Introduction to the Design and Behavior of Bolted Joints

Fourth Edition

Non-Gasketed Joints

John H. Bickford

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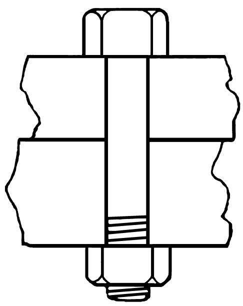
Introduction to the Design and Behavior of Bolted Joints



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Dedication

*Once again to my wife, Anne,
to our children David, Peter, and Leila,
and to Leila's children—our delightful
granddaughters—Zoe and Arden Sawyer.
Also to their father Hal Sawyer and to
Greg McCarthy. None of these people will
ever have to read this book, though
Hal probably will.*

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Preface to the Fourth Edition

This fourth edition stands on the shoulders of the first three editions, so I have included large excerpts from their prefaces once again. These detail the steps taken—and the themes developed—to reach this point. The acknowledgments found in the third edition are still valid too, so they are also included. At the end of this preface I express my thanks for the additional help provided for this present text.

The third edition was just under 1,000 pages in length, so it was clearly undesirable to create a longer, single volume, fourth edition. Experience in conducting bolting seminars, and through contacts with readers has shown, furthermore, that the audience for this text comes in two flavors. Many users deal primarily with gasketed, pressure vessel, and piping joints. The rest deal with the types of non-gasketed joints found in the auto, aerospace, structural steel, heavy equipment, mass production, and other industries. So it was decided that this fourth edition should be published in two volumes, one for each group. It was further decided that Volume 2, for gasketed joint users, should be coauthored by me and by Jim Payne. Jim is an internationally recognized expert in PVP joints, and is very active in ASME, the Pressure Vessel Research Council, and other groups that sponsor research and write standards dealing with gasketed joints. Jim will write all of the chapters whose focus is the gasketed joint. I will contribute those chapters pertinent to any bolted joint: on the basic behavior of joints and bolts, on materials, on threads, on torque and other preload control means, on failure modes common to gasketed and non-gasketed joints, etc. This generic material will also, of course, be included in the volume designed for those dealing with non-gasketed joints, so there will be a great deal of redundancy between the two volumes. We expect that only a few readers will need or want both volumes.

Previous editions have been used by practicing engineers, and have rarely if ever been used as a classroom text. An attempt has been made this time to make it more attractive not only to people in the field but also to teachers. A set of problems or exercises has been included at the end of each chapter. Answers to these will be found in the Appendix. All of the information required to answer the questions or do the exercises can be found in the book, either in the text or in the tables of data found in the Appendices. In fact, many of the exercises have been designed to force the student to search for information or data not in the chapter containing those exercises but elsewhere in the book, to encourage him to learn how to use the book more effectively. These exercises should also help to fix the material in the mind of a home-based student. An attempt has also been made to create a leaner, meaner text: long winded historical discussions, redundancies, irrelevancies and the like have been excluded this time so that basic ideas, data, and themes will be easier to find and use. The overall goal is a useful text that can also be used for training purposes.

Much material has been eliminated, but a lot of new information has been added. This is scattered throughout both volumes and generally involves an update of material already included in the previous edition. These updates are based on the latest revisions to various bolting standards, on information obtained from colleagues who are active in the field, and from that wonderfully helpful source of information that was not available to me when I created previous editions—the Web. The latter is so useful that, in several places including the

Appendices, I've given the addresses of many Web sites you will find especially useful when working with or studying bolted joints.

There are no new chapters in this Volume 1. Changes and additions have sometimes only required a sentence or two, more frequently a new paragraph, and occasionally a couple of pages. New information includes such things as revised designations for several bolting materials; new products and procedures to fight self-loosening; new ways to control preload, including the SquirterTM, a tension-indicating washer; new ultrasonic equipment for measuring bolt stretch or preload, including a plasma-coated, thin-film, ultrasonic transducer; a NASA procedure for selecting preload for space shuttle bolts, revised specifications and definitions for the design of joints loaded in shear; and current practices and tools for mass production bolting.

Acknowledgments for Volume 1 of the Fourth Edition

Colleagues who provided important new information for this fourth edition include Donald Bidwell, CEO of The Bidwell Industrial Group; Wayne Wallace of Applied Bolting Technology; Paul Wallace, consultant and ex-VP of SPS and Ingersoll-Rand; Ian Chapman of Ingersoll-Rand; Jesse Meisterling of PFW Technologies; Charles Wilson of the Industrial Fasteners Institute; Dr Pierfranco Mauri of Henkel-Loctite; and, most of all, colleague and now coauthor Jim Payne, consultant of JAPAC, Inc. Jim was retained by Taylor & Francis to create an electronic copy of the third edition, which he and I could then convert to Volumes 1 and 2 of a fourth edition. This turned out to be a surprisingly difficult and frustrating task. The electronics had a mind of its own and refused, at first, to do many of the things we wanted it to. Several versions of the software, and after struggles and surprises, which continued to the end of the project, we managed to fight it to a draw and finally produced the much revised electronic text used to print the words you are now reading. Jim stuck to the task with a dedication few would have shown. As I write this he still faces much of the work required to complete Volume 2 on gasketed joints.

Many other colleagues also contributed material for this revised edition. They and the material they provided are cited in the references at the end of many chapters and, as already mentioned, in the acknowledgments to the first three editions which you'll find at the end of the prefaces.

Previous editions of this text were published by Marcel Dekker of New York, but this fourth edition will come to you from the Taylor & Francis Group of CRC Press, and I am much indebted to the sympathetic assistance and encouragement I have received from Theresa Del Forn at Taylor & Francis. She understood the difficulties we were having with the electronic copy of the third edition, and she did what she could to help us and to ease that burden.

Once again—to the surprise of both of us—my MIT classmate, architect, wife, and love of my life, Anne, has had to put up with a distant and preoccupied husband for a whole year. I could never have done it without her tolerant cooperation for which I am extremely grateful. This is definitely the last time I do this to you, Anne.

Preface to the Third Edition

Although their exact birth date is unknown, it's certain that threaded fasteners have been around for at least 500 years. They're still the fastener of choice when we want an easy and relatively low-cost way to assemble anything from a frying pan to a satellite. They're virtually our only choice if we want to create a specific clamping force to hold a joint together. The bolt and its cousins are, in fact, marvelously simple mechanisms for creating and maintaining this force. And, of course, there's no better fastener if we also need to disassemble and reassemble something for maintenance or other purposes. Considering its tenacious hold on life and its persistent popularity, it's interesting that we still don't know all we'd like to about the bolt, or about the bolted joint and the way it behaves in service. Until a few years ago, moreover, most engineers knew virtually nothing about these things, and an "introduction" seemed in order. The response to the first and second editions of this text seems to confirm that premise. The number of engineers who know the basic concepts of joint behavior, however, has grown substantially; if my seminar students are any indication. The time has come, therefore, to deal with the subject in greater depth than in previous editions. As a result, this third edition contains far more specific information concerning design and behavior; with many new equations and numerical examples. For example, three new chapters, 21–23, are devoted specifically to design, covering joints loaded in tension, gasketed joints, and joints loaded in shear. In addition, several other chapters have been expanded to include specific design procedures or recommendations.

A lot of this new material—especially that dealing with joints loaded in tension—is based on a modified version of the design procedure defined some years ago by the German engineering society Verein Deutscher Ingenieure (VDI). I have modified their equations to account for—or at least to make more visible—such phenomena as elastic interactions, gasket creep, and differential thermal expansion, which can have a major impact on joint life and behavior. My debt to VDI, however, is substantial. If I have misinterpreted or misrepresented them, the fault is mine and I apologize.

This edition also contains a detailed discussion of the new gasketed joint design procedure developed by the Pressure Vessel Research Committee (PVRC) and now being incorporated into the *ASME Boiler and Pressure Vessel Code*. This procedure is based on the new gasket factors developed by the PVRC, and I have included the most recently published list of these factors. I have also shown, in a numerical example, how to use the new factors, and how the results obtained compare with the results of calculations based upon the historical Code procedure. Chapter 22, on the design of gasketed joints, also includes a discussion of alternate design procedures suggested by other people or groups in Europe.

Chapter 19 deals separately with the behavior of gaskets and contains much new material on important gasket properties such as creep and blowout resistance. Gasket test procedures and the new gasket rating factors proposed by the PVRC are also discussed at length for the first time.

Chapter 23, on the design of shear joints, is based on both VDI techniques and the design recommendations of the AISC, with the latter being more useful and informative for this type of joint. As in the other chapters on design, I have included numerical examples, this time for the design of friction-type, bearing-type, and eccentrically loaded joints.

Other new material in this edition includes Chapter 3, on threads configurations, nomenclature, and strength, as well as additional material in Chapters 6, 15, 16, and 17, dealing with joint assembly, fatigue, self-loosening, and corrosion.

This edition, like the previous ones, is based almost entirely upon the work of others, as shown by the many references cited at the end of each chapter. Each and every one of those authors deserves my respect and my thanks. Many of those who contributed the most to my education are listed in the following acknowledgments. My current debt is so broad, however, that I'll let the references serve for this edition.

I do, however, want to add an acknowledgment that I should have included in both the first and second editions: my debt to my publisher. I owe a great deal to Graham Garratt, vice president and publisher, who first suggested that I write such a text, and who later convinced me that a second and now a third edition were desirable. Writing a book is not a trivial task, and I probably would not have attempted it without his gentle urging and continued support.

Revising a text, I was surprised to find, is more challenging than writing one. Correcting, updating, and improving a text while adding new material could challenge more nimble minds than mine, and here I have been blessed with the friendly and helpful guidance of Walter Brownfield, who supervised the production of both the second and third editions.

So, my thanks to both of these people at Marcel Dekker, Inc. and to the many engineers and scientists listed in the references.

Preface to the Second Edition

When I wrote the first edition of this book, most people, including most engineers, were generally unaware of the importance of the bolted joint in our high-tech world. The few who were experts were often considered remnants of that previous age when large iron and steel railroads, ships, tractors, and bridges first evolved.

In recent years, however, a series of newsworthy events, many of them tragic, have made us realize that the threaded fastener still plays a major role in our lives. Oil drilling platforms have tipped over, airplane engines have failed, roofs have collapsed, and astronauts have died because of bolted joint failures. The Nuclear Regulatory Commission has declared ''bolting'' to be an ''unresolved generic safety issue with number one priority,'' even though no bolt-related accidents or equipment failures have occurred in that industry. And, most recently, the realization that substandard or counterfeit bolts are flooding the country, with safety implications for our defense, and our nuclear, aerospace, auto, and other industries, has led to congressional hearings and has even been reported on network television.

Even though our general awareness has been raised, the technology of bolted joints is still in its infancy. We know a lot more than we used to (some of that new knowledge is reflected in this new edition), but we still have a long way to go. Like weather forecasters, bolting engineers must still deal with very large numbers of unknowns and variables. As a result, our predictions and attempts to solve or prevent problems must often be based on past experience, trial and error, overdesign, and so forth, as in the past, rather than on the hard-and-fast answers so preferred by engineers.

Each of us, however, can benefit from the prior experience, the success and failure of others. Years ago, I designed a bolted joint seminar based on the material in the first edition. This seminar, which is still being given, has been sponsored by Raymond Engineering, the University of Wisconsin, and most recently by the ASME. Students have been drawn from the automobile, aerospace, power, marine, heavy equipment, and other industries that face bolting problems. The students have included people who design, build, and use bolted equipment. And I think that, over the years, they have contributed as much to my education as I have to theirs, offering tips, suggestions, and examples of things that have worked and have not worked. Their questions and problems have certainly forced me and the other instructors to dig more deeply than we might have into the literature and elsewhere, to try to shed light on some of the problems that still plague us.

Much of that digging is reflected in this new edition, in which I have attempted to include information that will answer the most commonly asked questions. The first edition, I'm afraid, raised as many questions as it resolved, and, although neither I nor anyone else at the present has all the answers to the questions that face bolting engineers, I have attempted to include far more concrete tips and suggestions and data than I did in the earlier edition.

The new material in this edition includes the following.

Specific suggestions for optimizing the results obtained when assembling bolted joints. Tips are given for assembly procedures based on torque control, torque–turn control, turn of nut, stretch control, ultrasonic measurement of bolt stress, and the like (Chapters 5–11).

A variety of suggestions on how to pick preload (or torque) for a given application, starting with simple methods for relatively unimportant joints and proceeding to more sophisticated methods (Chapter 21).

A new chapter devoted to the material properties that affect the strength of the fastener or the stability of the preload or clamping force on the joint in service (Chapter 3). Also, more data on such things as nut factors (Chapter 3), gasket stiffness (Chapter 4), the elevated temperature properties of bolting materials (Chapter 3), gasket creep (Chapter l8), and the relative costs of bolting materials (Chapter 3).

A greatly expanded discussion of stress corrosion and other stress cracking phenomena, with data on the stress corrosion resistance of a variety of bolting alloys (Chapter 19).

A tabulation of key bolting equations in calculator (or computer) format (Appendix H). A discussion of fastener coatings, with their uses, strengths, and weaknesses, including substitutes for cadmium plating (Chapter 19). An expanded discussion of fatigue failure, with new data (Chapter 17). A discussion of a phenomenon I call "elastic interactions," which occurs when we tighten groups of bolts and which can have a significant influence on the amount of clamping force developed in a joint (especially a gasketed joint) during assembly. Most people, myself included, were unaware of this phenomenon when I wrote the first edition. Interactions can cause assembly preloads to vary by 4:1 or more, even if tensioners are used to tighten the bolts (Chapter 6).

A simple procedure that will allow you to make a rough estimate of the stiffness of a bolted joint, a procedure based on experimental data generated by several different groups (Chapter 4). Although the procedure is only approximate, it is much cheaper than the experiments or finite element analysis required for a more exact answer, and it will be good enough for many applications.

A nearly complete revision of the discussion of ultrasonic measurement of bolt stress or strain to reflect the significant advances that have occurred in this technology in recent years (Chapter 11).

Major revisions to and extension of the discussion of gaskets, with a description of recent results of research sponsored by the Pressure Vessel Research Committee and a discussion of the new gasket factors now being proposed as replacements for the m and y factors of the *ASME Boiler and Pressure Vessel Code* (Chapter 18).

A procedure for estimating the effect of a change in temperature on preload or on the clamping force on the joint (Chapter 14), plus a discussion of the other ways in which elevated temperature can affect a gasketed joint (Chapter 18).

A structured procedure for answering bolted joint questions and for predicting results when the joint is assembled and put in service (Chapters 20 and 21).

I think that you will find that the information listed above, plus that carried over from the first edition, will help you deal with this complicated thing called a bolted joint.

Preface to the First Edition

To "get down to the nuts and bolts" of a topic has always meant to get to the heart of it, and rightfully so. After all, the joints are the weakest element in most structures. This is where the product leaks, wears, slips, or tears apart. I have heard that the improper use of fasteners—in joints, of course—is the largest single cause of the warranty claims faced by U.S. automobile manufacturers. An air force engineer told me that the cost of a modern military airplane is a linear function of the number of fasteners involved. These claims may be apocryphal, but the problems are real.

In spite of their importance, bolted joints are not well understood. Mechanical engineering students may receive a brief introduction to the subject in a design course, but only a small percentage of them—in school or afterward—will ever get involved enough for a real understanding. The specialists who design things which must not fail—airplanes, nuclear reactors, or heavy equipment costing hundreds of thousands or even millions of dollars—are forced to learn all there is to know about the design and behavior of bolted joints. The rest of the engineering fraternity, even designers, is guided by guesswork, experience, or handbooks, and they still have problems.

As a matter of fact, even sophisticated designers have problems at the present state of the art because the behavior of a bolted joint involves a large number of variables difficult or impossible to predict and control. There are widely used design theories and equations, many of which we shall study in this book, but these are usually simplifications and approximations. They have been used, successfully, on all sorts of joints in all sorts of products, but they are not sufficient for critical joints. Most of them, furthermore, have been around for years, and they have fallen behind the demands being placed on contemporary designs, e.g., higher operating temperatures and pressures, new materials, the increased clamor for more safety or environmental protection, and better strength-to-weight ratios. Even the thorough, widely used, and often-modified *ASME Boiler and Pressure Vessel Code* has failed to keep pace with the needs of the designer.

The engineering societies are aware of these problems, of course, and are currently funding extensive experimental and theoretical studies to advance the science (or is it an art, at present?) of bolted joint design. It is believed that this work will make accurate joint design possible, but not until the end of this decade. That forecast, coming from the most knowledgeable people in this business, gives you an idea of the magnitude of the problem. None of us, of course, can wait 10 years for solutions to our current design problems. We have to function at the current state of the art. Even this is a challenge, given the complexity of the subject, but currently available information can help us minimize joint problems even if we can't eliminate them. Hence, this book will serve as an introduction to the design and behavior of bolted joints and a primer for engineers or students who are struggling with the subject in depth for the first time. It will also help plant engineers, maintenance engineers, production engineers, and other nondesigners understand the nature of and reasons for their bolted joint problems, and give them some help in solving or reducing these problems.

The information in this book has come primarily from two sources. My employer, Raymond Engineering Inc., has manufactured for some years unusual tools and equipment for assembling and disassembling large bolted joints. With a desire to increase our knowledge of bolted joint technology, we commissioned, in 1978, a computerized literature search. This search, directed by Stephen Ford of the Battelle Memorial Institute, uncovered thousands of articles: some unique, some repetitive; some "correct," some ridiculous; some well written but some not. File drawers full of articles, including, by and large, all that was known, or at least all that had been published, about bolted joints at that time.

Since then, we have sponsored a biweekly computerized "update" search of many different engineering files, including EI, DOE, BHRA, NASA, ISMEC, ASM, INSPEC, CPI, CAC, NTIS, USG, and many others. These updates are made for us by the New England Research Center at the University of Connecticut.

These updates have kept our library current—and our readers busy! This present book is, to a large extent, an overview of the state of the art as revealed by this literature search, so, as the author, I am much indebted to Mr Ford for starting my education, and to UCONN for continuing it.

I'm even more indebted to the many engineers and scientists who wrote the articles: Bob Finkelston, Gerhard Meyer, and Dieter Strelow of SPS; G.H. Junker of Unbrako-SPS; Nabil Motosh of Asslut University in Egypt; John Fisher of Lehigh University; Wayne Milestone of the University of Wisconsin; Ed Rice of Ingersoll Rand; and Sam Eshghy of Rockwell stand out as key influences, but there are hundreds of others. Any errors in my book, of course, should not be blamed on them, but rather on my inability to understand.

But there's more to it than that. We're not scientists or academics. We're engineers and businessmen, and although we're deeply interested in the theories and explanations, our goal is to understand and solve, or prevent, field problems. It's nice to know that "the equations don't always work because . . . "; but we still have to tighten those joints, right now, in such a way that they stay put for the life of the product, or at least until the next maintenance shutdown. And so we kept looking for equations, information, rules-of-thumb, divine guidance, or anything that would get us there. And this led us in two directions that have produced results.

First, our search for something better led us to an impressive new instrument called the ultrasonic extensometer—invented by Donald Erdman of Pasadena and Howard McFaul of Douglas Aircraft Corporation. This instrument is designed specifically to measure the actual strain in a bolt before, during, and after tightening. Here, for the first time, we had a way to measure tightness in bolts, with a high degree of accuracy, under any and all field conditions, statically or dynamically, and across the board. Prior methods were only practical for samples you strain gauged a few bolts, for example, used load cells under the heads of a few, or made a laboratory experiment. In many cases you modified field conditions simply by taking the measurements. If nothing else, the results you obtained were unnaturally good, because the person using the wrench was more careful with those bolts than before or after your experiment.

Several types of extensometer are described at length in Chapter 9. I'm indebted to Donald Erdman, incidentally, for reading and correcting this chapter, as well as for making it possible for us to measure bolt stretch ultrasonically.

The extensometer makes it possible to check unmodified bolts assembled by unsuspecting people, and to monitor such elusive things as dynamic loads or long-term relaxation. Engineers have long been able to measure the inputs to the system, e.g., torque applied to the nut, composition of the lubricant, and angle of turn of the nut. Now we can see the immediate effects and results, as a function of job conditions, time, or both. We felt just as the electrical engineer must have when someone handed him the first oscilloscope.

We have used this instrument extensively in our laboratory to study bolt problems and to analyze and check some of the information and theories uncovered by the literature search. But more significantly, we have used it in the field. We organized a bolting services group which sent technicians to many parts of the country and overseas to help customers tighten or disassemble problem joints. This hands-on experience provides the second major source of information on which this book is based. I'll tell you what really happens when you tighten various kinds of joints, under often difficult conditions, with a variety of tools and procedures even if no one, at present, can fully explain why they behave this way. And I'll describe some of the techniques we and our customers have used to solve or minimize today's problems.

Most of our work has involved very large bolted joints, e.g., pressure vessels, pipe joints, heat exchangers, engine heads, and helicopter transmissions; but we've also been involved with small aerospace assemblies, and have had some exposure to the tools and techniques used by automobile and other mass producers. So, although the case history emphasis in this book will be on large fasteners, the design and behavior information is applicable to most types of bolted joints.

One warning for those involved in the design and construction of buildings: You will not find much information here on structural steel joints. Many of the topics covered would be pertinent to such joints, but I make little or no attempt to relate them to those applications. This is an area in which my company and I have had very little experience, and it's an area that is very well covered by Fisher and Struik's excellent *Guide to Design Criteria for Bolted and Riveted Joints* (Wiley, 1974). This work, on the other hand, doesn't cover liquid joints, or the problems faced by production engineers.

I am sure that some of you will find the subject of bolted joints as interesting as I do, at least by the time you finish the book. Before we start, let me add one more note of appreciation for my secretary, Tressa Battista, who faced too many drafts with too little time, but did it all.

We're ready. Let's learn about bolted joints.

Acknowledgments for the First Three Editions

As before, most of the information in this book is not original with me. I am merely passing along to you the education that I have received from so many other people and sources, including those mentioned in the preface to the first edition. For recent years, I am especially indebted to those listed below. In the preface to the second edition, I mentioned the seminar students, who are too numerous to name but to whom I owe a great deal. My education has been advanced even more, I think, by my participation in a number of technical societies and groups that are struggling to resolve a variety of bolting issues. I was involved, for example, with the subcommittee on bolted flange connections of the Pressure Vessel Research Committee; I was a member of their task group on gasket testing and was the chairman of the joint task group on the elevated temperature behavior of bolted flanges. I have learned a great deal from many of the engineers who have attended these meetings and who serve as consultants to these groups. I am especially indebted to Andre Bazergui of l'École Polytechnique in Montreal, Jim Payne of JPAC, Inc., George Leon of the Electric Boat Division of General Dynamics, and J. Ronald Winter of the Tennessee Eastman Company. Dr Bazergui was kind enough to review Chapter 18 on gaskets in the second edition. His comments have influenced the final version of that chapter; any remaining errors are mine.

I was also the vice-chairman of The Research Council on Structural Connections and have benefited significantly from their work and deliberations, which involve special inputs from John Fisher of Lehigh University, Joseph Yura and Karl Frank of the University of Texas at Austin, Geoffrey Kulak of the University of Alberta at Edmonton, Thomas Tarpy of Stanley D. Lindsey and Associates of Nashville, Peter Birkemoe of the University of Toronto, Michael Gilmor of the Canadian Institute of Steel Construction, and Bill Milek, now retired, formerly with the AISC.

Dr Kulak revised the valuable and influential text *Guide to Design Criteria for Bolted and Riveted Joints* (Wiley, 1987), originally written by John Fisher and John Struik. I cited this text and my debt to it in the preface to the first edition.

I also participated in meetings of the Atomic Industrial Forum/Metals Properties Council Task Group on Bolting and was chairman of a working group on bolting organized by the ASME Committee on Operations and Maintenance (Nuclear Codes and Standards). Both of these groups were established to define and resolve bolting issues that concerned the NRC. Key players to whom I am indebted here include Ed Merrick, at that time with TVA and now with APTECH in California, Russell Hansen of GA Technologies, and Joe Flynn, Jr., of INPO. I benefited from information on nuclear bolting problems provided by Ed Jordan, Robert Baer, and William Anderson of the Nuclear Regulatory Commission. I am also grateful for more recent input from Richard Johnson of the same organization.

The Electric Power Research Institute of California played a significant role in the work of the AIF/MPC Task Group, funding much of the group's research. They also funded the development at Raymond Engineering of a reference manual on good bolting practices as well as three training cassettes for bolting engineers and mechanics in the nuclear power industry. I was a coauthor of the manual and participated in the preparation of the cassettes. The material in both was developed with the help of, and was reviewed by, maintenance and operating engineers in a number of nuclear plants as well as by members of the ASME Working Group. Many of the tips and suggestions that found their way into the manual and cassettes have also been included in the second edition.

I am also grateful to the Industrial Fastener Institute of Cleveland. Raymond Engineering was an invited member of that group for several years, and I have learned much from my contacts with their members and from technical discussions with Charles Wilson, their Director of Engineering.

Colleagues at Raymond Engineering contributed to my education as well. Special mention should go to Jesse Meisterling and to Michael Looram. (The Bolting Products Division of Raymond Engineering, Inc., is now the Industrial Tool Division of Bidwell lndustrial Group, Inc.)

Stan Johnson of Johnson Gage deserves mention, too, for his input on thread strength, thread gaging, and the like.

Bolting products customers of Raymond Engineering have provided much information about bolting problems in nuclear power, petrochemical, aerospace, automobile, and other industries. Unfortunately, the number of teachers here is so great that individual mention is impossible, but my debt to them is nonetheless considerable.

Last but not least, I would like to acknowledge this latest of many debts to my wife, Anne, who once again lost a husband to a word processor. Ready or not, Anne, I'm now coming back!

Editor

John H. Bickford is a retired engineer living in Middletown, Connecticut. A former vice president of the Raymond Engineering Division of Kaman Aerospace, and manager of Raymond's Power Dyne Bolting Products Group, Mr Bickford holds 15 U.S. patents and several foreign patents. He is the author of numerous articles and book chapters on various aspects of bolting technology, and is coauthor of a reference manual for nuclear power plant maintenance personnel. Mr Bickford is a member of the American Society for Testing and Materials (ASTM) and a past member of the American Society for Mechanical Engineers. He is a founder and past president of the Bolting Technology Council, which has become Committee F16.96 of the ASTM. He holds a BS degree in business and engineering technology from the Massachusetts Institute of Technology in Cambridge, Massachusetts.

Abstract

Threaded fasteners, including all varieties of bolts, make our industrial, commercial, and even civil worlds possible, by providing simple, economic means to join an infinite variety of small parts together to create such large and useful objects as automobiles, airplanes, buildings, plows, looms, and printing presses. This book describes the design, behavior, misbehavior, failure modes, and analysis of the bolts and bolted joints which play such a large, even ubiquitous, role in all this. The reader will learn why proper bolt tension—often called preload—is critical to the safety and reliability of an assembled joint. He'll be introduced to many ways to create that preload as well as ways to measure or inspect it. He'll learn how to design joints, which are less apt to misbehave or fail, using the guidelines, procedures, and simple algebraic mathematics included in the text. Numerous tables, charts, graphs, and appendices will give the reader all the information and data he or she needs to become a competent designer or user of non-gasketed bolted joints. Gasketed joints are also covered, briefly, but will be the focus of a second volume to appear in the future.

1 Basic Concepts

This book is intended to give you an introduction to the design and behavior of bolted joints and hopefully will help you become better designers, assemblers, or users of such joints, or help you analyze and prevent joint failures. The subject is a complex one, which is why a two volume text of well over 1000 pages can be considered only an introduction. The material presented here, however, should be all the information that many or most people need. Numerous references at the end of each chapter lead the way to further details for those who need or want to know more.

This first chapter gives an overview of the material to be covered in the rest of the book; it's an introduction to the introduction, if you will. You might find it useful to come back to this chapter and reread it if you get bogged down in the detail of subsequent chapters and have trouble seeing how that subject or detail fits the overall picture.

1.1 TWO TYPES OF BOLTED JOINTS

Bolted joints come in two flavors, depending on the direction of the external loads or forces acting on the joint. If the line of action of the forces on the joint is more or less parallel to the axes of the bolt, the joint is said to be loaded in tension and is called a tension or tensile joint. If the line of action of the load is more or less perpendicular to the axes of the bolt, the joint is loaded in shear and is called a shear joint. Both types are illustrated in Figure 1.1. Some joints support combined tensile and shear loads and are named after the larger of the loads placed on them, be it tensile or shear.

The distinction between tensile and shear joints is important, because the two types differ in the way they respond to loads, the ways in which they fail, the ways in which they are assembled, etc. In general, the tensile joint is the more complex of the two—as far as behavior and failure are concerned—and it's the more common type of joint. Most of this text, therefore, is devoted to it. Another reason for this bias: Messrs. Kulak, Fisher, and Struik have written an excellent text, *Guide to Design Criteria for Bolted and Riveted Joints*, second edition (John Wiley & Sons, New York, 1987), which is devoted almost entirely to shear joints.

1.2 BOLT'S JOB

The purpose of a bolt or group of bolts in all tensile and in most shear joints is to create a clamping force between two or more things, which we'll call joint members. In some shear joints the bolts act, instead, primarily as shear pins, but even here some bolt tension and clamping force is useful, if for no other reason than to retain the nuts.

1.2.1 Tensile Joints

Specifically, in tensile joints, the bolts should clamp the joint members together with enough force to prevent them from separating or leaking. If the joint is also exposed to some shear loads, the bolts must also prevent the joint members from slipping.

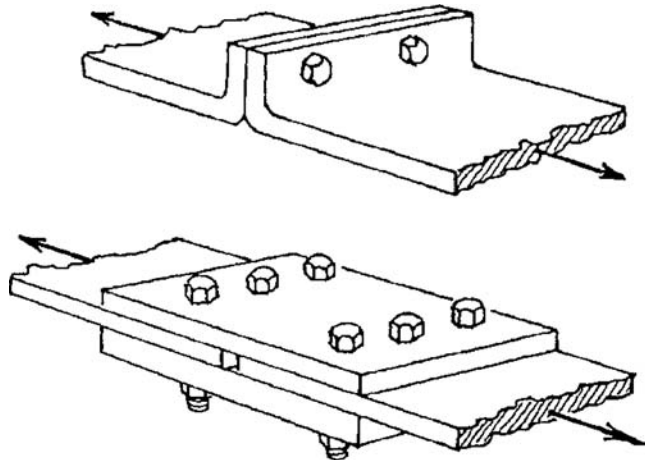


FIGURE 1.1 Bolted joints are classified by the service loads placed on them. If those loads and forces are applied in a direction more or less parallel to the axes of the bolts, as in the upper sketch here, the joint is called a tensile or tension joint. If the line of action of the forces is essentially perpendicular to the axes of the bolts, as in the lower sketch, the joint is called a shear joint.

Coincidentally, the tension in the bolt must be great enough to prevent it from self-loosening when exposed to vibration, shock, or thermal cycles. High tension in the bolt can also make it less susceptible to fatigue (but sometimes more susceptible to stress cracking). In general, however, we usually want the bolt in a joint loaded in tension to exert as much force on the joint as it and the joint members can stand.

There are two important facts you should keep in mind when dealing with tension joints. First, the bolt is a mechanism for creating and maintaining a force, the clamping force between joint members.

Second, the behavior and life of the bolted joint depend very much on the magnitude and stability of that clamping force.

Note that I did not say "magnitude and stability of the preload" or of the tension in the bolt or of the torque applied to the bolt. Those parameters are related to the clamping force, often closely related, but the key issue as far as joint behavior is concerned is the force the two joint members exert on each other (the clamping force), created, of course, by the force the bolts are exerting on them.

The key issue as far as bolt life and integrity are concerned is, however, the tension in it; so we must keep our eye on both interjoint clamping force and bolt tension to be successful.

The clamping force on the joint is initially created when the joint is assembled and the bolts are tightened by turning the nut or the head of the bolt. This act, of course, also creates tension in the bolt; the tension is usually called preload at this stage.

Although there may be some plastic deformation in some of the threads when a bolt is tightened normally, most of the bolt and the joint members respond elastically as the bolt is tightened. The joint members are compressed a slight amount, and the bolt is stretched by a larger amount.

In effect, both joint members and bolts behave like stiff springs, one being compressed and the other stretched as suggested in Figure 1.2. Like springs, furthermore, they acquire potential (or stored) energy. If we released them after tightening them, they would suddenly snap back to their original dimensions. It is this stored energy which allows bolts to maintain that all-important clamping force between joint members after we remove the wrench.

We might even say that the tensile joint, unlike its welded or bonded joint cousin, is "alive," filled with energy and able to do its job only because it's filled with energy.

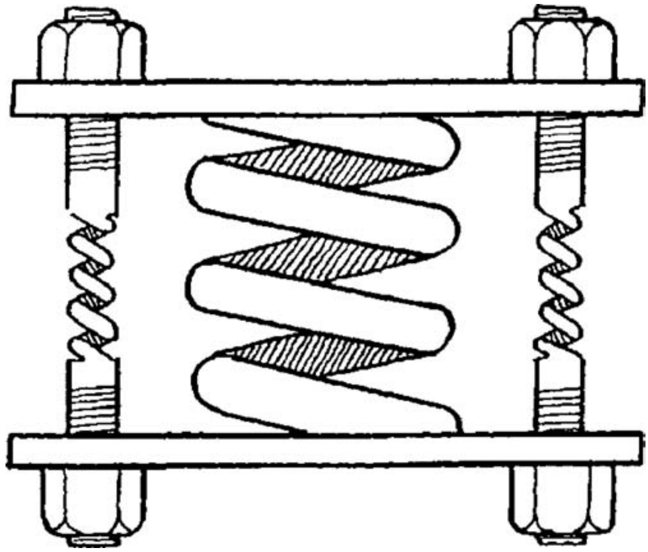


FIGURE 1.2 Bolts and joint members deform elastically when the bolts are tightened. In effect, they act like stiff springs as suggested by this sketch. This fact that they act like springs greatly influences the behavior of the joint.

1.2.2 Shear Joints

The bolt's main job in a shear joint is to keep the joint from slipping or from tearing apart in the slip direction. If the joint must also support some tensile load, the bolt must resist that too.

In some shear joints, as already mentioned, the bolts resist slip by acting as shear pins, and joint integrity is determined by the shear strength of the bolts and joint members. There are a number of reasons why we will often want to tension these bolts, as we'll see, but the exact amount of tension, or of the energy stored in them, is not a critical factor.

In other shear-loaded joints, slip is prevented by friction restraint between joint members. These friction forces are created by the clamping load, which in turn is created by heavily tensioned bolts. Here again, therefore, the bolt is a mechanism for creating and maintaining a force, and the magnitude and life of that force depend on the potential energy stored in the bolts during assembly. Even here, however, we're usually less concerned about creating an exact amount of tension in the bolts during assembly than we are when we're dealing with tensile joints, because service loads don't affect bolt tension and clamping force in shear joints.

1.3 THE CHALLENGE

The bolted joint presents users and designers with many problems. In part this is because it is "alive"—it keeps changing state in response to service and environmental conditions, as we'll see. A more common source of problems, however, is the fact that the assembly process and the in-service behavior are affected by literally hundreds of variables, many of which are difficult or impossible to control or to predict with accuracy. As a result, when we deal with bolted joints we must inevitably deal with a lot of uncertainty. What follows is a quick review of some of the sources of this uncertainty. We'll take a closer look at most of these things in later chapters.

1.3.1 Assembly Process

Bolts and joint members in both tension and shear joints respond in the same way to the act of tightening the bolts. There are differences in the accuracy with which we must tighten them, but most of the discussion which follows applies to all joints.

As far as all tension and most shear joints are concerned, the goal of the assembly process is to establish an initial clamping force between joint members, to introduce the first energy into bolt and joint springs. And, in tension joints, we're usually interested not just in tensioning the bolts but in tensioning them by a desired amount, because the life and behavior of such joints are so dependent on the right amount of clamping force. We want enough clamping force to prevent a variety of failure modes, but we must also make sure that the bolt tension and clamping force do not exceed an upper limit set by the yield strengths of the materials, the anticipated loads to be placed on the joint in service, and other factors. Unfortunately, as already mentioned, hundreds of variables affect the results when we tighten a group of bolts, so predicting or achieving a given clamping force is extremely difficult.

We attempt to control the buildup of clamping force by controlling the buildup of tension or preload in the bolt. In most cases, we do that by controlling the amount of torque applied to the nut or head.

The work we do on the fastener while tightening it is equal to one half the applied torque times the angle (measured in radians) through which the nut turns. Typically, about 10% of this input work ends up as potential energy stored in the joint and bolt springs. The rest is lost in a variety of ways.

Most of the kinetic energy is lost as heat, thanks to friction restraints between the nut and joint surface and between male and female threads. Some energy is used to twist and, often, to bend the bolt a little. Some energy may be lost simply in pulling heavy or misaligned joint members together or dragging a bolt through a misaligned or interference fit hole. More is lost by spreading the bottom of the nut, a process called nut dilation.

A major problem for the designer and assembler is that it is virtually impossible to predict how much of the input work will be lost due to factors such as these. The amount lost can and usually will vary a lot from one bolt to another, even in the same joint.

In spite of these uncertainties and losses, some potential energy is developed in each bolt as it is tightened, and it starts to create some clamping force in the joint. But then the bolt relaxes—loses some energy—for a couple of reasons.

A process called embedment occurs as high spots on thread and joint contact surfaces creep out from under initial contact pressure and the parts settle into each other. More drastically, a previously tightened bolt will relax somewhat when its neighbors in the joint are subsequently tightened. We call this process elastic interaction, and it can eliminate most or even all of the tension and energy created in the first bolts tightened in the joint. We'll examine this phenomenon in detail in a later chapter.

The amount of relaxation a bolt will experience is even more difficult to predict than the amount of initial tension it acquires when first tightened, increasing the challenge of the assembly process.

Anything that reduces the amount of energy stored in a bolt reduces the force it exerts on the joint. Too little torque, too much friction, rough surface finish, twisting, bending, hole interference, and relaxation all can result in less stored energy, less preload, and less clamping force.

Anything that increases the energy stored will increase the force. There are a couple of things which can do this during assembly: too much torque or too little friction, thanks perhaps to a better-than-anticipated lubricant.

Again, all of these factors are difficult to predict or control, making it very difficult to achieve a particular amount of preload or clamp force at assembly. Because many factors can give us less preload than desired and only a couple can give us more, we often—perhaps usually—end up with less than expected at assembly.

Bolts in shear joints are subjected to the same assembly problems and variables as are bolts in joints loaded in tension. There's a difference, however.

In tension joints we always care about the amount of preload, tension, clamping force, and potential energy developed during assembly because of the way such joints respond to service loads. We're not so concerned about this when dealing with shear joints. We'll see why when we examine the in-service behavior of such joints.

1.3.2 In-Service Behavior

The in-service behavior of tensile joints differs substantially from that of shear joints, and this is reflected in the different ways we design and assemble the two types. Here's a preliminary look at the differences.

1.3.2.1 Joints Loaded in Tension

We encountered many uncertainties when we assembled a tensile joint. Further uncertainties are introduced when we put such a joint to work—when we load it, expose it to vibration or shock, subject it to change in temperature, anoint it with corrosive fluids, etc. Being alive, it responds to such things; and as it responds, the tension in the bolt and the clamping force between joint members change.

First, and most important, the tensile load on the joint will almost always increase the tension in the bolts and simultaneously decrease the clumping force between joint members. This is undesirable and unavoidable, and it is the major reason why we care so much about the exact amount of bolt tension and clamping force developed at assembly.

If the assembly preloads are too high, the bolts may yield or break when they encounter the service loads. On the other hand, if assembly preloads are too small, the clamping force on which the joint depends may all but disappear when service loads decrease it.

Other service factors can also change bolt tension and clamping force and will affect our choice of assembly preload. For example, relaxation processes like embedment or gasket creep are increased by loads and by elevated temperatures. Vibration, shock, or thermal cycles can cause the bolt to self-loosen. Differential expansion between bolts and joint members can increase bolt tension and clamping force simultaneously, or can reduce both. In this case, heat energy is being used to increase or redistribute the energy stored in the parts.

Chemical energy, exhibited as corrosion, can increase the clamping force as corrosion products build up under the face or the nut or head of the bolt.

These factors present an additional challenge to the designer. They increase the difficulty of predicting joint behavior, because the designer can rarely predict the exact service loads or conditions the joints will face. The joint's response, furthermore, will be influenced by such hard-to-pin-down factors as the condition of the parts or the exact dimensions and material properties of the parts. Behavior will also be influenced by the hard-to-predict amount of preload in the bolts, which the designer must somehow specify.

Once again, however, the factors that lead to less clamping force are more common than the ones that can lead to more clamping force. Since this is also true, as we've seen, of the assembly process, we are forced to recognize Bickford's little-known First Law of Bolting: Most bolted joints in this world are providing less clamping force than we think they are.

1.3.2.2 Shear Joints

Shear loads do not affect the tension in the bolts or the clamping force between joint members, at least until such loads become so high that the joint is about to fail. Predicting behavior and avoiding failure are therefore easier when we're dealing with shear joints than when we are dealing with tensile joints. This, in part, explains why people who design airframe, bridge, or building structures rely so heavily on shear joints and avoid using tension joints whenever possible.

I don't mean to imply that shear joints won't respond to service loads and conditions; they will. Bolt tension and clamping force will change if temperatures change. Vibration or shock can loosen the bolts, parts can rust, and corrosion products can build up and alter bolt and joint stresses. If the loads on the joint are cyclical, the stresses in bolts and joint members will fluctuate. But the in-service uncertainties the designer faces, and their consequences, are usually less than those he'll face when dealing with tensile joints.

1.4 FAILURE MODES

The main reason we want to control or predict the results of the assembly process and the in-service behavior of the joint is to avoid joint failure. This can take several forms.

A joint will obviously have failed if its bolts self-loosen, shake apart, or break. Self-loosening is a complicated process and is described in a separate chapter along with ways to combat it. In general, however, it's caused by vibratory or other cyclical shear loads which force the joint members to slip back and forth. A major cause of self-loosening is too little preload, and hence too little clamping force. Both tensile and shear joints are subject to this common mode of failure.

Bolts in both types of joints can also break because of corrosion, stress cracking, or fatigue—all of which are also covered in later chapters and two of which are encouraged by the wrong preload. Stress cracking occurs when bolts are highly stressed; fatigue is most apt to occur when there's too little tension in the bolts. Even corrosion can be indirectly linked to insufficient preload, if a poorly clamped joint leaks fluids that attack the bolts.

If the bolts fail for the reasons just cited or if they exert too little force on the joint, perhaps because of the assembly or in-service conditions discussed earlier, the shear joint may slip or the tension joint may separate or leak. Each of these things means that the joint has failed.

It's obvious that a leak is a failure, but what's wrong with a little slip or with separation of a joint that doesn't contain fluid?

Slip can misalign the members of a joint supporting shear loads, thereby cramping bearings in a machine, for example. Or it can change the way a structure absorbs load, perhaps overstressing certain members, causing the structure to collapse. Slip can lead to fretting corrosion or to fatigue of joint members. As already mentioned, cyclical slip can lead to self-loosening and perhaps loss of the fasteners. Vibration loosening of bolts and fatigue failure of shear joint members are of particular concern to airframe designers.

Separation of the members of a joint supporting tensile loads can encourage rapid fatigue failure of the bolts. It can also destroy the integrity of a structure or machine. It can allow corrosants to attack bolts and joint surfaces. Separation means the total absence of clamping force, which means, in effect, that the joint is not a joint at all.

A gasketed joint can leak if the initial clamping force between joint members during assembly is not great enough or if the in-service clamping force (which will almost always differ from the assembly clamp) is too low. The joint does not have to separate for leakage to occur. Gasketed joints are the subject of volume 2 of this two volume text, but they will be considered briefly in this volume as well.

Note that most joint or bolt failure modes are encouraged by insufficient bolt tension or insufficient clamping force or both. Self-loosening, leakage, slip, separation, fatigue—all imply too little clamp.

A few problems can be caused by too much tension or clamping force, however. Stress corrosion and hydrogen embrittlement cracking of bolts can occur in both shear and tensile joints and are more likely if bolt stresses are high. Joint members and gaskets can be damaged by excessive clamp. Joint members can also be distorted by excessive bolt loads: the "rotation" of raised face, pressure vessel flanges is a common example. Fatigue life can sometimes be shortened by high stress, although more commonly it's caused by insufficient clamping force.

But failures caused by too little clamping force are more common in either tensile or shear joints than are failures caused by too much clamp. And, as we've seen, assembly and service conditions are more apt to give us too little clamp than too much. Welcome to the world of bolting!

1.5 DESIGN

1.5.1 In General

The design of bolted joints, like the design of anything else, involves a detailed consideration of function, shapes, materials, dimensions, working loads, service environment, etc. Every industry has characteristic or "typical" joint configurations and needs, and it would be impossible to detail each in a single text. We can, however, look at some generalities, which must apply to most joints, whatever their specific application. And we can review design procedures that have been accepted and used by many.

Specifically we'll focus on a design procedure developed by Verein Deutscher Ingenieure (VDI), a German engineering society, but with some modifications and extensions. We'll also examine in some detail the design rules for flanged, gasketed joints found in the ASME Boiler and Pressure Vessel Code, with emphasis on changes, which are currently being introduced to those rules. One chapter will be devoted to the design of structural steel, shear joints, with frequent reference to the text by Kulak, Fisher, and Struik, which was mentioned earlier. We will also examine a joint design procedure created by NASA for space shuttle and other joints.

1.5.2 Specific Goals of the Designer

The joint designer, of course, is faced with all the assembly and in-service uncertainties detailed earlier. In spite of these uncertainties, he must do two things when designing a joint that will be loaded all or in part in tension:

1. He must pick bolt and joint sizes, shapes, and materials which will guarantee enough clamping force to prevent bolt self-loosening or fatigue, and to prevent joint slip, separation, or leakage when clamping forces are at a minimum (because of the factors we've described) and those hard-to-predict service loads are at a maximum.
2. In addition, he wants to select bolts that are able to support a combination of maximum assembly stress plus the maximum increase in stress caused by such service conditions as applied load and differential thermal expansion.

If his joint is loaded only in shear, and will depend for its strength only on the shear strength of the bolts and joint members, then those strengths will determine the design. Such joints must not be subjected to varying or cyclical loads, or self-loosening and fatigue problems might be encountered. If service conditions permit it, however, such joints are safe and greatly simplify the design process.

There are other things that the designer must worry about when designing tensile joints and some shear joints. He'll consider the bearing stresses the bolts create on joint surfaces, the amount of change in load the bolts see (which can affect fatigue life), the accessibility of the bolts (which can affect assembly results), and the flexibility or stiffness of bolts and joint members. If he's designing a tension joint he'll be especially interested in the so-called stiffness ratio of the joint, because this affects the way in which a given service load changes bolt tension and clamping force.

In any tension joint and in shear joints where clamping force is important, the designer will want to do everything he can to improve the energy storage capacity of his bolts. He'll find that long thin bolts and thick, metal joint members can store more energy than short stubby bolts or nonmetallic joints, hence our historic problems with sheet metal joints and our emerging problems when we try to bolt composite materials.

As we'll see, the many assembly and service uncertainties the designer has faced have traditionally forced him to overdesign—"oversize" might be better—the bolts and joint members, with resulting penalties in weight and parts cost. We'll quantify this oversizing and suggest ways it can sometimes be reduced.

Although most of the book will deal with subjects like assembly practices, in-service behavior of the joint, and failure modes rather than design specifics, everything relates to and should affect the design of bolted joints.

1.6 LAYOUT OF THE BOOK

We'll start with some background material on the strength of bolts and threads, the stiffness of bolts and joint members, and a review of the properties of the materials usually used for these things. We'll focus on properties that affect basic strength and which affect the stability of the parts (several material properties can encourage changes in clamping force).

Next we'll look in considerable detail at the many options we have for controlling the assembly process, looking at torque, torque and turn, strength, direct tension, and ultrasonic control of preload and clamping force.

Then we'll turn our attention to the joint in service: how it responds to service loads and conditions, how it fails, how to improve its response and minimize the chances of failure.

Finally, having learned in detail what we're up against, we'll develop procedures for estimating results and will close with a detailed discussion of a modified version of the VDI design procedure.

Appendices in the back will give you a variety of reference data to aid design activities or analytical calculations as well as answers to the exercises and problems given at the end of each chapter.

Enough introduction! Let's begin our serious study of this thing called a bolted joint.

EXERCISES

Because there is no mathematics in Chapter 1, the following are essay-type questions rather than problems. Writing the answers down in your own words is an excellent way to remember them, so don't just copy sentences from the text as you do these exercises. The points covered here are critical to a true understanding of bolted joints and so are well worth the effort required to put them in your memory bank.

1. Name the two types of bolted joints and tell why they're called that.
2. What is the main mission of the bolts in any joint?
3. What factor largely determines the behavior and life of a tensile joint?
4. Why are bolted joints often called live joints as compared to welded, riveted, or bonded joints?

5. What must the bolts in a shear joint do to prevent its failure?
6. Describe some of the factors that make it difficult to create an exact and predicted amount of clamping force during assembly.
7. How does an external load on a tensile joint affect the clamping force, and what else does it affect?
8. Name the principal modes of failure of both tensile and shear joints.
9. Describe or list the criteria a bolted joint designer must consider when selecting bolt size and material for a new joint.
10. Will short, fat bolts generally work better than long, thin ones? If so, why? If not, why not?

2 Materials

In the previous chapter we learned that the bolt's job is to clamp two or more joint members together, and that the joint designer's job is to select bolts strong enough to do this. The strength of these bolts will depend upon their size, geometry, and on the strength of the materials of which they are made. In this chapter, we learn about those material strengths.

Chapter 4 in the previous edition of this text covered bolt and joint materials at encyclopedic length. I listed every bit of data about every one of these materials I could find. The result: long discussions and even longer tables, with a great deal of information which was of resize little, or at least of very infrequent, practical use. The present edition is hopefully a leaner document. The data required by most practicing engineers, or of use for classroom exercises, are still here and should be easier to find. Tables of material properties have been significantly shortened, especially if the data they contained can now be conveniently found online. Web site addresses, which are included in this chapter and in Appendix C, will lead the reader to complete data if that given here is not sufficient for his or her needs. Some of the data included in previous editions, however, are anything but easy to find elsewhere, and are probably not available online. These include the shear strength of bolt materials, the stress relaxation properties of materials used at elevated temperatures, the relative density (weights) of various materials, etc. Because this information is hard to find it has been republished in this edition.

2.1 PROPERTIES THAT AFFECT THE CLAMPING FORCE

What properties are we interested in when we pick a bolting material? Because the most important purpose of the bolts is to clamp the joint members together, we're interested in any physical or chemical or other properties which affect the "magnitude" of the clamping force we can create at assembly, and the "stability" of that clamping force. How will that force be modified by use or age or temperature change or some other mechanism?

From an energy standpoint we want to know how much potential energy we can store in the bolt and how much will be retained by it when it's put to work. The first consideration is related to the magnitude of the force, the second to its stability.

2.1.1 Magnitude of the Clamping Force

From a bolting material point of view, the magnitude of the initial clamping force will depend primarily on the basic tensile and shear strengths of the material. For a given diameter and thread configuration, a stronger material means a stronger bolt. And the stronger the bolt, the greater the clamping force it can produce. We saw how to estimate the strength of a bolt and its threads in the last two chapters. You'll find examples of material strengths in this chapter. I don't mean to imply that we'll always tighten bolts to the limit of their strength, but stronger bolts can and usually are tightened to higher tension than weaker ones. Otherwise, we waste the extra money we spend on better materials.

Also note that there are a lot of things other than material that affect the clamping force actually achieved at assembly. We'll look at these factors in depth in later chapters. At present we're considering only the clamping capacity of the bolts, the maximum force they could generate if tightened to their full strength.

2.1.2 Stability of the Clamping Force

The stability or reliability of the clamping force is a more complex issue, because a number of material properties can affect it, often without affecting the strength of the parts. For example, the clamping force introduced at assembly can be modified by temperature changes, by corrosion, or by external loads on the joint, depending in part on the way the bolt material responds to these things. We'll look at a number of pertinent properties in this chapter.

Most of the ways in which stability can be affected will be discussed in later chapters. I've put examples of all of the relevant material properties in the present chapter, however, because it's easier to pick a material (once you know the properties you're interested in) if all of the necessary information is in one place. Comparisons and trade-offs are easier. So, properties are discussed in this chapter, and how to use them in subsequent chapters.

As an introduction to the concept of stability, however, here's a brief summary of some of the ways in which the clamping force can be modified by environmental factors and by our choice of bolt material.

2.1.2.1 Thermal Expansion or Contraction

A change in temperature will change the length of the bolts and the thickness of joint members. Knowing the thermal coefficients of linear expansion will allow us to estimate how much change each part will experience.

If the parts are made from different materials—or are raised to different temperatures—the clamping force on the joint and the tension in the bolts will be modified by differential expansion or contraction. This can increase or decrease the clamping force. It can also break bolts or totally eliminate the tension in them. We'll see how to estimate these changes in Chapter 11 but will look briefly at the problem in Section 2.10.

2.1.2.2 Corrosion

The resistance of the bolt material to corrosion will determine how long our clamp will survive in the anticipated service environment. The buildup of corrosion products (e.g., rust) can increase clamping forces; additional corrosion can eat through the bolts. We'll consider corrosion mechanisms and stress corrosion cracking in Chapter 16.

2.1.2.3 Fatigue Rupture

Many materials have an "endurance limit" which, unfortunately, is only a fraction of their apparent (static tensile) strength. If cyclic stress levels are above this endurance limit, the bolt will eventually break and clamping force will be lost. So, the endurance limit is another property we'll be interested in. Fatigue will be the topic of Chapter 15.

2.1.2.4 Loss of Strength with Temperature

As already mentioned, a change in temperature can cause a change in clamp force because of differential expansion between bolts and joint members. Temperature can create problems even if these things have identical thermal coefficients and identical temperatures, however. The basic strength of the material can be affected enough by high temperature to put the joint in jeopardy.

2.1.2.5 Loss of Clamping Force with Temperature

Elevated temperature can also lead to stress relaxation (discussed in Chapter 11), which can reduce or eliminate the clamping force without any visible or measurable change in the parts. So resistance to stress relaxation is another material property that can affect the integrity of the clamp. Figure 2.3 and Table 2.3 provide stress relaxation data on a number of fastener materials.

Stress relaxation can and often does take place over an extended period of time. The lower the temperature, the longer it takes the bolt to shed stress. Clamping force can also, however, be lost very rapidly if common bolt materials are subjected to high temperature—during a fire, for example. Typical results are shown in Figure 2.4.

2.1.2.6 Elastic Stiffness of the Parts

The modulus of elasticity is another property we'll often be interested in. Modulus, in part, determines the stiffness of bolts and joint members, and stiffness in turn determines how the clamp force introduced at assembly will change when the joint is put in service. Factors like working loads (pressure, weight, shock, etc.), gasket creep, embedment of thread surfaces, elastic interactions between bolts—all to be considered in later chapters—will work to change those initial clamping forces even at room temperatures. The amount of change will depend on the relative stiffness of bolt and joint members (discussed at length in Chapter 5).

2.1.2.7 Change in Stiffness with Temperature

The modulus of elasticity is also affected by temperature; so the stiffness of bolts and joint members will change as the temperature changes. As one result, a 10% reduction in modulus means a 10% loss of tension in the bolt because it has become a less stiff spring. Note that a reduction in modulus occurs with an increase in temperature, and this increase may cause differential thermal expansion which partially or wholly offsets the loss in stiffness.

Any change in stiffness may also mean a change in the bolt-to-joint stiffness ratio, and this means a change in the way the system responds to external loads. So, there are many ways in which a change in temperature can modify the clamping force, with modulus playing several roles.

2.1.2.8 Brittle Fracture

Ductility can be another important consideration, especially if the bolts are to be tightened past yield (a common practice in structural steel work, as we'll see in Chapter 8). Very hard materials can be very strong—but brittle. The brittleness often leads to unexpected failure at loads below the theoretical strength of the parts. We'll look at some brittle fracture data in this chapter (Figure 2.2).

2.1.3 Miscellaneous Properties

Although things which determine or threaten the clamping force produced by the bolt will always be our main concern, there are times when other material properties must also be considered. Low-weight fasteners, for example, have always been important in aerospace applications and are of growing importance in automotive design as well (lower weight means lower fuel consumption). Some material weights are given in Table 2.6.

The cost of a fastener sometimes influences our choice, but the asking prices are subject to too much change to report them in a textbook. The electrical or magnetic properties of the fasteners can also be a consideration, but so rarely that we won't worry about them here.

The strength of joint members is usually not as big an issue as the strength of the bolts, but some data are useful and will be found in Table 2.7.

So, there are a number of material properties that will determine the ability of our bolt to clamp things in service, and other properties which will influence our material decisions in special situations. How can we select an appropriate material? The most obvious source of material information on which to base a decision would be an existing fastener "standard."

2.2 FASTENER STANDARDS

It is believed that some 500,000 fasteners have been defined by standards of some sort. Certainly hundreds of different specifications, recommendations, etc. are available today. The impact of fastener standards on our economy as a whole must be enormous—when you consider the alternative that "everyone designs and builds his own."

Fastener standards are published by several types of organizations, including the following:

Government organizations—for example, the National Institute of Standards and Technology (NIST), the Army, the Navy, the Air Force.

Engineering societies—important fastener standards are published by the Society of Automotive Engineers (SAE), the American Society for Testing and Materials (ASTM), the American Society of Mechanical Engineers (ASME) etc.

Trade associations—for example, the American Bureau of Shipping and the Association of American Railroads publish well-known fastener standards.

Fastener manufacturers—the principal U.S. source is the Industrial Fastener Institute (IFI), an association of fastener manufacturers. IFI publishes, among other things, a complete list of other people's standards. More importantly they sponsor basic research to develop new standards, some of which are later adopted by other organizations such as the ASTM.

Standards associations—general-purpose groups that publish standards on all sorts of things. Principal ones at the moment are the American National Standards Institute (ANSI) and the International Standards Organization (ISO).

Trade associations, military services, engineering societies, etc. tend, of course, to publish standards affecting fasteners in which they have a special interest. Groups such as NIST, ANSI, and IFI publish standards on all sorts of fasteners.

In general, standards cover such things as fastener materials, mechanical and physical properties, strengths, configurations, dimensions, usage, definitions, finishes, test procedures, grade markings, and manufacturing procedures. This does not mean that every standard covers each of these things—just that standards exist for all of these topics and for others.

The full names and addresses of the organizations from which you can buy standards are given in Appendix C. Web site addresses are also included for the first time.

2.3 SELECTING AN APPROPRIATE STANDARD

The purpose of a fastener standard is to define a group of materials and fastener configurations which are appropriate for the typical needs of a particular industry or a particular class of applications. The standard then makes it unnecessary for each engineer to be a metallurgist when trying to determine what would be appropriate in his application. This saves a great deal of time and money. Standards also reduce product and inventory costs, control quality, enhance product and system safety, and do other important things. For our purposes, we're interested in them as a source of material information.

If you work for an automobile manufacturer, your first choice of standards is simple—you try those prepared by the SAE. If you're involved with pressure vessels, you'll be guided by the ASME.

What do you do if your own industry has not produced a set of bolting standards? There are two readily available sets that are widely used by miscellaneous industries and designers. I'm sure that the most commonly used bolting standard, whether users realize they're using it or not, is SAE's J429, which defines automotive Grades 1 through 8. Bolts of these materials are made in large quantities and are therefore relatively inexpensive as specified or as standardized bolts go. They're readily available, and they cover a wide range of strength specifications, with tensile strengths ranging from 60 ksi (414 MPa) for the cheaper materials to 150 ksi (1034 MPa) for the more expensive Grade 8s. They're available in sizes ranging from ¼ to 1½ in. in diameter. Therefore, they're widely used for small- and medium-sized bolting jobs.

Another widely used set of bolting standards is published by ASTM. Several dozen ASTM standards are available, covering a wide range of threaded fastener materials, sizes, special applications, etc. You can buy individual standards or buy a bound volume containing many of them, and you can obtain them through ASTM's Web site. These standards will be your first choice if you're working with large equipment or systems—structures, off road equipment, pressure vessels, power plants, and the like. Bolting materials are managed by two ASTM committees. Committee A01 manages materials used for pressure vessel, piping, and special purpose applications. These include such popular materials as A193, A194, and A540. Committee F16 manages the rest, which are published in vol. 01.08 of their *Annual Book of Standards* [65].

In most situations your needs will be covered by fasteners defined by either the ASTM or the SAE. In every industry, however, including the automotive, there are special applications where something better is required. And in some industries, such as aerospace, the unusual is usual. The tables and descriptions that follow are intended to help you identify available fasteners which are not normally used in your own industry, whether or not these are common in someone else's industry.

In earlier editions of this text I tried to list every property of interest for every material listed—tensile, yield, and proof strengths; thermal coefficients of expansion; endurance limits; and all the rest. But much of this information was not and is not available. Different industries face different problems and specify only those properties of interest to them. If the shear strength of an A193 B7 bolt wasn't of interest to the normal user, then no one is going to specify or report it. All of the information I could find is given in the third edition and is summarized in this present edition or is represented by typical or popular examples.

Before we get to the data, however, it is useful to discuss what is still a hot topic in some parts of the bolting world (in spite of the antiquity of the topic!)—the issue of metric fasteners versus English (U.S. or Unified or inch series) fasteners. The tables in this chapter include data on both kinds, where available. But, generically, should you be interested in inch series fasteners or in metric?

Automotive manufacturers in the United States have gone metric, as have some other manufacturers who depend a great deal on exports; but most manufacturers here are still using English or inch series fasteners. For those facing a decision, the choice will probably be based on economic, political, or marketing considerations. From a purely technical point of view, inch and metric fasteners are available in the same general strengths, with the same general properties. Currently specified metric fasteners have, however, been designed and specified more recently than most of their English counterparts. Past experience has revealed some problems with English materials and geometry; the latest metric standards have attempted, at least, to overcome these shortcomings. So metric fasteners may be slightly better technically; but the differences are slight.

Note that the decision to choose metric involves two things: fastener material and fastener configuration (or at least basic dimensions). We're primarily interested in materials in this chapter (and in this book). We'll discuss metric fasteners in Section 2.6 of this chapter.

2.4 BOLTING MATERIALS

Soon we'll look at some specific properties of materials. But first, a word of caution: The data which follow are for general reference only. The properties of an individual batch of bolts can differ from the norm. The specific heat treatment used by one manufacturer can differ from that used by another, and this will affect properties. You'll sometimes find different values for a given property of a given material in the standards published by different organizations. The ultimate strength, for example, may differ depending upon the needs or design practices of various groups.

Differences of this sort are especially common for the more exotic materials used in aerospace or extreme-temperature applications. The strength of such materials, as a matter of fact, is often tailored for a specific application.

The diameter of the fastener can also affect its strength and other properties, because some bolting materials cannot be through-hardened. Large diameters don't get fully hardened in the center, smaller diameters do; so large diameters have lower average tensile strengths (in ksi or MPa). Material near the outer diameter of the fastener will support more load than material near the center; so the average strength of the larger fastener is less.

The shape of a fastener can also affect its strength. We're going to assume normal ANSI or equivalent shapes in this text, and so will ignore the ramifications of special configurations. But you'll usually recognize an unusual configuration on sight and should be wary of assigning normal properties to it.

Another factor causing variation in the data is the slow but fairly continuous evolution of most bolting standards and specifications. The ASTM and other committees who write these documents meet periodically and alter them whenever new experience, new conditions, previous misunderstandings, etc. suggest that a change would be desirable. Some of the data in the following tables come from standards first published 20 or more years ago. I've reviewed more recent versions of most of these documents and, if I have found only minor changes, have probably left the earlier numbers alone. Once again, however, if your designs will affect life or safety, or if failure will have severe economic consequences, you should take your design data from the current editions of the pertinent documents rather than from a text of this sort. Use the tables that follow only as a shopping guide.

The data in the tables were taken from the ASTM or other standards or specifications cited unless otherwise noted by reference numbers. The latter refer to the references at the end of the chapter.

2.5 TENSILE STRENGTH OF BOLTING MATERIALS

Now we're going to look at the room temperature proof, yield, and ultimate tensile strengths of several traditional groups of fasteners, each group characteristically used by a different industry or in a particular set of situations. These tables will help the beginner fish in the correct pool. They are not intended to define every fastener material used in those industries or applications, but merely representative examples. If a range of strength is given it usually means that the larger diameters of fasteners (typically diameters over 1 in.) made from that material cannot be through-hardened, and therefore have a lower average strength.

Proof strength: The proof strength of a bolt is defined as the highest stress it can support without permanent deformation.

Yield strength: The yield strength is the maximum useful stress a bolt can support in most applications and probably involves slight permanent deformation.

Ultimate strength: A load equal to its ultimate strength would break the bolt.

All data given in the following tables are in ksi. To convert to megapascals (MPa), multiply by 6.895.

2.5.1 General Purpose/Automotive Group

The most commonly used fasteners that are described by a specification (unspecified ones are even more common) may be those defined in Standard J429 published by the SAE. Typical examples include:

Grade	Proof	Yield	Ultimate
2	55–33	57–36	74–60
5	85–74	92–81	120–105
8	120	130	150

2.5.2 Structural Steel Group

These fastener materials are used in buildings, bridges, and other structures. Two of them, A325 and A490, are commonly tightened past yield, on purpose, as discussed in Chapters 8 and 11. The specifications listed below are published by the ASTM (Committee F16).

Spec.	Proof	Yield	Ultimate
A325	85–74	92–81	120–105
A354	105–95	109–99	125–115
A449	85–55	92–58	120–95
A490	120–117	130	170–150

2.5.3 Petrochemical/Power Group

The following materials are commonly used in petrochemical and power plants, as well as in marine, mining, manufacturing, and other industries using heavy equipment. The structural steel group of materials is also commonly found in such applications. The specifications listed below are published by the ASTM (Committee A01). The wide range of yield strengths listed here, for a given material, results from the fact that these fasteners are often made in large sizes—up to several inches in diameter—and the materials aren't always through-hardened. As mentioned earlier, the ASME publishes equivalent standards in which the materials listed below are called SA193 B7, SA193 B16, SA540 B21, and SA540 B24.

Spec.	Grade	Proof	Yield	Ultimate
A193	B7	NA	105–75	125–100
A193	B16	NA	105–85	125–100
A540	B21	NA	150–105	165–120
A540	B24	NA	150–105	165–120

2.5.4 Metric Group

There are probably as many different specifications for metric fasteners as there are specifications for English ones. But, as discussed in Section 2.6, most metric standards have adopted a common series of strength designations. The data below, for example, can be found in either ASTM A568 or SAE J1199, among other places. Again, to convert the following data to metric MPa, multiply the number given below by 6.895.

Class	Proof	Yield	Ultimate
4.6	33	35	58
8.8	87	96	120
10.9	120	136	151
12.9	141	160	177

2.5.5 Extreme-Temperature Materials

2.5.5.1 American Society for Testing and Materials (ASTM) F2281 Materials

The new ASTM F2281 standard describes a large number of materials which can be used in high-temperature service [65]. Here are some examples. The data are given in ksi. Multiply by 6.895 to convert to MPa.

Grade	Temperature	Tensile Strength	Yield Strength
600	600°F (316°C)	89.0	34.0
600	1800°F (982°C)	11.0	5.0
660	800°F (427°C)	138.0	93.0
660	1200°F (649°C)	104.0	88.0
660	1500°F (816°C)	36.5	33.0
718	600°F (316°C)	184.0	156.0
718	1400°F (760°C)	124.0	116.0

Standard F2281 also includes, in its appendix, a "Guide to Service Applications," which lists the maximum temperatures at which several materials can be used for continuous or intermittent service. Here are a few continuous service examples.

Grade	Temperature Limit
304 or 316	1600°F (871°C)
309	2000°F (1093°C)
330	2200°F (1204°C)

2.5.5.2 Traditional High-Temperature Materials

I consider many of the fasteners in this group "exotic"—they're uncommon and often expensive (see Table 2.4). I've also listed temperature limits of traditional ASTM A193 materials in Table 2.5. An aircraft designer might not consider an A193 temperature limit of 800°F high temperature—but the petrochemical engineer does so. As you can see some of the materials listed in Tables 2.4 and 2.5 can be used at temperatures as high as 3000°F (1648°C) or can be used at cryogenic temperatures.

Spec.	Grade	Temperature °F	Temperature °C	Yield Strength[a] ksi	Yield Strength[a] MPa	Ref.
ASTM A193	B7	752	400	76	524	
	B8	800	427	17	117	[1]
	B16	800	427	76	524	
BS4882	B80A	1400	760	73	503	[3]
A286[b]		1200	649	88	607	[3]
MP35N[b]		−320	−196	345 UTS	2379 UTS	[2]
		1000	538	225 UTS	1551 UTS	[2]
H-11		1000	538	141	972	[3]
Inconel, X-750[b]		1500	816	44	303	[3]
Waspaloy		1600	871	75	517	[3]
Rene 41		1600	871	80	552	[3]

[a] Values given are yield strengths for the cited temperature except for those values marked UTS, where strength is given as ultimate tensile strength instead of yield.
[b] Also recommended for cryogenic applications.

2.5.6 Corrosion-Resistant Group

As discussed in Chapter 16, no material will resist all types of corrosion. But ASTM A193, BS4882, A286, and MP35N, listed above, are all considered corrosion resistant. Other materials with this reputation include those listed below. See Chapter 16 for a more complete list.

Spec.	Grade	Yield Strength ksi	Yield Strength MPa	Ref.
Carpenter Gall-Tough	430	40	276	[6]
Nitronic 50		70	483	[7]
Inconel	600	37	255	[3]
AISI	316	45	310	
AISI	416-H	95	655	
Titanium	6A1-4V	128	883	[3]
Stainless steel	17-4PH	128–185	883–1276	[9]

2.5.7 Two New ASTM Bolting Standards

Virtually all of the bolting standards we deal with have been around, in one form or another, for decades. They are under constant review and are often revised slightly, but they are old friends. Two new bolting standards, however, have been adopted by the ASTM in recent years. These are ASTM F2281 "Standard Specification for Stainless Steel and Nickel Alloy Bolts, Hex Cap Screws and Studs for High Temperature Service" published in 2004 and F2282 "Standard Specification for Quality Assurance Requirements for Carbon and Alloy Steel Wire, Rod and Bars for Mechanical Fasteners," first developed by IFI and adopted by the ASTM in 2003. Although the latter is described as a quality control standard, it tabulates the mechanical properties of 90% of the materials used by the fastener industry, including many or most of those specified by ANSI, ASME, SAE, ISO, NIST, and others [66].

Here are some examples of the room temperature tensile strength of the wide range of materials covered by these new standards. The data are given in ksi. Multiply the numbers in the tables by 6.895 to convert them to metric MPa.

2.5.7.1 Room Temperature Strengths of ASTM F2281 and F2282 Materials

F2281 Grade	Grade	Tensile Strength
316	75	30
431	125–180	140
600	130	85
718	180	150
600	130	85
718	180	150

F2282 Grade	Tensile Strength
IFI-1006	60–62
IFI-1033	82–89
IFI-1514B	93–99
IFI-4140	90–102

So much for fastener groups. Again, these tables are far from complete. They're merely intended to give you typical data about the most widely used fastener materials. Additional information and data closely related materials will be found in the referenced standards.

2.6 METRIC FASTENERS

The current effort to get countries now using English units to adopt the more widespread metric system includes, as it must, a new, international metric standard for fasteners. Note that, heretofore, there hasn't been just one metric standard. Engineers could and did write as many different metric standards as we have English ones. But since most countries have to accept some change in their own standards to comply with a new international standard, the current effort is seen as a new—and maybe last—chance to reduce the vast number of fastener types, sizes, materials, etc. now currently available. Having standards of any sort was a start—there was a time when we didn't even have that. But now we have a chance to simplify things still further. In the long run, such simplification could more than partially offset the cost of changing drawings, tools, inventories, manuals, procedures, etc. as we adopt the new fasteners.

The numbers used to define the metric grades (called classes) have useful meaning. The first number is equal to the minimum tensile strength of the material, in MPa divided by 100. The second number represents the approximate ratio between minimum yield and minimum ultimate strengths for the material. Hence, Class 5.8 has a minimum ultimate strength of approximately 500 MPa, and its minimum yield strength is approximately 80% of its minimum ultimate strength.

All this is far more useful than calling out "Grade 5" or "B7" and letting the uninitiated struggle to find out what that means—as we have always done in the past.

When we're dealing with metric fasteners, we'll want to use metric units for such things as torque, stress, force, etc. You'll find conversion factors (English to metric and vice versa) in Appendix D.

2.7 EQUIVALENT MATERIALS

A review of Section 2.5 will show that the sets or groups of materials favored by different industries usually cover approximately the same range of yield strengths, with the strength of common sets ranging from something like 30 to 105 ksi or so. As a result, you'll often find that a material, or at least a room temperature strength rating, in one group is matched by a similar or equivalent material or strength in one or more other groups. Substitutions are sometimes possible. In critical applications, however, you should look at the original specifications for both materials. In many applications "equivalent strength" isn't the only criterion for selection. Response to a change in temperature or to corrosion or other factors may differ.

But equivalent materials are available. As one example, all of the following define basically material of the same strength.

AISI 4140	ASTM A193 B7
ASTM A194—GR 7	ASTM A320—GR L7
Metric 9.9	SAE J429 GR 5+
BS 970-En 19A	BS 1506-621 A
DIN 267—9.9	ASTM SA193—B7

As mentioned earlier, the new ASTM standards F2281 and F2282 cover most popular bolting materials; so additional equivalents can be found there. As an example, F2282 Grade IFI-4140 material is the equivalent of AISI 4140, and therefore of all of the other materials listed above.

2.8 SOME COMMENTS ON THE STRENGTH OF BOLTING MATERIALS

2.8.1 In General

The preceding tabulations give typical room temperature strengths of various male fastener materials (bolts, studs, screws) under static, tensile loads. This strength is expressed in three different ways—proof, yield, and ultimate tensile—depending on the available data.

Other aspects of the room temperature strength of bolts are covered later: namely, shear and brittle fracture strengths. A chart giving the relationship between yield strength and material hardness is also included. This can help you identify or estimate the strength of unmarked or suspect materials. Again, only male fastener materials are included here. Nut materials are covered in Table 2.2.

It is sometimes necessary to determine what material an unmarked bolt is made from, or to estimate its strength. A metallurgical analysis is required for a completely accurate answer; but it has been shown that there's a rough correlation between the hardness of a bolt and its yield strength (within reason—see the discussion of brittle fracture strength below). Studies of low-alloy, quenched, and tempered steels (LAQT steels) for example, sponsored by the Atomic Industrial Forum/Metals Properties Council Joint Task Group on Bolting, resulted in the curve shown in Figure 2.1. Examples of an LAQT steel would be AISI 4140, ASTM A193 B7, ASTM A490, SAE J429 GR 5, etc.

2.8.2 Shear Strength

Since most bolts are used as clamps, not as shear pins, most bolting specifications and standards list only one or more forms of tensile strength (proof, yield, or ultimate) and not shear strengths. Table 2.1 lists those few materials for which I've found published or reported shear strengths.

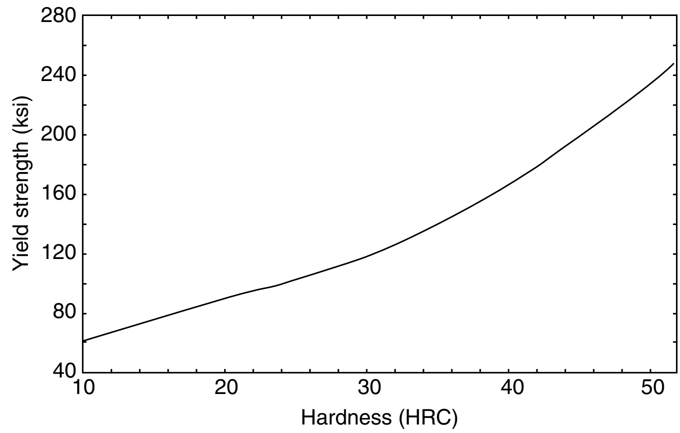


FIGURE 2.1 Yield strength versus hardness for low-alloy, quenched, and tempered steels such as ASTM A193 B7.

TABLE 2.1
Shear Strength

Material	Ultimate Strength (ksi)	Shear Strength (ksi)	Notes	Ref.
ASTM A325	132	79		[16]
ASTM A490	165	96		[16]
Stainless steel				
A286	232	119		[17]
431	230	128		[17]
Custom 455	269	151		[17]
PH12-9Mo	280	148		[17]
PH13-8Mo	243	142		[33]
Cryogenic materials				
A286	214	116	At 70°F	[18]
A286	291	282	At 423°F	[18]
Unitemp 212	214	132	At 70°F	[18]
Unitemp 212	278	166	At 423°F	[18]
Inconel 718	226	138	At 70°F	[18]
Inconel 718	291	168	At 423°F	[18]
Miscellaneous materials				
H-ll	238	135		[33]
H-ll	269	159		[33]
Marage-300	277	149		[33]
MP35N	283	162		[33]
Ti 6A1-4V	200	180		[18]
Beryllium	75	40		[18]
Steel	200	240		[18]
MP159		132		[58]
Fastener grade aluminum				
2024-T4	68	41		[57]
6061-T6	45	30		
6262-T9	58	35		
7050-T73	75	45		
7075-T6	83	48		
7075-T73	73	44		

Source: ASTM specifications A325, A490, A286; and references cited at the end of the chapter.

If the material you're interested in is not included in Table 2.1, it might help to know that most of the common steels we'll use, with hardness to 40 HRC or so, will have shear strengths in the neighborhood of 60% of ultimate tensile strength, or, in MIL-Handbook 5 terms, Fsu = 0.6 Ftu [63]. The stainless steels are an exception to this rule of thumb; they have shear strengths which are about 55% of their ultimate strengths, or Fsu = 0.55 Ftu [63].

Note that these are only rules of thumb and that other sources give other rules. The Unified Inch-Series Thread Standard ASME Bl. 1-1989, for example, says that the shear strength of threads is half the tensile strength of the material from which the external threaded part is made. This reflects the fact, I suppose, that nut materials are supposed to be slightly weaker than the bolt materials they're used with, or there's a small safety factor introduced here to reduce the chances that the threads will strip.

A still different rule comes from "the" book on structural steel joints [49], which suggests that the shear strength of most common (structural steel) joint materials is 70% of the ultimate strength of those materials.

2.8.3 Bearing Yield Strength

Another material property we often need when dealing with the shear strength of the bolts is the bearing yield strength of the joint material. We want the bearing yield strength of the joint members in a shear-loaded joint to be less than the shear strength of the bolts. If this is the case, the walls of the bolt holes in the joint plates will yield before the bolts shear. This will usually bring more bolts into bearing, reducing the shear loads on the first bolts to contact the walls of their holes.

As another rule of thumb, the bearing yield strength of common joint materials is about 1.5 times the material's tensile yield strength, or, in MIL-Handbook 5 nomenclature, Fby = 1.5 Fty [63].

The strengths in Table 4.1 are given in ksi. To convert to metric MPa, multiply the value given by 6.895. For example, a shear strength of 132 ksi would be the equivalent of $6.895 \times 132 = 910$ MPa.

2.8.4 Hardness versus Strength

Simplistically, a harder steel is always a stronger steel. A small fastener can do a big fastener's job—saving weight, reducing joint size, etc.—if the small one is hard enough. It's appealing!

Harder fasteners tend to be more brittle, however, and are therefore more susceptible to unexpected brittle fracture, the failure originating from an undetected crack or flaw in the surface. The harder the fastener, the greater the chances of such failure. Attempts have been made, therefore, to specify the safe limits of hardness and strength which we can count on [14,15]. Figure 2.2 shows the results for LAQT steels such as AISI 4140, A193 B7, and similar materials.

Curve A shows the yield strength of LAQT steels as a function of increasing hardness, a repeat of Figure 2.1. An increase in hardness produces an apparently limitless increase in strength [14].

These results are supported, in the HRC 40–50 region, by curve B, which shows the results of a series of tensile tests made on alloy steel bolts heat-treated to different hardnesses and tested in a tensile machine [15]. This time the information plotted is ultimate strength, not yield strength; so curve B is slightly above curve A in the region of overlap.

As before, curve B shows a steady increase in strength as hardness is increased—but only up to the point described as "critical hardness" [15]. Increasing the hardness progressively beyond this point (or region: the data showed quite a bit of scatter) resulted in brittle

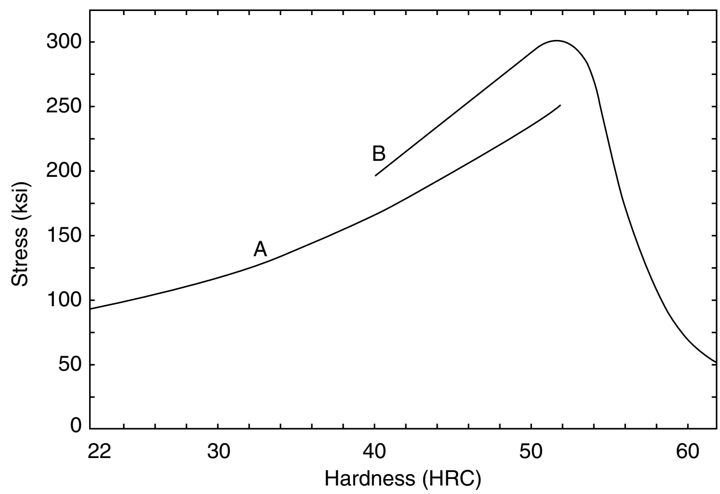


FIGURE 2.2 Curve A is a repeat of the yield strength versus hardness curve for low-alloy, quenched, and tempered steels (Figure 2.1). Curve B is a plot of ultimate strength versus hardness for similar materials. The dramatic fall in curve B beyond hardness HRC 52 reflects the tendency of very hard steels to fail early by brittle fracture.

failure at progressively lower and lower tensile loads. Presumably stress concentrations were created by cracks or flaws, and the material became less and less able to shed or reduce these concentrations by localized plastic yielding in the vicinity of the crack. Eventually the bolts would break. They were truly supporting high stress levels, as an increase in hardness says they should, but only in a local area. The average stress across the full bolt, when it broke, was less than the average stress a softer, more ductile bolt would support.

2.9 NUT SELECTION

A threaded fastener consists of a bolt and nut (or tapped hole). Knowing the strength of the bolt alone is not sufficient, since it's never used alone. We also want a nut or tapped hole which will develop the full strength of the bolt.

Specifications such as ASTM A194 and A563, SAEJ995, etc. tabulate the proof, yield, and ultimate strengths of nuts the way ASTM A193, A449, and SAE J429 tabulate bolt strengths. Listing all of that information would nearly double the number of tables in this chapter. But I don't think that's necessary.

Most designers and users focus on the strength and other characteristics of the bolt or stud. Having selected a bolt, they then want to choose an appropriate nut, or design a suitable tapped hole. As far as tapped holes are concerned, you can use the thread strength equations of Chapter 3 to select such things as length of thread engagement. When it comes to nuts, most designers will use those recommended in the pertinent bolt specification they're using—or in the nut standard to which the bolt standard refers.

Table 2.2 summarizes these bolt–nut recommendations. If you want to confirm that the recommendations are correct for your application, you can refer to the standards cited in the table. If you're bothered by the fact that several grades of nuts are sometimes listed for a single bolt material—that's the way the nut or bolt standard does it too. Any of the several nuts listed are acceptable for that bolt. You can base your choice on

TABLE 2.2
Which Nut Material to Use

Bolt Spec.	Grade	Nut Spec.	Grade
ASTM	B5	ASTM	Any
A193	B6	A194	Any
	B7		2H, 4, 7, or 8
	B7M		2HM, 4M, 7M, 19, 8M
	B8CL1		8
	B8CL2		8
	B16		4, 7, or 8
ASTM		ASTM	A, B, C, D, DH,
A307	A or B	A563	DH3
ASTM	L7	ASTM	4 or 7
A320	L43	A194	4 or 7
	L7M		7M
	LI		4 or 7
	B8CL1		8
	B8CL1A		8
	B8CL2		8
	B8MCL2		8
ASTM		A194 or A563	2 or 2H
A325			
A354		A563	C3, D, DH, DH3
A449		A563	C3, D, DH, DH3
A453		A453	C3, D, DH, DH3
A490		A563 or A194	DH, DH3, 2H
A540		A540	
SAE	1	SAE	2
J429	2	J995	2
	4		2.5
	5		5.8
	7		5,8
	8		8
Metric	4.6		4
	4.8		4
	5.8		5
	8.8		8
	9.8		9
	10.9		12
	12.9		12
Special combinations to resist galling			
Stainless steel			
	316	Stainless steel	400 series
Nitronic 60		Nitronic 50	
Cold drawn	316	Cold drawn	316
Low-alloy steel A193			
	B7, B16	Stainless steel	300 series

Source: ASTM specifications A194, A563, A453, A540; SAE specification J995; and references cited at the end of the chapter.

availability, cost, your desire to standardize your own inventories, or on a special requirement for your application.

In some cases you'll be using a bolt which is not included in Table 2.2. Nut recommendations are not made, usually, for the more exotic bolt materials, for example. In such cases, you'll find the following general guidelines of use:

1. In general, we want the nut or tapped hole to support more load than the bolt, because bolt failures are easier to detect than stripped nuts. One rule of thumb: the nut's proof load should approximately equal the ultimate tensile strength of the bolt [21].
2. In an apparent contradiction of the above, nuts are usually (but not always) made of slightly softer (less strong) material than the mating bolts, so that the nut threads will yield locally and better conform to the bolt threads when loaded. This better distributes the stresses in both parts.

Note that the strength of a part depends on dimension, shape, etc. as well as choice of material. This is why it is possible to make a stronger nut from a weaker material.

3. If you need a nut for a bolt made of a special material such as titanium, you'd presumably make the nut of the same material. The nut might be heat-treated differently, however, to satisfy point 2 above. Again, this is not a universal custom. Note that a nut made of a different material than the bolt could cause galvanic corrosion.
4. The strength of a nut depends, in part, on the length of thread engagement. To increase safety or reduce failure rates, you can consider increasing the length of engagement. Doubling the length of the nut, however, won't double its strength; the relationship is more complex than that, as discussed in Chapter 4.
5. A nut's strength also depends on its width across flats. Thread surfaces at 60° to the axis of the bolt create radial forces which tend to spread the most heavily stressed portion of the nut (the first few threads of engagement) as the fastener is tightened. This process is called "nut dilation." Nuts with thinner walls dilate more—partially disengage themselves from the bolt under load—and so have less strength. This is one of the reasons why standard nuts are available in several configurations—hex nuts, thick hex nuts, and heavy hex nuts. These differ in height (thread length) too. Your choice will be based on economics and on the fact that most bolts (and nuts) aren't loaded anywhere near the limit of their strengths. As a result, a regular hex nut—or equivalent in a tapped hole—is sufficient for most applications. Thicker, heavier nuts would be preferred if the bolts are to be loaded to target preloads beyond 60%–70% of yield perhaps, or if the consequences of failure suggest that an extra degree of safety would be prudent.

Note a curious fact that bothers many bolting engineers. There are no standards which define the strength or behavior or properties of a bolt-and-nut system. Instead, we have separate standards for male and for female fasteners. And in some situations the dimensional tolerances on threads are such that the resulting thread engagement can be poor, sometimes resulting in a bolt–nut system which will be unable to develop the full load either could support if tested alone. Mechanically galvanized inch series ASTM A325 bolts and nuts, for example, can cause problems. See Yura et al. [16] for a lengthy discussion of the nut–bolt specification problem, as related to A325 and A490 fasteners and their metric counterparts.

One frequently asked question is "What nut should I use to minimize galling between nut and bolt?" Table 2.2 ends with a few combinations I have heard recommended by petrochemical or nuclear power engineers. They may be worth trying in your application.

2.10 EFFECTS OF TEMPERATURE ON MATERIAL PROPERTIES

The clamping force first exerted by the bolt on the joint at the time of assembly is affected by changes in temperature. If the operating temperature of the system is higher than room temperature, then the strength of the bolt will be less than it was at assembly. As a result, too high a temperature can cause a bolt—especially a heavily preloaded one—to fail. This is certainly a type of instability we want to avoid.

2.10.1 Thermal Expansion

Thermal expansion also encourages clamping force instability. Since few joints operate at a perfectly constant temperature, thermal expansion or contraction of bolts and joint members is virtually universal. This is only a problem, however, when we're dealing with dissimilar materials having different coefficients, or are dealing with a system in which bolts and joint members reach different temperatures in operation. Generally speaking, the change in temperature must be significant too, before we run into problems. Normal changes in ambient temperature don't, in my experience at least, cause problems.

Typical, room temperature coefficients of linear expansion for LAQT bolting materials range from 6 to 7×10^{-6} in./in./°F, with the full range for less common materials running perhaps from 5.5 to 10.2×10^{-6} in./in./°F. The values tend to be slightly higher at elevated temperatures, e.g., 7–10 at 800°F (427°C).

The data are in inches/inch/°F, as stated. To convert to metric units of mm/mm/°C, multiply the tabulated value by 5⁄9. For example, a coefficient of 6.5 in./in./°F would be the equivalent of $6.5 \times 5/9 = 3.61$ mm/mm/°C.

Let's look at an example. The temperature of a 1⁄4–20 × 4, J429 GR 8 bolt is raised from room temperature (70°F) to 400°F, and increase of 330°F. The length of the bolt will, as a result, be increased by

$$4 \times 6.5 \times 10^{-6} \times 330 = 8580 \times 10^{-6} = 0.0086 \text{ in.}$$

That may not seem like much but, as we'll see in Chapter 5, a 4 in. LAQT bolt will probably be stretched no more than 0.010 in. during assembly; so a change in length of 0.0086 could cause it to loose most of its preload if the joint doesn't expand under it by a similar amount.

Variations in the modulus of elasticity can also affect the clamping force, because the modulus, in part, determines the stiffness of bolts and joint members. Stiffness and stiffness ratios affect the degree to which the clamping forces introduced at assembly will change when the joint is subsequently subjected to external loads, temperature changes, and the like, as we'll see in Chapters 6, 10, and 11, for example.

Typical values for the modulus are 31×10^3 ksi at cryogenic temperatures, the familiar 30×10^3 ksi at room temperature, and perhaps 24×10^3 ksi at 800°F. This is true of many joint materials, too, but there are many exceptions here. For example, cast iron has a modulus of $12–14 \times 10^3$ ksi, ductile iron 22×10^3 ksi, and magnesium only 6.1×10^3 ksi: all at room temperature. Table 5.1 in Chapter 5 lists the modulus of elasticity for many bolt and joint materials.

Because the modulus changes as temperature changes the response of the bolt and joint to loads is often different at different temperatures. In fact, the temperature-induced change in modulus can alone change the tension in a bolt, all other effects aside, as we'll see in Chapter 11. So, modulus with and without temperature change affects the stability of the clamping force in several ways.

The modulus data above are in ksi. To convert these data to metric units in GPa, multiply the two digit numbers tabulated by 6.895. A modulus of 30×103 ksi, therefore, equates to $30 \times 6.895 = 207$ GPa.

Stress relaxation is a cousin to the more familiar phenomenon called creep. Creep involves the slow change in dimension (the strain) of a part subjected to a heavy load (stress). If we threaded one end of a stud into a ceiling, hung a heavy weight from it, and turned up the temperature in the room, the stud would slowly stretch and eventually break.

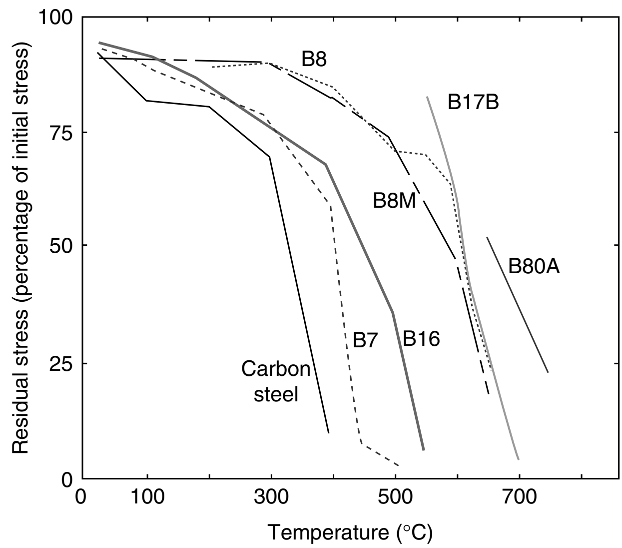


FIGURE 2.3 Stress relaxation of petrochemical bolting materials as a function of service temperature. Exposure in each case was for 1000 h at the temperatures shown. (From Standard BS 4882:1973, British Standards Institution, London, 1973.)

Stress relaxation, on the other hand, involves the slow shredding of load (stress) by a part under constant deflection (strain). A bolt which has been tightened into a joint, for example, is held in a constant, stretched condition by the joint members. The initial tension in the bolt will gradually disappear if stress relaxation occurs. Again, high temperature encourages the process. So stress relaxation is another possible source of "instability" of the clamping force created on the joint when we first tightened those bolts.

We'll look at stress relaxation in Chapter 11. As we'll see, it's only a problem at elevated temperatures.

Figure 2.3 and Table 2.3 give data on stress relaxation for a number of fastener materials. Data in Table 2.3 are from BS 4882 on petrochemical bolting materials [20].

Figure 2.3 shows the residual stress in a bolt after 1000 h of exposure to the temperatures shown. As an example, a carbon steel bolt will lose approximately 30% (retain 70%) of its initial preload if exposed to 300°C for 1000 h. The material references in Figure 2.3, B7, B8, B8M, and B16, are from BS 4882:1973 and correspond to equivalent materials in ASTM A193.

TABLE 2.3
Stress Relaxation High-Temperature Service Limit

Material (from BS 4882)	Temperature
Carbon Steel	572°F (300°C)
Mild Steel	572°F (300°C)
B7	752°F (400°C)
B16	932°F (500°C)
B16	968°F (520°C)
B8, B8T, B8C	1067°F (575°C)
B8M	1112°F (600°C)
B17	1202°F (650°C)
B80A	1382°F (750°C)

B17 is an A1SI660 austenitic alloy. B80A is more commonly known by its trade name, Nimonic 80A. Neither of these materials is found in A193.

Stress values in Table 2.3 are given in ksi. To convert to metric MPa units, multiply the value shown by 6.895. For example, a stress of 30 ksi would be the equivalent of $30 \times 6.895 = 207$ MPa.

A final word about Table 2.3. This table shows the maximum service temperatures which BS 4882 and equivalent materials can experience continuously without losing most of their preload or tension [20]. Comparison of some of these data with Figure 2.3 will show that these upper service limits define the final knee in the residual stress–temperature curves for the materials. Some stress relaxation will occur below these temperatures, but it won't be as extreme as the relaxation which will occur above them. See Figure 2.3 for identification of B7, B8, B17, etc.

In spite of stress relaxation there are many exotic aerospace bolting materials that can be used at very high temperatures and still retain enough strength and energy storage capacity to be useful. A number of these are listed in Table 2.4.

We're usually concerned about stress relaxation if our bolted joints will be exposed, on purpose, to high in-service temperatures for long periods of time. We use high-temperature materials for such applications and take their stress relaxation into account when estimating in-service clamping forces. We face different problems if our products are subjected to high temperature accidentally, during a fire, for example.

TABLE 2.4
Service Temperature Limits

Material Type	Name	UTS (ksi)[a]	Temperature Limit °F	°C	Ref.
Iron-base alloys	4340[b]	180	450	238	[58]
	PH13-8 M	220	650	344	
	Custom 455	220	650	344	
	Marage 300[b]	260	900	483	
	H-11[b]	260	900	483	
	A286	200	1200	649	
	TD-Ni-Cr	18[c]			
Nickel-base alloys	MP35N	260	700	372	[58]
	Inco 718	180	1200	649	
	Rene 95	230	1200	649	
	Rene 41	150	1400	760	
	Waspaloy	150	1400	760	
	Astroloy	190	1600	871	
Titanium-base alloys	Ti 1-8-5	200	300	148	[58]
	Ti 6-6-2	180	500	260	
	Ti 6-4	160	500	260	
				1093	
Haynes alloys	HA 188	45[c]	1800	982	[58]
	HA 214		2000	1093	
	HA 230		2000	1093	
Columbium base	Cb752	35[c]	2500	1371	[58]
	C 1294	35[c]	2500	1371	
Tantalum base	T-222	20[c]	3000	1648	[58]
	Ta-l0W	18[c]	3000	1648	

[a] At room temperature unless otherwise specified.
[b] Protective coating required.
[c] At maximum service temperature, not at room temperature.

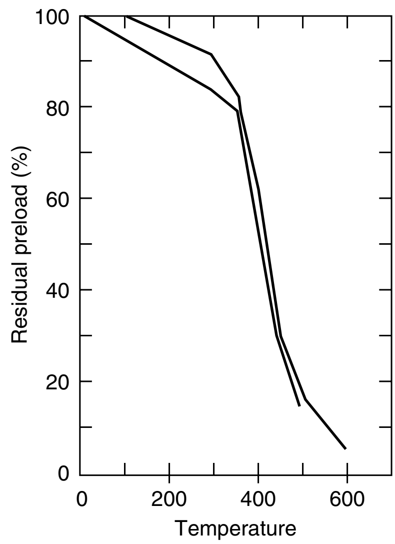


FIGURE 2.4 The residual preload in a group of structural steel bolts, as a percentage of the initial, room temperature preload, after 1 h soaking at the various temperatures shown. From this data the investigators concluded that fire temperatures above 350°C (662°F) and soak times greater than 1 h could seriously damage the integrity of a structure. Note that the diameter of the bolt influences the rate at which it loses preload. The upper curve is for an M30 bolt, 30 mm (about 1.2 in.) in diameter. The lower curve is for a smaller MI6 bolt. (From Wakiyama, K. and Tatsumi, A., *Trans. A.I.J.*, 19, 313, 1982.)

Figure 2.4 shows the response of structural steel bolts when exposed to high temperatures for only 1 h. These tests were conducted in Japan, but I assume that their structural steel bolts would be comparable to our A325 and A490 materials.

The people who conducted these tests conclude from the data summarized in Figure 2.4 that a slip critical structural steel joint (see Chapter 12 for definition of a slip critical joint) could safely be exposed to a fire temperature of 350°C (662°F) and still retain structural integrity [59]. Bolts tightened to a standard preload would lose perhaps 10% of this preload in 1 h. Bolts tightened to only 90% of standard preload would lose as much as 20%, however.

By further experiments they found that there's a cumulative time and temperature effect here. A clamping force loss of 20% was encountered in bolts exposed to any of the following combinations of time and temperature [60]:

250 h at 250°C (482°F)
20 h at 300°C (572°F)
0.7 h at 350°C (662°F)

Other bolted joints showed a 30% loss of clamping force after:

4100 h at 250°C (482°F)
880 h at 300°C (572°F)
3.2 h at 350°C (662°F)

As Figure 2.4 shows, higher temperatures resulted in greater loss; 1 h at 500°C (932°F) reduced tension in bolts to only 10% of their initial preload, for example. Rapid cooling—e.g., in water—increased the loss, incidentally. I still think it would be a good idea to let the firemen squirt water on your burning buildings!

TABLE 2.5
Cryogenic Bolting Materials Used to −423°F (−253°C)

Type	Name	Room Temperature UTS (ksi)	Ref.
Iron-base alloys	A286	140	[58]
	C.R. A286	200	
	PH13-8 Mo	220	
	Custom 455	220	
Nickle-base alloys	Rene	150	[58]
	Waspaloy	150	
	Inco 718	180	
	Astroloy	190	
	C.R. Inco 718	220	
	MP35N	260	
Titanium-base alloys	Ti 5Al-2.5Sn	110	[58]

So much for elevated temperatures. We're also sometimes concerned with extreme temperatures in the other direction. Table 2.5 and Figure 2.5 provide some data about the strength of certain exotic materials to very low, cryogenic temperatures [58]. Unfortunately, my source gave only the room temperature strengths for most of these materials, as you'll see from Table 2.5. Figure 2.5 suggests that their cryogenic temperature strengths could be higher than their room temperature strengths, but that there will be exceptions such as the particular titanium alloy shown in the chart. You'll have to contact aerospace bolt manufacturers for further information.

2.10.2 Miscellaneous Temperature Problems

Prolonged exposure to high temperature can embrittle a fastener, reduce its resistance to corrosion, and reduce its impact strength [65]; so, if safety is involved the designer or user would be well advised to seek expert advice if dealing, for the first time, with elevated temperatures.

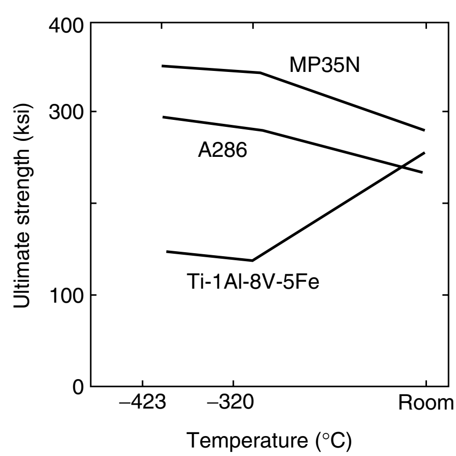


FIGURE 2.5 The ultimate strength of three exotic bolting materials at cryogenic temperatures. As suggested here, the strength of some materials is increased by a very low temperature, but it would be unwise to assume that this is true for all materials. The strength of another titanium alloy, Ti-5Al-5Sn-5Zr, increases between room temperature and –320°C but then decreases a bit if further cooled to –423°C. Its strength at that temperature is about 210 ksi, still above its room temperature ultimate of about 160 ksi. (From notes for a bolting seminar prepared by Ray Toosky of McDonnell Douglas, 1991.)

2.11 OTHER MATERIAL FACTORS TO CONSIDER

2.11.1 Fatigue Properties

Cyclic loads can cause a bolt to fatigue and break, destroying the clamping force on the joint. The number of load cycles a bolt will tolerate before failure is called its fatigue life.

This fatigue life depends on many factors, as we'll see in Chapter 15. The data are given in two different ways: as endurance limit or as fatigue strength. The endurance limit is the maximum tensile stress the bolt can stand for an infinite number of cycles, infinite life. Fatigue strength is the maximum stress the bolt will stand for an average life of given number of load cycle. As an example, an AISI 4340 bolt, heat-treated to a room temperature ultimate tensile strength of 165 ksi, can be expected, on the average, to tolerate 1000 cycles of load if that load never exceeds 10 ksi. Your bolts might stand more or less load than this, for 1000 cycles; but your results would probably be similar to those reported here. You'd be unlikely to get 10 times this much life, or one-tenth of it.

2.11.2 Corrosion

Predicting the useful life of a fastener in a corrosive environment is perhaps even more difficult than predicting its life under cyclic, fatigue loads. Much depends on the nature of the electrolyte, which can take an almost infinite number of forms, concentrations, etc. Temperature can play a large role. So can geometry of the parts, stress levels, stress concentrations, the properties of mating parts, surface flaws, crevices between parts, fluid velocity, and many other factors—as we'll see in Chapter 16. When selecting a fastener material, however, it can be useful to have a checklist of available materials which are considered "corrosion resistant" in general. A list of corrosion resistant materials can be found in Section 2.5 of this chapter. You'll find a longer list in Chapter 16.

2.11.3 Miscellaneous Considerations

The data in Table 2.6 often influence the selection of a fastener material and may dominate the selection in some cases. Weight is especially important in aerospace applications and is coming to mean more and more in automotive work. Cost is always of concern to the responsible designer.

As far as cost is concerned, however, remember that it is influenced by many factors—the quantities purchased, the popularity of a particular size or configuration of fastener, economic conditions at the time of purchase, the competition for a particular market, trade laws and tariffs, etc. The prices charged for such commercially available materials are also subject to more or less frequent change.

On the other side of the argument, however, remember that the material cost of a fastener is only part of the job cost. If you use a more expensive (stronger) material, you may be able to use fewer and smaller fasteners. Joint members can also be scaled down, in many situations. Assembly costs may be reduced because you don't have to tighten as many bolts or open preload tolerances. Warranty and liability claims may be reduced by fewer fastener failures. In short, true cost is a complex issue.

2.12 JOINT MATERIALS

We've already considered some joint material properties—the coefficient of linear expansion and the modulus of elasticity, for example—because these things affect the stability of the clamping force. Now we're going to look at the room temperature strength of some typical joint materials. I call these typical because the choice of joint material is virtually unlimited. But the data in Table 2.7 can be found in ASME, SAE, or other standards.

TABLE 2.6
Relative Weights of Fastener Materials

Fastener Material	Relative Weight
Nylon	0.05
Beryllium	0.066
Aluminum (2024-T4)	0.100
Titanium 6A1-4V	0.16
Steel	0.28
H-11	0.282
Waspaloy	0.286
A286	0.286
Rene 41	0.298
Inconel X-750	0.298
Inconel 600	0.301
Naval brass	0.304
K-Monel	0.305
Phosphor bronze	0.320

Source: From Hood, A.C., *Mach. Des.*, December 1981; Sproat, R.L., *Mach Des.*, 38, 107–122, 1966; *1984 SAE Handbook*, SAE, Warrendale, 1, 10.137–10.139, 1984; *1984 SAE Handbook*, SAE, Warrendale, 1, 10.22, 1984.

TABLE 2.7
Room Temperature Strength of Typical Joint Materials (ksi)

	Strength			
Joint Material	**Yield**	**Tensile**	**Shear**	**Ref.**
Structural steels				
Low-carbon steels (A36, Fe37)	33–36	58–80	41–56	[48]
High-strength steel (A588)	42–50	63–70	44–49	[48]
High-strength, low-alloy steel (A242, A441, A572, Fe52)	40–65	60–80	42–56	[48]
Quenched and tempered carbon steel (A537)	50–60	70–100	49–70	[48]
Quenched and tempered alloy steel (A514, A517)	90–100	100–130	70–91	[48]
Automotive materials				
Steels				
SAE J414				
1010 hot rolled	26	47	35	
1010 cold drawn	44	53	37	
1020 hot rolled	30	55	41.2	
1020 cold drawn	51	61	43	
1035 hot rolled	39.5	72	54	
1035 cold drawn	67	80	56	
Aluminum die castings	14–24	46	19–29	
SAE J453				
Grade 303, 306, 308, 309				
Grey iron castings				
SAE J859				
G1800	—	18	—	
G2500	—	25	31	[55]
G3000 etc.	—	30	38	

continued

TABLE 2.7 (continued)
Room Temperature Strength of Typical Joint Materials (ksi)

Joint Material	Strength: Yield	Strength: Tensile	Shear	Ref.
Malleable iron castings				
SAE J158				
M3210	32	50		
M4504	45	65		
M5003	50	75		
M5503	55	75		
M7002	70	90		
M8501	85	105		
Ductile iron castings				
SAE J434				
D4018	40	60		
D4512	45	65		
D5506	55	80		
D7003	70	100		
Steel castings				
SAE J435				
ASTM A27	30	60–65		
ASTM A148 low alloys	50–60	80–90		
ASTM A148 high-strength alloys	85–145	105–175		
Pressure vessel and piping flanges				
Ferritic steels for high-temperature service				
ASTM A182, GR. Fl, F2, F5, F7, Fll, F12, etc.	40	70	49	[50]
ASTM A182, GR. F6a Cl 2	55	85	60	[50]
GR. F6a Cl 3	85	110	77	[50]
GR. F6a Cl 4	No	130	91	[50]
GR. F6b, F6NM	90	110–135	77–95	[50]
Austenitic stainless steels for high-temperature service				
ASTM A182, GR. F304, F310, F316, F347, F321, etc.	30	75		[50]
Grades F304N, F316N	35	80		[50]
Grade FXM-19	55	100		[50]
Gray iron castings				
ASTM A278 Cl 20	—	20		[50]
C130	—	30		[50]
C140	—	40		[50]
C150 etc.	—	50		[50]
Ductile iron castings				
ASTM A395	40	60		[50]
Alloy steel forgings for low-temperature service				
ASTM A522	75	100		[50]
Austenitic ductile iron castings for low-temperature service				
ASTM A571-71	30	65		[50]

Source: ASTM specifications A36, Fe37, A588, A242, A441, A572, Fe52, A537, A514, A517; SAE specifications J414, J453, J859, J158, J434, J435; Kulak, G.L., Fisher J.W., and Struik, J.H.A., in *Guide to Design Criteria for Bolted and Riveted Joints*, Wiley, New York, 1987, 9–11; *ASME Boiler and Pressure Vessel Code*, Section II, Part A, ASME, New York, 1980; Juvinall, R.C., in *Engineering Considerations of Stress, Strain and Strength*, McGraw-Hill, New York, 1967, 559.

The strength of joint materials is usually not a major factor in bolted joint analysis or design. If the bolt holes are placed too close to the edge of joint members, however, they can tear out under shear loads, as we'll see in Chapter 3. More commonly, we'll manage to strip the threads from a tapped hole, another failure in shear of the joint material.

Although such failures are reasonably common, you'll find that most bolted joint failures can be traced to failure of the bolts, or, even more commonly, to poor assembly practices. Nevertheless, some joint strength data are useful.

Incidentally, although many of the joint material failures we'll experience are in shear, information on shear strength is especially difficult to find. Apparently, shear tests are difficult to make and give scattered results. Therefore, in Table 2.1, I've often resorted to Fisher's comment that the shear strength of most common steels is usually about 70% of the tensile strength [49]. He's dealing with structural steels, so I wouldn't extend that rule of thumb to castings or aerospace materials or other things which may be common in other industries. But I must confess I haven't found much data.

The information in Table 2.1 comes from the standards cited and from the references given in the final column.

The values in the table are given in ksi. To convert to metric MPa units, multiply the number shown by 6.895. For example, a yield strength of 33 ksi would be equivalent to $33 \times 6.895 = 228$ MPa.

EXERCISES

1. When choosing a bolting material we must consider its affect on the clamping force to be created by those bolts in service. Name two important aspects of the clamping force which will be affected, in part at least, by the choice of bolt material.
2. Name several factors that can lead to a change in clamping force in an assembled joint.
3. Name at least three sources of bolting standards.
4. Three different tensile strengths are given for many bolt materials. Name them.
5. Which of the three is most commonly used to specify bolt size (diameter) for a given application?
6. What's the maximum proof strength (in ksi) of a J429 GR 8 bolt and that of an ASTM A490 bolt?
7. What's the proof strength of a metric Class 10.9 bolt in MPa?
8. What is the ultimate strength and the yield strength (in MPa) of metric Class 4.6 bolt?
9. Does an increase in hardness increase or decrease the tensile strength of a steel bolting material?
10. Define nut dilation. Why can it be a problem?
11. Compute the approximate thermal expansion of a ½–13 × 2, SAE J429 GR 8 bolt when it is subjected to a temperature rise of 150°F (e.g., from 70°F to 220°F).
12. Define stress relaxation of bolts. Under what conditions is it apt to be a problem?
13. What is a cryogenic bolt material?
14. Which definition of strength are we most often concerned about when selecting a joint configuration or material?
15. Name the two committees who manage ASTM bolting standards.

REFERENCES AND BIBLIOGRAPHY

Note: In addition to the references listed below, a great deal of information was taken from the ASTM, SAE, ASME, and other standards cited in the various tables.

1. *ASME Boiler and Pressure Vessel Code*, Section III-Div 1, 1983 ed., Table 1-13.3, ASME, New York, 1983.

2. Hood, A.C., MP35N—a multiphase alloy for high strength fasteners, *Metal Progress*, May 1968.
3. *1984 SAE Handbook*, vol. 1, *Materials*, SAE, Warrendale, PA, pp. 10.140–10.142, 1984.
4. Chung, Y., Threshold preload levels for avoiding stress corrosion cracking in high strength bolts, Report No. 0284-03EV Rev 1, Bechtel Group, Inc., San Francisco, April 1984.
5. Mcllree, A.R., Degradation of high strength austenitic alloys X-750, 718 and A-286 in nuclear power systems, International Symposium on Environmental Degradation of Materials in Nuclear Power Systems—Water Reactors, Electric Power Research Institute, Palo Alto, CA, August 22–24, 1983.
6. Carpenter Gall-Tough Stainless, Brochure of Carpenter Technology Corp., Reading, PA, 1987.
7. Fighting corrosion with new specialty steels, *Product Engineering*, November 1979.
8. *ASME Boiler and Pressure Vessel Code*, Section III, Div. 1, 1983 ed., Table 1-1.3, ASME, New York, 1983.
9. Winter, J.R., Excerpts from bolt document, privately published, 1974.
10. Grade and material markings, Part VIII, *Fastener Technology*, pp. 50–58, December 1986.
11. Hood, A.C., Corrosion in threaded fasteners—causes and cures, *Machine Design*, December 1981.
12. Hood, A.C. and R.L. Sproat, Ultrahigh strength steel fasteners, Special Technical Publication no. 370, ASTM, Philadelphia, 1965.
13. Brochure No. 3275-681-2M, MP35N, SPS Technologies, Jenkintown, PA, 1987.
14. From data presented at a meeting of the Atomic Industrial Forum, Metal Properties Council Joint Task Group on Bolting, Knoxville, TN, 1983.
15. Gnuchev, V.S., Investigation of the strength of bolts, *Problemy Prochnosti*, no. 4, pp. 113–115, April 1977.
16. Yura, J.A., K.H. Frank, and D. Polyzois, High strength bolts for bridges, PMFSEL Report no. 87-3, University of Texas, Austin, TX, May 1987.
17. Zelus, P. and Peter M. Rush, Survey of fastener technology, *Metals Engineering Quarterly*, vol. 10, pp. 1–5, August 1970.
18. Sproat, Robert L., The future of fasteners, *Machine Design*, vol. 38, no. 30, pp. 107–122, December 22, 1966.
19. Standard BS 4882:1973, British Standards Institution, London, 1973.
20. Mountford corrosion resistant fasteners, GKN Fasteners Corrosion Lab Reprint 32, Fred Mountford, Ltd., Warley, UK.
21. Blake, A., *What Every Engineer Should Know about Threaded Fasteners*, Marcel Dekker, New York, 1986.
22. *ASME Boiler and Pressure Vessel Code*, Section III-Div. 1, 1983 ed., Table 1-5.0, ASME, New York, 1983.
23. *ASME Boiler and Pressure Vessel Code*, Section III-Div. 1, 1983 ed., Table 1-7.3, ASME, New York, 1983.
24. *1984 SAE Handbook*, vol. 1, *Materials*, SAE, Warrendale, PA, pp. 10.137–10.139, 1984.
25. *1984 SAE Handbook*, vol. 1, *Materials*, SAE, Warrendale, PA, p. 10.22, 1984.
26. *Handbook of Chemistry and Physics*, Charles D. Hodgman (Ed.), Chemical Rubber Publishing Co., Cleveland, OH, 1943.
27. *ASME Boiler and Pressure Vessel Code*, Section III-Div. 1, 1983 ed., Table 1-6.0, ASME, New York, 1983.
28. *Machinery's Handbook*, 20th ed., Industrial Press, New York, p. 1181, 1975.
29. Bickford, J.H. and M. Looram, Good bolting practices: a reference manual for nuclear power plant personnel: vol. 1, large bolt manual, Document no. NP-5067, Electric Power Research Institute, Palo Alto, CA, p. 44, 1987.
30. Compilation of stress relaxation data for engineering alloys, ASTM, Data Series, Publication DS 60, Publ. Code #05-060000-30, ASTM, Philadelphia, 1982.
31. Private conversation with Paul Bonenberger, General Motors Corp., Detroit, MI, 1981.
32. Private conversation with Robert Finkelston, SPS Laboratories, Jenkintown, PA, 1980.
33. Patel, S. and E. Taylor, New high strength fastener materials resist corrosion, *Metal Progress*, September 1971.
34. Finkelston, R.J. and P.W. Wallace, Advances in high performance mechanical fastening, Society of Automotive Engineers, no. 800451, Warrendale, PA, 1980.
35. Yoshimoto, I., K. Maruyama, K. Hongo, and T. Sasaki, On improvement of fatigue strength of high strength bolts, Bulletin P.M.E., no. 43, Tokyo Institute of Technology, Tokyo, Japan, March 1979.

36. Crispell, C., New data on fastener fatigue, *Machine Design*, Penton Publications, Cleveland, OH, p. 74, April 22, 1982.
37. Thread forms and torque systems boost reliability of bolted joints, *Production Engineering*, p. 37, December 1977.
38. Fritz, R.J., Cyclic stresses for bolts and studs, General Electric Co., Schenectady, NY.
39. Tensioning of fasteners (studs and bolts), Boiler and Machinery Engineering Bulletin no. 4.3, American Insurance Association, New York, 1981.
40. Aaronson, S.F., Analyzing critical joints, *Machine Design*, pp. 95–101, January 21, 1982.
41. Crispell, C., New data on fastener fatigue, *Machine Design*, pp. 71–74, April 22, 1982.
42. From a seminar on threaded fasteners, presented by a FORTUNE 500 manufacturing concern, September 1980.
43. List and description of various aerospace fastening systems used by the Boeing Commercial Airplane Co., Seattle, WA.
44. Erker, A., Design of screw fastenings subject to repeated stresses, Proceedings of International Conference on Fatigue of Metals, September 10–14, 1956, Institution of Mechanical Engineers, London, 1956.
45. Chapman, I., J. Newnham, and P. Wallace, The tightening of bolts to yield and their performance under load. *Journal of Vibration*, *Acoustics*, *Stress and Reliability in Design*, Transactions of the ASME, vol. 108, p. 213, April 1986.
46. Hood, A.C., Corrosion in threaded fasteners—causes and cures, *Machine Design*, pp. 153–156, December 7, 1961.
47. Snow, A.L. and B.F. Langer, Low cycle fatigue of large diameter bolts, Transactions of the ASME, *Journal of Engineering for Industry*, pp. 53–60, February 1967.
48. Kulak, G.L., J.W. Fisher, and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, 2nd ed., Wiley, New York, pp. 9–11, 1987.
49. Kulak, G.L., J.W. Fisher, and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, 2nd ed., Wiley, New York, p. 114, 1987.
50. *ASME Boiler and Pressure Vessel Code*, Section II, Part A, ASME, New York, 1980.
51. *Manual of Steel Construction*, 8th ed., AISC, Chicago, pp. 6–7, 1980.
52. *Handbook of Chemistry and Physics*, 27th ed., Chemical Rubber and Publishing Co., Cleveland, OH, p. 1631, 1943.
53. *ASME Boiler and Pressure Vessel Code*, Section VIII, Part UF, Table UF-27, ASME, New York, 1983.
54. Elevated temperature properties of cast iron, SAE standard J125, SAE, Warrendale, PA, 1969.
55. Juvinall, R.C., *Engineering Considerations of Stress*, *Strain and Strength*, McGraw-Hill, New York, p. 559, 1967.
56. Blake, A., *Practical Stress Analysis in Design*, Marcel Dekker, New York, p. 12, 1982.
57. Design factors for threaded aluminum fasteners, *Machine Design*, p. 109, June 26, 1980.
58. From notes for a bolting seminar prepared by Ray Toosky of McDonnell Douglas, 1991.
59. Wakiyama, K. and A. Tatsumi, Residual force in high strength bolts subjected to heat, *Transactions of the A.I.J.*, No. 313, p. 19, March 1982.
60. Wakiyama, K. and A. Tatsumi, Residual force in high strength bolts subjected to cyclic heat, reprinted from *Technology Reports of the Osaka University*, vol. 29, no. 1515, p. 533, 1979.
61. Raymond, L., Titanium alloys, *American Fastener Journal*, p. 8, May/June 1992.
62. Faller, K. and W.A. Edmonds, The incentives for titanium alloy fasteners in the Navy/Marine environment, *American Fastener Journal*, p. 44, September/October 1992.
63. Barrett, R.T., Design criteria, Part eight of a ten part series . . . fastener basics, *American Fastener Journal*, p. 56, May/June 1992.
64. Conversations with Ray Toosky of McDonnell Douglas, August 1993.
65. *Annual Book of ASTM Standards*, *2006*, *Volume 01.08 Fasteners: Rolling Element Bearings: Standards F2281*, *F2282* ASTM International, West Conshohocken, PA.
66. Conversation with Charles Wilson of the Industrial Fasteners Institute, February 2006.

3 Stress and Strength Considerations

In the last chapter we studied the strengths of the materials from which threaded fasteners are made. Now we will look at the resulting strengths of those fasteners themselves. As we do this we should remember that the bolt's job is to clamp the joint together firmly enough to prevent slip, separation, or leakage, and that the bolt must be strong enough to support the maximum preload it receives at assembly, plus the maximum additional loads it sees in service as a result of forces applied to the joint, differential thermal expansion, etc. When designing, evaluating, specifying, or selecting a bolt for a particular job, therefore, one of our first questions will be: "Is this bolt strong enough to clamp this particular joint?" As we're about to see, the question is much simpler than the answer, because there are many aspects to the concept of strength when we're dealing with a bolt.

3.1 TYPES OF STRENGTH

In general, of course, the strength of any machine part is determined by such things as the size and shape of the part, the material it's made from, the heat treatment of that material, its operating temperature, and its condition (has it been abused, is it corroded, etc.). Engineers also define strength in a variety of ways. Here are the more important definitions we use when we're dealing with bolts.

3.1.1 Tensile Strength

First, we must worry about the capacity of the bolt to generate and sustain a sufficient tensile force, since that force will be one of the main factors which determine the clamping force between joint members. (It's not the only factor, as we'll see in Chapter 6.) In most applications, the room temperature tensile strength of the bolt under static loads will be one of only two strength factors we need to be concerned about.

We'll describe this strength in several ways, however, depending on our needs. We'll use the term ultimate strength, yield strength, or—unique to threaded fasteners—proof strength. Each of these is defined and explained in this chapter. Each term defines the amount of tension we can exert on a bolt before exceeding that definition of its strength. If we apply more than its ultimate strength, the bolt will break.

3.1.2 Thread-Stripping Strength

The amount of tension we can create in a bolt depends not only on the strength of its body but also on the shear strength of its threads. If we're designing a nut or deciding how deep to make a tapped hole, we'll want to be sure that the thread engagement length will be great enough to allow us to develop the full ultimate strength of the bolt. A broken bolt is easier to detect than a stripped thread, so we never want the threads to strip.

Thread-stripping strength is the only other strength we'll have to worry about in most static load applications. In most situations we'll consider it simply by using a standard nut with the bolt we have selected, more carefully, above. Such nuts have been carefully designed to develop the full strength of the bolts (as long as nuts of the proper material are used with a given bolt, see Table 2.3). For tapped holes or special nuts, however, we'll have to compute thread-stripping strength.

Tensile strength will be considered at length in this chapter, thread-stripping strength in Chapter 4. Before going on, however, you should be aware of other strength considerations which will also be treated in later chapters.

3.1.3 Shear Strength

The primary load on most bolts is a tensile load along the axis. In some situations, however, the bolt also sees a load at right angles to the axis, usually called a shear load. This is especially common in structural applications. We'll consider ways to estimate such loads, and their effect on the bolt, in Chapter 12.

3.1.4 Brittle Fracture Strength

In the discussion so far we've been talking about the "normal" strength of a "normal" bolt under fairly static loads, at room temperature. Things change, sometimes drastically, when one or more of these qualifications are absent. For example, if the bolt is made of a very hard material it can theoretically support very high tensile loads. If a tiny crack or flaw exists in that bolt's surface, however, it might fail suddenly and unexpectedly under loads well below its theoretical strength. We took a brief look at brittle fracture in Chapter 2.

3.1.5 Strengths at High and Low Temperatures

Another deviation from the norm we must often face is an "extreme" temperature. Temperatures that are higher or lower than "normal ambient" (which is usually taken to mean 70°F or 20°C) will alter the tensile and stripping strength of the bolt, because of basic changes in the tensile and shear strengths of the material from which the bolt is made. We considered this point in Chapter 2 when we looked at some specific bolt materials and their properties.

3.1.6 Fatigue Strength

Tensile and stripping strengths are a measure of the resistance of the bolt to static or slowly changing loads. Cyclic loads lead to entirely different types of failure, as we'll see in Chapter 15 when we discuss fatigue. A bolt which promises to be "better" (stronger) from a static tensile strength point of view can actually be "weaker" (more likely to fail) under cyclic loads, a fact which has "burned" many an unsuspecting designer.

3.1.7 Stress Corrosion Cracking Strength

Similarly, the resistance of a bolt to stress corrosion cracking (SCC) or hydrogen embrittlement can be inversely proportional to its conventional tensile strength. We'll look at SCC in Chapter 16 when we deal with corrosion failure in general.

So strength is a many-faceted topic which we'll return to again and again in this text. To get started, let's look at what we usually mean when we talk about the strength of a bolt, its room temperature tensile strength.

3.2 BOLT IN TENSION

3.2.1 Elastic Curves for Bolts in Tension

If we place a relatively ductile fastener (such as an ASTM A325 or an SAE Grade 5) in a tensile-testing machine and gradually apply a pure tension load between the head of the bolt and the nut, it will generate a tension versus change-in-length curve such as that shown in Figure 3.1.

The initial, straight-line portion of the elastic curve for a bolt is called the "elastic region." Loading and unloading the bolt repeatedly to some point on this portion of the curve will never result in a permanent deformation of the bolt (although it may result in ultimate fatigue failure, as discussed in Chapter 15).

The upper end of the straight line ends at the "proportional limit," where the line is no longer straight, followed closely by the "elastic limit" (tension loads beyond this point will produce some permanent deformation), followed by the "yield strength point." Loading the fastener to this last point will create a particular amount of permanent deformation—usually chosen as 0.2% or 0.5% of the initial length. A definition of this sort is necessary because the point at which an engineering body can be said to have yielded is not obvious. As we'll see in a minute, some portions of the bolt will have yielded long before the body as a whole has been loaded to its yield strength, and other portions of the bolt are not even close to yield when the bolt has taken a permanent set of 0.5% or the like.

Another point of interest is the "ultimate strength" (often called tensile strength) of the bolt. This is the maximum tension which can be created by a tensile load on the bolt. It is always greater than the yield strength—sometimes as much as twice the yield—but always occurs in the plastic region of the curve, well beyond the point at which the bolt will take a permanent set.

A final point of interest on the elastic curve is the rupture point, where the bolt breaks under the applied load. This and the other points we've discussed are all shown in Figure 3.1.

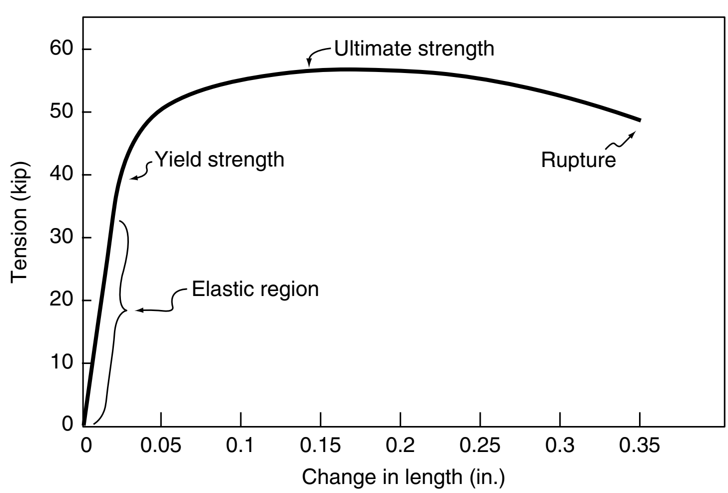


FIGURE 3.1 Points of interest on the elastic curve of a 1–16, A325 bolt with a 5 in. grip length. The proportional limit and elastic limit are located near each other at the upper end of the elastic region.

3.2.2 Elastic Curves under Repeated Loading

If we load the bolt well into the plastic region of its curve and then remove the load, it will behave as suggested in Figure 3.2. Note that it returns to the zero-load point along a line that is parallel to the original elastic line, but offset from the original line by an amount determined by the permanent deformation created by the earlier tension load on the bolt [11].

If we now reload the bolt but stay below the maximum tensile load applied earlier, the bolt will follow this new straight line and will again function elastically. In fact, its behavior will be elastic well past the tensile load, which caused permanent deformation in the first place. The difference between the original yield strength and the new yield strength is a function of the work hardening, which has been done on the bolt by taking it past yield on the first cycle, as noted in Figure 3.2. Loading it past this new yield point will create additional permanent deformation; but unless we take it well past the new yield point we won't damage or break the bolt by yielding it a little—in fact, we'll have made it a little stronger, at least as far as static loads are concerned.

Many bolt materials can be taken past initial, and new, yield points a number of times before they will break. The stronger, more brittle materials, however, can suffer a loss of strength by such treatment, as shown, for example, by Fisher and Struik [1]. Loss of strength in several ASTM A490 bolts, because of repeated cycling past yield (under wind and water loads), has been publicly cited as a contributing factor in the 1979 collapse of the roof of the Kemper Auditorium in Kansas City, for example [2].

3.2.3 Stress Distribution under Tensile Load

Let's place a bolt in a joint and load it in pure tension (this is possible if we use a hydraulic tensioner instead of a wrench to tighten the bolt). If the bolt is perfectly symmetrical, the faces of the head and nut are exactly perpendicular to the axis of the threads, joint surfaces are flat

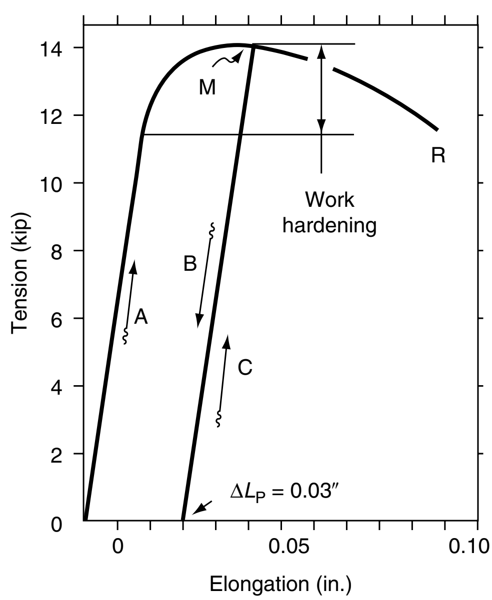


FIGURE 3.2 Elastic curve for a 1–16 × 4 socket-head cap screw loaded (A) to point M, well past the yield strength, and then unloaded (B) to reveal a permanent deformation $L_P = 0.03$ in. If reloaded, it will follow path (C).

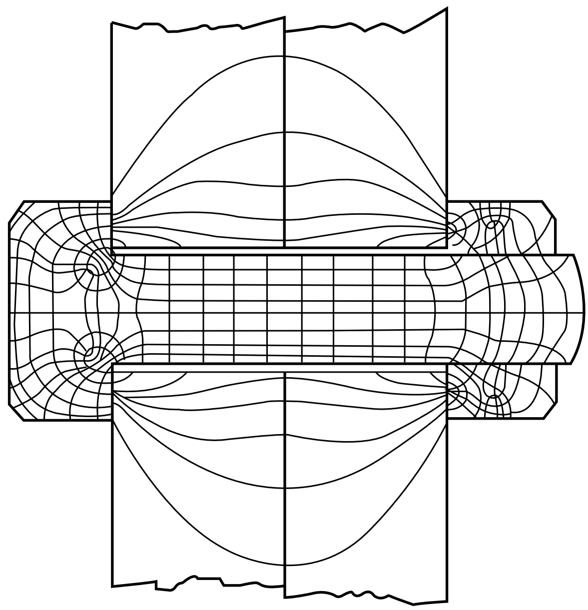


FIGURE 3.3 Lines of principal tension and compression stress in a bolt loaded in pure tension (and lines of principal compression stress in the joint).

and parallel, etc. (most of which we'll never encounter in practice); loading the bolt and joint this way will produce the stress distribution shown in Figure 3.3 [3,4].

Two points are worth noting:

Even though the bolt has been loaded in pure tension, it is well supplied with compressive stress (for example, in the shank), thanks to Poisson's reduction and thanks to the fact that the tension built into the bolt during the tightening process must subsequently be held there by compressive forces in the nut, bolt head, and joint members.

Complex though the picture is, it's a far cry from the truth in most applications, where it is complicated by imperfect geometry as well as by the presence of torsional, bending, and shear stresses, as we'll discuss later. Furthermore, our picture ignores localized, but often significant, stress concentrations in the threads.

3.2.4 Stress Concentrations

Figure 3.3 gives us a simplified view of the directions of stress in a loaded bolt and joint. An analysis of stress magnitudes would reveal three danger points, where stress concentrations create stress levels well beyond the average. These points are the fillet, where the head joins the body; the thread run-out point, where the threads meet the body; and the first thread to engage the nut. As we'll see, these are the points at which the fastener will usually fall.

3.2.5 Magnitude of Tensile Stress

In much of what follows, we'll find it useful to know something about the magnitude of tensile stress within the fastener. For example, many of our calculations will be based on the assumption that tensile stress is zero at the free ends of the bolt and that it rises uniformly through the head to the level found in the body, as suggested by Figure 3.4. There's a similar pattern in the threaded end, but the average stress in the threaded section is higher than the average in the body because the cross-sectional area is less in the threads.

A finite element analysis made by General Dynamics, Fort Worth [4], however, suggests a far more complex pattern. Scientists at General Dynamics say that the magnitude of tensile

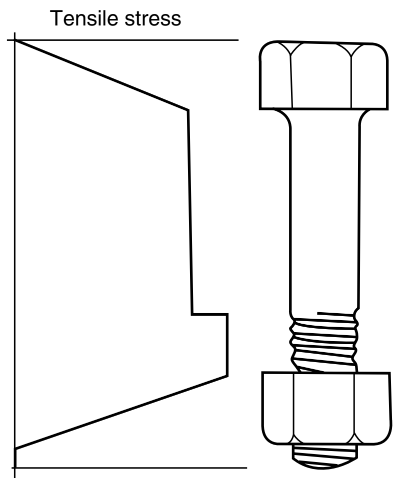


FIGURE 3.4 The magnitude of tensile stress in a bolt—the simplistic view often assumed in bolt calculations.

stress along the axis of the bolt does approximate that shown in Figure 3.4, but that the magnitude of tension along other lines parallel to the axis of the bolt looks more like that shown in Figure 3.5, with the peak stresses (at the fillet and nut–bolt engagement points) being two to four times the average stress in the body. To complicate things still further, the above is true only for long thin bolts, by which is meant bolts that have a grip length-to-diameter ratio greater than 4:1. In short, stubby bolts, the picture shows a general variation in tension stress from side to side, as well as end to end, as shown in Figure 3.6. There is now no such thing as ‘‘uniform stress level,’’ even in the body.

If you stop and think for a minute, you’ll realize that 4:1 isn’t very short and stubby. A ¼–20 bolt having a grip length of 1 in. would be considered short and stubby by this definition, which means that the majority of fasteners we’re going to deal with are probably short and stubby.

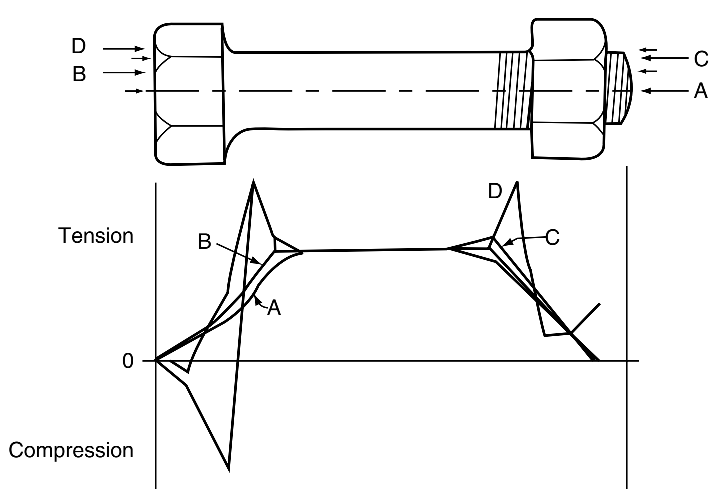


FIGURE 3.5 More accurate view of the tensile stress along four lines parallel to the axis of the bolt (see also Figure 3.6).

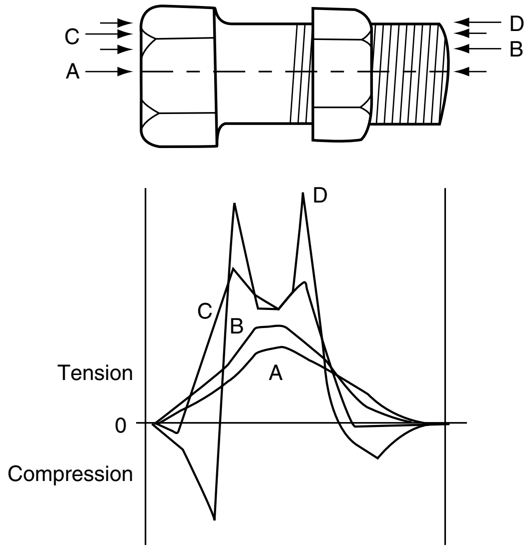


FIGURE 3.6 Magnitude of tensile stress along four separate paths in a bolt having a length-to-diameter ratio less than 4:1.

As a result, in the majority of applications, we're dealing with fasteners in which there is no uniformity of tensile stress. And this has all sorts of implications when we come to compute such things as stress level, preload, spring constants, and elongation, as we'll see in subsequent discussions.

Stresses in the threaded portion of the bolt are concerned. They do not, for example, take into account the stepped difference between average body stress and average thread stress shown in Figure 3.4. They also ignore stress concentrations at thread roots and thread run-out points. In fact, they assume a threadless fastener with uniform take-out of load between nut and bolt. This assumption probably doesn't affect their estimates of stress within the body of the fastener as much as it affects their estimates of stress levels at the surface of the fastener, but it's something to keep in mind.

3.2.6 Stress in the Nut

A slightly more accurate plot of the peak stresses in nut or bolt threads is shown in Figure 3.7 [3,6]. The fall-off in stress is not linear, as in the previous figures, but curved. Note that adding more threads (a longer nut) doesn't reduce the peak stress by much. The first three threads carry most of the load in any case.

Obviously, this stress picture is not an attractive one. Since most of the load is on the first thread or so, most of the nut isn't doing its share of the work. This situation can be improved in a number of ways—tapering the threads or altering the pitch on either nut or bolt to force more uniformity in load distribution, for example. Perhaps the most popular way is to use a nut that is partially in tension, such as one of those shown in Figure 3.8 [9].

Figure 3.9 shows the stress levels in the nut illustrated in Figure 3.8B.

One study [9] of titanium tension nuts, similar to that shown in Figure 3.8C, with most threads in tension, resulted in the computed stress distribution shown in Figure 3.10.

Another analysis, confirmed by experiment [10], shows that if the pitch of the bolt threads is 0.13% longer than the pitch of the nut threads—a difference of only 0.000065 in. per pitch

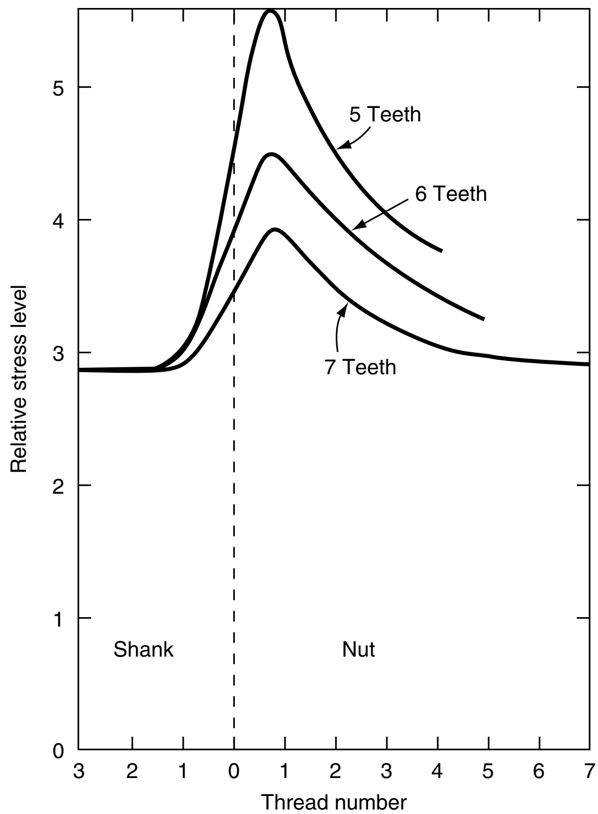


FIGURE 3.7 Peak stresses in three different nuts, having five, six, and seven teeth, respectively.

in a 1/4–20 thread, for example—then the outermost threads of the nut will be stressed more heavily than the innermost, as shown in Figure 3.10. This sort of variation must be common in practice, and it helps explain the difficulties of predicting how a given fastener or joint will behave.

Stress distribution similar to that shown in Figure 3.10 can also occur when the thread engagement is very long. Tapped holes often have more threads than conventional nuts,

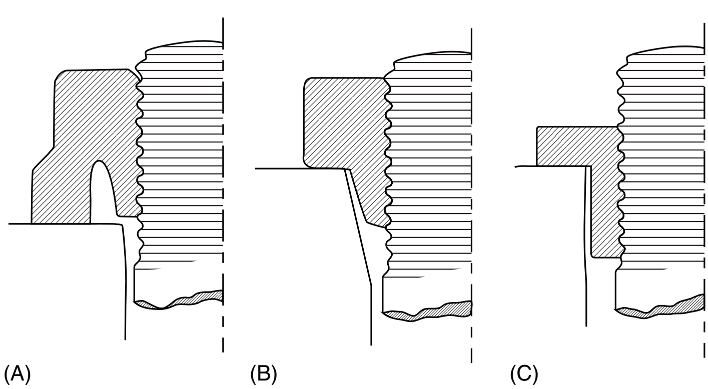


FIGURE 3.8 Nuts which are partially loaded in tension, such as those shown here, see a more uniform tooth stress distribution than do conventional nuts.

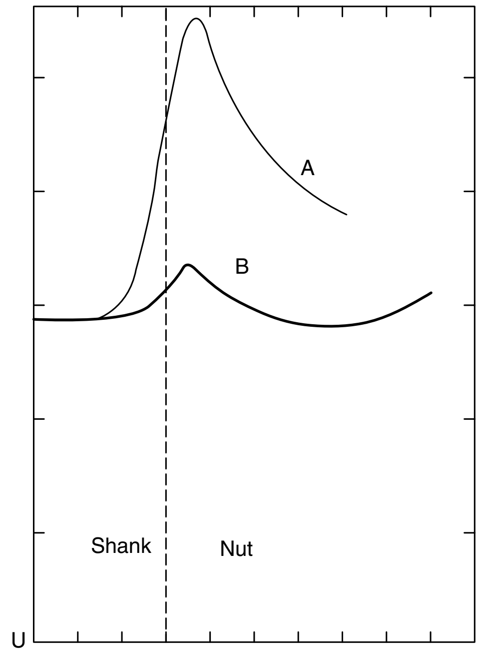


FIGURE 3.9 Relative stress level in nut and bolt threads for a tension nut (curve B) and for a conventional nut (curve A).

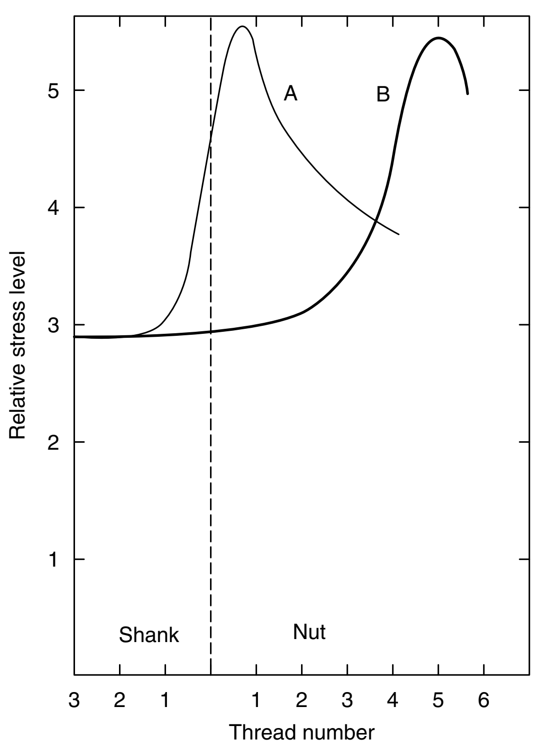


FIGURE 3.10 Stress distribution in threads with nominal pitch (A) and when there is a +0.13% error in the pitch of the bolt threads (B).

which, typically, have five threads. I've encountered tapped holes with 18 or 20 threads. Large cylindrical nuts, used for example on large pressure vessel flanges, can have a similar number of threads. Even if both male and female threads are cut or rolled to tight, correct tolerances, the threads will usually have slightly different pitch distances, and this can place the maximum thread contact stress at the far end of the thread engagement, or in the middle, or "somewhere else."

3.3 STRENGTH OF A BOLT

3.3.1 Proof Strength

It's useful at this point to look at the procedure used to determine and define such things as the proof strength and yield strength of bolts. We'll need this information when we study the various means by which we control the tightening of bolts. Recognize, however, that this is just an introduction to the important subject of strength (or bolt failure) and that there are many practical factors that can modify the apparent strength predicted by the conventional procedures.

In this chapter we want to estimate the static strength of the body and threads of the fastener, starting with the tensile strength of the threaded portion of the body (the weakest section). Because the stress picture is complicated, computing strength would be difficult, so the engineering societies have devised a way to determine strength experimentally and to base design and manufacturing specifications on the resulting data. A large number of fasteners, made with well-defined materials and standard thread configurations, were subjected to tensile loads to determine

- Highest tensile force they could withstand without taking any permanent deformation (called the proof load or proof strength). See Table 3.1 for a few examples.
- Tensile force that produces 0.2% or 0.5% permanent deformation (used to define yield strength).
- Highest tensile force they could support prior to rupture (used to define ultimate tensile strength).

The resulting data were then published. You don't have to worry about stress magnitudes or variations; the tables tell you how many pounds of force a fastener of a given shape and material can safely stand. Fastener manufacturers are required to repeat these tests periodically to make sure that their products meet the original standards.

TABLE 3.1
Proof Loads of a Few Typical Bolts in Kips

	Bolt Size			
Bolt	**¼–20**	**1–8**	**2–8**	**4–8**
ASTM A307-GR A	2.70	20	91.2	364
SAE J429 GR 8	3.82	72.7	NA	NA
ASTM A193 B7	NA	57.3	262	946
H11	NA	117	537	2146

Note: Load in kips $\times$ 4.448 $\times$ 10^3 $=$ load in Newtons.

There was also a need to compute design limits for nonstandard fasteners. Here the path becomes a little more murky but will still give you a useful answer unless your need for accuracy and safety is critical.

The people who tested the fasteners found, again by experiment, that if they divided the experimentally determined load at yield, in pounds, by a cross-sectional area based on the mean of the pitch and minor diameter of the threads, the result was a theoretical tensile stress at yield, which agreed closely with the stress at which a cylindrical test coupon of the same material would yield, and with similar results for ultimate strength, proof load, etc. The following data can now be published:

- Stress area (A_s) of a standard thread, based on the mean of pitch and root diameters
- Proof strength of a given thread, in pounds per square inch (psi), computed by dividing the experimentally determined proof strength, in pounds, by the stress area
- Ultimate strength or yield strength, of a given thread, in pounds per square inch, by dividing the experimentally determined ultimate or yield load in pounds by the same stress area

3.3.2 Tensile Stress Area

Since the mean of pitch and root diameters works for all thread sizes, it's also possible to write an expression for the stress area (A_s) of a standard 60° thread, as follows [7]:

$$\begin{aligned} A_s &= \pi/4[D - (0.9745/n)]^2 \\ &= 0.785[D - (0.9743/n)]^2 \end{aligned} \quad (3.1A)$$

where D is the nominal diameter and n is the number of threads per inch. Appendix E lists the stress areas of all standard, unified, and metric threads. As an example, let's compute the stress area of a ¼–20 UNC, Class 2A thread.

$D = 0.25$ in.
$n = 20$
$A_s = 0.785\ [0.25 - (0.9743/20)]^2 = 0.0318$ in.^{2}

This expression for area is used to compute the tensile or shear strength of a fastener. For example, to compute the ultimate load, in pounds, that your own fasteners should tolerate, it's only necessary to look up (or compute) the stress area of the threads and then multiply this by the published values for the ultimate strength (UTS) in pounds per square inch of that particular material. And you get the correct answer, for most purposes. For example:

$$F_{\text{Ten}} = \text{UTS} \times A_s \quad (3.1B)$$

A technique similar to that used for bolts is used for specifying the static strength of nuts. The stress area used to compute average nut stress levels is, in fact, the cross-sectional area of the "male" threads, A_s, just discussed (showing again how "artificial" these calculations are). The "strength" of a nut, of course, is simply the stripping strength of its threads. We get to that in Chapter 4.

3.3.3 Other Stress Area Equations

Although Equation 3.1, or its metric equivalent in Equation 3.4, is the expression most often used for the stress area of a thread, there are times when a more conservative (smaller) area is used for an added factor of safety. For example, before leaving this topic, we should mention that a slightly modified equation for stress area is recommended for fastener materials having an ultimate strength (as determined by coupon tests) in excess of 100,000 psi. This time the stress area is based on the mean of the minimum pitch diameter (E_{smin}) permitted by thread tolerances and the nominal root diameter. This gives a slightly smaller stress area (partially canceling the load benefits you'd derive from an assumption of higher ultimate strength). Anyway, the equation is [25]

$$A_s = \pi[(E_{smin}/2) - (0.16238/n)]^2 \tag{3.2}$$

To see how this expression for A_s differs from that given in Equation 3.1, let's use it to repeat our calculation of the stress area of another ¼–20 UNC Class 2A thread; this one presumably of a stronger material.

$$\begin{aligned} E_{smin} &= 0.2127 \text{ in.} \\ n &= 20 \\ A_s &= 3.14159\,[(0.2127/2) - (0.16238/20)]^2 \\ A_s &= 0.0301 \text{ in.}^2 \end{aligned}$$

So this A_s is 5% smaller than that we computed earlier.

There's at least one other tensile stress area in common use. The U.S. military uses a stress area based on maximum pitch diameter for UN and UNJ threads when dealing with fastener material strengths between 160 and 260 ksi—and when the threads are rolled after heat treatment. The resulting stress areas are tabulated in military specification NAS 1348. The areas are larger than the A_s values computed by Equation 3.1: about 17% larger for the smallest screws tabulated (0.600–80 threads) to about 4% larger for the largest (1.500–12). Again, these stress areas must reflect test results which show, in this case, that rolling threads after heat treat leads to a much stronger bolt than does rolling before heat treat or cutting.

There is another still more conservative stress area that is used, however. It is based on the root diameter of the threads rather than on the mean of pitch and root diameter, as for the true stress area. It differs from the areas we've talked about so far in that this one is not based on experimental data. It is designed to introduce a factor of safety in thread strength calculations. The designer purposely assumes a stress area smaller than the "real" (Equation 3.1) tensile stress area to be sure the bolts aren't overstressed in service.

The use of root diameter area rather than tensile stress area is mandated by the ASME Boiler and Pressure Vessel Code, for example. The expression for this area is

$$A_r = 0.7854(D - 1.3/n)^2 \tag{3.3}$$

where D and n have the same meaning as in Equation 3.1.

You'll find tensile stress and thread root areas for UN and UNC threads in Table 3.2 and a complete list of tensile stress areas for all inch series and metric threads in Appendix E. Figure 3.12 can help you visualize the differences between the various stress areas we've been discussing.

TABLE 3.2
Thread and Bolt Stress Areas

	Tensile Areas		Stripping Areas	
Thread	A_s (Stress)	A_r (Root)	Class 2A Thread	Class 3A Thread
1/4–20	0.0318	0.0269	0.092	0.096
5/16–18	0.0524	0.0454	0.147	0.157
3/8–16	0.0775	0.0678	0.216	0.232
7/16–14	0.1063	0.0933	0.296	0.321
1/2–13	0.142	0.126	0.390	0.427
9/16–12	0.182	0.162	0.502	0.548
5/8–11	0.226	0.202	0.624	0.681
3/4–10	0.334	0.302	0.908	1.01
7/8–9	0.462	0.419	1.25	1.38
1–8	0.606	0.551	1.66	1.82
1 1/8–8	0.790	0.728	2.13	2.329
1 1/4–8	0.969	0.890	2.65	2.913
1 3/8–8	1.23	1.16	3.22	3.55
1 1/2–8	1.49	1.41	3.86	4.26
1 5/8–8	1.78	1.68	4.55	5.04
1 3/4–8	2.08	1.98	5.30	5.03
1 7/8–8	2.41	2.30	6.09	6.81
2–8	2.50	2.30	6.96	7.72
2 1/4–8	3.56	3.42	8.84	9.83
2 1/2–8	4.44	4.29	10.95	12.18
2 3/4–8	5.43	5.26	13.28	14.80
3–8	6.51	6.32	15.84	17.67
3 1/4–8	7.69	7.49	18.62	20.8
3 1/2–8	8.96	8.75	21.63	24.15
3 3/4–8	10.34	10.11	24.79	27.79
4–8	12.18	11.57	28.28	31.64

Source: *Unified Inch Screw Threads*, ANSI Standard B 1.1-1974, ASME, New York, 1974; Blake, A., in *What Every Engineer Should Know about Threaded Fasteners*, Marcel Dekker, New York, 36–37, 1986.

Note: All areas are in square inches.

All of the stress area equations given so far are for American Standard Unified thread forms with an included angle of 60° as shown in Figure 3.11. Different equations must be used for 55° Whitworth, Acme, Buttress, or other thread forms.

3.3.4 Stress Areas—Metric Threads

The expressions for the stress areas of metric threads differ from those used for inch series threads because in metric threads, the pitch of the threads, rather than the number of threads per inch, is used to define the thread. The following metric equation corresponds to Equation 3.1, defining, as it does, the common tensile stress area [24]:

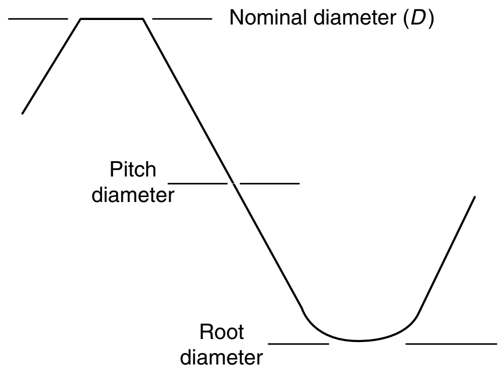


FIGURE 3.11 Relationship between root diameter, pitch diameter, and outside or nominal diameter in a standard 60° thread. The centerline of the thread is out of the picture, toward the bottom.

$$A_s = 0.7854(D - 0.938p)^2 \tag{3.4}$$

where
A_s = tensile stress area (mm)
p = pitch of the threads (mm)
D = nominal diameter of the fastener (mm)

The more conservative expression for higher-strength steels (over 686 MPa—99.5 ksi—this time) is [17]

$$A_s = 0.7854(E_{smin} - 0.268867/p)^2 \tag{3.5}$$

And the expression for thread root area is [24]

$$A_r = 0.7854(D - 1.22687p)^2 \tag{3.6}$$

As before, E_{smin} = minimum pitch diameter (mm); D and p have the same meanings as in Equation 3.4.

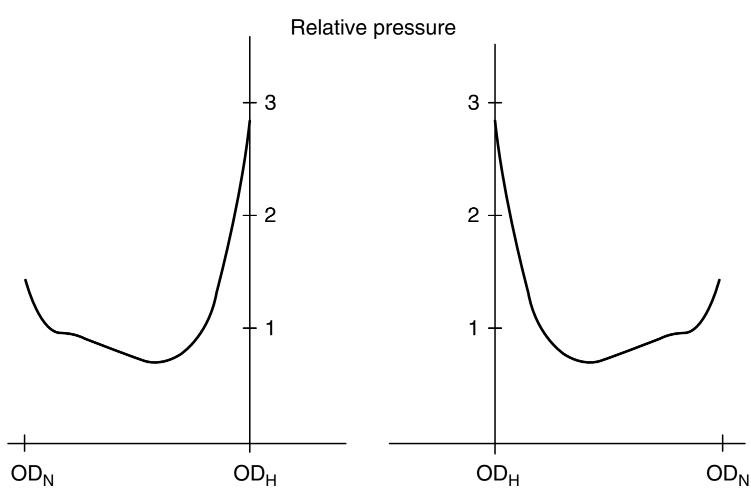


FIGURE 3.12 Relative contact pressure between head of bolt, or nut, and the joint surface. OD_H is the outer diameter of the hole. OD_N is the outer diameter of the contact surface of the nut (or bolt head).

3.3.5 Strength of the Bolt under Static Loads

The bolt or nut will fail under static loads if those loads exceed the strength of the fastener. To understand and avoid static failure, therefore, we must learn how to calculate the static strength of the parts. We'll concentrate on tensile loads; we will compute shear strength of the joint in Chapter 12. In estimating the static strength of a fastener under tensile loads, we must consider four possibilities:

- Body of the bolt will break (usually at one of the three stress concentration points).
- Bolt threads will strip (fail in shear).
- Nut threads will strip.
- Both threads will strip simultaneously.

To determine the static strength, we must explore each of these four possible failure modes. The weakest link, of course, will decide the strength of the chain. We'll consider bolt strength now, and thread strength in the next chapter.

The static strength of the body is usually computed for that section which is threaded, i.e., has a reduced cross-sectional area. The tensile force required to yield or break the bolt (you'll have to decide which of these definitions of failure you prefer) is, simply,

$$F = \sigma A_s \tag{3.7}$$

where
F = force which will fail the bolt (lb, N)
σ = ultimate tensile or yield strength of the bolt material (psi, MPa)
A_s = effective stress area of the threads (in.^{2}, mm^2) (see Equation 3. l or 3.4)

This equation may be used to compute any kind of tensile strength—ultimate, proof, yield, etc. As an example, let's compute the room temperature yield strength of an Inconel 600 bolt with a ¼–20 UNC Class 2A thread. The yield strength of this (and other) bolt material is given in Chapter 2.

σ = 37 ksi
A_s = 0.0318 in.^{2} (see the example with Equation 3.1)
$F = 37 \times 0.0318 = 1177$ lbs

What would the strength of this same bolt be at 1200°F (649°C)? With reference to the Table 2.1 in Chapter 2,

σ = 17 ksi
A_s = 0.0318 in.^{2}
$F = 17 \times 0.0318 = 541$ lbs

Remember that part of the strength of the bolt might be absorbed by torsion stress if we use a wrench to tighten it. This could reduce the apparent tensile yield or ultimate strength (σ) by 10%–20%, but only during tightening. This is why many people say, "If it doesn't break when you tighten it, it won't ever break." If loads are static only and are less than the preload developed during tightening, and corrosion or high temperature doesn't enter the picture, the statement is probably true. The strength of a bolt, of course, determines how much clamping force it can exert on a joint. Table 3.1 gives a few examples of the forces bolts can sustain, to give you a general feeling for the capabilities of individual bolts. The forces tabulated are proof loads or equivalents—about 90% of yield. The forces are given in thousands of pounds (kips). To convert to metric Newtons, multiply the tabulated figures by 4.448×10^3.

The examples in Table 3.1 show the impressive strength of bolts. They suggest that a single ¼–20 A307 bolt, properly mounted and supported, could support the weight of a car. Or that one 4–8 H-11 bolt could alone support the weight of nearly 1000 cars! (The strength of these and other materials is discussed at length in Chapter 2.)

3.4 STRENGTH OF THE JOINT

Before looking at combinations of loads on the bolt, let's take a quick look at the joint.

3.4.1 Contact Stress between Fastener and Joint

The contact pressure between the head of the bolt and the joint is not uniform. Neither is the contact pressure between the nut and the joint. For either case, the contact pressure might look something like that shown in Figure 3.13, as suggested by an experimentally confirmed, finite-element analysis made in Japan [10].

The contact pressure pattern shown in Figure 3.13, although nonuniform, assumes that the contact surfaces of the bolt head and nut will be perfectly parallel to the surfaces of the joint, at least when loading begins. Since this will rarely be the case—these surfaces are usually not exactly perpendicular to the thread or hole axes, for example—the actual pressure distribution will probably be even more irregular than suggested in the drawing.

The contact pressure between nut or bolt head and the joint can have an important influence on the way in which a loaded fastener retains the potential energy stored in it during assembly (see Chapter 1). Excessive pressure will allow the head or nut to embed itself gradually in the joint surfaces, allowing the fastener to relax, to shed some of its stored energy. These contact

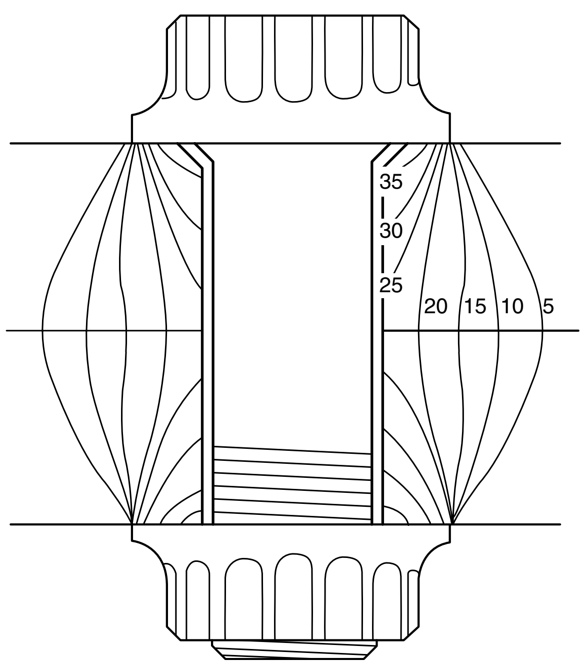


FIGURE 3.13 Lines of equal compressive stress in joint members when the bolt is loaded to 100 kips. Values given are in ksi.

TABLE 3.3
Contact Stress between Bolt Head and Joint When Bolts Are Tightened to 75% of Proof Load

Fastener Grade	Size	Head	Contact Stress Ksi	Contact Stress N/mm
10–9	M10	Flanged	36.3	250
	M16		39.9	275
10–9	M10	Plain hex	52.2	360
	M16		94.3	650
10–9	M10	Hex with washer	27.6	190
	M16		33.4	230
12–9	M10	Plain hex	63.8	440
	M22		131	900
12–9	M10	Hex with washer	29	200
	M22		50.8	350

stresses can be impressive, as suggested by the data in Table 3.3, which shows the stresses generated when the tabulated fasteners are tightened to 75% of their tensile proof load [20]. (For those primarily interested in inch series fasteners, M10, M16, and M22 metric fasteners are approximately ⅜, ⅝, and ⅞ in. in diameter, respectively.) As can be seen from the data, a flanged head or washer can significantly decrease these contact stresses. The thickness of the washer can also make a difference. Studies made at the Newport News Naval Shipyard confirm the contact stress pattern shown in Figure 3.14 and show how nonuniform head or nut-to-joint stress distribution creates high stress gradients in the joint members.

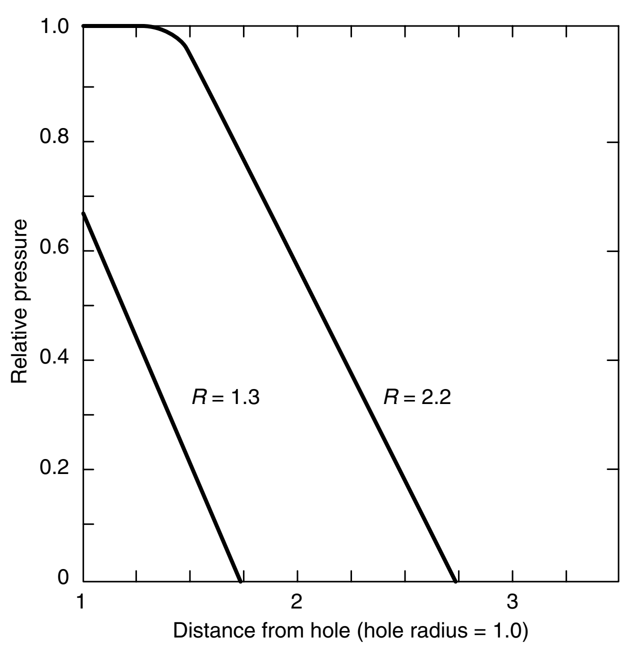


FIGURE 3.14 Relative interface pressure between joint members as a function of the distance from the edge of the hole. *R* is the ratio between the contact diameter of the head of the bolt (or nut) and the diameter of the hole.

We'll look at joint stresses in a minute. The studies further suggest that standard thickness of ANSI washer lacks sufficient stiffness to distribute the bolt loads with acceptable uniformity. Thicker washers are more effective; they reduce stresses in washer and joint members and, as a result, reduce joint deformation. The numerical results of these studies have not yet been published [21].

What's an acceptable stress? This is considered to be a stress slightly higher than the compressive yield strength of the joint material [20,21].

3.4.2 Stresses within and between the Joint Members

As a result of all this, the joint is loaded in a nonuniform manner by the bolt, and these nonuniform stresses are passed down into the joint members. Figure 3.13 shows the resulting, barrel-shaped, lines of equal compressive stress within the joint [5]. Notice that the relative magnitude of stress varies by 8 to 1 between the most highly stressed region and the outer rim of the barrel-shaped stress pattern.

The fact that the joint members are stressed nonuniformly means that the clamping pressure exerted by the bolt through the joint members on the interface between upper and lower joint members is also nonuniform, a fact that can cause problems in gasketed joints. A number of finite-element analyses and experimental studies have been made of the contact interface pressure. The results of one such study [8] are shown in Figure 3.14. Note that the contact pressures can be zero only two or three bolt diameters away from the bolt hole. The solution for this problem of course, is to place bolts close together in a multibolt joint so that no portion of the interface is entirely free of contact pressure. It is never possible, however, to produce exactly uniform pressure at the joint interface. Too many bolt holes, furthermore, will weaken the flange members and create interference and wrenching problems. As a rule of thumb, therefore, bolt holes are usually placed approximately 1½ diameters apart.

3.4.3 Static Failure of the Joint

Static failure of joint members is even less common than static failure of the fasteners, except perhaps in structural joints loaded in shear. Failures there are frequent enough to warrant a brief look at this point. We'll examine shear joints in greater detail in Chapter 12.

If the designer knows which cross sections might fail, he can usually avoid failure. The failures shown here are all of what used to be called bearing-type joints. Friction joints fail statically, too, but they do so by slipping into bearing and then failing by one of the ways shown below.

Joints in bearing can fail in a number of ways as suggested by Figure 3.15. The actual failure mode will depend on the relative strength of the bolts versus the strength of the cross section of the joint members at various load points. It will also be affected by the distance between the bolts and the edge of the plates, by the distance between bolts within the group, etc.

Typical failure modes include [14]:

1. Tear-out or marginal failure, where the bolts are located too close to the edge of the plate (Figure 3.15A)
2. Failure of the "net section" of the plate because the bolts are spaced too closely, or because the plate is too thin or too soft (Figure 3.15B)
3. A zigzag failure when there is too short a distance between bolt holes (Figure 3.15C)

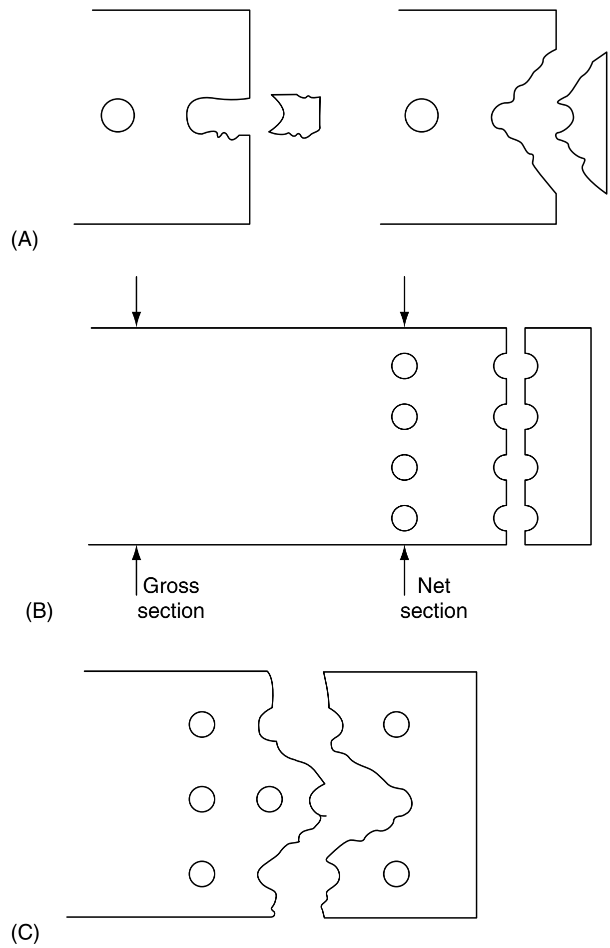


FIGURE 3.15 Some static failure modes of axial shear joints. (A) Tear-out or marginal failure; (B) failure through the "net section"; and (C) zigzag failure.

3.5 OTHER TYPES OF LOAD ON A BOLT

We have looked, at length, at the stress distribution and strength of a fastener under pure tensile load, because this is by far the most important type of loading. A bolt is always put into severe tension when it is properly tightened. Subsequent external loads won't modify this basic tension load very much if the joint is designed properly, as we'll see in Chapter 12, but the tension we create in the bolts and the resulting clamping force on the joint members usually determines the performance and life of that joint. There are times, however, when the bolts are exposed to other types of load, in addition to tensile loads. These additional burdens can reduce their capacity to support tensile stress and may therefore make it more difficult for the bolts to do their job as clamps.

Shear loads are the most readily understood and predictable of these other types of load. Shear loads of the sort shown in Figure 3.16, for example, are often encountered in structural steel joints. Note that there can be one or several shear planes through the bolt, depending on the nature of the joint and the number of joint members being clamped together. Note, too,

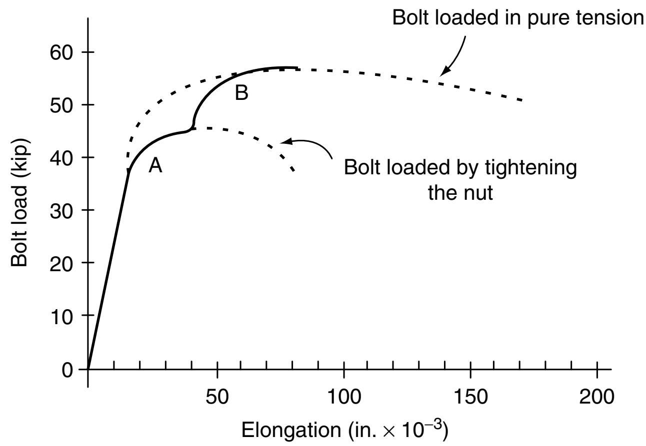


FIGURE 3.16 A fastener tightened with a torque tool, and therefore exposed to simultaneous torsion and tension, will yield at a slightly lower level of tensile stress (A) than a fastener subjected to pure tension. This same fastener, however, will support a higher tensile stress in service before yielding any further (B), because the torsion stress component will, in general, disappear rather rapidly after initial tightening in most situations. Data are for a g A325 bolt with a 4⅛ in. grip length [I].

that these shear planes could pass either through the body of the bolt or through the threads, or through both (Figure 3.16).

To compute the shear strength of a bolt we multiply the shear strength by the strength of the material (S_u) by the total cross-sectional area of the shear planes, taking all of them into account. For shear planes through threads, we use the equivalent thread stress area (A_s) given earlier in our analysis of tensile loads. For example, if there are two shear planes through the body of the bolt and one through the threads, then,

$$F = S_u \times (2A_B + A_s) \tag{3.8}$$

where A_B is the cross-sectional area of the body (in.^{2} or mm^2). We'll consider shear loads at greater length in Chapter 12.

When we tighten and stretch a bolt we inevitably twist it a little. This creates torsional stress in the bolt, which combines with the tensile stress and may take the bolt material past its yield point or even, in extreme cases, past its ultimate strength. Unless the bolt has broken, however, the torsional stress tends to decay and disappear a few minutes after tightening as the heavily loaded bolt relaxes, thanks to embedment of contact surfaces, elastic interaction between bolts, gasket creep or other factors discussed at length in Chapter 6.

The top and bottom surfaces of a joint are often not parallel to each other or are misaligned. In this situation a bolt will be bent slightly as it is tightened, increasing the tensile stress along its convex side. If we wished to analyze the effects of this distortion we'd have to have detailed information about the geometry of the joint surfaces so that we could define the effective "radius of curvature" of the bent bolt and resulting bending moment. As far as I know this is only done when dealing with the raised face flanges used in gasketed joints. Bending will, therefore, be discussed only in Volume 2 of this text.

The mathematics of torsion and bending are described in the third edition of this text if anyone is interested; but this information is of little or no practical use and will, therefore, not be allowed to interrupt us here. In that edition I also went to considerable length to define an equation summarizing the combined effects of all loads and forces on a tightened bolt. The result was a long work–energy equation, which has appealed to a few academically minded readers, but which is of no practical value. One reader, A.R. Srinivas of the Space Application

Center in India, outdid the rest of us by showing how the work–energy equation reduced to the definitely practical, long-form, torque turn equation, which we'll consider at length in Chapter 7. He eliminated all of the terms in the equation which describe forces and factors which have an insignificant effect on the behavior of the bolt and joint.

3.5.1 Strength under Combined Loads

We are interested primarily in the tensile strength of bolts, less often in the shear strength. The tensile strength determines the amount of preload we can safely put into a bolt on tightening it and the amount of tensile working load it can see thereafter. It's important to recognize, therefore, that the tensile strength of a given fastener is reduced if the fastener also sees torsion or shear loads. In an extreme case, for example, let's assume that nut and bolt threads have galled near the end of the tightening operation. This will result in an abnormally high level of torsion in the bolt. If we now apply tension, we'll find that the bolt will break at a tension level well below normal—perhaps even below proof load. The torsion stress has robbed part of the total strength of the bolt. Only if the bolt is loaded, statically, in pure tension can we count on it to support a proof load without deformation, or to yield at a particular level of tension.

Note that torsion stress will rob part of the total strength of the bolt only while the torsion stress is present. As we torque a bolt, for example, it is subjected to some torsion. Such a bolt will yield at a certain level of tensile stress. After we remove the torque wrench from the nut, however, the torsion stress will tend to disappear (thanks to embedment relaxation, as we will learn in Chapter 6). If we now apply an external tensile load to the fastener, we will discover that it will support a higher level of tension than that which caused it to yield in the first place, as suggested in Figure 3.16. A number of different fastening tools and strategies take advantage of this fact, as we will see later [1].

EXERCISES AND PROBLEMS

1. Which is greater, the tensile strength of a bolt or its stripping strength? Why?
2. Where in the bolt would we find the maximum tensile stress? Where in a conventional nut?
3. Define proof strength.
4. Compute the conventional tensile stress area of ⅜–16 and ½–13 threads. Compute the ratio in stress area between the two.
5. Using the stress areas computed in problem #4 compute the static, room temperature proof strengths of ⅜–16 and ½–13 bolts made from SAE J429 Grade 5 and Grade 8 materials. See Table 3.1 for proof strength data.
6. Compute the ratio in proof strength between the two threads.
7. Compute the shear strengths of those two threads if the fasteners are made from Grade 8 material.
8. Compute the ratio between shear strength and proof strength for each of these two threads.
9. A threaded fastener will often be subjected to loads other than tensile loads. List those other kinds of load. Which, if any, of them will we sometimes take into account when analyzing a bolted joint?

REFERENCES AND BIBLIOGRAPHY

1. Fisher, J.W. and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, Wiley, New York, 1974.
2. "Rocking that Fatigued Bolts Felled Arena Roof, ENR," *McGraw-Hill's Construction Weekly*, August 16, 1979.

3. Wiegand, H. and K.H. Illgner, *Berechnung und Gestaltung von Schraubenuerbindungen.* Springer-Verlag, Berlin, 1962, p. 48.
4. Fastener preload indicator, Contract no. F33615–76-C-5151, Report prepared for Air Force Materials Laboratory, Air Force Systems Command, Wright Patterson Air Force Base, Ohio, by General Dynamics-Ft. Worth, June 15, 1978.
5. Meyer, G. and D. Strelow, Simple diagrams aid in analyzing forces in bolted joints, *Assembly Engineering*, January 1972.
6. Seika, M., S. Sasaki, and K. Hosono, Measurement of stress concentrations in threaded connections. *Bulletin of the JSME*, vol. 17, no. Ill, September 1974.
7. Unified Inch Screw Threads, ANSI Standard B 1.1-1974, ASME, New York, 1974.
8. Gould, H.H. and B.B. Mikic, Areas of contact and pressure distribution in bolted joints, *Transactions of the ASME*, vol. 94, no. 3, August 1972.
9. Motosh, N., Load distribution on threads of titanium tension nuts and steel bolts, *Journal of Engineering for Industry—Transactions of the ASME*, vol. 97, no. 1, pp. 162–166, 1975.
10. Maruyama, K., Stress analysis of a bolt–nut joint by the finite element method and the copper-electro-plating method, *Bulletin of the JSME*, vol. 19, no. 130, April 1976.
11. Finkelston, R.J., Preloading for optimum bolt efficiency, *Assembly Engineering*, August 1974.
12. Roark, R. and W.C. Young, *Formulas for Stress and Strain*, 5th ed., McGraw-Hill, New York, 1975, p. 287.
13. Eshbach, O.W., *Handbook of Engineering Fundamentals*, 2nd ed., Wiley, New York, 1953, pp. 5–28.
14. Fisher, J.W. and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, Wiley, New York, 1974, p. 84.
15. Blake, A., *What Every Engineer Should Know about Threaded Fasteners*, Marcel Dekker, New York, 1986.
16. Crispell, C., A better way to design small fasteners, *Machine Design*, Cleveland, Ohio, January 22, 1987.
17. BS 3643: Part 2: 1966, British Standards Institution, London, 1966.
18. Oberg, E., F.D. Jones, and H. Horton, *Machinery's Handbook*, 20th ed., Industrial Press, New York, 1975.
19. Unified Inch Screw Threads, ASME B 1.1-1989 (Revision of ANSI Bl.l-1982), American Society of Mechanical Engineers, New York, 1989.
20. Eccles, W., Bolted joint design, *Engineering Designer* (UK), November 1984, p. 11.
21. Barron, J., in Report to the Bolting Technology Council, at a meeting in Cleveland, Ohio, April 19, 1993.
22. Junker, G.H., Principle of Calculation of High Duty Bolted Joints: Interpretation of Directive VDI *2230*, published by SPS, Jenkintown, PA (no date given).
23. VDI 2230: Systematic Calculation of High Duty Bolted Joints, VDI Society for Design and Development, Committee on Bolted Joints, Issued by the Verein Deutscher Ingenieure (VDI), VDI Richtlinien. (Dusseldorf, October 1977, Translated from German by Language Services, Knoxville, TN, published by Oak Ridge National Laboratories as ORNL-tr-5055, Table 14.)
24. Screw Thread Standards for Federal Services, Section 2, Unified Inch Screw Threads—UN and UNR Forms, FED-STD-H28/2B, 20 August 1991, Defense Industrial Supply Center, Philadelphia, PA, p. 57.
25. Srinivas, A.R., Determination of Torque Values for Metric Fasteners—An Approach, Space Payload Group, Space Applications Center, Indian Space Research Organization, Ahmedabad, India, Document 380 053, May 1992.

4 Threads and Their Strength

The threads are obviously an important element of the threaded fastener. They give this sturdy industrial product its unique ability to be installed, removed, and reinstalled as many times as we wish. They also affect fastener performance in a major way. As we'll see, thread type, thread class, thread configuration, the way in which the threads are produced, and the fit between male and female threads can affect not only thread strength—and, therefore, fastener tensile strength—but also the resistance of the fastener to such things as self-loosening and fatigue. The amount of preload achieved for a given torque can be influenced by thread configuration and by whether the threads have been cut or rolled. Finally, and not least, the strength of the threads helps to define the strength of the fastener. All things considered, it's worthwhile to take a close look at threads.

4.1 THREAD FORMS

4.1.1 Thread Forms in General

Literally hundreds of thread forms have been designed, and many are still in common use in a wide variety of applications. Fortunately, we only have to worry about the few that are currently used in threaded fasteners. To clear the decks, however, let's start by taking a quick look at three other forms which are often mentioned and which many beginners to bolting assume that they should know about. These forms are illustrated in Figure 4.1, along with a currently popular 60° form for comparison. Anyway, the three we don't have to worry about are [1,25]:

The Acme thread: this is used for power transmission, for example, to produce traversing motion on machine tools.

The Buttress thread: used when the thrust on the screw is in one direction only, for example, airplane propellor hubs and columns for large presses [26].

The Whitworth thread: this form, which had a 55° included angle instead of the now universal 60°, was for decades a British standard form but has now been replaced by an ISO inch series. It was the first screw thread form to have rounded roots, I believe.

All of the modern fastener thread forms we need to know about—the metric as well as inch series forms—are based on an arrangement of 60° angles. As we'll see this basic geometry is modified in several different ways, but it's the starting point for all contemporary fastener thread profiles.

4.1.2 Inch Series Thread Forms

In the United States our principal inch series thread form standards are ASME B1.1 1989 [2] and Federal Standard FED-STD-H 28/2B [3]. Both of these describe the basic unified thread form, identified by the code letters UN/UNR.

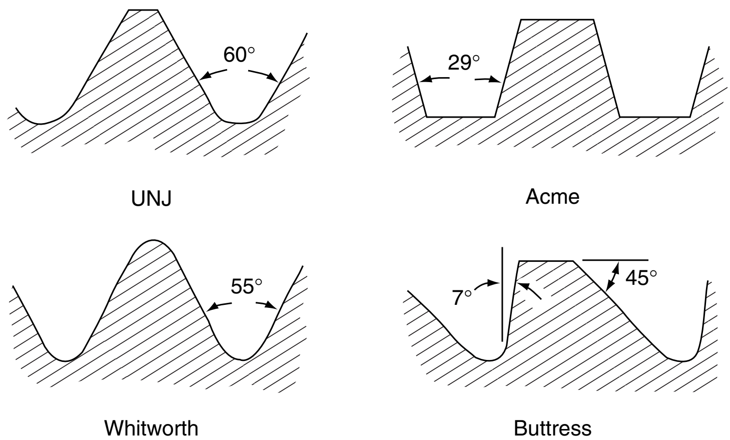


FIGURE 4.1 Three well-known thread forms that are not currently used with threaded fasteners and, for comparison, one which is (the UNJ form). The Acme is a well-known machine tool thread used for traversing screws. The Whitworth is a now-obsolete fastener thread form once used in the U.K. It has now been replaced by a 60 in. series ISO form.

A slightly modified version of the UN/UNR thread is defined in Military Specification MIL-S-8879 C and is called the UNJ form [4]. An ANSI/ASME standard for J threads, B1.15, is in preparation.

The differences between these three forms, UN, UNR, and UNJ, are shown in Figure 4.2. As you can see, the differences are very slight and consist entirely of the way the sharp roots of the teeth are filled in. These differences occur only in the external or male thread form. The same internal thread form is used with each. In any event, the differences are:

UN form has flat-bottomed or, optionally, slightly rounded roots.
UNR form has slightly rounded roots.
UNJ form has generously rounded roots.

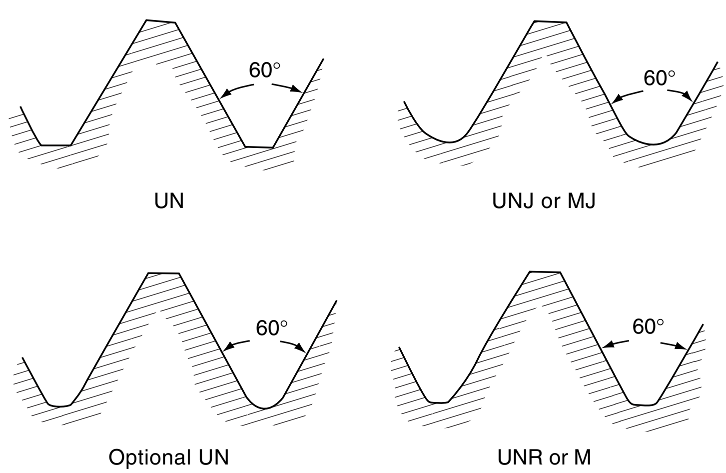


FIGURE 4.2 These are the thread forms most commonly used in the Western world at the present time. Each is a 60° included angle form. They differ from each other primarily in the way the roots of the external (male) threads are shaped. The UN form has flat, or, optionally, slightly rounded roots. The UNJ and metric MJ forms have generously rounded roots. The UNR and metric M forms have slightly rounded roots.

4.1.3 Metric Thread Forms

The currently popular metric threads are identified by the code letters M and MJ. The basic geometry of metric and inch series is identical but the way we define the metric threads differs from the way we define the inch series ones, as we'll see.

U.S. standards for metric threads include ANSI/ASME B1.18M-1982 [22] for standard commercial fasteners and ANSI/ASME B1.13M-1983 (Reaffirmed 1989) for fasteners having nonstandard pitch–diameter combinations or special lengths of engagement [5]. Both define M profile threads. Another standard, ANSI B1.21M-1978, defines MJ threads [6]. The M profile does not include an absolutely flat root option (equivalent to the UN form) but only a "radiused" option—flat with rounded fillets blending into the thread flanks—and a "rounded root" option similar to the UNR profile.

4.2 THREAD SERIES

Design profiles can be applied to threads of any size, and this leads to what are called "thread series." In the unified thread system, for example, we have [2]:

1. Threads called just UN/UNR or UNJ: the constant-pitch series. Each thread in such a series has the same number of teeth per inch. For example, there's an "8 pitch" series, which means that each thread in the group has eight threads per inch. The angle the helix of the thread makes around the fastener varies with the diameter of the fastener, but the depth of the teeth is constant, regardless of diameter, because of the rigid 60° geometry on which the form is based.

   Altogether there are eight constant-pitch thread series, including those with 4, 6, 8, 12, 16, 20, 28, and 32 threads per inch, and they're available in both the basic UN/UNR and the UNJ forms.
2. In addition to the constant-pitch series, there are several groups classified by coarseness. This refers not to their quality but to the relative number of threads per inch produced on a common diameter of fastener. For example, the codes UNC, UNCR, and UNCJ identify coarse-pitch threads.
3. UNF, UNFR, and UNFJ all designate fine-pitch threads.
4. UNEF, UNEFR, and UNEFJ are used for extra fine threads.
5. Provision has also been made for a special series called UNS, UNRS, or UNJS having pitch–diameter combinations not found in any of the standard series above. Most of us will never have to worry about these.

As I mentioned, coarseness designations refer to the relative number of threads per inch. For a fastener of a given diameter, a UNC thread has fewer threads per inch than a UNF, which in turn has fewer than a UNEF. For example, here is the nomenclature we'd use to define the five standard threads currently specified for a fastener having a nominal (body) diameter of 5 in.

The UNC (coarse) thread for that diameter has 13 threads per inch and is encoded as a UNC ½ × 13 thread. Our other options here would be UNRC ½ × 13 or UNJC ½ × 13.

Next in line we have a UN (constant-pitch) thread, which has 16 threads per inch and is called UN ½ × 16 or UNR ½ × 16.

Next there's a UNF (fine) thread with 20 threads per inch, called UN ½ × 20. Then a UNEF (extra fine) thread with 28 threads per inch, or UNEF ½ × 28. Finally, there's another constant-pitch thread for 5 in. fasteners, UN ½ × 32. Like all of the others in the group, it can have UNR or UNJ form too.

These numbers all change as the diameter of the fastener changes. For example, the following group of threads has been codified for 2 in. diameter fasteners:

UNC 2×4–½	UN 2×12
UN 2×6 UN	2×16
UN 2×8 UN	2×20

All the threads in this group are constant-pitch threads except for the first, coarsest thread. This is true of most large-diameter fasteners. In fact, above a 4 in. diameter there are nothing but constant-pitch threads. And once again, each of these threads can have a UNR or UNJ form instead of UN if you wish.

Metric threads generally duplicate the inch series threads, but with threads considered to be only coarse or fine. Instead of using threads per inch to classify them, however, we'll use the pitch distance between two teeth, in millimeters. For example, M6 $\times$ 1 would specify a thread having a nominal diameter of 6 mm and a pitch distance of 1 mm.

4.3 THREAD ALLOWANCE, TOLERANCE, AND CLASS

We still haven't finished classifying threads. The next important consideration is the way the male and female threads fit together. Is the fit loose and sloppy, or is it tight? Rough applications require the first, precision ones the second.

The fit between mating threads is determined by the basic clearance between them, a clearance determined by the way the threads are dimensioned and by the tolerances placed on those dimensions. Although the philosophy is the same for both metric and inch series threads, the nomenclature used is different [2,3,7]; so we'll consider the two types separately.

4.3.1 Inch Series Threads

4.3.1.1 Allowance

The basic clearance is established by a "thread allowance," which determines the minimum clearance between threads of a given class. Another way of saying this is that it determines the minimum distance between male and female threads when both nut and bolt are in their maximum material conditions: the fattest bolt and the thickest nut. The lower sketch in Figure 4.3 shows male and female Class 1A or 2A external threads separated only by the basis allowance.

4.3.1.2 Tolerance

Manufacturing tolerances are now placed on the allowance. These are always in the direction of less material; they always make the clearance between nut and bolt threads greater than the clearance determined by the allowance, never less. The upper sketch in Figure 4.3 shows the clearance between a pair of external 1A or 2A threads when the full tolerance is added to the allowance.

In the unified thread system, the female threads are always dimensioned to a basic profile. It's only the male thread which is reduced a little (made smaller in diameter) by the allowance. Tolerances, again always in the direction of less material, have to be placed on both male and female threads, of course.

4.3.1.3 Class

Three basic fits or "classes" are defined for unified threads. These are given the codes 1A, 2A, and 3A for male threads and IB, 2B, and 3B for female threads. The pair 1A and IB define the loosest fit, 3A and 3B define the tightest.

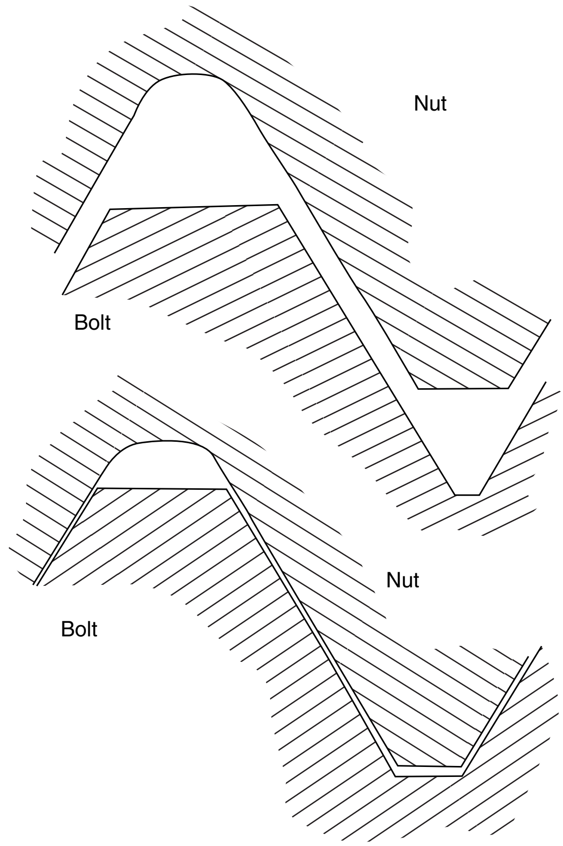


FIGURE 4.3 The lower sketch shows the clearance between male and female Class 2A UN threads when only the basic allowance separates them. The upper sketch shows how much this clearance increases when the full manufacturing tolerance is added to the allowance. In effect, the lower sketch shows the maximum material condition for bolt and nut, the upper sketch the minimum material condition for both. The bolt thread is reduced in diameter by the allowance and tolerance, and the roots of the teeth are rounded. The diameter of the nut teeth has been increased slightly by the manufacturing tolerance, and the roots of the teeth are rounded.

Class 1A, IB threads are used for rough work, for example, where some thread damage can be expected or conditions are very dirty. These are also the easiest threads to assemble. Class 2A, 2B threads are for normal applications. The bolts and nuts you buy in the hardware store are all of this class. Class 3A and 3B fasteners are used for applications requiring an extra degree of precision. Class 1A and 2A threads are assigned the same allowance in the unified system: but more generous tolerances are placed on the l's than on the 2's: so the average fit is looser.

Class 3 threads are assigned a zero allowance; so the fit can be line-to-line. A small tolerance on each thread makes assembly possible.

4.3.2 Metric Threads

The clearance between male and female metric threads is also determined by a basic allowance and by tolerances in the direction of less material. The number of tolerance and allowance options is greater with metric threads than with inch series threads, and different names are used to describe these things [5–7].

4.3.2.1 Tolerance Position (the Allowance)

The basic clearance between male and female threads, called the allowance in inch series threads, is called the tolerance position for metric threads and is identified by letter symbols G or H for internal threads and e, f, g, or h for external threads.

G and e define the loosest fits and the greatest clearance (greatest allowance). H and h define zero allowance, no deviation from the basic profile. Note that although internal metric threads can be assigned zero allowance (H) as with internal UN threads, they can optionally be assigned a tangible allowance (G)—made a little less fat—to accommodate plating or other coatings or perhaps just to provide a looser fit.

4.3.2.2 Tolerance Grade (the Tolerance)

What we call the tolerance of inch series threads is called the tolerance grade for metric threads and is identified by a number symbol. Seven tolerance grades have been established for external threads and are identified by the numbers 3 through 9 inclusive. Nine defines the loosest fit (the most generous tolerance) and 3 defines the tightest. To complicate our lives further, different groups of options have been established for external and internal threads, and tolerances have been placed on several thread dimensions. The possibilities listed in ANSI/ASME 1.13-1983 are shown in Table 4.1.

4.3.2.3 Tolerance Class (the Class)

The tolerance grade and tolerance position symbols are now combined into a "tolerance class" code of alphanumeric symbols. Here are some examples:

6g is a general-purpose callout for external threads. It defines the tolerance position and tolerance grade for the major diameter. Used in conjunction with a 6H nut, the fastener would be a reasonable substitute for applications previously using a Class 2A/2B pair.

4g6g is also a general-purpose callout for external threads, this time when a tighter fit is required. It defines the tolerance grade and position for both the pitch diameter (4g) and the major diameter (6g). Note that in inch series practice these two diameters cannot be toleranced separately as they can here. Class 4g6g, however, is considered to define an external thread which is an approximate equivalent to an inch series Class 3A thread. A nut of tolerance class 6H could be used here. 4H5H defines the tolerance grade and position for a common internal thread which could also be used with the male threads defined above. The 4H classifies the pitch diameter and the 5H the minor diameter. The symbol 5H can also be used alone for applications where a looser fit is acceptable.

4.3.3 Inch Series and Metric Thread Classes, Compared

It's useful to be able to equate inch series and metric allowances and tolerances. A couple of examples were given above. Table 4.2 defines some more options [1]. It doesn't define all of

TABLE 4.1
Tolerance Grades Assigned to Metric Threads

Dimension Controlled	Specified Tolerance Grades
Minor diameter, internal threads	4, 5, 6, 7, 8
Pitch diameter, internal threads	4, 5, 6, 7, 8
Major diameter, external threads	4, 6, 8
Pitch diameter, external threads	3, 4, 5, 6, 7, 8, 9

TABLE 4.2
Approximately Equivalent Classifications: Inch Series and Metric Threads

Inch Series		Metric	
Bolts	**Nuts**	**Bolts**	**Nuts**
1A	IB	8g	7H
2A	2B	6g	6H
3A	3B	4h	5H

the possibilities but only some of the more common "approximate equivalents." Pitch diameter tolerances are not included here for the metric threads, for example. This is not uncommon, but adding them would presumably identify still more approximations.

Coarse- and fine-inch series threads are compared to metric threads in Table 4.3.

4.3.4 Coating Allowances

If fasteners are to be plated or otherwise coated, some clearance must be provided for the coating. The way this is done is presumably of interest only to fastener manufacturers, who would be guided by the applicable standards. Users (buyers) might find the following summary of interest, however.

Inch series: The allowances specified for Class 2A external threads can accommodate coatings of reasonable thickness. Special provisions must be made for other classes of external thread, all internal threads, and threads to be given heavy coatings. Major and pitch diameter limits before and after coating must be specified on engineering drawings, for example [2].

Metric series: External thread tolerance classes 6g and 4g6g provide allowances which can accommodate normal coatings. For heavy coats, or if position h or H tolerance positions are involved, one should consult the standards, e.g., ANSI/ASME B1.13M, for manufacturing allowances [5].

Some problems have occurred in the past with mechanically galvanized fasteners. The male thread dimensions had been reduced rather drastically, to accommodate the coating, and this resulted in a significant loss of tooth strength. This is the kind of problem one sometimes encounters when using low-cost suppliers.

4.3.5 Tolerances for Abnormal Lengths of Engagement

The allowances and tolerances specified in thread standards assume normal lengths of engagement between male and female threads. The definition of "normal" is spelled out in the specifications, but, typically, it means lengths of engagement ranging from 1 to 1½ times the nominal diameter of the thread. Lengths for fine-pitch threads are alternately given in number of pitches, with a range of 5–15 being considered normal [2].

If the length of engagement is to be abnormally short, then it's wise to reduce the clearance between male and female threads by reducing the allowance or the tolerance. If the engagement is to be unusually long, tolerances must be relaxed or pitch mismatch may make it impossible to assemble the fastener or to run the bolt into a deep, tapped hole.

The standards, again, define the procedures for modifying the allowance or tolerance. ANSI/ASME B1.13M is especially clear on this point. It says that for very short lengths of engagement the tolerance on the pitch diameter of the external thread should be reduced by one number. For example, instead of 4g6g one might specify 3g6g.

For extra long lengths of engagement B1.13M says that the allowance on the pitch diameter should be increased. A normal 4g6g would become 5g6g, for example.

TABLE 4.3
Coarse- and Fine-Inch Series and Metric Fastener Series Compared

Thread Diameter (in.) (tpi)	Metric Series	Thread Diameter (in.) (tpi)	Metric Series
0.0600–80UNF		0.7500–10UNC	
0.063–72.6	M1.6 × 0.35	0.7500–16UNF	
0.079–63.5	M2 × 0.4	0.79–10.2	M20 × 2.5
0.0860–56UNC		0.87–10.2	M22 × 2.5
0.0860–64UNF		0.8750–9UNC	
0.098–56.4	M2.5 × 0.45	0.8750–14UNF	
0.1120–40UNC		0.94–8.5	M24 × 3
0.1120–48UNF		1.0000–8UNC	
0.12–50.8	M3 × 0.5	1.0000–12UNF	
0.1380–32UNC		1.06–8.5	M27 × 3
0.1380–40UNF		1.1250–7UNC	
0.14–42.3	M3.5 × 0.6	1.1250–12UNF	
0.1640–32UNC		1.18–7.3	M30 × 3.5
0.16–36.3	M4 × 0.7	1.2500–7UNC	
0.1640–36UNF		1.2500–12UNF	
0.1900–24UNC		1.3750–6UNC	
0.1900–32UNF		1.3750–12UNF	
0.20–31.8	M5 × 0.8	1.42–6.4	M36 × 4
0.2500–20UNC		1.5000–6UNC	
0.24–25.4	M6 × 1	1.5000–12UNF	
0.2500–28UNF		1.65–5.6	M42 × 4.5
0.3125–18UNC		1.7500–5UNC	
0.32–20.3	M8 × 1.25	1.89–5.1	M48 × 5
0.3125–24UNF		2.0000–4.5UNC	
0.3750–16UNC		2.20–4.6	M56 × 5.5
0.39–16.9	M10 × 1.5	2.2500–4.5UNC	
0.3750–24UNF		2.5000–4UNC	
0.4375–14UNC		2.52–4.2	M64 × 6
0.4375–20UNF		2.7500–4UNC	
0.47–14.5	M12 × 1.75	2.83–4.2	M72 × 6
0.5000–13UNC		3.0000–4UNC	
0.5000–20UNF		3.15–4.2	M80 × 6
0.55–12.7	M14 × 2	3.2500–4UNC	
0.5625–12UNC		3.5000–4UNC	
0.5625–18UNF		3.54–4.2	M90 × 6
0.6250–11UNC		3.7500–4UNC	
0.63–12.7	M16 × 2	3.94–4.2	M100 × 6
0.6250–18UNF		4.0000–4UNC	

4.4 INSPECTION LEVELS

Several levels of inspection have been defined for threads, depending upon the nature of the application and the consequences of failure. There is considerable debate at present about which fasteners should be required to pass which tests. We'll take a much longer look at this subject, and the broader subject of thread/fastener strength, later on in the chapter. Since inspection levels are sometimes tacked onto thread designations, it's useful to review them briefly at this point [23].

Level 21 is the least rigorous and is designed to guarantee functional assembly of male onto female threads, plus functional size control of maximum material limits. This level is used with most fasteners at present. The inspection can be performed with fixed GO, NO GO gages.

Level 21A is similar but is used only for metric threads. Level 22 controls the above and also controls the minimum material size limits over the full length of engagement of the thread. Level 23 controls all of the above and also controls, within established maximum–minimum limits, such things as thread flank angles, lead, taper, and roundness. This kind of inspection can be performed only with indicating gages or optical comparators or other devices that allow the inspector to measure all such parameters.

4.5 THREAD NOMENCLATURE

We can now put all of the above together to give the complete alphanumeric code or description of a thread.

4.5.1 Inch Series

An example of an inch series external (bolt) thread "code" would be ¼–20UNC 1A (21),

where
¼ = nominal diameter in inches
20 = number of threads per inch

UNC shows that this is a UN thread from the coarse series.

1A shows that this is a loose fitting, external thread (A) with a finite allowance and a maximum tolerance on both pitch and major diameters (Class 1).

21 shows that the thread is to be inspected with simple GO, NO GO gages.

If that thread were used on a bolt with a 1 in. long body, the code used to define the fastener would be ¼–20 × 1.

Coarseness and fit would not usually be added to the fastener code. The number of threads per inch gives the user coarseness information (a quarter-inch UNF fastener has 28 threads per inch; a quarter-inch UNEF one has 32). A fit of 2A and inspection level 21 would presumably be assumed for such a bolt. Another example would be 0.2500-32 UNJEF 3A, safety critical thread. Here we see the quarter-inch nominal diameter of the external thread given in decimal form followed by the number of threads per inch, 32; the series, UNJ extra fine; and the allowance and tolerance level, 3A. We're also told that this fastener is intended for safety critical applications. This statement defines the quality control and gaging procedures used with it, namely level 23.

4.5.2 Metric Thread

An example of a complete code for an external metric thread would be MJ6 × l-4 h 6 h,

where
M shows that this is a metric thread
J shows that the teeth have rounded roots with larger than standard radii
6 = nominal diameter in millimeters
1 = distance between successive thread crests (i.e., the pitch) in millimeters
4 h = the tolerance grade (4) and tolerance position (h) for the pitch diameter of the thread (position h specifies zero allowance; Grade 4 is used for normal applications)
6 h = the tolerance grade and position for the major diameter (again h signifies zero allowance; Grade 6 is also used for normal applications)

4.6 COARSE- VERSUS FINE- VERSUS CONSTANT-PITCH THREADS

Which is best, coarse-pitch, fine-pitch, or constant-pitch threads? It depends on your application. Each has advantages over the other [2–4,8,9].

4.6.1 Coarse-Pitch Threads

Coarse pitch is generally recommended for routine applications. Such threads will have greater stripping strengths when used with weak nut or joint materials—or when used on larger-diameter fasteners. Some say bolts over 1 in. in diameter should always have coarse threads; others put the crossover point at 1½ in.

It's easier to tap brittle material if coarse-pitch threads are used. Such threads are also easier to use in most cases: easier to start, faster rundown, etc.

4.6.2 Fine-Pitch Threads

Fine-pitch threads must be close fitting—made to Class 3 tolerances—to have acceptable stripping strength, but if this is done the bolts these threads are used on can have higher tensile strengths because the thread root and pitch diameters—and therefore the tensile stress area, A_s—are greater than they would be for a coarse-pitch thread on the same nominal diameter. This advantage can be obtained, however, only with a suitably long length of engagement between male and female threads. We'll study this subject later on in this chapter.

Fine-pitch threads are stronger in torsion, which means that they can be loaded to higher preloads before yielding. They also resist self-loosening under vibration or shock, and resist stress corrosion cracking, better than coarse-pitch threads do.

4.6.3 Constant-Pitch Threads

Constant-pitch threads are designed for applications where there will be repeated assembly and disassembly or where it may be necessary to rethread the part in service. They're used for adjusting collars, for thin nuts or threaded sleeves on shafts. They're also used in the design of compact parts [2,4].

The 8-thread series is used on large-diameter fasteners and was originally intended for bolts used in gasketed joints containing high pressure. It's also widely used as a substitute for coarse series fasteners when the basic fastener diameter exceeds 1 in.

The 12-thread series is used as a continuation of the fine thread series when bolt diameters exceed 1½ in. It was also originally intended for pressure vessels but has now found wider use.

The 16-thread series is also used on large-diameter fasteners, again for those requiring fine-pitch threads. It's used as well for adjusting collars and as a continuation of the extra fine-pitch series for bolt diameters over 1$\frac{1}{16}$ in.

4.6.4 Miscellaneous Factors Affecting Choice

We'll see other thread characteristics that may affect our choice of thread as we proceed through the book, but a few miscellaneous comments may be in order here.

A tighter fit, i.e., 3A versus 2A, gives a 10% increase in thread-stripping strength, because there's more root cross-section to be sheared. The rounded roots of the J profile will increase the strength still further.

The UNJ or MJ threads also have more resistance to fatigue than do the UN/UNR or M threads [21].

Threads tend to strip before the bolt breaks if the male–female fit is loose [10].

The number of threads in the grip (between the face of the nut and the head of the bolt) affects the ductility and stiffness of the fastener. Since we (usually) want ductility and low stiffness (a more resilient spring for better energy storage) it would seem that we'd usually want fully threaded fasteners. We'll be especially interested in ductility if using yield control to tighten the fasteners. (See Chapter 8.)

Factors like the shear strength of the fastener and its fit with its hole, however, often argue instead for partial threads and an unthreaded body of nominal or reduced diameter.

4.7 THE STRENGTH OF THREADS

There's a surprising amount of disagreement on what parameters determine the strength of a thread and on how best to evaluate the quality—including the strength—of a threaded fastener before use. Let's take a look at some conventional wisdom concerning thread strength and then look at some recent thoughts and concerns about thread strength and quality.

4.7.1 Basic Considerations

As we saw in Chapter 1, one of our principal design goals is "a fastener strong enough to support the maximum preload it might receive during assembly, plus the maximum additional loads it might see in service, as a result of forces applied to the joint, differential thermal expansion, etc." The larger the nominal diameter of a fastener, of course, the stronger it will be. As far as static loads are concerned, therefore, we'd like the shank or body of the bolt to be the full, basic, or nominal diameter of the thread, or at least to be greater than the root diameter of the threads [2].

We must then specify a length of thread engagement capable of developing the full strength of that body. This is just another way of saying that we want the body to break before the threads strip, because a broken bolt is easier to detect than a stripped thread.

When the threads strip they do so by shearing in one of three ways. If the nut material is stronger than the material from which the bolt is made, the threads will strip at the roots of the bolt teeth. If the bolt material is stronger, stripping will occur at the roots of the nut threads. If the materials have equal strengths, both nut and bolt threads will strip simultaneously, at their pitch diameters.

Studies made at the National Bureau of Standards many years ago showed that the shear strength for most common fastener materials varied between 50% and 60% of their ultimate tensile strengths. As a result, the stripping areas (A_{TS}), defined in the formulas below (on which the recommended lengths of thread engagement are based) are set at twice the tensile stress area (A_s) of the same thread [5].

If the fastener is to be subjected to fatigue or impact loads, we'd like it to be more resilient than a fastener subjected to static loads. Some recommend a shank (body) diameter about 60% of that used for static loads if the fastener will see impact loads, or a shank diameter of 90% of the static diameter if it will experience fatigue loads (repeated load cycles) [2].

4.7.2 Thread Strength Equations

You'll find both long form and short form thread strength equations in earlier editions of this book. Those equations are complete and accurate but they're not convenient to use because each requires input data for various thread parameters: the maximum inner diameter of the nut threads or the minimum pitch diameter of the bolt threads, for example. Such things are tabulated in long tables in *Machinery's Handbook* or ANSI/ASME B1.1, or FED-STND-H28, each of which also includes a group of thread strength equations. Since you have to go to one of these sources for the data it makes sense to go to them for your thread strength equations as well.

You'll find that the equations allow you to compute two different factors: A_{TS} of the threads and the length of thread engagement (L_e) required to develop the full strength of the fastener. You'll also find that these two are, understandably, directly proportional to each other. Longer engagement means more threads to strip, which means a larger stripping area. You can't compute A_{TS} without knowing or assuming an L_e or compute length of engagement without making assumptions about A_{TS} since each is a function of the other.

One of my goals is to give you all of the data you need to deal with most on-the-job, everyday questions and to answer the problems at the end of each chapter. In pursuit of this goal I have included Tables 4.5 through 4.7. They will give you the thread-stripping areas for a length of thread engagement equal to one nominal diameter of the bolt. This is the common length for thick or heavy hex nuts; which are stronger than thinner, regular hex nuts. You can use an A_{TS} from one of these tables to compute the force required to strip that length of thread, using Equation 4.1 below. You can then compare that with the force required to break your bolt (using Equation 3.1). Remember, nuts are usually made of material weaker than the bolt material but, in an apparent contradiction, you want the bolt to fail before the nut. If your calculations show that the threads would strip before the bolt would break then you'll have to change something: probably increase the length of thread engagement.

In a minute I'll show you how to deal with lengths of engagement greater or less than one diameter. But first, here's the basic procedure.

4.7.3 Thread Strength Computations When $L_e = D$

$L_e = D$, where L_e = length of thread engagement (in., mm) and D = nominal diameter of the bolt (in., mm).

First we compute the force required to strip the threads:

$$F_{St} = S_u A_{TS} \tag{4.1}$$

where

- S_u = ultimate shear strength (psi, MPa) of the nut or bolt materials (from Table 2.7 in Chapter 2)
- A_{TS} = cross-sectional area (in.^{2}, mm^2) through which the shear occurs for a length of engagement equal to D (from Tables 4.5 through 4.7)
- F_{St} = the force (lb, N) required to strip that length of threads of a bolt or nut

Next we compute the tensile force required to break the bolt (F_{Ten}) and compare it to the force required to strip the threads:

$$F_{Ten} = \text{UTS} \times A_S \tag{3.1B}$$

You can find A_S in Tables 4.5 through 4.7, or can compute it from

$$A_S = 0.7895[D - (0.9743/n)]^2 \tag{3.1A}$$

In Equations 3.1A and 3.1B

- UTS = ultimate tensile strength of the bolt material (psi, MPa)
- A_S = the tensile stress area of the bolt (in.^{2}, mm^2)
- D = nominal diameter of the bolt (in., mm)

n = number of threads per inch
F_{Ten} = the tensile force required to break the bolt (lb, N)

If the computed F_{ST} is greater than F_{Ten} you're finished. If not, or if the difference in strength is unnecessarily large (too much length of engagement) proceed as suggested later. First, though, let's look at an example of the basic procedure.

4.7.4 Basic Procedure—An Example

Let's assume that we have the following:

Bolts: ⅝ in. diameter fasteners with 0.625–11UNC Class 2A threads (see Table 4.3) made from ASTM F2281 GR 600 material having a UTS of 130 ksi (see Chapter 2; Section 5).

Joint material: SAE J414 GR 1035 having a shear strength of 54 ksi (see Table 2.7). We'll tighten the bolts in tapped holes in this material rather than in nuts in this example.

From Table 4.6

A_{TS} of a 0.625–11 Class 2A thread $= 0.891$ in.^{2} if $L_e = D$
$A_S = 0.226$ in.^{2}
$F_{St} = S_u \times A_{TS} = 54{,}000 \times 0.891 = 48{,}114$ lbs
$F_{Ten} = \text{UTS} \times A_S = 130{,}000 \times 0.226 = 29{,}380$ lbs

So the bolt will break before the threads will strip if $L_e = D = 0.625$ in. That's the desirable outcome so we may be done. But that's a very large difference between thread and bolt strengths, and suggests that we've asked for a greater length of thread engagement that required. This also suggests "more nut threads than we need" which, in turn, might lead to a slight possibility of pitch interference. In any event, it's useful to know what to do next if we don't like the results of the preceding calculation.

4.7.5 Thread Strength Calculations When $L_e \neq D$

As mentioned earlier the L_e and the A_{TS} are directly proportional to each other. It is, therefore, very easy to compute stripping areas for engagement lengths other than nominal diameter, D:

$$A_{TS}' = (\text{Desired } L_e/D) \times A_{TS} \tag{4.2}$$

where A_{TS} is the value found in Tables 4.4 through 4.6. It is 0.891 in.^{2} in the present example, for a 0.625–11, Class 2A thread (in.^{2}, mm^2).

A_{TS}' = Recalculated stripping area for a length of thread engagement greater to or less than the nominal diameter of the fastener (D) (in.^{2}, mm^2).

For example, what if the joint plate in the previous example were only ½ in. thick rather than ⅝ in. thick?

$$A_{TS}' = (0.5/0.625) \times 0.891 = 0.7128 \text{ in.}^2$$

and

$$F_{St} = 54{,}000 \times 0.7128 = 38{,}491 \text{ lbs}$$

The bolt will still break before the threads tapped in a ½ in. plate will strip, so we could stop here. What if the plate is only ¼ in. thick?

$$F_{St} = 54{,}000 \times (0.250/0.625) \times 0.891 = 19{,}245 \text{ lbs}$$

TABLE 4.4
Selected UNC/UNF/8UN Tread Stress and Shear Areas

Thread	Tensile Stress Area	Bolt Tread Strip Area, Le + 1 Diameter		Nut Thread Strip Area, Le = 1 Diameter	
		2A/2B	3A/3B	2A/2B	3A/3B
0.1900–24UNC	0.0175	0.050	0.053	0.076	0.080
0.1900–32UNF	0.0200	0.052	0.055	0.074	0.078
0.2500–20UNC	0.0318	0.092	0.096	0.135	0.141
0.2500–28UNF	0.0364	0.093	0.101	0.130	0.137
0.3125–18UNC	0.0524	0.147	0.157	0.213	0.222
0.3750–16UCN	0.0775	0.216	0.232	0.310	0.322
0.3750–24UNF	0.0878	0.217	0.242	0.299	0.314
0.4375–14UFC	0.0163	0.296	0.321	0.428	0.443
0.5000–13UFC	0.1419	0.389	0.427	0.562	0.581
0.5000–20UNF	0.1599	0.400	0.444	0.541	0.567
0.5625–12UNC	0.182	0.503	0.548	0.717	0.741
0.6250–11UNC	0.226	0.624	0.683	0.891	0.919
0.6250–18UNF	0.256	0.624	0.708	0.853	0.892
0.7500–10UNC	0.334	0.910	1.00	1.29	1.33
0.7500–16UNF	0.373	0.924	1.04	1.24	1.30
0.8750–9UNC	0.462	1.24	1.38	1.77	1.83
0.8750–14UNF	0.509	1.26	1.43	1.71	1.78
1.0000–8UNC	0.606	1.66	1.82	2.33	2.40
1.1250–8UN	0.790	1.92	2.14	2.95	3.04
1.2500–8UN	1.000	2.65	2.91	3.65	3.75
1.3750–8UN	1.233	3.22	3.55	4.41	4.54
1.5000–8UN	1.492	3.86	4.26	5.25	5.41
1.6250–8UN	1.78	4.55	5.03	6.17	6.35
1.7500–8UN	2.08	5.30	5.87	7.15	7.36
1.8750–8UN	2.41	6.10	6.76	8.20	8.45
2.0000–8UN	2.77	6.96	7.72	9.32	9.61
2.2500–8UN	3.56	8.85	9.83	11.78	12.16
2.5000–8UN	4.44	10.96	12.19	14.53	15.01
2.7500–8UN	5.43	13.30	14.80	17.57	18.16
3.0000–8UN	6.51	15.85	17.67	20.88	21.59
3.2500–8UN	7.69	18.62	20.79	24.47	25.32
3.5000–8UN	8.96	21.63	24.16	28.37	29.36
3.7500–8UN	10.34	24.83	27.78	32.51	33.67
4.0000–8UN	11.81	28.28	31.65	36.96	38.30

Note: All areas are given in square inches.

Now the threads will strip before our ⅝–11 bolt would break, which is not a good idea. In this case we should use a regular or heavy nut to tighten the bolts, rather than tapping the holes in the plate.

4.7.6 Other Stress Area Formulas

In Chapter 3, we were introduced to the most common formula for the thread stress area (Equation 3.1A). We will usually use this expression when computing the strength of a thread and, in fact, have been using it in Section 4.7.3. It could be important for you to know, however, that different expressions must be used for some of the other thread forms illustrated in Figures 4.1 and 4.2. Here are some examples [27].

TABLE 4.5
Selected Metric M Series Thread Stress and Shear Areas

Thread	Tensile Stress Area	Bolt Thread Strip Area, Le = 1 Diameter		Nut Thread Strip Area, Le = 1 Diameter
		6H/6g	6H/4g6g	6H6g and H/4g6g
M5 × 0.8	14.183	35.366	37.085	49.994
M6 × 1	20.123	51.875	54.174	73.145
M8 × 1.25	36.609	97.290	100.74	134.63
M10 × 1.5	57.990	155.80	160.73	214.77
M12 × 1	84.267	227.72	234.86	313.41
M14 × 2	115.44	313.94	323.24	434.47
M16 × 2	156.67	417.55	429.92	569.12
M20 × 2.5	244.79	665.58	682.06	907.76
M22 × 2.5	303.40	814.67	—	1,100.3
M24 × 3	352.50	971.10	994.23	1,320.2
M27 × 3	459.41	1,246.7	—	1,674.3
M30 × 3.5	560.59	1,549.2	1,582.5	2,086.5
M36 × 4	816.72	2,271.5	2,315.8	3,026.2
M42 × 4.5	1,120.9	3,120.2	3,175.2	4,168.3
M48 × 5	1,473.2	4,119.5	4,187.3	5,482.0
M56 × 5.5	2,030.0	5,680.3	5,769.4	7,510.2
M64 × 6	2,676.0	7,483.6	7,596.4	9,847.3
M72 × 6	3,459.8	9,574.2	9,718.5	12,478
M80 × 6	4,344.1	11,922	12,101	15,419
M90 × 6	5,590.8	15,217	15,446	19,534
100 × 6	6,994.7	18,856	19,170	24,059

Note: All areas given in square millimeters.

UNJ threads rolled after heat treatment:

$$A_S = 0.7854(d_{2\,\max})^2 \tag{4.3}$$

UNJ and MJ threads rolled before heat treatment or not rolled at all:

$$A_S = 0.7854[0.5(d_{2\,\max} + d_{3\,\max})]^2 \tag{4.4}$$

M-form metric threads:

$$A_S = 0.7854(D - 0.9382P)^2 \tag{4.5}$$

Metric MJ threads rolled after heat treatment:

$$A_S = 0.7854(d_{3\,\max})^2[2 - d_{3\,\max}/d_{2\,\max})^2] \tag{4.6}$$

where
D = basic pitch diameter (in., mm)
P = thread pitch (in., mm)
$d_{2\max}$ = maximum pitch diameter (in., mm)
$d_{3\max}$ = maximum rounded root minor diameter

4.8 WHAT HAPPENS TO THREAD FORM UNDER LOAD?

When torque is applied to a nut, the nut moves along the bolt thread and is pressed against the surface of the part to the fastened. This force at the bearing surface compresses the nut; the nut, in turn, transmits this force to the bolt and develops a tensile stress in the bolt. Since the nut is compressed, its thread lead is reduced. The tension in the bolt causes it to stretch, so its thread lead is increased.

Before the load is applied to the nut, the thread leads of nut and bolt are the same. But now that the joint is loaded, the threads no longer match. This puts an uneven load on the threads, with the highest load concentrated at the threads nearest the bearing surface of the nut. Localized yielding results in some of the load being transmitted to subsequent threads. The problem is that high stresses are concentrated in just a few threads. Even if the threads do not strip, the fatigue life of the joint is reduced.

Under load, forces are transmitted from nut thread flanks to bolt thread flanks. When nut and bolt flank angles are the same, contact between threads is across the full flanks and the effective load acts as if it were at the centers of the flanks. This force produces a moment around the thread roots, like that of a cantilever beam, and results in stresses concentrated at the roots.

If the flank angle of a thread ridge in the bolt thread is smaller than that of the mating nut thread ridge, contact will be at the crest of the bolt thread and the cantilever beam stress at the bolt thread root will be approximately twice that which was experienced when flank angles were equal. But if the flank angle of the bolt thread is larger, contact between nut and bolt threads is near the bolt thread root and cantilever beam stresses at the root are minimized (see Figure 4.8). Minimizing the bolt thread root stresses is known to reduce bolt fatigue. (It has been recognized that at the same time stresses at the root of the bolt thread are minimized, stresses at the root of the nut threads are increased. Nuts and other tapped parts are generally less likely to fail from fatigue, however.)

In the 1960s, Standard Pressed Steel Company (now SPS Technologies, Inc.), on the basis of the known effects of changes in thread form under load, developed modifications to the standard UNJ form thread (also a development of SPS) to improve bolt fatigue life. One modification reduced the lead of the bolt thread just enough that under load it would be the same as the lead of the loaded nut thread while still permitting assembly prior to applicable of the load. The other modification increased the pressure flank angle of the bolt thread by 5° to ensure contact of the nut thread ridges near the bolt thread roots, to minimize bending stresses at the root. This thread is called the asymmetric thread and is used on bolts that must have the best possible fatigue performance.

4.9 THINGS THAT MODIFY THE STATIC STRENGTH OF THREADS

4.9.1 COMMON FACTORS

We saw earlier that there are a number of factors which can modify the anticipated tensile strength of a bolt—such things as high temperature, corrosion, torsion, or cyclic loading. These things can also modify the strength of threads. So can some other factors which aren't quite as obvious, given below.

Nut dilation [12,13]. If the walls of the nut are not thick enough, the wedging action of the threads will dilate the nut, partially extracting the nut threads from the bolt threads. This reduces thread engagement, and therefore reduces the cross-sectional areas which support the shear load, reducing shear strength. The standard width to nominal diameter ratio for medium diameter threads (e.g., 5/8 in.) is 1.7:1. The ratio for large diameters (e.g., 1 1/2 in.) is 1.6:1. The ratio for small diameters (e.g., 1/4 in.) is 2:1 [28]. If the ratio between width

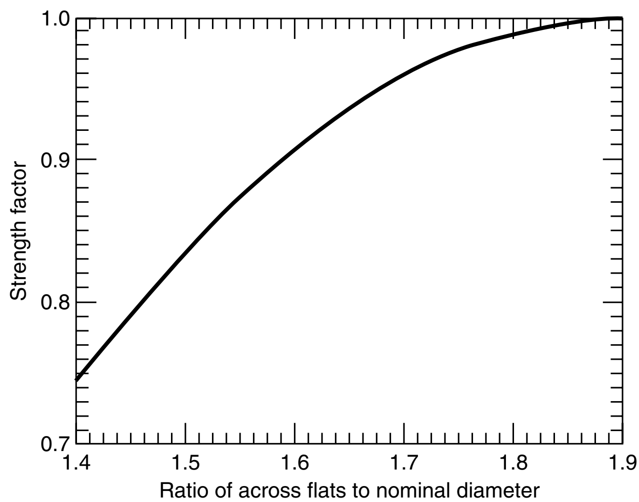


FIGURE 4.4 Strength reduction factor for nut dilation, as a function of the ratio of the across-the-flats distance to the nominal diameter of the fastener. (Modified from the formula for calculating the stripping strength of internal threads in steel, Report to ISO/TCI/WG4 by Sweden-Bultfabrike, AB, 1975.)

across flats and nominal diameter is only 1.4:1, for example, strength will be reduced by 25% as shown in Figure 4.4. The ratio for small diameters (e.g., 1/4 in.) is 2:1 [28]. Note that the reduction applies to both nut or bolt threads, the failure occurring in the weaker of the two.

Oversized holes are often used in structural steel, shipbuilding, and other applications involving massive joint members which are difficult to align. Dilation of regular hex nuts can be increased if only the tips of the nut hex contacts the joint. Washers and heavy hex nuts are recommended in such situations [29].

Relative strength of nut-to-bolt threads [14–16]. As we have seen, the relative strength always determines which members will fail. If there is too big a difference between the two materials, another factor must be considered: The weaker of the two threads will deflect under the relatively stiff action of the other, creating a form of thread disengagement that again reduces the area supporting shear stress. Note that it doesn't matter which thread—nut or bolt—is substantially weaker than the other. The result is shown in Figure 4.5.

Coefficient of friction. If the coefficient of friction between nut and bolt threads is too low, then both nut dilation and thread bending become more likely because the threads can pull apart more readily. A lubricant such as phosphate and oil, for example, is said to reduce resistance to thread stripping by as much as 10% [14].

Rotary motion [14,17]. Dynamic friction is usually less than static friction. As we have seen above, anything that reduces friction between nut and bolt threads makes it easier for the nut to dilate and for the threads to bend. This means that the threads are a little more likely to strip during torquing operations when the nut is moving relative to the bolt than they are under static loads. The reduction in strength is estimated to be approximately 5%.

To compute the modified potential strength of a nut thread, therefore, one multiplies the apparent strength in pounds by the appropriate nut dilation factor from Figure 4.4 and by the appropriate thread bending factor (for nuts) from Figure 4.5. If the threads are lubricated, the computed strength should be reduced by an additional 10%; if torque is used to tighten the nuts, a final 5% reduction is required.

Similar calculations are used to estimate the strength of the bolt threads, the only difference being that the thread bending factor used (from Figure 4.5) will be that for bolts

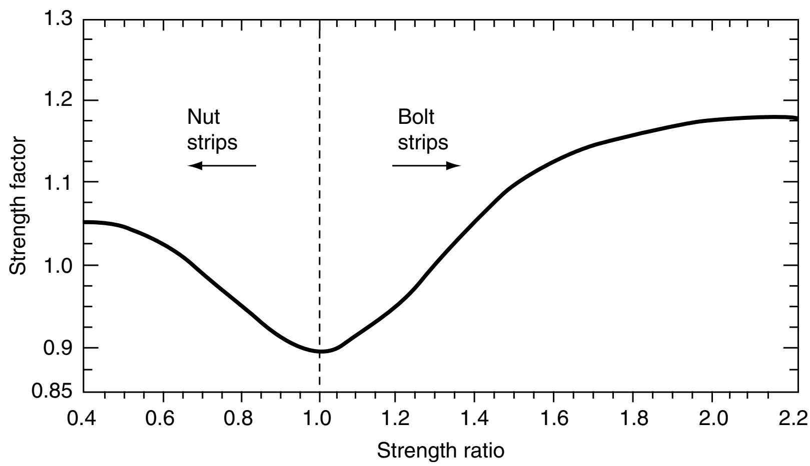


FIGURE 4.5 Strength reduction factor for thread bending. The horizontal axis gives the ratio of nut strength to bolt strength. (Modified from Alexander, E.M., Design and strength of screw threads, Transactions of Conference on Metric Mechanical Fasteners co-sponsored by ANSI, ASME, ASTM, and SAE, presented at American National Metric Council Conference, Washington, DC, 1975.)

rather than for nuts. As an example, let's compute the strength of the threads for the 0.625-11, 2A bolt whose thread-stripping area from Table 4.5 is 0.624 in.^{2}. Let's assume that because of space limitations we're using a nut with a ratio of width across flats to nominal diameter of 1.45:1, a little less than normal.

The nut is 0.547 in. thick (see Appendix F). Let's now assume that the nut material is 25% weaker than the bolt material, with a shear strength of 98 ksi. We're going to use it with an ASTM F2281 GR 600 bolt whose UTS is 130 ksi (Table 4.2). We can estimate the bolt's shear strength at 60% of its UTS or 78 ksi.

The bolts are to be lubricated with molydisulfide, an even better thread lube than phosphate and oil (see Table 7.1) and they'll be tightened with a torque wrench.

We use Equation 4.1 to compute the theoretical force required to strip the threads of the bolt.

$$F_{St} = S_u \times A_{TS}$$

where

S_u = 96 ksi
$A_{TS} = 0.891 \times (0.547/0.625) = 0.780$ in.^{2}
$F_{St} = 78{,}000 \times 0.780 = 60{,}825$ lbs

Now we apply the strength reduction factors as follows:

SR1 = strength reduction factor for nut dilation for a 1.45:1 ratio = 0.8 (from Figure 4.4).
SR2 = strength reduction factor when the nut material is 25% stronger = 1.1 (from Figure 4.5).
SR3 = coefficient of friction factor. Let's assume 15% loss of strength (which is probably conservative) because moly is almost 50% more lubricious than phos-oil is (see Table 7.1), so SR3 = 0.85.
SR4 = rotary motion factor, a loss of 5%, so SR4 = 0.95.

The reduced estimate for the strength of our threads is now:

$$F_{St}' = F_{St} \times SR1 \times SR2 \times SR3 \times SR4$$
$$F_{St}' = 60{,}825 \times 0.8 \times 0.975 \times 0.85 \times 0.95 = 38{,}311 \text{ lbs}$$

A significant difference! As we saw, the bolt has an F_{Ten} of 29,380 lbs so the bolt would presumably break before these threads stripped, which is desirable. Not much margin for safety, however.

Is this analysis valid? The reduction factors we've just used come from studies made by Alexander for the SAE [14] and the results, commonly called the "Alexander model," have been widely accepted and used. Conclusion: the estimate we've just made is valid. If we needed more strength than that implied by the results we should increase the length of engagement between the bolt and our abnormally thin-walled nut.

4.9.2 Which Is Usually Stronger—Nut or Bolt?

We have considered this issue several times, but it is important enough to elaborate. You will find that the proof strength of a standard nut is generally greater than the proof strength of the fastener with which it is supposed to be used. Designers would prefer bolt failure to nut failure because a failure of the bolt is more obvious. For example, the amount of torque we can apply to a bolt with a stripped thread is often greater than that we had applied just before stripping occurred. The increased torque indicates an increase in tension or preload in the bolt, when, in fact, all preload is lost when the thread fails. On the other hand, there's no chance of misreading the situation when the body of a bolt breaks; that's obvious. In an apparent contradiction, nuts are made of weaker (softer) materials than bolts. This encourages plastic yielding in nut threads to bring more threads into play in supporting the load. But the nut as a body will still withstand a higher tensile force than the mating bolt. For the same reason designers will want tapped holes to be deep enough to more than support the full strength of the bolt. Use of the suggested length of engagement design procedure (Equations 3.1 and 4.1) can be used to achieve this result. Most people will use this procedure, or its equivalent in ANSI/ASME B1.1, only to find the thread length of tapped holes. They won't design nuts.

Note that standard nuts come in several configurations. As far as hex nuts are concerned, a regular hex nut has a thread length equal to 0.875 times the nominal diameter of the bolt. Thick and heavy hex nuts have a length equal to the nominal diameter. All three should be able to develop the full strength of the bolt with varying factors of safety, but there can be problems. You'll find a further discussion, and some recommendations on which nut to use, in Table 2.3.

4.9.3 Tables of Tensile Stress and Shear Areas

Equations are nice, but it's often handy to have a table of "answers." Tables 4.4 and 4.5 give stress areas and thread-stripping areas. The stripping areas for internal threads are 1.3–1.5 times those shown in Table 4.6. Remember, we always want the bolt, not the nut, to fail.

As mentioned earlier, a more complete table of tensile stress areas will be found in Appendix E. The thread-stripping areas given above are in square inches for a length of thread engagement equal to one nominal diameter of the bolt (the common length for thick or heavy hex nuts). The stripping areas for the mating internal threads would be 1.3–1.5 those shown here.

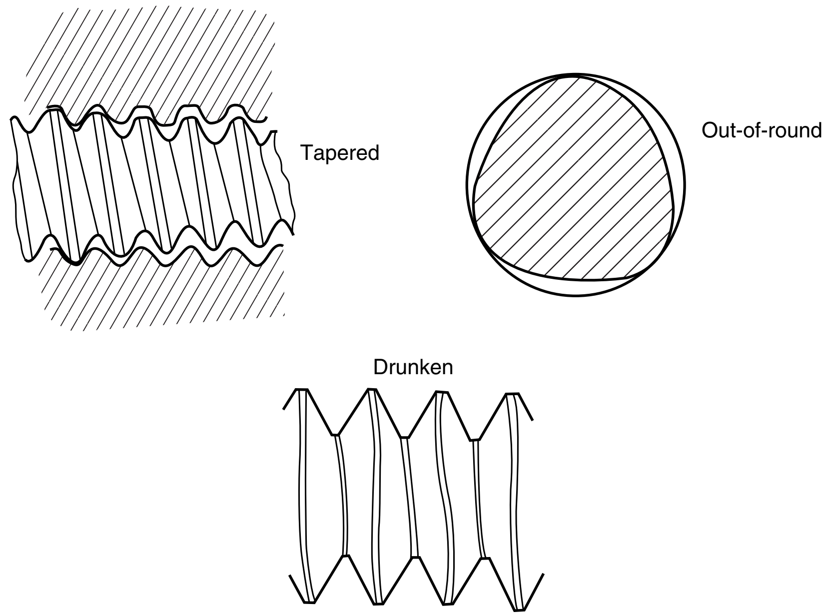


FIGURE 4.6 Tapered out-of-round or drunken threads all reduce thread-stripping strength.

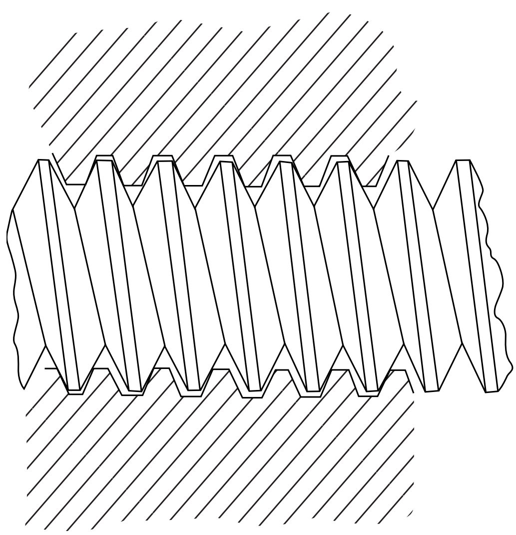


FIGURE 4.7 If the pitch of the male threads differs significantly from that of the female threads, they may be in contact for only part of the length of engagement.

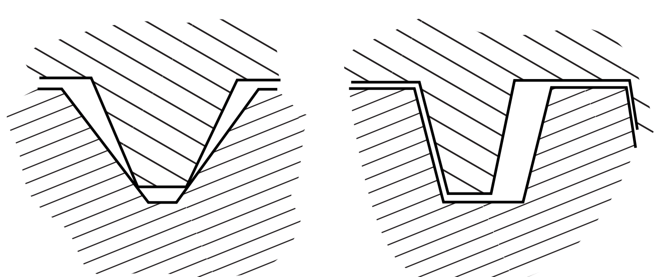


FIGURE 4.8 Incorrect tooth angles can also result in improper engagement and loss of thread strength.

4.10 OTHER FACTORS AFFECTING STRENGTH

So much for the conventional wisdom concerning thread strength. The formulas we've looked at were all based on the assumption that the threads would be manufactured within tolerances specified by ANSI/ASME B1.1 or equivalent. Recent aerospace and other experience has suggested that this may not be enough, that various thread distortions can be produced during manufacture—and may have a significant effect on the thread's strength and performance. The issue is currently being debated, and a research project to resolve it is under way or planned at this time (mid-2007). In any event, here are some of the factors whose importance has been questioned and is, therefore, currently being studied.

4.10.1 Pitch Diameter

If thread geometry gets too far away from that defined by the ASME B1.1 standard, the equations of this chapter may no longer work [19]. And small differences in thread dimensions, angle, etc. may make a significant difference in thread strength [20]. The pitch diameter of a 0.625–18UNF-3A/3B thread, for example, is a nominal 0.5889 in. ASME Bl.l allows a tolerance of +0, −0.0035 in. on that pitch diameter. Assuming that this bolt is used with a nut having a nominal pitch diameter (also 0.5889 in.), a bolt having the minimum pitch diameter allowed by the specification would have 16% less thread strength than a bolt with a nominal pitch diameter.

If the pitch diameter of that bolt is 10 mils less than nominal, it will have less than half the rated strength; at 20 mils, it will have only a quarter of its rate strength; all, still, if used with a nut having nominal PD.

A similar loss of strength occurs if the pitch diameter of the nut is greater than nominal. If both nut and bolt are wrong—the nut being too large and the bolt too small—the loss of strength can be almost complete.

4.10.2 Other Thread Parameters

Pitch diameter is not the only geometrical factor we must be concerned about. Anything which reduces the amount of contact between male and female threads will affect their strength. If the bolt or nut is slightly tapered, for example, the threads will be partially disengaged at one end of the engagement length. If the threads on either are slightly out-of-round, they will not be fully engaged during a portion of each turn. If the pitch of the male threads differs from that of the female threads, they will be in engagement only over a portion of the engagement length. If the helix angle of nut or bolt is irregular, we can get a condition called a "drunken" thread. If the flank (included) angle of the teeth is too great or too small, we will get improper engagement. Some of these problems are illustrated in Figures 4.6 through 4.8. All of them, again, may cause a significant loss of strength in the threads: but studies are needed to confirm this. It has also been suggested that improper thread profiles such as those shown in these figures can reduce the resistance of a fastener to shock or vibration and therefore will encourage self-loosening.

These problems, incidentally, have been more common in recent years, again thanks to manufacturers of low-cost bolts.

Note that simple GO and NO GO thread gages will not catch such problems as incorrect flank angle or incorrect pitch diameter. They really only check basic root and nominal diameter; so a bolt can pass such gages and still may have less than normal thread strength. Indicating gages are available, however, which check all of the necessary geometrical factors.

EXERCISES AND PROBLEMS

1. A thread is defined as ½–13UNC 3B (21). What does that tell us about the thread?
2. Another thread is defined as M8×1.25 6H. What does that tell us?
3. We have a ¼–20 Class 2A thread on an ASTM A325 bolt. How much force would be required to break the bolt? (refer Section 2.5.2, Table 4.4, and Equation 3.1)
4. How much force would be required to strip the threads of the bolt described in #3 above? (refer Sections 2.5.2 and 2.8)
5. How much force would be required to strip the threads of an A540 heavy hex nut with 1–8UNC threads? (refer Sections 2.5.3 and 2.8 and Table 4.5)
6. Could the bolt described in #3 be safely used with a tapped hole in an SAE J453 GR 306 aluminum die casting 3⁄16 in. thick? (refer Tables 2.7 and 4.5)
7. We are planning to use a special nut on the bolt described in #3 above. Here are the parameters of the nut:
   Material: 6061 T6 aluminum
   Across the flats dimension: 0.4 in.
   The nut will be lubricated. It will be tightened with a hand wrench.
   How much force would be required to strip the threads of this nut? (refer Section 2.5.2, Table 2.1, and Figures 2.4 and 2.5)
8. We are planning to use ASTM F2282 GR IFI-1006 bolts with 0.3750–16UNC threads tightened into tapped holes in a ¼ in. thick joint plate made of ASTM A36 low carbon steel. Is that a good idea? (refer Table 2.7)
9. How many threads are there in the tapped hole in Problem 8?

REFERENCES AND BIBLIOGRAPHY

1. Sharman, J.M., Threaded fastener selection, *Engineering* (U.K.), October 1975.
2. Unified Inch Screw Threads UN and UNR Thread Forms, ASME Standard B 1.1-1989, ASME, New York, 1989.
3. Screw Thread Standards for Federal Services—Section 2, Unified Inch Screw Threads—UN and UNR Forms, FED-STD-H28/2B, Federal Supply Services, General Services Administration, Philadelphia, August 20, 1991.
4. Screw Threads, Controlled radius root with increased minor diameter, General specification for, MIL-S-8879C, U.S. Government Printing Office, Washington, DC, July 25, 1991.
5. Metric Screw Threads—M Profile, ANSI/ASME Bl.13M-1983, ASME, New York, 1983.
6. Metric Screw Threads—MJ Profile, ANSI B1.21M-1978 (Reaffirmed 1991), ASME, New York, 1991.
7. Nomenclature, Definitions, and Letter Symbols for Screw Threads, ANSI/ASME B1.7M-1984, Reaffirmed 1992, ASME, New York, 1992.
8. Thread forms and torque systems boost reliability of bolted joints, *Product Engineering*, December 1977, p. 37.
9. Waltermire, W.G., A fresh look at a basic question: coarse or fine threads? *Machine Design*, March 17, 1960.
10. Comments made by Richard T. Barrett of NASA Lewis Research Center at a bolting conference jointly sponsored by the Industrial Fasteners Institute and the Federal Highway Commission at the NASA Lewis Research Center, Cleveland, November 4, 1992.
11. Unified Inch Screw Threads, ANSI Standard Bl.1-1974, ASME, New York, 1974.
12. Ellison, H.W., *Effect of Nut Geometry on Nut Strength*, General Motors Corp., Warren, MI, 1970.
13. Formula for calculating the stripping strength of internal threads in steel, Report to ISO/TCI/WG4 by Sweden-Bultfabrike, AB, 1975.
14. Alexander, E.M., Design and strength of screw threads, Transactions of Conference on Metric Mechanical Fasteners co-sponsored by ANSI, ASME, ASTM, and SAE, Presented at American National Metric Council Conference, Washington, DC, 1975.

15. Gill, P., *The Static Strength of Screw Threads*, G.K.N., U.K.
16. Parisen, J.D., *Length of Thread Engagement in Nodular Iron*, General Motors Corp., Warren, MI, 1969.
17. Wiegand, H. and K.H. Illgner, *Boltbarkeit von Schraebenuer-bindungen mit ISO*, Gewindeprofil, Konstruktion, 1967.
18. Blake, A., *What Every Engineer Should Know about Threaded Fasteners*, Marcel Dekker, New York, 1986.
19. Yura, J.A., K.H. Frank, and D. Polyzois, High strength bolts for bridges, PMFSEL Report 87-3, University of Texas, Austin, May 1987. From information received from Stanley P. Johnson of the Johnson Gage Company, Bloomfield, CT, May 1988.
20. McNeill, W., A. Heston, and J. Shuetz, A study of factors entering into the calculation for connecting rod joints according to VDI 2230 guidelines, *VDI Berichte, No. 766*, VDI, Germany, 1989.
21. Metric Screw Threads for Commercial Mechanical Fasteners—Boundary Profile Defined, ANSI B1.18M-1982 (Reaffirmed 1987), ASME, New York, 1987.
22. Screw Thread Gaging Systems for Dimensional Acceptability—Inch and Metric Screw Threads (UN, UNR, UNJ, M, and MJ), ANSI/ASME B1.3M-1986, ASME, New York, 1986.
23. *Machinery's Handbook*, the 14th or any more recent edition, The Industrial Press, New York.
24. ACME Screw Threads, ASME/ANSI Bl.5-1988, ASME, New York, 1988.
25. Buttress Inch Screw Threads, ANSI Bl.9-1973 (Reaffirmed 1985), ASME, New York, 1985.
26. *Handbook of Bolts and Bolted Joints*, Bickford, J.H., and Nassar, S. (Eds.) Marcel Dekker, New York, 1998.
27. Shigley, J.E. and C. Mischke, *Fastening, Joining and Connecting*, reprinted from *The Mechanical Designers Workbook*, McGraw Hill, NY, 1990.
28. Barron, J., "Where to Use Heavy Hex Nuts," Newport News Naval Shipyard, March 2006.

5 Stiffness and Strain Considerations

As we learned in Chapter 1, both bolt and joint members are, in effect, stiff springs. They deflect under load. They relax when the load is removed. They store potential energy and can create an appropriate clamping force—and therefore function as an effective joint—only as long as they retain "enough" of that energy.

Because of all this, one of the most important properties of the bolt and joint members is their stiffness. Less stiff—I'll often call them "softer"—springs can often store energy more effectively than very stiff ones, so we'll be interested in how stiff they are. In addition, the "joint stiffness ratio," which I'll define near the end of this chapter, is an extremely important design parameter which affects the way the bolts and joint members absorb external loads, respond to changes in load, respond to changes in temperature, etc.

Let's look in detail, then, at this concept of stiffness and at the related deflection of or strain in the bolt and joint members. We'll start with the bolt, then examine the joint, and then take a brief look at the stiffness ratio.

5.1 BOLT DEFLECTION

5.1.1 Basic Concepts

Let's apply equal and opposite forces to the ends of a rod of nonuniform diameter, as shown in Figure 5.1. If the tension stress created in the rod is below the proportional limit, we can use Hooke's law and the relationship between springs in series to compute the change in length of the rod.

The combined change in length of the rod will be equal to the sum of the changes in each section:

$$\Delta L_C = \Delta L_1 + \Delta L_2 + \Delta L_3 \tag{5.1}$$

Hooke's law tells that the change in one section will be

$$\Delta L = \frac{FL}{EA} \tag{5.2}$$

where

ΔL = change in length (in., mm)
A = cross-sectional area (in.^{2}, mm^2)
L = length of the section (in., mm)
E = modulus of elasticity (psi, GPa)
F = applied tensile force (lb, N)

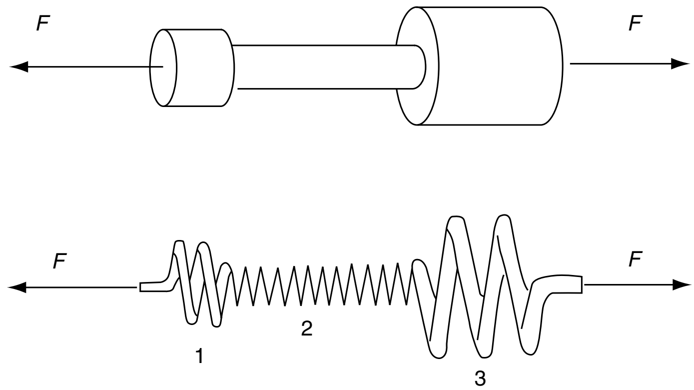


FIGURE 5.1 Rod of nonuniform diameter, loaded in tension, and equivalent spring model.

Since the various sections are connected in series, they each see the same force, so we can combine Equations 5.1 and 5.2 above and write

$$\Delta L_C = F\left(\frac{L_1}{EA_1} + \frac{L_2}{EA_2} + \frac{L_3}{EA_3}\right) \tag{5.3}$$

Now, the spring constant of a body is defined as

$$K = \frac{F}{\Delta L} \tag{5.4}$$

where
K = spring constant or stiffness (lb/in., N/mm)
ΔL = change in length of the body under load (in., mm) F
F = applied load (lb, N)

The spring constant of a group of bodies, connected in series, is

$$\frac{1}{K_T} = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3} \tag{5.5}$$

where
K_T = combined spring constant of the group (lb/in., N/mm)
$K_1, K_2, \ldots$ = spring constants of individual members of the group (lb/in., N/mm)

Now, the equation for the spring constant of a body can be rewritten as

$$\Delta L = \left(\frac{F}{K}\right) \quad \text{or} \quad \Delta L = F\left(\frac{1}{K}\right) \tag{5.6}$$

Comparing our equation for the spring constant for a group of bodies to the equation for the stretch or change in length of a group of bodies, we see that

$$\left(\frac{1}{K_T}\right) = \frac{L_1}{EA_1} + \frac{L_2}{EA_2} + \frac{L_3}{EA_3} \tag{5.7}$$

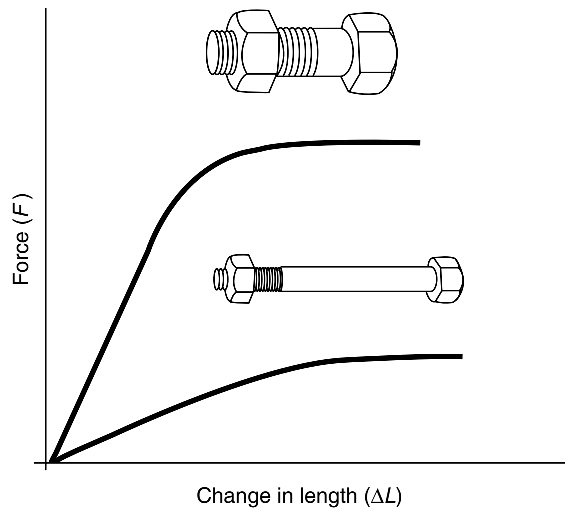


FIGURE 5.2 Elastic curves for a short, stubby bolt and a long, thin bolt cut from the same material.

Note that the stiffness of either a plain or complex body is very much a function of the ratio between length and cross-sectional area—it's a function, in other words, of the "shape" of body just as much as it is a function of the material from which the body is made. If we take one piece of alloy steel and make two bolts from it, one a short, stubby bolt and the other long and thin, and we then place the bolts in tension and plot elastic curves for them, we will end up with two curves such as shown in Figure 5.2.

The equations used for a rod having several different diameters are basically the equations we would use for computing change in length and stiffness of bolts. If we can compute or predict the lengths, cross-sectional areas, and modulus of the material, we should be able to compute the deflection under load. There is some ambiguity, however, about each of these factors when we're dealing with bolts. We'll take a closer look.

First, though, note that I included the modulus of elasticity in the list of things we'll need to know. We have not needed that material property until now. Most engineers dealing with bolts and joints don't need to know the modulus because they're primarily interested in the tensile or shear strengths of bolts, threads, joint members, etc. Those, like us, who are also interested in the response of bolts and joints to service loads, temperature cycles, etc. will find the modulus a very important factor. Table 5.1 lists the modulus for many bolt and joint materials.

5.1.2 Change in Length of the Bolt

5.1.2.1 Effective Length

Tensile loads are not applied to bolts "from end to end"; they're applied between the inner face of the nut and the undersurface of the head. The entire bolt is not loaded, therefore, the way the test rods are. There is zero tensile stress in the free ends, for example.

There is, however, some stress in portions of both the head and the threads (see Figure 3.5). We cannot assume that the bolt is merely a cylinder equal in length to the grip length. Instead, we have to make some assumption concerning the stress levels which will allow us to estimate an "effective length" for the bolt that is somewhere between the true overall length and the grip length.

TABLE 5.1
Modulus of Elasticity at Room Temperature ($\times 10^6$ psi)

Bolt Materials	Modulus	Ref.
ASTM A193 B5	30.9	[13]
B7	29.7	[13]
B8-Cl 1	28.3	[13]
B16	29.7	[13]
ASTM A325	29.5	[14]
ASTM A354	29.3	[13]
ASTM A490	29.3	[13]
H-11	30.6	[14]
Inconel 600	31.4	[14]
Ti 6Al-4V	16.5	[14]
Metric SAE J1199		[17]
4.6/4.8/5.8	29.5	
8.8/9.8/10.9	29.3	
Joint materials		
Steels low carbon	29.5	[15]
Medium carbon	29.3	[15]
Chrome-Moly	29.7	[15]
Austenitic	28.3	[15]
Aluminum		
2024	10.6	[15]
Cast	8–10	[16]
Iron		
SAE J158 Malleable	25–26	[18]
SAE J434		
Ductile	22	[16]
Cast	12–14	[16]
Wrought	26–29	[16]
Cold rolled brass	13.1	[16]

We know from Chapter 3 that tensile stress in a bolt is maximum near the inner faces of the head and nut, and that tensile stress is zero at the outboard faces of the nut and head. Assuming that there is a uniform decrease in stress from inboard to outboard faces of the head, as suggested by Figure 3.4, we can make the assumption that the average stress level in the head of the bolt is one-half the body stress; or we can make a mathematically equivalent assumption and say that one-half of the head is uniformly loaded at the body stress level and that the rest of the head sees zero stress. Similarly, we can say that one-half of the threads engaged by the nut are loaded at the "exposed thread" stress level. We are now in a position to say that the effective length (L_E) of the fastener is equal to the length of the body (L_B) plus one-half the thickness of the head (T_H) added to the length of the exposed threads (L_T) plus one-half the thickness of the nut (T_N), as suggested by Figure 5.3 or by

$$L_E = (L_B + T_H/2) + (L_T + T_N/2) \tag{5.7a}$$

Compare the actual stress levels sketched in Figure 5.3 with those shown in Figures 3.5, 3.6, and 3.7. We have taken the simplest case for estimating the effective length of our bolt. There's really no simple way we could deal with the "true" stress distribution, which would involve a finite-element analysis or the like and would require more information about the geometry of a particular bolt and joint than we'll ever have in practice. We'll find, however, that the assumptions we have made give us reasonable predictions in many applications,

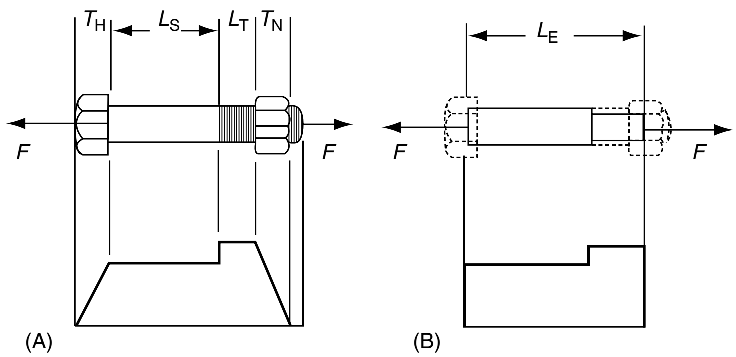


FIGURE 5.3 Illustration of actual bolt configuration and average tensile stress levels (A), and the equivalent configuration and stress distribution assumed for calculation purposes (B).

because the bulk of the fastener is stressed at, or near, the levels we have assumed. It is only the surfaces of the fastener that exhibit the maximum deviations from these averages.

At least that's true as far as long, thin bolts are concerned. As the length-to-diameter ratio of the bolt decreases, and the bolt becomes more and more short and stubby, our assumption of effective length becomes more and more suspect. More about this in Chapter 9.

I have suggested that we use one-half the thickness of the head and one-half the thickness of the nut to compute the effective length of the equivalent fastener. I should mention in passing that other sources recommend slightly different correction factors, such as $0.4 \times$ nominal diameter for the head, and another $0.4 \times D$ for the nut [1]; or $0.3 \times \mathrm{D}$ for each [2]; or nothing for the head and $0.6 \times \mathrm{D}$ for the nut [3]. The thickness of a standard heavy hex nut, incidentally, is equal to the nominal diameter, so the correction I have suggested is equal to $0.5 \times D$ for a heavy hex nut and a little more for a light nut.

5.1.2.2 Cross-Sectional Areas of the Bolt

We also have to make some assumptions concerning cross-sectional areas of the bolt when computing change in length. The body area is no mystery; it's merely equal to $\pi \times D^2/4$, where D is the nominal diameter of the fastener.

For the cross-sectional area of the threads, however, we must use the effective or "stress area" discussed in Chapter 3.

5.1.3 Computing Change in Length of the Bolt

We can now compute the approximate change in length of the bolt under load:

$$\Delta L_C = F_P\left(\frac{L_{be}}{EA_B} + \frac{L_{se}}{EA_S}\right) \tag{5.8}$$

where

L_{be} = the effective length of the body (true body length plus one-half the thickness of the head of the bolt) (in., mm) (see Appendix F)

L_{se} = the effective length of the threads (length of exposed threads plus one-half the thickness of the nut) (in., mm) (see Appendix F)

ΔL_C = combined change in length of all portions (in., mm)

A_S = the effective stress area of the threads (see Chapter 3) (in.^{2}, mm^2) (see Appendix E)

A_B = the cross-sectional area of the body (in.^{2}, mm^2)

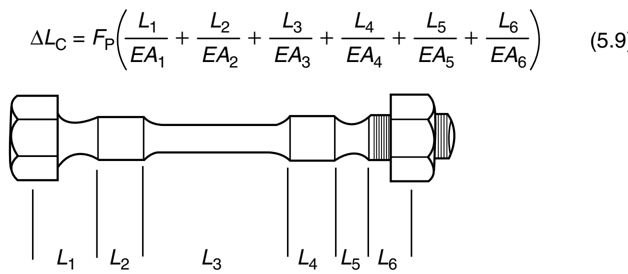


FIGURE 5.4 Each cross section of a complex fastener must be computed separately.

E = the modulus of elasticity (psi, N/mm^2) (Table 5.1)
F_P = tension in bolt (lb, N)

If the fastener has a more complex shape, as shown in Figure 5.4, then additional sections must be computed, but there is otherwise no change in the procedure. The change in length for the fastener shown in Figure 5.4, for example, would be

$$\Delta L_C = F_P\left(\frac{L_1}{EA_1} + \frac{L_2}{EA_2} + \frac{L_3}{EA_3} + \frac{L_4}{EA_4} + \frac{L_5}{EA_5} + \frac{L_6}{EA_6}\right) \tag{5.9}$$

5.2 BOLT STIFFNESS CALCULATIONS

5.2.1 Basic Concepts

Once we know how to compute the change in length of the fastener, we can also estimate the spring constant or stiffness, using the relationship

$$K_B = F_P/\Delta L_C \tag{5.10}$$

Before using this equation let me mention that Equation 5.12 gives us an alternative—and perhaps more convenient—way to estimate bolt stiffness. But for now, let's continue with Equation 5.10.

5.2.2 Example

Let's compute the stiffness and change in length of a 3/8–16 × 1/2 SAE, Grade 8 hex bolt (shown in Figure 5.5) in a joint having a 1 in. grip length (L_g). We can get the nominal dimensions we'll need by measuring a sample bolt, or, more safely, by referring to the pertinent specifications, or to the data in Appendices E and F. These tell us the following:

Specification	Dimension
SAE J 104 (nut)	Height of nut (T_N) = 0.3285 in.
SAE J 105 (bolt)	Height of head (T_H) = 0.2345 in.
ANSI Bl.1-1974 (thread)	Thread length (L_T) = 1.000 in.
	Tensile stress area of threads (A_S) = 0.0775 in.^{2}

From the description of the bolt, we already know, of course, that nominal body diameter (D) = 0.375 in. and nominal shank length (L) = 1.500 in.

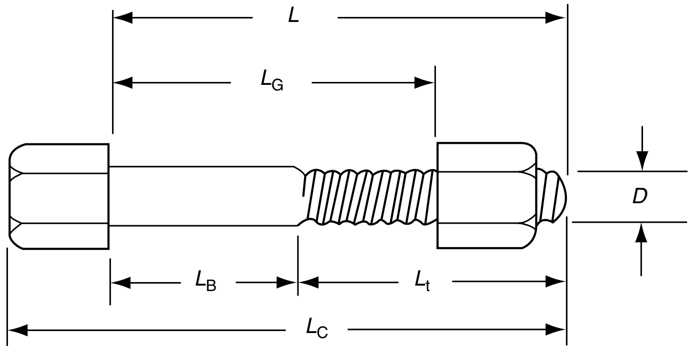


FIGURE 5.5 The bolt whose stiffness and elongation are computed in the text.

$$A_B = \frac{\pi D^2}{4} = \frac{\pi (0.375)^2}{4} = 0.1104 \text{ in.}^2$$

We must now compute the nominal cross-sectional area (A_B), the nominal length of the body (L_B),

$$L_B = L - L_t = 1.5 - 1.0 = 0.5 \text{ in.}$$

and the effective lengths of body and threaded sections, remembering that we're interested only in the threads that are actively engaged in carrying load:

$$L_{be} = L_B + T_H/2 = 0.5 + 0.1173 = 0.6173 \text{ in.}$$
$$L_{se} = L_G - L_B + T_N/2 = 1.0 - 0.5 + 0.1643 = 0.6643 \text{ in.}$$

We're now ready to compute the reciprocal of the spring constant of this bolt in this joint:

$$\frac{1}{K_B} = \frac{L_{be}}{EA_B} + \frac{L_{se}}{EA_s} = \frac{0.6173}{30 \times 10^6 \times 0.1104} + \frac{0.6643}{30 \times 10^6 \times 0.0775} = 0.4708 \times 10^{-6} \text{ in./lb}$$

Once we have this, we can compute the change in length for a given force. What force should we use? A typical tightening specification would be "60% of yield strength," which, for an SAE Grade 8 bolt, would be $0.6 \times 130{,}000$, or 78,000 psi. (You'll find yield strengths in Chapter 2.)

To convert this desired stress level to axial force, we multiply the stress value by the tensile stress area of the threads:

$$F_P = S_Y \times A_S = 78{,}000 \times 0.0775 = 6045 \text{ lbs}$$

We can now compute the change in length, which this tensile force would create in this bolt in this joint,

$$\Delta L_C = F_P\left(\frac{1}{K_B}\right) = 6045 \times 0.4708 \times 10^{-6} = 0.00285 \text{ in.}$$

and, for later reference, we note that

$$K_B = \frac{1}{0.4708 \times 10^{-6}} = 2.124 \times 10^6 \text{ lb/in.}$$

5.2.3 Actual versus Computed Stretch and Stiffness

The equations we have given are widely used, but the stretch and stiffness they predict can be quite inaccurate. The exact dimensions of a given bolt won't be the nominal dimensions in most cases, because of manufacturing variations. The modulus of elasticity will vary a little. Stress concentrations of the sort discussed in Chapter 3 can make a large difference in the actual relationship between applied force and change in length in a given bolt. Bending can also distort the relationship. These and other factors will be discussed at length in Chapter 9, where we discuss stretch measurement as a way to control preload.

5.2.4 Stiffness of Bolt–Nut–Washer System

So far we have considered only the stiffness of the bolt itself. The joint is never clamped by a bolt, however; it's clamped by a bolt-and-nut system—or by a bolt–nut–washer system. The stiffness of this combination of parts is found by

$$\frac{1}{K_T} = \frac{1}{K_B} + \frac{1}{K_N} + \frac{1}{K_W} \tag{5.11}$$

where

K_T = total stiffness of the system (lb/in., N/mm)
K_B = stiffness of the bolt (lb/in., N/mm)
K_N = stiffness of the nut (lb/in., N/mm)
K_w = stiffness of washer (lb/in., N/mm)

As we'll see in Chapter 11 when we examine the nonlinear behavior of a joint, the fact that the joint is clamped by a bolt–nut–washer system, instead of by the bolt alone, makes a big difference. The stiffness of the system, for example, is only about half the stiffness of the bolt alone [6]. Interactions between parts also make the behavior—including the apparent stiffness—drastically nonlinear, especially at low load levels.

None of this is taken into consideration in classical joint design, however, which assumes linear, elastic behavior, and which assumes that the stiffness of the clamping element will be that of the bolt alone. We'll know better by the time we study Chapter 11.

Before leaving the subject of washers we should note that the washer can have a significant impact on the stiffness of the joint. A large-diameter, heavy washer will allow a bolt to apply clamping force to more joint material than will a light washer—or no washer. The more joint material involved, the more force it takes to compress the joint by a given amount; i.e., the joint becomes a stiffer spring. As we'll see in later chapters, this has many implications for the design and behavior of the bolted joint.

5.2.5 Alternative Expression for Bolt Stiffness

Although the procedure we've used to compute bolt stiffness is correct and emphasizes the all-important deflection or change in length of the bolt, it's cumbersome. If we're interested in the stiffness of a conventional bolt we can often use a simpler expression as follows. We start, again, with Equation 5.10.

$$K_B = \frac{F_P}{\Delta L}$$

But now we use Hooke's law to eliminate the F term.

Hooke's law

$$E = (F_P/A_S)/(\Delta L/L_e)$$

Rewriting it,

$$F_P = \frac{EA_s\Delta L}{L}$$

Substituting this expression for F_p in Equation 5.10 gives us

$$K_B = \frac{EA_s}{L_e} \tag{5.12}$$

where
E = modulus of elasticity of the bolt (psi, GPa)
A_S = tensile stress area of the bolt (in.^{2}, mm^2)
L_e = effective length of the bolt (in., mm)

This equation assumes that the stiffness of the body of the bolt is the same as the stiffness of the threaded section—or that the bolt is fully threaded. If neither is true, Equation 5.12 is not as accurate as Equation 5.10. Nevertheless, it's widely used and gives us a convenient way to approximate K_B. When we compute D the stiffness of the bolt shown in Figure 5.5 we used the following equation:

$$\frac{1}{K_B} = \frac{L_{be}}{EA_B} + \frac{L_{se}}{EA_s}$$

As you can see, this is an extended version of Equation 5.12. Like Equation 5.12, furthermore, it doesn't require us to compute the change in length of the bolt. Although a little more complicated than Equation 5.12, it's still easy to use and gives us a far more accurate estimate of K_B than does the more common Equation 5.12.

5.2.6 Energy Stored in the Bolt

Lest we forget!—we're interested in the bolt as an energy storage device. It can create that all-important clamping force on the joint only as long as it retains potential energy. Although we'll always be interested in whether or not our bolts are "good" or "bad" energy storage devices, we won't often have to compute the exact amount of that energy. If we do want to compute it, we use the following expression:

$$\mathrm{PE}_B = \Delta L \times F_B/2 \tag{5.13}$$

where
F_B = tension in the bolt (lb, N)
ΔL = deflection in the bolt (in., mm)
PE_B = the potential energy stored in the bolt (in.-lb, mm-N)

Note that if we were to plot the tension in the bolt versus its deflection as we tightened the nut, we'd generate a curve like that shown in Figure 5.6: a straight line as long as the bolt deforms elastically, curving over at the top if we tighten it so much that it yields (deforms plastically).

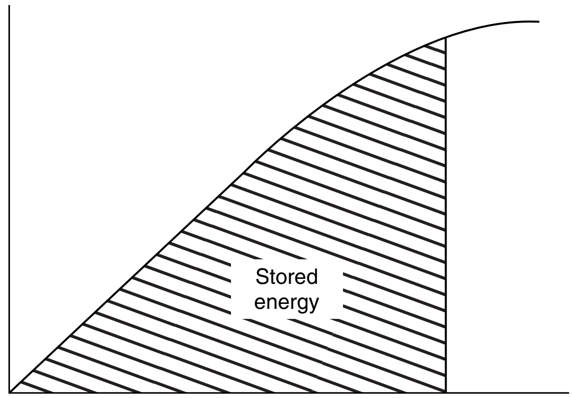


FIGURE 5.6 A plot of the tension or preload in a bolt versus its deflection. The curve will be a straight line as long as the bolt deforms elastically. When it yields, the line becomes a curve as shown. The energy stored in the bolt is equal to the area under the curve.

Equation 5.13 is good only for the straight-line portion of the curve; it defines the area under that curve. We'd have to use graphical techniques, calculus, or a computer to estimate the energy stored in a bolt that had deformed plastically—but this would still be equal to the area under the F_B–AL curve.

5.3 THE JOINT

5.3.1 Basic Concepts

We can also treat joint members as springs in series when we compute joint stiffness and deflection. The loads, of course, are compressive rather than tensile, but the basic equations are the same. For example, if we apply equal and opposite forces to a pair of blocks as shown in Figure 5.7, the change in thickness (ΔT_3) of the system of blocks and the spring constant (K_3) will be

$$K_J = \frac{F}{\Delta T_J} \tag{5.15}$$

where

$$\frac{1}{K} = \frac{1}{K_1} + \frac{1}{K_2} = \frac{T_2}{EA_1} + \frac{T_2}{EA_2} \tag{5.16}$$

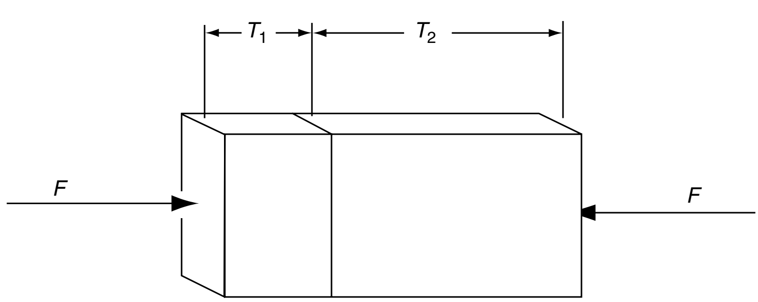


FIGURE 5.7 Two blocks in compression.

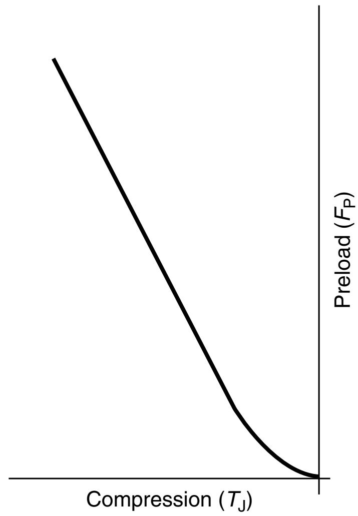


FIGURE 5.8 The deflection (T_J) of joint members can be nonlinear levels (low F_P).

$$\Delta T_J = \Delta T_1 + \Delta T_2 = F\left(\frac{T_1}{EA_1} + \frac{T_2}{EA_2}\right) \tag{5.14}$$

and

$$K_J = \frac{F}{\Delta T_J} \tag{5.15}$$

where

$$\frac{1}{K} = \frac{1}{K_1} + \frac{1}{K_2} = \frac{T_2}{EA_1} + \frac{T_2}{EA_2} \tag{5.16}$$

Theoretically, the relationship between applied compressive force and deflection for our pair of blocks should be linear as long as the force stays within the elastic limit of the material. In practice, however, we will often find that the stiffness of a joint is not linear and may not be fully elastic. Some report the preload–compression relationship shown in Figure 5.8 [1]. Others report a variety of nonlinear effects. We'll look at some of these in Chapter 13. Before we consider these complexities, however, it is useful to review the "classical" theories, which have been used to evaluate joint behavior in the past. Although simplified, they are often used as a basis for more complex theories. They're also good enough for many applications. So let's take a look at them now.

5.3.2 Computing Joint Stiffness

We assumed a simplified, equivalent body shape for a bolt, to make routine calculations of stiffness and deflection less complicated. We have to do the same sort of thing for the joint.

That portion of the joint which is put in compressive stress by the bolt can be described as a barrel with a hole through the middle, as suggested in Figure 3.3. Some workers, therefore,

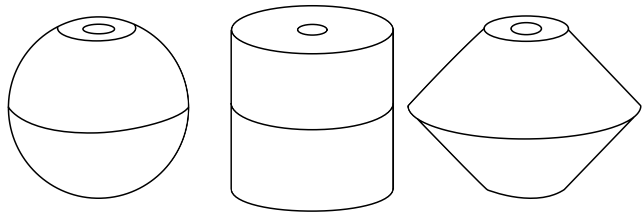


FIGURE 5.9 Equivalent shapes, substituted for joint members in calculating joint stiffness and deformation.

have substituted an "equivalent barrel" for the joint [4], but more common substitutions are hollow cylinders [1] or a pair of frustum cones [5] as in Figure 5.9.

5.3.2.1 Stiffness of Concentric Joints

A discussion of eight proposed ways to estimate the stiffness of a hard (non-gasketed) joint is given by Motosh [4]. The equivalent cylinder approach is described at length by Meyer and Strelow [1]. Unfortunately, each of these techniques assumes that

1. Joint behavior will be linear and fully elastic.
2. There is only one bolt in the joint and it passes through the center of the members being clamped together (this is called a "concentric" joint).
3. The external load applied to the joint is a tension load and it is applied along a line that's concentric with the bolt axis.

Your own experience, I'm sure, will tell you that limitations 2 and 3 are substantial ones and mean that these equations and recommendations may not apply—at least not very accurately—to many of the joints with which we will be dealing. They're our only choice at the present state of the art, however, except as noted below. At least they're our main "theoretical" choice. If the approximations they give us aren't good enough, we have to determine joint stiffness experimentally.

We will use the equivalent cylinder approach, in this book, to estimate stiffness. This involves the general equation

$$K_J = \frac{EA_C}{T} \tag{5.17}$$

where
K_J = stiffness of joint (lb/in., N/mm)
E = modulus of elasticity (psi, MPa)
A_C = cross-sectional area of the equivalent cylinder used to represent the joint in stiffness calculations (in.^{2}, mm^2)
T = total thickness of joint or grip length (in., mm)

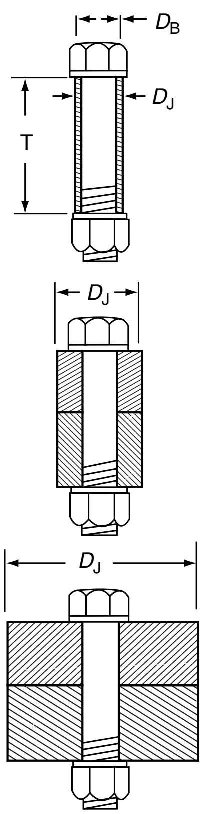
Note the similarity of this equation to Equation 5.12. The big difficulty here is A_C, the cross-sectional area of the equivalent cylinder. The equations we'll use for A_C are summarized in Figure 5.10. Note that there are three different equations, depending on the diameter of contact (D_B) between the bolt head (or washer) and the joint, and its relationship to the outside diameter of the joint (D_J) [1,7]. If the joint has a square or rectangular cross section, its diameter is the length of one side (or of the shortest side of the rectangle). D_H is the diameter of the hole.

5.3.2.2 Stiffness of Eccentric Joints

Most bolts don't run through the centerline of the joint or external tension loads don't align themselves with bolt axes. If the bolt, load, or both lie away from the joint centerline, the joint is called "eccentric" and our choice of stiffness equations is diminished still further. The German engineering society, Verein Deutscher Ingenieure (VDI), however, has published equations that can be used to estimate the stiffness of eccentric joints as long as the cross-sectional area of that portion of a joint which is loaded by one bolt is not much larger than the contact area between bolt (or nut or washer) and joint [7]. With reference to Figure 5.11, the area we assume to be loaded by the bolt is A_J. The stiffness equations which follow assume that

$$A_J = b \times W \quad \text{if} \quad W \leq (D_B + T_{\min}) \tag{5.18a}$$

$$A_J = b \times (D_B + T_{\min}) \quad \text{if} \quad W > (Z > u + T_{\min}) \tag{5.18b}$$



if $D_B \geq D_J$

then
$$A_C = \frac{\pi}{4}(D_J^2 - D_H^2) \tag{5.19}$$

if $D_B < D_J \leq 3\,D_B$

and $T \leq 8\,D$

then
$$A_C = \frac{\pi}{4}(D_B^2 - D_H^2) + \frac{\pi}{o}\left(\frac{D_J}{D_B} - l\right)\left(\frac{D_B T}{5} + \frac{T^2}{100}\right) \tag{5.20}$$

if $D_J > 3D_B$; and $T \leq 8D$

then
$$A_C = \frac{\pi}{4}\left[\left(D_B + \frac{T}{10}\right)^2 - D_H^2\right] \tag{5.21}$$

FIGURE 5.10 Equations used to compute the stiffness of concentric joints using the equivalent cylinder method. We'll call this stiffness K_{ic}.

$$A_J = b \times (D_B + T_{min}) \quad \text{if} \quad W > (D_B + T_{min}) \tag{5.18a}$$

$$A_J = b \times W \quad \text{if} \quad W \leqq (D_B + T_{min}) \tag{5.18b}$$

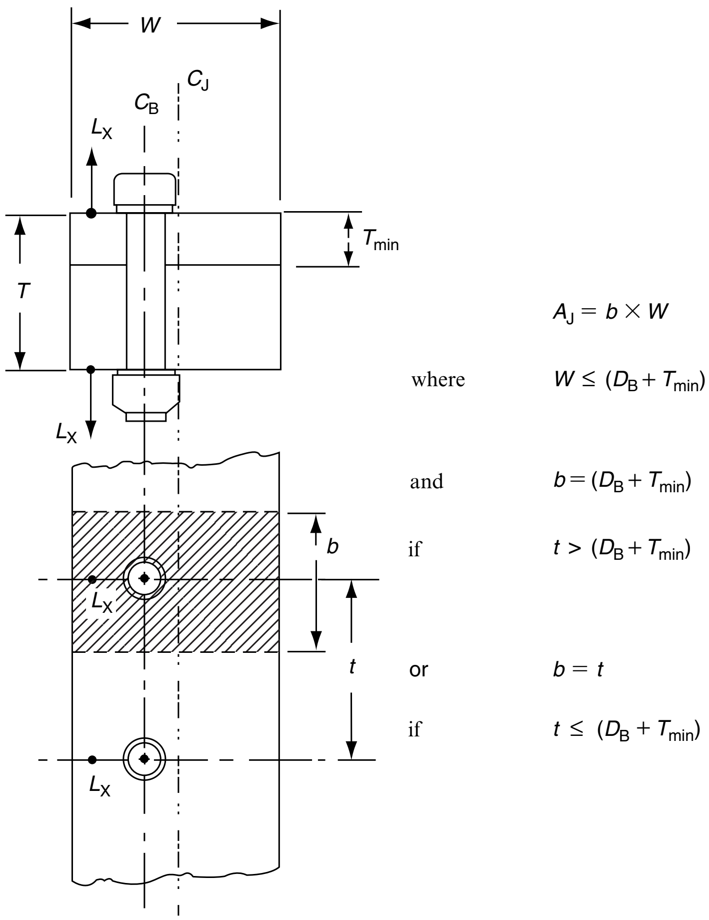


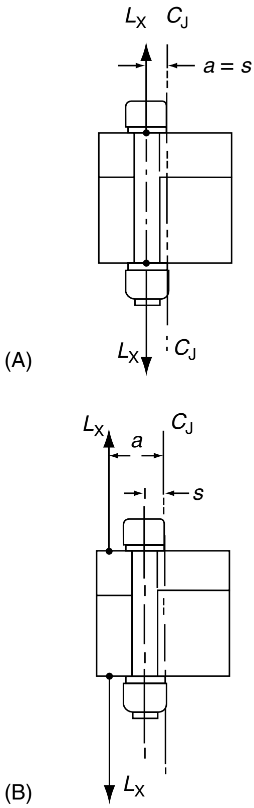
FIGURE 5.11 Sketch of an eccentric joint. The shaded are, $b \times W$, can be considered that portion of the joint interface which is loaded by a single bolt. See text for the equations used to estimate this area, A_J.

In each case

$$b = t \quad \text{if} \quad t \leq (D_B + T_{min})$$
$$b = (D_B + T_{min}) \quad \text{if} \quad t > (D_B + T_{min})$$

where
W, t, b, and T_{min} are illustrated in Figure 5.11 (all in in., mm)
D_B = diameter of contact between bolt head (or washer) and the joint (in., mm)
D_H = diameter of the bolt hole (in., mm)

If joint dimensions exceed the limits suggested above (for W), the equations given in Figure 5.12 don't apply. If the joint satisfies the limitations, then its stiffness may be estimated from the equations given in Figure 5.12,



$$K_J = \frac{1}{r'_J} \tag{5.22}$$

where

$$r'_J = \frac{1}{K_{Jc}}\left(1 + \frac{s^2 A_C}{{R_G}^2 A_J}\right)$$

$$K_J = \frac{1}{r''_J} \tag{5.23}$$

where

$$r''_J = \frac{1}{K_{Jc}}\left(1 + \frac{s\,a\,A_C}{{R_G}^2 A_J}\right)$$

FIGURE 5.12 Equations used to compute the stiffness of eccentric joints when the line of action of the external load (L_X) coincides with the bolt axis (A) and when it does not (B).

where
- C_J = centerline of joint
- L_X = external load (lb, N)
- A = distance between external load and joint centerline (in., mm)
- s = distance between bolt axis and joint centerline (in., mm)
- A_C = cross-sectional area of equivalent concentric cylinder (see Figure 5.10) (in.^{2}, mm^2)
- k_{jc} = stiffness of equivalent concentric cylinder (see Figure 5.10) (lb/in., N/mm)
- K_J = stiffness of eccentric joint (lb/in., N/mm)
- r'_t = resilience of eccentric joint when load and bolt are coaxial (in./lb, mm/N)
- r''_t = resilience of eccentric joint when load and bolt fall along different axes (in./lb, mm/N)
- R_G = radius of gyration of joint area A_B (in., mm)
- A_J = cross-sectional area of joint (see Figure 5.11) (in.^{2}, mm^2)

For reference, the radius of gyration for a square cross section is [8]

$$R_G = 0.289d \tag{5.24}$$

where d = length of one side.

For a rectangular cross section it is

$$R_G = 0.289d \tag{5.25}$$

where $d =$ length of the longer side.

For a circular cross section it is

$$R_G = 0.25d \tag{5.26}$$

where $d =$ diameter of the circle.

5.3.3 Stiffness in Practice

Experience shows that the stiffness of a "typical" joint (whatever that may be) is about five times the stiffness of the bolt that would be used in such a joint. Very thin joints—sheet metal and the like—will be substantially stiffer, although the stiffness of the bolt will also increase rapidly as it gets shorter, as suggested in Figure 5.13. In this figure, incidentally, we have used the equivalent cylinder approach to estimate the possible stiffness of a concentric, hard joint.

5.3.3.1 A Quick Way to Estimate the Stiffness of Non-Gasketed Steel Joints

Here's another way to use Motosh and VDI data to estimate the stiffness of a non-gasketed steel joint. Both sources have published charts on which are plotted the joint-to-bolt stiffness ratio (K_J/K_B) as a function of the bolt's slenderness ratio L/D, where $L =$ the effective length of the bolt and $D =$ nominal diameter.

Figure 5.14 shows a combined version of the published data for slenderness ratios varying from 1:1 to 16:1. The straight line represents the Motosh data; the curved line is from VDI. As you can see, they're in good agreement above a slenderness ratio of about 4:1.

Figure 5.15 shows a similar plot for thinner joints, with L/D ratios of 1.2:1 or less [9]. Projections of the lower end of the VDI and Motosh curves are also shown in Figure 5.15, showing that the agreement in the data for thin joints is less than perfect. Nevertheless, for any slenderness ratio, this is the best information I'm aware of.

These curves can be used to estimate joint stiffness as follows:

1. Use Equation 5.10 or 5.12 or a version thereof to compute the stiffness of your bolts (K_B).
2. Compute the L/D ratio of your bolt, using the effective length ($L_{be} + L_{se}$) for L.
3. Use Figures 5.14 or 5.15 with your L/D ratio and find the corresponding K_J/K_B stiffness ratio.
4. Multiply the K_B computed in step 1 by the K_J/K_B ratio to estimate K_J.

Note that the data in Figures 5.14 and 5.15 is good only for steel bolts used in non-gasketed steel joints. If your joint is made of something else, complete the above steps and then modify the estimate of stiffness as follows.

$$K_J{}' = K_J \frac{E_m}{30 \times 10^6} \tag{5.27}$$

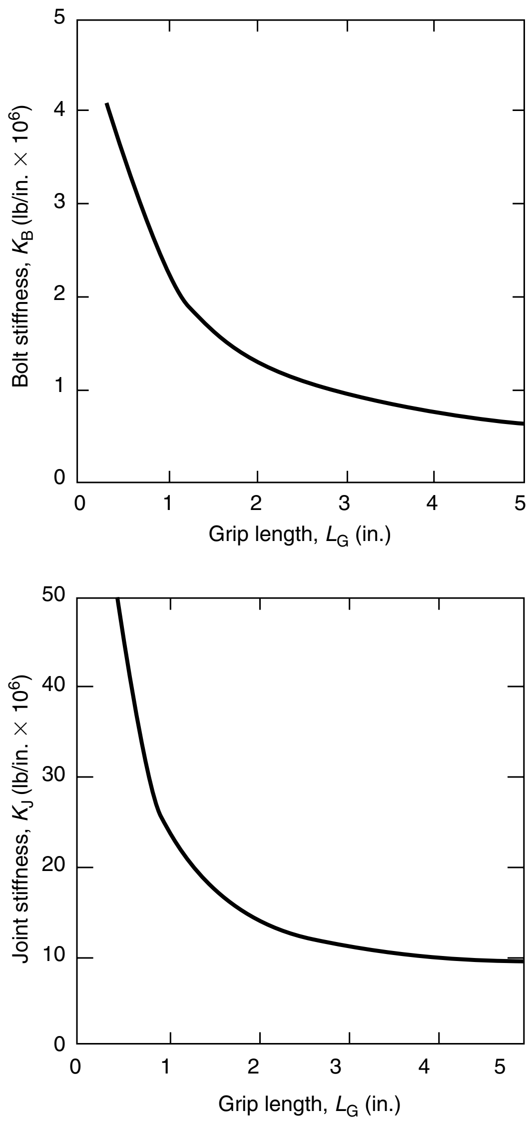


FIGURE 5.13 Stiffness of 3/8–16 bolts (K_B) and of the joints (K_J) they must be used in. Bolt hole to joint edge distance has been assumed constant at 1.25 in. Note 10:1 difference in vertical scale.

where

K_J = stiffness of a steel joint as estimated from the procedure above (lb/in., N/mm)
K_J' = stiffness of the same joint, but made from an alternate material (lb/in., N/mm)
E_m = modulus of elasticity of the alternate material (psi, MPa)
E_{steel} = modulus of the steel joint material (psi, MPa)

5.4 GASKETED JOINTS

We've analyzed both the bolt and the joint as groups of springs in series. In such an arrangement, if the stiffness of one spring is substantially less than the stiffness of the others,

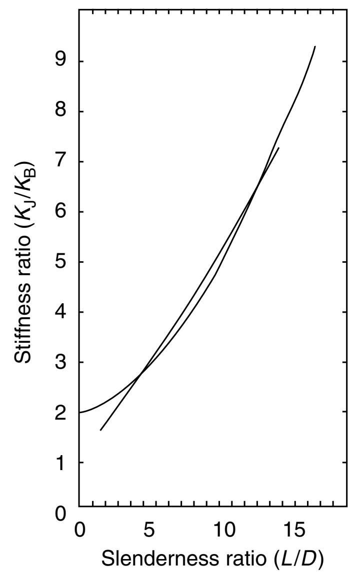


FIGURE 5.14 Plots of experimentally determined joint-to-bolt stiffness ratio.

the "soft" one will dominate the behavior of the group. Minor changes in the stiffness of springs A and C in Figure 5.16, for example, won't have much influence on the overall deflection of the train of springs under applied load F. By the same token, changes in the applied force will create a much larger change in the deflection of spring B than it will in the deflection of A or C.

Gaskets are relatively soft bodies compared to other joint members; they have to be in order to do their job of plugging leak paths. As a result, the stiffness of a gasketed joint is essentially equal to the stiffness of the gasket. This creates fatigue and other problems, as we'll

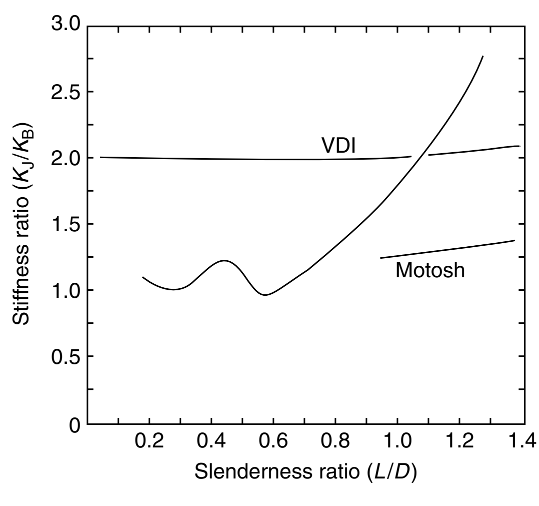


FIGURE 5.15 Plot of stiffness ratio versus slenderness ratio for thin joints. These data, like those shown in Figure 5.14 can be used to estimate the stiffness of non-gasketed joints, as explained in the text. The lower ends of the VDI and Motosh curves of Figure 5.14 are repeated here for comparison.

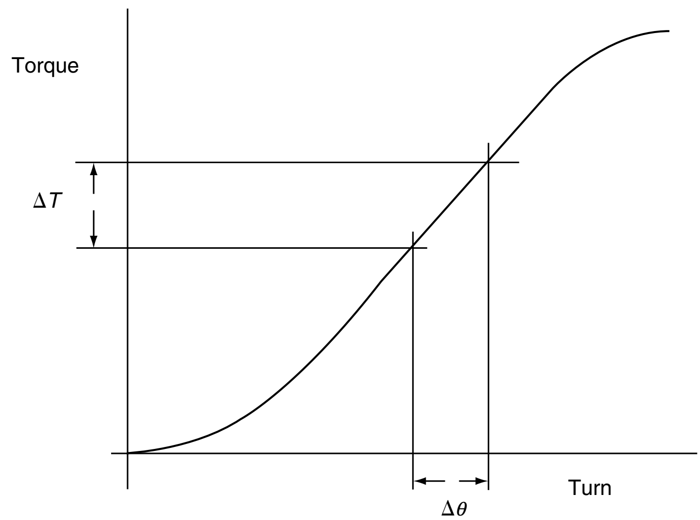


FIGURE 5.16 This curve shows the typical relationship between applied torque and the turn of the nut when a bolt is tightened (see Chapter 8). The middle portion of the curve is usually an approximately straight line. A change in torque (ΔT) along this straight line, and the corresponding change in angle ($\Delta\theta$), can be used to estimate the stiffness of the joint, as explained in the text.

see in later chapters. And this is true, incidentally, even if the deformation of the gasket is basically plastic, rather than elastic, as is often the case.

The force-deflection behavior of a gasket is strongly nonlinear and irreversible (the gasket exhibits hysteresis and creep). Its stiffness, therefore, varies as it is loaded and then unloaded. In estimating joint behavior we would be interested in the gasket's behavior as it is unloaded (by pressurizing the vessel, for example, and partially relieving the joint. All of this complicates the analysis of gasketed joints. You'll find a complete discussion in Volume 2 of this text.

5.5 AN ALTERNATE WAY TO COMPUTE JOINT STIFFNESS

Shoberg and Nassar [11] have shown that the stiffness of the joint and the stiffness ratio can be determined in an experiment that measures the torque applied to the nut and the angle through which the nut turns as it is being tightened through the straight-line portion of the torque–turn curve shown in Figure 5.10, 6.3, or 8.1. The equation they have derived is

$$K_J = (\Delta T/\Delta\theta)/[(KDPK_B/360) - (\Delta T/\Delta\theta)] \tag{5.28}$$

where

P = pitch of the threads (in., mm)
ΔT = increase in the torque applied to the nut (lb-in., N-m)
$\Delta\theta$ = resulting increase in turn of the nut (degrees)
K = nut factor defining the torque to preload relationship (see Chapter 7)
D = nominal diameter of the fastener (in., mm)
K_B = stiffness of the bolt (lb/in., N/mm)
K_J = stiffness of the joint (lb/in., N/mm)

I'll give the derivation of this equation in Chapter 8, where we'll take a close look at the torque–turn behavior of the joint during assembly.

5.6 JOINT STIFFNESS RATIO OR LOAD FACTOR

Now that we know how to compute or estimate the stiffness of the bolts and of the joint members we're ready to use this data to compute an important design factor called the "joint stiffness ratio" of the bolted joint. Note carefully that this is not simply the bolt-to-joint stiffness ratio; it's more complicated than that. In this book, and in much of the bolting literature, this ratio is expressed in terms of the stiffness of the joint elements, or

$$\Phi_K = \frac{K_B}{K_B + K_J} \tag{5.29}$$

where
Φ_K = joint stiffness ratio or load factor (a dimensionless constant)
K_B = stiffness of the bolt (lb/in., N/mm)
K_J = stiffness of the joint material around a bolt (lb/in., N/mm)

In other places, for example, in the German VDI Directive 2230 [10], the stiffness ratio, called a "load factor," is expressed in terms of the resilience of the parts. Resilience is the reciprocal of stiffness (i.e., $r = 1/K$), so

$$\Phi_K = \frac{r_j}{r_s + r_j} \tag{5.30}$$

where
Φ_K = joint stiffness ratio or load factor (a dimensionless constant)
r_s = resilience of the bolt (in./lb, mm/N)
r_j = resilience of the joint (in./lb, mm/N)

We'll see how to use this ratio or load factor in Chapter 10 and in the various chapters devoted to the design of the joint.

5.7 STIFFNESS—SOME DESIGN GOALS

5.7.1 Energy Stored in the Joint Members

Once again we could plot a curve showing the relationship between the deflection of the joint and the clamping force on it, as in Figure 5.17; and once again the energy stored in this "spring" is equal to the area under the curve. Theoretically that curve will consist of a straight line with a curve at the upper end if the joint members—or gasket—start to deform plastically. As suggested in Figure 5.8 and as we'll see in more detail in Chapters 11 and 16, this picture is a very much simplified illustration of the actual behavior of the joint spring, especially if this is a gasketed joint; but Figure 5.17 can be taken as a first approximation and is used in most cases to analyze the behavior of a bolted joint. For more information about the design of gasketed joints see reference [12] or Volume II of this text. We'll use it, for example, in Chapter 11 when we first look at the response of a bolted joint to tensile loads, and in Chapter 18 when we learn how to design a non-gasketed joint.

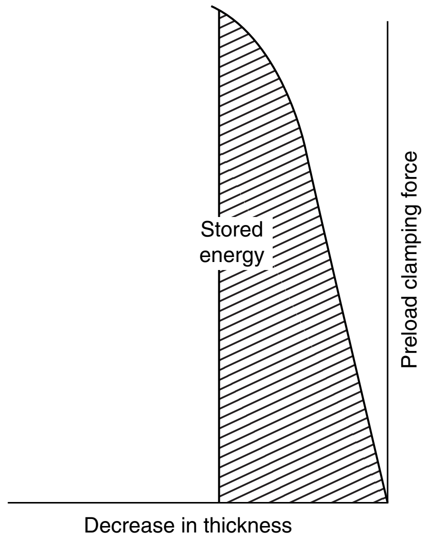


FIGURE 5.17 A plot of the deflection of the joint members as a function of the clamping force exerted on the joint by the bolts. The plot shown here assumes that joint deflection is linear as clamping force starts to build up, then becomes nonlinear when the joint yields and starts to deform plastically. This assumption is often used for design purposes but doesn't reflect the fact that the true deformation curve of the joint is nonlinear throughout, especially if the joint contains a gasket.

So now—in Figures 5.6 and 5.17—we have pictures of the energy stored in both bolts and joint members. How do we use this information? The answer: we use it only to remind ourselves that stiffness is an important design consideration—because it affects the amount of energy stored in the bolt–joint system. The more energy we can store there, the more abuse the joint will be able to withstand before it fails. And stiffness is the key to the "amount stored." We won't have to measure or compute the energy itself; we'll spend our time worrying about, computing, and manipulating stiffness. Let's take a look.

5.7.2 Relationship between Stiffness and Stored Energy

Take another look at Figure 5.6. Let's replace the bolt theoretically illustrated there with a bolt having a larger diameter. Let's tighten the new bolt to the same preload as we tightened the old one. Because the larger bolt is stiffer than the old one, it will deflect less than the old did. Its preload-deflection curve will be steeper—"more vertical" if you will see. This means that the area under the curve, estimated by Equation 5.13, will be less. And this means that the same loss of deflection will mean a greater loss of preload in the new, fatter bolt than in the original one. The same amount of thermal change or vibration loosening or relaxation will cause a greater loss of preload in the bolt that has stored less energy.

Alternately, of course, a bolt of smaller diameter taken to the same preload would be less sensitive to the changes mentioned above. If we couldn't take a thinner bolt to this preload we might achieve the same effect by using two thin bolts in place of the original fat one, or by using a longer bolt of the original diameter. More length means less stiffness as suggested by Equation 5.12.

As an example, consider two 3⁄8–20 SAE Grade 5 bolts tightened to proof load (85 ksi). Same applied torque; same preload (7106 lbs in these bolts having a tensile stress area of 0.0836 in.^{2}). But one bolt is ½ in. long, the other ¾ in.—and they're used in joints of those thicknesses. As we'll see in Table 9.2, these bolts will have been stretched approximately 0.003 in. per inch of grip length, when taken to proof load (just under yield). So the short bolt has stretched 0.0015 in., and the longer bolt 0.0023 in. If any of the various relaxation effects reduces the deflection of the shorter bolt by 0.001 in. its preload would drop to 33% of proof load, a residual clamping force of only 2345 lbs. The same 0.001 in. reduction in the stretch of the longer bolt would drop its preload to 56% of proof; it retains 3979 lbs of clamping force.

The bottom line is this: we almost always want "less stiff" bolts if we have a choice. By this means we hope to avoid the chronic problems often associated with short, stubby fasteners—such as sheet metal screws—which loosen so readily in service.

What about the joint? Do we want it to be as resilient as possible? Probably not. First of all, the bolt, being less stiff, will almost always store much more energy than the joint. You might think of the bolt as the active element in the system, with the joint as the passive or resistive element. Second, and more important, we also want a "good" stiffness ratio in our joints.

5.7.3 Stiffness Ratio

We can't go into it at this point, but as we'll see in Chapter 11 and in the chapters devoted to joint design, we usually want a low stiffness ratio. The lower this ratio the less the clamping force will be affected by external loads, by thermal change, by vibration, by fatigue, etc. So we'll often try to minimize the stiffness ratio. Fortunately, it helps to minimize the bolt stiffness at the same time; for once we're not trying to achieve conflicting design goals.

EXERCISES AND PROBLEMS

1. Compute the change in length of a medium carbon steel rod of uniform cross section, 0.5 in. diameter, and 4 in. long, when subjected to a tensile force of 10,000 lbs (refer Equation 5.2 and Table 5.1).
2. Compute the change in length of a medium carbon steel rod 4 in. long, subjected to a force of 10,000 lbs. The rod has three cross sections: The first 0.5 in. long, 0.95 in. in diameter; the second 3 in. long, 0.5 in. in diameter; the third 0.5 in. long, 1.2 in. in diameter. The forces are applied to the outer ends of the rod as in Figure 5.1 (refer Equation 5.3).
3. Using data in Chapter 2 and Appendices E and F, compute the change of length of a ¾–10 × 5 medium carbon ASTM A490 bolt tightened to its yield strength in a joint where the grip length is 4.25 in. (refer Table 5.1, Figure 5.3, and Section 2.5.2).
4. Compute the stiffness of the bolt used in example 3 above using Equations 5.10 and 5.12. If the results obtained with these two equations differ, which is more accurate and why?
5. What is the resilience of the example 3 bolt?
6. Assume that the example 3 bolt is to be used to clamp medium carbon steel joint members together. Use Equations 5.17 and 5.21 to estimate the stiffness of the bolt, the stiffness of the joint, and the stiffness ratio for this assembly.
7. Estimate the safe stiffness ratio using Figure 5.14 or Figure 5.15 (refer Equation 5.7a).
8. What will the stiffness ratio be if that bolt is used in a cast iron joint (refer Table 5.1 Equation 5.27)?

REFERENCES

1. Meyer, G. and D. Strelow, Simple diagrams aid in analyzing forces in bolted joints, *Assembly Engineering*, January 1972.
2. *Fastener Preload Concepts*, Contract no. F33615-76-C-5251, for Air Force Materials Laboratory, Air Force Systems Command, WPAFB, Dayton, Ohio, by General Dynamics, Fort Worth, June 15, 1978.
3. Sawa, T. and K. Maruyama, On the deformation of the bolt head and nut in a bolted joint, *Bulletin of the JSME*, vol. 19, no. 128, February 1976.
4. Motosh, N., Determination of joint stiffness in bolted connections, *Transactions of the ASME*. August 1976.
5. Osgood, C.C., How elasticity influences bolted joint design, *Machine Design*, pp. 92ff, February 24, 1972.
6. Pindera, J.T. and Y. Sze, Influence of the bolt system on the response of the face-to-face flanged connections, Reprints of the 2nd International Conference on Structural Mechanics in Reactor Technology, Berlin, Germany, September 10–14, 1973, vol. Ill, part G-H.
7. Junker, G.H., Principle of the Calculation of High-Duty Bolted Joints: Interpretation of Directive VDI 2230, Unbrako technical thesis, published by SPS, Jenkintown, PA, p. 7.
8. Oberg, E., F.D. Jones, and H.L. Horton, *Machinery's Handbook*, 20th ed., Industrial Press, New York, 1975, p. 378.
9. Osgood, C.C., *Fatigue Design*, Wiley-Interscience, New York, 1970, pp. 196–197.
10. Junker, G.H., Principle of the Calculation of High-Duty Bolted Joints: Interpretation of VDI Directive 2230, SPS, Jenkintown, PA, date unknown.
11. Shoberg, R.S., *Engineering Fundamentals of Torque–Turn Tightening*, R.S. Technologies, Ltd., Farmington Hills, MI, 1991 or 1992.
12. Nau, B.S., On the design of gasketed joint, 12th International Conference on Fluid Sealing, Brighton, U.K., May 10–12, 1989.
13. ASME Boiler and Pressure Vessel Code, Section III-Div. 1, 1983 ed., Tables 1–6.0 and 1–7.3, ASME, New York, 1983.
14. *1984 SAE Handbook*, Vol. 1, Materials, SAE, Warrendale, PA, pp. 10.137–10.139, 1984.
15. ASME Boiler and Pressure Vessel Code, Section VIII, Part UF, Table UF-27, ASME, New York, 1983.
16. *Handbook of Chemistry and Physics*, 27th ed., Chemical Rubber and Publishing Co., Cleveland, OH 1943, p. 1631.
17. SAE Standard J1199, SAE, Warrendale, PA.
18. SAE Standard J158, SAE, Warrendale, PA.

6 Introduction to Assembly

That all-important clamping force which holds the joint together—and without which there would be no joint—is not created by a good joint designer, or by high-quality parts. It is created by the mechanic on the assembly line or job site, using the tools, procedures, and working conditions we have provided him with. The force is brought into being as energy in the mechanic or power tool is converted to potential energy stored in the joint and bolt members. The correct amount of force cannot be created if the design is faulty or the parts don't fit together properly or they break; but getting all this right, while necessary, is not enough. The final, essential "creator" of the force is the mechanic, and the time of creation is during assembly. So it's very important for us to understand the assembly process.

Because of this, the next four chapters will be devoted to assembly, starting, in this chapter, with an overview of the process. How do the bolts and the joint members respond as we tighten the bolts? We'll see that the behavior of the parts, during assembly, is complex. We'll take a close look at several unseen, difficult to detect, difficult to quantify factors which can have a significant impact on the results, on the amount of clamping force developed in the joint. This in turn will teach us that it isn't easy to control the buildup of clamping force in the joint and that those mechanics need all the help we can give them, both as product designers and as production engineers.

In the following three chapters, we'll look at the many options we have for control of the bolt-tightening process, starting with relatively simple, crude methods and proceeding on to ever more elaborate and accurate ones. Our knowledge of the behavior of the bolts and joint during assembly will help us evaluate the merits of these options.

In still later chapters, we'll learn why accurate control of the clamping force is so necessary, how too much or too little clamp can degrade the behavior and life of the joint in service. As we learned in Chapter 1, if the bolts and joint members don't contain the correct amount of stored energy and therefore create the correct amount of clamping force, we'll have joint problems. In other words, proper assembly is essential.

6.1 INITIAL VERSUS RESIDUAL PRELOAD

The clamping force a bolt exerts on the joint is usually called or equated to the so-called preload in the bolt. This term is used in general in most of the literature on bolting to describe the tension in the bolt at any time, but this, in my opinion, is a mistake. I like to think of the preload created in an individual fastener when it is first tightened as "initial" preload, even though that term may be redundant. As you'll see, the effects we're about to discuss will frequently modify this preload as the fastener relaxes or as we tighten other fasteners in the joint. I call the final preload in the bolts the "residual" preload.

When the joint is put into service, a variety of things can act to modify the preload in individual fasteners still further. This could be called "in-service tension in the bolts."

Each of these preloads or tensions is directly proportional to the amount of potential energy stored in the bolt, as it is first tightened, or after relaxation occurs, or in service.

In most cases these preloads or tensions will also be directly proportional to the clamping force between joint members; but there are exceptions, as we'll see.

But—now that we have these definitions under our belt—let's get on with it. What happens during assembly?

6.2 STARTING THE ASSEMBLY PROCESS

We're going to assemble a hypothetical joint, using as our example a round, gasketed, pipe flange joint held together by 16, ⅛–8, ASTM A193 B7 bolts (see Figure 6.1). The large diameter and the presence of a gasket make this assembly a little more difficult than most, but will therefore allow us to look at a more complete range of assembly problems than would a simpler example. Most of the discussion would apply to joints in general.

We're also going to measure the torque we apply to the nuts to control the buildup of initial preload in these bolts. This is the most common, and one of the simplest, types of control. It will be the subject of Chapter 7, so we won't go into a lot of detail here about it's pros and cons. We'll just use it for now.

6.2.1 ASSEMBLING THE PARTS

We start by roughly aligning the flanges so that we can insert the bolts by hand. When we finish pushing and pulling on the flanges, their mating surfaces are not exactly parallel and the holes aren't aligned perfectly; so we have to tap a few of the bolts with a hammer to get them through their holes, and some of them stick out a little farther, on the nut end, than do others. Now we're going to apply a preliminary "snugging torque" to run the nuts down and pull the flanges together.

6.2.2 TIGHTENING THE FIRST BOLT

To load the joint (and gasket) evenly, we'll apply the snugging torque in a cross or star pattern, as shown in Figure 6.2. We'd use a similar pattern on a square or rectangular joint if the bolts were all around the edge. In a rectangular, structural joint pattern, with several rows of bolts, we'd start snugging at the center of the bolt pattern and work our way out to the free edges.

We'll use 225 lb-ft of torque for this first, snugging pass. This is about a third of the final torque we're planning to use, and we'll follow it with a second pass at two-thirds of final torque, and then with a third and final pass at full torque. In a structural steel joint, we would follow the snugging pass with a second (last) pass at the final torque. Note that in each case

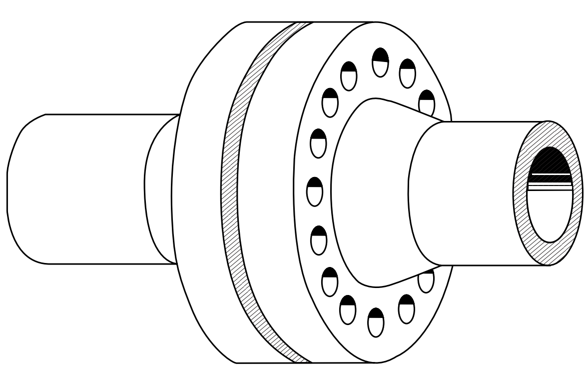


FIGURE 6.1 This is a sketch of the large-diameter, pressure vessel joint used as an example in this chapter. We see what happens when we install and tighten the bolts.

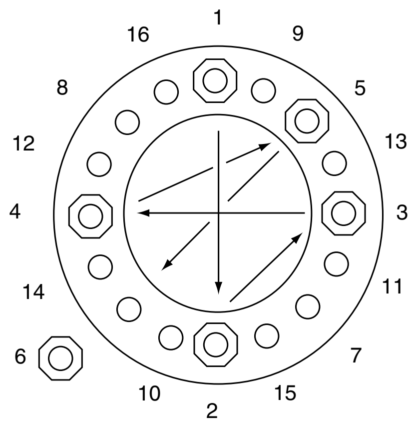


FIGURE 6.2 We'll tighten the bolts of our example joint in the "star pattern" sequence shown here. We'll use three passes, at one-third, two-thirds, and final torque, following the same sequence on each pass.

we're following basically a two-step procedure: pull the joint together and then tighten it. Because this is a learning experiment we'll use ultrasonic equipment (Chapter 9) to measure the preload in each bolt as we tighten it. We'll also measure the angle through which the nut turns after it contacts the surface of the joint, and we'll measure the amount by which the bolt stretches and the amount by which the joint is compressed.

We now apply the snugging torque to the first bolt and use the resulting preload, torque, and turn data to plot the curves shown in Figure 6.3. We're doing work on this fastener as we tighten it. The amount of work is equal to the area under the torque turn curve (measured in lb-ft or N-m times radians). Ideally, all of this work would be converted to potential energy in the bolt and in those portions of the joint members which surround it. If that were the case, all of the work we do on this fastener would end up contributing to the clamping force. Unfortunately and unavoidably, most of our input work is lost.

Typically, about 90% of the work we do on a nut is converted to heat, thanks to the frictional resistance between the face of the nut and the surface of the joint, and between male and female threads. About 50% is lost under the nut, and about 40% within the threads, as shown in Figure 6.4. Only 10% of the input work typically ends up as potential energy in the bolt; so only 10% ends up as bolt preload or as clamping force between joint members.

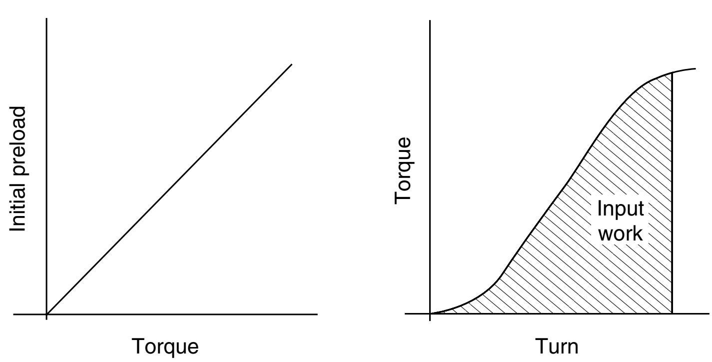


FIGURE 6.3 As we tighten the first bolt in our example joint we plot the buildup of initial preload versus applied torque (left-hand diagram) and applied torque versus the angle through which the nut turns (diagram on the right). The area under the torque–turn curve is equal to the amount of work we're doing on the nut and to the energy delivered to the fastener joint system.

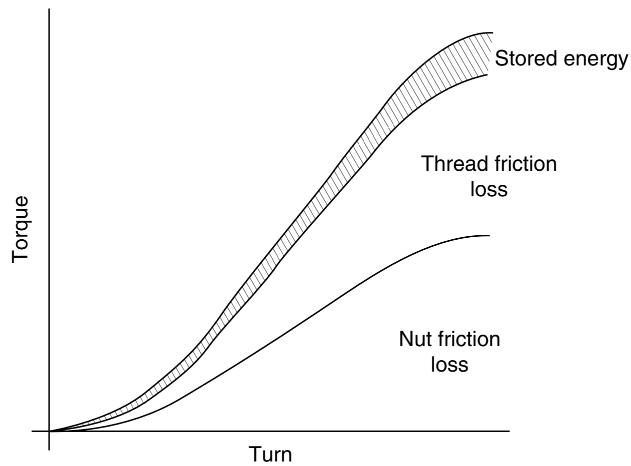


FIGURE 6.4 This diagram shows the approximate way in which the energy delivered to the fastener joint system is absorbed by it. About 50% of the input is lost as friction-generated heat between the face of the nut and the surface of the joint. Another 40% is lost as heat between male and female thread surfaces. Only about 10%, on the average, ends up as potential energy stored in the bolt and joint springs; and only that 10%, therefore, ends up as preload in the bolt and clamping force on the joint.

We'd like to apply a given torque to each bolt and create a given amount of initial preload (the same amount) in each bolt. But the fact that most of the work we do on the nuts is converted to heat makes this virtually impossible, because these frictional losses are extremely difficult to predict or control. Let's assume, for example, that this first nut we're tightening is a little drier than average. As a result, let's assume that 52% of the input work is converted to heat at the nut joint interface, rather than the typical 50%. A 4% increase in friction—from 50% to 52% of the input work—is easy to come by.

This 4% increase in friction loss, that extra 2% of the input work going into heat, means that 2% less of the input work will be converted to the thing we're interested in, the tension in the bolt. We started with the assumption that an average of only 10% of the input would be going into preload; now we've lost a fifth of that. This bolt will, therefore, end up with only 80% as much preload as we expected it to. A 4% swing in friction has caused a 20% change in assembly preload, a very bad leverage situation. And, as we'll learn in the next chapter, there are a lot of factors which can cause this kind of variation in friction.

Although in our learning experiment we're measuring both torque and bolt tension, we won't attempt to compensate for the frictional differences between bolts; we'll apply the snugging torque of 225 lb-ft to the first bolt and let the initial preload end up where it may.

We also plot the deflections in the bolt and in the joint material surrounding the bolt versus the preload we create in the bolts and the presumably equal and opposite clamping force on the joint (see Figure 6.5). We then combine these force–deflection curves, plotting the preload on a common axis, as also shown in Figure 6.5. This creates what the bolting world calls a "joint diagram." The pure of heart among you may complain that the preload in the bolt and the clamping force on the joint are equal and opposite, action and reaction forces, and that both should not be shown as positive values, but this joint diagram is a great convenience so we'll draw it as shown.

Since the diagram records the forces developed in bolt and joint and the deflection of each part, it also gives us a visual indication of the stiffness of the bolt and the stiffness of the joint.

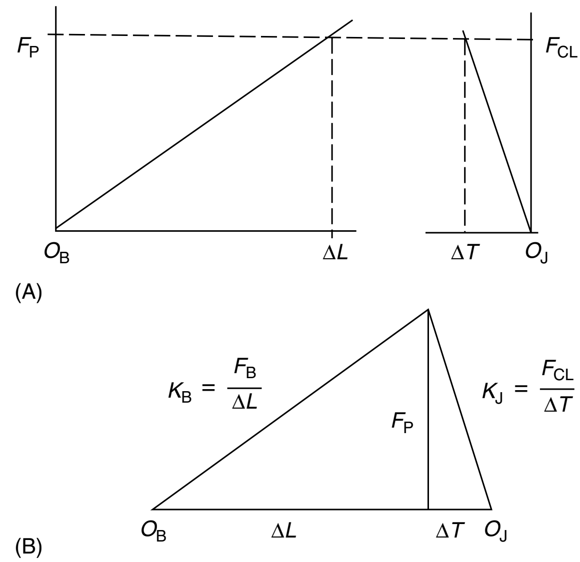


FIGURE 6.5 As we tighten the first bolt we also plot the buildup of preload (F_P) in the bolt versus the increase in length (ΔL) of the bolt, and the buildup of clamping force on the joint (F_{CL}) versus the compression or change in thickness (ΔT) of the joint. At this point we assume that the preload will be equal to the clamping force. These two plots are shown at the top of this illustration. We then combine those two plots, as shown at the bottom of this illustration, to start constructing what we'll call a joint diagram.

These are proportional to the slopes of the two straight lines, and, as we saw in Chapter 5, can be computed as follows:

$$K_B = F_P/\Delta L \tag{6.1}$$

$$K_J = F_P/\Delta T \tag{6.2}$$

where

F_P = preload in the bolt and joint (lb, N)
ΔL = deflection (stretch) of the bolt (in., mm)
ΔT = deflection (compression) of the joint (in., mm)
K_B = stiffness of the bolt (lb/in., N/mm)
K_J = stiffness of the joint material being loaded by this bolt (lb/in., N/mm)

Note that the areas under the bolt and joint curves also equal the amount of energy stored in these parts, as shown in Figures 5.6 and 5.17 in the last chapter. So this simple diagram contains a lot of useful information. We'll extend this diagram in Chapter 10 to add the effects of external loads on the joint. We'll also use joint diagrams when we design joints. For now, however, we're merely interested in using the joint diagram to illustrate the preloading of the bolts.

Our boss, who has already read a previous edition of this book, uses Equation 7.4 in the next chapter to compute the average preload he expected us to get in this first bolt when we applied 225 lb-ft of torque to it. He tells us that we should have created 12,000 lbs of tension in the bolt. Because of the slightly higher than average friction loss described earlier, however, this bolt has ended up with only 80% of that preload, or 9,600 lbs.

Has this really created 9,600 lbs of clamping force between joint members? Our joint diagram assumes it has, but the correct answer is ‘‘probably not’’—at least as far as this first bolt is concerned. Remember that we had to tap some of those bolts into their holes? This implies that there was contact between the sides of those bolts and their holes—bolt–hole interference. Furthermore, the flange surfaces were not pulled into full contact when we first assembled the parts. They were slightly misaligned, as we could tell by the fact that some of those bolts stuck out farther on the threaded end than did others. Before we go on to snug tighten the remaining 15 bolts in our example joint, let's take a look at how hole interference and nonparallel flanges might affect the buildup of clamping force during the assembly process.

6.3 BOLT PRELOAD VERSUS CLAMPING FORCE ON THE JOINT

The main purpose of the bolts is to clamp the joint members together. A common misconception is that there is always an equal and opposite action–reaction relationship between the tension in the bolts in a joint and the thing we're interested in, the clamping force between the joint members. If there are eight bolts in the joint and an average tension of 10,000 lbs in each of the eight bolts, then, simplistically, the joint is clamped together with an interface force equal to eight times 10,000, or 80,000 lbs. That's usually true, but there can be some significant exceptions.

6.3.1 EFFECTS OF HOLE INTERFERENCE

Consider the situation shown in Figure 6.6. We are tightening this stud by turning the upper nut. Our goal is to clamp the joint members together. The hole in the upper joint member is undersized, so the stud is a press fit in this member.

Thanks to frictional and embedment constraints between the sides of the bolt and the walls of the hole, it will take some positive force to pull the bolt through the hole, and then to stretch it within the hole. Where does this force come from?

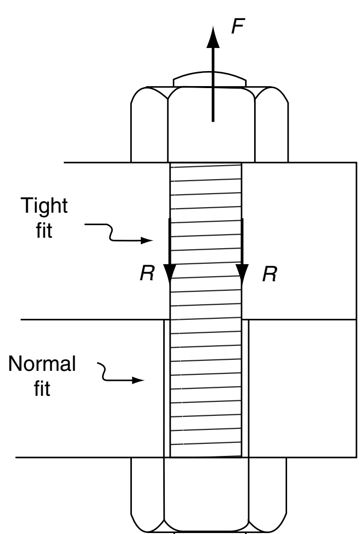


FIGURE 6.6 If there is interference between the bolt and the hole, the clamping force between joint members may be less than the tension in the bolt, changing the torque–clamping force relationship.

The force is created, obviously, when we turn the nut, creating tension in the bolt. Thanks to hole interference, some of this tension will not end up as clamping force between joint members. Part of it will be lost as the bolt fights its way past the walls of the hole.

I'm sure you can envision the extreme case in which the holes are grossly undersized and the torque normally specified for a bolt of this diameter is insufficient even to bring the joint members into contact, much less provide any real clamping force. Note that misalignment between the holes of upper and lower joint members could create a similar hole–bolt interface problem.

This is obviously not a good situation, but hole misalignment, undersized holes, press-fit fasteners, etc. are relatively common in the bolting world.

Hole interference is used on purpose by the airframe industry, for example, to reduce the possibility of fatigue failure of the shear-loaded joints used in airframe structures. (Compressive stress built up in the walls of the bolt holes fights the formation or growth of fatigue cracks.)

The holes are purposely drilled smaller than the diameter of the bolts to create this interference. There is, of course, a manufacturing tolerance on the diameters of both holes and bolts, so the amount of interference varies. The greater the interference, the greater the force required to pull the bolt through its hole. It also requires more force to pull a bolt through a thick plate than through a thinner one, for a given amount of interference.

One airframe manufacturer recently measured the amount of force required to pull bolts of a given nominal diameter through holes drilled in plates of varying thickness. Some of the results are shown in Figures 6.7 and 6.8. The bolt and hole diameters used in the experiment varied through the full range of the manufacturing tolerance. It was found that the force required to pull some of these bolts through their holes exceeded the average preload which would be developed, by the specified torque, in a bolt of that diameter, even if the bolts were used in regular holes with normal clearance. In other words, the specified torque could not be

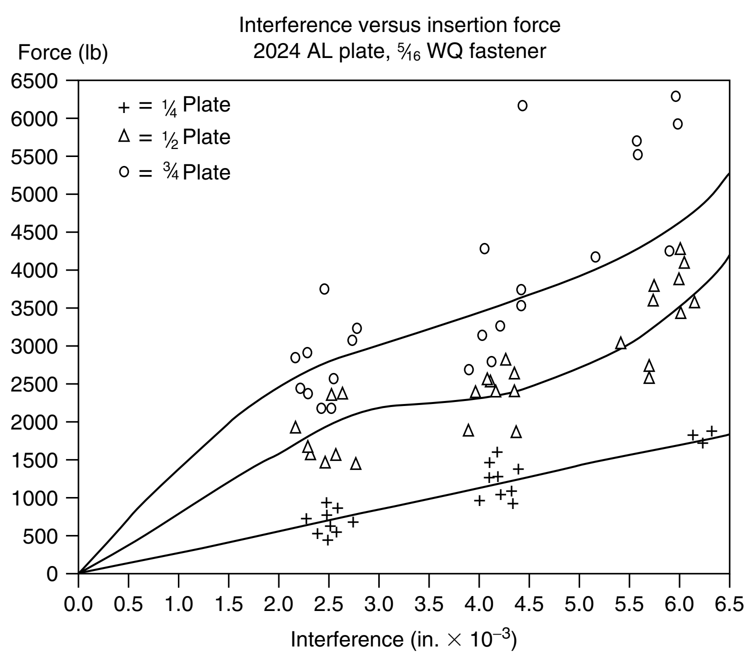


FIGURE 6.7 Chart showing the force required to push, 5/16 in. fasteners through interference fit holes in aluminum plates varying in thickness from 1/4 to 3/4 in. Bolt diameters are larger than the hole diameters by the amounts shown on the horizontal axis. Note the wide scatter in the results, which are summarized in Figure 6.8.

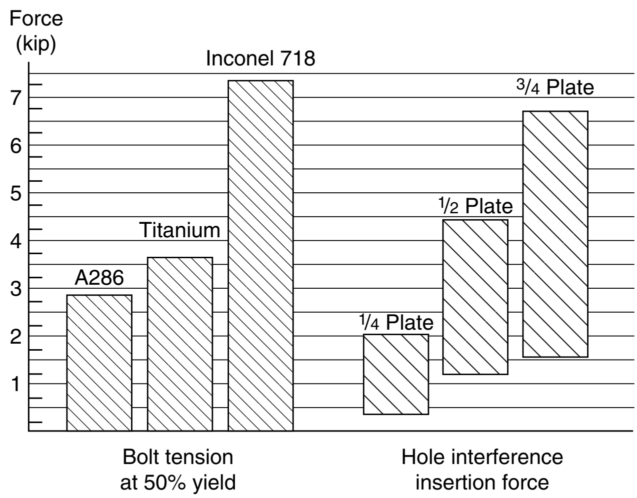


FIGURE 6.8 The three vertical bars on the right side of this chart show the range of force required to thrust 5⁄16 in. fasteners through interference fit holes in aluminum plates of various thickness, summarizing some of the data shown in Figure 6.7. The three bars on the left side of the chart show the nominal preload which would be generated by 5⁄16 in. bolts if tightened to 50% of yield. Three bolt materials are shown. Note that only the Inconel bolts would have enough preload to overcome the worst-case interference forces and still provide some interface clamping force on the joint if ½ in. or ¾ in. plates were involved.

counted on to pull all of the bolts through their holes, much less go on to develop any clamping force between joint members. In spite of this, the same torque is specified by the airframe manufacturer for all bolts of a given diameter, without regard to the amount of hole interference seen by a given bolt or the thickness of the plates in which the bolt is used.

As a result, torque is not used in this application to pull the bolts through the holes. A bolt puller does that job, and temporary clamps are used to hold the joint members together until the bolts have been installed and tightened. But in some joints the act of tightening the bolts is supposed to create some clamping force between upper and lower joint members, and the amount of this clamp must vary widely.

6.3.2 Resistance from Joint Members

Another factor which can rob from the clamping force between joint members is shown in Figure 6.9. A heavy cover is being lifted up against a flange on a pressure vessel. As shown in the figure, the joint members have not yet been brought into contact, but we are turning the nut on the stud to bring them into contact. At the moment, as shown in the figure, there is already tension in the stud equal to the weight of the cover. As a result, it will take torque to advance the cover up against the flange, thanks to the normal frictional constraints between male and female threads and between the nut and the cover.

So, at the point shown, there is torque, tension, and friction loss; there probably will be torsion in the stud, but there is zero clamping force between joint members.

Eventually the two joint members will be brought into contact. Further torque, at this point, will be required to create a clamping force between joint members to load the gasket. That torque should presumably be added to the torque required to pull the joint members together in the first place, but it rarely is in practice.

There aren't many applications in which a heavy weight is raised against a mating joint member by tightening the bolts, of course. We have, however, encountered some

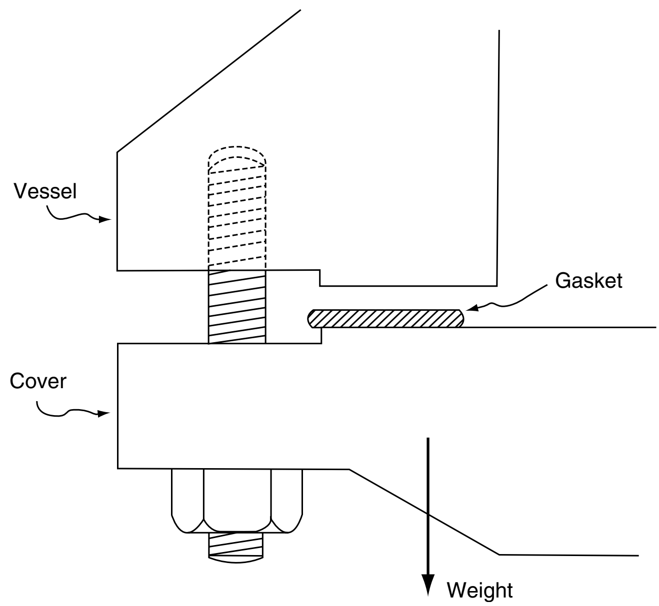


FIGURE 6.9 Heavy and misaligned or warped joint members can also affect the relationship between the torque applied to the fastener, the tension in the fastener, and the clamping force between joint members.

such situations. We learned of one situation in which large windmills are lowered to a horizontal position so that new, heavy blades could be attached. And these blades were raised by tightening the bolts, which fastened them to the structure. Although this kind of thing is rare, it is fairly common for large joint members to be misaligned or nonparallel as the assembly process starts. Getting a pipe flange, for example, to mate with the flange of a pump or valve often requires a lot of motion in the flange members. The forces required to align such systems will have the same effect that the force created by the weight in Figure 6.9 has on that joint.

I wish I could tell you how much extra torque to add for misalignment or the like, but I can't. I have never seen anything in the literature on this subject either. Yet I'm convinced that this factor can seriously degrade the relationship between tension in the bolts and clamping force between joint members, especially in large joints, as one example. Warped or nonflat (e.g., wavy) joint members, incidentally, could create the same sort of problem.

I once met a maintenance supervisor in a large petrochemical plant who took this problem seriously. He insisted that his crews align gasketed flanges within 12 mils before bolting them together. Bolting nonparallel flanges, he said, was a waste of time; "they'll always leak."

The Puget Sound Naval Shipyard has also studied this problem [18]. They made theoretical calculations and conducted experiments to determine the forces, stresses, and moments in pipes and flanges when the flanges are misaligned. They used a hydraulic tensioner, for example (see Chapter 9), to measure the force required to pull misaligned flanges together, as shown in Figure 6.10. The chart in Figure 6.11 shows some of the results of their work: the stresses in the pipe adjacent to the flange as a function of nominal pipe diameter and with flange gaps of 0.015, 0.020, and 0.025 in. The forces (and torques) required to pull the flanges together would be proportional to these stresses.

In each case documented in Figure 6.11, it is assumed that there is a length of pipe equal to 100 times the nominal pipe diameter between the flange and the first rigid pipe support (a rod hanger or intersection with a larger pipe).

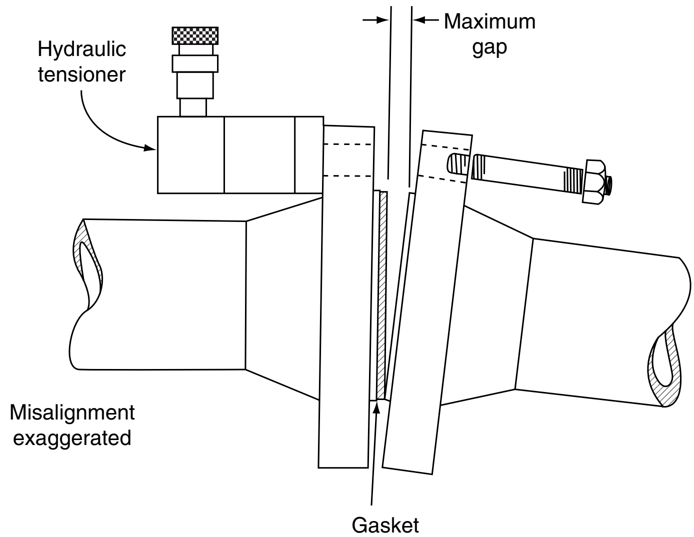


FIGURE 6.10 Puget Sound Naval Shipyard used a hydraulic tensioner as shown in this sketch to learn how much force would be required to pull misaligned flanges together. The data was recorded as a function of the size of the ‘‘gap’’ shown here.

As a result of these studies, Puget Sound developed some flange parallelism criteria, designed, as I understand it, to keep pipe stress below 3 ksi near any flange which is connected to turbines or other rotating equipment. Pipe stresses beside flanges located 50 or more pipe diameters away from rotating equipment were allowed to go slightly higher.

As a final note, if misalignments and stresses are too high, they are sometimes reduced by the use of bellows-type expansion joints at some point in the pipeline.

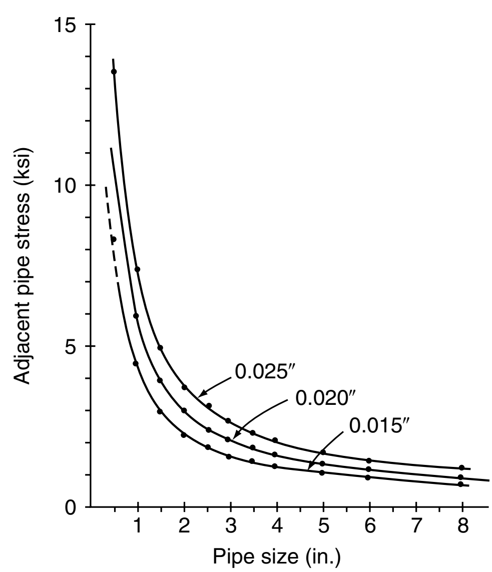


FIGURE 6.11 A plot of the stress in the pipe adjacent to a misaligned flange versus the nominal diameter of the pipe, for misalignment gaps of 0.015, 0.020, and 0.025 in. It is assumed that the nearest rigid support for the pipe is located 100 pipe diameters away, along the pipe, from the misaligned flange. The moment on the flange and the force required to pull the two halves into full contact would be proportional to the pipe stress.

6.4 CONTINUING THE SNUGGING PASS

We're now going to continue tightening the 16-bolt gasketed, flange joint we're taking as an example in this chapter. We apply the snugging torque of 225 lb-ft to each of those bolts, following the cross-bolting pattern shown in Figure 6.2. We measure the preload, turn, and deflection in and around each bolt as we tighten it and we record these data. We find that, on average, we've created the anticipated preload in these bolts, but that individual bolt results vary from the average by $\pm 30\%$. That, our boss assures us, is a typical result of torque-tightening a group of unlubricated, as-received, steel bolts and nuts against steel joint surfaces. Everyone's happy, so we now go out and have lunch.

When we come back from lunch we find our boss's boss on the job site, reviewing our data. He wants to see how we managed to measure the tension or preload in these bolts, so we plug in our ultrasonic instrument and remeasure the preload in bolt number one, the first bolt we tightened to a preload of 9600 lbs. To our embarrassment we find only 900 lbs of tension in that bolt. And we find a wide range of residual preloads in the other 15 bolts. We think that our instrument is misbehaving, but the boss disagrees. He gives us a lecture on bolt relaxation, which some people call "torque loss." Here's what he says.

6.5 SHORT-TERM RELAXATION OF INDIVIDUAL BOLTS

Whether or not there's a one-to-one relationship between bolt tension and interface clamping force, there will often be some initial loss of tension in individual bolts after they are initially tightened. Let's call this "short term" relaxation, to distinguish it from other effects to be discussed in Chapter 11, which will cause further loss of tension over a long period of time.

In general, short-term relaxation occurs in a bolted joint because something has been loaded past its yield point and will creep and flow to get out from under the excessive load. This can be a component, such as a soft bolt or a gasket; more commonly it's only a portion of a component, such as the first threads in a nut. Let's look at some examples.

6.5.1 Sources of Short-Term Relaxation

Here are some things which can cause relatively short-term relaxation, starting with the most common of all—embedment.

The surfaces of the threads in the nut, the bolt, and the faying surfaces of the structural members, washers, etc. are never perfectly flat, even if such parts are given a high polish, which is rarely the case with industrial structures and fasteners. Under a microscope they are a series of hills and valleys.

When such parts are first loaded, they contact each other only through high spots on the metal surfaces. Even a small bolt, however, is able to exert extremely high surface pressures on structural members or on its own threads. Thread dimensions have been selected to support these high loads, but only if a significant percentage of the total thread surface shares that load.

Since initial contact areas are relatively small, the metal at the contact points cannot stand the pressures. Plastic deformation occurs until enough of the total thread surface has been brought into play to stabilize the situation and support the load without further deformation.

The same thing happens in the faying surfaces of the structures, though perhaps to a lesser extent because larger surfaces are involved and initial contact areas are larger. Embedment is illustrated in Figure 6.12.

Many of these surface high spots are smoothed away during the tightening process [1]—at least they will be if the fasteners are torqued. Hydraulic tensioners don't load the active threads, or even the joint surfaces underneath the nut or washer, until they let go off the bolt. As a result, there is often more embedment relaxation after tensioning than there is after torquing.

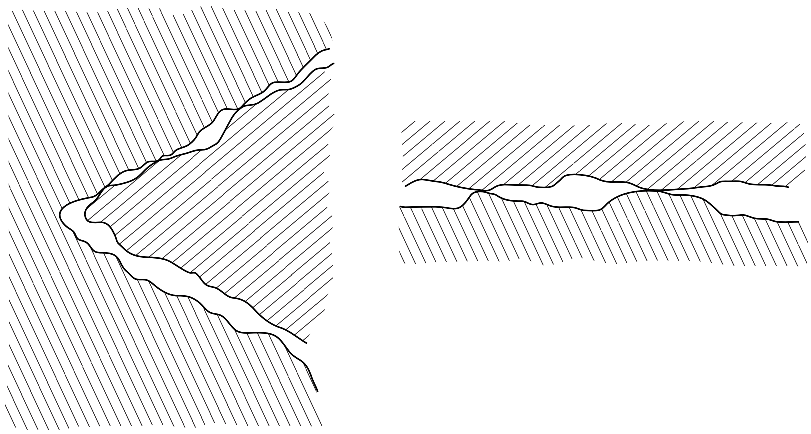


FIGURE 6.12 High spots on thread and other contact surfaces will yield and creep under initial contact forces. As a result, the surfaces will settle into each other until enough contact surface has been brought into contact to stabilize the joint. The process is called embedment.

Embedment is worse on new parts than on reused ones. In critical applications it can be minimized by tightening, loosening, and retightening the fasteners several times. This is done on camera mounts in space satellites, for example [2].

6.5.1.1 Poor Thread Engagement

If the bolt is undersized, or the nut oversized, thread contact areas will be less than those planned by the designer, and substantial plastic deformation may occur, as shown in Figure 6.13 [3].

6.5.1.2 Thread Engagement Too Short

The length of thread engagement for steel fasteners should be at least 0.8 times the nominal diameter of the fastener. If the engagement length is too short (too few threads support the load), thread contact areas are again smaller than those intended by the fastener manufacturer and excessive relaxation can result. One author claims that if thread engagement length is greater than 1.25 times the nominal diameter, "permanent set is negligibly small" [1].

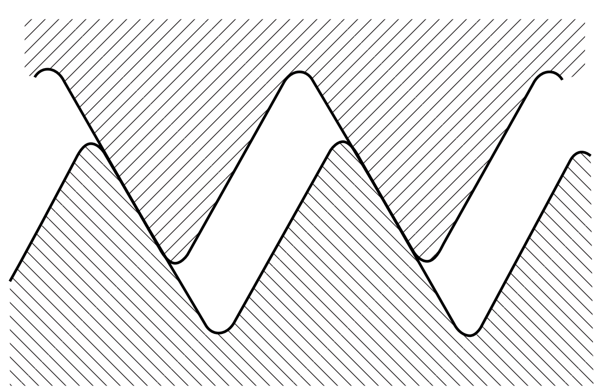


FIGURE 6.13 Poor thread engagement may be a major source of plastic deformation and therefore joint relaxation.

6.5.1.3 Soft Parts

If parts are softer than intended by the designer—perhaps because of improper heat treat or incorrect material—they may creep and relax substantially even if the geometry is correct and loads are normal.

6.5.1.4 Bending

If the fastener is bent as it is tightened, it will see higher stresses along one side than along the opposite side. These higher stresses mean more plastic flow and therefore greater than normal embedment or thread relaxation.

6.5.1.5 Nonperpendicular Nuts or Bolt Heads

The contact faces of nuts and bolt heads are never exactly perpendicular to the axis of the threads or to the axis of the bolt hole. This means that only a portion of the contact surface of the nut or bolt head is loaded when we first tighten the fastener. These abnormally loaded surfaces will creep until enough additional contact area has been involved to reduce contact pressures and stabilize the joint.

6.5.1.6 Fillets or Undersized Holes

If the head-to-body fillet contacts the edge of the bolt hole as shown in Figure 6.14, the edge of the hole will break down under initial contact pressures. This may result in a complete loss of preload, since such effects are usually large compared to the amount by which the bolt was stretched when it was initially tightened [4].

6.5.1.7 Oversized Holes

Undersized holes can be a problem; so can oversized holes. Now there is too little contact between nut and joint surface or between bolt head and joint surface. Unless a washer or something is used to distribute contact pressures and limit contact stresses, the head, nut, or both will embed itself in the joint surfaces, as suggested in Figure 6.15 [3]. The amount of relaxation will, of course, depend on the strength of the surface supporting the nut or washer. The oversized or slotted holes used to aid the assembly of structural joints, for example, don't increase relaxation appreciably [5].

6.5.1.8 Conical Makeups

Surface irregularities will exist on conical joint surfaces as well as flat ones. The effect on axial tension in the fastener, however, is magnified if the embedment occurs on conical surfaces.

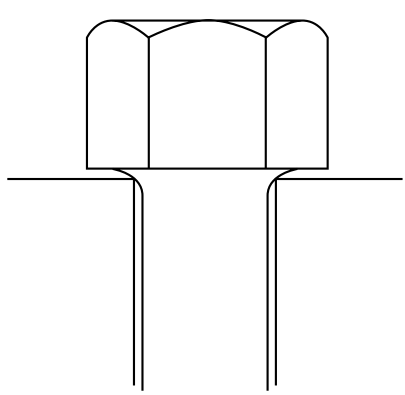


FIGURE 6.14 Oversized fillets and undersized holes may result in total relaxation of a preloaded fastener.

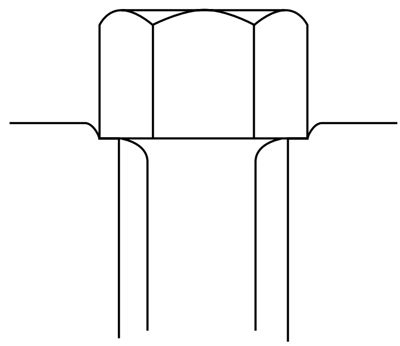


FIGURE 6.15 Oversized holes may also increase contact stress levels and therefore increase embedment relaxation.

A given amount of relaxation perpendicular to the surface may mean substantially greater relaxation in the axial direction, as suggested in Figure 6.16 and Equation 6.3.

$$r = \frac{e}{\sin \theta} \tag{6.3}$$

where

e = embedment relaxation perpendicular to the surface of a conical joint member (in., mm)
r = resulting relaxation parallel to the axis of the fastener (in., mm)
θ = half angle of the cone (deg)

6.5.2 Factors Affecting Short-Term Relaxation

Overstressed parts relax. Overstressing may be created in a number of different ways, as we have seen above. The amount of relaxation that overstressing causes in a given bolt and joint, however, can depend on a number of secondary factors. Here are some of them.

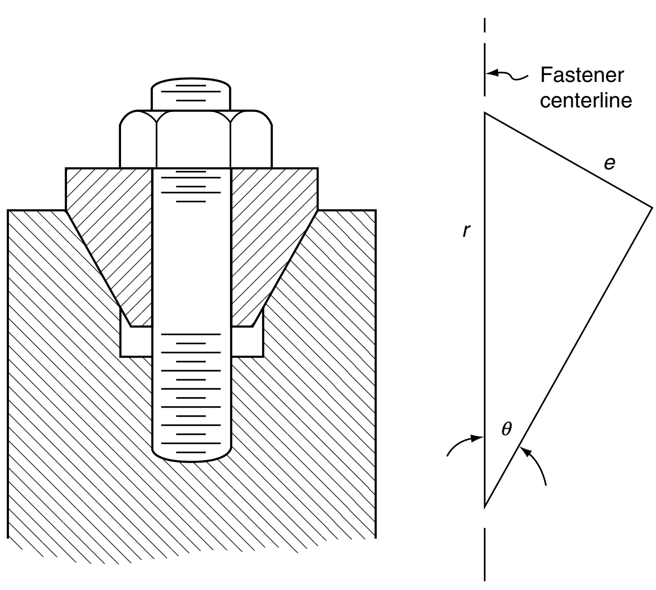


FIGURE 6.16 Conical or tapered joints usually relax more than flat ones for reasons given in the text.

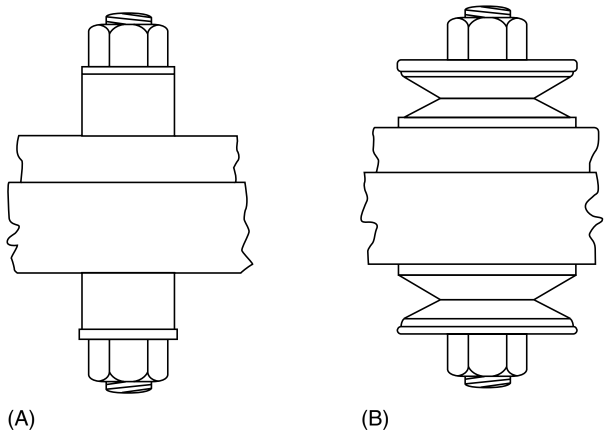


FIGURE 6.17 Since long, thin bolts will relax by a smaller percentage than short, stubby ones, many people use bushings as shown in (A) to reduce percentage relaxation in a given joint. Stacks of Belleville washers (B) are also effective.

6.5.2.1 Bolt Length

Long, thin bolts will relax by a smaller percentage than short, stubby ones. The total embedment relaxation or the like will be the same for a given initial preload, but that embedment will be a different percentage of the total length of the bolt and therefore will mean a different percentage loss in length. Preload loss will be proportional to the change in length (see p. 61 in Ref. [5]).

Many people take advantage of this fact. They add bushings above and below flange surfaces for example, as shown in Figure 6.17. This makes it possible for them to use longer bolts on a given joint.

6.5.2.2 Belleville Washers

Another common way to reduce the change in clamping force produced by a given amount of embedment is to use Belleville washers, as also shown in Figure 6.17. These springs have a very flat rate, compared to the stiffness of either bolt or joint members. They will therefore determine (limit) the preload and clamping force in the system (see Figure 5.16). Because of their flat rate, a small deformation in the bolt or joint won't make an appreciable difference in force levels. For the same reason—and more commonly—Bellevilles are used to compensate for the effects of differential temperature expansion. Spring manufacturers have some trouble controlling the stiffness of Bellevilles, however, so this solution to a temperature or relaxation problem can increase the basic scatter in preload.

6.5.2.3 Number of Joint Members

Increasing the number of surfaces in a joint may increase relaxation effects because there are now more high spots to embed and settle in together. Doubling the number of contact surfaces, for example, will almost double relaxation in many cases [6].

6.5.2.4 Tightening Speed

Creep and flow take time. Fasteners that are tightened very rapidly won't have time to settle in together during the tightening process and will relax more after tightening [7]. This is one of the advantages of pulsed hesitation tightening or of torque-recovery tightening, which we examined in Chapter 7. You tighten the fasteners with high-speed tools, but pause once or repeatedly to give the parts time to relax.

Tightening bolts in a series of passes, rather than applying full torque on the first pass, allows time for relaxation. This procedure also pulls the joint together uniformly. For both of these reasons, progressive tightening is a virtual necessity on large gasketed joints.

6.5.2.5 Simultaneous Tightening of Many Fasteners

Some experiments [8] have suggested that tightening a group of fasteners one at a time results in more relaxation in a given fastener than does tightening several or all of them at once. Presumably a fastener tightened before its fellow sees higher stress concentrations than it does if it is tightened simultaneously with the rest and all share the developing load. Elastic interactions between fasteners, discussed in Section 6.6 of this chapter, are almost certainly involved here as well.

6.5.2.6 Bent Joint Members

If joint members are soft or warped or bent, tightening one fastener can cause relaxation (or additional stress) in other fasteners. This sort of ‘‘cross talk’’ between fasteners is very common, although it is not usually seen or recognized. More about this in Section 6.6.

6.5.3 Amount of Relaxation to Expect

The factors that cause and contribute to relaxation are many and hard to predict. Although attempts have been made to write equations for the amount of relaxation to expect [6,8,9], in most cases the amount must be determined experimentally. And, as is our common fate when dealing with bolted joints, it won’t be ‘‘an’’ amount, but rather a distribution of values around some anticipated mean.

In general, fasteners relax rapidly following initial tightening, then relax at a slower rate, following the pattern shown in Figure 6.18. The amount of relaxation varies greatly, depending on the condition of the parts, finishes, initial and local tension levels, fit of parts, and all of the other factors discussed earlier. Here are some of the relaxation amounts and times found in the references. Fisher and Struik report that tests of A325 and A354 Grade BD bolts in A7

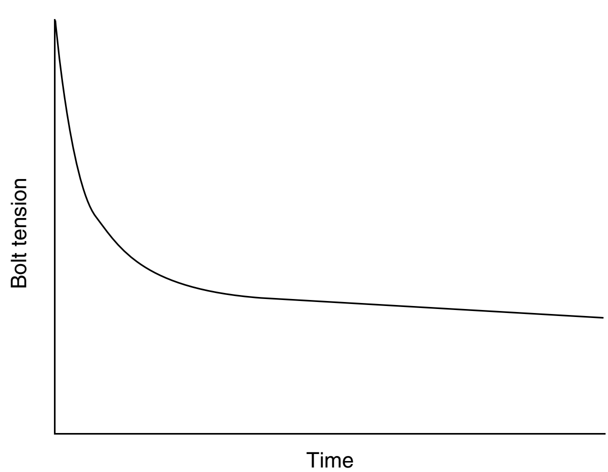


FIGURE 6.18 Most short-term relaxation occurs in the first few seconds or minutes following initial tightening, but continues at a lesser rate for a long period of time.

structural steel showed a loss of 2%–11% of preload immediately after tightening, followed by another 3.6% in the next 21 days, followed by another 2% in the next 11.4 years [5, p. 61]. Bethlehem Steel reports that only 5% of the initial tension will be lost in structural bolts set by turn-of-nut techniques 5% over the total life of the structure [10]. Chesson and Munse report a variety of results on a variety of structural bolts, different types of bolt heads, different nuts, with and without washers, etc. [11]. As one example, an A325 bolt with a regular (not heavy) head, a flanged nut, and no washer relaxed 2.6% in the first minute after tightening (most of this in the first 15–20 s). It had relaxed by 6.5% after 5 days [12]. Hardiman reports that most relaxation occurs in the first few seconds, but that relaxation, usually, never stops [13]. Southwest Research Institute suggests that fasteners lose an average of 5% right after tightening, "because of elastic recovery" [14]. The 24–8 × 124 Nitronic 50 top guide studs in a BWR relaxed an average of 43% after tensioning. Grip length was 4.75 in.; studs were hydraulically tensioned to 160,000 lbs; nuts were run down with a measured torque of 500 in.-lb. This is not all embedment, as we'll see in Section 6.6.

Gasketed joints will relax substantially, whether the bolts are torqued or tensioned. This is especially true during preliminary passes, when loss of as much as 80%–100% of initial tension is not at all uncommon for reasons to be discussed soon. Gaskets will eventually stabilize, however, and will retain the tension introduced in final tightening operations.

6.5.4 Torsional Relaxation

We've been looking at preload or tension relaxation. This is of prime importance to us, because of the general importance of correct preload. We mustn't forget, however, that torsional stress is also built up in a fastener as it is tightened, and that this stress is also subject to varying amounts of relaxation. Many people, in fact, will insist that torsional stress disappears immediately and completely when the wrench is removed from the fastener. Others find that it doesn't disappear until a breakaway torque is applied [15]. Our experience indicates that, like tension relaxation, torsional relaxation depends on many factors; the amount and rate of torsional relaxation will vary substantially from bolt to bolt as well as from application to application.

Figure 6.19 shows the tension and torsion relaxation we measured in an experiment with a 2¼–8 × 12 B16 stud, which had been lubricated with moly. Torsion relaxed 50% when the wrench was removed; tension actually increased 1%–2% during this period. We have subsequently seen this phenomenon on many other types and sizes of bolt. Our guess is that, as embedment allows relaxation of both tension and torsion stress to occur, some of the torsional stress is turned into a little more tension stress. The twisted bolt screws itself farther into its own nut. The exchange is encouraged if you lubricate the threads but do not lubricate the face of the nut.

Sizable relaxation of tension, occurring as the torsional stress disappears, can, of course, mask this exchange of torsion for tension. Thus we've observed this exchange only on hard joints. But it's relatively easy to reproduce, and we think its common. Many bolts in large joints mysteriously "grow" a little between passes, for example, even when there is no temperature change or the like to explain the growth, and even when neighboring bolts remain at constant length. Presumably elastic interactions (Section 6.6) play a role here as well.

A torque wrench appears to respond to torsional stress levels in the bolt as well as to preload levels. The torque required to restart a nut can be less than that required to tighten it in the first place, even if there has been no loss in preload in the meantime if torsional stress has disappeared or been reduced. Repeated "torque recovery" can, as a result, gradually increase the tension in a fastener until it is substantially above initial anticipated levels. In one set of measurements on tank tread end connector bolts, for example, we first tightened the ⅝–18 × 1¾ Grade 8 bolts with a torque of 150 lb-ft. This stretched the bolt 0.0015 in. This was a tapered joint, so the bolt

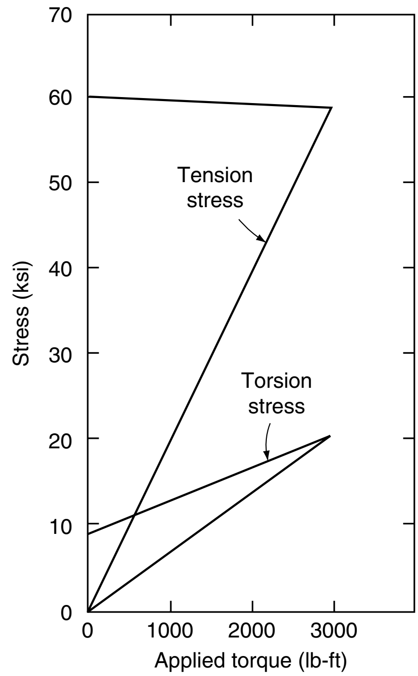


FIGURE 6.19 Relaxation of torsional stress in a bolt can be accompanied by an actual increase in tension. The bolt screws itself into its nut. Data shown were taken in tests on a 2¼–8 × 12 B16 stud.

now relaxed to a stretch of only 0.001 in. We reapplied 150 lb-ft of torque, and the stretch returned to 0.0014 in. Incidentally, it took only 100 lb-ft to restart the nut.

The bolt now relaxed again, was tightened again, relaxed some more, and so on, as shown in Figure 6.20. Final preload (stretch) was 33% greater than that achieved in the first pass,

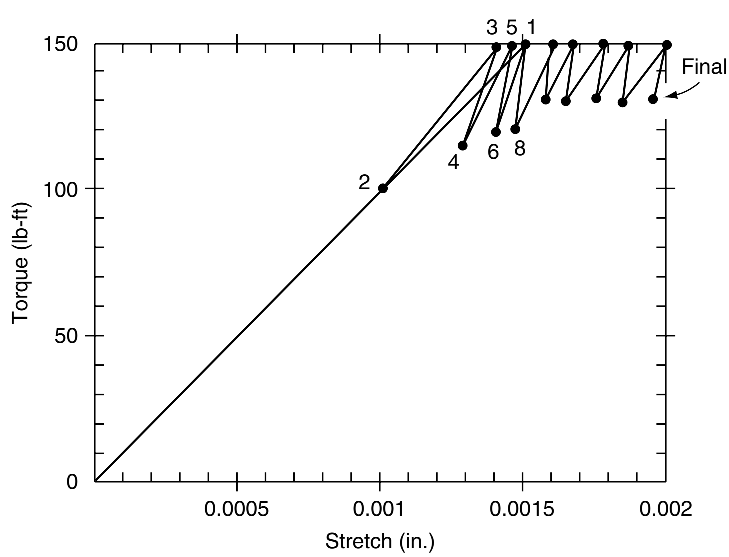


FIGURE 6.20 Torque–stretch-relaxation history of a ⅝–18 × 1¾ Grade 5 bolt. A torque of 150 lb-ft was applied repeatedly to this fastener, with a pause for relaxation between each pass. Final preload was 33% greater than that achieved on the first pass.

even though we never applied more than the initial 150 lb-ft of torque. The final restarting torque was still only 87% of the rated torque. Many mechanics would conclude from this that preload was still 13% below the initial value. This possible interaction between torsional and tension stress further complicates the task of predicting how much a given fastener will relax, of course. It's another complex situation.

6.6 ELASTIC INTERACTIONS BETWEEN BOLTS

Even if we can avoid the problems cited above, and can count on achieving a certain amount of preload in the bolts we tighten at assembly one by one, there are going to be many times when that preload will be significantly modified as we tighten other bolts in the same joint, thanks to "elastic interactions" between bolts, as we'll see in a minute [19–22]. Let's look at an example.

Let's assume that we're planning to tighten a circular, flanged joint that contains eight bolts. We're going to use ganged hydraulic wrenches or hydraulic tensioners to tighten these bolts two at a time. The bolts we tighten simultaneously, of course, will be opposite each other on the flange, 180° apart.

To explain the process of elastic interactions, we're going to think of the joint as a large spring connected by rigid top and bottom plates to the bolts (smaller springs), which are going to be used to clamp it. This arrangement is suggested in Figure 6.21, which shows the first two bolts to be tightened.

Now, let us assume that we have tightened bolts 1 and 2 in this joint, and that magically we have achieved exactly the initial preload we wanted in each bolt. Let's say that this preload is 10,000 lbs of tension in each bolt.

The 20,000 lbs of force that these two bolts are creating on the joint partially compress the joint. We now go on to tighten bolts 3 and 4 located 90° away from bolts 1 and 2. Again, our tools work magic for us and we create exactly 10,000 lbs of initial preload in bolts 3 and 4 when they are tightened (Figure 6.22).

We now have four small springs (bolts) compressing the joint spring rather than the two small springs we had a moment ago. If bolts 1 and 2 had retained their full preload, we would now have 40,000 lbs of force on this joint instead of 20,000 lbs. Doubling the compressive force on the joint spring would, of course, double the amount by which it is compressed. But what happens in bolts 1 and 2 when we tighten 3 and 4? Bolts 1 and 2 are allowed to relax a little as the joint is compressed by bolts 3 and 4.

At this point in the process, therefore, bolts 1 and 2 have a slightly lower amount of preload in them than bolts 3 and 4—even though each of the four bolts started with the same

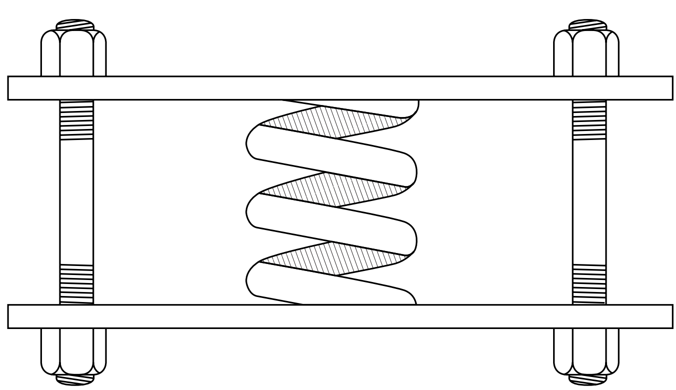


FIGURE 6.21 A simulated model of a bolted joint, in which the joint members are represented by a large spring, here "loaded" by the first two bolts to be tightened.

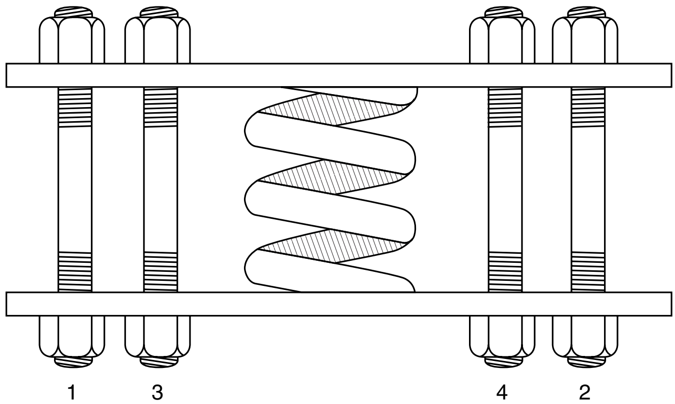


FIGURE 6.22 The joint model of Figure 6.21, but now with four bolts to be tightened.

initial tension of 10,000 lbs. When we now go on and tighten bolts 5 and 6 in a third step, bolts 3 and 4 will relax a little, and bolts 1 and 2 will relax further. Tightening bolts 7 and 8 to complete the assembly will create relaxation in each of the six bolts tightened earlier.

The result is four different levels of residual preload in the eight bolts when they are tightened two at a time, even though the initial preload in each one was identical to start with. And this is not just a theoretical possibility; it's a very common occurrence. Most people are not aware of this interaction, however, which is visible only if you use ultrasonics or strain gages or something to monitor the tension in the bolts.

Figures 6.23 through 6.25 show some actual elastic interaction data. A raised face gasketed joint with a spiral-wound, asbestos-filled gasket was tightened in three passes using a cross-bolting pattern and tightening one bolt located 180° away from bolt 1. The third and fourth bolts tightened are halfway between bolts 1 and 2, etc. The difference between the "x"s and solid line shows the loss of initial preload in the bolts as a result of elastic interactions at a time. Figure 6.23 shows the results after the first pass in which 100 ft-lb of torque was applied to each bolt.

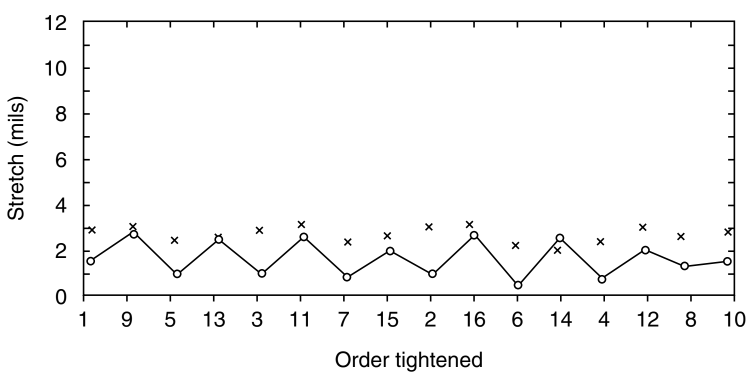


FIGURE 6.23 The elongation or stretch achieved in the 16 bolts of a gasketed flanged joint as the bolts are initially tightened one by one (x's) and after all have been tightened (solid line). Numbers on the horizontal axis show the location of the bolts, and the order in which they were tightened. The second bolt tightened, bolt 2, is located 180° away from bolt l. The third and fourth bolts tightened are halfway between bolts 1 and 2, etc. The difference between the x's and solid line shows the loss of initial preload in the bolts as a result of elastic interactions.

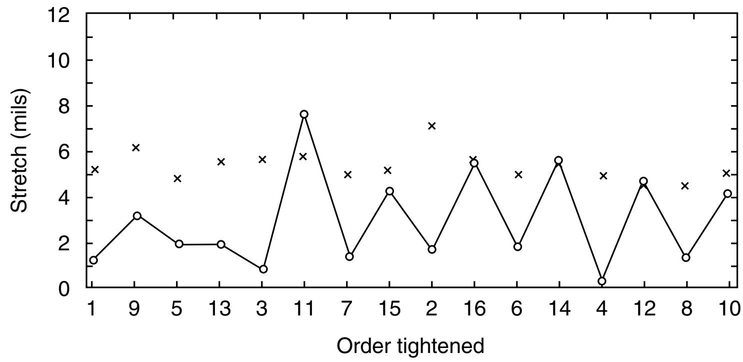


FIGURE 6.24 Initial and residual preloads in the 16 bolts of the joint shown in Figure 6.23 after a second tightening pass at a higher torque.

The isolated "x"s show the initial change in length achieved in each bolt as it was tightened individually. This change in length, or stretch, is proportional to the tension in the bolt, as we saw in Chapter 5 (or will see at greater length in Chapter 9). The sawtooth line shows the pattern of residual preload (stretch) in all of the bolts in this joint following completion of the first pass. The numbers on the horizontal axis of the figure define the sequence in which the bolts were tightened and their relative position around the joint. For example, bolt 1 was tightened first; bolt 2, 180° away from bolt 1, was tightened second; bolt 3, halfway between bolts 1 and 2, was tightened third; and so on.

Note that when bolt 1 was originally tightened, approximately 3 mils of stretch was created in that bolt by the 100 ft-lb of torque. After all the bolts in the joint had been tightened, however, the tension in bolt 1 was remeasured, and it was found to have only about 1.5 mils of stretch. Bolt 3 started and ended about the same place. Bolt 6 started with about 2 mils, but lost all but 0.25 mil as the other bolts in the joint were tightened.

In pass 2, 200 ft-lb of torque was now applied to each of the bolts, again one at a time and in the same cross-bolting pattern used for the first pass. Figure 6.24 shows the tension created in individual bolts during this pass (x's) and the final residual tension in each bolt at the end of the pass (line). Note that the sawtooth curve, which was very regular in Figure 6.23, has now started to break up and become more erratic. Note, too, that at this point in the procedure the scatter between maximum and minimum residual tension in the bolts is nearly 20:1, ranging from less than 1 mil of stretch in the bolt tightened fourth to nearly 8 mils of stretch in the

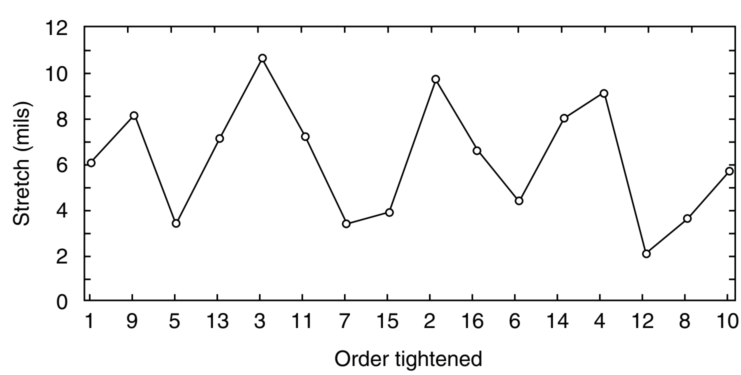


FIGURE 6.25 Final tension in the 16 bolts of the joint of Figures 6.23 and 6.24 after a final cross-bolting pass at a final (highest) torque.

eleventh bolt tightened, and this despite the fact that the scatter between maximum and minimum tensions created in individual bolts as they were first tightened is fairly normal. In pass 1, for example, the scatter between applied torque and achieved initial preload was roughly $\pm 30\%$, as anticipated for as-received steel-on-steel bolts. But by the end of pass 2, the scatter in residual tension is 20:1.

Figure 6.25 shows the results after a third and final pass at 275 lb-ft of torque. The solid line shows the final residual preload in all bolts. The maximum to minimum range of preload is about 5.5–1, a far cry from $\pm 30\%$.

As mentioned, the joint we have just been examining contained a spiral-wound, asbestos-filled gasket. The fact that a gasket was involved certainly contributed to the amount of elastic interaction observed here. But even hard metal-to-metal joints show this effect to some extent.

The fact that we were using a torque wrench to tighten these bolts contributes somewhat to the initial bolt-to-bolt scatter in tension, of course. But the elastic interaction contributes far more to the final scatter, and has nothing to do with the type of tool used. Figure 9.11, for example, shows similar results obtained when a large joint was tightened with four hydraulic tensioners. The range between maximum and minimum residual bolt stretch in this joint was 5–28 mils, thanks largely to elastic interactions. This was a foundation joint with no gasket.

Although usually invisible, elastic interactions of the sort shown here have always been with us. They are one of the reasons why joints have always had to be grossly over-designed to function properly. Although worse in gasketed joints than in metal-to-metal joints, they may be less significant in the gasketed joint because this joint has a "memory." Its leak behavior depends as much upon the initial preload developed in its bolts as it does to the residual preload. You'll find a complete discussion of this in Volume 2 of this text.

Is there any way to prevent or eliminate elastic interaction?

Attempts have been made to define a torquing procedure that will minimize the effects of elastic interactions [16,17]. The authors suggest that if different amounts of torque were applied to each of the groups of bolts in the example joint illustrated in Figure 6.22, for example, it would be possible to end up with the same amount of tension in each bolt after a single pass. One might, for example, apply 400 ft-lb of torque to bolts 1 and 2; 300 to bolts 3 and 4; 200 to bolts 5 and 6; and 100 to bolts 7 and 8. Bolts 1 and 2 relax three times; bolts 3 and 4 relax twice; bolts 5 and 6 relax once; and all end up with the same amount of tension finally created in bolts 7 and 8.

Several of us have attempted to do this, but without success. One problem is that the amount of torque applied to the groups tightened earliest can be astronomical—unless a large number of bolts are tightened simultaneously. More important, however, the elastic behavior of the joint and individual bolts is basically unpredictable. Tightening the same bolts in the same joint repeatedly usually produces different final patterns of residual preload.

This is illustrated in Figure 6.26, which shows the results achieved in another experiment on the joint described in Figures 6.23 through 6.25. The bolts were tightened in three passes as before, but the final pattern of preload differs significantly from that shown in Figure 6.25.

This time, in an experiment to reduce the residual scatter, we made a fourth pass at the final torque of 275 lb-ft, but in reverse order. The bolt tightened last on pass 3 was tightened first on pass 4. The bolt tightened next to last was tightened second, etc. As you can see, this reduced the scatter in residual from about 7:1 to about 3:1, which could be helpful in some situations, but which is probably not worth the effort in others. Note that even 3:1 is a long way from the $\pm 30\%$ often claimed for torque procedures in general.

We've tried other torquing procedures to reduce final scatter. For example, we've tried making a final pass at 75% of final torque on the odd-numbered bolts only; tried using a large number of passes (15 or 20) at the final torque; tried making several passes at the final torque, using a clockwise pattern, rather than cross-bolting, for these final passes, etc. Most of the things tried seem to help somewhat, but the main key seems to be "more passes," regardless of

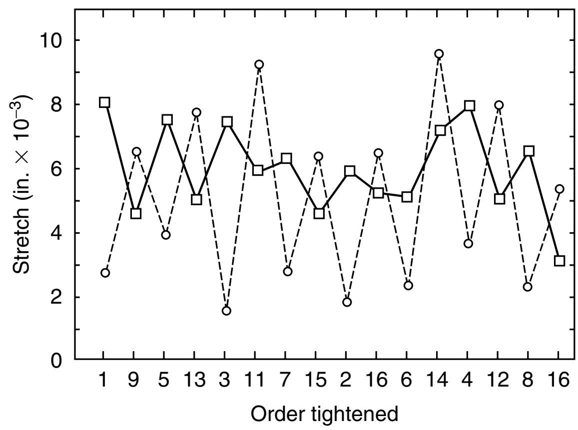


FIGURE 6.26 The dashed line shows the pattern of final, residual tension in the 16 bolts of the joint described in Figures 6.23 through 6.25 after that joint has been loosened and then retightened with the same torques and procedure used earlier. The solid line shows the change in pattern of residual tensions after a fourth and final pass in reverse order (the last bolt was tightened first).

what pattern or format is used. Scatter is reduced slowly as the number of passes increases. But scatter cannot be eliminated by any "torquing" procedure I know of. If ultrasonics or datum rods or load cells can be used to monitor results, a different torque can be applied to each bolt and scatter can then be reduced to $\pm10\%$ or less, depending on the time you spend on it and the skill of your mechanics. But this isn't a torque control procedure; it's stretch or strain control. Pure torque control cannot avoid the type of results described.

The only sure way to eliminate elastic interactions entirely is to tighten all the bolts in the joint simultaneously. That is actually done on some reactor pressure vessels and on automotive engine heads, for example. The closer you can come to simultaneous tightening, the less elastic interaction effect you will get. For example, if you can tighten half the bolts in the joint simultaneously, you will see less interaction than if you tighten them individually or tighten one-quarter of them at a time.

What saves most of us is the fact that it's not necessary to eliminate or compensate for elastic interactions. The life and behavior of most joints—including most gasketed, pressurized ones—depends more on the average tension in the bolts than on the range of tension or minimum tension. Knowledge of elastic interactions can help you solve a chronic leak or other joint problem, but in most cases the common solution is an increase in average preload rather than a decrease in preload scatter, although both are sometimes required. We can now modify our joint diagram to show the loss in preload created by embedment and by the average elastic interaction effect, as in Figure 6.27. The dashed lines show the situation—bolt and joint forces and deflections—before relaxation. The solid line shows the residual preload situation at the end of the assembly operation. It is this solid triangle we'll build on in later chapters as we add the effects of external loads etc.

As a final note, Dr. George Bibel of the University of North Dakota has been conducting studies of elastic interactions sponsored by British Petroleum America and by the Pressure Vessel Research Committee [19–22]. He has found a way to achieve final bolt tensions, with a scatter of only $\pm2\%$ or so, in large-diameter gasketed or ungasketed joints. (His tests involved flanges with diameters of 16 and 24 in.) Using a cross-bolting pattern, he first tightens the bolts to an arbitrary, experimental tension, which he monitors with strain gages or ultrasonics. During this pass he determines the amount of interaction between each bolt and (primarily) its neighbors. He then loosens the bolts and retightens them in a

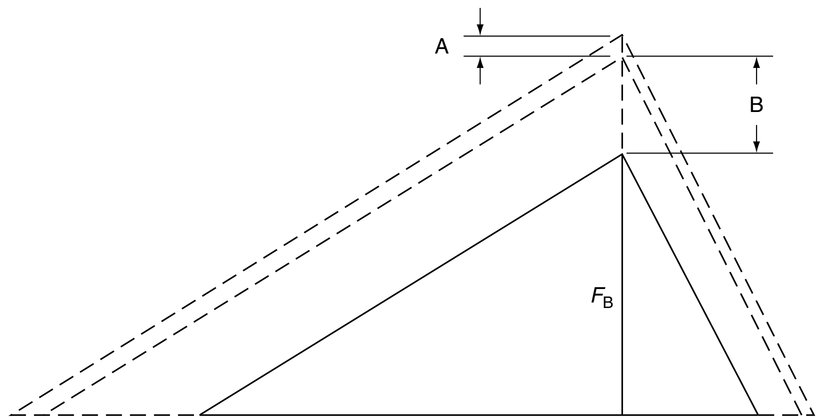


FIGURE 6.27 The joint diagram of Figure 6.5 modified to show the effects of embedment and elastic interaction on initial preload. The largest triangle reflects the initial, "just tightened" situation. Embedment reduces the initial preload by an amount shown as A. The average loss in a group of bolts, caused by elastic interactions, is shown as loss B. We use the average loss instead of the individual loss because joint behavior will more likely be determined by average loss than by worse case. The final, average, residual preload in this group of bolts is represented by the solid triangle.

single, cross-bolting pass, determining the amount each is to be tightened by use of a matrix of interaction coefficients derived from the data taken during the experimental pass. Once again, tension and not torque is used to determine the initial preload developed in each bolt during this final pass. He believes that this procedure could, in one possible application, reduce the radiation exposure of maintenance workers in nuclear power plants.

Dr. Bibel reports that elastic interactions during a conventional three-pass bolting procedure can result in an average loss of preload of 25%–50%. This is the case, at least, if relatively thick, spiral-wound gaskets are involved. Thinner gaskets reduce the average loss, as does metal-to-metal contact, but some loss occurs in each case.

He also reports that the elastic interactions in a given joint appear to be repeatable, and therefore predictable, which would suggest that the interaction coefficients would have to be determined only once for a given joint. Finally, he reports that the first bolt tightened in a cross pattern can be overloaded by up to 50% when later bolts are tightened, again, during a normal bolt-up procedure.

6.7 THE ASSEMBLY PROCESS REVIEWED

After learning about elastic interactions, we make one more pass around our 16-bolt joint, in the reverse order, to reduce the scatter in residual preload. We remeasure the preloads in each bolt after that pass and find that each has changed. We accept the final results and then draw the block diagram shown in Figure 6.28 to summarize the things we now know can affect the amount of clamping force created on a multibolt joint when it's tightened. We include a few things learned in earlier chapters or elsewhere; for example, the fact that some bolts will be bent slightly if tightened against nonparallel joint surfaces (as discussed in Chapter 2). We also include some minor factors not discussed, such as the very small heat loss in the tools and drive bars we're using. We also include the possibility of heat loss through "prevailing torque" when interference of some sort between male and female threads makes it necessary to use a wrench simply to run a nut down, before the nut contacts the joint. We'll learn more about this in Chapter 14.

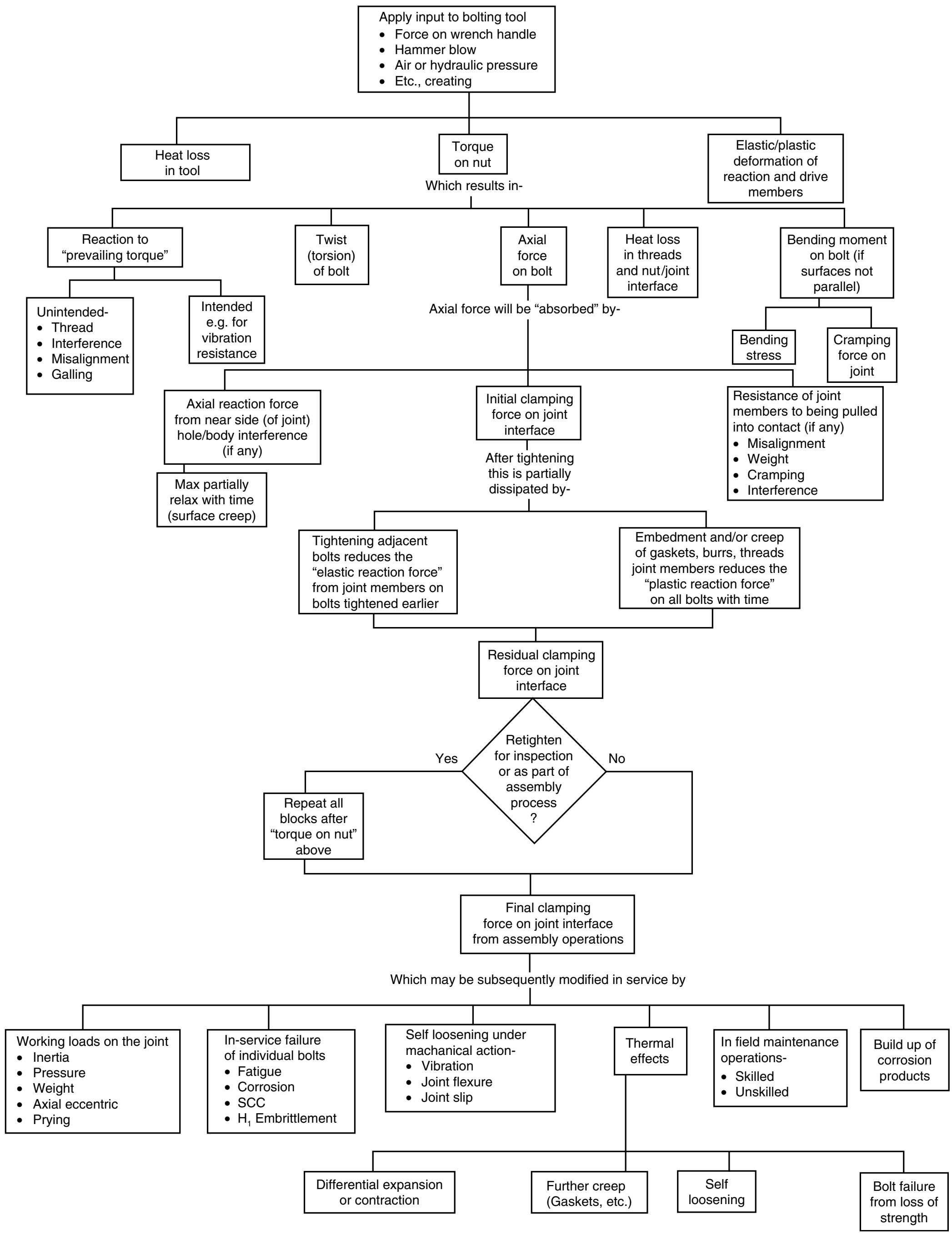


FIGURE 6.28 Block diagram showing most of the factors which affect the relationship between the input work done by the tool on the bolt or nut, and the subsequent in-service clamping force between joint members.

Because of our interest in stored energy, we also list the many ways the input work we do on the nuts is absorbed by the bolts and joint members. The work

- Heats the parts and tools
- Enlarges interference fit holes
- Bends and twists the bolts
- Dilates the nuts (see Chapter 3)
- Pulls the joint members together
- Overcomes prevailing torque
- Deforms the parts plastically (embedment and gasket creep)
- Deforms the bolts and joint members elastically

It stretches the bolts and compresses the joint. Only the work done deforming the parts elastically ends up as stored energy, creating clamping force; but then some of this work is lost as the parts embed and relax.

We now realize why accurate control of the assembly process is so difficult. We would have to predict or control such unpredictable things as the friction forces between parts, the effort required to pull the joint together, the amount of hole interference, and the elastic interactions between bolts to achieve exactly the same amount of residual preload in each bolt. I think that it's safe to say that we'll never be able to afford the effort required to predict or control the many variables involved.

6.8 OPTIMIZING ASSEMBLY RESULTS

This doesn't mean, however, that all is lost. A great deal of work has been and is still being done on assembly tools and techniques. There are ways to get better results than we achieved with our example joint in this chapter. In the next three chapters, we're going to look at most of these techniques. This education will allow us to pick the most appropriate assembly methods for our own applications.

Regardless of the type of control we adopt, furthermore, there are a number of relatively simple things we can do to improve our chances of success during assembly operations. In fact, the things listed below can often make more improvement than the use of elaborate or expensive preload control equipment. As you can see, most of the items on the list would tend to make your assembly practices and procedures more consistent. Although you can't predict or control the many variables affecting results, you can reduce their variation by being consistent.

1. Be as consistent as possible in your choice and use of tools, procedures, calibration frequency, etc.
2. Train and supervise the bolting crews. This can help far more than a better lubricant or more expensive tool. Let the people know how important good results are, and how difficult they are to obtain. Enlist their help.
3. Make sure the fasteners are in reasonable shape. Wire-brush the threads if they're dirty and rusted. Use stainless steel bristles on alloy steel materials. Chase threads with a tap or die if they're damaged. Or replace the bolts with new ones!
4. Use hardened washers between the turned element (nut or bolt head) and joint members.
5. If lubricants are to be used, make sure they're clean and fresh. Apply them consistently, the same amount to the same surfaces by the same procedure. Preload scatter will be minimized if lubes are used on both thread and nut (or head) contact surfaces.
6. Run the nuts down by hand. If you can't, the threads may need to be cleaned or chased.

7. Hold wrenches perpendicular to the axes of the bolts.
8. Apply torque at a smooth and uniform rate. Avoid a stick-slip situation as you approach the final torque. If necessary, back the nut off a little so that the specified torque can be reached with the wrench in motion.
9. If hydraulically powered wrenches are used, be sure that adequate reaction points are used, and that the tools aren't twisting or cramping as a result of cocked or yielding reaction surfaces.
10. Snug the joint first, with a modest torque; then tighten it. Try to align the joint members before tightening the bolts.
11. Tighten from the center of bolt patterns toward the free edges if the bolt pattern is rectangular (as in a structural steel joint). Work in a cross-bolting pattern on circular or oval joints.
12. Keep good records of the tools, operators, procedures, torques, and lubricants used. Bolting is an empirical "art" at present. If you record details of your "experiment" you'll have the information you need to make a controlled modification in the procedure the next time, if the first procedure doesn't give the results you want.
13. If possible, develop your own nut factors, experimentally, rather than relying on a table. Perform the experiment under actual job conditions if possible, though a lab test on your joint (or a simulation of your joint) is better than nothing.

EXERCISES

1. What is the difference between initial and residual preload?
2. Typically, what percentage of the work we do with a torque wrench ends up as preload in an individual bolt?
3. Is the preload mentioned in question 2 above initial or residual preload?
4. What would be the result if we lubricated the bolts in a given joint for the first time and reduced the friction loss during tightening by 3%?
5. Why do individual bolts relax a bit when first tightened?
6. Name several factors that can increase the relaxation.
7. Which will tend to loose more preload after initial tightening, a short bolt or a long bolt?
8. What are the implications of your answer to question 7 above?
9. Name several ways by which the loss of initial preload can be reduced.
10. Roughly how much embedment relaxation could we expect from new parts used in a non-gasketed, steel joint?
11. Define elastic interactions.
12. How can elastic interactions be reduced?

REFERENCES

1. Landt, R., *Preload Loss and Vibration Loosening*, SPS Technologies, Jenkintown, PA, 1979.
2. Conversation with Carl Osgood, Consultant, Cranbury, NJ, 1974.
3. Friesth, E.R., Performance characteristics of bolted joints in design and assembly, SME Technical Paper AD77-715, 1977.
4. High strength bolted joints, SPS Fastener Facts, SPS Technologies, Jenkintown, PA, section IV-C-4.
5. Fisher, J.W. and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, Wiley, New York, 1974, p. 179.
6. Meyer, G. and D. Strelow, How to calculate preload loss due to permanent set in bolted joints, *Assembly Engineering*, February and March 1972.
7. Ehrhart, K.F., Preload—the answer to fastener security, *Assembly and Fastener Engineering* (*London*), vol. 9, no. 6, pp. 19–23, June 1971.

8. Prodan, V.D., et al., More precise load-relief factor for tightened screw connections, *Vest. Mashin.*, vol. 54, no. 1, pp. 27–28, 1974.
9. Junker, G., *Principles of the Calculation of High Duty Bolted Connections-Interpretation of Guideline VDI 2230*, VDI Berichte, no. 220, 1974, published as an Unbrako technical thesis by SPS, Jenkintown, PA.
10. *High Strength Bolting for Structural Joints*, Booklet 2867. Bethlehem Steel Co., Bethlehem, PA, p. 9.
11. Chesson, E., Jr. and W.H. Munse, *Studies of the Behavior of High Strength Bolts and Bolted Joints*, University of Illinois College of Engineering, Engineering Experiment Station Bulletin 469, 1964.
12. Mayer, K.H., Relaxation tests on high temperature screw fastenings, *Wire*, vol. 23, pp. 1–5, January–February 1973.
13. Hardiman, R., Vibrational loosening-causes and cures, Presented at the Using Threaded Fasteners Seminar at the University of Wisconsin-Extension, Madison, May 8, 1978.
14. Investigation of threaded fastener structural integrity, Report prepared by Southwest Research Institute, San Antonio, Texas, under contract no. NAS9-15140, DRL Tii90, CLIN 3, DRD MA 129 TA, SWRI project no. 15-4665, October 1977.
15. Blake, T.C. and H.J. Kurtz, The uncertainties of measuring fastener pre-load, *Machine Design*, September 30, 1965.
16. Van Campen, D.H., A systematic bolt-tightening procedure for reactor pressure vessel flanges, ASME First International Conference on Pressure Vessel Technology, Delft, Netherlands, Part 1, pp. 131–141, September 29–October 2, 1969.
17. Rumyantsev, D.V., V.D. Produn, and A.F. Pershin, A tightening procedure for fastening bolts applicable to high-pressure apparatus, *Khimicheskoe i Neftyanoe Mashinostroenie*, no. 2, pp. 3–5, February 1973.
18. Tripp, Owen S., and Earl D. Stevens, Flange parallelism criteria, presented to the PVRC Subcommittee on Bolted Flanged Connections, San Francisco, CA, January 1983.
19. Bibel, G. and R. Ezell, An improved flange bolt-up procedure using experimentally determined elastic interaction coefficients, *Journal of Pressure Vessel Technology*, vol. 114, November, p. 439, 1992.
20. Bibel, G. and R. Ezell, Bolted flange assembly: preliminary elastic interaction data and improved bolt-up procedures, draft of paper submitted to the Pressure Vessel Research Committee, May 18, 1993.
21. Bibel, G.D. and D.L. Goddard, Preload variation of torqued fasteners, a comparison of frictional and elastic interaction effects, *Fastener Technology International*, February 17, 1994.
22. Bibel, G., "Tightening groups of fasteners in a structure and the resulting elastic interaction," in *Handbook of Bolts and Bolted Joints*, John H. Bickford and Sayed Nassar (Eds.), Marcel Dekker, NY, 1998.

7 Torque Control of Preload

We have now reviewed the assembly process and have seen why the control of that all-important clamping force on the joint is so difficult. During this review we assumed that we were using the torque applied to the nut to control the tension or preload built-up within the bolt. Now we're going to take a much closer look at torque control itself. We'll learn how to estimate the initial preload we'll achieve by applying a given torque to a fastener; we'll take a brief look at torque tools; and we'll look at "restarting" or "breakaway" torque as a means of evaluating the residual preload in a bolt.

In later chapters we'll study other means of controlling assembly preloads. But make no mistake: at this point in time, torque is king. It's by far the best-known, most common, and usually the least expensive way to control preload. It doesn't do a perfect job, but it produces results which are good enough for the vast majority of applications requiring control. So—it certainly deserves our full attention.

7.1 IMPORTANCE OF CORRECT PRELOAD

In earlier chapters we learned that the main purpose of most bolts is to clamp the joint members together. The clamping force is created when we tighten the bolts during assembly, when we preload the bolts, create tension in them, by turning the nut or bolt while holding the other.

During initial assembly of an individual bolt there's a one-to-one relationship between the tension in a bolt and its preload. As we saw in the last chapter, however, the tensile load in a bolt will change as we tighten other bolts or put the joint in service, so the preload is an initial and short-lived affair. Nevertheless, it's extremely important.

As I mentioned in the last chapter, we're interested in the tension in a bolt at three different times.

Initial preload: The tension created in a bolt when it is first tightened.

Residual preload: The tension remaining in a bolt at the end of the assembly process, after all bolts have been tightened.

The tension in a bolt in service: We're most interested in the last one on this list, the tension in the bolts while they're in service, but in this and the next two chapters we're going to focus on the initial preload we get when we first tighten a bolt. This is the preload over which we have the most direct control, and it will usually determine and often dominate the residual and in-service conditions. Unless the joint is very poorly designed, subsequent tightening of other bolts or working loads, thermal loads, vibration loads, etc. won't modify the initial preloads enough to cause joint problems. In other words, there'll usually be a direct relationship between this transient, initial preload and the ultimate behavior of the joint. Correct initial preload is essential.

7.1.1 Problems Created by Incorrect Preload

We'll take a closer look at these points in later chapters, but here's an introduction to the problems created by incorrect preload.

1. *Static failure of the fastener*: If you apply too much preload, the body of the bolt will break or the threads will strip.
2. *Static failure of joint members*: Excessive preload can also crush or gall or warp or fracture joint members such as castings and flanges.
3. *Vibration loosening of the nut*: No amount of preload can fight extreme transverse vibration, but in most applications, proper preload can eliminate vibration loosening of the nut.
4. *Fatigue failure of the bolt*: Most bolts that fail in use do so in fatigue. Higher preload does increase the mean stress in a fastener, and therefore threatens to shorten fatigue life. But higher preload also reduces the load excursions seen by the bolt. The net effect is that higher preload almost always improves fatigue life.
5. *Stress corrosion cracking*: Stress corrosion cracking (SCC), like fatigue, can cause a bolt to break. Stresses in the bolt, created primarily by preload, will encourage SCC if they're above a certain threshold level, as we'll see in Chapter 16.
6. *Joint separation*: Proper preload prevents joint separation; this means that it reduces or prevents such things as leaks in a fluid pipeline or blow-by in an engine. The latter, of course, means that proper preload allows the engine to produce more horsepower.
7. *Joint slip*: Many joints are subjected to shear loads at right angles to the axis of the bolt. Many such joints rely for their strength on the friction forces developed between joint members, forces created by the clamping force exerted by the bolt on the joint. Again, therefore, it is preload that determines joint integrity. If preload is inadequate, the joint will slip, which can mean misalignment, cramping, fretting, or bolt shear.
8. *Excessive weight*: If we could always count on correct preload, we could use fewer and smaller fasteners, and often smaller joint members. This can have a significant effect on the weight of our products.
9. *Excessive cost*: The cost of many products is proportional to the number of assembly operations. Correct preload means fewer fasteners and lower manufacturing costs—as well as lower warranty and liability costs.

Notice that in all of the above we want correct preload, not just preload. Too much will hurt us—so will too little.

In most situations we also want uniform preload, although this will usually be less important than a particular preload. Loading a group of fasteners irregularly can warp or damage joint members or gaskets. Nonuniform preload will also mean that only a few of the bolts carry the external loads. If they don't share the burden planned for them by the designer, the joint may fail.

7.1.2 How Much Preload?

We always want the maximum possible preload, but in choosing this, we must consider:

- Strength of the bolt and of the joint members under static and dynamic loads
- Accuracy with which we expect to tighten the bolts
- Importance of the joint, i.e., the factor of safety required
- Operating environment the joint will experience in use (temperature, corrosive fluids, seismic shock, etc.)
- Operating or working loads which will be placed on the joint in use

Again, we'll consider details of these requirements in later chapters, and will then reconsider the important question, "How much preload do I want?" in Chapters 17 through 19. In this and the next two chapters, however, we'll examine the problem of tightening the bolts accurately; we want enough preload, but we can't stand too much. This implies control of some sort.

7.1.3 Factors That Affect the Working Loads on Bolts

We'll start our detailed study of the control problem soon, but first, to see how control-at-assembly enters the picture, let's quickly summarize the main factors which will determine the "working" loads in our bolts. Important though control at assembly is, we don't want to lose sight of the fact that it's only one of our many concerns. All of the following will influence our results.

1. *Initial preloads*: The preloads we develop in those bolts when we first tighten them one at a time. Initial preload will be the principal subject of this and the next two chapters.
2. *Sequence/procedure*: The procedure with which a group of bolts are tightened can affect final results substantially. Procedure includes such things as the sequence with which they're tightened, whether they're tightened with a single pass at the final torque, or in several passes at steadily increasing torques, etc.
3. *Residual preloads*: The preloads left in the bolts after embedment and elastic interactions.
4. *External loads*: External loads add to or subtract from the tension in the bolts, and therefore from the clamping force on the joint. Such loads must be predicted and accounted for when the joint is designed and when the "correct" preload is chosen. External loads are created by such things as pressure in the pipeline or engine, snow on the roof, inertia, earthquakes, the weight of other portions of the structure, etc.
5. *Service conditions*: Severe environments can affect operating conditions in the joint and bolts. This is especially true of operating temperatures. These can create differential expansion or contraction, which can significantly alter bolt tensions and clamping force. Corrosion can cause change as well. Contained pressure will affect clamping forces.
6. *Long-term relaxation*: There are some long-term relaxation effects that must also be considered: relaxation caused by corrosion, or stress relaxation or creep, or vibration. And again, we want correct bolt loads for the life of the joint, not just for a while.
7. *The quality of parts*: We won't get correct preload, or satisfactory performance from the joint, unless the parts are the right size, are hardened properly, and are in good condition. This factor can't be handled separately; it gets in the act by affecting the others. If the bolts are soft, for example, we won't get the expected preload for a given torque, and relaxation will be worse. If joint members are warped or misaligned, it may take an abnormal amount of tension in the bolts (created by an abnormal amount of preload) to create the necessary clamping force between joint members.

That's only a partial list but it covers the main factors. Obviously, we've got our work cut out for us if we expect to overcome all of the possible problems. But we have no choice except to try—or change professions. The more we know about the possibilities and probabilities, the better our chances of success. We're going to start by looking at the most common and popular way to control that all-important initial preload on which all the other factors depend.

7.2 TORQUE VERSUS PRELOAD—THE LONG-FORM EQUATION

Whenever we tighten a bolt we perform two acts. We "do" the job—we tighten that bolt; and we "control" the job—we tighten it to a certain point and then make a decision to stop. Since the threaded fastener is designed to be tightened by twisting the nut with respect to the head, or vice versa, we usually find it convenient to do the job by applying a torque. This does not mean, however, that we must control the job with torque. We have, as you will see, many options when it comes to control. Let's look at some.

When we tighten a bolt we apply "torque" to the nut, the nut "turns," the bolt "stretches," creating "preload."

As we'll see, we can enter this sequence of events at any point to control the thing we're interested in—which is always preload. We can control, in other words, through torque or through turn or through stretch or through a combination of these things. In a few special situations we can control through preload directly.

In general, the closer we enter the chain to preload itself, the more accuracy we can achieve—and the more it will cost us.

We will find it easiest and least expensive to control preload with torque or turn, because these are the inputs to the system. Obviously, we'd like to be able to predict results; to predict the amount of preload we get when we tighten a bolt. There are two well-known equations that give us a shot at doing this, though, as we'll see, with less than perfect results. The first, which we'll look at now, is called the long-form torque–preload equation. As Srinivas has shown, this is a simplification of the more complete, but virtually impossible to use work–energy equation (Equation 2.29) given in Chapter 2. As a matter of fact, even this long-form simplification presents the user with some problems, but it is, nevertheless, widely used. Let's take a look at it. Experience and theoretical analysis say that there is usually a linear relationship between the torque applied to a fastener and the preload developed in a given fastener, as in Figure 7.1. In other words,

$$\text{Torque} = (\text{preload}) \times (\text{a constant})$$

$$T_{\text{in}} = F_{\text{P}} \times C \tag{7.1}$$

But what's the constant? A number of equations have been derived that attempt to define it. Here's one which has been proposed by Motosh [1]:

$$T_{\text{in}} = F_{\text{P}}\left(\frac{P}{2\pi} + \frac{\mu_{\text{t}} r_{\text{t}}}{\cos\beta} + \mu_{\text{n}} r_{\text{n}}\right) \tag{7.2}$$

where

T_{in} = torque applied to the fastener (in.-lb, mm-N)
F_{P} = preload created in the fastener (lb, N)
P = the pitch of the threads (in., mm)
μ_{t} = the coefficient of friction between nut and bolt threads
n = the effective contact radius of the threads (in., mm)
β = the half-angle of the threads (30° for UN or ISO threads)
μ_{n} = the coefficient of friction between the face of the nut and the upper surface of the joint
r_{n} = the effective radius of contact between the nut and joint surface (in., mm)

This equation involves a simplification, and so the answer it gives us is in error by a small amount. But I think it is the most revealing of the so-called long-form equations that have been proposed.

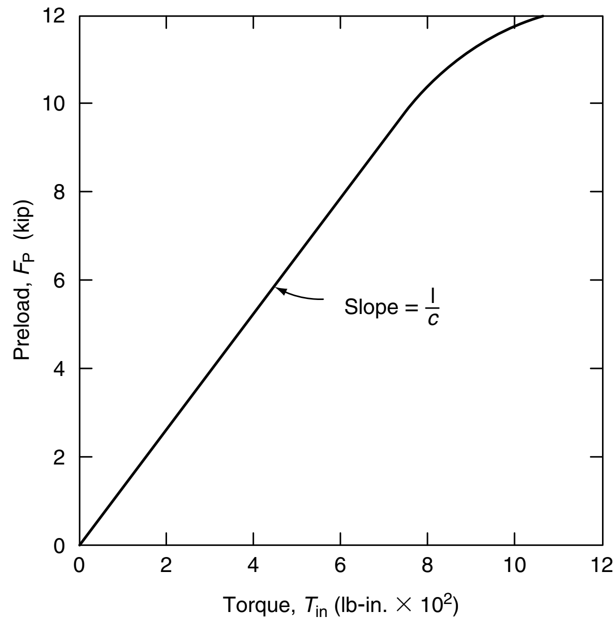


FIGURE 7.1 Normal relationship between applied torque (T_{in}) and achieved preload (F_P). Note that the straight line becomes a curve at the top when something—the bolt or joint—starts to yield. The bolt is an SAE 5429, Grade 8, ⅜–16 × 2.

The equation shows that the input torque (left side of the equation) is resisted by three reaction torques (three components on the right-hand side). These are as follows:

$$F_P \frac{P}{2\pi}$$

is produced by the inclined plane action of nut threads on bolt threads. This could be called the "bolt stretch component" of the reaction torque. It also produces the force, which compresses the joint and the nut, and it is part of the torque, which twists the body of the bolt.

$$F_P \frac{\mu_t r_t}{\cos\beta}$$

is a reaction torque created by frictional restraint between nut and bolt threads. It also provides the rest of the torque, which twists the bolt (unless there is some prevailing torque—discussed later—which would add a third component to the twist torque).

$$F_P \mu_n r_n$$

is a reaction torque created by frictional restraint between the face of the nut and the washer or joint.

If you plug typical fastener dimensions into Equation 7.2, then assume a coefficient of friction and input torque, you can compute the magnitude of each of the three reaction components separately. If you do this, you will find that the nut friction torque is approximately 50% of the total reaction, thread friction torque is another 40%, and the so-called bolt stretch component is only 10%, as shown in Figure 7.2.

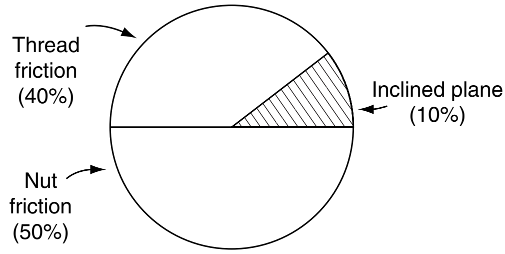


FIGURE 7.2 Relative magnitudes (typical) if the three reaction torques oppose the input torque applied to a nut.

Note that Figure 7.2 is just another way of illustrating the point made in Figure 6.4, where we were primarily concerned with charting the way in which input work was absorbed by the bolt and joint. Since input work is equal to torque times turn of nut, there's a linear relationship between energy and torque. Both Figures 6.4 and 7.2 give an oversimplified—but very useful—summary of the real situation, however, by suggesting that the only energy (or torque) losses are those due to thread or under-nut friction. In fact, there are other losses and we're about to discuss some of these.

When we use the torque equation or the energy equation, however, we're concerned not with how much of the input goes into preload, but by the degree of control we can maintain over this process. And here we run into all sorts of difficulties.

Remember the "leverage" problem we discussed in Chapter 6. Let's assume that the coefficient of friction between nut and joint surface is 10% greater than average. As we'll see, a 10% variation in friction is common; it can be caused by all sorts of things. This 10% increase raises the torque required to overcome nut–joint friction from 50% to 55% of the input torque. Taken alone, that would be no problem; but where does that extra torque come from? It can't come from the operator because he has no way of telling that this set of parts is absorbing more torque between nut and joint. It won't come from the thread friction component, unless the coefficient of friction at that point decreases magically to offset the other increase. So the only place it can come from is the inclined plane or bolt stretch component, the preload component—the thing we're after when we tighten the bolt. But taking an extra 5% of the input torque away from the stretch component reduces that component from 10% of the input to 5%. That's a 50% loss in bolt tension, thanks to a 10% increase in nut friction—a very bad leverage situation. And finding things that increase nut or thread friction is not difficult, as we'll see next.

7.3 THINGS THAT AFFECT THE TORQUE–PRELOAD RELATIONSHIP

Now let's take a closer look at some of the many factors which affect the amount of initial preload we get when we tighten a fastener. Many of these factors were included in our discussion in Chapter 6—and in the block diagram of Figure 6.28—but we'll consider them at greater length now, and will add some new ones!

7.3.1 Variables That Affect Friction

The coefficient of friction is very difficult to control and virtually impossible to predict. There are some 30 or 40 variables that affect the friction seen in a threaded fastener [30]. These include such things as

Hardness of all parts
Surface finishes
Type of materials
Thickness, condition, and type of plating, if present
The type, amount, condition, method of application, contamination, and temperature of any lubricants involved
Speed with which the nut is tightened
Fit between threads
Hole clearance surface pressures
Presence or absence of washers
Cut versus rolled threads

A word about the last item, cut versus rolled threads. Ron Winter of Tennessee Eastman has presented data showing the results he obtained by tightening separate groups of bolts: one group having cut threads, the other rolled threads. The bolts with rolled threads achieved a higher average preload for a given torque, and the bolt-to-bolt preloads were less scattered than the bolts with cut threads. Since a lower average and more scatter are characteristic results for any effect which increases thread friction, I assume that this item should be included in the above list.

In any event, you can see that there are many factors that affect friction, and therefore preload. Some of these factors can be controlled to some degree—but complete control is impossible.

Together, all of the factors listed above determine what I call "control accuracy," the ability of the variable we've selected for control—in this case, torque—to create the thing we're after, which is always preload. As far as torque is concerned, conventional wisdom—and much experience—suggests that a given torque will create a given initial preload with a scatter of $\pm 30\%$ in a large group of as-received bolts.

7.3.2 Geometric Variables

Friction is often cited as the only villain in the torque–preload equation, but that is not the case. Although we think we know the pitch of the threads, or the half-angle, or the effective contact radii between parts, in practice we have surprising variations in all of these things.

The bolt is not a rigid body. It is a highly stressed component with a very complex shape and severe stress concentrations. The basic deformation is usually elastic, but there are always portions of the bolt—for example, in thread roots and the like—which deform plastically, altering geometric factors r_t and pitch.

The face of the nut is seldom exactly perpendicular to the axis of the threads. Holes are seldom drilled exactly perpendicular to the surface of the joint, so contact radius r_n is usually unknown. Some experiments indicate that these factors can introduce even more uncertainty than does friction.

The Bolting Technology Council (now Committee F16.96 of the ASTM) sponsored a research study of the torque–preload relationship [29]. The study was done at École Polytechnique in Montreal. The purpose of the study was to find an economically viable way to conduct bolting experiments, the problem being that very large numbers of variables are often going to affect results.

Eleven variables that were believed to have a significant impact on the torque–preload relationship were selected for the École Polytechnique study. Taguchi methods were used to statistically design the experiment. The variables chosen included "perpendicularity."

Some bolts were tightened against parallel joint surfaces; others were tightened against joints having a 5° taper. Results of the experiment, which were confirmed by a second round, showed that perpendicularity affected the amount of preload achieved for a given torque more than did any other variable—including whether or not a lubricant had been applied to the threads and nut face.

7.3.3 Strain Energy Losses

All of the above variables are at least visible in the long-term equation. There are other sources of error, however, to which this torque–balance equation is blind. When we tighten a nut, we do work on the entire nut–bolt–joint system, as we saw in Chapter 2. Part of the input work ends up as bolt stretch or friction loss, as suggested by the long-form equation; but other portions of the input work end up as bolt twist, a bent shank, nut deformation, and joint deformation. The "true" relationship between input torque and bolt preload, therefore, must take these outputs into account, as we attempted to do in Equation 2.29. In one extreme case, for example, if the threads gall and seize, input torque produces just torsional strain and no preload at all. The long-form equation would suggest that all is lost in thread friction torque for this situation (infinite coefficient of friction in the threads). This isn't true, but thread friction torque is a twist component, so the equation doesn't lie. It's just that the result is strain energy, not heat. In fact, I don't mean to imply that the torque–balance equation is incorrect. It does, indeed, describe the action and reaction torques on the system correctly. But you will get into trouble if, as is common, you then add, "every part of the input energy which is not converted to preload must end up as friction loss because the equation shows only preload or friction terms." The torques are only cam action or friction torques, but what this means from an energy distribution standpoint is not revealed by the torque equation.

7.3.4 Prevailing Torque

Another factor not included in the long-form equation is prevailing torque: the torque required to run down a lock nut which has a plastic insert in the threads, for example. The insert creates interference between nut and bolt threads, and thereby helps the fastener resist vibration. The torque required to overcome this interference doesn't contribute to bolt stretch. It might be considered an addition to the thread friction component of torque, but it is a function of the design of the lock nut, and of the materials used, as well as the geometry, so it's best handled as a separate term, as suggested below:

$$T_{\mathrm{in}} = F_{\mathrm{P}}\left(\frac{P}{2\pi} + \frac{\mu_{\mathrm{t}} r_{\mathrm{t}}}{\cos\beta} + \mu_{\mathrm{n}} r_{\mathrm{n}}\right) + T_{\mathrm{P}} \tag{7.3}$$

where T_{P} = prevailing torque (lb-in., N mm), and all other terms were defined earlier.

Note that the prevailing torque is not a function of preload, the way all the other terms are. Note too that prevailing torque may not be a constant; it may change as the lock nut is run farther down the bolt, or is reused.

7.3.5 Weight Effect

Heavy or misaligned joint members resist being pulled together. This may not affect the torque–preload relationship, but it will reduce the amount of input torque, which ends up as clamping force between joint members.

7.3.6 Hole Interference

If the hole is undersize or misaligned it will take some effort merely to pull the bolt through the hole. This, too, can reduce the amount of torque available to create bolt preload.

7.3.7 Interference Fit Threads

If threads are damaged, or if they're designed to have zero clearance, it can take some torque to run the nut down on the loose bolt.

7.3.8 The Mechanic

Lest we forget, there are people involved here too. The results we get for a specified torque will depend very much on whether or not the person using the wrench has been well trained, knows what he's doing, cares about doing it right, can reach the bolts easily, can see the dial gage on his wrench, etc. The operator can be a more important factor than all of the others combined.

7.3.9 Tool Accuracy

We must also remember that tools aren't perfect. They will produce a requested torque with some tolerance or error, depending upon their construction, the accuracy with which the gage reports their output, how recently they have been calibrated, etc.

7.3.10 Miscellaneous Factors

There are many other factors that have been found to have some effect on the torque–preload relationship. Joseph Barron of the Newport News Shipbuilding Co. reported on a situation in which a regular hex nut, being used without a washer in an oversized hole, dug into the surface of the joint "like a plow." That would drive the nut factor through the roof. Many other, more common, factors have been identified, which have generally a smaller effect than the ones listed above. Some years ago, for example, R. Stewart of the Wright-Patterson Air Force Base gave me a list of 75 variables, which they had found had a statistically significant impact on the torque–preload relationship [30]. The list included most—but not all—of the factors listed above, plus things like type, thickness, and consistency of plating; the type of bolt head; the treatment of the hole, hole finish, hole concentricity, hole size, countersunk angle; gaps; burrs; type of wrench used to tighten the bolts; whether it was torqued from head or nut ends; number of times the bolt and nut had been used; number, type, and size of the washers used if any; etc. The list ended with item 76: "Any combination of the above."

The École Polytechnique study mentioned above also included, as test variables, such factors as lubricant, grade of bolt, type of wrench used, whether or not the fastener was covered with rust, whether or not it was plated, the number of times it was tightened, full versus partial thread engagement, and the stiffness of the joint in which the bolts were tightened. All were found to have some effect on the amount of preload achieved for a given torque, but most had a relatively—and sometimes surprisingly—small effect. Corrosion, joint stiffness, and amount of thread engagement were expected to have a fairly large effect, for example, but did not.

Since each of the effects listed above is itself affected by many secondary variables, you can see that literally hundreds of factors can influence the results when we tighten a single bolt. As we saw in the last chapter, additional factors—such as elastic interactions—further complicate our lives when, as usual, we tighten not just one but a group of bolts.

7.4 TORQUE VERSUS PRELOAD—THE SHORT-FORM EQUATION

There's another equation which I feel is more useful than the long-form equation. It's called the short-form torque–preload equation and it boils everything down to the "fact" that the initial preload created in a bolt is equal to the applied torque divided by a constant. Simple—but only if we know the constant! Remember, we are trying to define the constant in the torque–preload equation. The so-called short-form equation gives us

$$T_{\text{in}} = F_{\text{P}}(KD) \tag{7.4}$$

where

T_{in} = input torque (lb-in., not lb-ft; N-mm)
F_{P} = achieved preload (lb, N)
D = nominal diameter (in., mm)
K = nut factor (dimensionless)

If a prevailing torque fastener is used, the equation must be written

$$T_{\text{in}} = F_{\text{P}}(KD) + T_{\text{P}} \tag{7.5}$$

The discussion that follows assumes no prevailing torque, which is usually the case.

Note that the nut factor K is not a coefficient of friction. It is, instead, a general-purpose, experimental constant—our old friend the bugger factor—which says that "when we experimentally applied this torque to the fastener and actually measured the achieved preload, we discovered that the ratio between them could be defined by the following expression":

$$\frac{T_{\text{in}}}{F_{\text{P}}} = KD \tag{7.6}$$

The nice thing about K is that it summarizes anything and everything that has affected the relationship between torque and preload in our experiment—including friction, torsion, bending, plastic deformation of threads, and any other factor that we may or may not have anticipated.

The un-nice thing about K, of course, is that it can only be determined experimentally, and experience shows that we really have to redetermine it for each new application. Even then it is not a single number. Experience shows that for accurate prediction we have to make a number of experiments to determine the mean K, standard deviation. Having done this, however, we can indeed predict the minimum and maximum initial preload we're going to achieve for a given input torque, at a predictable confidence level. We cannot do this with the long-form equation.

Figure 7.3 will give you some idea of the variations you might expect to encounter in the K value for as-received steel fasteners. The histogram in Figure 7.3 is a compilation of the Ks reported in the literature obtained in a computerized search, plus the Ks reported by Raymond Engineering field service engineers on maintenance and construction sites. The standard deviation for this K data is 0.05, and the mean is 0.199. Plus or minus three standard deviations, therefore, takes us from a K of 0.15 to a maximum K of 0.25.

The data suggest that we can expect a scatter of $\pm 26\%$ in the preload achieved at a given input torque. That is very close to the $\pm 30\%$ figure you will see quoted by many authors; and a 0.199 mean is equal to the 0.2 value usually cited for steel fasteners used in steel joints.

It's possible, as Shigley did, to combine the long-form and short-form equations mathematically, and so end up with an expression for K in terms of fastener geometry and the

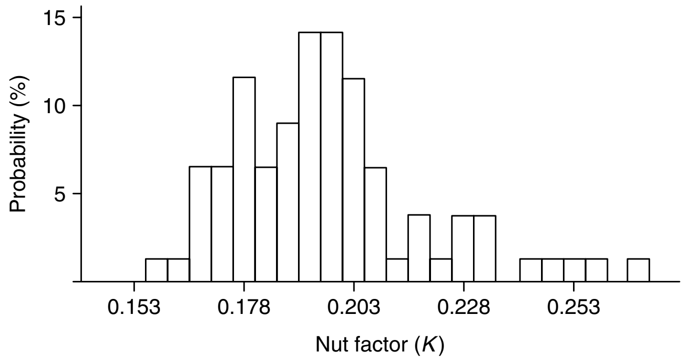


FIGURE 7.3 Histogram of *K* values reported for as-received steel fasteners from a large number of sources.

coefficient of friction [2]. There's nothing wrong with this approach from a mathematical standpoint, but I think it destroys all that is useful about *K*. As long as we treat *K* as an experimental constant it can help us estimate the relationship between torque and preload—if we make the right experiments and if we interpret them correctly. These are big "ifs." Finding a mathematical equivalent for *K*, however, just turns the short-form equation into another version of the long-form equation; and since we can't solve one, we really can't solve the other. Let's free *K* from any preconceptions about where it comes from. Ultrasonic techniques, to be discussed in Chapter 9, make on-site measurement of *K* a practical reality, and so drastically increase the usefulness of *K* and of the short-form equation.

7.5 NUT FACTORS

7.5.1 Some General Comments

The thing we now call the nut factor has been with us for many decades but the way it has been used has changed. Shigley called it a "torque coefficient" and used it to summarize the long-form equation [2]. It was used, in other words, to describe the theoretical relationship between torque and preload in mathematical terms. Later it was recognized as a convenient way to summarize the results of the torque-preload relationship when estimating the torque required for a particular job, or when reporting the results of an experiment. That is how I've used it in previous editions of this text. Recently, however, I've learned that people who make lubricants and antiseize compounds are using the nut factor to qualify their products. They print the expected nut factor on the can in which you receive the lube. The implication is that this lube will thereby control the torque–preload relationship, in a manner defined by that *K*, under all practical conditions. That's a big order. Each of the lubes so qualified, however, was also expected to do other things not usually associated with the torque–preload relationship: namely prevent galling, prevent or reduce the build-up of rust, facilitate disassembly even after long-term exposure to high temperatures, etc.

The Pressure Vessel Research Committee (PVRC) sponsored a research program at École Polytechnique entitled "Nut Factor Claims vs Experimental Reality." [39] The nut factors and other characteristics of six high temperature, antiseize lubricants were evaluated under a variety of conditions. One was a ceramic-based material, two each were nickel and copper based, the sixth was based on molybdenum disulfide. The moly was rated for use to 750°F (400°C); the others were supposed to be usable in even higher temperatures: ceramic based as high as 3,000°F (1,650°C); copper based 1,800°F (982°C); nickel based 2,400–2,500°F

(1,315°C–1,370°C). Only the moly-based material, however, prevented galling and therefore facilitated disassembly after one week of exposure to a temperature of 650°F. Break out (disassembly) torques for some lubricants were even higher than break out torques for unlubricated bolts.

The torque–preload relationship was measured at ambient temperatures up to 390°F (200°C) to evaluate the accuracy of the nut factor claimed for a given material: the *K* printed on the can. (The PVRC does not expect people to tighten bolts at temperatures exceeding 100°F.) Here's a summary of their results which, in my opinion, show surprisingly little scatter.

Ceramic-based lube (*K* not claimed): $K = 0.183 \pm 13\%$
Copper-based lube (*K* of 0.16 claimed): $K = 0.165 \pm 16\%$
Copper-based lube (*K* of 0.12 claimed): $K = 0.166 \pm 12\%$
Moly-based lube (*K* of 0.10 claimed): $K = 0.120 \pm 31\%$
Nickel-based lube (*K* not claimed): $K = 0.169 \pm 18\%$
Nickel-based lube (*K* of 0.15 claimed): $K = 0.142 \pm 24\%$

Note that the high end values of several of these nut factors come close to or even overlap the 0.2 considered the typical nut factor for unlubricated steel on steel fasteners. So these lubes don't affect the lubricity as much as we might expect, or as much as suggested by the numbers printed on their cans. Moly was an exception to this.

Surprisingly, the nut factor for the moly material decreased steadily in almost straight line fashion from 0.15 at the minimum test temperature of 23°F (−5°C) to 0.06 at the maximum test temperature of 390°F (200°C). Nut factors for the other materials might be less at mid-range temperatures than at either extreme, for example, but showed no strong pattern of growth or decay.

Nut factors were found to vary with the diameter of the bolts on which the lubes were used. Larger diameters always resulted in larger nut factors. For example the nut factor on a ⅞ in. bolt thickly lubed with a nickel-based antiseize compound was 0.2, while that on a similarly lubed ⅝ in. bolt was only 0.1. The amount of torque applied to a fastener also affected the nut factor: the nut factor decreased as torque increased.

In summary then, some lubricant manufacturers are now qualifying their products by nut factor, but these must be seen as approximations only. Let's take a closer look at *K* to see why.

7.5.2 Nut Factor Examples and Case Histories

The nut factor as defined by the short-form torque–preload equation sums up everything which affects the torque–tension relationship and so constitutes an ideal way to report on or analyze the results of torque control procedures. If we had a handy table of the mean *K*, and scatter in *K*, associated with various procedures or lubricants or types of tools or combinations of such things, we'd have everything we need to pick tools, procedures, lubricants, etc. for our own applications. As already mentioned, however, an accurate *K* for a new application must be derived by experiments which measure torque and tension, and therefore derive *K*, on that application itself, not on similar applications.

Many investigators have found, in fact, that nut factors determined on a sample or prototype joint, in a laboratory, can often differ significantly from nut factors determined on the actual joint in the field or on a production line. This merely reflects the fact that the nut factor does indeed summarize such things as tool accuracy, operator skill, bolting procedure, etc. as well as the more obvious factors such as lubricity and condition of the threads.

If you need further proof that the nut factor is a soft number, which must be used with caution, consider the following case histories.

Brookhaven National Laboratories measured the coefficient of friction between heavily loaded metal surfaces coated with a number of thread lubricants. They tested, for example, molydisulfide lubricants received from four different sources, and tested each one wet and dry to simulate bolted joint conditions. The coefficients of friction ranged from 0.026 to 0.273 [3], a variation of over 10:1. Other tests reported by the NRC [4] on graphite-based lubricants revealed a 3:1 spread in coefficient: copper–graphite and nickle–graphite lubes had a 2:1 range. These are coefficient of friction tests, not nut factor tests, but had the same lubes been tested in bolts the nut factor would presumably have shown similar or even greater dispersion.

A diesel engine manufacturer reported privately that the torque required to achieve a desired preload in engine head bolts increased by 50% with four reassembly operations using the same parts (and re-lubing them each time). Preloads were measured accurately, with ultrasonic equipment, described in Chapter 9. The nut factor, in this case, increased 50% with reuse of the fasteners. Field studies on a large number of 3 in. diameter fasteners in a nuclear power plant revealed the same trend: a steady increase in the torque required to achieve a given, ultrasonically measured preload (meaning a progressively higher nut factor) when the fasteners were reused.

An aerospace manufacturer, however, obtained very different results. A given ⅞ in. MP35N bolt was tightened, loosened, and retightened repeatedly to a load of 46,000 lbs. The torque required during 20 such re-tightenings dropped from 500 lb-ft to less than 200 lb-ft as shown in Figure 7.4. The nut factor in this case decreased with repeated re-tightenings.

A completely trustworthy nut factor, therefore, is like the Holy Grail—something everyone yearns for, but which no one will ever find. All of this means that we're back in a box, which is becoming painfully familiar. We'd like to pick a material property or a gasket stiffness or now a nut factor from a handy table and then proceed with our design or maintenance work; but we

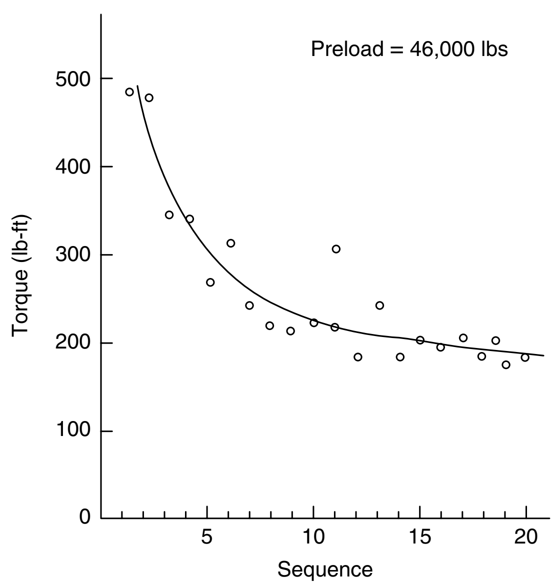


FIGURE 7.4 Torque required to produce the same 46,000 lbs of tension in a single bolt, tightened 20 times in the same hole. Note that the bolt, a ⅞ in. MP35N fastener, became easier to tighten as it was reused.

find every table preceded by warning labels saying, "For General Use Only: Make Experiments of Your Own to Determine the True Values for Your Application."

Given the fact that most bolted joints are grossly overdesigned, however, and the fact that an occasional failure would be acceptable or the fact that most people can't afford the time or money required to make experiments, a table is still welcome. You'll find some nut factors, therefore, in Table 7.1. These data merely report results obtained by others under conditions that will usually differ from the conditions you face. In most instances not enough information is given on the original conditions to allow much comparison anyway. But the data will give you an order-of-magnitude feeling for the way various lubes, fastener materials, rust, etc. affect the torque–tension relationship.

TABLE 7.1
Nut Factors (K)

	Reported Nut Factors		
Fastener Materials and Coatings	**Min.**	**Mean**	**Max.**
Pure aluminum coating on AISI 8740 alloy steel [23]	0.42	0.52	0.62
Electroplated aluminum on AISI 8740 alloy steel [23]	—	0.52	
As-received, mild or alloy steel on steel	0.158	0.2	0.267
As-received, stainless steel on mild or alloy steel		0.3	
As-received, 1 in. dia. A490 [7] as received		0.179	
Very rusty[b]		0.389	
With Johnson 140 stick wax	—	0.275	—
Black oxided' ⅞ A325 and A490, slightly rusty [5]	0.15	—	0.22
Black oxide	0.109	0.179	0.279
Cadmium plate (dry)	0.106	0.2	0.328
Vacuum cadmium + chromate [23]		0.21	—
Copper-based antiseize	0.08	0.132	0.23
Cadmium plate (waxed)	0.17	0.187	0.198
Cadmium-plated A286 nuts and bolts [33]	0.15		0.23
Cadmium plate plus cetyl alcohol on A286 nuts and bolts [33]	0.11		0.16
Cadmium-plated nuts used with MP35N bolts [33]	0.18		0.29
Dag (graphite + binder) [25]	0.16		0.28
Dicronite (tungsten carbide in lamellar form)	0.045		0.075
Emralon (PTFE + resin) [25]	0.10		0.15
Everlube 810 (MoS_2/graphite in silicone binder) [24]	0.09		0.115
Everlube 811 (MoS2/graphite in silicate binder) [24]	0.09		0.115
Everlube 6108 (PTFE in phenolic binder) [24]	0.105		0.13
Everlube 6109 (PTFE in epoxy binder) [24]	0.115	—	0.14
Everlube 6122	0.069	0.086	0.103
Fel-Pro C54	0.08	0.132	0.23
Fel-Pro C-670	0.08	0.095	0.15
Fel-Pro N 5000 (paste)	0.13	0.15	0.27
Mechanically galvanized A325 bolts [5]			
As received	0.35	—	0.49
Clean and dry		0.46	
Slightly rusty°	0.36		0.39
Mechanically galvanized A325 bolts; lubed with 1 part water, 1 part Jon Cote 639 wax [5]	0.11		0.26
Hot-dip galvanized ⅞ A325 [5]			
As received	0.14		0.31
Slightly rusty	0.09		0.17
Clean and dry	0.09		0.37
Hot-dip galvanized ⅞ A325 lubed with 1 part water, 1 part Jon Cote 639 wax [5]	0.10		0.16

The data in the table are organized by thread coatings—lubricants, rust, platings, or as-received surfaces. This reflects the fact that lubricity has a major influence on the nut factor. A slippery surface will lead to a lower nut factor than a stickier one.

As a result, many people want to assign a different nut factor to each type of lubricant, and if other things are equal, this is possible. I do it in Table 7.1. But we must remember the earlier discussion of things that affect the initial preloads and the working loads in bolts. Many were lubricity factors, but not all. Operator errors and tool calibration were also involved, for example. And we also have the sobering Brookhaven studies in which they measured the coefficient of friction between heavily loaded (and undoubtedly controlled) metal surfaces—test blocks of some sort, not bolts. They found a 10:1 variation in lubricity of things called molydisulfide by a variety of manufacturers. So surface coatings are important, and are used to define our table, but we should remember that this is only one of the many factors that determine K.

As a result, the nut factor data in Table 7.1 are typical only. They may not represent your own application. In critical applications you should always develop your own nut factors by an appropriate experiment.

7.5.3 Coefficient of Friction versus Nut Factor

Lubricant manufacturers and test laboratories are usually interested in friction in general, not just friction in threaded fasteners. Because of this they report test results in terms of coefficient of friction (μ) instead of nut factor. We could use Equation 7.2 to convert coefficient data to a nut factor, but it's usually less trouble to use the following approximation:

K is approximately equal to μ plus 0.04.

For example, if we're told that a new thread lube provides a steel-on-steel coefficient of friction of 0.12, we can estimate that a fastener using that lube would have a nut factor of about 0.16. Again, this is only an approximation, but many factors in addition to friction affect the nut factor for a given application, so an approximation is useful.

7.6 TORQUE CONTROL IN PRACTICE

7.6.1 What Torque Should I Use?

Perhaps the most common question in bolting is "What torque should I use on this bolt?" The purpose of picking a good torque is to end up with a good preload, leading to a good clamping force on the joint. So, in order to answer the torque question we must first decide what clamping force is required in our application.

We're not ready to do that yet. We must first learn about bolting tools and the way they affect the results. And, more important, we must learn about bolt and joint failure modes, and the correlation between bolt tension, clamping force, and failure. We'll get around to an answer for the "what torque?" question—but not until Chapter 17.

We have, however, acquired one piece of information which will be vital to our final choice of assembly torque. We've learned that we can't count on that torque achieving a single value of tension in all the bolts we tighten. The resulting tensions will be scattered; we'll end up with a range of tensions as broad, perhaps, as one of the ranges of nut factor reported in Table 7.1. To pick a torque to use during assembly we'll pick an optimum or target preload, selected to avoid or minimize subsequent joint problems. Then we'll identify the mean nut factor, which best defines the conditions of our application. We'll use that nut factor and target preload in the short-form equation (Equation 7.4) to define the torque we'll specify for assembly purposes.

We'll also, if the joint is important, use the reported scatter in possible nut factor value to estimate the max.–min. range of preload that torque might give us. And we'll decide whether those maximum and minimum tensions could lead to joint problems. If so, then we'll have to find a better way to control the assembly process (to be discussed in the next few chapters)—or we'll have to redesign the joint to be more tolerant of scatter in clamping force.

The following discussion gives further case histories on nut factor and preload scatter, using actual data. But first: when discussing preload scatter we must be careful to distinguish between the initial preload achieved bolt by bolt as we first tighten them and the residual preloads in that same or a different group of bolts after all in the group have been tightened and embedment and elastic interactions have done their work. Many experimenters report only the first and lull us into a false sense of security. Residual scatter is always greater—and sometimes much greater—than the initial scatter.

7.6.2 Initial Preload Scatter

Figure 7.5 shows initial preload results for a large number of fasteners tightened one at a time. Note that the distribution of preload achieved for a given torque is skewed right in this case, rather than being Gaussian or normal [15].

If your sample is smaller, there is little you can predict about the distribution or range of results beyond what you know about the possible (but not certain) range in K. Since this can vary by $\pm 26\%$, the preload can vary from -21% to $+35\%$ of the value corresponding to the mean K.

The short-form equation is useful when we want to describe the relationship between applied torque and achieved preload as the fastener is being tightened, or at the instant the tightening process is completed. A tightening experiment might show the sort of scatter in nut factor suggested by the data in Table 7.1. If we record the final torque applied to a fastener, however, and then measure the residual some time after tightening, and then use the short-form equation to compute K, we'll usually conclude that the nut factor varies far more than "usual." The scatter in final tension may be much greater than that shown in Table 7.1.

The reason: postassembly relaxation effects further disperse the already scattered tensions created in bolts as they were tightened individually. Embedment relaxation and elastic interactions between bolts in a group are the main culprits. Both are discussed in the last chapter. As we saw, the resulting scatter in residual (as opposed to initial) tensions in the bolts can be 4:1 or 10:1 or worse. The short-form equation can be applied to such data; you're free to define the terms and conditions for your experimentally derived nut factors any way you like; but the scatter (and therefore the uncertainties) in the results becomes so great as to make

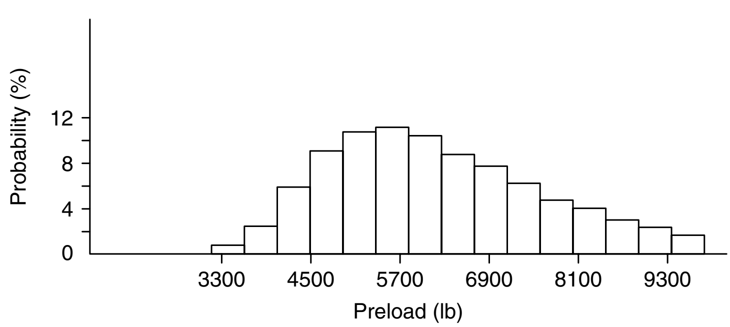


FIGURE 7.5 Histogram of the initial preload achieved in a large number of 5⁄16–24, SAE Grade 8 fasteners with a 2.3 in. grip length tightened to a uniform torque level. (Modified from Eshghy, S., *Fastener Technology*, July 8, 1978.)

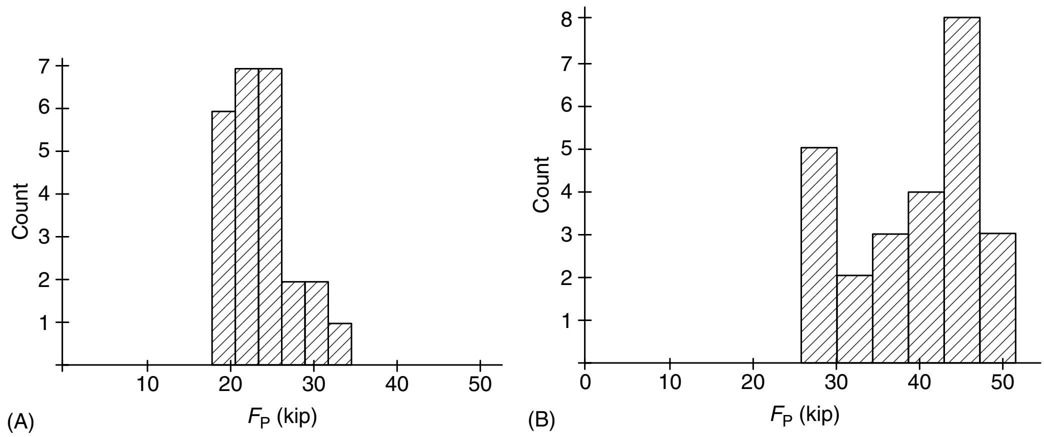


FIGURE 7.6 Histograms of the residual preload produced in 25, 1⅜–8 × 10 B7 studs coated with solid-film PTFE when torqued to (A) 400 lb-ft or in a second test to (B) 800 lb-ft.

your calculations almost useless. It's best to use nut factors only to describe the initial tightening of individual fasteners—groups of them, sure, but one at a time.

7.6.3 Low Friction for Best Control

Friction affects the efficiency with which input work on the nut is converted to preload. We're primarily interested, however, in the accuracy of this process. Does a lubricant on the threads increase or decrease the preload scatter for a given input torque?

If we use the long-form torque equation to compute the preload achieved in a given fastener for a given input torque at different coefficients of friction, we'll get the results shown in Figure 7.6. At first glance, Figure 7.6 suggests that a higher coefficient would provide more accurate control than a lower coefficient. A $\pm 20\%$ uncertainty in a coefficient of 0.4, for example, would mean a 14 kip scatter in preload; while a $\pm 20\%$ uncertainty in a coefficient of 0.1 would mean a 28 kip scatter in preload, although percentage changes in preload would remain about the same.

A similar curve can be drawn for the relationship between K and preload for a given torque. So, does low friction mean poor control?

No. In practice, lubricants usually reduce scatter in preload. Histogram A in Figure 7.7, for example, shows the scatter in torque required to bring 140, M12 steel bolts to the ultimate tensile load point when the bolts were lubricated with machine oil [8]. Note that the torque required for a given preload has a normal distribution, at least in this case.

Histogram B in Figure 7.8 shows the higher torques—and slightly greater scatter in torque—measured during similar tests on 140, M12 steel bolts, which had been cleaned in gasoline and dried. Since all 280 bolts had been taken to ultimate strength, the final preload was approximately the same in each, and "more scatter in torque" means "more variation in μ or K." The difference is not great, however, because machine oil isn't a very good thread lubricant.

Here's some practical advice: if certainty you need in clamp, just make sure those bolts are damp.

7.6.4 The Lines Aren't Always Straight

The equations predict, and many experiments confirm, that the relationship between applied torque and achieved preload is usually linear until something in the joint yields. There are many times, however, when this is not true. Sometimes thread lubricants migrate or break

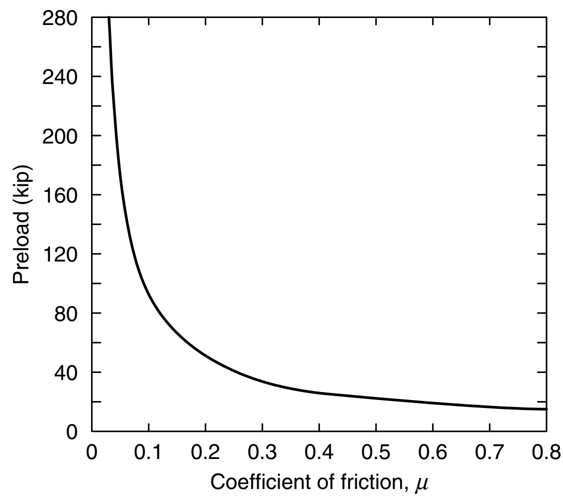


FIGURE 7.7 Theoretical preload achieved for a given input torque as a function of the coefficient of friction.

down, or minor galling occurs, so that it requires larger and larger increments of torque to produce the next increment of preload.

7.6.5 Other Problems

We can summarize the things we've talked about so far—the things which affect the torque–preload relationship—with an acronym, FOGTAR, as follows:

Friction
Operator
Geometry = FOGTAR
Tool Accuracy
Relaxation

Friction includes not only lubricant but also surface finish, speed of tightening, type of materials involved, and many, many more variables. Geometry includes not only the

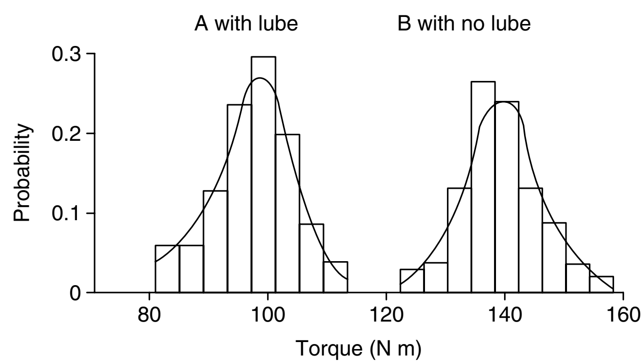


FIGURE 7.8 Histogram of torques required to tension (A) 140 lubricated bolts and (B) 140 unlubricated bolts to ultimate tensile strength. Bolts were M12, Steel St.3. (Modified from Bezborod'ko, M.D., et al., *Vest. Mash.*, 58, 1978.)

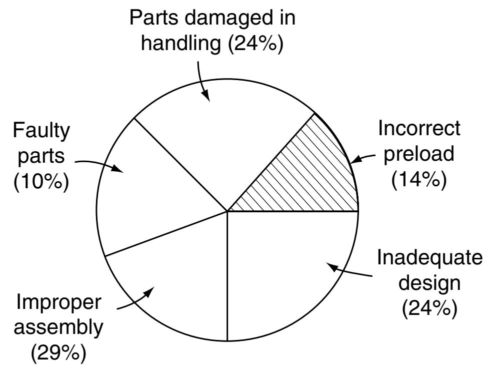


FIGURE 7.9 A summary of the causes of bolted joint failure on the Skylab program. All fasteners have been torqued. (Modified from Investigation of threaded fastener structural integrity, Final report, Contract no. NAS9–15140, DRL-T-1190, CLIN 3, DRD MA 129 TA, Southwest Research Institute, Project no. 15–4665, October 1977.)

manufacturing tolerances on parts but that important perpendicularity between nut face, hole axis, and joint surface. Relaxation includes embedment and elastic interactions, both of which occur as we tighten a group of bolts. Operator and tool accuracy are self-explained.

FOGTAR! It's probably the Tibetan word for trouble, and it reminds us that it's not just variation in friction which makes torque control less than perfect. It's FOGTAR and all that implies.

Remember this and you can't go wrong.
Your tools are weak and the FOGTAR's strong!

And that's not all. We must also worry about poor design, plus damaged or incorrect parts, if we're the project manager rather than the tool user. Figure 7.9 shows an example of the problems other things can cause. A study [14] of joint failures during live missions on the Skylab program showed that only 14% of the failures were caused by what the author of the final report called "incorrect torque" [14]. "Incorrect preload" would probably have been a more accurate term, and it would be the result of most of the factors we've discussed in this chapter.

In any event, 86% of the failures were traced to poor design, bad parts, or operator problems—and this on a program where the quality control activity must have been intensive, to say the least.

Our choice of the long-form or short-form equation, our guess for μ or K, and the accuracy with which we measure torque may have only a small influence on the number of joint failures we must face. To solve the total problem we must control design, FOGTAR, and parts—DEFOG TARP? Anyway, as we'll see later, some tools and control systems are designed to compensate for some, at least, of the FOGTAR problems—and even for faulty parts. Tool designers have gone well past tool accuracy in contemporary designs.

7.7 SOME TOOLS FOR TORQUE CONTROL

7.7.1 Some Generalities

A few of the large number of tools that have been developed for applying torque to threaded fasteners are shown in Figures 7.7 through 7.10. We'll look at some of them in detail in a minute but first, some definitions and generalities.

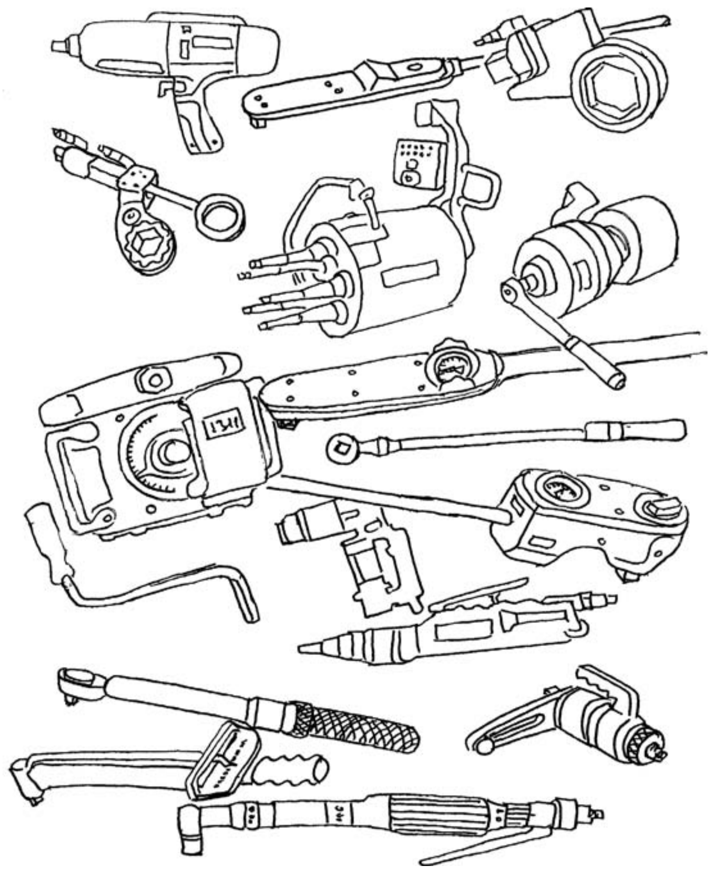


FIGURE 7.10 Sketch showing some of the very large number of manual and powered torque tools available today.

Torque range for manual and powered tools [36]: Conventional manual tools (wrenches) can be used to create torques as high as 500 ft-lb (680 N-m), but no one could do this repetitively. In production operations, therefore, such tools are usually limited to applications requiring torques of 20 ft-lb (27.2 N-m) or less. There are also unconventional manual tools, called geared wrenches or multiplier, with which the operator can create torques as high as 20,000 ft-lb (27,200 N-m) while turning an input crank which requires only a few ft-lbs of torque. Pneumatic and electric-powered tools are used to create torques ranging from 0.5 in.-lb (4.34×10^{-4} gm cm) to 1,000 ft-lb (1,360 N-m). The king of the jungle, hydraulically powered tools, are used for torques ranging from 500 ft-lb (680 N-m) to 20,000 ft-lb (27,200 N-m).

Power supplies: Most electric tools are driven by DC motors, but are powered by regular two phase AC. Multiple spindle electric tools, however, often require a three phase power source. Pneumatic tools are driven by compressed air, of course, and this should be lubricated to protect and extend the operating life of the tools. As a result these tools emit an oily mist and shouldn't be used in clean environments. Hydraulic tools require hydraulic power sources able to create pressures, in some cases, as high as 5,000 psi (34.5 MPa).

Tool maintenance [36]: Pneumatic tools require little maintenance if used with a properly lubricated air supply. DC-powered tools are generally less durable than air tools. A common problem: cable failure if the tools are not properly supported. Impact wrenches require surprisingly little maintenance, in spite of the noise and vibration they create. Pulse tools require regular changes of hydraulic fluid, gaskets and seals, etc.

Influence of the joint hardness on tool selection [38]: Tool selection will be based, in part, on a joint property called "hardness" because this affects the rate at which the final assembly torque builds up. Sheet metal joints, for example, are very hard; fasteners are short and, as we've seen in Chapter 5, retain very little potential energy. Nuts run down by powered tools slam into the joint and stop abruptly, so the tightening tools need to be able to react and stop almost instantaneously. By comparison, joints clamped by long bolts, especially gasketed joints, can be very soft. The nut continues to rotate even after it contacts the joint, so tools have more time to measure and react to the build-up of torque.

Tool overshoot: The unavoidable inertia of a high-speed, powered tool can lead to "overshoot," especially on hard joints. The tool fails to shut off fast enough and delivers a final torque slightly higher than that desired. This can be a problem in mass production applications where speed is often necessary [38]. The torque settings of the tools can sometimes be adjusted to compensate for overshoot, but the hardness of seemingly identical joints can vary enough to create an overshoot problem. If a sheet metal joint member is not perfectly flat, for example, its hardness may be reduced enough to eliminate overshoot. A tool adjusted to compensate for overshoot would, in this case, stop before a desired torque level had been reached. In other situations torque rate can be affected if a washer or gasket is inadvertently left out of the assembly.

Classification of joints: Automotive production people have defined three joint classifications [36]: Class A joints are safety related (wheels, brakes etc.), Class B joints are reliability related (engines), and Class C joints are those affecting customer satisfaction. These classifications can also influence the selection of a tool for a particular job.

7.7.2 Reaction Forces Created by the Tool

Tools that generate torque also, unavoidably, create shear loads on the fastener and reaction torques on the operator or on a supporting structure of some sort. The greater the torque the higher these reactions become and the more necessary it is to do something about them.

7.7.2.1 Shear Loads Created by Torque Wrenches

A torque wrench does not just create torque on the nut or bolt head. It also creates a side load, which is equal and opposite to the force with which the mechanic pulls the handle of the wrench. This reaction side load is necessary to establish what the professors call "equilibrium." (We better call it that, too!) The input torque—mechanic's pull times the length of the wrench—must equal the reaction torque created by thread and nut friction and by the inclined plane action of the threads, turning torque into stretch of the bolt. At the same time the sum of forces in the x, y, and z planes must each equal zero; action forces must be balanced by reaction forces. Fortunately, we only have to worry about the x plane if we choose our coordinates properly.

Let's look at an example. The mechanic is pulling on the handle of the wrench in Figure 7.11 with a force of 20 lbs. The distance between the center of the bolt and the mechanic's hand is 2 ft. Input torque is clockwise and is

$$T_{\text{in}} = 20 \times 2 = 40 \text{ ft-lb.}$$

For equilibrium the input moment—which equals the input torque—must equal the reaction moment or

$$\Sigma M = 0.$$

The fastener reacts with a 40 ft-lb torque in the counter-clockwise direction.

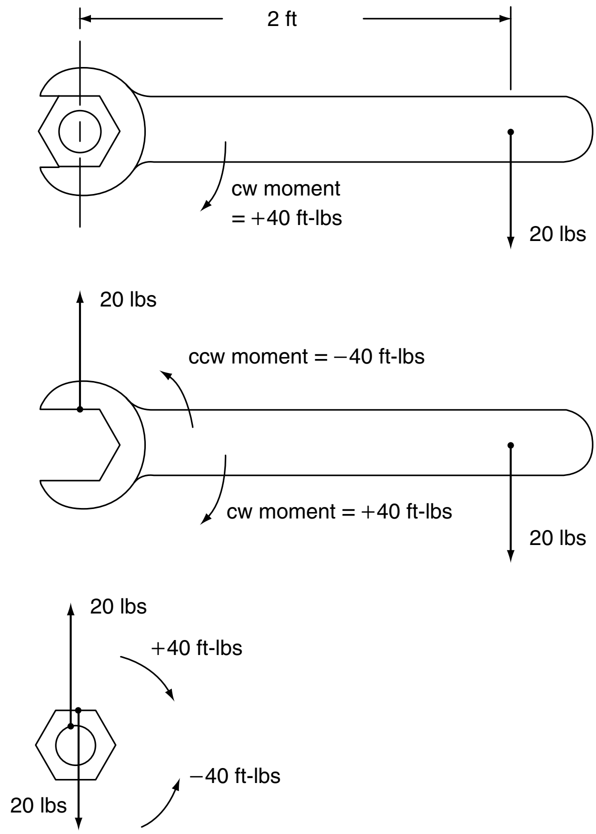


FIGURE 7.11 A manual torque wrench of any kind produces not only torque on a nut, but also a reaction side load equal to the force exerted by the mechanic on the handle of the wrench. The middle drawing here is a free-body diagram of the wrench. The lower drawing is a free-body diagram of the nut and shows that the reaction side load must be supported by the bolt (which in turn is supported by the joint member). Although the 20 lbs side load shown in this example is trivial and would have a negligible effect on the torque–preload relationship, other types of torque tools can create very large side loads of this sort, as described in the text.

At the same time the fastener must support the sideways pull of 20 lbs. If it didn't, the bolt, mechanic, and wrench would just move sideways. Looking at the free-body diagram of the nut we see that the act of tightening it has created a side or shear load of 20 lbs between the nut and the bolt. (This is supported by a 20 lbs force between the bolt and its hole.)

Anything which increases the contact pressure between male and female threads increases the frictional drag as the nut is turned around the bolt. The axial forces created here, as the bolt is stretched, will be much greater than this 20 lbs side load, but this example illustrates the forces that are created when any unbalanced torque tool is used to tighten a bolt. There are some situations, as we'll see later, where these reaction side loads can become very high.

7.7.2.2 Reaction Torques

Reaction torques are a far more serious problem than shear loads in most situations. As Newton has taught us, for every action there's a reaction so for every purposely created and desired torque there's an unavoidable and undesirable reaction torque. If the desired torque is

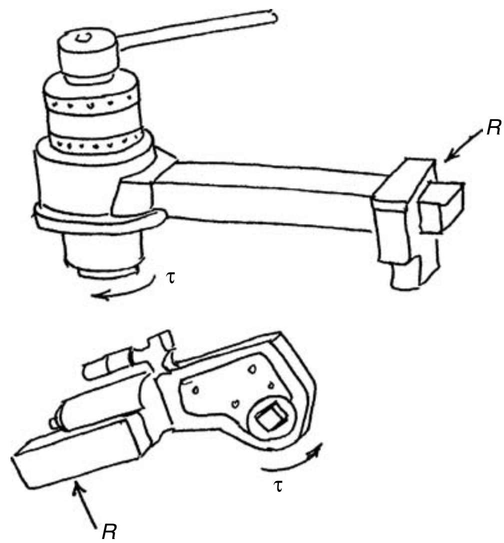


FIGURE 7.12 Tools that apply high torque to a fastener also generate high reaction torques on the operator or structure whose bolts are being tightened. The upper sketch shows a geared torque multiplier, with a reaction bar. The lower sketch shows an hydraulic wrench with a reaction foot. Tools of this sort can produce torques of hundreds or thousands of ft-lbs. Something must support these torques by providing reaction forces, *R*, in the direction shown.

in the clockwise direction, the reaction torque is counter-clockwise. If output torques are small, the reaction torques can usually be absorbed by the operator. Experts warn, however, that if the tool has a pistol grip, some means must be provided to absorb the reaction torque if it exceeds 18–20 in-lb ($16–17 \times 10^{-3}$ gm cm) or the operators wrist could be injured. Reaction support must also be provided for in-line or angle wrenches used to create torques of over 15 ft-lb (20 N-m) [36].

Tools producing high torques require significant reaction support structures. Things like torque multipliers and hydraulic wrenches require sturdy "reaction arms" like that shown in the upper sketch of Figures 7.7 through 7.12, or a "reaction foot" shown in the lower sketch. If the multiplier in that illustration were producing 2,000 ft-lb (2,720 N-m) and its reaction arm were one foot (0.305 m) in length something must be strong enough to generate a reaction force *R* of 2,000 lbs (8,896 N) to prevent the wrench from turning counter-clockwise. This something is usually the engine or reactor vessel or pipe flange whose bolts are being tightened. Reaction force requirements would be even higher for the hydraulic wrench shown in the same illustration if it was less than one foot in length.

So much for generalities; now let's take a closer look at some of the more popular tools.

7.7.3 In the Beginning—A Search for Accuracy

7.7.3.1 Manual Torque Wrenches

Man's quest for more reliable bolted joints probably started with the manual torque wrench—an attempt to get improved torque accuracy. Today, after years of development, there are many kinds of torque wrench, ranging in output from a few ounce-inches to 1000 lb-ft. Output torque accuracies range from $\pm 2\%$ to $\pm 20\%$ of full scale.

As soon as man had acquired some semblance of torquing accuracy, he discovered that accuracy was not enough. Operator problems—lack of skill, carelessness, etc.—often wiped out the gains made in accuracy. Many of the wrenches on the market, therefore, have been designed to reduce operator problems by providing such things as better gauges, presets, signal lights, audible outputs (clicks), etc. The latest manual wrenches are equipped with electronic measuring systems and digital readouts.

7.7.4 More Torque for Large Fasteners

The higher the torque, of course, the more difficult it becomes to produce it manually. Several tools, therefore, have been designed to help tighten larger fasteners. Those of us who assemble heavy equipment often need one or more of the following tools.

7.7.4.1 Torque Multipliers and Geared Wrenches

Torque multipliers are gearboxes that multiply the torque produced by a manual torque wrench. Typical ratios are 4:1 or 10:1, with a few up to 100:1 or so. Thanks to friction losses in the gear trains, multipliers tend to decrease the accuracy of a manual wrench somewhat, but they produce output torques of up to 83,000 lb-ft.

Note that the reaction side loads and reaction torques will tend to increase as the applied torque increases. Note too that something has to hold the gearbox—the multiplier—from rotating or moving sideways as it applies torque to the fastener. Multipliers, therefore, are equipped with a reaction arm, which leans against another bolt in the group, as for example the one shown in Figure 7.12 or against some part of the structure being assembled. Larger multipliers are equipped with pins in their base, which engage a reaction arm designed for a particular application or engage holes drilled in the joint member itself.

Geared torque wrenches combine the readout of a torque wrench with the gearing of a torque multiplier. This time the readout is of output torque (rather than input to a gear train), and the multiplication ratio is higher—ranging from 125:1 up to 2,400:1 on wrenches producing outputs of 600 lb-ft up to 20,000 lb-ft. The high gear ratio means that input torques are very low; the tools can be driven by nut runners. So in fact these could be called "nut runner multipliers" rather than "torque multipliers."

Like multipliers, geared wrenches must be provided with a reaction arm of some sort. If provided with a double-sided reaction arm—and if that reacts against two points in the structure which are also equidistant from the drive spindle—the tool will produce a pure torque couple on the fastener. As a result, such tools have been used to tighten the main spindle nuts of high-speed jet engines and other rotating equipment, where the concentricity of the final assembly is very important (to prevent vibration), and it's very useful, therefore, to avoid large side loads on the fastener and fastened parts during assembly.

7.7.4.2 Hydraulic Wrenches

Another way to get a lot of torque in a small space is to use a hydraulically actuated tool in which a piston drives a short, stubby ratchet wrench through as many cycles as necessary to tighten a bolt. This is probably the most popular type of production tool when torques in the range 1,000–5,000 lb-ft are required, although the tools are available in torques of up to 100,000 lb-ft of output. Output torque accuracies of $\pm 2\%$–10% of full scale are possible in most cases. This is one of the few power tools available for extreme torques (Figure 7.12). Most of these tools have one-sided reaction arrangements and do, therefore, exert fairly large side loads on the fasteners being tightened.

7.7.5 Toward Higher Speed

None of the tools described so far are very fast, obviously. For high speed production applications we must have something else. The most common "something else" is an electric or air-powered impact wrench or nut runner. Broadly speaking electric tools are usually preferred by production engineers today. They are quieter, cleaner, and ergonomically better than air powered tools in most applications. They are almost all powered by brushless DC motors. They're economical to run but tend to be more expensive and less robust than air tools. They're much preferred for applications torque and turn transducers used to monitor or control the assembly operation. We're warned, however, not to use DC electric tools at more than 85% of their rated capacity in mass production applications because they can overheat and burn out.

7.7.5.1 Impact Wrenches

Tiny hammers within impact wrenches give repeated blows on an output anvil, allowing a relatively small, lightweight, inexpensive tool to produce surprisingly high output torques—tens of thousands of pound-feet in some cases—at least on stiff joints. Impact wrenches are notoriously noisy and inaccurate. The amount of torque produced by a given tool on a joint, however, depends very much on the springiness of that particular joint. Even changing the length of the drive bit between wrench and socket can change output substantially—a short bit is a stiffer spring than a long one.

One nice feature of an impact wrench is the fact that it produces remarkably little reaction torque of the operator, because the impacts it generates care of such short duration. Impact wrenches also require less maintenance than other air or electric-powered tools. Because their action is intermittent impact wrenches cannot provide data output to an electronic control system.

The accuracy claimed for most impact wrenches runs from ±20% to ±40%, so they should not be used on safety or performance related (Class A or B) joints. Some impacts, however, are provided with torsion bar outputs that can be used to limit the final torques created by the tool. The bars twist enough at a predetermined torque to activate a shut-off switch. Torsion bars do not, however, reduce the scatter in torque created by these tools [37].

7.7.5.2 Pulse Tools

There's a family of tools called pulse tools, which apply torque to a fastener in a rapid series of pulses, but not with the violence of an impact wrench. The torque is not created by hammer blows but rather by hydraulic pulses. Air pressure to the wrench pressurizes a small quantity of contained hydraulic fluid which does the rest. These tools are much quieter than impact wrenches and, like them, create very little reaction torque on the operator. Another feature they share with their impact cousins: they cannot provide an electronic data stream to a computer.

7.7.5.3 Nut Runners

Things called nut runners are probably the most popular and widely used production bolting tool. They are fast, air- or electric-powered tools used to tighten fasteners ranging in diameter from ¼ in. to perhaps ½ in. (M6-M14) requiring torques ranging from 5–110 ft-lb (10–150 N-m). These tools have elongated handles to help the operator absorb the reaction torques, but a reaction bar or foot adapter of some sort is usually employed for repeated production operations involving the higher torques. Both in-line and right angle output spindles are common [38].

Smaller sizes of the same tool are often called screwdrivers and create torques so low that the operator can easily absorb the reaction torque which rarely exceeds 11 ft-lb (15 N-m). Many of the electric-powered tools are cordless and battery powered. They are quiet, clean and convenient, with torque accuracies approaching ±5%–10%.

Air powered, pneumatic nut runners contain a small rotary-vane turbine whose low torque and high speed are converted to usable output torque and speed by a multistage planetary gear train. Nut runners are available with outputs of up to 650 lb-ft or so. They are quieter than impact wrenches and, even in their simplest form, more accurate. This accuracy, however, depends upon the type of clutch used to disengage the tool when the desired toque has been reached. Here are some of the options [37]:

1. *Direct drive clutch*: Used for soft joints requiring little torque control. The operator controls the torque by controlling the flow of air.
2. *Adjustable cushion clutch*: A good, general purpose, torque-limiting clutch which can be used on both hard and soft joints.
3. *Positive jaw clutch*: Useful if the run down torque may exceed the final torque because this is a thread-forming screw or a vibration resistance one with a prevailing torque feature of some sort. (See Chapter 14.)
4. *Adjustable precision shut off clutch*: For critical applications requiring precise torque control or difficult-to-assemble joint materials such as composite or plastic.

One way to get higher speed, of course, is to tighten several fasteners simultaneously. Most air-tool manufacturers, therefore, provide multispindle as well as single-spindle tools. Multispindle tools react against one nut while turning another. Each nut being turned, in fact, acts as a reaction point for the other nuts being turned. This is convenient and makes it possible to tighten groups of fasteners very rapidly. But the short reaction distances can create substantial side loads. Consider the situation, for example, in which a two-spindle nut runner is applying 600 ft-lb of torque to two 1 in. diameter bolts which are 3 in. apart.

Torque = 600 ft-lb
Reaction arm = 3 in. = 0.25 ft
Reaction force = 600/0.25 = 2400 lbs

If the bolts are only 2 in. apart the reaction force rises to 3600 lbs.

If the tool is tightening a large number of bolts simultaneously my guess is that each bolt will see a different amount of reaction force, that these forces will be statically indeterminate, and that one or a couple of them could be even higher than would be suggested by considering them in pairs.

7.7.6 Add Torque Calibration or Torque Monitoring

Stall-torque or clutched air tools give some degree of output torque accuracy only as long as they are properly adjusted and within calibration. Air tools tend to wear rapidly, however, and good control on input air supply is not always possible. In some cases, operator skill and carelessness can also make a difference. One of the next steps on the road to better accuracy, therefore, is torque monitoring [16,19].

Torque calibration is the simplest way to monitor torque. The user periodically measures the output torque being produced by the tool, using a calibration stand of some sort. The length of time between calibrations depends on such things as the importance of the joint being tightened, the environment in which the tools are being used, the stability of the tools, etc. In some cases the calibration period is specified by a standard-setting body. Structural steel torque wrenches, for example, must be recalibrated daily, per the specification on structural joints issued by the Research Council on Structural Connections of the Engineering Foundation.

The calibrator most frequently used in structural work is shown in Figure 7.13. The heart of the device is a short hydraulic cylinder with a hole through the middle. A bolt is run

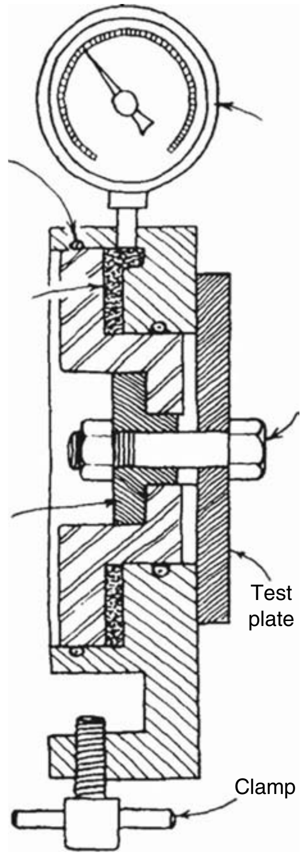


FIGURE 7.13 Skidmore–Wilhelm calibrator. A fastener is mounted in the calibrator (rather than in the actual joint) and is tightened by applying torque to head or nut. The instrument shows the actual tension achieved in the fastener under these conditions.

through the hole, a nut and washers are added, and the bolt is tightened. This raises the pressure in the cylinder, and a suitably calibrated pressure gauge interprets the increase in pressure in terms of clamping force. It's a simple, effective, and popular tool [20]. It's certainly accurate enough for most purposes, but the torque versus preload relationship is affected somewhat by the stiffness of the joint in which the fastener is used, as we'll see in later chapters. Since the stiffness of the hydraulic cylinder is a lot less than the stiffness of most joints, the accuracy of this calibrator would be a problem in critical applications.

One problem with calibration is the fact that the torque produced by some tools can be influenced by the characteristics (e.g., the stiffness) of the joint. This is especially true of impacting tools, but it can also be true for other tools. Since the stiffness of the calibrator is never the same as that of the joint, frequent calibration is not as reliable a means of control as you might think.

A good way to monitor the torque being delivered to the joint is to measure the output of the tool as it actually tightens the fasteners. DC-powered tools are especially useful for this sort of thing, and can readily be equipped with transducers to report both output torque and drive angle information. Sophisticated torque-monitoring systems are available for this purpose—electrical or electronic systems give a digital or control signal readout as the fastener is being tightened. Some systems are used only to monitor a few bolts once in a while—a sample. Other systems are designed to monitor all of the bolts tightened at a given production station. Monitored systems still use $\pm10\%$ air tools, but since they are monitored they really do produce $\pm10\%$ [16]. Don't use data acquisition tools near a source of electrical noise; however, noise can confuse the tool.

Still other systems are intended to monitor the tension in sample joints as well as the torque applied to the fastener. One such, for example, interposes a tension load cell between the head of the fastener (or nut) and the joint. A torque transducer is interposed between the normal production tool and the drive socket. The readout system—which is portable—now watches applied torque and achieved tension. The system is not intended to be used on every assembly. It is used by quality control inspectors or engineers to evaluate the results achieved on a sample fastener in a sample joint—perhaps to recalibrate a production tool or to adjust it in the first place.

Here's another way to monitor results: some tools and systems automatically mark the parts after a desired torque has been produced, for a rough visual inspection.

7.7.7 Add Torque Feedback for Still Better Control

A good torque monitor will usually show that you're not getting consistent output torque. The next step toward accuracy, therefore, is to provide some sort of feedback control based on torque. The transducer signal used for monitor purposes is now massaged, amplified, and used to actually control the tool—probably through an electrically actuated valve of some sort. Torque accuracies of $\pm1\%-\pm5\%$ are now possible [16].

A variety of such tools are available with different principles of operation, response times, control accuracies, etc. Most controls are based on output torque, but some more recent systems base the control on the output speed of the tool. Some control systems tool control as well as digital or signal outputs.

7.7.8 For More Information

Go online to get more and up-to-date information on bolting tools. Visit the Web sites of companies like Ingersoll-Rand, Atlas Copco, Gardner Denver, Thor, Bidwell Industrial Group's Power Dyne Division, Chicago Pneumatic, Skidmore-Wilhelm, and Allen Bradley and Biach Industries.

7.8 FASTENERS THAT LIMIT APPLIED TORQUE

Assembly torque is usually controlled by a torque tool of some sort, but this is not always the case. The structural steel and airframe industries favor, instead, special fasteners which limit the amount of torque which can be applied to them, and, thereby, limit and control the preload developed in them during assembly. The accuracy with which torque—or, more important, preload—can be controlled by such fasteners is less than the accuracy of the best torque tools, but these two industries deal primarily with joints loaded in shear, and the accuracy requirements are less severe than they are for critical joints loaded in tension. Some of these special fasteners are described below. Note that all suffer permanent deformation in some way as the limiting torque is reached. This prevents the mechanic from tightening them further. It also allows an inspector to decide, from visual observation alone, whether or not they have been fully torqued. And it prevents them from being (easily) loosened. Neither industry needs to loosen and reinstall the bolts in the joints they're used in, however, so this is not a problem. This fact does limit their use in other applications, however.

7.8.1 The Twist-Off Bolt

The twist-off bolt cannot be held or turned from the head. (You'll note in the figure that it has an oval head.) Instead, the bolt is held by the assembly tool from the nut end. An inner

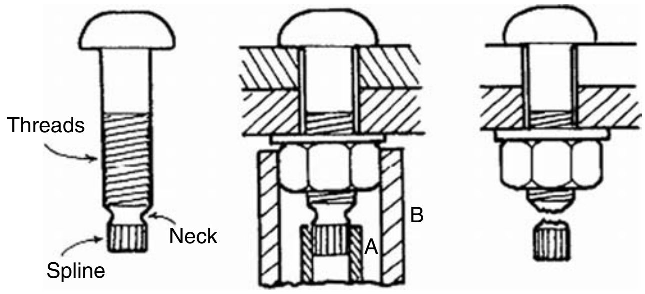


FIGURE 7.14 A twist-off bolt designed to indicate that a minimum amount of torque has been applied to the bolt. The tool holds and reacts against (A) spline, while turning the (B) nut.

spindle on the tool grips a spline section connected to the main portion of the bolt by a turned-down neck. An outer spindle on the tool turns the nut and tightens the fastener, with the tool reacting against the spline section. When the design torque level has been reached, the reaction forces on the spline snap it off, as shown in sketch 3 in Figure 7.14. The building inspector can determine whether or not a minimum amount of torque was applied to the fastener by looking to see whether or not the spline sections have indeed been snapped loose from the bolts.

If, between calibration and use, the bolts are allowed to become rusty or in any other way suffer a change of lubricity, then the amount of tension actually achieved in field assembly can be quite different from that achieved in the calibration stand, as suggested by some of the data in Table 7.1.

The fact that this fastener can be calibrated in the as-used condition, however, and, even more important, the fact that the inspector has a way to determine whether or not a minimum torque was applied to the fastener make this a popular item.

7.8.2 The Frangible Nut

The twist-off bolt is used only in the structural steel industry, as far as I know. The airframe people use the same principle, but this time applied to the nut. The outer section of the nut is unthreaded but has a normal hexagon shape which is driven by the assembly tool. It's fastened by a reduced cross-section breakaway collar to a cylindrical inner section which is threaded. When the limiting torque is reached the hex section breaks away from the cylindrical section, which remains to hold the bolt. This fastener is shown in Figure 7.15.

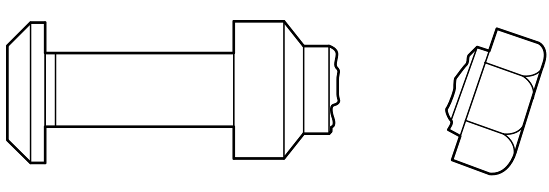


FIGURE 7.15 This type of twist-off fastener is used extensively on airframes. The cylindrical end of the nut is threaded; the outer, hex section is not. When the torque applied to the fastener reaches a predetermined value the hex end snaps off, as shown in the lower sketch, leaving the cylindrical end fully engaged with the bolt. Only visual inspection is required to prove that a given torque has been applied to the fastener.

7.9 IS TORQUE CONTROL ANY GOOD?

Does the fact that there are control problems mean that torque control is not good? No, emphatically not. Torque is still the most versatile and easiest control means. Design engineers, furthermore, are aware of the limitations of torque control and have long since learned to overdesign their products to offset the scatter one will normally get in preload—except in critical joints.

Using torque to control preload is somewhat like driving a car to work. Statistics prove that every time you get behind the wheel there is some possibility that you will not arrive at your destination. This doesn't mean that you should walk instead—or take a commercial airplane (which has a better per-passenger-mile safety record than the automobile). These other options are just not practical or economical in most cases—they don't suit other aspects of your total "get-to-work" problem.

If you know the inherent dangers and limitations of automobile travel, however, and do things to compensate for them, there's a better chance that you will arrive at your destination.

Torque control of preload also has obvious limitations and dangers, but in most cases it will be the only practical or economical choice. Knowing its limitations will improve your chance of success. And success, in this case, is not "preload accuracy"; it's "joints which don't fail." There can be a big difference between those two, and we're doing the person who pays the bills a big disservice if we ignore that fact. We don't want "accuracy" simply because it's technically challenging or interesting. We want it only if we need it.

Besides, there are a lot of things we can do to improve the results we get when we use torque control. Some of these things are listed at the end of the last chapter. I've put them there because most of them can be used to improve assembly results in general, regardless of which type of tool or control system we use.

7.10 TESTING TOOLS

Remember the "T" in FOGTAR? It stood for tool accuracy. You'll never get acceptable results at assembly, even if all other factors are working for you, if your tool misbehaves. Reliable, accurate tools are a must. Ford approves a tool for production use only if it meets their accuracy standards when new and after 250,000 cycles of use. (They certify a tool as preferred if it remains within specifications for 500,000 cycles.) [38] The tools are tested in accordance with ISO standard #5393, "Rotary Tools for Threaded Fasteners—Performance and Test Method."

Tests for accuracy must be based on test joints whose hardness is similar to that of the joints on which the tool is to be used, to evaluate overshoot and other factors affecting accuracy. Testing impact tools is especially difficult because they produce only a rapid series of "instantaneous torque" pulses. In general, tools should be tested by knowledgeable experts whenever possible. A large facility for testing the accuracy and reliability of automotive assembly tools has been set up by the mechanical engineering department of the Lawrence Technological University in Southfield, Michigan, near Detroit.

RS Technologies of Farmington Hills, Michigan has developed an impressive line of equipment and procedures for testing and evaluating the performance of bolting tools. They can measure and plot drive torque, angle of turn, overshoot, embedment relaxation, breakaway torque, and other bolting variables. In one of the tests they have developed the "real time," dynamic friction forces between male and female threads and between nut and joint surface are measured and reported separately. You can learn more if you visit their Web site.

7.11 THE INFLUENCE OF TORQUE CONTROL ON JOINT DESIGN

Remember that a bolted joint is a mechanism for creating a force, the clamping force designed to hold two or more pieces of a structure or product together. That being the case, anything which affects or helps determine that clamping force must be of interest to a conscientious bolted-joint designer. And the type of control we use at assembly is certainly such a factor.

If we use torque control, with its typical scatter of $\pm30\%$, some bolts in a large group will acquire 30% more initial preload than the target or average value; other bolts will start their active lives with 30% less; and most will start with something in between. We could estimate this average, of course, by using the short-form torque–preload equation (Equation 7.4).

The thoughtful designer, therefore, cannot assume that the torque he specifies will achieve average initial preload in every bolt. He must assume that the initial preload seen by some bolts will be the average preload plus 30%. At the same time he must assume that the initial preloads seen by other bolts will be average less 30%.

It's important to understand that, as a result of this, the designer must select bolts and joint members large and strong enough to support the stresses encountered at average initial preload plus 30%. At the same time he must include enough bolts so that there's still sufficient clamping force to hold the structure together—after embedment and elastic interaction relaxation occur—even if some of the bolts have as little as average minus 30%.

Theoretically, from a simple, statistical point of view, the average preload in any group of bolts would be the target preload, with as many of those bolts above average as below average. The clamping force on the joint would just be the average preload times the number of bolts—less embedment and elastic interaction losses, of course. If things were as simple as that, we really wouldn't have to worry about this $\pm30\%$ business.

The trouble comes from the fact that most joints don't contain enough bolts to guarantee a statistical average preload. And, in a small lot of bolts obtained from the same source at the same time or all subjected to the same prior service conditions, all may be more, or all less, lubricious than "average." As a result, and at least in safety-related joints, the safest course of action is to assume that average initial preloads will be something less than the short-form equation would lead us to believe. And then to subtract the further reduction in average preload caused by relaxation and interactions. What's left will be our estimate of the average residual preload; and this is all we can count on to generate the clamping force between joint members as we put that joint to work. All this is illustrated in Figure 7.16, an extension of Figure 6.27.

Should we assume that the average initial preload will be a full 30% less than the theoretical average? Only an experiment on the actual product can answer that question with certainty, but 30% would certainly be a defensible number. Consider it a safety factor, if you will, and increase or decrease it depending on the consequences of failure, and on your reading of the care with which you expect this product to be assembled and used. Before considering a more optimistic average, however, you should consider the following.

In Section 7.3, we looked at some of the many factors which can affect the amount of preload we'll get for a specified applied torque. Some of these scatter factors can give us either more or less than average preload; examples include friction, tool accuracy, and the mechanic. None of the factors can give us only more than average preload. But—and here's the problem—many of the things listed can only give us less than average. These include things like joints that are difficult to pull together, nonparallel joint surfaces, nonperpendicular holes, interference fits, and embedment relaxation. When we consider that the other major source of scatter in assembly results—elastic interactions—is also a "less only" factor, we're forced to conclude that it's far safer to assume that we'll end up with a less than average residual clamping force than with a greater than average one.

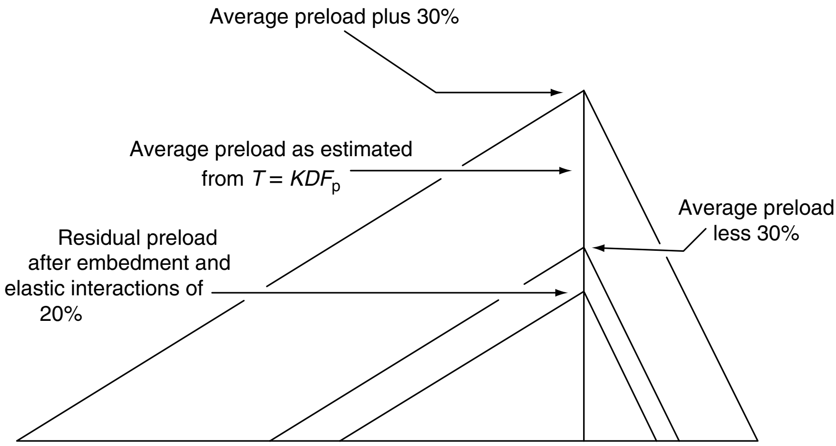


FIGURE 7.16 This is an extension of the joint diagram shown in Figure 6.27. This one now includes the effect of scatter or uncertainty in the relationship between applied torque and the initial preload created in the bolt. Conventional wisdom says that the scatter in preload for a given torque, applied to as-received steel-on-steel fasteners, will be $\pm 30\%$, as shown here. Embedment relaxation and elastic interactions then reduce the initial preloads to some lesser residual value; an average loss of 20% is suggested above. Bolts and joint members must be sized to support the maximum force of average plus 30%; yet we must assume a much lower residual preload when estimating worst-case clamping forces on the joint.

This forces us to overdesign the joint. We must include many, large-diameter bolts and equally large joint members in order to be able to count on a relatively modest residual clamping force. And the accuracy—or lack of it—of the assembly control system we have selected affects the difference between the size of the parts required to support maximum assembly preloads and the amount of clamping force we can count on.

7.12 USING TORQUE TO DISASSEMBLE A JOINT

We have now taken a long look at the use of torque to control the buildup of preload in bolts when a joint is assembled. This is, however, only one of the many ways to do that job. In the next couple of chapters we'll examine half a dozen or more other ways to create and control preload. Each way has its advantages and disadvantages, its supporters and its detractors. The blood of many salesmen has been shed in these struggles to identify the best procedure for a given application. Torque has won its share of these contests but has also lost many individual battles. There is, however, one place where torque is undisputed king: whether measured or not torque is always used to loosen fasteners during disassembly of a joint.

The literature is generally mute on the disassembly process unless galled fasteners or some other problem is encountered. In Section 7.5.1 we learned of a PVRC lubricant study that encountered galling. In Chapter 13 of this text we'll go farther and learn some of the techniques used to disassemble a joint if the fasteners have galled. But joint disassembly deserves more than that. There are correct and incorrect ways to disassemble a joint, using a torque tool of some sort.

In Chapter 6 we saw that large diameter joints should be tightened by a series of torque passes applied in a cross or star pattern to the bolt circle. The first pass might involve a torque that is only one-third of the torque that is to be used on the final pass. The fasteners in rectangular joints are best first snugged then tightened starting in the center of the bolt pattern before proceeding to the bolts on the outer edges of the pattern.

Small, relatively light-weight joints involving few bolts can usually be disassembled by loosening individual bolts fully, in any convenient pattern. The bolts in large, heavy joints such as those described above, however, should be loosened the same way they were tightened: gradually, in two or more passes. Remember that as such bolts were tightened they progressively compressed the joint members, creating sometimes huge amounts of potential energy in the joint as well as in themselves. Each time we fully loosen a bolt the remaining, still tightened, bolts have to absorb some of the potential energy previously supported by the bolts just loosened. The last bolts to be loosened can actually break—actually fly apart—under these conditions. Bolts tightened by a series of increasing torques should be partially loosened, then further loosened, and then finally loosened to avoid this problem. Even if the final bolts don't break the tensile stress, resulting thread contact pressures in the last ones to be loosened could be great enough to gall the threads or increase breakout torques to difficult or impossible levels.

EXERCISES AND PROBLEMS

1. Name some of the problems created by incorrect torque.
2. How much preload do we want, in general?
3. How much torque versus initial preload can you expect to encounter when tightening a group of unlubricated, as received, steel bolts?
4. What is meant by the term prevailing torque?
5. What do we mean by the terms initial preload and residual preload? Why do they differ?
6. What limits the usefulness of the so-called, long-form torque–preload equation?
7. Explain the difference between the coefficient of friction (μ) and the nut factor (K).
8. How would you identify the best nut factor for your application?
9. How much torque would be required to tighten an ASTM A193 Gr. B16 bolt with a 3¼–8 thread to 60% of yield?
10. Why does a wrench create a side load on the bolt being tightened?
11. What do we mean by the term torque reaction?
12. A right angle nut runner is used to produce 500 ft-lb of torque on a bolt having a ⅞–9 UNC thread. The tool is 18 in. long. How much reaction force, created by the reaction torque, will the operator feel? Is this a practical situation?

REFERENCES AND BIBLIOGRAPHY

1. Motosh, N., Development of design charts for bolts preloaded up to the plastic range, *Journal of Engineering for Industry*, August 1976.
2. Shigley, J.E., *Mechanical Engineering Design*, 3rd ed., McGraw-Hill, New York, 1977.
3. Czajkowski, C.J., Investigations of corrosion and stress corrosion cracking in bolting materials on light water reactors, *International Journal of Pressure Vessels and Piping*, vol. 26, 1986, pp. 87–96.
4. Anderson, W. and P. Sterner, Degradation of threaded fasteners in reactor coolant pressure boundary of pressurized water reactor plants, *U.S. NRC*, *NUREG*-1095, May 1985.
5. Yura, J., K. Frank, and D. Polyzois, High strength bolts for bridges, PMFSEL Report no. 8–3, University of Texas, May 1987.
6. Brenner, H.S., Results of special evaluation of Pepcoat 6122 and nuclear grade N5000 antiseize lubricant systems on torque–tension performance of ASTM A325 series hex-head bolts submitted by Power and Engineered Products Company, Inc. (PEPCO), South Plainfield, NJ, Report no. C 16034–1, Almay Research and Testing Corp., Los Angeles, October 25, 1982.
7. Tests on 1 in. A490 conventional and twist-off high strength bolts and direct tension indicators conducted in the as-received condition and after weathering, Report no. 870702, Pittsburgh Testing Laboratory, Pittsburgh, PA, April 10, 1986.

8. Bezborod'ko, M.D., et al., How to secure the reliable tightening of screwed joints, *Vest. Mash.*, vol. 58, no. 1, 1978.
9. Bickford, J., Study of the tension, torsion and bending stresses in 2¼–8 × 12, B16 studs during torquing operations, Raymond Engineering, Inc., In-house report, April 1976.
10. Brenner, H.S., Development of technology for installation of mechanical fasteners, Almay Research and Testing Corp., Los Angeles, Report prepared for NASA under contract no. NA3–3–20779, 1971.
11. Chesson, E., Jr., N.L. Faustino, and W.H. Munse, High strength bolts subjected to tension and shear, *Journal of Structural Division, Proceedings of the ASME,* October 1965.
12. Breeze, W.D., Tightening torque and clamping force relationship for a range of threaded fastener sizes, Naval Ship Engineering Center, Hyatsville, MD, April 1974.
13. Zmieuskii, V.I., et al., Greases for stainless-steel threaded joints, *Vest. Mash.*, vol. 54, no. 1, 1974.
14. Investigation of threaded fastener structural integrity, Final report, Contract no. NAS9–15140, DRL-T-1190, CLIN 3, DRD MA 129 TA, Southwest Research Institute, Project no. 15–4665, October 1977.
15. Eshghy, S., The LRM fastening system, *Fastener Technology*, July 8, 1978.
16. Rice, E.E., Reliability in threaded fastener tightening, Society of Manufacturing Engineers, Paper no. AD75–769.
17. Ehrhart, K.F., Preload—the answer to fastener security, *Assembly and Fastener Engineering*, June 1971.
18. Lesner, R.S., Power tool concepts for precision torque control, Society of Manufacturing Engineers, Paper no. AD76–638, 1976.
19. Wakefield, S.B., Portable electronic joint analysis, Society of Manufacturing Engineers, Paper no. AD75–769, 1975.
20. Wilhelm, J., Torque–tension testing equipment, *Compressed Air Magazine*, vol. 75, no. 8, pp. 20–23, August 1970.
21. Hung, N., Clamping force for electrically conductive fasteners, *Machine Design*, October 1984.
22. From information received from the E/M Corporation, West Lafayette, IN, March 1988.
23. Taylor, E., SerMeTel W (TM) aluminum coating for aerospace fasteners, SPS Laboratory Report no. 5392, SPS Technologies, Inc., Jenkintown, PA, October 30, 1980.
24. Gresham, R.M., Bonded solid-film lubricants for fastener coatings, *Fastener Technology International*, April/May 1987.
25. Emrich, M., Corrosion protection for fasteners, Part 1, *Assembly Engineering*, October 1982.
26. Pepcoat, a study in protection and performance, G*Chemical Corp., Wayne, NJ, undated manufacturer's literature.
27. Investigation of the torque–tension relationship for Monel, K-Monel and steel threaded fasteners, Report MPR-920, MPR Associates Inc., prepared for Attack Submarine Acquisition Program, NAVSEA PMS393TM6, Washington, DC, June 1986.
28. Report of torque tests to determine the torque-axial load relationship for Monel and steel bolts lubricated with Neolube and MIL-L-24479B, Report MPR-1092, MPR Associates, Inc., prepared for Attack Submarine Acquisition Program, NAVSEA PM393TM6, Washington, DC, October 1988.
29. Clement, B. and A. Bazergui, A study of the preload relationship in bolting technology: experimental design and analysis, prepared for the Bolting Technology Council, New York, October 1989.
30. Stewart, R., Torque/tension variables, list prepared at Wright-Patterson Air force Base, April 16, 1973.
31. Toosky, R., Threaded Fasteners, Torque and Tension Control, notes for a seminar given at McDonnell-Douglas, Long Beach, CA, December 1991.
32. From a presentation made by Ralph Shoberg of R.S. Technologies, Farmington Hills, MI, to the Bolting Technology Council, May 1991.
33. Crispell, C., Torque vs induced load of A-286 and MP35N nuts and bolts with cadmium, dry film and cetyl alcohol lubricants, Request No. 8–1–7-EH-07694-AP29-B, NASA Contract No. NAS8–32525, November 14, 1978.
34. Srinivas, A.R., *Determination of Torque Values for Metric Fasteners—an Approach*, Spacecraft Payload Group, Space Applications Center, Indian Space Research Organization, Ahmedabad, India, May 1992.
35. From information provided by the manufacturer, VSI. VOI-SHAN, Redondo Beach, CA, 1992.

36. Chapman, I., "Choosing the Optimum Fastening System," privately published by Ingersoll-Rand, n.d.
37. Chapman, I. and S. Sharma, "Choosing the Optimum System for the Assembly of Threaded Fasteners," privately published by Ingersoll-Rand of Annandale, NJ and Harley Davidson of York, PA, n.d.
38. "Pocket Guide to Tightening Techniques," available on line from www.atlascopco.com.
39. *Nut Factor Claims vs Experimental Reality*, Report on PVRC project 01-BFC-06 presented by James Payne at a PVRC Bolted Flanged Connections Workshop in New Orleans on May 23–25, 2005.

8 Torque and Turn Control

8.1 BASIC CONCEPTS OF TURN CONTROL

We have tried to use torque as a means of controlling the preload in the fastener and have found problems. Even perfect input torque can give us a $\pm 25\%$–30% variation in preload.

But when we apply torque, the nut turns. Can we use the angle through which the nut turns, instead of torque, to control preload? At first glance this looks very promising. After all, when we turn the nut on a machine-tool lead screw by 360°, the screw advances or retracts with a linear displacement equal to exactly one pitch of the threads. Won't a bolt stretch by this amount when we rotate the nut one turn? If so, we could use the lead screw equation to relate bolt stretch to turn of the nut, or

$$\Delta L = P\frac{\theta_R}{360} \tag{8.1}$$

where ΔL is bolt stretch (in., mm), P is the pitch of the threads (in., mm), and θ_R is the angle of nut rotation (degree) with respect to the bolt. We could then get bolt preload very easily, assuming that we knew the spring constant or stiffness of the bolt, from

$$F_P = K_B \times \Delta L = K_B P\frac{\theta_R}{360} \tag{8.2}$$

where K_B is bolt stiffness (lb/in., N/mm) and F_P is preload (lb, N).

But life isn't this simple. As illustrated in Figure 8.1, the lead screw moves a distance of one pitch when we turn the nut only if the nut is rigidly restrained and the lead screw is perfectly free. If the nut were free and the screw restrained, of course, it is the nut which would move, as it does when we're running a free nut down against a joint. We still have relative motion, between the bolt and nut, equal to one pitch; but turning a loose nut obviously produces no preload whatsoever.

After the nut has bottomed, further turning will indeed stretch the bolt. But during this portion of the tightening process neither nut nor bolt is rigidly restrained. Rotation of the nut produces, in part, displacement of the nut downward into the joint, as joint members and nut compress. It also produces displacement of the bolt upward—elongation of the bolt, in other words. The relative displacement between nut and bolt is still one pitch for one input turn (if we ignore bolt twist), but only a portion of that relative displacement is bolt elongation.

If the spring constant of the body of the bolt equals the combined spring constant of everything else in the joint (including the nut), then half of the displacement (or, more accurately, the deformation) will occur in the bolt and the rest will be distributed in the nut and joint members. Figure 8.2 shows an analog of this situation. The forces on bolt and joint are equal and opposite. If the spring constants are equal, then it follows that deformations

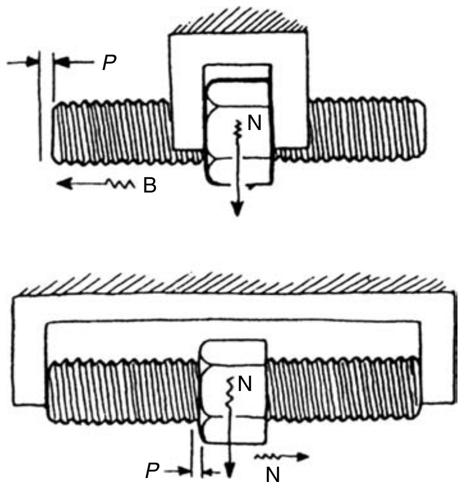


FIGURE 8.1 In a lead screw, one revolution of the nut will produce a linear displacement of one pitch in the screw or in the nut, depending on which is restrained and which is free to move.

produced by equal forces must also be equal. Rotating the nut one turn will only stretch the bolt one-half a pitch.

If the bolt is less stiff than the joint, then it will absorb a larger share of the overall deformation and the joint will see less. This is the situation we will find in a heavy joint with a hard makeup. Forces on bolt and joint are still equal and opposite. Deformations will be in inverse relationship to bolt and joint stiffness. Rotating the nut one turn will stretch the bolt more than one-half a pitch, but less than one pitch.

If the joint is relatively soft compared to the bolt—as it sometimes is if a gasket is involved—then most input turn will be absorbed in deformation of the joint rather than of the bolt, as suggested in Figure 8.3. Rotating the nut one turn stretches the bolt less than half a pitch—usually much less for large gasketed joints.

Note that we can never generate more force in one spring than we can in any other spring to which it is connected in series. This is why the softest member of a joint—whether it's the bolt or a gasket or something else—will dominate the behavior of the joint, both during tightening and in use. This is true not only when behavior is elastic, such as above, but also if

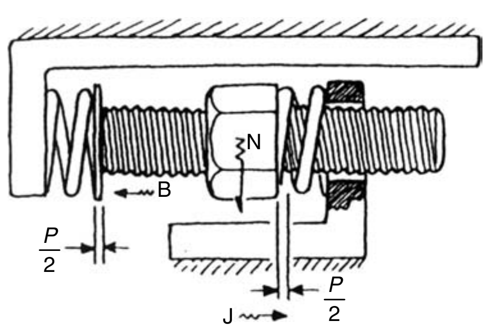


FIGURE 8.2 Analog of a bolted joint in which the spring constant of the bolt equals the spring constant of joint members. The sketch assumes rigid bodies restrained by equal springs. Displacement of the bolt to the left equals displacement of the nut to the right. The bulkhead restraining the nut spring (joint) is shown partially cut away for clarity.

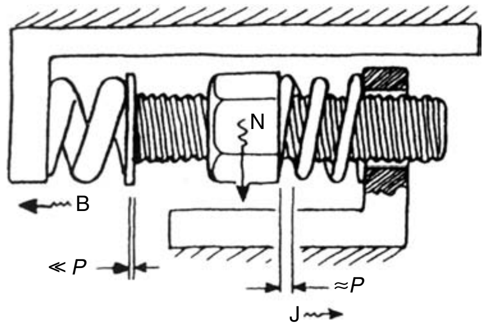


FIGURE 8.3 In a gasketed joint, the spring constant of the bolt is greater than the combined spring constant of joint members, so turning the nut produces more deformation in the joint than in the bolt.

the softer member yields, limiting the force it will support. More of this when we discuss turn-of-nut procedures later in this chapter.

In any event, we now realize that before we can use input turn as a control means, we must be able to predict the relative stiffness of bolt and joint members. As we saw in Chapter 5, computing bolt stiffness is a relatively simple procedure, at least for long thin bolts. Computing joint stiffness is more difficult; it must be determined experimentally in most cases. In fact, there will always be some uncertainty, especially if short bolts, unusual bolt materials [1], thin joint members, or gaskets are involved (see Chapter 9). And any uncertainties in our estimates of the stiffness ratio lead to similar uncertainties in our predictions of the relationship between turn and preload in a given application.

But that's not the crux of the turn control problem. Let's take a closer look at what happens when we apply turn to a nut.

8.2 TURN VERSUS PRELOAD

8.2.1 Common Turn–Preload Relationship

Let's assume that we're tightening a hard joint. Here's what happens step by step as we rotate the nut. We'll assume that turn with respect to the body of the machine (ground) equals the turn with respect to the bolt.

The first few turns of the nut produce no preload at all, because the nut has not yet been run down against joint members and they are therefore not yet involved. This situation is shown in Figure 8.4A.

Finally the nut starts to pull joint members together. There may be frictional restraint between joint members and surrounding structures. Joint members may not be perfectly flat. There may be a bent washer. As a result, although we start to produce some tension in the bolt, most of the input turn is absorbed by the joint and the bolt sees only a small increase in preload, as suggested by Figure 8.4B. This process is called snugging the joint, and the amount of turn required varies unpredictably, even between apparently identical bolts or joints.

After the joint has been snugged, all bolts and joint members start to deform simultaneously, with individual deformations in inverse proportion to individual spring constants. Preload now starts to build more rapidly in the bolt, following a straight line whose slope is equal to

$$\text{Slope} = \frac{\Delta F_{\mathrm{P}}}{\Delta \theta_{\mathrm{R}}} = \left(\frac{K_{\mathrm{B}} K_{\mathrm{J}}}{K_{\mathrm{B}} + K_{\mathrm{J}}}\right) \frac{P}{360} \tag{8.3}$$

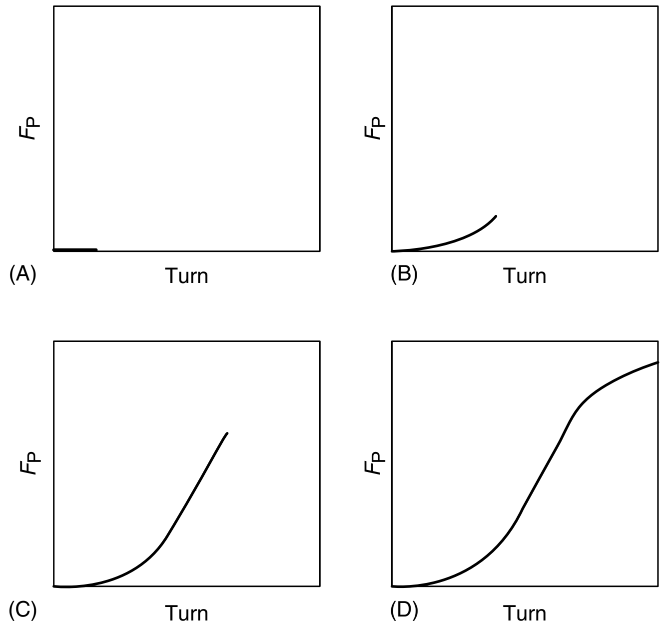


FIGURE 8.4 Step-by-step buildup of preload in a joint when the turn of the nut relative to the bolt equals turn with respect to ground. (A) Nut run-down produces no preload. (B) Snugging pulls joint members together, flattens washers, etc. and produces a little preload. (C) Bolt and joint members all deforming elastically. (D) Something in the joint—usually the bolt or gasket—yields to limit further buildup of preload.

where K_B and K_J are the spring constants of bolt and joint members (lb/in., N/mm), P is the pitch (in., mm), F_P is preload (lb, N), and θ_R is the input angle of turn in degrees. Note that $(K_B K_J)/(K_B + K_J)$ is the combined stiffness of the total bolt–joint system, so this equation reduces to a revised version of the F_P-θ Equation 8.2.

During this straight-line portion of the process, there is usually a linear relationship between additional input turn and additional preload, as shown in Figure 8.4C. If we could predict the spring constants involved—and if we could determine where this straight-line portion of the curve starts—measuring turn would give us good control of preload. Unfortunately, however, we will find it very difficult to determine where the straight-line portion of this curve starts. It will vary from bolt to bolt, from assembly to assembly, and from application to application, adding to the uncertainties in spring constant.

If we continue to turn the nut, something in the joint will eventually start to yield. This ends the linear buildup of preload in the joint as suggested in Figure 8.4D. More important for control purposes, however, it ultimately limits the preload created by turning the nut further. This is really a torque–turn technique, rather than a turn technique, so we'll leave it for later.

8.2.2 Other Turn–Preload Curves

The S-shaped curve shown in Figure 8.4 describes the torque–preload behavior of a conventional, moderately stiff joint. There are, however, other possibilities.

8.2.2.1 Sheet Metal Joint

If joint members are very thin, they are also very stiff. This can mean a nearly vertical rise in the elastic portion of the torque–turn curve, again creating control problems in high-speed,

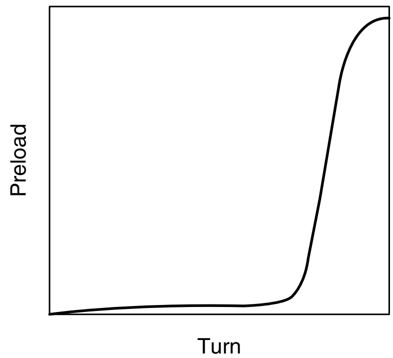


FIGURE 8.5 Turn–preload curve for a sheet metal screw. (Modified from Ehrhart, K.F., *Assembly and Fastener*, June 1971.)

automatic operations. Figure 8.5, for example, shows the typical curve for a bolt tightened against sheet metal [1].

8.2.2.2 Gasketed Joint

Gasketed joints are often tightened in stages, since irregular loading can distort and damage a gasket, and it is desirable to pull the joint together simultaneously to the extent possible. A typical procedure is to tighten all fasteners to 30% of the final desired preload, then go around again and tighten them all to 60%, and then go around again and tighten them to 100%. Gaskets will usually creep and relax between passes, however, and gasketed joints will experience the elastic interaction relaxation discussed in Chapter 6, so if one were to plot the relationship between the turn applied to a particular fastener and the preload produced in that fastener, he would usually see a pattern such as that shown in Figure 8.6.

If all of the bolts in a gasketed joint are tightened simultaneously all the way, we would expect the turn–preload curve to be relatively normal—except that the elastic region would have a very low slope, and there would be relaxation following tightening, as shown in Figure 8.7.

8.3 FRICTION EFFECTS

It is commonly believed that turn control is better than torque control because the relationship between turn and preload is independent of friction. All we have to worry about are those spring constants and "where do we start measuring turn?"

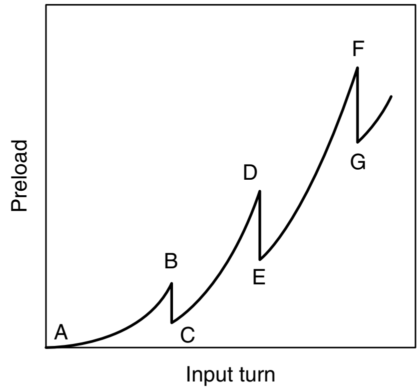


FIGURE 8.6 The preload in gasketed joints partially relaxes between passes, because of gasket creep and relaxation. The first pass took the bolt from point A to point B, the second pass from C to D, etc.

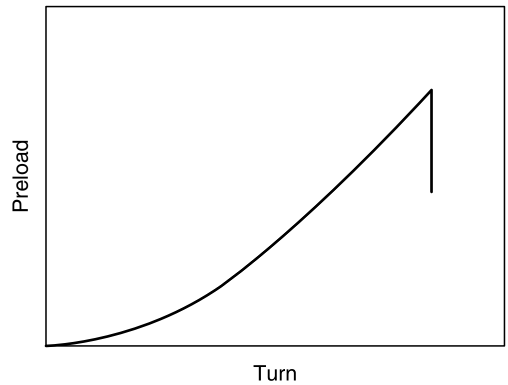


FIGURE 8.7 If all bolts in a gasketed joint are tightened simultaneously, relaxation occurs only after final tightening—but it still occurs.

But this is not true. It's a popular misconception based on the fact that most of the turn control equations you'll find in the literature are based on the relative turn between nut and bolt (θ_R). If we could indeed measure relative turn, we could ignore friction.

I don't know of any way to do that, however, except in a laboratory. In practice, we always measure turn-of-nut relative to the frame of the machine, or the floor, or some fixed reference (θ_G) other than the body of the bolt. From our frame of reference, we could give the nut one-half revolution after snugging and produce no preload at all if the threads had seized. On the other hand, one-half revolution could stretch the bolt past yield if someone had used a better-than-normal lubricant on the threads. In-between coefficients of friction in the threads would give us in-between results, as far as the relationship between input turn and preload is concerned.

As a result, we will find that when we measure turn with respect to the same reference point from which we measure input torque, the floor on which we are standing, the turn–preload curve becomes a family of curves like the torque–preload family—a different curve for each coefficient of friction. And the slope of the center portion of each curve is no longer just a function of spring constants; it is also affected by the amount of input work that is going into torsional energy or heat loss in the system (see Figure 8.8).

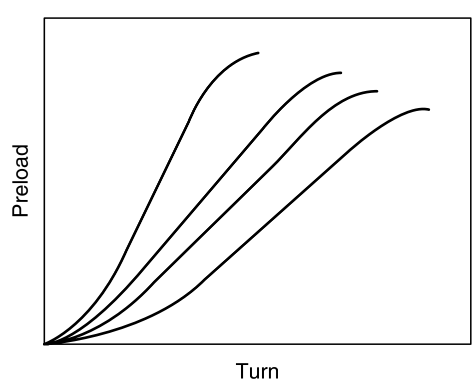


FIGURE 8.8 If one measures input turn relative to the machine rather than relative to the bolt, the preload–turn relationship becomes a family of curves, again depending on the coefficient of friction in the threads.

As a result of all this, pure turn is no more accurate than pure torque control. But all is not lost. We apply torque and the nut turns. If we use both torque and turn to control the process, we do gain some accuracy over using either torque or turn alone, as we'll see.

Note that the slope of the straight-line portion of these curves is no longer given by Equation 8.3, because the angle of turn used in Equation 8.3 is the relative angle between nut and bolt, and we're now using turn with respect to ground. The correct equation for the present curve is far more complex as we'll see.

8.4 TORQUE AND TURN IN THEORY

8.4.1 Torque, Turn, and Energy

The area under the torque–turn curve defines the amount of energy being delivered to the fastener during the tightening process, as shown in Figure 6.3. If we have a computer-generated curve we can use standard software to compute that area—that energy—by integration. Those without this capability can estimate it by plotting the torque–turn curve, using a planimeter or "counting the squares" between the curve and the turn axis to estimate the area (as we all had to do in the not-so-good old days).

No one estimates the input energy at present, but people who design and build automated, production torque–turn equipment come closer to doing this than they realize. We'll take a look at this equipment in Section 8.7.

The fact that torque–turn information defines the input energy is one reason why torque–turn control is significantly better than torque control.

8.4.2 Torque–Turn–Preload Cube

Let's pause a moment to recognize the fact that we can never deal with just torque versus preload or just turn versus preload as we tighten a fastener. We apply torque, the nut turns, and the bolt stretches, creating preload. All of this is going on simultaneously. Preload is being developed in the fastener as a function of both torque and turn simultaneously. We can plot the resulting "space curve" on the three axes of a cube as shown in Figure 8.9. Each face of this cube gives us a different two-dimensional view of the total preload versus torque–turn curve, as suggested in Figure 8.9. Note that the turn–preload and torque–preload views of the space curve agree with Figure 8.4 and with Figure 7.1 as well.

Anything that affects one view of this space curve will affect the other views as well. Figure 8.10 shows what a change in friction does to the torque-turn-preload cube. Note that the torque-versus-turn view of this process (looking down from the top) is basically S-shaped—like the turn–preload curve but generally with a different slope, etc.

8.4.3 The Broader View

Have we just involved more variables and therefore have made things worse by trying to measure torque and angle (turn) at the same time? No. There are, in fact, at least three different ways in which measuring torque and turn simultaneously can improve our control over preload: Torque and angle give us sufficient information to tighten safely until something in the joint yields. The yielding limits, and therefore controls, preload. One such technique is called "turn-of-nut" control.

Torque and angle information allows us to spot a large number of serious practical problems. We haven't considered these yet, because they don't enter into the equations, and neither torque nor turn can reveal them alone. These problems include such things as blind

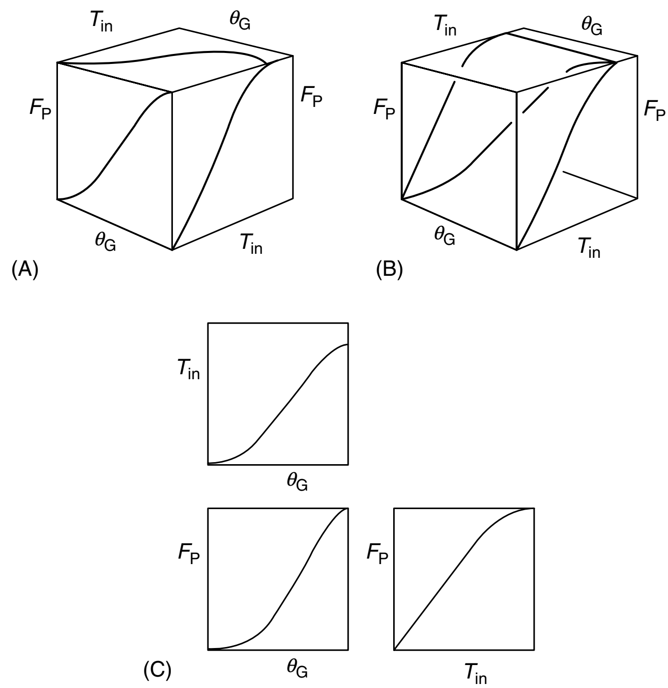


FIGURE 8.9 Torque and turn cannot be isolated from each other. Together they produce preload. View C is an orthographic projection of the cube. View A shows these projections on the surfaces of the cube, and view B shows the true T–θ–F_P curve snaking around inside the cube.

holes, wrong parts, crossed threads, etc. and are of major concern in automatic production operations. They usually cause less trouble in manual operations, but only if operators are well trained, properly motivated, and careful.

Torque and angle information can be fed to microprocessor or computer-control systems where sophisticated analysis of the information can be used in many different ways to give us

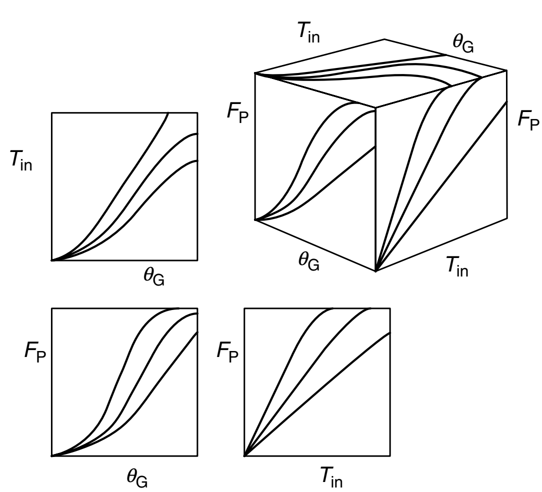


FIGURE 8.10 Anything—such as a change in friction—which affects one view of the torque–turn–preload process must affect the other views as well.

more accurate control of preload than would torque or turn information alone. Let's look at some possibilities in detail.

8.5 TURN-OF-NUT CONTROL

8.5.1 The Theory

The so-called turn-of-nut method is widely used, especially in structural steel applications. Historically it was the first torque–turn control technique; it is a manual technique, although similar, computer-control strategies are now used in mass production operations, as we'll see in a minute.

In the original turn-of-nut procedure, the nut is first snugged with a torque, which is expected to stretch the fastener to a minimum of 75% of its ultimate strength. The nut is then turned "three flats" (half a turn) or the like, which stretches the bolt well past its yield point, as shown in Figure 8.11 [5].

The preload produced by the snugging torque, of course, varies because it's affected by all the normal variations in friction, geometry, etc. Subsequently turning the nut past yield, however, always produces about as much tension as the bolt can support. Final variation in preload in a large number of bolts is probably closer to $\pm5\%$ than to the $\pm25\%$–30%, which would be the case if we used torque or turn control alone.

Torque instead of turn is used as the control means during the snugging process because torque is better able to compensate for start-up variables. If the head of the bolt slips, for example, when we first start to tighten it, we merely keep turning the nut until we have produced enough torque to guarantee that everything is truly snugged and we can start measuring turn.

Turn is used for final control, however, since it is a more accurate way of determining that we have really stretched the bolt past yield. The bolt really does behave like a lead screw—an elastic one—during some portion of its behavior. It has to stretch past yield if we turn the nut 180° past almost any point on the linear portion of the torque–turn curve.

Note that the final torque required to yield the bolt can vary drastically, making torque a poor means of determining yield. We could do it by looking at the rate of change in torque as a function of turn, but this is an awkward thing to do manually. It is, however, the basis for some computer-controlled techniques, as we'll see later.

Note that this classical turn-of-nut procedure cannot be used on brittle bolts. It can be used safely only on ductile bolts having long plastic regions, such as the ASTM A325 or A490 fasteners used in structural steel work.

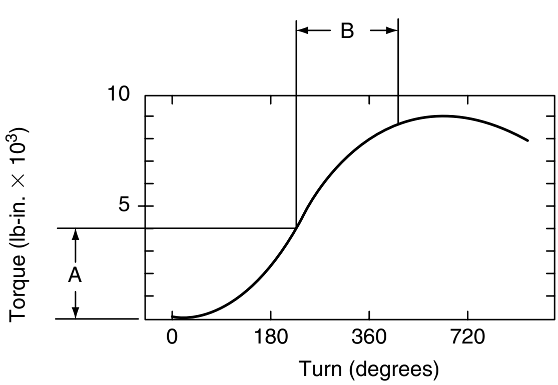


FIGURE 8.11 In turn-of-nut techniques the nut is first tightened with an approximate torque (A) and then further tightened with a measured turn (B). 7/8 × 5 1/2 ASTM A325 bolt.

Furthermore, it should never be used unless you can predict the working loads to which the bolt will be subjected in use. Anything which loads the bolt above the original tension will create additional plastic deformation in the bolt. If the overloads are high enough, the bolt will break. I don't mean to suggest that A325 bolts tightened this way will be unable to support tensile loads. As shown in Figure 3.17, the bolts will yield under combined torsional and tensile stress when first tightened. The torsional stresses will disappear soon after tightening, thanks primarily to embedment relaxation. This returns some tensile load capacity to the bolt (perhaps 5%–10% of its yield strength). Subsequent tensile loads would have to exceed this value to cause further yielding or rupture.

8.5.2 The Practice

8.5.2.1 Structural Steel

The turn-of-nut procedure for structural bolts was first proposed in the mid-1950s by the Association of American Railroads, influenced, perhaps, by similar techniques used by the automobile industry [6]. The technique was later modified by Bethlehem Steel Corporation and subsequently adopted in that form by the Research Council on Riveted and Bolted Joints of the Engineering Foundation. In the present form, bolts are first tightened with an impact wrench until the tool starts to impact. The bolt has now been snugged. The position of the socket, which is marked at 90° intervals, is now noted or marked, and the impact wrench is used to turn the drive socket another 180°–270°, depending on bolt length and whether or not the surfaces of the joint are perpendicular to the axis of the bolt threads or are sloped (as is common in structural steel). Beveled washers are often used to compensate for sloped surfaces. Table 8.1 shows the amount of turn recommended by the research council [5].

As an alternative, bolts can be snugged by "the full effort of a man using a spud wrench"; they are then turned the amount shown in Table 8.1 by an impact wrench. This procedure is not considered safe on bolts less than ¾ in. in diameter, however, as a man can tighten small bolts past the yield point with the spud wrench. Smaller wrenches or a different procedure must then be used [7].

TABLE 8.1
Nut Rotation from Snug Tight Condition for Turn-of-Nut Procedure[a,b]

	Disposition of Outer Face of Bolted Parts		
Bolt Length (Underside of Head to End of Bolt)	**Both Faces Normal to Bolt Axis**	**One Face Normal to Bolt Axis and Other Sloped not More Than 1:20 (Beveled Washer not Used)**	**Both Faces Sloped not More Than 1:20 from Normal to the Bolt Axis (Beveled Washer not Used)**
Up to and including 4 diameters	⅓ turn	½ turn	⅔ turn
Over 4 diameters but not exceeding 8 diameters	½ turn	⅔ turn	⅚ turn
Over 8 diameters but not exceeding 12 diameters	⅔ turn	⅚ turn	1 turn

Source: Specification for Structural Joints Using ASTM A325 or A490 Bolts, Research Council on Riveted and Bolted Structural Joints of the Engineering Foundation, November 13, 1985.

[a] Nut rotation is relative to bolt regardless of the element (nut or bolt) being turned. For bolts installed by ½ turn and less, the tolerance should be ±30°; for bolts installed by ⅔ turn and more, the tolerance should be ±45°.
[b] Applicable only to connections in which all material within the grip of the bolt is steel.

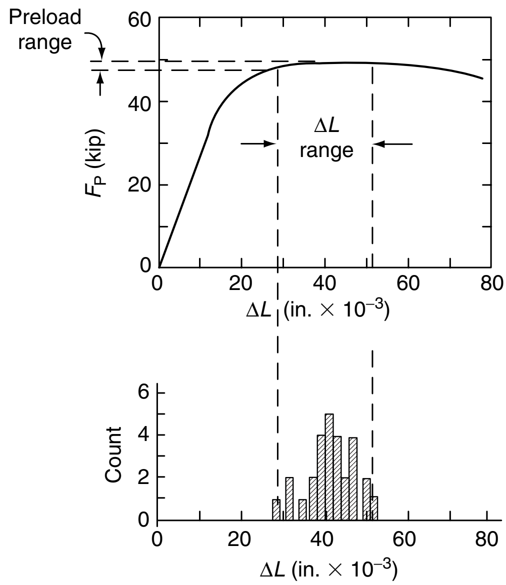


FIGURE 8.12 Elastic curve for a ⅞ in. A325 bolt with a 4 in. grip length, and a histogram of the elongations produced in 28 such bolts by a ½-turn-past-snug tightening procedure. Note the small variation in preload (tension) in spite of the almost 2:1 scatter in elongation. (Modified from Specification for Structural Joints Using ASTM A325 or A490 Bolts, Research Council on Riveted and Bolted Structural Joints of the Engineering Foundation, November 13, 1985.)

The success with which the turn-of-nut procedure controls bolt tension can be seen by referring to Figure 8.12. Note that this is a tension–elongation curve, not a torque–turn or preload–turn curve, so it is not S-shaped. It is, instead, the elastic curve for this ⅞ in. A325 bolt, and is based on tests reported by Fisher and Struik [7]. The histogram under the graph shows the elongations produced, in a sampling of bolts, by ½ turn-past-snug torque. Projecting maximum and minimum elongations up to the tension–elongation curve shows that there was very little variation in the preload achieved in this group of bolts, because A325 bolts are made of a ductile material having a long, flat plastic region.

8.5.2.2 Turn-of-Nut Procedure in Production Operations

Modifications of the original turn-of-nut procedure are often found in industry; a torque-then-turn procedure is used, and the bolts are stretched past their yield point. Fasteners are stretched perhaps 0.001–0.002 in. (0.025–0.050 mm) and preload accuracies of $\pm 8\%$ are achieved [12]. Other torque–angle strategies are also used, perhaps more commonly, in manufacturing. We look at some of these below.

Figure 8.13 shows the results of an experiment in which a large number of 5⁄16–24, SAE Grade 8 bolts, with a grip length of 2.3 in., were tightened with torque–turn air tools against a load washer and a pair of steel blocks [4]. The fasteners were not tightened past yield so the scatter in preload is about 1.7:1. Not as good as yield control, but substantially better than the scatter obtained in a similar group of bolts with torque control, however, as you can see from Figure 7.5.

8.5.2.3 Turn-of-Nut Procedure in Aerospace Assembly

A number of knowledgeable companies have developed manual torque–turn procedures, which they call "turn-of-nut," but which do not involve tightening the fasteners past the

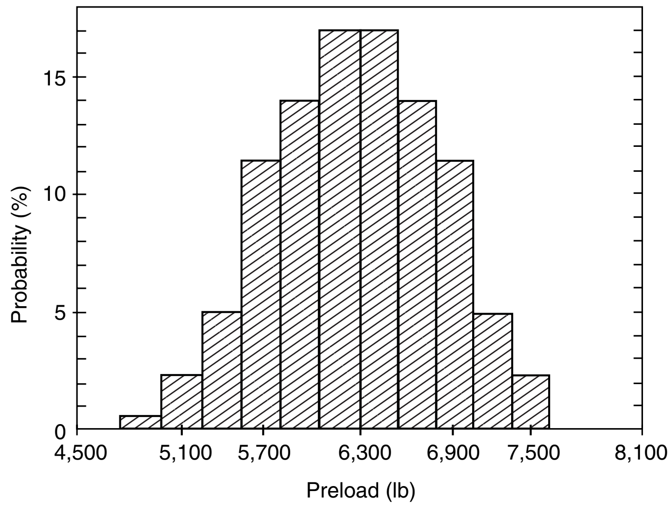


FIGURE 8.13 Histogram of the preload achieved in a group of 5⁄16–24 SAE Grade 8 bolts with a 2.3 in. grip length when they were tightened to less than the yield point by torque–turn procedures. (Eshghy, S., *Fastener Technology*, pp. 47ff, July 8, 1978.)

yield point. Experience shows that some of those systems provide additional accuracy over systems using torque or turn alone. Here's an example from the aerospace industry.

An aircraft engine manufacturer applies a seating torque of 3000 lb-in. to a large nut located on the central axis of the engine. This is one of several nuts along the axis; they clamp together the rotating parts of the turbine engine. The nut is now loosened completely and is then retightened to 5000 lb-in.

The nut is loosened a second time and is now retightened to a snug torque of only 500 lb-in. A turn protractor on the wrench is now set at 0°, and the nut is turned a specified number of degrees and minutes. This final turn can require as much as 120,000 lb-in. of torque.

The final torque is measured and recorded as a cross-check, to make sure that nothing has gone wrong, but turn and turn alone is used for the final control.

There are probably two reasons why preload accuracy is improved here, even though the fastener is not taken anywhere near yield by this particular torque–turn procedure. First, the initial cycling of the fastener will reduce subsequent embedment relaxation. Also, the initial cycling guarantees that all parts of the assembly have been pulled together before the official snug torque is applied. This procedure therefore provides a more stable starting place for the final cycle, reducing some of the major uncertainties of the torque–turn procedure. Note, however, that the technique is being used on a high-quality assembly, whose parts are subjected to far more quality control than are the parts of most bolted joints.

8.6 PRODUCTION ASSEMBLY PROBLEMS

Production assembly lines require high-speed bolting tools that must be able to detect and respond to torque, turn, and other variables more sensitively and rapidly than is possible for a human operator. The tools used are the same as or similar to the nut runners and screwdrivers described in Chapter 7, but these now must include angle as well as torque transducers. (Some suppliers use "current limit control" to control torque indirectly and to save the cost of the torque transducer. [13]) Computer or microprocessor control is a must for all torque–angle tools, of course. Before getting into details let's look at some of the problems the tool must deal with.

In the discussion so far in this chapter we have treated preload as if it depended only on the torque or turn accuracy of the tools, and on such parameters as the coefficient of friction, fastener geometry, the elasticity of various parts, etc. In practice, however, the engineer—especially the production engineer—faces many serious problems which have nothing to do with the theoretical behavior of an ideal fastener.

These problems include:

1. Blind holes
2. Holes not tapped deep enough
3. Wrong size holes
4. Wrong size bolts
5. Dirt in holes
6. Crossed threads
7. Partially stripped threads
8. Soft parts
9. Gross misalignment of parts
10. Chips under the head or in the hole
11. Tool malfunctions
12. Warped mating parts
13. Burrs

to which must be added the sometimes gross relaxation of parts that have been properly preloaded to start with. As we'll see in a minute, measuring both torque and turn makes it possible to spot problems of this sort.

Even systems that are capable of responding to the assembly problems listed above can not always provide enough control in really critical joints, especially when very high speed is a must. The controls might work fairly well on a "normal" joint. This means one with a relatively ductile bolt, having a conventional shape and a length-to-diameter ratio that is at least 2:1, used in a joint that contains no gaskets and has a "reasonable" volume of joint material. These same systems, however, would not always work when we were dealing with short, stubby bolts, very high-strength bolts, (low-ductility) sheet metal screws, some types of prevailing torque, etc. In situations of this sort the basic S-shaped torque–turn curve becomes distorted, as suggested by Figure 8.14.

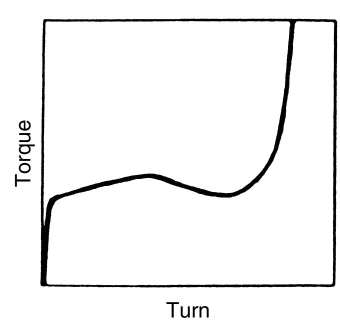


FIGURE 8.14 Torque-turn curve for a fastener having a prevailing torque lock nut. (Modified from Chapman, I. and Sharma, S., *Assembly*, May 2002.)

Each of these situations is characterized by a relatively high run-down torque, followed by a very sharp and sudden rise to final torque. In effect, the fastener sees enough starting torque to trigger many shut-off systems, and then slams into a torque wall, putting excessive demands on the response time of control systems and components [10].

The best way to cope with demanding joints of this sort is to provide more sophisticated control systems, to be described next. These systems also provide substantially improved preload accuracy for normal joints as well.

8.7 POPULAR CONTROL STRATEGIES

Presently available computer and microprocessor systems use many different algorithms to deal with the problems described above while controlling the tightening process. Here are just a few of the options available to the production engineer. Some and not all involve turn as well as torque. I should add that "turn" is called "angle" in most production strategies. The phrase turn-of-nut is restricted to those strategies in which the nut is first snugged and then turned a measured amount, as in the structural steel procedure. Strategies where turn is used to find things like blind holes, crossed threads etc. are classified as torque–angle methods.

8.7.1 Torque–Angle Window Control

An electronically controlled air or electric tool will produce first a preset torque, (typically 30%–50% of the final torque). The control system will then start to measure the angle through which the nut is turned, as it increases (and continues to measure) the applied torque. The system will evaluate the angle through which the fastener has turned when a predetermined torque has been reached and the tool is stopped. Note that the angle of turn is not controlled; it is simply measured [12]. If everything is all right, the torque and turn values will fall somewhere within an acceptable "window" of values, as suggested by Figure 8.15. If there is a blind hole or the threads are galled, however, then it will take far too much torque to produce the desired turn (and the tool will shut off when it exceeds the maximum acceptable torque). If the bolt is too soft, or the hole is grossly oversized, on the other hand, it will require much less than the rated torque to produce the anticipated turn, and the tool will again cut off.

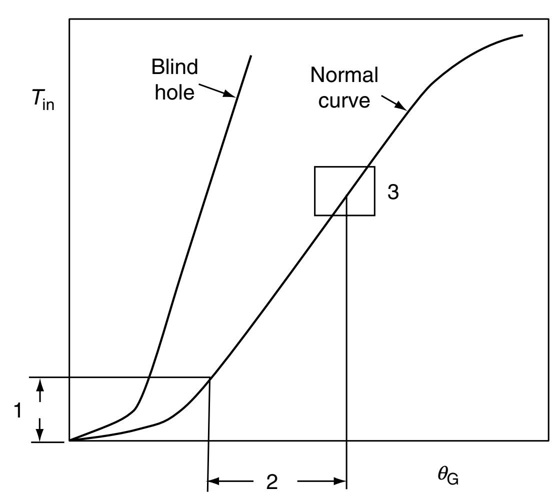


FIGURE 8.15 A typical torque–turn system will first apply a threshold torque (1), then move the nut a controlled angle of turn (2), and then inspect final (cutoff) torque and turn to see if final results lie within the window (3) of desired tolerance. If there's a blind hole, for example, the curve will miss the window, as shown.

Controlling torque and measuring angle-past-snug does not give better control over preload, which is controlled with a typical torquing accuracy of $\pm$25%–30%, but it does make it possible for the control system to detect problems one can't detect with torque control alone. And it can do this even though these are high-speed, automatic assembly operations.

Incidentally, torque–angle control can be fooled by certain combinations of the problems listed above. The combination of a soft bolt with a high-friction surface, for example, will often be interpreted by the control system as an acceptable fastener. Bolts which are too hard can't be detected, either. So, the equipment is very useful, but not infallible.

8.7.2 Torque–Time Window Control

Another, closely related, approach is to monitor the amount of time required for torque to build up to the desired level. It's easier and cheaper to measure time than turn, and it does approximately the same thing—it allows you to spot gross aberrations caused by crossed threads, soft parts, etc. Unlike torque–turn control, however, torque–time control probably doesn't give you improved control over preload when everything is normal. A typical torque–time curve is shown in Figure 8.16.

8.7.3 Hesitation and Pulse Tightening

Hesitation and pulse tightening are both torque–time strategies. As we've seen, neither turn nor time measurement do much about the relaxation problem. The only cure for this is to "give it time to relax and then retighten it." Production system designers have attempted to cope with this problem by hesitation or pulse tightening of some sort. In hesitation, tightening the tool tightens a fastener part way, and then hesitates to give the fastener some time to relax before tightening it the rest of the way (Figure 8.17).

Some tools do all their tightening in a series of pulses, rather than continuously. Figure 8.18 shows the torque–time curve for a nut runner control system in which torque pulses, separated by brief relaxation times, are applied to the fastener after initial tightening to compensate for relaxation in the fastener and joint.

Hesitation or pulsed tightening also gives better control at higher speeds. By turning the tool off automatically, whether or not it has reached final torque, you provide your control

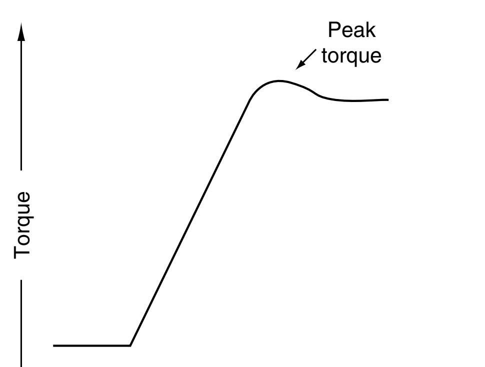


FIGURE 8.16 Torque–time curve of a standard stall-type air motor. A torque–time control system inspects final results to make sure that they are within the desired window of values specified by the engineers. (Courtesy of Ingersoll-Rand.)

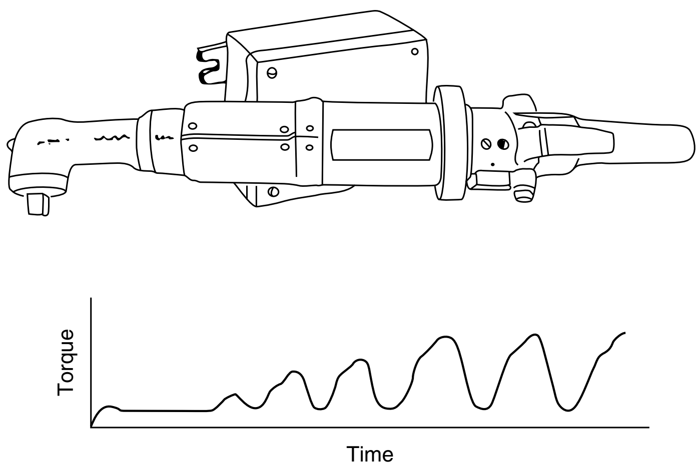


FIGURE 8.17 A simple air-tool control system in which the bolt is tightened in a series of pulses (Thor).

system with a number of opportunities to say "no more." Control decisions are always made with the tool at rest, and there is much less danger of overrunning the set point.

8.7.4 Yield Control

Some of the most sophisticated control systems available today are designed to tighten every fastener to the threshold of yield. Using various strategies, they watch the torque and angle relationships building up in a given fastener, recognize and measure the straight-line portion of the curve, and then shut off when they reach the upper bend in the curve—the point at which something starts to yield (see Figure 8.19). Note that since the torque–preload or torque–angle or turn–preload curves all "flatten out" at the yield point, further input to the fastener does not produce much more preload. This is why tightening to yield gives more accurate control of preload. Accuracies varying from ±3%–5% to ±8% have been claimed [9,11].

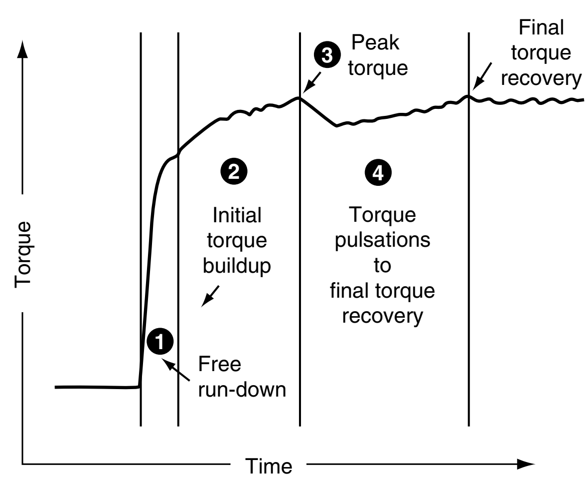


FIGURE 8.18 Equi-torq motor. Torque pulses are applied to the fastener after initial tightening, to compensate for relaxation. (Courtesy of Ingersoll-Rand.)

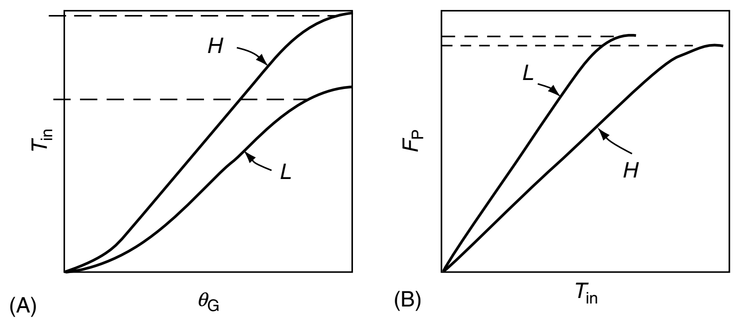


FIGURE 8.19 (A) A yield-point control system will cut off at point L on the torque–turn curve if friction is low. It will cut off at H if friction is high. (B) Looking at the same cutoff points on the torque–preload view of the torque–turn–preload curve, we can see that yield control does improve preload accuracy substantially. High friction still results in lower preload at yield because the additional torsion stress robs some of the bolt's tension capability.

The following is called the yield equation [11,12].

$$Y_S{}^2 = T_S{}^2 + 3S_S{}^2 \tag{8.4}$$

This suggests that the preload created at the yield point is

$$F_P = A_S\left(T_S{}^2 + 3S_S{}^2\right)^{1/2} \tag{8.5}$$

where
- F_P is the preload (lb, N)
- A_S is the tensile stress area (in.^{2}, mm^2)
- Y_S is the tensile yield strength (psi, N/mm^2)
- T_S is the tensile stress (psi, N/mm^2)
- S_S is the shear stress (psi, N/mm^2)

The shear stress is created by friction forces developed between male and female threads, which twist the fastener as it is tightened.

Note that tightening a group of fasteners to the yield point does not mean that each one will be tightened to exactly the same preload. Geometric and material variations will introduce some scatter. So will variations in friction, however. As we learned at the end of Chapter 3, torsion stress will rob some of the strength of a bolt. If the friction is high, therefore, torsion stress in the bolt will be high at the end of the tightening operation and the bolt will yield at a lower tensile stress level, i.e., lower preload. The differences probably won't be great in normal operation, but they could be substantial if you inadvertently forgot to lubricate normally lubricated bolts or the like.

The fact that torsion or shear stress absorbs part of the strength of the bolt during tightening is, in fact, useful in this situation. Remember that the fastener will recover its full tensile capacity after the torsion stress has disappeared. And it will disappear shortly after the tightening operation, thanks to embedment relaxation, at least in most applications. The tensile stress required to further yield the fastener is higher than that required to yield it in the first place. This means that the fastener which has been "tightened to yield" can support additional static and cyclic working loads without yielding any further—a very important factor in many applications. It can't support an indefinite increase in load, of course, but it can support perhaps 10% more than was required to yield it in the first place.

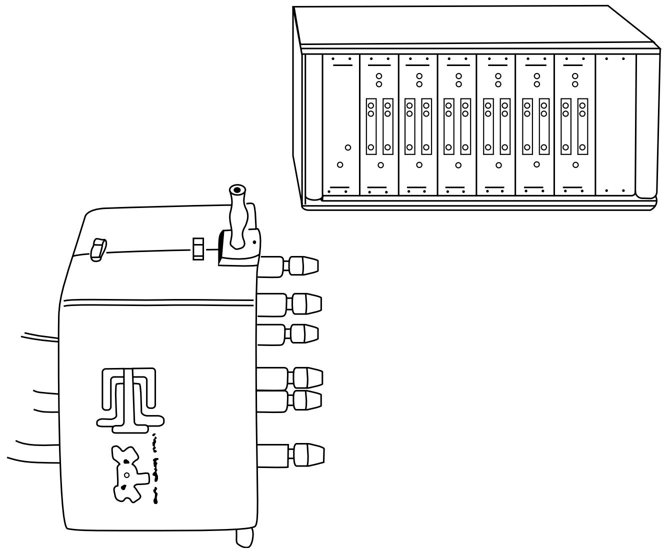


FIGURE 8.20 Multispindle air-tool control system designed to tighten fasteners to the yield point (SPS).

A sophisticated yield control system, shown in Figure 8.20, was introduced by SPS many years ago and is now available from Ingersoll-Rand, who acquired the SPS line of tools several years ago. In this strategy, rate of change of torque is monitored as a function of the angle of turn. The derivative (slope) of this curve peaks as the tool climbs the straight-line portion of the torque–turn curve; then it falls suddenly as the yield point is passed [8]. Variations in run-down torque or snugging angle of turn are ignored, because the computer has been programmed to look for a change only after it has seen a relatively high and relatively constant torque–turn ratio. The control curve can be seen, superimposed on the torque–turn curve, in Figure 8.20. This control system also uses torque–angle windows to spot gross problems such as crossed threads or soft bolts.

One of the advantages usually cited for yield systems is that they tighten each fastener to the maximum safe preload available from that fastener: to the yield point. This means, however, that for a given fastener they would offer the designer no choice of preload. He could not, for example, get "60% of yield" to accommodate dynamic loads, temperature changes, or the like. As a result, currently available yield control systems have been designed to operate in other torque–turn modes as well (torque–turn windows, for example).

Note that a ductile fastener tightened to yield still has a substantial useful life. It hasn't been "damaged": it has just been work-hardened a little. Thanks to joint relaxation following preload, the designer can often count on purely elastic behavior from a bolt originally tightened to yield, as we have seen.

Yield control was originally used on applications where the customer wanted the maximum preload with minimum scatter: in automotive assembly operations involving things like connecting rods, main bearings, cylinder heads, and transmission ring gears. It may have been more widely used than it is at present, but there are still some who use it. It was always more popular in Europe than in the United States [13].

8.7.5 Turn-of-Nut Control

In a tightening strategy called turn-of-nut by automotive production engineers, the fastener is first snug tightened and then further tightened by a measured angle of turn which, according to the experts, produces an amount of bolt elongation (ΔL), which can be predicted by Equation 8.1.

The calculated ΔL is then used to program the controlling microprocessor or computer [12]. The bolts are not tightened past yield. Note that the bolt-to-joint stiffness ratio is ignored here: the engineers are assuming that all nut turn is converted to bolt stretch, that the joint is not compressed at all. The automotive joints on which this procedure is used apparently involve rigid joint members and flexible bolts. Perhaps they massage the computed ΔL in some cases. In any event, a resulting preload scatter of $\pm 15\%$ is claimed for this strategy.

8.7.6 Prevailing Torque Control

Some of the more sophisticated of the present day automotive-tightening procedures are called "prevailing torque strategies." The term "prevailing torque" has long been used in many industries to describe the torque required to run down a fastener when a thread locking feature (see Chapter 1) or something else resists that run down. In today's automotive world many fasteners are called upon to cut female threads in the joint members as they are tightened into aluminum or light alloy joint members. The prevailing torques required to start such a fastener and the often slightly lower prevailing torques required to cut female threads may often exceed the final torque used to seat the fastener. The microprocessor recognizes and accepts these three torques—starting, run-down, and seating—while controlling the assembly. The controlling algorithm stops the process if any of these torques are above predetermined limits. Once the fastener has been correctly seated the controller uses torque or angle or yield control to tighten the fastener [13].

8.7.7 Plus—Permanent Records

Once you have a computer, of course, it becomes possible to manipulate the torque, turn, preload, or all these data in a number of different ways. Fastener assembly systems can be tied to larger computers used for production-control purposes, for example. Some systems provide a continuous statistical analysis of the problems encountered on the assembly line. This helps quality assurance inspectors to spot things like a high percentage of faulty parts, improper procedures at previous stations on the line, etc. Hard-copy records of torquing operations can also be kept for warranty or liability protection purposes, as well as for production-control purposes.

8.7.8 Meanwhile, Out in the Field

Many manufacturers are reluctant to switch to a new production method of fastener control if they cannot provide the same sort of control to field service and maintenance people. New controls, for one thing, mean new software: engineering drawings, specifications, product manuals, etc. These add to the cost of adopting a new system. If production people are working with one set of specifications and drawings, and service people in the field are working with a different set, all sorts of confusion and uncertainty can arise.

Fortunately, the microprocessor makes it possible for companies that manufacture sophisticated production systems to offer the same "brain power" in semiportable or handheld tools, so that field people and production people can, indeed, use the same tightening strategies. Such equipment is not inexpensive, of course. For critical joints, however, where "accuracy" can mean better product life, safety, strength-to-weight ratios, fewer warranty claims, etc. a switch to the new systems can often be justified.

Manual turn-of-nut techniques are often used in the field to retighten fasteners originally tightened to the yield point, if the mechanic does not have the new microprocessor-controlled tools.

8.8 MONITORING THE RESULTS

The only thing we need to complete the picture is the ability to monitor the results achieved with the production or field systems. We need some means that is more accurate than the assembly systems—something which can be used by quality control or engineering personnel to set up the systems, calibrate them, recalibrate them, etc. Since the systems themselves have now become more accurate than manual torque wrenches, a tension-monitoring device of some sort is almost mandatory.

Tension load cells can sometimes be used, but these tend to alter the characteristics of the joint enough to affect the accuracy of their results (when compared against the inherent accuracy of the best control systems).

Ultrasonic equipment is used to monitor tool performance on critically important joints. Ultrasonic extensometers, which measure the stress or strain in a bolt, are discussed in Chapter 9. Such equipment is already being used by a few companies for quality control evaluation and analysis of joints tightened by yield control and other torque–turn systems. Results have been impressive, and we expect that this technology will play a large part in the future.

The fact that torque–turn systems cannot be monitored on line, in many situations, has led to the development of laboratory techniques for adjusting and supporting them. The equipment is used to assemble samples of the actual joint, in a laboratory where strain gages, ultrasonic equipment, load cells, and other test equipment can be used to inspect results. The torque and turn settings revealed by the laboratory tests are used on the production line. These settings are rechecked in the laboratory at frequent intervals, sometimes with each new batch of bolts. By this process the equipment is indirectly monitored.

It is also useful to subject the equipment to many load cycles, to determine whether or not set points shift, for example. The ideal way to do this would be to use the equipment to tighten many bolts, since that would perfectly duplicate the anticipated load patterns it will encounter in use. This would be very time consuming and expensive, however, so engineers have sought ways to simulate loads which absorb torque and require turn.

Tools can be used to drive electrical generators, or to fight partially engaged clutches or brakes. Even with a small air tool, however, such tests will generate a significant amount of heat energy which must be disposed off. And the load devices tend to be short lived.

One interesting load device that minimizes these problems is diagrammed in Figure 8.21. One end of a shaft, 4–5 ft in length, is connected to an air brake. A rubber tube, nearly

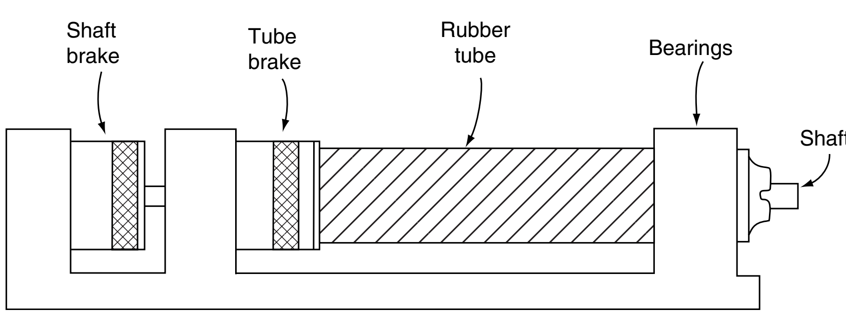


FIGURE 8.21 Diagram of a device used to load a torque–turn tool repeatedly for life tests. The load is provided by a long shaft and/or by a rubber tube, which are twisted by the tool as described in the text.

the same length, is held by another brake. The tool to be tested engages the far end of both shaft and tube. Depending on which brake is energized, the tool twists the shaft or tube until the tool stalls or is otherwise turned off. By varying the diameter of the shaft and the wall thickness or durometer of the rubber tube, operators can adjust the amount of torque and turn required from the tool. The equipment was developed by the Ford Motor Company.

8.9 PROBLEMS REDUCED BY TORQUE–ANGLE CONTROL

The block diagram Figure 6.28 illustrates the many factors that affect the results when we use a torque wrench to tighten a group of bolts. To what extent does the use of torque–turn control reduce the uncertainties refined there?

Torque–angle doesn't eliminate any of the factors shown in the diagram, but it does help us estimate and cope with several of them. For example, in the third row of boxes, having both torque and turn information will tell us whether or not we're encountering unintended prevailing torque, or excessive friction loss, or, sometimes, an abnormal amount of bending.

In the next row, it should help us detect the fact that there's bolt–hole interference and significant resistance to the joint members being pulled together. All this could be very useful, for each of these five factors can be a major source of uncertainty during the assembly process.

Torque–turn control does not, on the other hand, eliminate embedment or elastic interactions (although hesitation tightening will help). It does not change the effects that such things as working loads, vibration, thermal changes, etc. have on the clamping force in the joint. That's not unexpected. It's unfortunate, however, that this better assembly control technique does not provide us with a more accurate way to compensate, after initial tightening, for relaxation or in-service effects. We can't go back to a joint previously tightened by torque–turn and reapply the original torque-followed-by-turn, to compensate for post-assembly changes and reestablish the desired clamping force on the joint. In this respect, torque–turn is as "blind" as pure torque. But for initial tightening of individual bolts at least, it can give us some significant advantages over torque control.

8.10 HOW TO GET THE MOST OUT OF TORQUE–ANGLE CONTROL

In Chapter 6 we reviewed some of the things we could do to optimize torque control (see Section 6.7). Many of those steps are also appropriate for torque–turn control. You should, for example, train and supervise the crew, keep good records, make sure that bolts and joint members are in reasonable condition, use properly calibrated tools, etc.

In addition, the following steps can improve the results you get with torque–turn:

1. Determine the torque and turn you should use on your application by making one or more calibration tests on the actual joint, or on a model that simulates the actual joint as closely as possible. Use strain gages or ultrasonics or micrometer measurements to estimate bolt preloads. Repeat the tests periodically; for example, if you're about to use a new lot of bolts (especially if they've come from a new supplier).
2. Be sure that the bolt doesn't rotate while you're applying a measured turn. The turn values specified by the AISC are based on relative rotation between nut and bolt. Bolt twist is ignored (and is usually a small factor). If you define your own turn specifications, they should be based on the same assumptions.
3. Control the lubricity of your fasteners as well as possible. You may not need the degree of control you do for pure torque control, but you want to be sure that your snugging torques develop the desired minimum preloads, yet don't raise bolt tension so high

that subsequent turn will break the bolts. For example, protect the bolts used on structural steel jobs so they don't rust before use.

4. Be sure that joint members are properly aligned and in good contact before applying final snugging torques and final turns. Torque–turn controls can sense hole interference or joint resistance problems, but not overcome such problems. Use drift pins to align structural joints, for example, and snug the joint with a few bolts before inserting the final bolts. It you can't insert these easily, give more attention to alignment.
5. Be alert for anything that changes the stiffness of bolts or joint members. Earlier we learned that thin washers, used on oversized holes, had resulted in low tension in the bolts. Warped joint members, or springy joint members that have not been brought into full contact, can do the same thing.
6. In manual torque–turn operations, such as structural steel work, be sure to mark the joint surface and the nut before applying the final turn, to assure that the correct amount of turn has been used.

EXERCISES AND PROBLEMS

1. How do bolt stiffness and joint stiffness affect the angle of turn versus preload relationship?
2. A rubber washer has been used instead of a metal one, by mistake. How will preload be affected if the same predetermined torque and turn are used to tighten that bolt?
3. A nut having an effective length of 2 in. and a ½–16 UN thread is snug tightened and then turned 30°. How much preload will that create in the bolt?
4. Is that an acceptable preload?
5. Describe the typical torque–turn tightening process as described by an S-shaped torque–turn curve.
6. How is that process—that curve—modified if this is a sheet metal joint?
7. Do variations in friction between male and female threads affect the torque–turn relationship?
8. Describe the turn-of-nut procedure used in structural steel work.
9. Why does the structural steel turn-of-nut process create preload more accurately than does a torquing procedure?
10. Why is a combination of torque and angle measurement or control so popular in mass production operations?
11. What is meant by window control?
12. What is meant by yield control in mass production applications?
13. Does tightening a fastener to yield weaken or damage it? If so, why? If not, why not?
14. Name some of the bolt tightening problems or uncertainties not reduced by torque–angle or torque–turn control.

BIBLIOGRAPHY AND REFERENCES

1. Ehrhart, K.F., Preload—the answer to fastener security, *Assembly and Fastener*, June 1971.
2. Goodman, E., T. Hogland, and J. Miller, Establishment of standardization data for Monel and K Monel fasteners, Value Engineering Co., Report no.8-RD-65, Contract no. NObs-90493, AD 643 667, April 27, 1965.
3. Rice, E.E., Reliability in threaded fastener tensioning, Society of Manufacturing Engineers, Paper no. AD75-795, 1975.
4. Eshghy, S., The LRM fastening system, Society of Manufacturing Engineers, no. AD77-716, 1977.
5. Specification for Structural Joints Using ASTM A325 or A490 Bolts, Research Council on Riveted and Bolted Structural Joints of the Engineering Foundation, November 13, 1985.
6. *High Strength Bolting for Structural Joints*, Bethlehem Steel Corp.

7. Fisher, J.W. and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, Wiley, New York, 1974, p. 57.
8. Boys, J.T. and G.H. Junker, Modern methods for controlling the tightening of fasteners with power tools, *Design Engineering*, p. 21, January 1975.
9. Eshghy, S., The LRM fastening system, *Fastener Technology*, pp. 47ff, July 8, 1978.
10. Ehrhart, K.F., Preload—the answer to fastener security, *Assembly and Fastener Engineering*, June 1976.
11. Chapman, I., *Choosing the Optimum Fastening System*, Ingersoll-Rand Corporation.
12. Chapman, I. and S. Sharma, Choosing the optimum fastening system for the assembly of threaded fasteners, *Assembly*, May 2002.
13. Information received via e-mail from Ian Chapman of Ingersoll-Rand, Annandale, NJ. May 3, 2006.

9 Other Ways to Control Preload

We applied torque, and the nut turned, stretching the bolt and creating preload. We have seen that control through torque or turn, or both, did not give us ideal control of preload. We need something better for critical applications. In this chapter we're going to look at several other ways to control bolt preload. These include "stretch control" in which the change of length of the fastener is monitored, and a variety of control means or systems called "direct preload control." Some of these actually come close to being that; others don't, as we'll see. Most of these other ways to control preload control it more accurately than torque or torque–turn control, but most are also more expensive to apply. To complete the survey we're going to look at ultrasonic control of stretch and preload.

9.1 STRETCH CONTROL: THE CONCEPT

With torque or turn, we're trying to control the tightening process through the forces applied to, or the motion of, the nut. What we're really interested in is the bolt, however, since this is the thing which is being stretched to produce the clamping force on the joint.

As we saw in Chapter 5, we can consider the bolt to be a stiff spring. The relationship between the change in length of the bolt and the preload within it can be described by

$$\Delta L = F_{\mathrm{P}}\left(\frac{1}{K_{\mathrm{B}}}\right) \tag{9.1}$$

where

ΔL is the change in length of the bolt (in., mm)
K_{B} is the stiffness of the bolt (lb/in., N/mm)
F_{P} is the preload in the bolt (lb, N)

This equation says that the change in length of a bolt is equal to the preload within it times a constant. We can use Hooke's law to determine the constant in terms of the bolt properties and dimensions, as we saw in Chapter 5. For a common hex bolt, with a body, for example,

$$\Delta L_{\mathrm{C}} = F_{\mathrm{P}}\left(\frac{L_{\mathrm{be}}}{EA_{\mathrm{B}}} + \frac{L_{\mathrm{se}}}{EA_{\mathrm{S}}}\right) \tag{9.2}$$

where

L_{be} = effective body length (in., mm) (see Chapter 5)
A_{B} = body cross-sectional area (in.^{2}, mm^2)
E = modulus of elasticity (psi, MPa)
L_{se} = effective thread length (in., mm) (see Chapter 5)
A_{S} = effective cross-sectional area of threads (in.^{2}, mm^2)
ΔL_{C} = combined change in length of all sections (in., mm)

Note that we have now apparently eliminated most of the factors which deal with the relationship between two or more parts in the system—factors which gave us major control problems when we were dealing with torque and turn. We've eliminated, for example, the friction between nut and bolt threads and that between nut and workpiece. We've eliminated the ratio of the spring constant or stiffness of bolt with respect to that of the joint. We've eliminated the need to measure the turn-of-nut with respect to the bolt or the work piece. When dealing with stretch control we are looking, basically, at the bolt alone, and this provides an enormous simplification—and a subsequent improvement in control accuracy. Equation 9.2 suggests that if we can measure the change in length of the bolt accurately, we can determine the preload with the same degree of accuracy.

One interaction between bolt and joint does remain, however. The effective length of the threads depends on the grip length of the joint—on the combined thickness of joint members. The relationship between bolt stretch and bolt preload will be affected by variations in grip.

Note that all of the variables that remain could be measured and controlled if we needed the ultimate in preload accuracy. Such unmeasurables as the coefficient of friction or relative turn between nut and bolt have all been eliminated.

One attractive feature about stretch control is the fact that we can use it to measure residual preloads long after the fastener has been tightened. To do this we must keep a permanent log of the original length of each fastener, which can be a nuisance. But, when necessary, we can track preload during the life of the bolt merely by comparing its present length to its initial length. By comparison, we can never return to a previously tightened bolt and measure residual preload, accurately, with torque or turn tools.

Another attractive feature of stretch control, for those of us who like to monitor the energy content of bolts and joint members, is the fact that stretch measurements, combined with the related preload estimates, give us our best estimate of the amount of energy stored in the bolt. Take another look at Figure 5.6 and Equation 5.13 to see what I mean. The world at large doesn't base preload control or maintenance on energy estimates yet, but those who use various forms of torque–turn control are coming close to doing this. Those who use stretch control come even closer, whether they realize it or not. I suspect that future designers of bolted joints and automated production tooling will pay more attention to energy content and loss than they do at present.

9.2 PROBLEMS OF STRETCH CONTROL

At first glance it looks as if we have now achieved our goal and have found a practical way to measure preload. The bolt is, after all, a relatively simple shape. The modulus of elasticity is a well-known and well-defined quantity. Bolt dimensions are defined and controlled by a variety of different specifications. Our problem is solved.

Or is it? Unfortunately—but not surprisingly—we find that we are still faced with variables and uncertainties. Here are some of the uncertainties.

9.2.1 Dimensional Variations

Every dimension on a fastener is, of course, subjected to manufacturing variation or tolerance, even for something as highly standardized as a bolt. The tolerances allowed by bolt standards on body and thread lengths, however, are quite wide in most cases and can introduce errors in measurements based on stretch, because the elasticity of the body is so much less than the elasticity of the threaded section [1].

9.2.2 Change in Temperature

All dimensions, including thread and body lengths, will increase or decrease as temperature increases and decreases. We must monitor and record temperatures if we're going to measure bolt lengths some time after starting to tighten them.

9.2.3 Plastic Deformation of the Bolt

Stretch control assumes that the fastener has stretched elastically. It can't be used for fasteners tightened past yield.

9.2.4 Bending and Nonperpendicular Surfaces

A bent bolt will stretch more along the convex side than the concave side. We're interested in the average or centerline stretch, so significant bending caused by tapered joint surfaces or nonperpendicular holes can cause stretch measurement errors.

9.2.5 Grip Length

We'll need to know the in-service grip length of the bolt to compute the amount of stretch required to achieve a desired preload. Grip lengths are predictable in most applications, but can vary quite a bit in things like large-diameter, gasketed joints.

9.3 STRETCH MEASUREMENT TECHNIQUES

There are several traditional ways to measure bolt elongation in practice. None of them are practical for mass production operations, but they are used quite frequently for critical joints in heavy equipment, process systems, construction work, and the like.

9.3.1 Micrometer Measurements

The traditional way to measure bolt elongation, of course, is with a micrometer. If we have access to both ends of the bolt, as suggested in Figure 9.1, we can use a C-micrometer. There are a number of problems associated with micrometer measurements, however.

9.3.1.1 Irregular Measurement Surfaces

As we've seen, most bolts bend slightly as they are tightened, or if the ends of the bolt are not flat and parallel to each other as received. The bending is always invisible to the operator, but it can make accurate center-line measurement difficult or impossible. Raised grade markings are

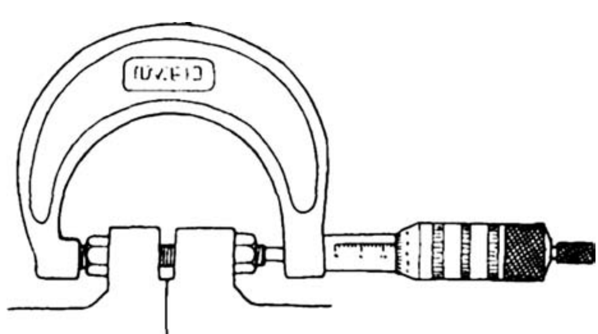


FIGURE 9.1 Using a C-micrometer to measure the change in length of a flange bolt.

another problem. One way to reduce measurement uncertainties is to embed small steel balls into the ends of the bolt and use the C-micrometer to measure the ball-to-ball length of the fastener.

9.3.1.2 Operator Feel

A certain amount of skill and feel is also required for C-micrometer measurements of conventional bolts or studs. On long bolts the operator has considerable difficulty in deciding whether or not the anvils of the micrometer are indeed flat against the ends of the fastener, even if the ends of the fastener are parallel. The result is that different operators will get different results.

9.3.1.3 Measurement Accuracy Required

A fairly high level of accuracy is required, especially if you are measuring small bolts (although the smaller the bolt, the more accuracy you can get with micrometer measurements). Figure 9.1, for example, shows what a 0.001 in. error means as a percent total elongation and as a function of the grip length of the bolt. If you're trying to control preload within $\pm 10\%$, at 50% of yield (of a Grade 5 or B7 bolt), then ± 0.001 in. accuracy is acceptable only for bolts more than 5½ in. in effective length. If you want $\pm 2\%$ control of preload in this bolt, then elongation measurements must be made to the nearest 0.0002 in. If the effective length is only 1 in., then $\pm 2\%$ control would require an accuracy of ± 0.00004 in. And in each case we're assuming that the only errors are measurement errors, which is highly unlikely.

9.3.1.4 Depth Micrometers

It isn't common to use C-micrometers to control preload in bolts. It is common, however, to use depth micrometers to control preload in very large studs, especially those tightened by heater rods or hydraulic tensioners. A loose rod having ground and parallel ends is placed down inside a hole which has been gun-drilled through the center of the stud, as shown in Figure 9.2. The lower end of the hole is capped by a threaded plug; or the rod has an enlarged end which is threaded, if you can afford or must have one rod per stud.

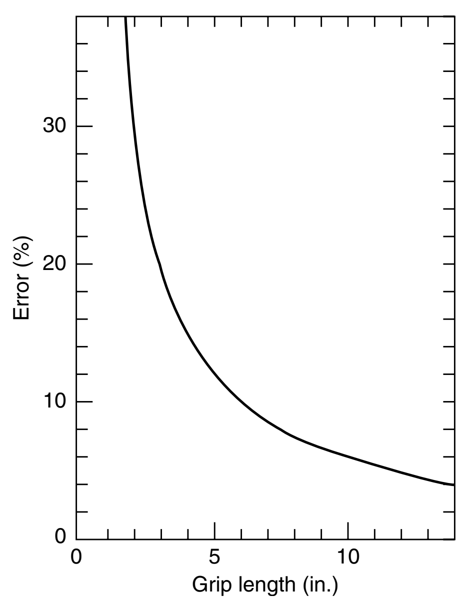


FIGURE 9.2 A measurement error of 0.001 in. leads to a larger percentage error in predicted preload in a short fastener than it does in a long one. The curve assumes that all fasteners are loaded to 50% of yield (SAE Grade 5 or ASTM B7 fasteners).

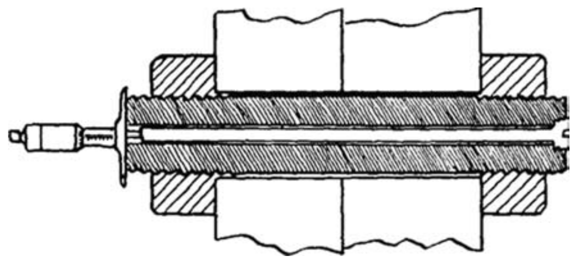


FIGURE 9.3 Using a depth micrometer and gage rod to measure the change in length of a stud.

The micrometer is now used to measure the distance between the end of the stud and the rod in the center. As the stud is stretched, this distance increases because the rod, being loose, is not stretched. The procedure is simple and the accuracy quite good. The distance measured is small, even when a large stud is involved, so problems of "feel," etc., are much less than with a large C-micrometer. The reference anvil of the depth micrometer should always be oriented in the same clock position to minimize errors from nonparallel or nonflat surfaces, however.

It is not necessary for the gage rod to run the entire length of the bolt. If it does not, of course, you will only be measuring stretch in a portion of the bolt and must take this into account. But that is usually easy to do. Under these circumstances, it is necessary to control the depth of the gage hole from bolt to bolt so that you will be measuring the same amount of bolt each time and don't have to make a separate calculation for each one. This is especially true if the hole ends in the threaded region of the bolt, since these portions of the bolt stretch more than does the body.

Except in very long bolts, I think it's better to run the gage rod through to the other end, as suggested in Figure 9.3. It can be threaded into the far end of the bolt or can be retained by a small threaded plug.

Note that the gage rod makes it possible for us to measure the stretch in one end of the bolt with respect to the other end. This is important if the bolt or stud is tightened into a blind hole and we have access to only one end. More important, however, it allows us to measure stretch without introducing the sort of errors we would encounter if we tried to use any other surface as a reference point. The bolt as a whole can move toward the nut as the nut is tightened, flange surfaces deflect, nuts embed themselves into flange surfaces, etc. The measurement accuracy required here demands that the other end of the bolt itself must somehow be used as the reference point.

Note that gage rods provide a built-in record of the change in length of the stud. A depth micrometer can be used at any time after assembly to determine the residual stretch in the stud. With other stretch measurement techniques, such as C-micrometers or ultrasonics, it's necessary to keep a log of the original lengths of the studs, to compare against present lengths, for post assembly measurements.

9.3.2 Other Techniques

9.3.2.1 Dial Gages

Dial gages can be used instead of depth micrometers on very-large-diameter studs or bolts. They're used to measure the distance between the end of the stud and the end of an internal gage rod, in the same fashion that a depth micrometer was used.

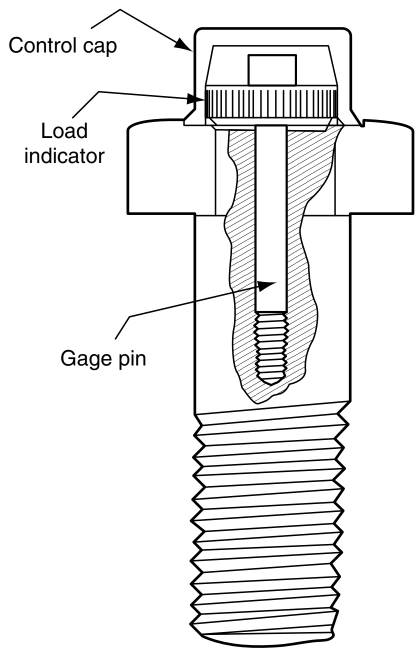


FIGURE 9.4 A preload-indicating bolt provided by RotaBolt Limited, in England. A rotating "load indicator" is preset at a prescribed air gap which is closed when the bolt is stretched during tightening.

Note that a dial gage cannot be set on the flange surface and used to check the motion of the end of the bolt as it is tightened because, again, of deflections and general motions in nut, bolt, flange surfaces, etc. and change in length only. The gage must be set on the bolt by hand, under the head of the gage screw. To my knowledge this system is not commercially available. It has been patented by NASA and could be used under license arrangements with them.

9.3.2.2 Commercially Available Gage Bolt

Figure 9.4 shows a commercially available gage rod bolt with an integral measuring device provided by RotaBolt Limited, in England. A rotating load indicator is preset at a prescribed air gap above the end of the fastener. The fastener is tightened until a control cap can no longer be turned by hand. The entire measurement system, load indicator, control cap, etc. remain a part of the fastener and can be used to recheck for residual stress at any time after assembly.

9.3.2.3 Ultrasonic Measurements

Any sort of micrometer measurement is clumsy and time consuming and leaves something to be desired in accuracy. These facts have minimized the number of applications in which bolt stretch is used to control preload. A relatively new technology involves the ultrasonic measurement of the change in length of bolts, and may make stretch control far more common. This technology will be discussed at length later in this chapter.

9.4 HOW MUCH STRETCH?

The amount of stretch you want will, of course, be determined by the amount of preload you want in the fastener. We won't be able to answer the question until after we've considered working loads on the bolt, and failure modes. Table 9.1, however, shows the amount of

TABLE 9.1
Typical Elongation Chart for Common Bolting Materials

Bolting Material	20% of Yield	40% of Yield	60% of Yield	80% of Yield	100% of Yield
Monel 40 K psi y.s.	0.3	0.5	0.8	1.1	1.3
SAE GR 2 55 K psi y.s.	0.4	0.7	1.1	1.5	1.8
SAE Grade 3; B7 and B16 over 4 in. diameter 80 K psi y.s.	0.5	1.1	1.6	2.1	2.7
SAE Grade 5; A325; B7 and B16 up to 4 in. diameter 96 K psi y.s.	0.6	1.3	1.9	2.6	3.2
SAE Grade 8; A490 120 K psi y.s.	0.8	1.6	2.4	3.2	4.0
Inconel 718 180 K psi y.s.	1.2	2.4	3.6	4.9	6.1
4340 steel, RC47 200 K psi y.s.	1.3	2.7	4.0	5.3	6.6
Best available high-bolt material 240 K psi y.s.	1.6	3.2	4.8	6.4	8.0
Titanium (6A14V) 134 K psi y.s. $E = 17 \times 10^6$	1.6	3.2	4.8	6.4	8.0

Source: Courtesy of Raymond Engineering Inc., Middletown, CT.

Note: Indicated elongation figures (in thousandths of an inch) are for various percentages of yield strengths (y.s.) of different bolts with a 1 in. grip length. (Modulus of elasticity assumed to be 30×10^6 unless otherwise noted.) To obtain desired elongation for a particular metal, read the elongation figure under the appropriate percentage of yield and multiply by the grip length in inches. For example, to obtain the expected elongation for an SAE Grade 5 bolt stretched to 80% of yield, with a 5 in. grip length, select the appropriate figure, which in this case is 2.6, and multiply by 5. The answer is 0.013 in. Note and warning: Many factors determine the ‘‘correct’’ stretch for a given fastener and application. Use this table with caution.

stretch you might see in various bolts, per inch grip length, if they were loaded to 50% of their yield strength. These are typical values, the way that the torque or nut factor values given in Chapter 7 are typical. It would be advisable for you to calculate the expected stretch in your application by using Equation 9.2 or, better still, by determining the preload–stretch relationship experimentally, especially if your bolt has a grip-to-diameter ratio below 4:1.

We’ll consider the question ‘‘How much stretch (preload)?’’ at greater length in Chapters 10 through 12.

9.5 PROBLEMS REDUCED BY STRETCH CONTROL

Figure 6.28 lists the factors affecting in-service clamping force. All of these factors were potential problems when we used torque to control the tightening operations. Although some of the factors were made less threatening by torque–turn control, none of them were eliminated. What about stretch control? Does it help reduce the uncertainties?

It does help. A lot. With stretch control, we can estimate the tension remaining in the bolt after the assembly tools have been removed. This feedback allows us to ignore large groups of variables—or to compensate for their effects. Prevailing torque, friction loss, bolt–hole interference, embedment relaxation, elastic interactions—all will still affect the amount of torque we must apply to the fastener to tighten it, but with stretch control, we merely apply as much torque as is required to create the final, residual tension we want in each bolt. We won’t have to measure the applied torque (unless we’re concerned about galling of other severe problems). In practice, if we do keep track, we’ll probably find that it takes a different amount of torque on each fastener to stretch them by the same amount; but who cares?

If we use gage rods, or keep a log of initial lengths, we can also return to the fastener after some in-service life, and estimate the effects of such things as external loads, vibration, thermal effects, etc. So stretch control is a significant improvement over torque–turn control. At least it is as far as the measurement of residual preload or in-service tension is concerned. It's probably not an improvement as far as job cost is concerned, however, especially in production operations. So don't use it unless it's genuinely necessary.

Note that stretch control is still blind to any factor that affects the relationship between tension in the bolt and clamping force between joint members. If the joint members resist attempts to pull them into contact, stretch control will accurately reflect the effort the bolts are making, but won't be able to tell us if the tension in the bolts is merely fighting the joint members or is also, for example, providing pressure on a gasket. In many situations, of course, other observations or measurements (e.g., of the gap between flanges) may warn us that we have a problem, but usually won't tell us how much of a problem.

And before we get too carried away, remember that stretch measurements do not provide perfect estimates of bolt tension. Variations in such things as grip length, bolt geometry, the modulus of elasticity, etc. will still introduce error (scatter) in our preload estimates.

Nevertheless, from an accuracy point of view, stretch is significantly superior to torque or torque–turn in most situations.

There are, however, some things that torque–turn can do better than stretch control can do. A stretch control system can't detect a soft bolt, for example, or a bolt which is the wrong size or made of the wrong material. Torque–turn systems can usually spot these problems. So, each has its strengths and weaknesses. In general, however, when preload accuracy is the main concern, stretch control is usually superior to torque or torque–turn control.

9.6 HOW TO GET THE MOST OUT OF STRETCH CONTROL

To maximize stretch control you should do many of the things suggested for torque or torque–turn control: keep good records, train and supervise the crews, be sure that the parts you're using are in good shape, be as consistent as possible in your bolting techniques and procedures, calibrate the tools frequently, etc. And there are some items which should be given special attention.

1. Monitor any and all dimensions that affect the stiffness of the bolt and, therefore, the relationship between bolt tension and elongation. Body length, thread length, and grip length can be especially important.
2. For the same reason, it's useful to monitor the modulus of elasticity of the bolts. Don't assume that the modulus is 30 million just because the fasteners are steel. Knowing and controlling the modulus is especially important on long bolts (several feet in length, for example).
3. Be alert to variations in the flatness or parallelism on bolt ends. Dished ends, burrs, damaged surfaces, etc. can introduce errors when mechanical or ultrasonic techniques are used to measure stretch.
4. Monitor bolt temperatures if the before and after measurements are to be made at different times.

9.7 DIRECT PRELOAD CONTROL—AN INTRODUCTION

We applied torque, the nut turned, and the bolt stretched. We tried to control the tightening process through torque, turn, and stretch—and found errors and uncertainties involved with each. Is there any practical way to control stress or preload directly?

In most applications the answer is no. But a number of techniques are emerging, which come close to tension control, and which are usually claimed to be such. For example, there are tension-control systems that are based on the measurement of torque and turn, or on the measurement of strain. We looked at the torque–turn techniques in Chapter 8 and will now look at some of the strain measurement techniques—many of which are useful in special situations—and then we'll look at some as-yet-unavailable ways in which actual stress could, theoretically, be measured. We'll also look at a useful device that is widely, but erroneously, believed to be a tension-control device: the hydraulic tensioner.

My definition of what constitutes true and direct preload or stress control, and what does not, will seem over restrictive to some. For the purposes of this book, however, I think it's important for you to be able to tell the difference.

I don't mean to imply that the techniques to be described in this chapter are not useful, or that they are somehow false. All truly control tension with some degree of accuracy, usually better accuracy than can be achieved with a torque wrench. So let's take a look at some of these techniques.

9.7.1 Strain-Gaged Bolts

One way to determine stress, of course, is to use strain (not stress) gages. The technology here is well advanced, and with the proper procedures and instruments, you can determine stress with far more precision than will usually be required. You'll be measuring the strain at a specific point on the surface of the fastener, however, and so must be careful in locating your strain gages if what you're interested in is average tensile stress (preload). You can also determine separately such things as bending stress or torsion stress by proper positioning of groups of strain gages. Used properly, strain gages are probably the most accurate way to measure bolt tension at the present time. Preload accuracies of $\pm 1\%$ to 2% are reported [2,3]. At least one company (Strainsert, W. Conshohocken, PA) sells bolts and studs in which strain gages have been mounted.

9.7.2 Strain-Gaged Force Washers

Another way to measure preload is to use a force washer—a compressible ring that has been provided with strain gages. These preload load cells can be used to measure preload continuously while the fastener is being tightened. An obvious disadvantage is their cost—they have to be left in place after use to be meaningful. As a result, force washers are useful only for experimental measurements and for very special applications. Like strain-gaged bolts, however, they are a very accurate way to measure bolt loads.

9.7.3 Direct Tension Indicators

The interest in a guaranteed minimum preload has led the structural steel industry to adopt several new types of fasteners which improve the chances that the fasteners will be preloaded properly and make it easier to inspect previously tightened fasteners for minimum tension. These fasteners are formally classified as either "alternate design bolts," to be discussed soon, or fasteners which allow "direct tension indicator tightening."

The most common type of direct tension indicator (DTI) at the present time is a washer with "bumps" on its upper surface. In one of several assembly procedures, a DTI washer is interposed between the head of the bolt and the surface of the joint. As the nut is tightened, the bumps on the DTI washer yield plastically, reducing the gap between the head of the bolt and the washer. A feeler gage is used to measure this gap. When the gap has been reduced below a preselected maximum value, the tightening process is stopped. No subsequent turn-of-nut is

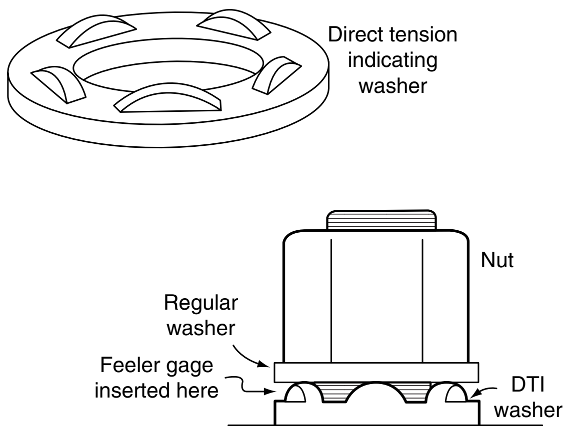


FIGURE 9.5 Direct tension indicator (DTI). This tension-indicating washer can be used under the head of a bolt or, as shown here, under the nut. If it is used under the nut, it is best to interpose a conventional washer between the DTI and nut. (J. & M. Turner, Inc.)

required in this case—this is a substitute procedure for turn-of-nut. In an alternate procedure, shown in Figure 9.5, the DTI is used under the nut rather than under the head of the bolt.

Several studies have been made to evaluate the accuracy with which the DTI washer controls initial preload in a fastener. In one series of experiments, the accuracy of the device, when used between parallel joint surfaces, ranged from +4%, −6% to +12%, −10%. When used on nonparallel surfaces (structural steel members are often tapered), the best-case accuracy was +15%, −11% and the worst-case, +23%, −15%. In every case, however, the minimum tension required in structural steel work was achieved [4].

Note that each DTI washer is built to compress the proper distance at a preselected preload. This means that you must use the right DTI to get the right preload. There have been situations in which steel erectors used DTIs built for ASTM A325 bolts in assemblies actually using ASTM A490 bolts. The results were low preloads—at A325 levels—instead of the higher preloads that were intended by the building designer.

Another type of crush washer, this one used by the aerospace industry and called a preload-indicating washer (PLI), is shown in Figure 9.6. The washer consists of four parts:

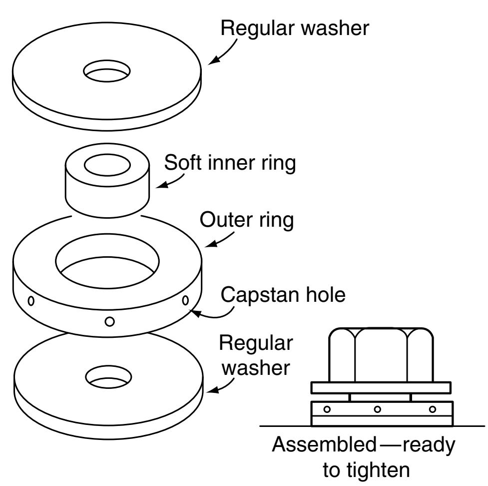


FIGURE 9.6 Preload-indicating (PLI) tension-control system used in aerospace applications.

two conventional washers, plus an inner ring and an outer ring with radial clearance in between. The inner ring is slightly thicker than the outer ring. As preload is built up in the bolt, the inner ring compresses. The operator stops applying torque to the bolt when the inner washer has been compressed to the thickness of the outer washer, and the outer washer can no longer be turned around the inner washer when a small pin is inserted in the capstan hole.

Again, studies have been made to determine the accuracy with which PLIs control initial preload. One investigator reports that preload varies from 65% to 95% of yield [5]. Others report characteristic accuracies of $\pm10\%$ of desired preload [6,7].

9.7.4 Squirter Self-Indicating DTIs

Applied Bolting Technology of Bellows Falls, VT, has patented a new type of DTI that has been well received by the structural steel industry. Small channels are stamped in the underside of a DTI, connecting the depressions (bumps) to the outer rim of the washer. The depressions are then filled with an orange colored silicone rubber compound which is allowed to cure. This so-called squirter DTI is now combined with a bolt, nut, and flat washer, following the procedure used with a conventional DTI and illustrated in Figure 9.5. When the fastener is first snug tightened, following the standard AISC or Research Council on Structural Connections (RCSC) procedure, a small amount of the orange material appears at the ends of the channels, allowing the building inspector to determine, at a glance, that the fastener has indeed been snugged. When the fastener is further tightened to the final tension more orange material is "squirted" out to indicate that. The building inspector no longer needs to use a feeler gage to determine whether or not the fastener has been properly tightened [8–11].

A Squirter DTI, like every other type of fastener used in structural steel, must be calibrated before use, with samples of a given lot being tested in a Skidmore–Wilhelm device (Figure 7.13) or equivalent. The assemblers must be trained to judge whether or not a sufficient amount of orange silicone has been ejected during assembly. But experience in the field shows that this is relatively easy to do. Tests made by independent groups have shown that desired bolt tensions can be achieved with a standard deviation of about 2.5% [11]. After inspection, the orange silicone is cleaned away by compressed air before the steel members are painted.

9.7.5 Twist-Off Tension-Control Bolts

The RCSC Specification for Structural Joints Using ASTM A325 or A490 Bolts used to allow the use of alternate design bolts, which were defined as those which incorporate a design feature intended to indirectly indicate tension—or automatically provide it. Figure 7.14 shows what was the most common form of this type of bolt—a twist-off bolt. In the latest, 2004 published version of this specification, however, the twist-off bolt is now recognized directly, with reference to ASTM F1852 [17].

The twist-off bolt cannot be held or turned from the head. (You'll note in the figure that it has an oval head.) Instead, the bolt is held by the assembly tool from the nut end. An inner spindle on the tool grips a spline section connected to the main portion of the bolt by a turned-down neck. An outer spindle on the tool turns the nut and tightens the fastener, with the tool reacting against the spline section. When the design torque level has been reached, the reaction forces on the spline snap it off, as shown in sketch 3 in Figure 7.14. The building inspector can determine whether or not a minimum amount of torque was applied to the fastener by looking to see whether or not the spline sections have indeed been snapped loose from the bolts.

Note that this is really a torque control system, not a direct tension-control system. The relationship between torque and tension, therefore, must be calibrated in a Skidmore–Wilhelm (Figure 7.13) or equivalent. If, between calibration and use, the bolts are allowed to become rusty or in any other way suffer a change of lubricity, then the amount of tension actually achieved in field assembly can be quite different from that achieved in the calibration stand.

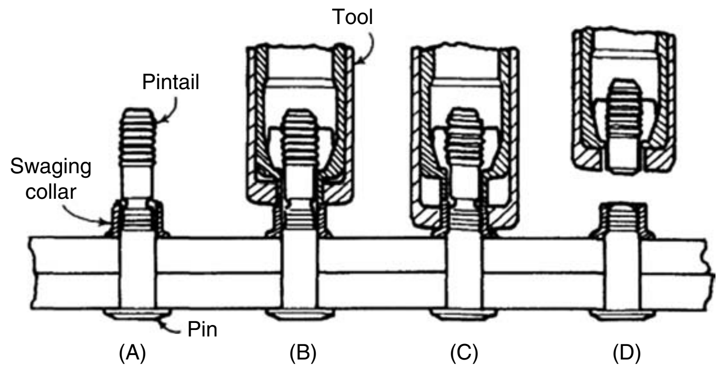


FIGURE 9.7 A "lockbolt." The tool applies tension by pulling on the pintail. A swaged collar, rather than a nut, is used to develop and retain further tension. The pintail is then broken free. (Huck Manufacturing Co.)

The fact that this fastener can be calibrated in the as-used condition, however, and, even more important, the fact that the inspector has a way to determine whether or not a minimum torque was applied to the fastener make this a popular item manufactured by a number of bolt suppliers. I include it here with fasteners which control tension more directly because the structural steel industry, at least, has accepted it as a bolt which automatically provides tension. In fact they're now calling these things "TC Bolts," the TC standing for tension control.

9.7.6 Alternative-Design Fasteners

The RCSC bolting specification still allows the use of otherwise undescribed alternative design fasteners" if they meet the mechanical, chemical, and physical requirements of A325 or A490 fasteners. One such fastener may be the lockbolt, as shown in Figure 9.7. At first glance it looks similar to a twist-off bolt, but is, in fact, quite different. In one configuration the nut is replaced by a swaging collar. The extended end of the bolt, called a pintail, has annular grooves on it rather than threads. The assembly tool grabs the pintail and pulls on it, creating a modest amount of tension in the bolt (which is called a pin in this case). The tool then swages the collar against annular grooves in the end of the bolt, creating substantial further tension. The tool now breaks the pintail section free from the bolt, once again providing an easy way for postassembly inspection. Although considered a bolt by many, this fastener could also be defined as a tension-controlled rivet. Another version, however, is more like a bolt. This bolt-like lockbolt, which is manufactured by the Huck Manufacturing Company, has threads instead of annular grooves on the pin. Once again, a collar is swaged onto the end of the fastener, but this time swaged into the threads. Also, this time, the swaged collar is hexagonal at its base, and so can be engaged with a conventional wrench if disassembly of the joint is necessary.

9.8 BOLT TENSIONERS

9.8.1 The Hardware

In all of the control strategies we have discussed so far, we have assumed that we used torque to tighten the fastener even if we used some other parameter to control the tightening. Because we were not measuring and controlling the buildup of bolt tension itself, we usually had to face a lot of uncertainty in the relationship between our control parameter—torque, for example—and the preload or tension we were after. It would be great if we could apply

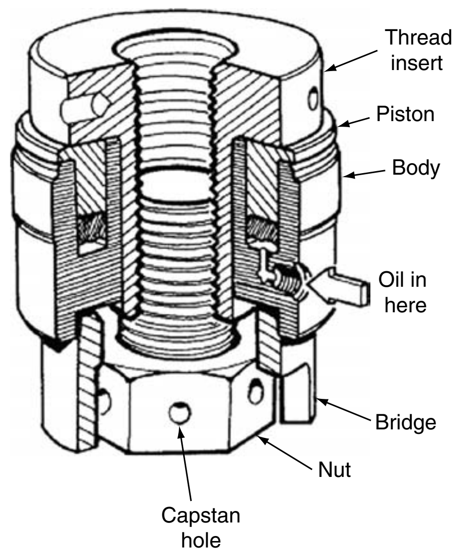


FIGURE 9.8 Cutaway view of an hydraulic tensioner. Oil is introduced under pressure to apply tension to the bolt. The nut is then run down with a capstan bar. (Raymond Engineering.)

and control the tension itself. This is the intent of that family of tools called bolt tensioners. As we'll soon see, these offer some significant advantages over the techniques we have talked about so far, but like the others are far from perfect.

Figure 9.8 shows a typical tensioner. Its operation can best be understood by reference to Figure 9.9. To start the tensioning process a threaded section of the tensioner, called a thread insert in Figure 9.8, is run down by hand over threads on the end of the bolt or stud to be tightened. Note that these stud threads extend beyond the threads engaged by the stud's own nut. This first step is diagrammed in Figure 9.9A.

Hydraulic fluid under pressure is now pumped into the tensioner, extending its piston and stretching or tensioning the stud, as shown in Figure 9.9B. At this point in the process, the stud contains a very precisely controlled amount of tension. Errors in the control of fluid

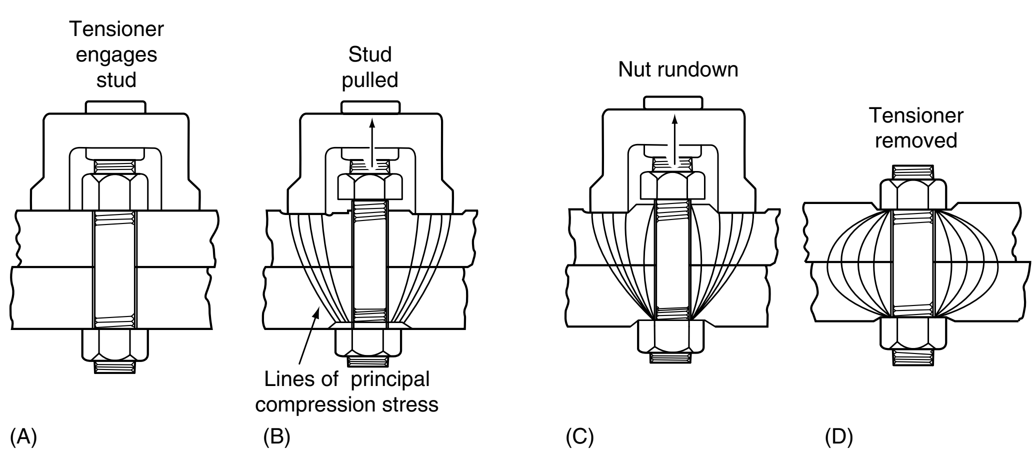


FIGURE 9.9 Diagram showing the sequence of operations by which a tensioner preloads a stud. See text for explanation.

pressure or friction between the piston and the rest of the tensioner, etc. will introduce a little uncertainty, but it will be minor.

If we could walk away from the system at this point, we would indeed have found a way to preload bolts with almost perfect precision. Instead, however, we now shift control of that tension from the tensioner itself to something else—in this case the stud's nut. We then rely on the nut to retain the tension introduced by the tool.

To accomplish this with the system shown in Figure 9.8, we insert a steel rod or a capstan bar in a hole in the stud's nut and use the capstan bar to run the nut down against the top surface of the joint, using as much torque as we can generate with the capstan bar (which isn't a great deal). This step is shown in Figure 9.9C.

We now depressurize the tensioner and remove it from the stud. At this point, only the stud's nut is retaining the tension introduced by the tool, as suggested in Figure 9.9D.

9.9 BOLT HEATERS

A bolt heater accomplishes the same thing as a bolt tensioner, although the equipment and procedures used are vastly different. A heating rod is inserted in a hole drilled down the axis of the bolt. The bolt expands (increases in both length and diameter) as it heats up. After it has increased in length by a desired amount, the nut is placed on the bolt and is run down hard against the top surface of the joint. The nut is supposed to retain the change in length of the stud. Preload builds up in the fastener as the bolt cools.

This is obviously not as well controlled a process as hydraulic tensioning. But people who use the technique report that a skilled crew can usually achieve the desired amount of residual tension in 60% of the studs in a joint with a first pass. Heater rods are then put back in the remaining studs in a second attempt to get the desired tension in them. By the third round, we're told, all studs have been brought to an acceptable tension.

The accuracy of the heater technique depends, of course, on the amount of stretch introduced into the fastener by the heater in the first place; then, on the way in which the nut is run down; then, on the way in which the nut creates and retains tension as the bolt cools. Embedment, elastic interactions, etc. will all still occur unless all bolts are tightened simultaneously and by the same amount, which probably never happens.

One big advantage of the bolt heater is that it is very inexpensive. The larger the stud (in diameter), the greater the cost advantage of the heater over the wrench or tensioner large enough to do the job. A disadvantage is that this is a relatively slow procedure, requiring a fair amount of skill on the part of the operators. Another possible disadvantage is the fact that decarbonization of thread surfaces can sometimes occur when the stud is heated. As we'll see in Chapters 15 and 16, decarbonization can increase the chances of fatigue failure or stress corrosion cracking.

9.10 PROBLEMS REDUCED BY DIRECT PRELOAD CONTROL

Refer to Figure 6.28, the block diagram showing factors which affect the in-service clamping force in a bolted joint if we use torque control during assembly. Which of these variables are eliminated or made less troublesome by direct tension control?

The answer depends to some extent, of course, on the type of direct tension control you're talking about. Let's take them in the order in which we have discussed them.

9.10.1 DIRECT TENSION INDICATORS

The direct tension-indicating fasteners, shown in Figures 9.5 and 9.6, would eliminate most of the problems diagrammed in row 3, such things as bolt twist, heat loss in the threads, and reaction to prevailing torque. They wouldn't be able to compensate for severe bending of the bolt, but this is rarely a real problem.

These indicating fasteners cannot, however, distinguish between tension in the bolt and clamping force on the joint interface, so they do not eliminate problems that would be caused by axial reaction force from the near side of the joint, by resistance of the joint members to being pulled together, etc. Recently, for example, a structural steel erector reported that the DTIs had flattened before the (misaligned) joint members had been pulled together. It was suggested that he continues tightening until the joint was snugged, then remove the bolts one by one, and replace and retighten them with fresh bolts and fresh DTIs.

9.10.2 Twist-Off Bolts

The fastener shown in Figure 7.14 is really, of course, a torque control device, not a tension-control device. It would, therefore, be subject to all of the uncertainties shown in Figure 6.28.

9.10.3 Hydraulic Tensioners

If used alone, without auxiliary control means, hydraulic tensioners again will only get around the problems shown in row 3 of the diagram in Figure 6.28. If, as is common, they're used in conjunction with a stretch control technique of some sort, they will also be able to compensate for such things as elastic interactions and subsequent vibration loosening. With or without stretch control they will be unable to detect the difference between bolt tension and clamping force on the joint interface, however. They will be blind to axial reaction forces from the joint or resistance of the joint members to being pulled together. Note that tensioners used without stretch control cannot detect elastic interactions, but they can eliminate them (if all studs in the joint are tensioned simultaneously) or reduce them (if several studs are tightened simultaneously).

9.10.4 Bolt Heaters

Bolt heaters reduce most of the problems reduced by hydraulic tensioners—the factors shown on row 3 of Figure 6.28. They are also as blind as tensioners to the differences between bolt tension and clamping force in the joint interface. Finally, they can probably reduce elastic interactions if all bolts in the joint are heated simultaneously. Since heaters cannot be coordinated as accurately as tensioners, however, they probably won't fully eliminate these interactions.

9.11 GETTING THE MOST OUT OF DIRECT PRELOAD CONTROL

As with the other control techniques we have discussed, there are certain universally useful things which you should do to get the most out of direct preload control: things like keeping accurate records of your tools, procedures, results; training and supervising the operators; making sure that the joint members are properly aligned and pulled together before final tightening; tightening the joint in several passes rather than in a single pass; working from the center or most rigid part of the joint outward toward the free edges; etc. See the end of Chapter 6 for further details.

There are also some special things you can do, depending on the type of direct tension control you are using.

9.11.1 Twist-Off Bolts and DTI Washers

Each of these devices require different procedures and precautions, and samples of each lot must be calibrated before use. The tools used in some cases are very special and should, of course, be those provided by the fastener manufacturer. The correct feeler gage should be used with a standard DTI washer. Assemblers using Squirter DTI's must be trained to do so. In general, you should obtain and follow the special instructions provided by the manufacturers of these products.

9.11.2 Bolt Tensioners

1. Be sure that the nut turns freely on the stud during rundown, while the stud is under tension. Remember that only the male threads are stretched by the tensioner. If close-fitting male and female threads are being used, stretching the male threads may actually create interference, which can affect the amount of torque required to run the nut down. It's best, therefore, to use coarse threads and to avoid a class 3 fit.
2. Run the nuts down with as uniform a torque as possible. Measure this torque if you can.
3. Tighten as many fasteners as you can afford simultaneously. The ideal thing would be to tighten them all at once, but this is often impossible. Nevertheless, the more the merrier, because this reduces elastic interaction effects.
4. Verify that the specified hydraulic pressure is applied to each tensioner. Make sure, for example, that all hydraulic connections have been properly made. Sometimes a press-fit hydraulic connection can appear to be completed, but not be, so that a tensioner could get no pressure at all.
5. Use data from the tensioner manufacturer to determine how much overtension you should introduce to compensate for elastic recovery.
6. Be sure that the base of the tensioner sits squarely on the joint surfaces, so that the tensioner will pull directly along the axis of the stud. A distorted base can cause interference with the nut, preventing smooth nut rundown.
7. For the same reason, be sure that studs are perpendicular to the joint surface. Tensioning will bend studs if this is not the case, again binding the nut during rundown and probably reducing the amount of residual tension created in the studs. Shimming can be used to compensate for nonperpendicularity.
8. If you are not tightening all studs simultaneously, make a final pass at the final tensioner pressure to compensate for elastic interactions. Apply the final rundown torque once more to each nut while its stud is under tension. If you get a lot of nut motion during this pass, it is probably wise to have still another pass under the same conditions.
9. Use a good thread lubricant on the fasteners to increase the preload created by a given torque.
10. Use thick, large-diameter washers at both ends of the bolt to increase the stiffness of the joint.
11. Elastic interactions, elastic recovery, and other relaxation effects will be reduced if the bolts are less stiff. You can make them less stiff by turning down the bodies, by gun-drilling them, or by using longer bolts of the same diameter (placing collars or Belleville springs under the nuts).

9.11.3 Bolt Heaters

1. Use as many heaters as possible simultaneously to minimize elastic interactions.
2. Go for the final stretch (preload) in a single pass to minimize the amount of time the heat is applied (and therefore minimize the possibilities of decarbonizing the studs).
3. Use gage rods, or ultrasonics, or some secondary control means to measure the residual preloads after the bolts have cooled.

4. Reheat and retighten those which aren't right after the first pass, second pass, etc.
5. Run the nuts down with a uniform torque, preferably measured.
6. Use less stiff fasteners or heavy washers, as with tensioners.

9.12 ULTRASONIC MEASUREMENT OF STRETCH OR TENSION

9.12.1 In General

We have discussed a variety of ways in which we can control the tools used to tighten bolts. In each case our goal has been to control the amount of tension produced in the bolts during assembly—or, even more important, to control the amount of clamping force created between joint members during assembly.

Every method we have considered—torque, torque–turn, stretch, and tension control—has had drawbacks and limitations. But each of these methods is good enough for many applications, thanks to the fact that most bolted joints are over designed or to the fact that we are usually not too concerned about the consequences of an occasional failure. In more and more applications, however, we are becoming concerned and would like to find a better way to control bolt tension and clamping force. Fortunately, a better way to control bolt tension, at least, has been slowly emerging in the last 30 years. I'm referring to ultrasonic measurement of bolt stretch or tension. These techniques allow us to get past dozens of the variables that affect the results we achieve with torque and turn control. They allow us to see and compensate for the elastic interactions and other factors which limit the accuracy with which we can tighten bolts with hydraulic tensioners. In short, they give us a new, more accurate, and often more convenient way to get the advantages of stretch control or strain gage control [12].

9.12.2 Principle of Operation

The basic concepts behind ultrasonic control of bolt stretch or tension are relatively simple. The most common systems available today are what we call "pulse-echo" or "transit time" instruments. A small acoustic transducer of some sort is placed against one end of the bolt being controlled. An electronic instrument delivers a voltage pulse to the transducer, which emits a very brief burst of ultrasound (typically two to three cycles). This burst passes down through the bolt, echoes off the far end, and returns to the transducer. The electronic instrument measures very precisely the amount of time required for this burst of sound to make its round trip in the bolt.

As the bolt is tightened, the amount of time required for the ultrasound to make its round trip increases for two reasons:

1. The bolt stretches as it is tightened, so the path length increases.
2. The average velocity of sound within the bolt decreases because the average stress level has increased. Both of these changes are linear functions of the preload in the fastener, so the total change in transit time is also a linear function of preload.

The instrument has been designed to measure the change in transit time, which occurs during tightening, and to interpret and report the results as a change in length of the fastener; or using the stretch measurement data plus several "constants" determined by a calibration exercise, the instrument can calculate and report the tension within the fastener. Because the change in length is measured directly, while the tension must be computed using that information plus other data about the bolt and joint, the stretch measurement is more accurate than the preload measurement. As a result, ultrasonics are more commonly used to for stretch control than for preload control.

Although the operation is easy to describe, it's not error free. The one and a half cycle signal pulse is often distorted a bit as stress builds up in the fastener and the bolt bends as it bends slightly. Most of the instruments incorporate a small oscilloscope, or are used in conjunction with one, so that the trained operator can compensate for such problems.

9.12.3 How It's Used

To make this text flow more smoothly, I'm going to call our transit time ultrasonic instrument a "bolt gage," a term used by Raymond Engineering Inc. and by its successor the Bidwell Industrial Group. Using such an instrument is simple. A drop of coupling fluid (glycerin, water, oil, grease, resins, etc. can be used) is placed on one end of the fastener to reduce the acoustic impedance between transducer and bolt. The transducer is placed on the puddle of fluid and is held against the bolt, mechanically or magnetically. The instrument is then zeroed for this particular bolt, because each one will have a slightly different acoustic length (even if their physical lengths are the same). If you wish to measure residual preload, relaxation, or external loads at some later date, you record the length of the fastener (at zero load) at this time. Next, the bolt is tightened. If the transducer can remain in place during tightening, it will show you the buildup of stretch or tension in the bolt. If it must be removed, it is placed on the bolt again, after tightening, to show you the results achieved by the torque, turn, or tension used.

If, at some future time, you wish to measure the present tension in the bolt, you enter the original length of that bolt in the computer in the bolt gage, then place the transducer back on the bolt. The instrument will show you the difference in length or stress between present and zero stress conditions.

Ultrasonic measurements require skill and are time consuming compared to, for example, control by torque. In many applications, therefore, they're used just to spot check a few bolts in a group after the bolts have been snug tightened, and are only used on all bolts during the final tightening pass. Many people, in fact, only use them to spot check the results achieved with a torque wrench or tensioner even on the final pass.

It's also possible to use a bolt gage to measure the residual tension in bolts whose preload was controlled by something other than ultrasonics. Let's assume, for example, that the bolts in a bridge were tightened years ago, with turn-of-nut being used to control preload, and the highway department now wants to know how much tension is left in the bolts.

In this case the bolt gage is used in a normal fashion, but the sequence of measurements is reversed. A transducer is placed on a previously tightened bolt and its acoustic length is measured. The bolt is now loosened, completely, and its acoustic length remeasured. The loss in length will be proportional to the tension that existed in the bolt before it was loosened.

If you wish to measure the residual tension in several bolts in the same joint, you must reinstall and retighten the first one to its original tension before measuring and loosening a second bolt. Elastic interactions between the bolts will change the residual preloads in others in the group when you remove the first.

9.12.4 Calibration of the Instrument

It is unfortunately necessary to calibrate an ultrasonic bolt gage for each application it is to be used on, because the velocity of sound varies with different bolting materials, and even for different lots of the supposedly same material. The ratio between the change-in-length effect of the transit time and the change-in-stress effect also varies from material to material and lot to lot. Ambient or service temperatures also affect sound velocities and must be accounted for. All of this mandates calibration.

Although there are cruder ways to do it, calibration is best performed in a tensile machine equipped with instruments which report the force being applied to the test specimen (in this case a fastener) and its resulting change in length. A bolt gage is also used to measure these things during the test, and the gage is adjusted so that its readings will agree with those of the tensile machine. These adjustments are recorded as several calibration constants, which can be used to reprogram the bolt gage, without further calibration, if it is to be used again on the same job.

Some companies also provide calibration gage blocks to help recalibrate an instrument they have originally calibrated before shipping it to the customer.

9.12.5 Presently Available Instruments

A number of different ultrasonic bolt gages are available at present, and more will presumably appear in the future. You can obtain updated information about them by going online to some of the people listed in the references at the end of this chapter, or by going online to "ultrasonic bolt gages" [13–16].

Instruments now on the market vary in size from handheld to small cabinets to modified laptop computers. They vary in resolution but most claim to be able to measure the elongation of shorter bolts to the nearest 0.00001 in. Those also capable of measuring the stretch of very long bolts; up to 450 in. (11.4 m) for the Power Dyne instrument [16] can do so to the nearest 0.001 in. This instrument can also store full data on up to 5000 fasteners on a data card. Additional cards can be used to store data on additional thousands of bolts. Another available bolt gage has internal storage for 8000 bolts [13].

Power Dyne also sells a transducer multiplexer box through which the bolt gage can be connected simultaneously to as many as 16 bolts. Measurements taken this way will usually be far more accurate and reliable than readings taken by periodically removing and replacing one or a few transducers. Each of the gages mentioned above can be used with ultrasonic transducers, which are manually held against one end of a bolt, or which are held mechanically or magnetically.

9.13 ULTRASONIC MEASUREMENTS USING PLASMA-COATED, THIN FILM TRANSDUCERS

It is now possible to buy bolts with thin film, piezoelectric, ZnO ceramic plus metal transducers permanently deposited on one end [15]. A variety of hand held or wrench mounted probes can be used to connect a conventional bolt gage to these transducers with none of the "expert feel" required to place a conventional transducer on the bolt. The bonded transducers are never removed and replaced; they have been successfully left in place and used after several years in service, even at operating temperatures of 350°C. In other applications probes mounted in right angle and in-line nut runners and other assembly tools have controlled the tightening of many thousands of bolts. Some of the probes are held magnetically to the fastener; still others are mounted in standard wrench sockets or even read through slip rings. Expensive "user training" is essential if one is dealing with a conventional bolt gage transducer: these deposited, thin film transducers make most of that unnecessary. The customer usually provides the bolts to be equipped with transducers if quantities are small: Intellefast gets larger quantities from the fastener manufacturer. At present (mid-2007) small quantities for test or development sell for about $30.00 each. Production quantities cost $5.00 to $7.00 each.

EXERCISES AND PROBLEMS

1. We want to tighten an ASTM A193 bolt having 4–8 UN threads and a yield strength of 96 ksi. The effective length of its body, L_{be}, is 12 in.; that of its threads, L_{se}, is 4 in. We're going to control the buildup of preload by measuring bolt elongation. How much

elongation do we want to create if the target tension is 80 ksi? (References: Equation 9.2 and Appendix E)
2. How much torque will it take to stretch the bolt that much if $K = 0.2$?
3. What will the preload be at that stretch?
4. Use Table 9.1 to check your answer by computing the grip length expected to give the combination of 84% yield and 41.4×10^{-3} in. stretch. What is that computed grip length? Does it seem to be approximately correct for the conditions described in problem 1?
5. In practice, how could you measure that much stretch in a bolt of this size?
6. The bolt of problem 1 is put in service and its temperature rises to 250°F. How much change in length of the bolt would you expect this increase in temperature to create? (Refer Table 4.5.)
7. If you were using a full length, centerline, gage rod to measure ΔL, how much change would you expect to see at 250°F?
8. What ΔL from "before tightening" would you expect to see at 250°F if you were using ultrasonics to measure stretch?
9. What change in preload from room temperature to 250°F would the answer to problem 8 suggest?
10. List three ways to control preload directly.
11. What are the advantages of a bonded ultrasonic transducer?
12. What is sometimes cited as a disadvantage?
13. Can torque be used to tighten a 6 in. diameter bolt? If not, why not?
14. Name two devices that are normally used to tighten bolts this size.

REFERENCES

1. Cornford, A.S., Bolt preload—how can you be sure it's right? *Machine Design*, vol. 47, no. 5, pp. 78–82, March 6, 1975.
2. Donald, E., Fatigue-indicating fasteners ("bleeding bolts"): a new dimension in quality control, *Fastener Technology*, March 4, 1979.
3. Osgood, C., How elasticity influences bolted joint design, *Machine Design*, March 1978.
4. Struik, J.H.A. and J.W. Fisher, Bolt tension control with a direct tension indicator, *AISC Engineering Journal*, vol. 10, no. 1, pp. 1–5, first quarter 1973.
5. Brenner, H.S., Development of technology for installation of mechanical fasteners, Report prepared under Contract no. NAS 3-20779 by Almay Research and Testing Corp., Los Angeles, for the Research and Technology Division, Manufacturing Engineering Lab, NAS 8-20779 CR-103179, 1971.
6. Investigation of threaded fastener structural integrity, Report prepared by Southwest Research Institute, San Antonio, Texas, under Contract no. NAS9-15140, DRL T1190, CLIN 3, DRD MA 129 TA, SWRI Project no. 15-4665, October 1977.
7. Rodkey, E., Making fastened joints reliable . . . ways to keep 'em tight, *Assembly Engineering*, pp. 24–27, March 1977.
8. Wallace, W., Squirters blind tested at the University of Idaho, *Distributor's Link*, Summer 2000.
9. Wallace, W., Leave your feeler gage at home, *Distributor's Link*, Fall 2003.
10. Curven, C., Lessons from the fields—Lesson 1: a torqued bolt isn't necessarily a tight bolt, *Distributor's Link*, Fall 2004.
11. Herr, J., Tests show Squirter™ DTI's are amazingly accurate, *Distributor's Link*, Spring 2003.
12. Hayman, Ed and W. Brown, Assembly of bolted joints, presented at a PVRC BFC workshop in Safety Harbor, FL, Feb. 6, 7, 2002.
13. Bolt tension meter, online brochure of the Electronic Equipment Co. of Cedarhurst, NY, May 2006.
14. Information received from American Remote Visions Co., Titusville, FL, May 2006.

15. Meisterling, J., Permanent thin-film ultrasonic transducers for real time tightening and long term fastener load measurement, PFW Technologies, East Hampton, CT, March 2006.
16. Information received from the Power Dyne Group of Bidwell Industries, Middletown, CT, March 2006.
17. *Specification for Structural Joints Using A325 or A490 Bolts*, June 3, 2004, Research Council on Structural Connections.

10 Theoretical Behavior of the Joint under Tensile Loads

We've now completed our study of the various ways in which the initial tension in a bolt—its preload—can be controlled during the tightening operation. Preload control has probably received more publicity and attention than any other bolting problem, but it's certainly not the only problem we face.

We're not just interested in the initial tension; we're interested in joints that don't fail in service during the expected life of the product. Correct preload is a critically important factor, but we also need to know and control the service or working loads on the bolt and joint, and need to understand the many ways in which joint condition and behavior change with time as a result of relaxation, corrosion, vibration, cyclic loads, etc. We'll consider these and other aspects of the problem in the next group of chapters, starting with an analysis of working loads and then looking at the many ways in which a joint can misbehave or fail—and what we can do about it.

Note that when we turn our attention from assembly to working loads, failure modes, and the like, we start to address a new, major topic. Our first topic, which has occupied most of our attention to date, was establishing the clamping force on the joint. Where does it come from? How can we get the clamping force we want? Now we begin our studies of an equally important, second topic: How stable will that clamping force be in service? What can change it? How much will it change? Under what conditions will it be lost altogether? Will anything increase it to a dangerous level? We're going to start this study by examining the way the joint responds when exposed to the normal, external loads it has been designed to support, loads created by external pressure, inertia, weight, etc. As we learned in Chapter 3, we often categorize joints by the type of loads they support—calling them "tensile joints" or "shear joints." In this and the next chapter we're going to look at the first of these—joints in which the bolts are loaded in tension. We define tension loads as those which are applied along a line of action more or less parallel to the axes of the bolts. Then, in Chapter 12, we'll study joints loaded in shear.

There are two key questions we must answer whenever we analyze the response of a joint to external loads:

- What is the maximum force or tension which, in the worst case, the bolts in the joint will be subjected to?
- What will be the minimum clamping force on the joint, again "worst case"?

We're interested in the answer to the first question because we don't want the bolts to break. We're interested in the answer to the second question because we now know, as we learned in Chapter 3, that the life and behavior of a bolted joint will be very short if there's too little clamping force between the joint members.

10.1 BASIC JOINT DIAGRAM

To understand the working behavior of a bolted joint—for example, its failure modes—we must first understand in some detail the forces and deflections in the joint—in other words, its elastic behavior. Figure 10.1, greatly exaggerated to illustrate the effects, shows what happens when we tighten a bolt on a flange or the like. Tightening the bolt sets up stress and strain in both the bolt and flange members. The bolt is placed in tension; it gets longer. The joint compresses, at least in the vicinity of the bolt. It always does this, regardless of how stiff it may appear to be.

10.1.1 Elastic Curves for Bolt and Joint Members

In Chapter 6 we constructed a "joint diagram," first for a preloaded bolt and then for the preloaded bolt after relaxation. We're now going to see how to extend those diagrams to include the effects of a tensile load on the joint. It's important for us to know where the diagram comes from, however, so let's first repeat the development of the diagram for a preloaded joint.

We start with elastic curves for the bolt and for that portion of the joint surrounding a single bolt, as in Figure 10.2. We then push those two curves together against a single, central, preload axis, as in Figure 10.3 to form the diagram of the preloaded joint. We then reduce the size of this triangle a little to account for embedment relaxation and for elastic interactions between bolts during assembly, as in Figure 6.27.

10.1.2 Determining Maximum and Minimum Residual Assembly Preload

10.1.2.1 The Equations

Now that we know all there is to know about preload scatter (thanks to Chapters 6 through 9) we must ask ourselves, "What preload are we talking about: average preload, maximum preload, or minimum preload?" Unfortunately, because it means we must construct two joint diagrams (or a more elaborate and slightly confusing one) we're interested in two things. First, we're interested in the maximum residual preload in individual bolts, because that will help us define the maximum loads seen by individual bolts in service. We don't want any bolts to break under excessive load [5].

We're also interested in the minimum average preload in the group of bolts which form this joint, because that will allow us to estimate the minimum clamping force created on the joint by all of the bolts working together. In a few cases we might be interested in the worst-case minimum residual preload in an individual bolt, but almost always that will be of much less interest than the minimum average because our main concern is for the total

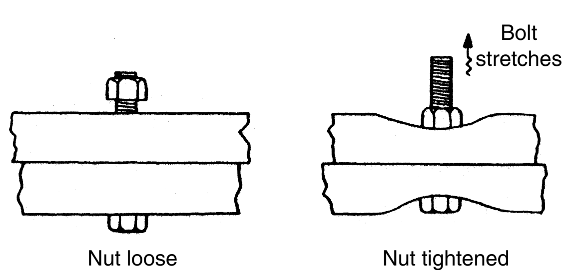


FIGURE 10.1 Tightening a bolt stretches the bolt and compresses the joint.

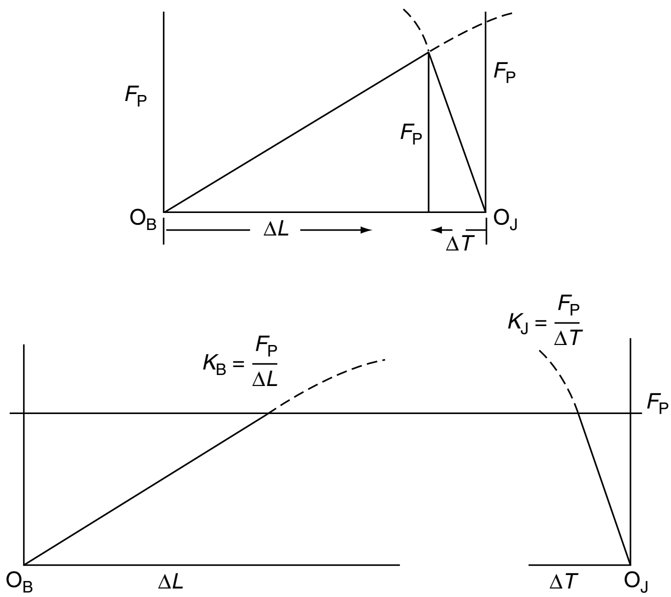


FIGURE 10.2 Elastic curves for bolt and joint members.

clamping force rather than the minimum, per-bolt clamping force. In spite of this, however, we start by considering both the maximum and minimum initial preloads created in individual bolts during assembly.

Let's call average, initial preload at assembly F_{Pa}. If the preload scatter during assembly is called $\pm s\%$ (with s expressed as a decimal), then

$$F_{Pmax} = (1 + s)F_{Pa} \tag{10.1}$$

$$F_{Pmin} = (1 - s)F_{Pa} \tag{10.2}$$

For example, if we're using torque control and expect the initial preloads achieved for a given torque to vary by $\pm 30\%$, then

$$F_{Pmax} = 1.30F_{Pa}$$

$$F_{Pmin} = 0.70F_{Pa}$$

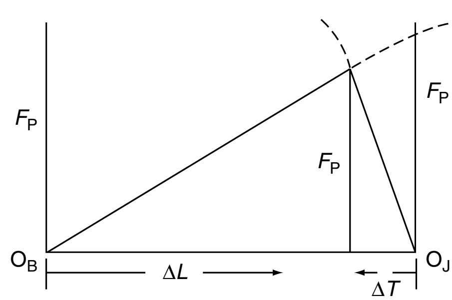


FIGURE 10.3 The elastic curves for bolt and joint can be combined to construct a joint diagram. O_B is the reference point for bolt length at zero stress. O_J is the reference point for joint thickness at zero stress.

As we saw in Chapter 6, the initial preloads will be decreased, during assembly, by embedment relaxation and by elastic interactions. We can extend Equations 10.1 and 10.2 to include these losses, thereby computing the maximum and minimum residual assembly preloads in individual bolts after relaxation loss. We'll call these Max and Min F_{Pr}.

$$\text{Max } F_{\mathrm{Pr}} = (1 + s)F_{\mathrm{Pa}} - \Delta F_{\mathrm{m}} - \Delta F_{\mathrm{EI}} \tag{10.3}$$

$$\text{Min } F_{\mathrm{Pr}} = (1 - s)F_{\mathrm{Pa}} - \Delta F_{\mathrm{m}} - \Delta F_{\mathrm{EI}} \tag{10.4}$$

When we use these equations we'll have to assume values for the embedment and elastic interaction losses, and we'll usually find it convenient to express the losses as fractions of the average preload. To do this we substitute the following in Equations 10.3 and 10.4:

$$\Delta F_{\mathrm{m}} = e_{\mathrm{m}} F_{\mathrm{Pa}} \tag{10.5}$$

$$\Delta F_{\mathrm{EI}} = e_{\mathrm{EI}} F_{\mathrm{Pa}} \tag{10.6}$$

where

e_{EI} = the percentage of average, initial preload lost as a result of elastic interactions; expressed as a decimal

e_{m} = the percentage of average, initial preload lost as a result of embedment relaxation; expressed as a decimal

10.1.2.2 An Example

Writing the equations is straightforward enough, but we encounter some difficulty when we try to apply them to a given case history or example. The problem comes in deciding what values to assign to the various terms. The many uncertainties associated with bolted joints force us to make a number of assumptions, based on prior experience, or on the published experiences of others.

As we've seen in previous chapters, for example, embedment relaxation might reduce initial preloads by 10%. Elastic interactions might reduce them further by an average of 18% for an ungasketed, two-piece joint; by 30% for a joint containing a sheet gasket; and by 46% for a joint containing a spiral wound gasket [6]. Let's assume we're dealing with an ungasketed joint. Let's also assume that we're going to use torque to control the buildup of initial preload in these as-received bolts as we assemble the joint, suggesting an initial preload scatter of $\pm 30\%$.

Theoretically, the last bolt tightened in a given region of the joint ends up with 130% of the average preload, less only by some embedment relaxation loss. It wouldn't experience any elastic interaction loss because it is the last bolt tightened in its region. It's highly unlikely, however, at least in my opinion, that a given bolt will see both the maximum initial preload and the minimum elastic interaction loss (none!). If the failure of an individual bolt in service would compromise a safety-related joint, and if the combination of initial preload plus working loads suggests that failure is possible, then we would have to accept $1.30F_{\mathrm{Pa}}$ as the maximum residual preload in an individual bolt. This would mean we'd have to use bolts and joint members large enough to support 30% more than average bolt tension; and that could seriously compromise the cost, weight, and size of the joint.

I'm going to avoid this by using engineering judgment to reduce the maximum anticipated preload. I'm going to assume that, worst case, an individual bolt will see the maximum possible initial preload (130% of average) but will also see average embedment loss of 10%

and average elastic interaction loss of 18%. From Equations 10.3, 10.5, and 10.6, therefore, I estimate maximum residual assembly preload to be

$$\text{Max } F_{\text{Pr}} = 1.30F_{\text{Pa}} - 0.1F_{\text{Pa}} - 0.18F_{\text{Pa}} = 1.02F_{\text{Pa}}$$

When the time comes I'll enhance the safety of these assumptions by using no more than 60% of yield as my assembly preload target. Note that "target preload" and "average assembly preload," F_{Pa}, are one and the same.

Now for the low end. As already mentioned, we're interested here in averages, not individual minimums. And once again some engineering judgment will be required to pick values. Theoretically, we could algebraically add the average preload, F_{Pa}, to the average embedment loss of 10% and the average elastic interaction loss of 18%. This would suggest an average residual preload of (1.0 – 0.10 – 0.18) or $0.72F_{\text{Pa}}$ but I think that this is too optimistic. It's extremely important that our joint end up with enough clamping force, worst case, to survive service loads and conditions. Furthermore, experience shows that there are far more things which result in less clamping force than expected rather than in more, as we saw in Chapter 6. So, I'm going to assume that the average residual will be based on an average initial preload of $0.70F_{\text{Pa}}$ less average embedment and elastic interaction losses of 10% and 18%, respectively. Equations 10.4 through 10.6 now give us

$$\text{Min } F_{\text{Pr}} = 0.70F_{\text{Pa}} - 0.1F_{\text{Pa}} - 0.18F_{\text{Pa}} = 0.42F_{\text{Pa}}$$

Incidentally, we obtain F_{Pa} by using our old friend the short-form torque–preload equation.

$$F_{\text{Pa}} = \frac{T}{KD}$$

where

T = torque in in.-lb (N-m)
D = nominal diameter of the fastener (in., m)
K = nut factor (see Table 7.1)

The above gives us the vertical scales for our maximum and minimum assembly joint diagram. Now we need to decide how long to make the baselines of our triangles.

We could use the methods of Chapter 2 to actually compute the deflections of bolt and joint members under the residual preloads, but this is unnecessary, and we're not much interested in these deflections. It's sufficient and convenient, therefore, to draw the horizontal lines defining the deflections of bolt and joint members to any convenient scale. It's only necessary that they be in the proper proportions to each other. To determine these proportions we use the methods of Chapter 5 to estimate the stiffness of bolt, and then obtain the joint-to-bolt stiffness ratio from Figure 5.14 or by computation. If that stiffness ratio is 5:1, then the joint-to-bolt deflection ratio will be the inverse of that or 1:5 and the joint's deflection line will be drawn one-fifth the length of the bolt's deflection line. Let's assume this 5:1 ratio for the present example, and draw our joint diagram accordingly.

The resulting "preloaded and relaxed" joint diagrams are shown, combined, in Figure 10.4. The tensile force in the bolts at this point is called the residual assembly preload (F_{Pr}), and it's assumed to be equal and opposite to the clamping force being exerted by that bolt or those bolts on the joint. We're ignoring complications like weight effect and hole interference in this analysis.

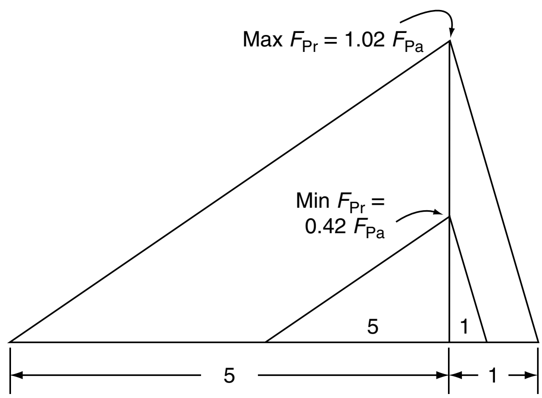


FIGURE 10.4 Joint diagram showing the anticipated maximum load we expect to see, worst case, in individual bolts (Max F_{Pr}) at the end of the assembly process; taking preload scatter, embedment relaxation, and elastic interactions into account; all with reference to a specific example described in the text. The diagram also shows the worst-case, minimum, average, residual preload (Min F_{Pr}) expected in the same example.

Next we're going to modify this diagram to include an external, tensile load on the joint. Before we do that, however, note that the energy stored in the joint and bolt springs, as a result of the assembly process, is equal to the area enclosed by the joint diagram.

10.1.3 Joint Diagram for Simple Tensile Loads

Let's assume that we grab the nut and the head of the bolt with powerful pliers and pull, producing equal and opposite tension forces on each end of the bolt, as in Figure 10.5 [7]. This is obviously an unrealistic method of loading. In fact, we'll never encounter it in practice. But it is the classical way to approach joint behavior—a useful way to further our understanding of the joint and the joint diagram. We'll look at more realistic loading methods later.

Remember, since we tightened the bolt, the joint has been pushing outward on the bolt, keeping the bolt in tension. The new external tension load we have just applied with the pliers

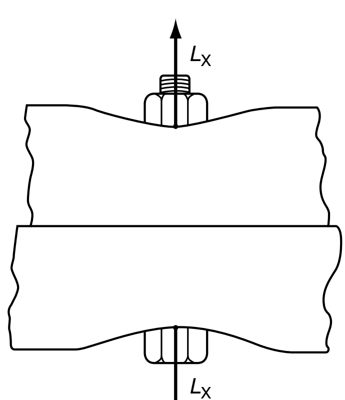


FIGURE 10.5 Let's assume that an external tensile load (L_X) is applied to the nut and to the head of the bolt as shown.

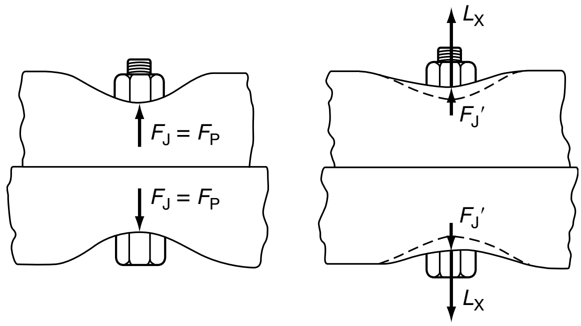


FIGURE 10.6 Forces on the tightened bolt and joint deflection before and after application of external tension load L_X.

helps the joint support the tension in the bolt. In other words, the new external force partially relieves the joint (Figure 10.6).

Since strain (deformation) is proportional to stress (applied force), the partially relieved joint partially returns to its original thickness, moving back down its elastic curve. Simultaneously, the bolt, under the action of the combined joint force and external force, gets longer—following its elastic curve.

Note that the increase in length in the bolt is equal to the increase in thickness (reduction in compression) in the joint. The joint expands to follow the nut as the bolt lengthens. This is an important point; it's a key to understanding joint behavior. Figure 10.7 summarizes the discussion so far.

Now remember: The stiffness of the bolt is only one-fifth that of the joint. This means that, for an equal change in deformation (strain), the change in load (force) in the bolt must be only one-fifth of the change of the load in the joint, as noted in Figure 10.8. The external tension load (L_X) required to produce this change of force and strain in bolt and joint members is equal to the increase in force on the bolt (ΔF_B) plus the reduction in force in the joint (ΔF_J), or

$$L_X = \Delta F_B + \Delta F_J \tag{10.7}$$

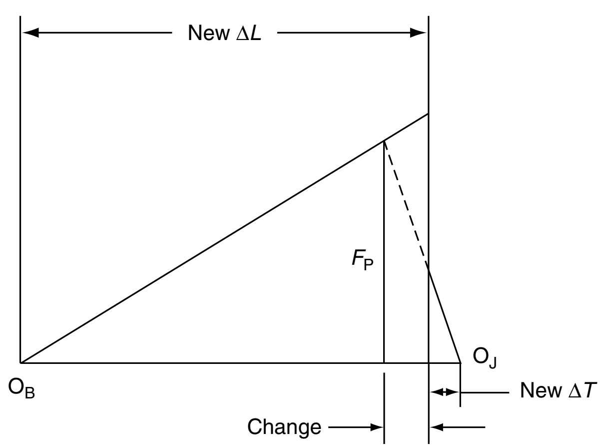


FIGURE 10.7 When an external tension load is applied, the bolt gets longer and joint compression is reduced. The change in deformation in the bolt equals the change in deformation in the joint.

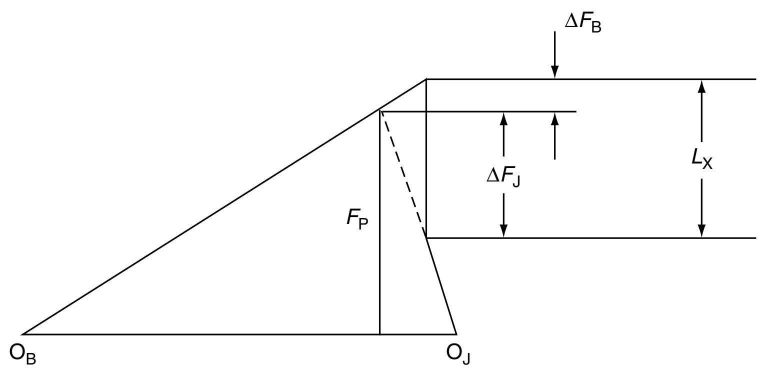


FIGURE 10.8 Because bolt and joint have different stiffnesses, equal changes in deformation mean an unequal change in force. ΔF_B is the increase in bolt force; ΔF_J is the decrease in clamping force in the joint. L_X is the external load.

Many people find this point difficult to accept. After all, we have applied the external load L_X just to the bolt, yet the increase in force seen by the bolt is only a small portion of this external load; the rest of the external load is "absorbed" by the joint. We pull the bolt, but it doesn't feel all the pull. This seems to violate a sacred concept, but it really doesn't. We're not, in fact, just "pulling on a bolt." We're applying a tensile load, through the bolt, to a group of springs. The bolt spring sees—absorbs—some of this load, and the other springs, the joint, sees or absorbs the rest.

It's important to have a clear understanding of all this. The way a bolted joint absorbs external load is another important key to understanding joint behavior. So I think it's worthwhile to look at a crude analogy that may help you see what's happening here.

10.1.4 The Parable of the Red Rolls Royce

You're walking up a steep hill in Big City when you see a funny event. A man has just parked his red Rolls Royce on the hill and has gone around back to get a briefcase full of municipal bonds out of the trunk. He's planning to throw them away in an empty lot conveniently provided for trash disposal. The action starts in Figure 10.9.

Another citizen, carrying a rock, happens by (on his way to collect his unemployment check). He hates guys with red cars, so he seizes a knife that is lying on the street, and, using his rock as a hammer, nails the first citizen's left foot to the road.

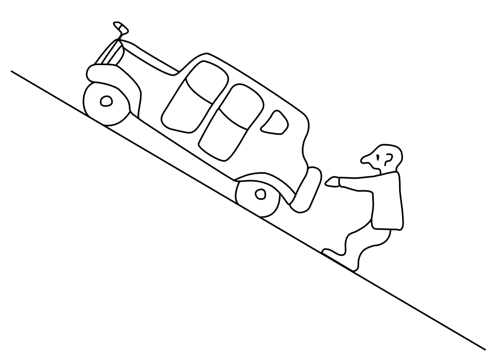


FIGURE 10.9 Scene 1 of the parable of the red Rolls Royce.

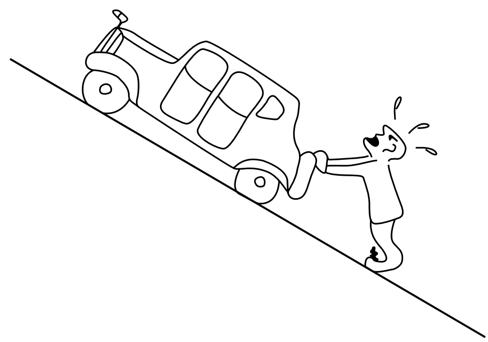


FIGURE 10.10 Scene 2: The system has been preloaded.

Citizen no. 2 then goes up front and releases the brakes on the car. He locks the doors and exits, stage left, with the key. Our victim must now try to keep the car from rolling downhill, rolling over him on its way (Figure 10.10).

Being from the country you don't know how to behave in Big City, so you decide to help this guy (whose name is Mr. Joint). You go to the front of the car and start to pull on it—you're applying an external tension force to the car (Figure 10.11).

But you find that you have a problem. You and Mr. Joint can, indeed, move the car up the hill, but the farther it moves, the less Joint can help you, because his foot is still pinned to the road. In order to pull the car away from him, then, you have to pull hard enough not only to add to the force he was originally applying, but also hard enough to replace the force he can no longer apply as the car moves away from him.

10.1.5 Back to the Joint Diagram—Simple Tensile Load

We have a similar situation in a bolted joint. Any external tension load, no matter how small, will be partially absorbed as new, added force in the bolt (ΔF_B), and partially absorbed in replacing the reduction in the force that the joint originally exerted on the bolt (ΔF_J).

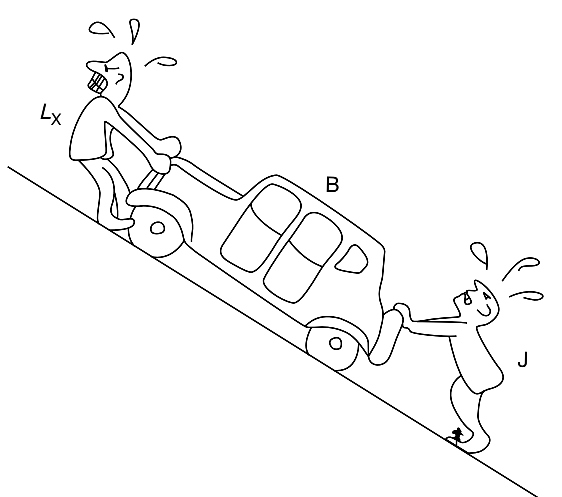


FIGURE 10.11 Scene 3: An external load (L_X) has been applied, helping the joint (J) support the load applied by the bolt (B).

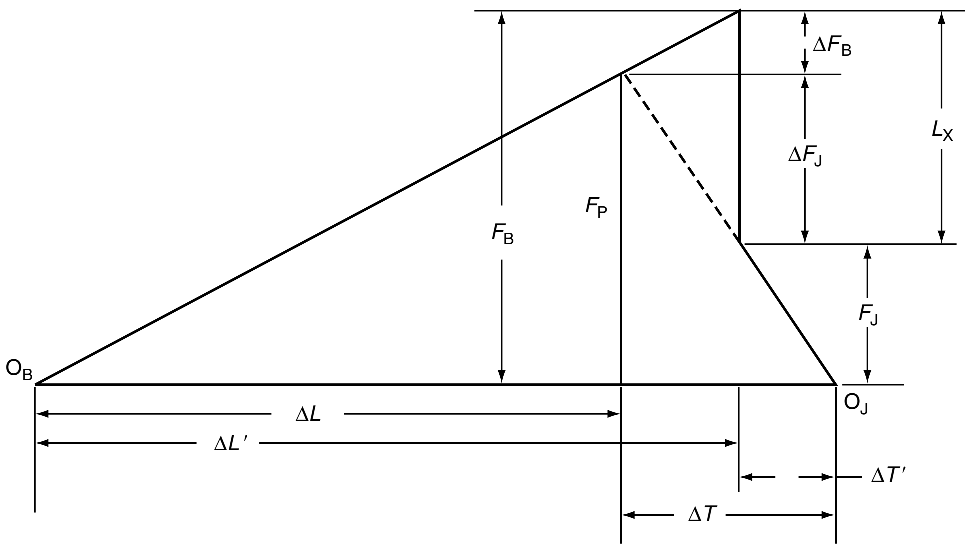


FIGURE 10.12 Summary of the discussion of the joint diagram. F_P is the initial preload; F_B is the present bolt load; F_J is the present joint load; L_X is the external tension load applied to bolt.

The force of the joint on the bolt, plus the external load, equals the new total tension force in the bolt—which is greater than the previous total—but the change in bolt force is less than the external load applied to the bolt. The joint diagram in Figure 10.12 shows all this. We'll look at the mathematics of this diagram in a minute. First, let's consider some of the things the diagram can illustrate for us.

10.2 DETAILS AND VARIATIONS

10.2.1 Changing the Bolt or Joint Stiffness

What happens if we change the stiffness (spring constant) ratio between bolt and joint?

Let's make the bolt a lot stiffer (steeper elastic curve) by using a bolt with a larger diameter. The new joint diagram can be seen in Figure 10.13. Note that the bolt now absorbs a larger percentage of the same external load. It's as if the owner of that red Rolls Royce were

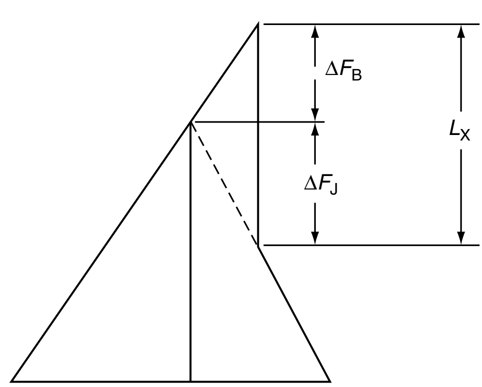


FIGURE 10.13 Joint diagram when the stiffness of the bolt nearly equals that of the joint.

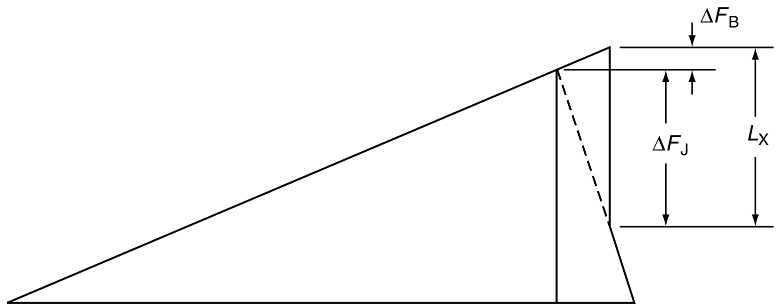


FIGURE 10.14 Joint diagram with softer bolt and stiff joint.

only a child—you work harder as you pull the car away from him, because you're tougher than he is.

If the bolt is made less stiff with respect to the joint, it will see a smaller percentage of a given external load (Figure 10.14).

You might get the same effect if the red Rolls Royce had been owned by a professional football player. You would not have been able to help him as much as you would an ordinary citizen; he's a lot "stiffer" than you are!

The fact that the bolt sees only a part of the external load, and the amount it sees depends on the "stiffness ratio" between bolt and joint, has many implications for joint design, joint failure, measurement of residual preloads, etc. as we'll see. But we're not done yet.

10.2.2 Critical External Load

If we keep adding external load to the original joint, we reach a point where the joint members are fully unloaded, as in Figure 10.15. This is called the critical external load. Note that this critical load is not, in general, equal to the original preload in the bolt, although many people think it is. It's often approximately equal to the preload, however, for several reasons. For example:

1. In many joints the bolt is relatively soft (low spring rate) compared to the joint members. Under these conditions there is a very small difference between the preload in the bolt and the critical external load required to free the joint members.
2. As we'll see later, joints almost always relax after they have first been tightened. Relaxation of 10% or 20% of the initial preload is not at all uncommon. Now, if a

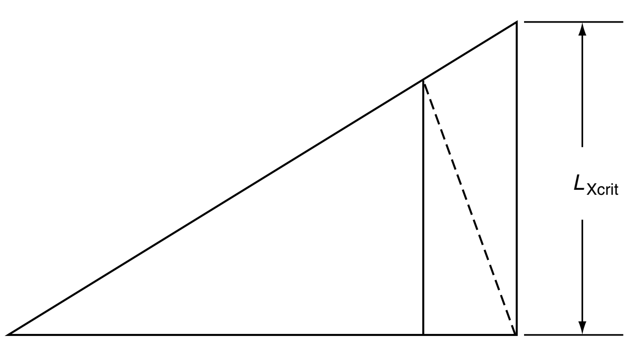


FIGURE 10.15 A critical external load (L_{Xcrit}) fully unloads the joint (but not the bolt).

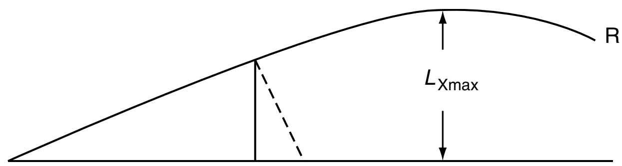


FIGURE 10.16 L_{Xmax} is the maximum external load the bolt can support before it ruptures (at R).

bolt has one-fifth the stiffness of the joint (which is also common), then the critical external load required to free the joint members is 20% greater than the residual preload in the bolt when the external load is applied. Under these conditions the difference between the critical external load and the present preload is just about equal and opposite to the loss in preload that was caused by bolt relaxation. Therefore (by coincidence!), the critical external load equals the original preload before bolt relaxation.

10.2.3 Very Large External Loads

Any additional external load we add beyond the critical point will all be absorbed by the bolt: You're now pulling on the Rolls Royce all by yourself; Mr. Joint has been left behind (Figure 10.16). Note that if the external load gets still larger, the line describing the action of the bolt becomes nonlinear. We must not forget that all of these joint diagram "triangles" are just portions of the elastic–plastic curves for the bolt and joint members.

Although it's usually ignored in joint calculations, there's another "curve" we should be aware of here. The compressive spring rate of many joint members is not a constant, as discussed in Chapter 3. A more accurate joint diagram would show this, as suggested by Figure 10.17. More about this in Chapter 11.

So much for our first look at a conventional joint diagram. We'll derive some related mathematics for it later. Before we get into that, however, I want to introduce you to a different type of joint diagram—one we'll find useful when we study fatigue failure [1].

10.2.4 Another Form of Joint Diagram

This time we're going to plot the tension in the bolt and the compression in the joint on different sides of the horizontal axis (which will represent the external load).

Before we apply any external load, we have equal and opposite preloads in the bolt and in the joint. As we apply external load (see Figure 10.18), the forces change in both bolt and

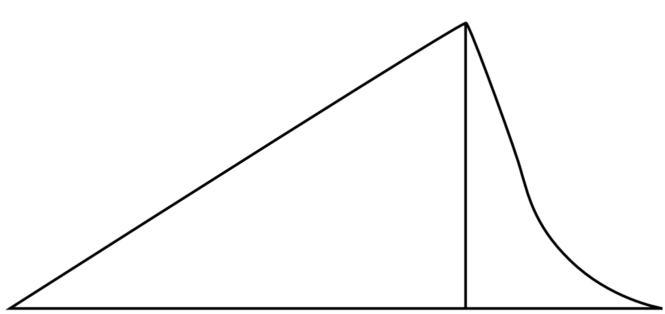


FIGURE 10.17 The spring rate of the joint is often nonlinear for small deflections of the joint.

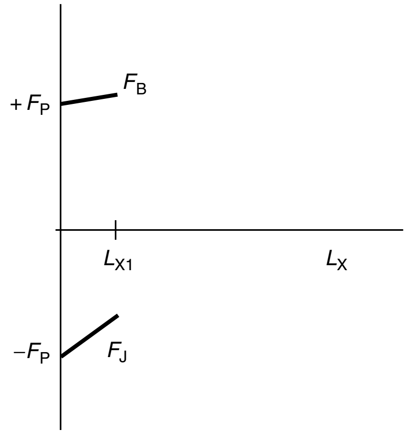


FIGURE 10.18 The initial preloads in bolt and joint are $\pm F_P$. The bolt load (F_B) increases and the joint load (F_J) decreases when we apply an external load L_{X1}.

joint; but the joint, still being five times stiffer than the bolt, sees more change in force for a given change in deformation.

If we apply a critical external load to the assembly, the joint finally becomes completely unloaded. There is no more compressive force in the joint, and the joint members cannot "grow" in thickness any further to follow further elongation of the bolt. You would find, if you increased the external load still further, that there would be an abrupt change in the slope of the bolt curve beyond this point. In fact, you'd find that you were now following the elastic curve for the bolt "alone." The bolt is the only member of the assembly being loaded, now that the joint is fully unloaded and is no longer able to absorb additional load. All of this is shown in Figure 10.19 [4].

We'll find both types of joint diagrams useful later. They are two different ways to look at the same phenomenon.

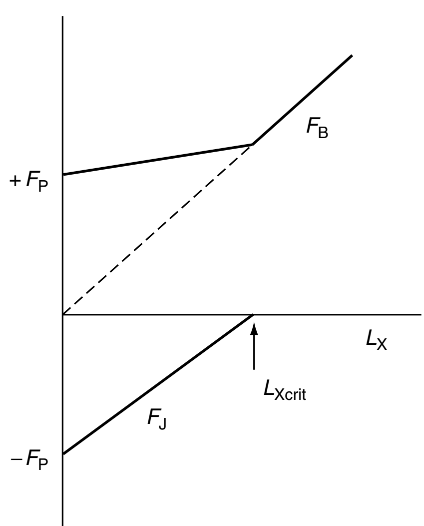


FIGURE 10.19 If you increase the external load until it exceeds the critical load, you'll see a sudden increase in the rate at which the bolt absorbs further external load.

10.3 MATHEMATICS OF THE JOINT

10.3.1 Basic Equations

Let's return now to the first joint diagram—which we'll use more than the second—and write some equations which will help us analyze and design tensile joints [2]. Figures 10.12 and 10.20 show a completed diagram. The central line, labeled F_P could be representing either maximum or minimum residual "relaxed" preload. We'll call it just F_P at this point to avoid two sets of essentially identical equations. So—the diagram illustrates

F_p = initial preload (lb, N)
L_X = external tension load (lb, N)
ΔF_B = change in load in bolt (lb, N)
ΔF_J = change in load in joint (lb, N)
ΔL, $\Delta L'$ = elongation of bolt before and after application of the external load (in., mm)
ΔT, $\Delta T'$ = compression of joint members before and after application of the external load (in., mm)
L_{Xcrit} = external load required to completely unload joint (lb, N) (not shown in the diagram)

The spring constants or stiffness of the bolt and joint can be defined as follows:
For the bolt:

$$K_B = \frac{F_P}{\Delta L} \tag{10.8}$$

For the joint:

$$K_J = \frac{F_P}{\Delta T} \tag{10.9}$$

By using trigonometry, and by recognizing similar triangles where they occur, we can now derive the following useful expressions:

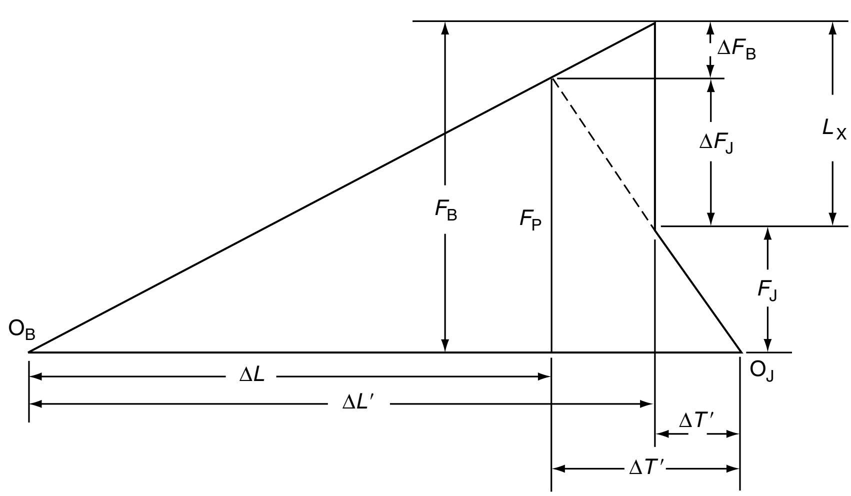


FIGURE 10.20 Completed joint diagram.

$$\Delta F_B = \left(\frac{K_B}{K_B + K_J}\right) L_X \tag{10.10}$$

(until joint separation, after which $\Delta F_B = \Delta L_X$), and

$$L_{Xcrit} = F_P\left(1 + \frac{K_B}{K_J}\right) \tag{10.11}$$

The ratio $K_B/(K_B + K_J)$ turns out to be so useful that we give it a name and a symbol of its own. Following VDI practice [6] we'll call it the "load factor" (Φ_K). It defines that portion of the external tensile load which is seen by the bolt, or

$$\Phi_K = \frac{\Delta F_B}{L_X} = \frac{K_B}{K_B + K_J} \tag{10.12}$$

and

$$\Delta F_B = \Phi_K L_X \tag{10.13}$$

where
L_X = the external, tensile load (lb, N)
K_B and K_J are expressed in lb/in. or N/mm

Note that the joint absorbs the rest of the external load, or

$$\Delta F_J = L_X - \Delta F_B = (1 - \Phi_K)L_X \tag{10.14}$$

We're now in a position to write more complete equations for the two things we're most interested in: the maximum load seen by the bolts and the minimum clamping force we can expect to see on the joint. To do this we must once again distinguish between maximum and minimum residual preloads after relaxation. We're going to write the "room temperature" versions of these equations at this point. In the next chapter we'll extend the equations to include the thermally induced effects of differential expansion.

Maximum anticipated bolt load in service (with reference to Equation 10.3)

$$\begin{aligned}
\text{Max } F_B &= \text{Max } F_{Pr} + \Delta F_B \\
\text{Max } F_B &= \text{Max } F_{Pr} + \Phi_K L_X \\
\text{Max } F_B &= (1 + s)F_{Pa} - \Delta F_m - \Delta F_{EI} + \Phi_K L_X
\end{aligned} \tag{10.15}$$

Minimum anticipated, per bolt, clamping force in service (with reference to Equation 10.4)

$$\begin{aligned}
\text{Min } F_J &= \text{Min } F_{Pr} - \Delta F_J \\
\text{Min } F_J &= \text{Min } F_{Pr} - (1 - \Phi_K)L_X \\
\text{Min } F_J &= (1 - s)F_{Pa} - \Delta F_m - \Delta F_{EI} - (1 - \Phi_K)L_X
\end{aligned} \tag{10.16}$$

As mentioned earlier, we'll usually be more interested in the total clamping force on the joint than in the clamping force created by an individual bolt. To get the total, since we're dealing with averages here, we merely multiply the per-bolt force by the number of bolts (N) or:

$$\text{Total Min } F_J = N \times \text{per-bolt Min } F \tag{10.17}$$

We'll extend these equations further in Chapter 11 to include the effects of differential expansion but Equations 10.14 through 10.16 will be all we'll need for many (perhaps most) applications. Incidentally, the load factor, Φ_K, is sometimes called the "force ratio" or "the joint stiffness ratio." The latter could be confused with the simpler joint-to-bolt stiffness ratio, so I'll call Φ_K the load factor to avoid confusion.

10.3.2 Continuing the Example

In the joint diagram example we started in Section 10.1.2, we computed the maximum and minimum residual assembly preloads; they were $1.02F_{Pa}$ and $0.42F_{Pa}$, respectively, where F_{Pa} was the average, initial, assembly preload. Now let's continue our example by adding the effects of an external load. Let's assume that the external load (L_X) will equal $0.25F_{Pa}$.

First, we must compute a load factor. Remember that the joint-to-bolt stiffness ratio in our example was 5:1. Therefore, from Equation 10.12:

$$\Phi_K = \frac{1}{1+5} = 0.17$$

We can now estimate the changes which will occur in bolt tension (F_B) and the clamping force on the joint (F_J) when L_X is applied. From Equation 10.13:

$\Delta F_B = 0.17L_X = 0.17(0.25F_{Pa}) = 0.043F_{Pa}$ and, from Equation 10.14, $\Delta F_J = (1 - 0.17)$ $L_X = (0 - 0.17)(0.25F_{Pa}) = 0.21F_{Pa}$.

Now we can compute the maximum load to be seen by the bolt, as a result of the assembly process plus the external load (Max F_B) and the minimum clamping force we can expect to achieve on the joint as a result of the same factors (Min F_J). From Equation 10.15

$$\text{Max } F_B = \text{Max } F_{Pr} + \Delta F_B$$
$$\text{Max } F_B = 1.02F_{Pa} + 0.043F_{Pa} = 1.06F_{Pa}$$

where, F_{Pr} is the residual preload after embedment and elastic interaction loss. Next, from Equation 10.16:

$$\text{Min } F_J = \text{Min } F_{Pr} - \Delta F_J$$
$$\text{Min } F_J = 0.42F_{Pa} - 0.21F_{Pa} = 0.21F_{Pa}$$

The results are plotted in Figure 10.21. Note that these results show that there is an approximately 5:1 ratio between the maximum force which some of the bolts, worst case, must be able to support; and the minimum clamping force we can expect, again worst case. These results are based, of course, on the assumptions I made in Section 10.1.2 and may not be entirely valid, but those assumptions were reasonable and the results are certainly not out of line. The results show us why poor control of the assembly process leads to "overdesign" of the joint. In order to get that minimum clamping force we must use bolts and joint members that are five times more massive than would be necessary if we could control residual assembly preloads more accurately, perhaps by using something other than torque control to reduce the scatter in initial preload or by using ultrasonics or an equivalent to allow us to compensate for elastic interactions. Incidentally, the energy contained within an externally loaded joint, in service, is equal to the area enclosed by the joint diagram of Figure 10.20. If you have computed both deflections and forces, you can compute the energy stored within the joint from

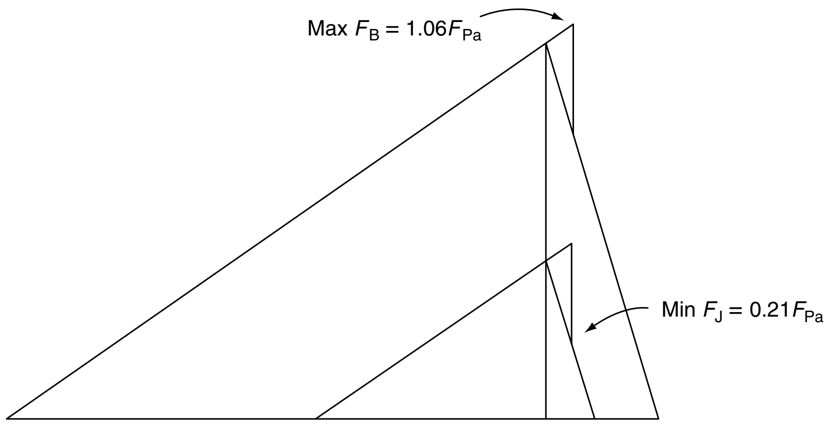


FIGURE 10.21 Continuing the example first illustrated in Figure 10.4, this time to include the effects of an external tensile load on the joint. The diagram shows the results of calculations to estimate the maximum, worst-case, load expected in individual bolts (Max F_B) and the worst-case, minimum, average, per-bolt clamping force on the joint (Min F_J). See text for details.

$$E_J = 0.5 \times \Delta T' \times F_J$$

and the energy stored in the bolt from

$$E_B = 0.5 \times \Delta L' \times F_B$$

These equations are of interest only to those having a morbid interest in bolted joints—like me!—and we won't find much if any use for them. So I'm not going to give them numbers.

So much for simple joint diagrams. They're very basic, and very useful, even though they describe the behavior of the joint under a very uncommon type of loading—a tensile load applied between the bolt head and nut (as with our pliers!) or at least applied to the bolt between the plane of the upper surface of the joint and the plane of its lower surface.

As we're about to see, our equations must be modified and different diagrams must be constructed if the loads are applied at some point other than head-to-nut; but the resulting bolt loads and clamping forces will often be very similar to those we've just computed. Even though head-to-nut loading is almost never encountered in practice, the joint diagram of Figure 10.12 and its related equations are usually used in design work because they give a worst-case result for the maximum loads—and load excursions—to be seen by individual bolts; and they give reasonable estimates of the minimum clamping force on the joint. They do not, however, give worst-case estimates for the clamping force. If that is the critical factor in our application, then we want to refine our equations and joint diagram to give us a more realistic view of the way tension loads are actually applied to a joint. We do this by introducing the concept of loading planes.

10.4 LOADING PLANES

We have taken a detailed look at the behavior of a bolt in a joint when tension forces are applied to both ends of the bolt. Now we're going to look at the behavior of the bolt and joint when tension loads are applied at other points.

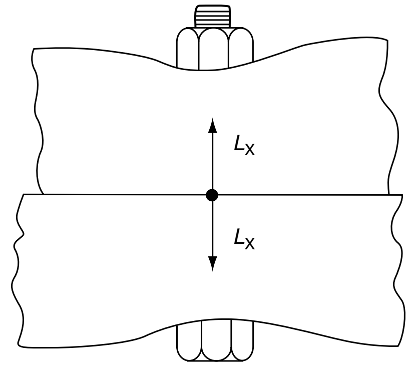


FIGURE 10.22 In this example the external tension load is applied at the joint interface, as shown here.

Note that loads are seldom, if ever, applied to a single "point" in a bolted joint. Loads are created by pressure, weight, shock, inertia, etc. and are transferred to the joint by the connected members. An accurate description of where that load is applied would require a detailed stress analysis (e.g., a finite-element analysis). The people who developed the classical joint diagram, however, have found a simpler way to "place" the load. They define hypothetical "loading planes," parallel to the joint interface, and located somewhere between the outer and contact surfaces of each joint member. They then assume that the tensile load on the joint is applied to these loading planes. Joint material between the loading planes will then be (theoretically) unloaded by a tensile load; joint material out-board of the planes will be "trapped" between plane of application of load and the head (or nut) of the fastener.

In our first example (Section 10.1.3), these planes coincided with the upper and lower surfaces of the flange or joint. For our next example, loading planes will coincide with the interface between upper and lower joint members, as suggested in Figure 10.22.

Note that the loading plane is pure fiction. It's a "bugger factor" used to correct a joint diagram analysis, to make the analysis agree, for example, with experimental results (which might show how an external tensile load actually affects the tension in a preloaded bolt). In any event, the loading plane is a useful concept. Here's how it works.

10.4.1 Tension Applied to Interface of Joint Members

Remember, in our first analysis:

Bolt was treated as a tension spring.
Upper and lower flange members were treated as compression springs.
Tensile loads applied to each end of the bolt stretched (loaded) the bolt and partially relieved (unloaded) the joint.

All of this was analyzed in a joint diagram which showed how one spring was loaded and the other unloaded by the external load.

If the same tensile force, however, were applied to the interface between the upper and lower joint members, then both the bolt tension spring and the joint compression spring would be loaded by the external load. What does the joint diagram look like in this situation?

The two flange pieces are originally exerting equal and opposite forces on each other—forces equal to the preload in the bolt. As we start to apply a small external load at the interface, we partially replace the forces that the two joint members are exerting on each other. We're relieving these flange-on-flange forces rather than adding to them, to start with.

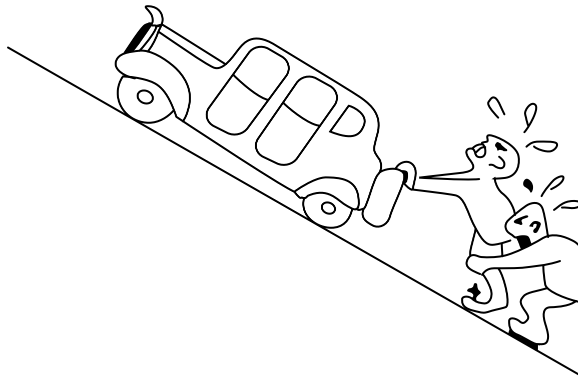


FIGURE 10.23 We're going to relieve Mr. Joint by picking him up and pushing through him.

Going back to the parable of the red Rolls Royce, this time you're trying to help Mr. Joint by picking him up, reducing the force he is exerting on the road, without changing the amount of force he is exerting on the car, as shown in Figure 10.23.

In the joint, of course, this means that the external load reduces the flange-on-flange force without increasing the total force in either the flange members or the bolt—yet. The joint diagram for this situation is shown in Figure 10.24 [2]. Note that I have chosen to draw both elastic curves (bolt and joint) on the same side of the common vertical axis (the axis that represents original preload or F_P). I do this because both springs are loaded by the external force.

When the external load equals the original preload in the bolt, it will have replaced all of the force that each joint member was exerting on the other. In the red Rolls Royce parable, you've just lifted Mr. Joint completely off the road, but he's still exerting the same amount of force on the car. Neither bolt deformation nor joint deformation has changed to this point.

Increasing the external load beyond this point will now add to the original deformation of both the bolt and the joint members. The bolt gets longer and the joint compresses more. The joint diagram merely "gets larger." (In Figure 10.25, the dashed lines represent the original joint diagram; the solid lines represent the new joint diagram.) Note that at all times, both

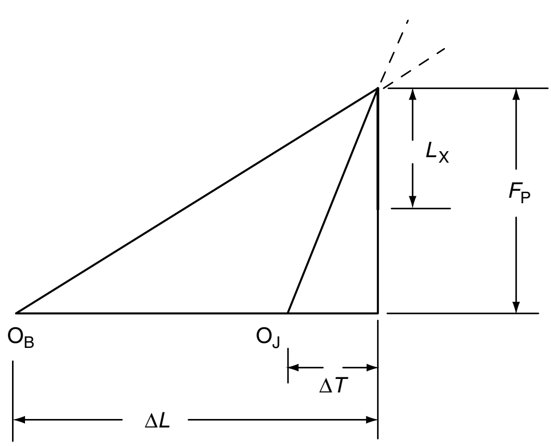


FIGURE 10.24 Joint diagram when an external tension load is applied at the joint interface. ΔL is the elongation of bolt (in., mm); ΔT is the compression of joint (in., mm); F_P is the original preload (lb, N).

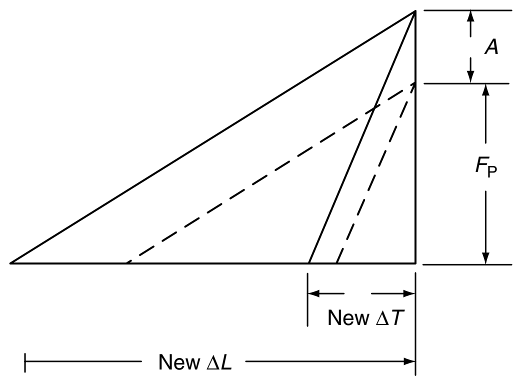


FIGURE 10.25 The external load applied to the joint interface has exceeded the initial load by amount A.

bolt and joint see the same total load, change in load, etc. In the Rolls Royce parable, you are now exerting more force on the car than Mr. Joint was, but you're doing it through Mr. Joint.

As one final comment, the spring rate of the joint members will still be nonlinear at small deflections, so there would be a curve in the bottom of that line in a more accurate joint diagram.

10.4.2 Mathematics of a Tension Load at the Interface

The mathematics of a tension load at the interface is very simple, and can be determined by inspection of the joint diagram.

The change in bolt force is

$$\Delta F_B = 0$$

until the external load exceeds the preload (F_P), after which

$$\Delta F_B = L_X - F_P \tag{10.18}$$

or

further $\Delta F_B =$ further ΔL_X

The critical external load required to cause joint separation is

$$L_{Xcrit} = F_P \tag{10.19}$$

Note that this is true regardless of the spring constants, or spring constant ratios, of the bolt and joint members.

One of the keys of our first joint diagram (considered in Section 10.1) was that the change in elongation of the bolt under an external load equaled the change in compression of the joint; but the changes in force in each were unequal. Note that in the present case, where the tension load is applied at the interface, the deflections are not equal, but the force in the bolt is always equal to the force in the flange. So this second example could be called the inverse of the first.

10.4.3 Significance of the Loading Planes

Now that we have examined two cases, we can start to see the significance of the loading plane. The mathematics above tells us that there will be no change in the force seen by the bolt when tension loads are applied at the interface until the external load exceeds the original preload in the bolt. As we'll see when we look at cyclic loads and fatigue, it would be very desirable to be able to apply external loads to a joint in such a way.

On the other hand, interface loading gives us a critical external load (the load required to cause joint separation) that is equal to the preload. This is less than the load required for separation when the tension loads are applied at joint surfaces. The load capacity of the interface joint, therefore, is less than the load capacity of the original joint.

Maximum bolt load, working change in bolt load, and critical external load are important design factors. They are different for each possible pair of loading planes, hence the importance of loading planes to our calculations.

Note that this situation describes a best-case situation for the bolts and a worst-case situation for the joint. Designers often, therefore, use this analysis and the analysis of Section 10.3.1 to analyze a design. The combination gives them a look at worst-case assumptions for both bolts and joint. If their designs can be expected to survive service loads and conditions under either assumption, it's unnecessary for them to agonize over the ''actual'' location of the loading planes, which is usually somewhere in between the two assumptions we've made so far. There are times, however, when we want a more realistic analysis. To do this we must assume that the loading planes are located neither at the outer surfaces of the joint members, nor at their interface, but somewhere within the joint members. Let's see how we do that.

10.4.4 Loading Planes within the Joint Members

Now let's consider a more complex situation: Let's assume that the loading planes are at some arbitrary point within the joint members, as suggested in Figure 10.26.

Upon reflection we will see that

- The bolt will still stretch when we apply the external load.
- The inner portions of each joint member—those portions nearest the interface—will unload when the external load is applied.
- The outer portions of each joint member, however, will be placed under additional compression load by the external load.

We could diagram this situation by placing the elastic curves for all the springs that are loaded (by the external load) on the left of the vertical F_P axis, and all of those that are unloaded on the right. This isn't particularly helpful, however. It's simpler, and more meaningful, to realize that the head-to-nut and joint interface diagrams developed earlier

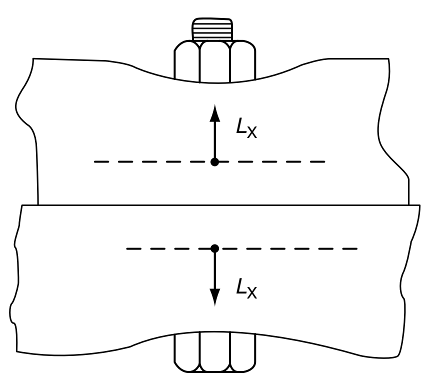


FIGURE 10.26 External loads applied at some point within the joint members.

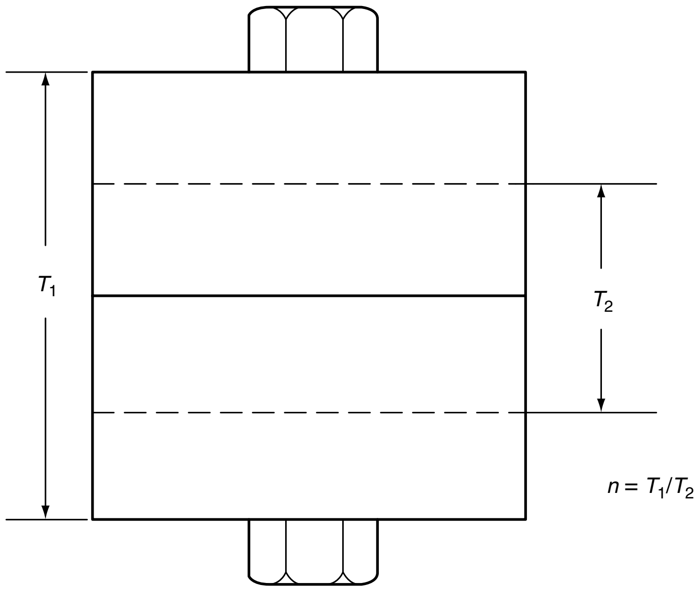


FIGURE 10.27 The "loading plane factor" discussed in the text is defined as the ratio between that portion of the joint which is unloaded by an external tensile load, 7%, and the total thickness of the joint, T_1.

represent limiting conditions. In-between loading planes lead to in-between values for all factors of interest—bolt loads, joint loads, critical external load, etc. We can compute values for these things by using a "loading plane factor" (n) which defines the ratio between the thickness of joint material being unloaded by the external force and the total thickness, or, with reference to Figure 10.27

$$n = \frac{T_2}{T_1} \tag{10.20}$$

We can use n to write new equations for the load factor, the change in tension seen by a bolt, and the change in the per-bolt clamping force, as follows:

$$\Phi_{Kn} = n\left(\frac{K_B}{K_B + K_J}\right) = n\Phi_K \tag{10.21}$$

$$\Delta F_B = n\left(\frac{K_B}{K_B + K_J}\right)L_X = \Phi_{Kn}L_X \tag{10.22}$$

$$\Delta F_J = (1 - \Phi_{Kn})L_X \tag{10.23}$$

The factor Φ_{Kn} can be substituted for Φ_K in Equations 10.15 and 10.16, to include the effects of loading planes. This gives us the following:

$$\text{Max } F_B = (1 + s)F_{Pa} - \Delta F_m - \Delta F_{EI} + \Phi_{Kn}L_X \tag{10.24}$$

$$\text{Min } F_J = (1 - s)F_{Pa} - \Delta F_m - \Delta F_{EI} - (1 - \Phi_{Kn})L_X \tag{10.25}$$

As before, we can express the embedment and elastic interaction losses as fractions of the average, initial preload.

$$\Delta F_{\mathrm{m}} = e_{\mathrm{m}} F_{\mathrm{Pa}} \tag{10.5}$$

$$\Delta F_{\mathrm{EI}} = e_{\mathrm{EI}} F_{\mathrm{Pa}} \tag{10.6}$$

And, once again, we can estimate the total clamping force on the joint from:

$$\text{Total Min } F_{\mathrm{J}} = N \times \text{per-bolt Min } F_{\mathrm{J}} \tag{10.17}$$

where $N =$ the number of bolts in the joint.

These five equations are very basic and very important. We'll use them—or modifications of them—when we learn how to design bolted joints. They allow us to estimate the maximum loads to be seen, worst case, by individual bolts in the joint, and the worst-case minimum clamping force created by the group of bolts on the joint; all as a function of

- Initial average preload created during assembly (F_{Pa})
- Anticipated scatter in preload during assembly (s)
- Loss of preload caused by embedment relaxation (e_{m})
- Further, average loss in preload caused by elastic interactions (e_{EI})
- External load (L_{X})
- Stiffness of bolts and joint members (Φ_{K})
- The way the load is applied to the joint (n)

We'll add other factors later: differential expansion in Chapter 11 for example, but the five equations above are all we'll need for many—perhaps most is more accurate—applications.

Note that although we've used the original, actual bolt stiffnesses K_{J} and K_{B} to compute Φ_{KB}, the effective stiffness ratio between bolt and joint seems to have changed. As Figure 10.28 suggests, not only the loads on these two parts but also the deflections of these two parts have

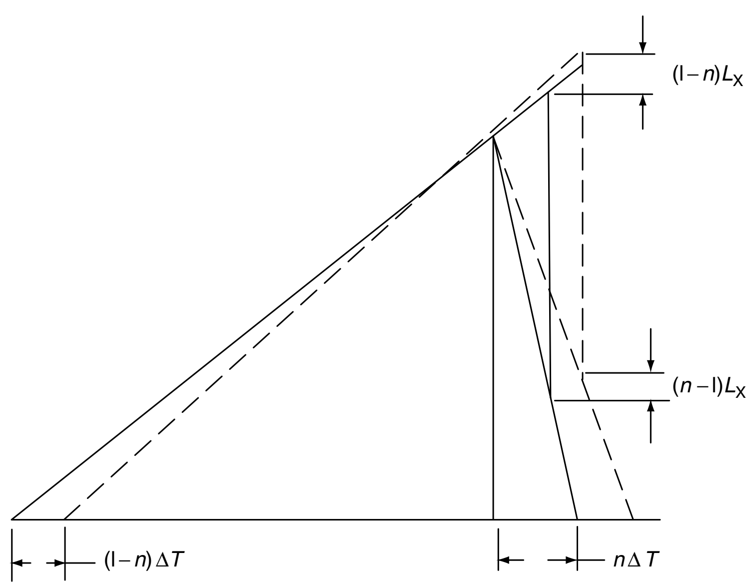


FIGURE 10.28 The dashed lines reflect the head-to-nut loading condition shown in Figure 10.12 or 10.20. The solid lines show how the diagram is modified to accommodate loading planes within the joint members. The location of these planes is defined by the loading plane factor, n, explained in Figure 10.27.

changed. The diagram tells us that the bolt is deflecting more than it did under our original assumption of head-to-nut loading; even though the force on the bolt is less than it was then. Is this reasonable? Yes, it is. The diagram reveals the fact that the bolt spring, as one of this system of springs, will deflect more for a given tension load if that load is applied within the joint than it would if the load were applied between bolt head and nut.

One reason for avoiding calculations based on the true loading plane location is that this location is difficult to find. Remember that it's not really a true plane. A finite-element analysis or the like would be required, I suspect, to determine what portion of the flange members is unloaded and what portion sees added load when the external load, is applied.

It's not obvious where the loading planes would be, either, as suggested by Figure 10.29 [2]. The location is determined by how much of the joint is "clamped" and how much is "clamping." Only an experiment can tell us for sure; though a finite-element analysis might come close. Those who can't afford either are advised to assume that $n = 0.5$. In most applications this will get you closer to the truth than will the assumption of head-to-nut loading ($n = 1.0$) or interface loading ($n = 0$).

Note that an inside-the-joint loading plane makes things easier for the bolt: it sees less increase in tension for a given external load. But it makes things worse for the joint: more loss in clamp force. All of which is supposedly our "most accurate" look at how a joint

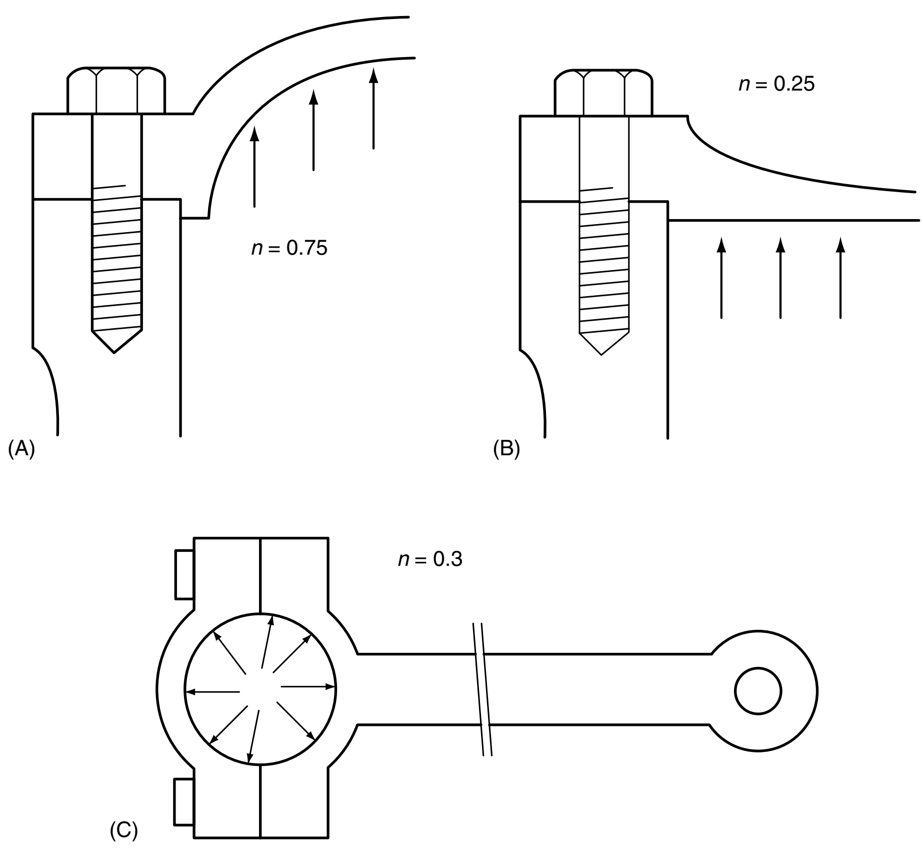


FIGURE 10.29 Loading plane factors from Ref. [2]. It's not obvious why the factors are so different for the two pressure vessel joints shown here, nor is it obvious why the factor for the engine crankshaft joint is nearly the same as that for the drastically different pressure vessel having the convex coverplate. This illustrates the difficulty of trying to guess the location of the loading planes within a new joint. (From Meyer, G. and D. Strelow, *Assembly Eng*., 28–32, 1972.)

responds to a tensile load. But it may not be our safest look, because it's based on so many assumptions.

Before moving on, a word about stored energy. The energy stored in the externally loaded joint is still equal to the area enclosed by the joint diagram, even though that has undergone the changes shown in Figure 10.28. In fact, the original and final joint diagrams superimposed in Figure 10.28 enclose the same amount of area. This must be true because each diagram is based on the same residual preloads (same amount of energy retained at the end of the assembly process) increased by applying the same external loads to the joints (same amount of additional, mechanical energy added to that stored during assembly). The only difference is in the point of application of the external loads, and that shouldn't—and doesn't—affect the resulting contained energy.

10.4.5 Modifying Our Example to Include the Effects of Internal Loading Planes

Before leaving the subject of loading planes, let's see what effect they have on the results we obtained for our example: Sections 10.1.2 and 10.3.2. Let's assume that $n = 0.5$. The external load, L_X, is still equal to $0.25F_{Pa}$. From Equation 12.21

$$\Phi_{Kn} = 0.5\left(\frac{1}{1+5}\right) = 0.08$$

From Equation 10.22

$$\Delta F_B = 0.08(0.25F_{Pa}) = 0.02F_{Pa}$$

From Equation 10.23

$$\Delta F_J = (1 - 0.08)(0.25F_{Pa}) = 0.23F_{Pa}$$

From Equations 10.15 and 10.24

$$\text{Max } F_B = \text{Max } F_{Pr} + \Delta F_B = 1.02F_{Pa} + 0.02F_{Pa} = 1.04F_{Pa}$$

From Equations 10.16 and 10.25

$$\text{Min } F_J = \text{Min } F_{Pr} - \Delta F_J = 0.42F_{Pa} - 0.23F_{Pa} = 0.19F_{Pa}$$

These figures are very close to the Max $F_B = 1.06F_{Pa}$ and Min $F_J = 0.21F_{Pa}$ we obtained in Section 10.3.2 when we were ignoring the loading plane factor; so the differences aren't impressive—and probably aren't significant—in this example. If this weren't a learning experience, we'd have wasted our time to introduce the loading plane factor. But that won't always be the case.

The final joint diagram for our example is shown in Figure 10.30.

10.5 DYNAMIC LOADS ON TENSION JOINTS

We have seen what happens when we apply an external load to a joint loaded in tension. The discussion implied that such loads were static. External loads, however, can also fluctuate. How does this affect the joint diagram?

First, let's consider a fluctuating load that is applied at the joint surface (bolt head to nut). The load we apply will have a maximum value a little less than the critical load.

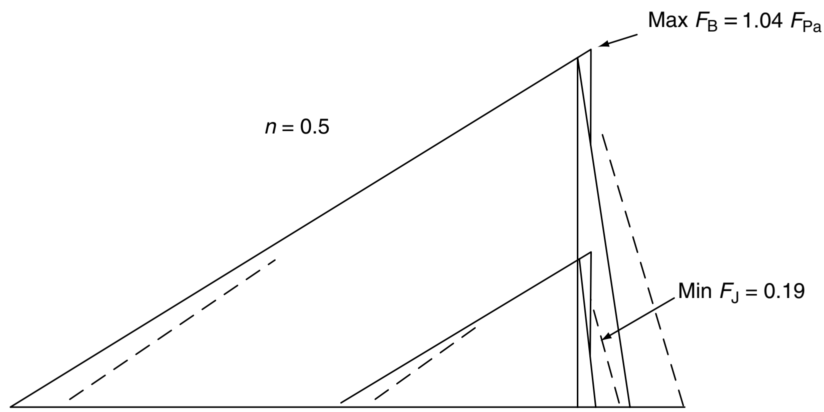


FIGURE 10.30 Final joint diagram for the example joint previously illustrated in Figures 10.4 and 10.21. This time a loading plane factor of 0.5 has been included in the calculations. As you can see by comparison with Figure 10.21, this has made very little difference in the final estimates of maximum individual bolt load (Max F_B) or minimum average, per-bolt clamp force on the joint (Min F_J).

Remember, with a static external load, the bolt sees a portion of the load and the joint sees the rest—the ratio depending on the relative stiffness of bolt and joint, at least until the external load exceeds the critical load. This fluctuating external load will cause the total tension load in the bolt and the load in the joint to fluctuate also (Figure 10.31).

Note that, in all of the above, the maximum external load is almost equal to, but slightly less than, the critical external load required for joint separation. What happens if the external load exceeds the critical value? Let's increase the external load by 50%.

A careful layout of this diagram (Figure 10.32), or suitable calculations, shows that (for the stiffness ratio we have assumed for this bolt and joint) the maximum load on the bolt will increase 25% when we raise the external load approximately 50% above the critical value. However, the magnitude of the excursion, or change, in bolt tension, (ΔF_B) increases by more than 250%. This increase in maximum bolt load and—much worse—the increase in the magnitude of the fluctuation of bolt load greatly increase the chance of fatigue failure of the bolt.

Notice, too, what happens to the clamping load on the joint under these circumstances. The joint is fully unloaded about one-third of the time, and so will be highly susceptible to leaks, slip, fatigue, and fretting problems.

In the discussion above we assumed that the bolt was loaded between head and nut. If the tension load is applied, instead, at the joint interface, the external load will produce no change in bolt tension until the external load exceeds the critical value required for joint separation. This is an excellent situation if fatigue is of concern, but zero excursion is difficult to arrange in practice.

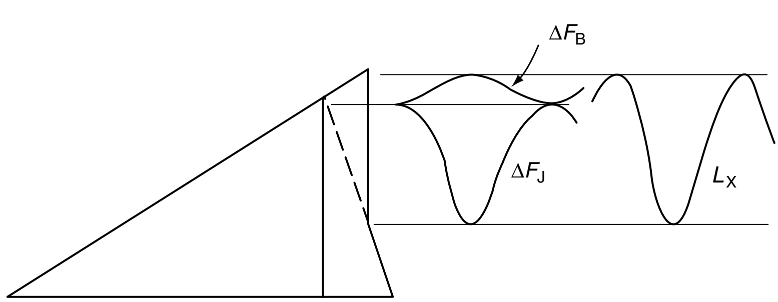


FIGURE 10.31 Bolt and joint loads when the external load fluctuates.

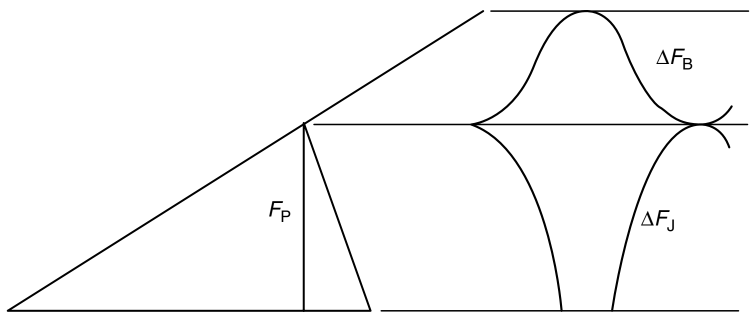


FIGURE 10.32 Joint diagram when the fluctuating external load exceeds the critical load.

10.6 THE JOINT UNDER A COMPRESSIVE LOAD

Before we leave the subject of joint diagrams let's look briefly at the response of a joint to a compressive rather than tensile load. Such loads would appear, at first thought, to be less common than tensile loads, but I believe we encounter them every time we assemble a multi-bolt joint. Tightening bolts 3 and 4 in Figure 6.22, for example, will apply a compressive load to the assembly, which previously consisted of bolts 1 and 2 and the joint members.

The joint between the piston rod and piston in Figure 10.33 is another example. This joint will be subjected to an alternating tensile-compressive load if the hydraulic cylinder is used in a push–pull application [3].

The joint diagram for a compressive load is shown in Figure 10.34. It differs only slightly from the diagram of Figure 10.12. This time the external load increases the compression of the joint, by ΔT, and simultaneously allows the bolts to relax by the same amount, ΔL. The new compressive force on the joint is F_J; the reduced tension in the bolts is F_B, and is less than the original preload of F_P. And all of this has been caused by exerting a compressive load of L_{XC} on the joint (in a direction along the axis of the bolts).

10.7 A WARNING

We now seem to have a good understanding of the behavior of a bolted joint under tensile loads. The theories we've discussed are widely used to design and analyze joints. As "experts,"

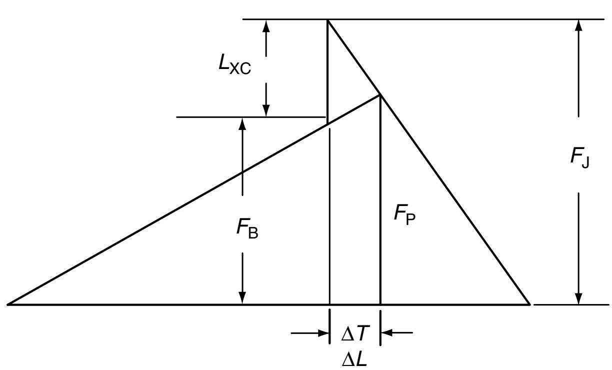


FIGURE 10.33 If a hydraulic cylinder is used in a push–pull mode, the joint between piston and rod will be subjected to alternating tensile and compressive loads. The tensile loads can be analyzed with the joint diagram of Figures 10.12, 10.21, or 10.30. The diagram of Figure 10.34 can be used for the compressive loads.

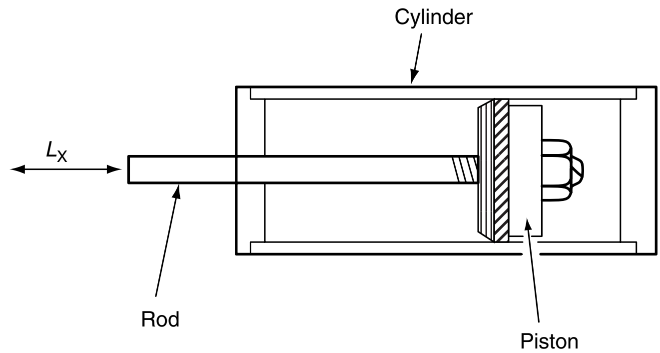


FIGURE 10.34 Joint diagram for a joint loaded in axial compression. The compressive load is L_{XC}; the resulting bolt tension and joint load are F_B and F_J, respectively. Bolt and joint deflections are ΔL and ΔT.

though, we should be aware of the fact that we've taken a very simplistic approach. We've assumed that joint behavior is fully elastic and linear. In fact, it is often neither of these things and, as a result, will not behave as we have predicted. We'll often have to assume linear behavior to estimate bolt loads or the like because the true behavior is so complex as to defy current theories, except in a few special cases. We need a general understanding of the true behavior, however, and will get that in Chapter 11.

Also note that we've been analyzing "what will we get?" if we apply an external tensile load to a preload bolted joint. We have not yet addressed the equally important question, "what results do we want?" in the way of clamping force and in maximum bolt tension. For example, what's the least clamping force we can accept? We'll start to get some answers to these questions when we look at failure modes in Chapters 13 through 16 and will try to finalize our answers when we deal with joint design in Chapters 18 and 19.

EXERCISES AND PROBLEMS

1. Sketch a joint diagram for a joint in the preloaded condition, before an external load is applied to the joint. Assume that $K_J = 5\ K_B$.
2. We apply 43 ft-lbs of torque to a ⅜-28 × 2 J429, GR 8 bolt. We believe that the nut factor will be 0.2 with a scatter of only ±17%. Compute the estimated, initial, maximum and minimum preload created in the bolt.
3. For the situation described in problem 2 above we expect elastic interaction loss of 6% and embedment loss of 4%. Compute the revised maximum and minimum preloads.
4. Sketch the revised, preloaded joint diagram.
5. Compute the load factor Φ_K for this joint.
6. We now apply an external tensile load of 3,000 lbs to the joint. By how much will the tension in the bolt increase?
7. By how much will the clamping force on the joint be reduced?
8. Sketch the new joint diagram, assuming maximum initial preload.
9. Compute the estimated maximum bolt tension and minimum clamping force under the 3,000 lbs external load.
10. Does the addition of the external load endanger the bolt?
11. Sketch an alternate form for joint diagram for this situation.
12. Define the "critical external load." Approximately what would you expect it to be for this joint?

REFERENCES

1. Radzinovsky, E.I., Bolt design for repeated loading, *Machine Design*, pp. 135ff, November 1952.
2. Meyer, G. and D. Strelow, Simple diagrams aid in analyzing force in bolted joints, *Assembly Engineering*, pp. 28–33, January 1972.
3. Horsch, R., Solve complicated force problems with simple diagrams, *Assembly Engineering*, pp. 22–24, December 1972.
4. Milestone, W.D., Fatigue design considerations in bolted joints, Paper presented at University of Wisconsin (Madison) Seminar, Using Threaded Fasteners, April 1979.
5. Webjorn, J., The bolted joint, a series of problems, Dissertation No. 130, Institute of Technology, Department of Mechanical Engineering, Linkoping, Sweden, 1985.
6. Bibel, G.D. and D.L. Goddard, Preload variation of torqued fasteners: a comparison of frictional and elastic interaction effects, *Fastener Technology International*, February 17, 1994.
7. Junker, G.H., Principle of calculation of high duty bolted joints—interpretation of VDI 2230 Directive, published by SPS Technologies, Jenkintown, PA (no date).

11 Behavior of the Joint Loaded in Tension: A Closer Look

In Chapter 10 we used joint diagrams to estimate the effects which an external load (L_X) would have on bolt tension (F_B) and on the force with which the joint members are clamped together (F_J). See Figure 10.20 or the more complex Figure 10.28 to refresh your memory if necessary.

In that discussion we assumed that the behavior of the nuts, bolts, and joint members under load would be linear and fully elastic. In practice that will rarely, if ever, be true. At the present time, I suspect, most designers of bolted joints ignore the analytical complexities which nonlinear behavior introduces. The factors "past experience" and "over-design"—often expressed as a codified safety factor or as a limit on "allowable stress"—compensate for any differences between estimated and actual bolt or joint loads. When optimum joint design is desired, however, and there is considerable concern about the consequences of joint misbehavior, the designer may want to consider the actual, nonlinear response of the bolt and joint to external loads. To help people in that situation we'll begin this chapter by reviewing some of the factors that lead to nonlinear behavior.

Following this discussion we'll go on to examine two other factors, which can cause bolt loads and clamping forces to differ from those predicted by the classical joint diagrams of Chapter 10. One of these factors is a change in temperature of the parts. As we'll see, this can affect the assembled joint in several ways. The second factor is a phenomenon called flange rotation, which affects only certain types of joints, but which can also be troublesome.

This doesn't complete the list of factors that can affect bolt loads and clamp forces in service. Other possibilities include gasket creep, self-loosening of the bolts, stress relaxation, corrosion, and bolt fatigue. We will, however, discuss these in later chapters rather than here.

One of the reasons why joint behavior is so complex is that the actual response of a joint to external loads is often elastoplastic and almost always nonlinear. Joint stiffness and bolt tension, for example, can be a function of the magnitude of applied loads, and of the point of application of applied loads, as well as a function of joint dimensions or the modulus of elasticity. Such behavior is very difficult to define mathematically. In practice, the actual behavior of most joints has to be determined by experiment, or by sophisticated finite-element analysis.

As a preliminary example of the complexities of joint behavior, here's one type of joint nonlinearity which has been reported. A finite-element analysis, partially confirmed by experiments conducted separately by another investigator, has suggested that the stiffness of a bolted joint is not merely a function of its geometry, material, etc. but is also affected by such things as the magnitude of the loads on the joint, the finish of contact surfaces, and the amount of friction between joint members [21]. The authors suggest that a change in joint stiffness occurs when a joint is loaded, a change caused by a reduction in the area of contact between joint members. The initial, preloaded, area of contact is quite large. This means that a lot of joint material is involved, and this means a stiff joint. When a tensile or moment load is applied, the joint members are partially pulled away from each other, reducing the interface contact area and therefore reducing the stiffness of the joint.

These authors reported a theoretical 50:1 reduction in joint stiffness as the external load rises from zero to something approaching initial preload. This would mean that a simple joint diagram could no longer be used to estimate bolt loads or the like. The shape of the diagram would keep changing as loads changed. Interface friction influences the behavior by inhibiting slip between joint members, making it harder for them to rotate away from each other. Higher friction, therefore, reduces the amount of change in joint stiffness as loads are applied.

I can neither confirm nor deny any of these theories, hypotheses, or experimental results. There's apparently no question, however, that actual joint behavior is far more complicated than is suggested by the classical joint diagram, even if we add the hypothetical loading planes. Time will tell whether or not a new, general-purpose (and hopefully easy to use) model of joint behavior will be developed, or whether, alternatively, each joint configuration will require its own finite-element analysis, or equivalent.

In any event, we cannot study the full elastic (or elastoplastic) behavior of joints in an introductory text. The analytical tools I've described (for example, in Chapter 10) have helped many in the past, and will help many more in the future. If they lead to overdesign, experiment and experience will often be used to modify the resulting designs. A designer will have to concern himself with the true, nonlinear behavior of a joint only if faced with a special critical design, or a special intractable problem.

There are, however, some other behavioral complexities most of us can't avoid. These include the phenomenon known as flange rotation, and the effects of a drastic change in temperature on a bolted joint. We'll look at these beginning with Section 11.4.

11.1 EFFECT OF PRYING ACTION ON BOLT LOADS

In the simple analysis of the behavior of bolted joints under tension loads, it is always assumed that the resultant external load is applied at some point along the axis of the bolt. This is a very useful simplification, since it leads to linear behavior, and therefore allows us to analyze the behavior of the joint and the bolt with simple mathematics.

Axial loads, however, are rarely, if ever, encountered in practice. It is far more common for external tensile loads to be applied off to one side of the bolt. This is called a "prying load"; such a load can drastically increase the amount of tensile and bending stress produced in the bolt by a given external force.

11.1.1 Definition of Prying

To see why, let's bolt a crowbar to a table. If we now apply an external load (L_X) to the outer end of the bar, the bolt will have to develop a force larger than L_X (determined by the lever ratio $1/f$) to hold the bar to the table, as in Figure 11.1.

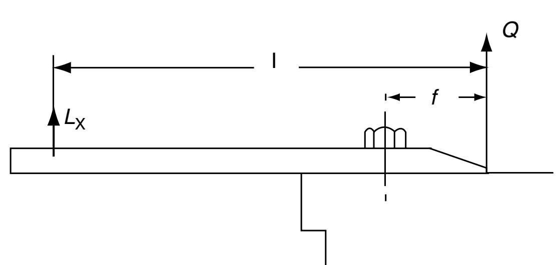


FIGURE 11.1 An offset external load exerts a prying action on the bolt, as suggested by this lever analogy.

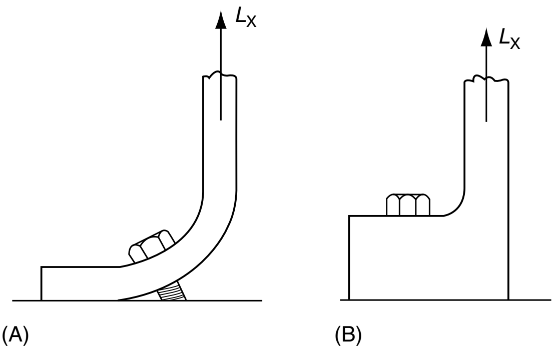


FIGURE 11.2 Joint members can act as levers to pry the bolt. The action is far more severe if the joint members are (A) flexible than if (B) they are not.

The same sort of thing happens in a structural joint such as that shown in Figure 11.2A, if the flange is flexible enough to become a lever. Note that you don't get much prying action if the flange is very rigid (Figure 11.2B). Under these conditions the bolt reacts to an external load just the way it would if the external load were applied axially. Both upper and lower flange members must be rigid for this to be the case.

Prying action is often encountered, as suggested by the sketches in Figure 11.3.

11.1.2 Discussion of Prying

Unfortunately, we can't use a simple lever equation, as we did for the crowbar, to compute bolt force under prying loads, because we're dealing with complex, distributed stress and strain in elastic–plastic bodies.

Empirical formulas have been derived to estimate the magnitude of bolt forces produced in special cases of prying [1–3], and finite-element analysis has been proposed for more

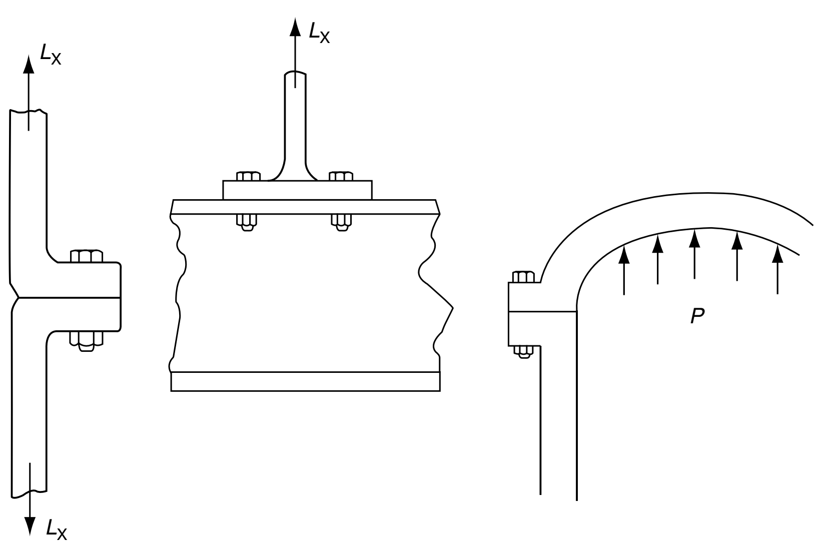


FIGURE 11.3 Tension loads on the joint are offset from the axis of the bolts in most joints, such as those shown here.

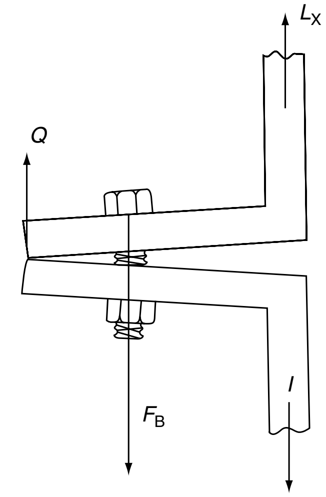


FIGURE 11.4 A bolt subjected to prying must support the external load L_X plus the reaction force Q (shown schematically).

general cases [4]. More usefully, the German engineering society Verein Deutscher Ingenieure has developed a set of equations for analyzing the effects of prying loads, and we will look at those in a moment. First, however, let's discuss the effects in general terms to better understand what we're trying to analyze.

1. A bolt subjected to a prying load must ultimately resist the full external load plus the full prying force (see Figure 11.4), or the joint will fail.

$$F_B \geq L_X + Q \tag{11.1}$$

2. Note that Equation 11.1 doesn't include preload, because it defines the load that the bolt ultimately must resist. The bolt won't see the full external load, plus prying load, at low values of external load, any more than it would fully see a small axial tension load. In each case it will see only some portion of the external load (or external load plus prying load), depending on the stiffness ratio between bolt and joint (K_B/K_J)—at least until joint separation. After separation, it will see all of the applied loads.

   We can illustrate prying with a modified joint diagram [5]. Compare Figures 11.5 and 11.6. Everything that the study of axial loads has taught us—about load sharing

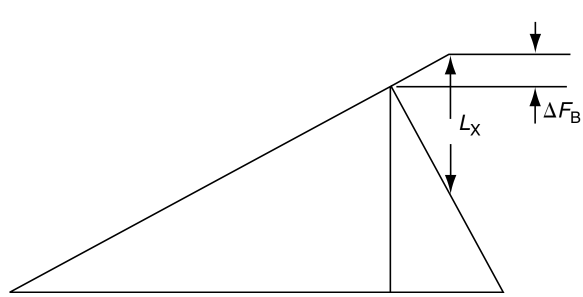


FIGURE 11.5 If the bolt load is axial, the bolt sees only a portion of the applied L_X until $L_X \geq L_{X\text{crit}}$.

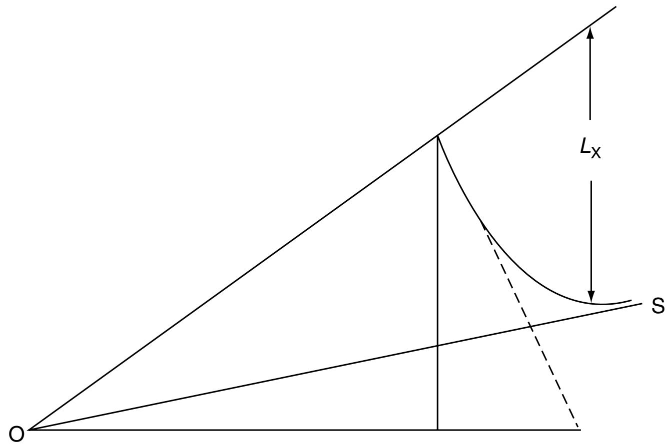


FIGURE 11.6 Joint diagram for a prying load. If the flange is flexible enough, the tension load can't unload it completely. The joint experiences one-sided liftoff, starting near the point of application of the tensile load. After liftoff the joint unloading line approaches the line O–S asymptotically.

between bolt and flange, about cyclical external loads, etc.—can be applied to these prying joint diagrams. That's why it was useful for us to study axial loads, even if they don't always exist in practise.

Before we leave Figure 11.6 note that the joint's curved, unloading line approaches the line O–S asymptotically as L_X increases. We'll discuss this, and its effect, later in the chapter.

3. Notice that the equation $F_B \geq L_X + Q$ reduces to $F_B \geq L_X$ when the stiffness of the joint becomes very high (or the bolt-to-joint stiffness ratio K_B/K_J becomes very small), confirming the sketch in Figure 11.2B.
4. We saw when studying axial loading that a small K_B/K_J ratio was very desirable. It reduced the percentage of external load seen by the bolt (at least until joint separation), and it therefore improved the static load capability and the fatigue life of the joint. Paragraph 3, above, gives us still another reason why a small K_B/K_J ratio is desirable: A stiff flange eliminates the force multiplication caused by prying action.
5. Prying forces can be reduced, with reference to Figure 11.7 by [1]: Increasing the distance a from the center of the bolt to the fulcrum, decreasing the distance b from the bolt center to the point of application of the external load, and increasing the thickness of joint members.
6. Adding additional outboard fasteners as in Figure 11.8 doesn't help much—especially beyond the second row. Almost all of the load is seen by the first (innermost) row.
7. Prying always bends the bolt, increasing stress on one side more than the other.
8. Experience shows that bolts which fail under prying loads fail in the threads. Longer bolts (to reduce K_B or bending stresses) don't seem to help much.
9. If a group of bolts are involved, then, altogether, they must support (ultimately) the external load and the prying load.

$$\sum F_B \geq L_X + Q$$

All comments made above for a single bolt still apply, but each bolt in a group of N bolts sees only $1/N$ of the total load.

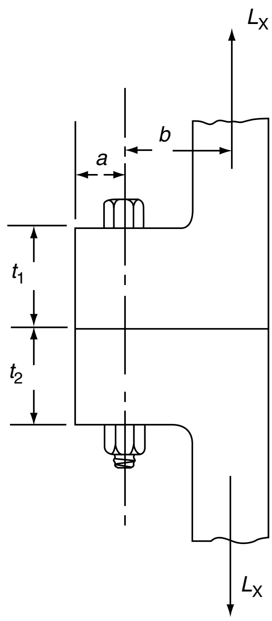


FIGURE 11.7 Prying forces can be reduced by increasing a, t_1, or t_2 and by decreasing b.

10. Prying can also be illustrated by our alternative joint diagram.

First, let's review the axial loading situation. When we first start to build up external load on an axially loaded bolt, there is little change in bolt force, since most of the newly applied external load will be absorbed by the flange. After we have reached the critical external load, however, the bolt absorbs all additional external load (Figure 11.9).

The same thing occurs if we have a prying situation but both flange members are very thick. If the flanges are flexible enough to create a prying action, however, the curve is altered. It starts along the same line it followed when the flange was rigid, but once the external load becomes large enough to flex the flange, and therefore to create

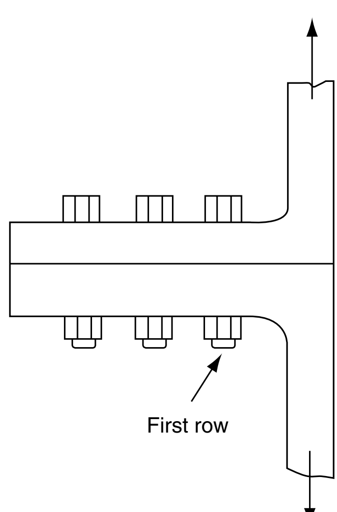


FIGURE 11.8 Only the bolts in the first row will see, or have to resist, prying forces.

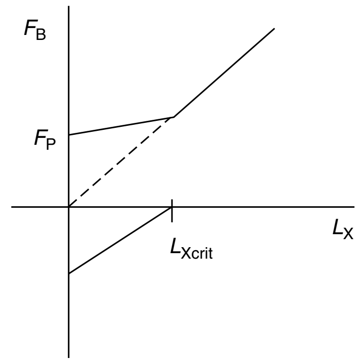


FIGURE 11.9 Alternative joint diagram for a bolt subjected to an axial tension load.

prying action, the force within the bolt becomes greater than that which would be produced by the same external load applied axially. The bolt force stays greater than expected, by an amount equal to the prying force (Q) until rupture occurs, as shown in Figure 11.10.

11. Changing the amount of preload in the bolt doesn't affect the ultimate rupture point. After all, the load in the bolt is well beyond preload at this point, as would be predicted by the joint diagram. Figure 11.11 shows the response to different preloads.

11.1.3 Prying Is Nonlinear

We began this chapter by saying that the behavior of most joints is nonlinear (even if it's elastic), and that such things as tension in the bolt can be a function of the point of application of external load to the joint. Analysis of prying action gives us insight into one of the reasons for these statements. Compare Figure 11.9 with Figure 11.10, for example. In Figure 11.9 we assumed axial loading. The bolt and joint, working together, behave linearly until joint separation (first leg of the line defining bolt load as a function of external load). After separation, the bolt follows a second, but still linear, path on its own. In Figure 11.10, by contrast, the bolt load is a roughly S-shaped function of the external load—not because the bolt itself has become a nonlinear spring, but because the mechanism by which the bolt is

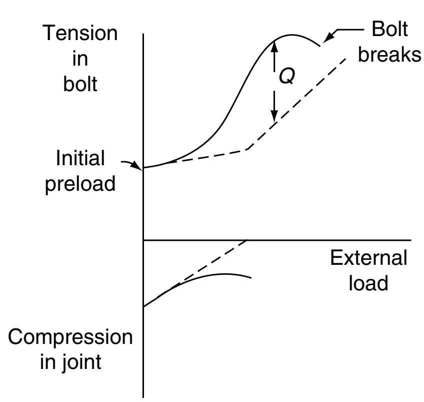


FIGURE 11.10 Alternative joint diagram when the bolt is subjected to a prying load. (Modified from Fisher, J.W. and Struik, J.H.A., *Guide to Design Criteria for Bolted and Riveted Joints*, Wiley, New York, pp. 260ff, 1974.)

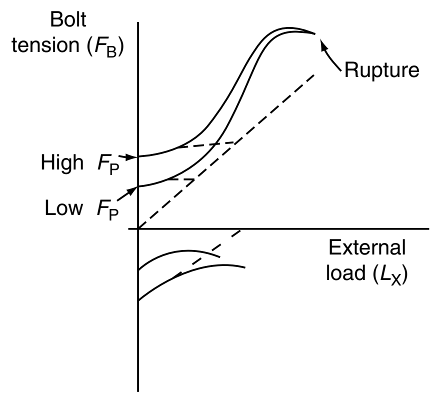


FIGURE 11.11 An increase or decrease in initial preload doesn't change the point at which a bolt breaks.

loaded in the system is nonlinear. Thanks to the lever-like prying action, the bolt sees a greater-than-expected increase in bolt tension as the external load is increased. The loading curve hooks over when the bolt can't stand any more load; it yields and then breaks.

If the external load were removed before bolt yield, the bolt tension would return to original preload along the original curve or thereabouts; so behavior of the system could be both fully elastic and strongly nonlinear.

11.2 MATHEMATICS OF PRYING

11.2.1 In General

Joint failure because of prying action was first noticed in structural steel joints, leading engineers to increase the stiffness of structural steel members to the point that prying was eliminated. At least, the problem was eliminated in structural steel work as long as the designer obeyed the rules. More recently, however, it has been realized that prying action is never truly eliminated from any joint which experiences an offset external load. In fact, some engineers believe prying occurs even if the load on a single bolt is apparently along the axis, because the face of the nut is never exactly perpendicular to the axis of the threads. In any event, prying is very common; it leads to nonlinear behavior of the joint, and it can cause problems in critical joints. Design calculations based on the assumption that the joint will behave in a linear manner can lead to conclusions that are wrong—conclusions concerning such things as the preload desired in a joint, the fatigue resistance of a joint, and the like.

11.2.2 VDI's Analytical Procedure

The German engineering society Verein Deutscher Ingenieure has published guidelines for the design of nonlinear joints, taking into account the fact that in practice almost all loads are applied at some point other than along the axis of the bolts [5–7,22]. They refer to such loads as "eccentric," incidentally, and I will do so here. We examined the stiffness of such joints in Chapter 5 (see Figure 5.12).

Let's consider the joint shown in Figure 11.12. The centerline of the bolt is offset from the centerline or "axis of gyration" of the joint by a distance s. The external, tensile load is applied along a line of action at a distance a from the axis of gyration. Experiment and analysis show that when such a joint is loaded the contact pressure between joint members is

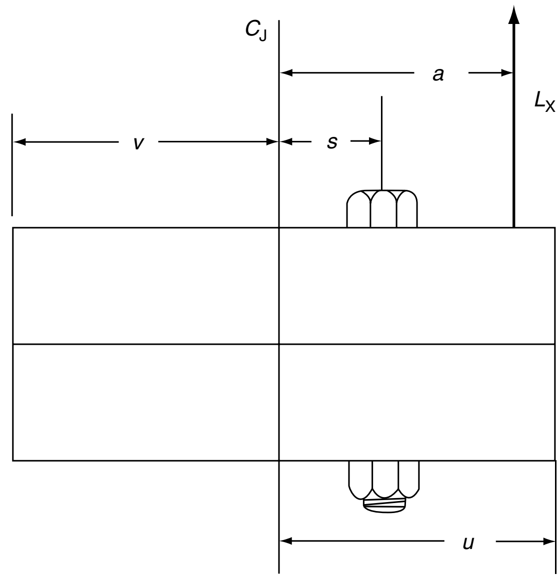


FIGURE 11.12 Eccentrically loaded joint and the symbols used by VDI to identify the distances from the axis of gyration of the joint (C_J) to the axis of the bolt (s), the line of action of the applied load (a), and the edges of the joint (v and u). The distance a is always taken as positive. The distance s is considered positive if it's on the same side of the axis of gyration as a; it's taken as negative if it's on the opposite side of the axis of gyration.

not uniform, as was assumed in the Chapter 10 derivation of the equations for linear elastic analysis of a joint, but is greater on the bolt side of the joint than on the other. In fact, if the bolt is offset from the centerline of the joint by a substantial amount, and the joint has sufficient elasticity, merely tightening the bolt can separate the joint along the free edge, as suggested in Figure 11.13.

As the external tension load is built up on the joint, the interface contact pressure changes. In general, it is reduced under the point of application of the external load, and it is increased at the opposite side of the joint, as suggested in Figure 11.14.

References 5, 7, and 22 give mathematical expressions which can be used to compute the tension in the bolt and the clamping force on the joint when the system is exposed to eccentric loads. These are derived with the help of two basic equations first encountered in Chapter 12. Let's see them again now.

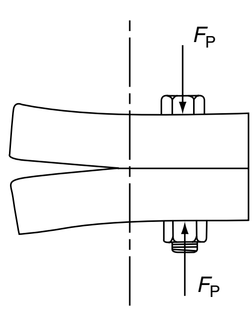


FIGURE 11.13 Preloading an eccentric elastoplastic joint can actually separate joint members, as shown here.

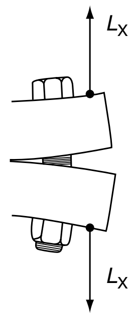


FIGURE 11.14 The interface contact pressure between joint members shifts as external load is applied to the joint, eventually opening the joint underneath the point of application of the load.

$$\Delta F_B = \Phi L_X \tag{11.2}$$

$$\Delta F_J = (1 - \Phi)L_X \tag{11.3}$$

where

ΔF_B = change in tension in the bolt created by external tensile load L_X (lb, N)
ΔF_J = change in clamp force created by external tensile load L_X (lb)
Φ = the load factor

We used ΔF_B and ΔF_J in Equations 10.15, 10.16, 10.24, and 10.25 to describe the response of concentrically loaded bolts and joint members to assembly variables, external loads, and service conditions. Later in this chapter we'll use revised versions of these equations to describe the response of eccentrically loaded bolts and joints. The difference between the Chapter 10 equations and the revised ones will be in our choice of the load factor Φ (and the added effects of differential expansion). At this point we're about to define new 3Φs for various types of eccentricity. In Chapter 10 we defined our 4Φs in terms of bolt and joint stiffnesses (K_B and K_J). When we write the more complex equations for eccentric joints we'll find it's easier to use bolt and joint resiliencies.

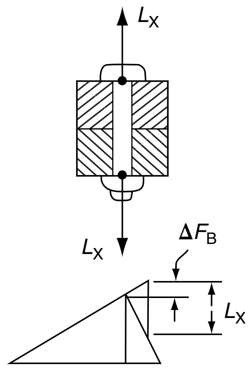
Resilience is the reciprocal of stiffness; for example,

$$r_J = \frac{1}{K_J} \quad r_B = \frac{1}{K_B} \tag{11.4}$$

where

r_J = resilience of concentric joint (in./lb, mm/N)
r_B = resilience of concentric bolt (in./lb, mm/N)
K_J = stiffness of joint (lb/in., N/mm)
K_B = stiffness of bolt (lb/in., N/mm)

In the same way, the resilience of an eccentric joint (r'_J or r''_J from Figure 11.15) would be the reciprocal of the stiffness of that joint (see Figure 5.12). Expressions for the load factor Φ for different situations are given in Figures 11.15 and 11.16. The eccentric joints pictured here must meet the conditions described in Chapter 5 (see Equation 5.19 and Figures 5.10 and 5.11). Here's a summary list of the load factors we have discussed, plus those we're approaching now.



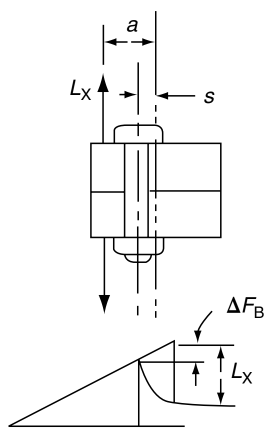
(A)

In general

$$\Delta F_B = \phi L_X \text{ and } r = \frac{1}{K}$$

For concentric joints

$$\phi_K = \frac{r_J}{r_J + r_B}$$



(B)

For eccentric joints

$$\phi_{ek} = \frac{r''_J}{r''_J + r_B}$$

where

$$r''_J = r_J\left(1 + \frac{a}{s}\lambda^2\right)$$

$$r''_J = r_J(1 + \lambda^2)$$

$$\lambda = \left(\frac{s}{R_G}\right)^2 \frac{A_C}{A_T}$$

FIGURE 11.15 Equations for the load factor Φ for (A) concentric and (B) eccentric joints when the external load is applied between the head of the bolt and the nut. r_J = resilience of a concentric, elastic joint (perhaps determined by the equivalent cylinder method described in Chapter 5); $r_J = 1/K_{jc}$ (in./lb, mm/N); r'_J = resilience of an eccentric joint in which the bolt and external load are coaxial (i.e., $a = s$) (in./lb, mm/N); r''_J = resilience of an eccentric joint when bolt and load are found at different distances from the centerline of the joint (in./lb, mm/N); r_B = resilience of the bolt (in./lb, mm/N), $r_B = 1/K_B$; A_C = cross-sectional area of an equivalent cylinder (in.^{2}, mm^2) (see Figure 5.10); A_J = cross-sectional area of eccentric joint (in.^{2}, mm^2) (see Figure 5.11); R_G = radius of gyration area A_J (in., mm) (see Figure 5.12 for alternative expressions for the resiliences).

Φ_K = load factor for a simple, concentric joint when the external load is applied to the bolt head and nut. This is defined in terms of bolt and joint stiffnesses in Chapter 12 and in terms of resiliencies at the top of Figure 11.15.

Φ_H = load factor for a simple, concentric joint when the external load is applied along loading planes within the joint members. This is described in terms of joint and bolt stiffnesses in Chapter 12 and in terms of resiliencies at the top of Figure 11.16.

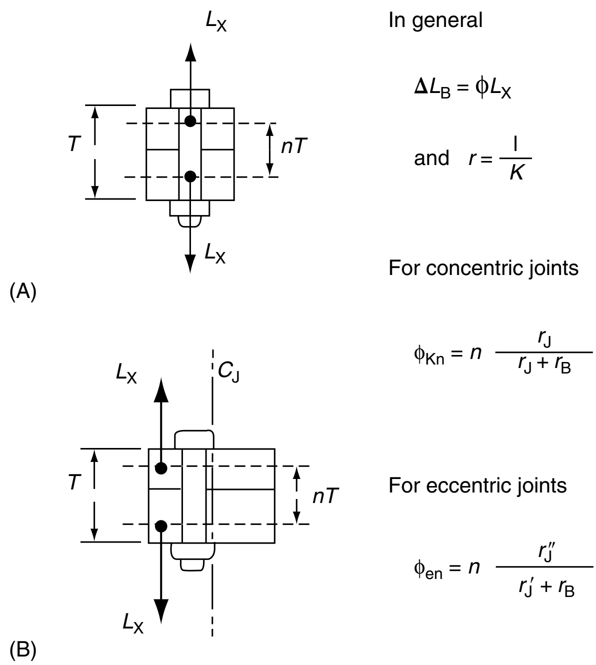


FIGURE 11.16 Equations for the load factor Φ for (A) concentric and (B) eccentric joints when the external load is applied between internal loading planes. See Figure 13.15 for definitions of most of the terms used. T = grip length (in., mm).

Φ_{ek} = load factor for a joint in which both the axes of the bolt and the line of action of the external load are offset from the axis of gyration—and when the external load is effectively applied to the head of the bolt and to the nut. This Φ is that given in Figure 11.15B. (The axis of gyration is the centerline of the joint. It is shown in Figure 13.15 but identified only in the bottom sketch in Figure 11.16.)

Φ_{en} = load factor for a joint in which both the bolt axis and the line of action of the external load are offset from the axis of gyration—and the external load is applied along loading planes within the joint members. This Φ is given at the bottom of Figure 11.16.

One other situation must be described, that in which the bolt axis coincides with the axis of gyration ($s = 0$) but the external load is offset from the axis of gyration ($a \neq 0$). In this case Φ_{en} reduces to Φ_{Kn}. This situation is not illustrated.

If $a = s$, i.e., where bolt and load are coaxial but both are offset from the axis of gyration, we use the equations for Φ given in Figures 11.15 and 11.16 and merely enter the same value for a and s. The expression for the resilience of the joint in this situation is given in Figure 5.11.

11.2.3 Critical Loads and the Preloads Required to Prevent Joint Separation

We are always interested in the critical external load ($L_{X\mathrm{crit}}$) required to reduce the contact pressure between the joint members to zero. In this case it will become zero only under the point of application of the load, of course; on the opposite side of the joint, pressure will actually have increased (because this is a prying load). Nevertheless, we are interested in determining this critical load.

It turns out that for a nonlinear, eccentrically loaded joint, the critical load, like the stiffness and the load factor, is not only a function of the dimensions of joint members, but is also a function of the distance between the centerline of the joint and the centerline of the bolt, and of the distance between the centerline of the joint and the centerline of the point of application of the external load. The bolt location is easy to determine, of course. But the point of application of the resultant external load on a joint is often unknown, since the external load is created by distributed forces (weight, pressure, inertia, etc.) rather than by a point force.

In general, if bolt and load are offset from the axis of gyration, the relationship between the residual assembly preload required in a joint to prevent a given external load L_X from causing one-sided liftoff near the point of application of the load is given by the expression:

$$F_{Pr} = F_{Pmin} + \Delta F_J \tag{11.5}$$

where

F_{Pr} = residual assembly preload (lb, N)

F_{Pmin} = clamp force at liftoff (and therefore the minimum clamp force required to prevent liftoff) (lb, N)

ΔF_J = the change in clamp force created by external load L_X and, therefore, our old friend $(1-\Phi)L_X$

The clamp force at liftoff can be computed from:

$$F_{Pmin} = \frac{(a-s)u}{R_G^2 + S \cdot u} L_X \tag{11.6}$$

where

F_{Pmin} = minimum preload required to prevent separation of an eccentric joint under the point of application of the load (lb, N)

L_X = external load (lb, N)

R_G = radius of gyration of the cross-sectional area of the joint (in., mm)—see Equations 5.23 through 5.27

a, s, and u are defined in Figures 11.12 and 11.17.

If the axis of the bolt coincides with the axis of gyration but the line of application of the external load is offset (a is positive), the clamp force required to prevent liftoff is computed from

$$F_{Pmin} = \frac{auL_X}{R_G^2} \tag{11.7}$$

Note that the eccentric joint we have been discussing has not necessarily been loaded to the point where something has yielded. The fact that its behavior is nonlinear does not mean that it is deforming plastically. It would not necessarily exhibit any of the other effects of plastic behavior—such as hysteresis—upon being unloaded. As far as load sharing and apparent joint stiffness are concerned, however, it appears to be behaving in an elastoplastic manner.

11.2.4 Bending Stress in the Bolt before Liftoff

The following expression can be used to estimate the total stress in the outer fiber in the root of the first load-bearing thread in an eccentrically loaded joint, before and up to the point at which one-sided liftoff will occur [5,22]. This stress will determine the endurance limit of the

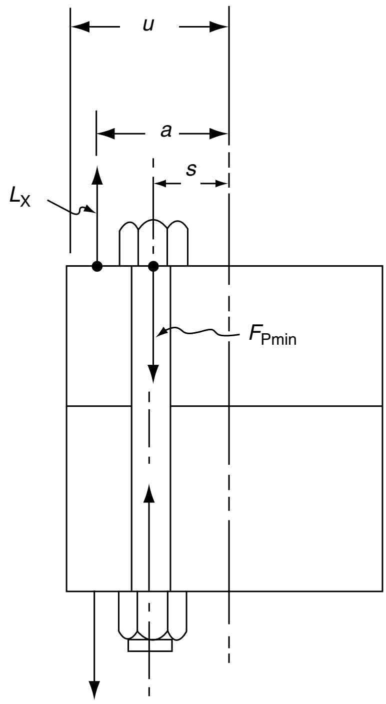


FIGURE 11.17 F_{pmin} is the minimum preload required to prevent separation of this eccentrically loaded joint, under the point of application of the load. (Modified from Junker, G., *Principles of the Calculation of High Duty Bolted Connections—Interpretation of the Guideline VDI2230*, VDI Berichte no. 220, 1974, an Unbrako technical thesis, published by SPS, Jenkintown, PA.)

bolt, as we'll see when we discuss fatigue in Chapter 17. It is caused by two things: the tension in the bolt which has been caused by the external load and magnified by the prying action, and the bending stresses created in the bolt as the joint members are pried apart.

$$\sigma = \left[1 + \left(\frac{1}{\Phi_{en}} - \frac{s}{\alpha}\right)\frac{L_G}{L_e}\frac{E_B}{E_J}\frac{\alpha\pi dm^2}{8A_J R_G^2}\right]\frac{\Phi_{en}L_X}{A_r} \tag{11.8}$$

where

a, s = dimensions illustrated in Figures 11.12 and 11.15 (in., mm)
Φ_{en} = load factor for an eccentrically loaded joint when the load is applied at some point within the joint members
A_J = effective cross-sectional area of the joint (in.^{2}, mm^2) (see Equation 5.18 and Figure 5.11)
R_G = radius of gyration of the joint (in., mm) (see Equations 5.23 through 5.27)
L_G = grip length of the joint (in., mm)
L_e = effective length of the bolt (see Chapter 4) (in., mm)
E_B = modulus of elasticity of the bolt material (psi, GPa)
E_J = modulus of elasticity of the joint material (psi, GPa)
L_X = the external load on the joint (lb, N)
σ = the maximum stress in the outer fiber of the root of the first, load-bearing thread (psi, Pa)

Note that this stress doesn't enter our basic joint equations (Equations 10.24 and 10.25) but we will use it when we design a joint, in Chapter 18, to evaluate the bolt recommended by those joint equations.

11.2.5 Effects of Very Large External Loads

Most of the discussion so far has dealt with external loads which are less than or just equal to the amount required to create one-sided liftoff or separation of joint members near the point of application of the external load. Rarely—but sometimes—we're interested in the effects of still higher external loads. I'm neither willing nor able to give you mathematical formulas for the effect of such a load on bolt tension or clamp force, but we can approximate the results by reference to Figure 11.6.

We see there that the joint's unloading line approaches the sloped line O–S asymptotically. We can construct line O–S by using the following expression to compute its slope.

$$L_X(a + v) = F_B(s + v) \tag{11.9}$$

where

L_X = the external load on the joint (lb, N)

F_B = the tensile force in the bolt (lb, N)

a, s, and v = dimensions illustrated in Figure 11.12. Note that we use the absolute value of v; a is always assumed to be positive; s is positive if it's on the same side of the axis of gyration as a; otherwise it's negative

11.3 OTHER NONLINEAR FACTORS

11.3.1 Nut–Bolt System

Prying or eccentric action is not the only cause of nonlinear behavior of a bolted joint. Here's another.

Let's assume that we apply tension to a steel rod of uniform cross-section by pulling on it with our fingers, as shown in Figure 11.18. As we pull, we're going to measure the distance between the tips of the fingernails on our two index fingers; we're also going to measure the change in length of the rod.

Because of the way in which our fingers are constructed, we would detect a large and visible change in the distance between our fingernails even though the balls of our fingertips had only rolled, not slipped, over the surface of the rod. The simultaneous change in length of the rod itself, however, would be very small, because we would not be able to exert much tension this way.

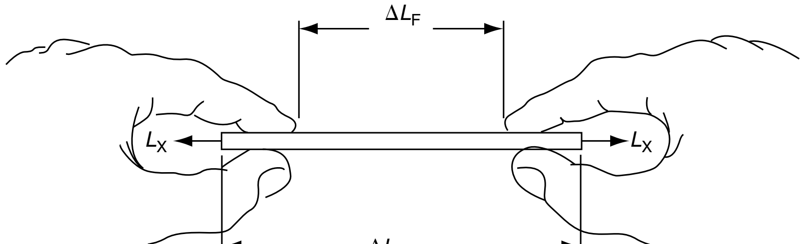


FIGURE 11.18 We pull on a steel rod to stretch it, measuring as we do the change in length of the rod (ΔL_R) and the change in the distance between our fingernails (ΔL_F).

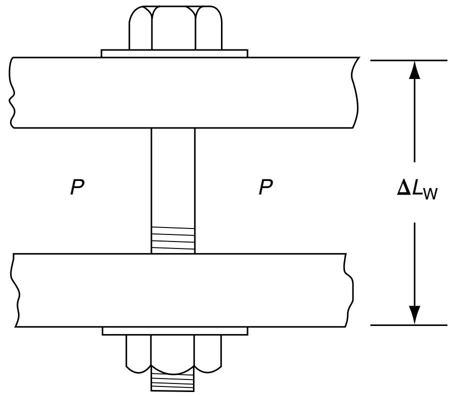


FIGURE 11.19 Internal pressure (P) applies a tension load to this bolt, nut, and washer system. (Modified from Pindera, J.T. and Sze, Y., *CSME Trans*., University of Waterloo, Waterloo, 1972.)

If we plotted the change in length of the rod as a function of the applied tension, we would find that it would be a straight line. If we plotted the change in distance between our fingernails as a function of applied load, however, we would find that it was, in general, not to be a straight line, but depended instead on the load deformation behavior of our flesh and muscles. I'm not prepared to suggest what the resulting curve would look like!

A similar situation occurs when we measure the change in spacing (ΔL_W) between the washers on a bolt, nut, and washer system, which is being subjected to internal pressure load, as in Figure 11.19.

If we also measure the change in length of the total bolt (ΔL_B) as a function of the applied load, we will find that it is a straight line. If we were to compute the stiffness of the bolt (the slope of the line) using the equations of Chapter 5, we would find that our calculations would probably approximate the measured stiffness. The bolt would behave in the anticipated linear elastic fashion, as long as we did not use too much pressure to load it.

If we also plot the change in the distance between washers (ΔL_W) in this situation as a function of applied load, however, we will find that the behavior is very nonlinear, as suggested in Figure 11.20 [8,9].

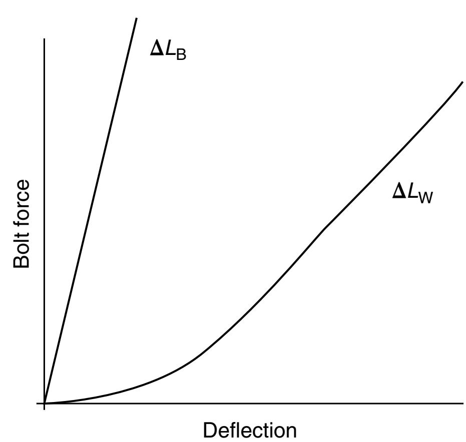


FIGURE 11.20 Distance between washers (ΔL_W) of the system shown in Figure 11.19 as a function of the applied load. Simultaneous change in length of the bolt alone (ΔL_B) is also shown. (Modified from Pindera, J.T. and Sze, Y., *CSME Trans*., University of Waterloo, Waterloo, 1972.)

The thing we are loading—the bolt—behaves in an elastic fashion, but our method of applying the load—through the nut-and-washer system—introduces nonlinearities if we measure the result at the wrong point. The reason for this nonlinear behavior, of course, is that the nuts and washers have to settle into the threads of the bolt in order to push on the bolt, some embedment occurs, the washers may flatten out a little, etc. It is the nature of the loading mechanism rather than the thing being loaded, which determines the apparent behavior.

Who cares? Well, it turns out that the joint designer cares—or should. After all, the joint neither knows nor cares about the behavior of the bolt as an isolated body. It is always loaded by a bolt, nut, and washer assembly. So the force versus change-in-length behavior which it sees is that reflected by the distance between the two washers which are used to clamp it together. As a result, the distribution of an external load between bolt and joint, the apparent stiffness of the joint, the apparent stiffness of the bolt as far as the joint is concerned, etc. are all drastically different than would be predicted by calculations based on the assumption that the joint and bolt will both behave as uniformly loaded, linear elastic members.

Some, at least, of this nonlinear behavior is caused by localized plastic yielding in the threads, embedment, etc., so the behavior of a joint which has been preloaded, released, and then reloaded will probably be more linear than the behavior of a fresh joint.

If sufficient load is applied to the bolt, furthermore, it will be operating in a region where its behavior is elastic (the upper or right-hand end of the ΔL_W curve shown in Figure 11.20). Even here, however, the stiffness of the bolt, as seen by the displacement between washers, is going to be only about half the stiffness computed by our equations (which consider merely the body of the bolt and not the bolt–nut–washer system). This is because the bolt, the nut, and the washers are each springs; they are each loaded, in series, and their combined stiffness will be a function of the stiffness of each one.

$$\frac{1}{K_T} = \frac{1}{K_B} + \frac{1}{K_N} + \frac{2}{K_W} \tag{11.10}$$

where

K_T = stiffness of the entire bolt–nut–washer system (lb/in., N/mm)
K_B = stiffness of the bolt (lb/in., N/mm)
K_N = stiffness of the nut (lb/in., N/mm)
K_W = stiffness of the washer (lb/in., N/mm)

Note that in this situation at least some of the nonlinear behavior is determined by plastic yielding, etc. within the system.

We could, therefore, expect to find hysteresis effects, etc. if we made a close enough examination.

Note, too, that in this case the apparent stiffness of the bolt (as seen by changes in the distance between the washers) is a function of preload or tension level as well as of the usual dimensions. The system has a very low stiffness at low load levels and a stiffness approaching half that of the bolt alone at high loads. This is similar to the nonlinear behavior of a block under compressive loads, as we saw in Chapter 5 (Figure 5.8) and is really caused by the same phenomenon: initial plastic deformation of the body under compressive stress.

The situation illustrated in Figure 11.20 was encountered in some recent experiments with a Superbolt torquenut shown in Figure 11.21. This device consists of a cylindrical nut (called a torquenut) which is run down, by hand, against a heavy, hard washer. A group of jackbolts are then tightened, in a cross-bolting pattern, to tension the large stud or bolt on which the

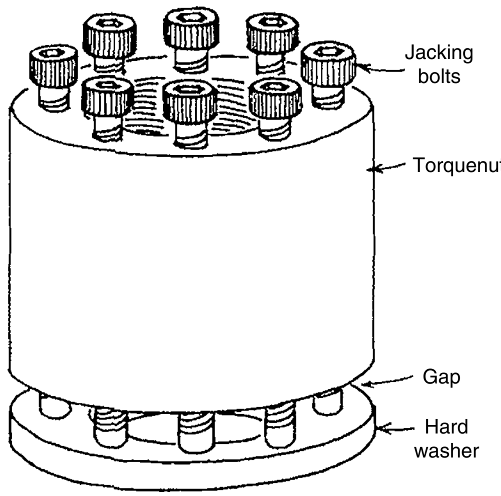


FIGURE 11.21 A Superbolt torquenut allows large-diameter fasteners to be tightened with small, low-torque tools. It also illustrates the behavior shown in Figure 13.20, as explained in the text.

torquenut has been placed. Since only a small wrench, and small amount of torque, is required to tighten the jacking bolts, the technique allows very large fasteners to be tightened in very inaccessible places. It also has other applications, of course.

I describe it here because an attempt was made recently to control the tension built up in the large stud by measuring the change in the gap between the torquenut and the washer as the jacking bolts were tightened. If there were a one-to-one relationship between the change in this gap and the stretch of the bolt, gap measurement would provide a ready means to control the tightening process.

The experiments revealed significant differences between gap change and bolt stretch, however, with the gap change far exceeding bolt stretch. The investigators felt that the difference resulted from the fact that the gap change reflected elastic and plastic deformation of the joint members, washer, and thread surfaces, as well as elastic stretch of the bolt—all as suggested in our earlier discussion [19].

11.4 THERMAL EFFECTS

Now let's look at what a change in temperature can do to bolt loads and the interface clamping force. We'll look at five effects: a change in elasticity or stiffness of bolt and joint members; a loss of strength of the bolt; modification of bolt loads and clamping force by differential thermal expansion or contraction; creep relaxation; and stress relaxation. Note that each of these factors acts to change the clamping force in the joint and the tension in the bolts. We've called changes of this sort "instability in the clamping force." It's important for us to know how to estimate the amount of change which will occur, and to learn how to reduce it, or compensate for it.

11.4.1 Change in Elasticity

The modulus of elasticity of bolting materials decreases as the temperature of the material rises. As a typical example, the modulus of an A193 B7 bolt drops by about 17% when the temperature of the bolt is raised from 70°F (20°C) to 800°F (427°C). Unless there are offsetting differential expansion effects (to be considered soon), the preload in the bolt will decrease in the same ratio, because the bolt gets less stiff. We can estimate the preload in a bolt at elevated temperature by using the simple expression:

$$F_{P2} = F_{P1} \frac{E_2}{E_1} \tag{11.11}$$

where
F_{P2} = preload at elevated temperature (lb, N)
F_{P1} = preload at room temperature (lb, N)
E_2 = modulus of elasticity at elevated temperature (psi, GPa)
E_1 = modulus of elasticity at room temperature (psi, GPa)

Note that temperatures below ambient will increase the preloads introduced during room temperature assembly. By the same token, preloads introduced during "hot bolting" procedures at elevated temperatures will subsequently increase when the system is shut down and the joint cools (again assuming no differential contraction).

11.4.2 Loss of Strength

The tensile strength of most bolts will also decrease as temperature rises. The yield strength of an A193 B8 Class 1 bolt, for example, drops from 30 ksi at room temperature to only 17 ksi at 800°F; and B8 is not recommended for use above 800°F. A heavily preloaded bolt could fail, therefore, if exposed to extreme temperatures—by a fire, for example.

This problem received considerable attention a few years ago. It was discovered that low-cost suppliers of J429 Grade 8 bolts had often made the bolts of boron steel instead of medium carbon steel. Boron steel is permitted for Grade 8.2, but many suppliers had instead marked and sold boron steel bolts as Grade 8. And this caused problems at elevated temperatures.

Boron steel and medium carbon alloy steel have similar properties at room temperature. But boron steel bolts can lose as much as 75% of their prestress after 80 h of exposure to temperatures as low as 700°F (371°C), compared to a 45% loss for the alloy steel bolts [11]. Attempts were made to identify the so-called counterfeit bolts in the system, and to prevent further substitutions of this sort.

11.4.3 Differential Thermal Expansion

One of the most troublesome thermal effects the bolting engineer must deal with is differential thermal expansion or contraction between joint members and bolts. To illustrate the problem, consider the automotive head joint shown in Figure 11.22. This is a sketch of an actual joint. Carbon steel bolts are used to bolt an aluminum head to a cast iron engine block, loading a steel and asbestos gasket. The relative coefficients of expansion for these materials might be as follows (all $\times$ 10^{-6} in./in./°F):

Carbon steel	6–7
Cast iron	6
Aluminum	12–13

Figure 11.23 shows the stress on the gasket (or equivalent tension in the bolts) as the engine in Figure 11.22 is assembled and then used. A certain amount of initial stress is created when the engine is assembled at room temperature (point A). When the engine is started, gasket stress rises sharply to point B, thanks to the fact that the aluminum heats up more rapidly than the bolts, and because the coefficient of expansion of the aluminum is approximately double

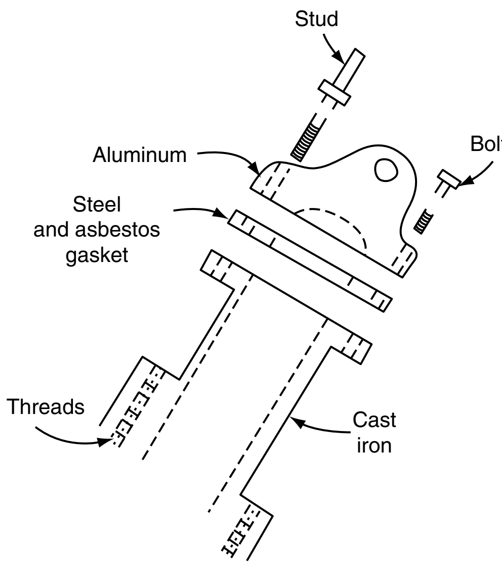


FIGURE 11.22 Automobile engine head. The variety of materials used here results in differential expansion between the aluminum head and the carbon steel bolts. This creates the changes in gasket stress seen in Figure 11.25.

that of the carbon steel bolts. The aluminum tries to expand but is trapped between the cast iron block and the carbon steel bolts. The differential expansion increases the tension in the bolts and the clamping force on the gasket.

As the engine continues to run, the temperature of the bolts rises to more nearly (approximate or equal) that of the aluminum head. As a result the bolts expand some

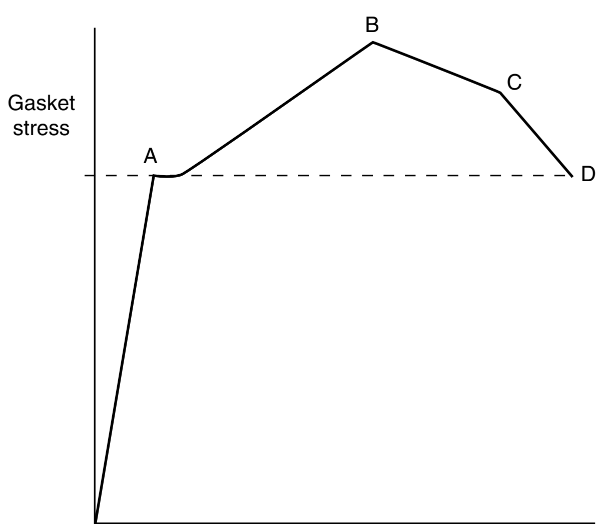


FIGURE 11.23 The automotive gasket of Figure 11.24 is initially loaded to point A during assembly at room temperature. As the engine heats up (ahead of the bolts), the stress rises to point B. It falls to point C when the temperature of the bolts "catches up" with the temperature of the head, then returns to the original preload value at point D when the engine is shut down and returns to room temperature.

more, somewhat reducing the clamping force and stress on the gasket. Because the aluminum still wants to expand more than the other materials, the steady-state stress on the gasket remains higher than the initial assembly stress. When the engine is turned off, and returns to room temperature, the stress on the gasket returns to the original value. At least it will do this if the added stress has not caused some irreversible plastic deformation in the gasket. Thermal cycles can ''ratchet'' all of the clamping force out of a gasketed joint, thanks to hysteresis and creep in the gasket. In the present example, however, the manufacturer says that the stress merely returns to the original value.

It's often useful to be able to estimate the change (increase or decrease) in bolt tension, and clamping force on the joint, created by thermal expansion. We can proceed as follows.

The relationship between initial preload in a bolt, and the change of length of that bolt, can be estimated with Hooke's law.

$$F_P = \frac{A_S E}{L_E} \Delta L_B \tag{11.12}$$

where
F_P = preload (lb, N)
A_S = tensile stress area (in.^{2}, mm^2)
E = modulus of elasticity of bolt (psi, N/mm^2)
L_E = effective length of bolt (in., mm)
ΔL_B = change in length of bolt (in., mm)

The additional tension (or loss of tension) created in the bolt by differential expansion between joint members and bolt (F_T) can be approximated by

$$F_T = \frac{A_S E}{L_E} (\Delta L_J - \Delta L_B) \tag{11.13}$$

where
ΔL_J = change in length (thickness) of the joint (in., mm)
ΔL_B = change in length of the bolt (in., mm)
F_T = the additional tension or loss of tension created by differential expansion (lb, N)

Both the tension in the bolt and the clamping force on the joint will be increased if ΔL_J is greater than ΔL_B. They'll be decreased if the bolt expands more than the joint. We can compute these changes in length/thickness as follows:

$$\Delta L_B = \rho_1 L_G (\Delta t) \tag{11.14}$$

$$\Delta L_J = \rho_2 L_G (\Delta t) \tag{11.15}$$

where
ρ_1 = coefficient of thermal expansion of the bolt material (in./in./°F, mm/mm/°C)
ρ_2 = coefficient of thermal expansion of the joint material (in./in./°F, mm/mm/T)
L_G = the grip length of the joint (in., mm)
Δt = the change in temperature (°F, °C)

It's important to note that the length/thickness changes used in Equations 11.13 through 11.15 are changes that would be caused by a change in temperature if the bolts had not been tightened. For example, if the bolts and joint members were lying on a bench, and the temperature in the room raised or lowered, the parts would experience the changes in dimension calculated by Equations 11.14 and 11.15. Note that Equation 11.13 can also be written in terms of K_B, the stiffness of the bolt, as follows:

$$F_T = K_B(\Delta L_J - \Delta L_B) \tag{11.16}$$

Now, all of the above is conservative. The increase in tension estimated by Equation 11.16 will probably be greater than the actual increase, because the equation assumes, in effect, that the total thermal load on the joint will be seen by the bolts. In practice, the stiffness ratio between bolts and joints, and effects such as flange rotation, will modify the tension created by the thermal load. For example, Evert Rodabaugh of Battelle, in the flange design program FLANGE [10], suggests that:

$$F_T = \frac{\Delta L_J - \Delta L_B}{\dfrac{1}{K_B} + \dfrac{1}{K_{J1}} + \dfrac{1}{K_{J2}} + \cdots} \tag{11.17}$$

where the various K_Js define the stiffness of individual joint members, bolts, gasket, joint rotation, etc.

In another, private, reference, an engineer in a large petrochemical company has used the expression

$$F_T = \frac{\Delta L_J - \Delta L_B}{\dfrac{1}{K_B} + \dfrac{1}{K_J}} \tag{11.18}$$

This reduces to

$$F_T = \frac{K_B K_J}{K_B + K_J}(\Delta L_J - \Delta L_B) \tag{11.19}$$

which says that the change in force in the bolt is equal to that portion of the external load a joint diagram would predict the bolt would see. (See Figure 11.24.)

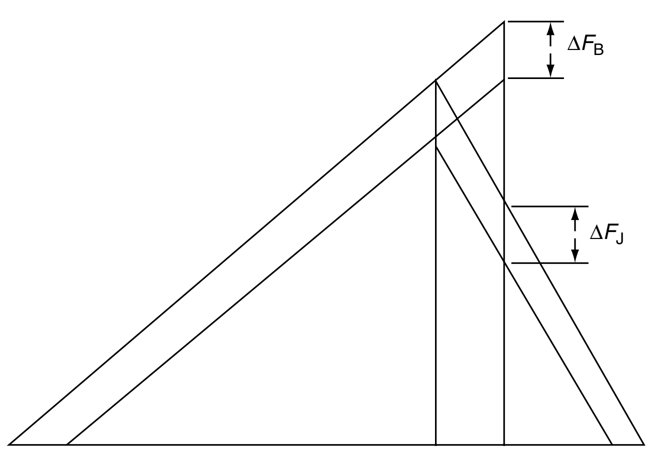


FIGURE 11.24 Differential thermal expansion between bolts and joint members can simultaneously increase—or simultaneously decrease—the tension in the bolts and the clamping force on the joint, causing the entire joint diagram to grow or shrink as shown here.

We can rewrite Equation 11.19 in terms of the load factor (Φ) first defined in Chapter 10 and further discussed in Section 11.2.

$$F_T = \Phi K_J(\Delta L_J - \Delta L_B) \tag{11.20}$$

where

Φ = any one of the load factors we've defined; Φ_K, Φ_{en}, etc.
K_J = the joint stiffness associated with that particular load factor (lb/in., N/mm)
F_T = additional tension or loss of tension (lb, N)

For example, if the joint is loaded eccentrically, the bolt is offset from the axis of gyration and the load is applied at some point within the joint members, we'd use the load factor Φ_{en} as defined in part B of Figure 11.15. We'd also, then, have to use the resilience of that sort of joint, r_J'' (also defined in Figure 11.15) to define the joint's stiffness ($K_J = 1/r_J$). This force F_T is seen by both the bolt and the joint, because it's created as these two elements fight each other. The force caused by differential thermal expansion, in other words, should not be treated like an external load.

We can draw a joint diagram to illustrate this, as in Figure 11.24. I've included an external load in this diagram because a change in temperature is usually accompanied or preceded by the application of load. We're interested in the changes which affect an in-service, working joint. But it's not the external load that causes the changes shown in Figure 11.24; it's the differential expansion. This makes the whole joint diagram—with the exception of the external load line—grow or shrink, depending upon whether F_T is positive or negative.

Note that two different bolt lengths must be used in these equations for accuracy. Since we're interested in the forces created by differential expansion between the joint and bolts, we must look first at the relative expansion of the joint—and of that portion of the bolt which traps the joint. We consider ΔL_B, in other words, only for the grip length of the bolt.

Consider what we might conclude if we focused, instead, on the overall length or on the effective length of the bolt. Assume, for a moment, that bolts and joint members are made of the same material and experience the same rise in temperature. If we computed the expansion of bolt and joint, using different lengths for each—grip length for the joint and effective length for the bolt, for example—we'd conclude that the bolt would expand more than the joint. After all, expansion is an "inch per inch" proposition. So we'd predict a drop in bolt tension. But this would clearly not occur if all parts were of the same material and experienced the same temperature changes. Thermal expansion of such a system would change its dimensions, but not the internal stresses or the forces between subassemblies. The bolt as a free body will expand more than the joint; it's true, but the expansion within and past the nut doesn't create the forces we're concerned about. So we focus on the relative expansion of the joint and of the bolt within the grip length.

When we want to estimate the effect this bolt–joint interference has on bolt tension or clamping force, however, we must use the correct values for bolt and joint stiffness. If we assumed at this point that the bolt was only as long as the grip, we'd conclude that it was stiffer than it really is. So we must now use the effective length of the bolt (grip length plus half the height of the head plus half the thickness of the nut) when computing bolt stiffness K_B. The length of the joint, of course, remains unchanged as the grip length. This is illustrated in Figure 11.25.

11.4.4 Stress Relaxation

Two other thermal effects we must be concerned about are the closely related phenomena of creep and stress relaxation. Creep is the more familiar of the two, and can be illustrated as follows.

Let's assume that we've fastened one end of a steel bolt to a ceiling. We now hang a heavy weight from the lower end, and raise the temperature in the room to 1000°F (538°C).

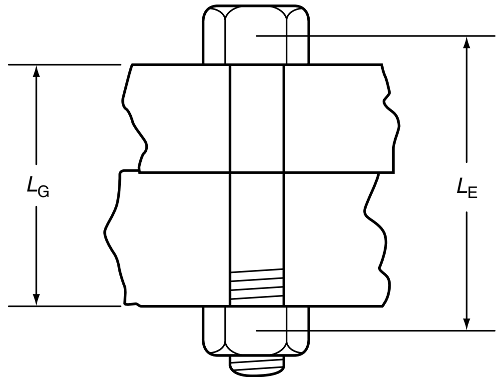


FIGURE 11.25 When computing the effects of differential expansion, one must use the grip length of the bolt (L_G) to estimate the amount of interference between bolt and joint, but use the effective length (L_E) of the bolt when computing the effect this interference will have on bolt length or tension. See text for discussion.

Since this temperature would place the bolt in its creep range, it will slowly stretch, necking down as it does so. Eventually it will get too thin to support the weight, and the bolt will break. The slow increase in length of a material under a heavy, constant load is called creep. It can occur in some materials (e.g., lead) at room temperature, but is more commonly encountered at elevated temperatures.

Stress relaxation is a similar phenomenon. This time, however, we're dealing with the steady loss of stress in a heavily loaded part whose dimensions are fixed. A bolt, for example, is tightened into a joint, which means it is placed under significant stress. But the bolt doesn't get longer. Its length is determined by the joint (and the nut). If exposed to 1000°F, however, the bolt will shed a significant amount of stress—of initial preload, if you will—as the molecules struggle to relieve themselves of the imposed load.

In Figure 3.3 we saw an estimate of the percentage of initial tensile stress various types of bolts would lose in 1000 h if exposed to temperatures up to 1472°F (800°C). A carbon steel bolt, for example, would lose about 90% of its initial preload in 1000 h at 752°F (400°C), while an A193 B8 bolt would lose only 10% or so at that temperature.

These losses, incidentally, would not be repeated. The carbon steel bolt would not lose 90% of the remaining 10% during the second 1000 h period; it would probably have stabilized at the 10% figure, and might serve indefinitely there, at 400°C, if left in place.

This is because the tendency to relax decreases as the driving force, the tensile stress in the bolt, decreases. This is illustrated in Figure 11.26, which shows the relaxation of several A-286 studs in 100 h at 1200°F (649°C) [12]. The stud whose behavior is illustrated by curve C in Figure 13.28 appears to have stabilized, at least. The others may lose a little more as time goes by; but won't lose the 50% or so they lost in the first 100 h.

The fact that a material stabilizes after a certain amount of stress relaxation means that we can compensate for it by overtightening the bolts during initial assembly. If they're going to lose 50%, we put in an extra 50% to start with, assuming that they and the joint can stand this much at room temperature. But we must realize that the loss can be substantial, and can continue for long periods of time. Mayer reports relaxation of 50% in CrNiMo fasteners tested at 500°C, with other materials, tested at 425°C–480°C, still relaxing after periods as long as 10^4 h [14]. Markovets says that E110 (25Cr2MoV) steel, normalized at 1000°C and tempered at 650°C, will relax 20% in the first 100 h at 500°C, 56% (and still relaxing) at 10^4 h [13].

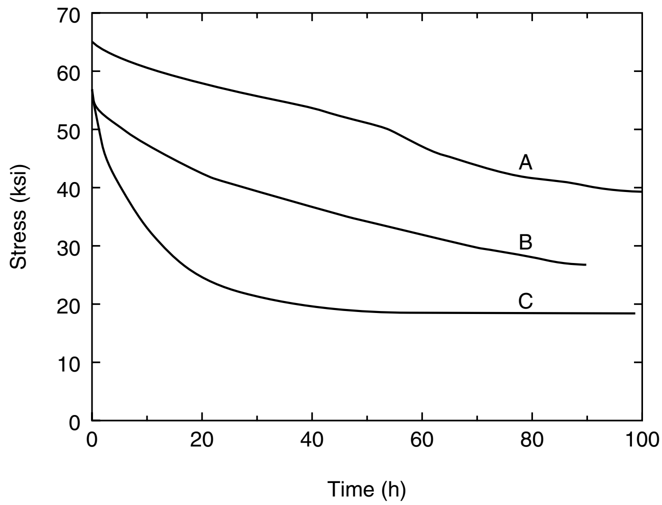


FIGURE 11.26 Stress relaxation of A-286 bolts exposed to 1200°F for 100 h. Bolt A was originally stressed to 70 ksi, B and C only to 60 ksi. In the test the bolts were allowed to creep a small amount, then the loads on them were reduced to reverse the creep. Bolt B was relieved in coarser steps than bolt A.

Note that there can be a penalty for using better materials to reduce stress relaxation. Bolts made from a material such as Nimonic 80A, which can be used to 1382°F (750°C), can cost five to six times as much as bolts made of a more common material such as A193 B7 [18].

Stress relaxation is affected by the geometry of the bolt as well as by the material from which it's made. A test coupon of a given material, for example, will usually relax much less than a bolt of the same material, because so much of the relaxation occurs in the threads. There are exceptions to this rule, but it's generally true [18].

Whether or not threads are rolled or cut can make a difference at the highest temperatures, but this is said not to matter at temperatures where relaxation properties are generally good (presumably below the service limits). Poor-quality threads, however, which we considered in Chapter 3, will relax much more than good-quality threads [18].

The importance of quality threads is also shown by the fact that Grade B16 bolts used with high-quality stainless steel nuts (Grade 8) relaxed, in one series of tests, much less than the same bolts with carbon or low-alloy nuts [18].

Attempts have been made to use readily available creep data to predict the amount of stress relaxation one might encounter in a given situation (because stress relaxation data are much less common). Creep data are subject to much variation, however, with one investigator reporting data which differ significantly from data reported by another. As one result, short-term stress relaxation estimates cannot be based on creep data. Estimates of long-term loss, however, are more reliable. One can use, for example, a relationship called "Gieske's correlation," which says that "the residual stress in a bolting material, after 1000 h at a given temperature may be taken as equal to the stress required to produce a 0.01% creep in 1000 h at that same temperature" [12].

11.4.5 Creep Rupture

Although bolts are loaded under constant strain (constant length) conditions, they can fail by a mechanism known as creep rupture. We tighten the bolt (i.e., load it heavily) at room temperature, then expose it to temperatures in the creep range. Stress relaxation will partially relieve it. Now we return it to room temperature, to perform maintenance on the system, for example. After completing our work, we retighten it to the original loads and put it back in service at elevated temperatures. It will experience a second cycle of stress relaxation.

During each such cycle the bolt experiences some creep damage, even though its length never exceeds its length when first tensioned. After a certain number of cycles it will develop a crack, and will eventually break. The failure is called creep rupture.

It needn't take many cycles to rupture a bolt or stud in high-temperature service. Seven cycles can do it if initial preloads are high enough. Lower preloads in the same studs might extend this to 25 cycles. With only two maintenance cycles a year, however, 25 cycles could equate to 12½ years of useful life [15]. Another source recommends a maximum of six retightening cycles for bolts used in elevated temperature service and recommends that the accumulated plastic deformation be restricted to a maximum of 2% [18]. In any event, designers are cautioned to consider such things when selecting bolting materials and specifying maintenance procedures [16].

In the discussion so far we've considered only creep or stress relaxation in the bolts. These phenomena can also occur in joint members and gaskets, with creep being the more common. The neck of a pipe flange, in high-temperature service, will creep, as might the flanges themselves [16,17]. Gasket creep is also common, and will be discussed in Chapter 19.

11.4.6 Compensating for Thermal Effects

The behavior of a bolted joint will ultimately depend to a large extent on the clamping force on that joint in service, as opposed, for example, to the clamping force created during assembly. Thermal effects which change the initial clamping force, therefore, can be a real threat to behavior.

Fortunately, all of the changes we've looked at have limits; they don't go on forever (at least not rapidly enough to be a problem). Stress relaxation can be severe in the first few hours or few hundred hours, but stresses will eventually stabilize. Most gasket creep occurs in the first few minutes. Differential expansion or contraction ceases when the temperature of the system stabilizes. So we're dealing with transitions from one stable point to another, rather than with continuous change. That's a big help!

We're also dealing, still, with a system of springs which will share loads and changes in load, as suggested by the joint diagram of Figure 11.26. That, too, can be a big help, because it means we can often adjust bolt-to-joint stiffness ratios to reduce the impact of thermal change.

Specifically, therefore, we overtighten bolts at assembly to compensate for anticipated losses from differential expansion, gasket creep, stress relaxation, and loss in bolt stiffness. Before overtightening them, of course, we must determine whether or not the higher preloads will damage anything during assembly or after temperatures (and therefore loads and strengths) have changed.

If we're designing the joint, using Equations 11.21 through 11.23, we'ld have anticipated the amount of overdesign required to compensate for differential expansion. If we're dealing with a troublesome, existing joint we could use the same equations to estimate the amount of overtightening required. We could then use the bolt strength equations of Chapter 3 to decide if the present bolts will take this much stress. If not, we could consider using bolts made of a stronger material.

In Chapter 1 we learned that we usually want to use the highest clamping forces the parts can stand. That's the goal in the process described above.

One common limitation on assembly preload will be the rotation of a raised face flange. It's not unusual to find yourself trapped between too little residual bolt load after differential expansion and stress relaxation and flange rotation great enough to open a leak path. The answer to this dilemma can be more uniform residual tensions in the bolts. Don't tighten them above a limit determined by rotation, but make sure that every one is left as tight as that limit allows.

If more preload isn't possible, or doesn't solve the problems created by thermal change, you should consider altering the stiffness ratio between bolts and joint members. Assuming that the nominal diameter of your bolts is set by the design of the joint, and that you can't change joint stiffness, you may want to increase bolt length. This will reduce bolt stiffness, which will mean less change in bolt tension for a given expansion-interference forced change in dimension. A less stiff bolt will also absorb a smaller percentage of a load change than will a stiffer one; as the joint diagram teaches us.

The normal way to increase bolt length is to stack Belleville washers under the head and nut or to place elongated collars there. Figure 11.27, for example, shows a very long collar used in a nuclear power application to solve a difficult, differential expansion problem in a normally inaccessible joint [20]. Where longer bolts are unacceptable, bolt stiffness can be reduced by gun-drilling a hole down the bolt axis, or by turning down a portion of the bolt.

Another way to reduce thermal effects, at least those involving differential expansion, is to use more similar materials for bolts and joint members.

Although you should avoid it if possible, hot bolting can also be used to compensate for thermal changes. Retighten the bolts after they've relaxed, gaskets have crept, etc. How much torque should you use for this? It's a frequent question and I've never heard a definitive answer. If the change in temperature didn't affect the lubricity of the parts, then reapplication of the original torque should reestablish the original clamping force. If the lubricity has decreased by 10%, torques should probably be raised a similar amount. Perhaps your lubrication supplier can tell you what changes, if any, to expect. If not, an experiment in an oven, testing the force required to slip a heavy block over a plate, might give you a way to estimate the "hot torque." Don't forget, however, that a change in temperature will also affect a lot of other factors which influence the torque–preload relationship—things like fits and clearances, the ease with which operators can reach and tighten the nuts, the calibration

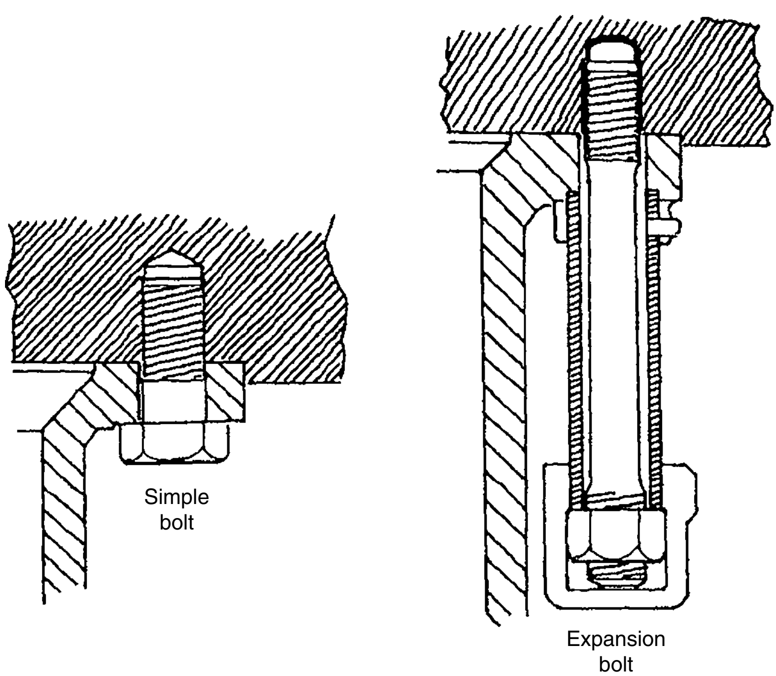


FIGURE 11.27 The elongated expansion bolt on the right replaced the conventional bolt on the left, to solve a severe differential expansion problem in a nuclear power application.

of the tools used in proximity to hot parts, etc. But a lubricant analysis should be a reasonable way to get a rough estimate.

Beyond that, keep track of the torques used (and the temperatures involved). Experience will show you whether those torques were "good" or "bad" for the next shutdown or repair.

11.5 JOINT EQUATIONS THAT INCLUDE THE EFFECTS OF ECCENTRICITY AND DIFFERENTIAL EXPANSION

11.5.1 The Equations

We can now extend the equations of Chapter 10 to include the effects of differential expansion and joint eccentricity (i.e., prying action). Remember that we're interested in two things: the maximum tensile load which must be supported by an individual bolt and the minimum clamping force we can expect to find, worst case, in the joint; both of these "in service." Remember, too, that the sign of the differential expansion force is not always the same. If the joint expands more than the bolts, it's positive; if the bolts expand more than the joint, however, there will be a loss of both clamping force and bolt tension, and the sign will be negative. With all that in mind, let's rewrite our equations using Φ_{en} as the load factor and $1/r_J''$ as the joint stiffness (which we'll call K_J''). Maximum bolt load (extending Equation 12.15):

$$\text{Max } F_B = (1+s)F_{Pa} - \Delta F_m - \Delta F_{EI} + \Phi_{en} L_X \pm \Phi_{en} K_J''(\Delta L_J - \Delta L_B) \tag{11.21}$$

Minimum per-bolt clamping force on the joint (extending Equation 10.16):

$$\text{Min } F_J = (1-s)F_{Pa} - \Delta F_m - \Delta F_{EI} - (1-\Phi_{en})L_X \pm \Phi_{en} K_J''(\Delta L_J - \Delta L_B) \tag{11.22}$$

Total minimum clamping force on a joint containing N bolts:

$$\text{Min total } F_J = N \times \text{per-bolt Min } F_J \tag{11.23}$$

In the three equations above,

- F_{Pa} = the average or target assembly preload (lb, N)
- ΔF_m = the change in preload created by embedment relaxation (lb, N); Equation 10.5 showed us that $\Delta F_m = e_m F_{Pa}$
- e_m = percentage of average, initial preload (F_{Pa}) lost as a result of embedment, expressed as a decimal
- ΔF_{EI} = the reduction in average, initial, assembly preload caused by elastic interactions (lb, N); Equation 10.6 told us that $\Delta F_{EI} = e_{EI} F_{Pa}$
- e_{EI} = the percentage of average, initial preload (F_{Pa}) lost as a result of elastic interactions, expressed as a decimal
- ΔL_B = the change in length of the grip length portion of a loose bolt created by a change of Δt (°F, °C) in temperature (in., mm); see Equation 11.14
- ΔL_J = the change in thickness of the joint members, before assembly, if exposed to the same Δt (in., mm); see Equation 11.15
- s = half the anticipated scatter in preload during assembly, expressed as a decimal fraction of the average preload; see Equation 10.1
- K_J'' = the stiffness of a joint in which both the axes of the bolts and the line of application of a tensile force are offset from the axis of gyration of the joint, and in which the

tensile load is applied along loading planes located within the joint members (lb/in., N/mm); K'_J is the reciprocal of the resilience of such a joint (see Figure 11.16)

Φ_{en} = the load factor for the joint whose stiffness is K'_J (see Figure 11.16)

N = number of bolts in the joint

Again, these equations can be rewritten using any appropriate combination of Φ and K_J. This completes our main bolted joint design equations.

11.5.2 An Example

Now let's run an example to practice using Equations 11.21 through 11.23, extending the example we worked out in the last chapter to include the effects of eccentricity and differential expansion. We'll assume the following values. Some of these are the same as those we used in Chapter 12; others are added to include eccentricity and expansion. This time, for example, we'll need to enter actual data for such things as bolt and joint dimensions and material strengths.

Assembly parameters:

s = tool scatter = 0.30
e_m = 0.1
e_{EI} = 0.18
F_{Pa} = target preload = 50% of bolt yield

Service conditions:

Operating temperature = 400°F
External load, $L_X = 0.25F_{Pa}$
n = decimal defining distance between loading planes = 0.5
The joint: dimensions and properties (see Figures 5.11 and 11.12)

Material: mild steel

E = 27.7×10^6 psi at 400°F
a = 0.4 in.
s = 0.2 in.
T = total joint thickness; also = grip length L_G = 1.0 in.
T_{min} = thickness of thinner joint member = 0.5 in.
b = distance between bolts = 0.75 in.
D_J = $W = 2 \times D_B$ = 1.0 in.
ρ_J = coefficient of expansion = $8.3 \times 10{\sim}^6$ in./in./°F
N = number of bolts in the joint = 8

Bolt dimensions and properties:

We'll use a 3/8–16 × 1½ Inconel 600 bolt, identical in dimensions to that used as an example of bolt stiffness in Section 5.2 (Figure 5.5).

A_S = tensile stress area of the threads = 0.0775 in.^{2} (Appendix F)
E = 30×10^6 psi at 400°F
K_B = 2.124×10^6 lb/in., as computed in Section 5.2
S_y = yield strength at 70°F = 37×10^3 psi
D_B = width across flats of the head of the bolt = 0.5 in.

$D_H = 0.4$ in.
ρ_B = coefficient of expansion = 9.35×10^{-6} in./in./°F

Now, using the data listed above, we need to compute several factors required for Equations 11.21 through 11.23. First let's compute the area of the joint (A_J) and of the equivalent cylinder of the joint (A_C). See Figures 5.10 and 5.11 for reference.
$A_J = b \times W = 0.75 \times 1.0 = 0.75$ in.^{2} (see Figure 5.11)

$$A_C = \frac{\pi}{4}\left(D_B^2 - D_H^2\right) + \frac{\pi}{8}\left(\frac{D_J}{D_B} - 1\right)\left(\frac{D_B T}{5} + \frac{T^2}{100}\right)$$

$$A_C = \frac{\pi}{4}\left(0.5^2 - 0.4^2\right) + \frac{\pi}{8}\left(\frac{1}{0.5} - 1\right)\left(\frac{0.5 \times 1}{5} + \frac{1^2}{100}\right) \tag{5.20}$$

$A_C = 0.11$ in.^{2}

Next we must compute the radius of gyration (R_G) for the rectangular area A_J, using Equation 5.26, where d = the length of the longer side (in this example $d = W$).

$R_G = 0.209d = 0.209(1.0) = 0.209$ in.
$R_G^2 = 0.0437$ in.^{2}

Next, we compute the stiffness (K_{JC}) and then the resilience (r_J) of the equivalent cylinder (refer Figure 5.10).

$$r_J = 1/K_{JC} = 0.328 \times 10^{-6}\text{in./lb}$$

Next we need the factor λ^2 (refer Figure 11.15)

$$\lambda^2 = \frac{s^2 A_C}{R_G^2 A_J} = \frac{0.2^2(0.11)}{0.0437(0.75)} = 0.134$$

Now we can compute resiliencies r_J', r_J'', r_B (reference Figure 11.15) and joint stiffness K_J''

$$r_J' = r_J\left(1 + \lambda^2\right) = 0.328 \times 10^{-6}(1 + 0.134) = 0.372 \times 10^{-6} \text{ in./lb}$$

$$r_J'' = r_J\left[1 + \left(\frac{a\lambda^2}{s}\right)\right] = 0.328 \times 10^{-6}\left[1 + \left(\frac{0.4 \times 0.134}{0.2}\right)\right]$$

$$r_J'' = 0.416 \times 10^{-6} \text{ in./lb}$$

$$r_B = \frac{1}{K_B} = \frac{1}{2.124 \times 10^6} = 0.471 \times 10^6 \text{ in./lb}$$

$$K_J'' = \frac{1}{r_J''} = 2.404 \times 10^6 \text{ lb/in.}$$

We can now compute load factor Φ_{en} (see Figure 11.16).

$$\Phi_{en} = n\left(\frac{r_J''}{r_J' + r_B}\right) = 0.5\frac{0.416 \times 10^{-6}}{(0.372 + 0.471) \times 10^{-6}}$$

$$\Phi_{en} = 0.247$$

Next we need to compute the changes in the length of bolt and joint when subjected to an increase in temperature of 330°F (from 70°F to 400°F).

$$\Delta L_J = \rho_J(L_G)\Delta t = 8.3 \times 10^{-6}(1.0)330 = 0.00274 \text{ in.}$$

$$\Delta L_B = \rho_B(L_G)\Delta t = 9.35 \times 10^{-6}(1.0)330 = 0.00309 \text{ in.}$$

Finally, let's select an assembly preload, which becomes our target or average preload (F_{Pa}). We said that we wanted F_{Pa} to equal 50% of the bolt's yield strength. That will be the room temperature yield strength because assembly is done at room temperature.

$$F_{Pa} = 0.5(S_y)A_S = 0.5(37 \times 10^3)0.0775 = 1.43 \times 10^3 \text{ lbs}$$

Now we're finally ready to compute the two things we're most interested in, the maximum force the bolts must be able to support (Max F_B) and the minimum clamping force on the joint ($N \times$ Min per-bolt F_J) using Equations 11.21 through 11.23. And let's compute each term in Equations 11.21 and 11.22 separately before combining them. It's instructive to see how much change in preload each factor contributes to the final result. Starting then with Equation 11.21:

Max assembly preload $= (1 + s)F_{Pa} = (1 + 0.3)1.43 \times 10^3 = 1.86 \times 10^3$ lbs
Embedment loss $= e_m = F_{Pa} = 0.1(1.43 \times 10^3) = 0.143 \times 10^3$ lbs
Average elastic interaction loss $= e_{EI}F_{Pa} = 0.18(1.43 \times 10^3) = 0.257 \times 10^3$ lbs
Increase in bolt tension caused by the external load $= \Phi_{en}L_X = \Phi_{en}(F_{Pa}/4) = 0.247$ $(1.43 \times 10^3/4) = 0.0883 \times 10^3$ lbs
Decrease in bolt tension caused by differential expansion $= \Phi_{en}K_J''(\Delta L_J - \Delta L_B) = 0.247$ $(2.404 \times 10^6)(0.00274 - 0.00309) = 0.208 \times 10^3$ lbs
We can now use Equation 11.21 to combine these to get the Max F_B in service: Max $F_B = 1.34 \times 10^3$ lbs

This shows us that, in this application, the maximum tension in the bolts will occur at room temperature, during assembly, on the most lubricious bolts (maximum positive tool scatter). This is a desirable situation; it means that if the bolts don't break during assembly they won't break in practise.

Now for minimum clamp force (Equation 11.22)
Minimum assembly preload $= (1 - s)F_{Pa} = 0.70(1.43 \times 10^3) = 1.00 \times 10^3$ lbs
Embedment and elastic interaction losses are the same as computed above.
Loss in clamp created by the external load $= (1 - \Phi_{en})L_X = (1 - \Phi_{en})(F_P/4) = (1 - 0.247)$ $(1.43/4)10^3 = 0.269 \times 10^3$ lbs.
Loss in clamp caused by differential expansion is the same as above.

We use Equation 11.22 to combine these and get the minimum per-bolt clamp force on the joint in service.

Min $F_J = 0.123 \times 10^3$ lbs
The total clamp force on the joint would be N times this, or $8 \times 0.123 \times 10^3$
Total Min $F_J = 0.984 \times 10^3$ lbs

Note that there's a 15:1 ratio between the maximum tensile force the bolts must be able to withstand and the minimum force we can count on for interface clamp.

$$\text{Ratio} = \frac{\text{Max } \mathrm{F_P}}{\text{Min per-bolt } F_J} = \frac{1.86}{0.123} = 15.1$$

Once again we see that inability to control assembly preload, elastic interactions, and embedment, plus the effects of external loads and differential expansion have forced a significant overdesign of the joint. We could improve results by using bolt and joint materials having similar coefficients of expansion or by using feedback assembly control to reduce tool scatter and relaxation losses; but the situation explored in this example is not uncommon and may be more economical than the improvements just listed.

Note that we've taken a very conservative view in running this example. Some of the factors we've dealt with—such as the effects of external load or of differential expansion—are presumably unavoidable. Each and every joint of this sort, used in the hypothetical service, will be exposed to those effects. But will each joint see a full $\pm 30\%$ scatter in the torque–preload relationship? I very much doubt it. If we had placed this joint in production we could assume that an occasional joint would see a full $+30\%$ or a full -30%; but I would suspect never both. Also, if this joint was to be used in a safety-related application we might assume both $\pm 30\%$ to be sure we had covered all possibilities. But if we were only concerned about a few joints, each containing only a few bolts, we could undoubtedly use less than $\pm 30\%$ for the anticipated scatter.

How much less? I'm afraid that I'll have to leave an accurate answer to the statisticians of this world. If the joint is not safety related, however, I would suggest that $\pm 10\%$ might be used if our bolts are new and as-received (unlubed) but were all obtained at the same time from the same source; and perhaps $\pm 5\%$ for new, single-source, lubed bolts. Higher figures would, of course, be used for old, reused bolts, especially if they show signs of handling or are slightly rusty or something. For design purposes, however, the full range illustrated in Figure 11.28 would apply.

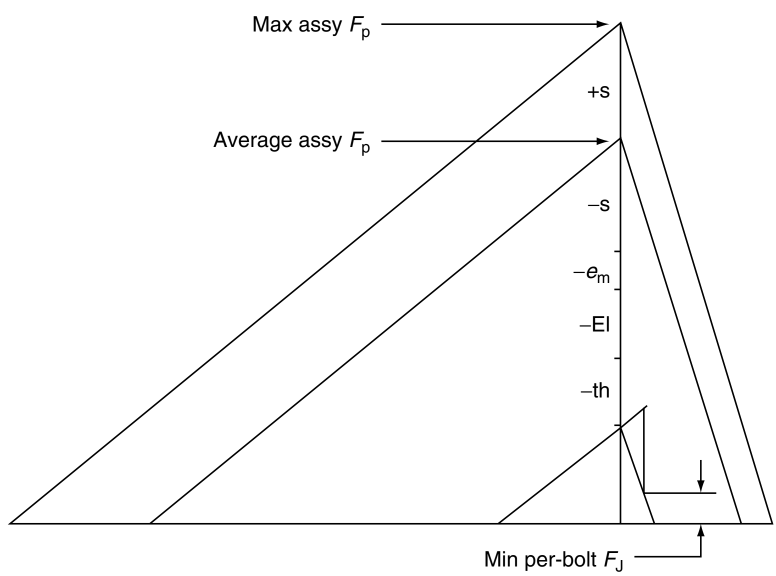


FIGURE 11.28 This joint diagram is based on the example given in Section 11.5. The maximum bolt load, in this case, is simply the average assembly preload, F_{pa}, plus the maximum anticipated scatter in the relationship between applied torque and achieved preload. The minimum per-bolt clamping force is the average assembly preload less the loss in preload caused by embedment relaxation (e_m), elastic interactions (EI), differential expansion (th), and external tensile load on the joint. The resulting maximum bolt load, which will define the size of the bolts, is 15 times the minimum per-bolt clamping force we can count on from those bolts.

EXERCISES AND PROBLEMS

1. Define prying action.
2. Sketch an alternative joint diagram showing the effects of prying action.
3. Name at least two things the joint designer can do to reduce prying action.
4. This and the following three questions are all in reference to a ductile iron joint whose width (D_J) = 1.625 in. Thickness (T) = 2.0 in. The joint is loaded by a single 3/8–20 × 2.5 J429 GR 5 bolt. Washers 0.625 in. in diameter are used under the head of the bolt and under the nut. The diameter of the bolt hole is 1/64 in. larger than the diameter of the body of the bolt. A tensile load (L_X) of 7.40 kips is applied to the joint along the axis of the bolt. What is the effective length (L_e) of this bolt?
5. What is the stiffness of the bolt and the stiffness of the joint?
6. What is the resilience of the bolt and of the joint members?
7. What is the stiffness ratio of this assembly?
8. What is the increase in bolt tension and the decrease in clamping force created by the external load?
9. Now consider a second joint of the same material, width and thickness as the joint in problem 4 above but eccentrically loaded. This time, however, the line of action of the external load is 0.563 in. removed from the centerline of the joint and the bolts are 0.2 in. removed from that centerline (i.e., the bolts are about half way between the centerline of the joint and the line of action of the external load). Assume that the bolt loads a square cross-section of the joint 1.625×1.625 in.^{2} in area and that the joint is loaded along a loading plane defined by $n = 0.7$. What is the radius of gyration (R_G) of this joint?
10. What is the resilience of the bolts and joint this time?
11. What is the stiffness ratio of this eccentric joint?
12. By how much does the tensile load of 7.40 kips increase the tension in the bolts and reduce the clamping force on this joint?
13. What is the minimum preload required to prevent separation of this eccentric joint?
14. The concentric joint of problem 4 is subjected to an increase in temperature of 100°F. Does this increase or decrease the tension in the bolts? By how much? (Assume that the coefficient of linear expansion of the bolts is 5.6×10^{-6} in./in./°F and that of the joint is 6.0×10^{-6} in./in./°F.)

REFERENCES

1. Fisher, J.W. and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, Wiley, New York, 1974, pp. 260ff.
2. Nair, R.S., P.C. Birkemoe, and W.H. Munse, High strength bolts subject to tension and prying, *Journal of the Structural Division*, *ASCE*, vol. 100, no. ST2, proc. paper 10373, pp. 351–372, February 1974.
3. Agerskov, H., Analysis of bolted connections subject to prying, *Journal of the Structural Division*, *ASCE*, vol. 103, no. ST11, November 1977.
4. Simon, M.W. and J.B. Hengehold, Finite element analysis of prying action in bolted joints, Paper no. 770545, Society of Automotive Engineers, Earth Moving Industry Conference, Central Illinois Section, Peoria, IL, April 18–20, 1977.
5. Junker, G., *Principles of the Calculation of High Duty Bolted Connections—Interpretation of the Guideline VDI2230*, VDI Berichte no. 220, 1974, an Unbrako technical thesis, published by SPS, Jenkintown, PA.
6. Finkelston, R.J. and P.W. Wallace, Advances in high performance fastening; Paper no. 800451, SAE, Warrendale, PA, February 25–29, 1980.
7. Grotewohl, A., *Calculation of a Dynamically and Eccentrically Loaded Bolted Connecting Rod Connection according to VDI 2230*, VDI Berichte no. 220, 1974.

8. Pindera, J.T. and Y. Sze, Response to loads of flat faced flanged connections and reliability of some design methods, University of Waterloo, Canada, *CSME Transactions*, March 1972.
9. Pindera, J.T. and Y. Sze, Influence of the bolt system on the response of the face-to-face flanged connections, Reprints of the 2nd International Conference on Structural Mechanics in Reactor Technology, vol. III, part G–H, Berlin, Germany, pp. 10–14, September 1973.
10. Rodabaugh, E.C., FLANGE, a computer program for the analysis of flanged joints with ring-type gaskets, ORNL-5035, U.S. Department of Commerce, National Technical Information Services, Springfield, VA, January 1985.
11. False Grade 8 engineering performance marks on bolting and improper marking of Grade 8 nuts, Research Report of the Industrial Fastener Institute, Cleveland, OH, April 4, 1986.
12. Halsey, N., J.R. Gieske, and L. Mordfin, Stress relaxation in aeronautical fasteners, National Bureau of Standards Report no. NBS-9485, February 1967.
13. Markovets, M.P., Graphic analytic method of calculating the time to failure of bolts under stress relaxation conditions, *Teploenergetika*, *NG*, pp. 52–54, June 1970; English translation, *Thermal Engineering*, vol. 17, no. 6, pp. 77–79, June 1970.
14. Mayer, K.H., Relaxation tests on high temperature screw fastenings, *Wire*, vol. 23, pp. 1–5, January–February 1973.
15. Valve studs—tightening, inspection and replacement recommendations (TIL-891), General Electric Company, Tampa, FL, December 1979.
16. Criteria for design of elevated temperature Class 1 components in Section III, Division 1, of the ASME Boiler and Pressure Vessel Code, ASME, New York, May 1976.
17. Kraus, H. and W. Rosenkrans, Creep of bolted flanged connections, *Welding Research Council Bulletin*, no. 294, Welding Research Council, New York, May 1984.
18. Sachs, K. and D.G. Evans, The relaxation of bolts at high temperatures, Report C364/73 on work done at the GKN Group Technological Center, Wolverhampton, U.K., September 6, 1973.
19. From information received from the manufacturer, Superbolt Inc., Carnegie, PA, 1988.
20. Recommended practices in elevated temperature design: a compendium of breeder reactor experiences (1970–1987), A.K. Dhalla, (Ed.), Published by the Pressure Vessel Research Committee of the Welding Research Council, New York, in *Welding Research Council Bulletin*, no. 362 (April 1991), no. 363 (May 1991), no. 365 (July 1991), and no. 366 (August 1991).
21. Grosse, I.R. and L.D. Mitchell, Nonlinear axial stiffness characteristics of axisymmetric bolted joint, available from the University of Massachusetts, Amherst, MA, or from the Virginia Polytechnic Institute and State University, Blacksburg, VA, 1988.
22. VDI2230, Systematic calculation of high duty bolted joints, issued by Verein Deutscher Ingenieure (VDI) VDI Richlinien, Dusseldorf, Germany, October 1977; translation by Language Services, Knoxville, TN, published by the U.S. Department of Energy as ONRL-tr-5055, p. 44.

12 In-Service Behavior of a Shear Joint

We've studied axial tension loads at length because they're always present (since we preload—tension—the bolts as we tighten them) and because they will often dominate the behavior of the joint even when other types of load are also present. Tension loads in general are also the most difficult to understand, furthermore, because of the intricate way in which bolt and joint share them.

Before leaving the subject of working loads, we should examine shear loading. This is also a common type of load, especially in structural steel joints; and it demands an entirely different type of joint analysis—and creates a different joint response—than does a tension load. In fact, as mentioned in the preface, it would require a second book to do justice to the subject of shear joints; I won't attempt to cover them here. Geoffrey Kulak, John Fisher, and John Struik have produced a well-written and comprehensive text on the subject, *Guide to Design Criteria for Bolted and Riveted Joints* [1,2], which I recommend.

Although we can't examine them at length, it's pertinent for us to take a brief look at shear joints—enough to see why they're different, and enough to know when to read another book!

12.1 BOLTED JOINTS LOADED IN AXIAL SHEAR

In a shear joint, the external loads are applied perpendicular to the axis of the bolt, as in Figure 12.1.

12.1.1 In General

A joint of this sort is called a shear joint because the external load tries to slide the joint members past each other or tries to shear the bolts. If the line of action of the external force runs through the centroid of the group of bolts, it's called an axial shear load, as shown in Figure 12.2.

The strength of such a joint depends on (1) the friction developed between the joint surfaces (called faying surfaces) and (2) the shearing strength of the bolts and plates. Until recently, joints loaded in shear were formally classified as either "friction-type" or "bearing-type" and were specified and analyzed accordingly. Laboratory studies were used to confirm design procedures. Now it is recognized, however, that, while it's possible to construct a pure friction-type or pure bearing-type joint in a laboratory, no such distinction exists in most field joints.

We'll look at the field situation a little later, and at contemporary shear joint classifications. First, though, I think that the original definitions are still a useful place to start learning about the design and behavior of shear joints. I also think that, although pure friction or bearing joints may be difficult or impossible to achieve in structural steel work, they are

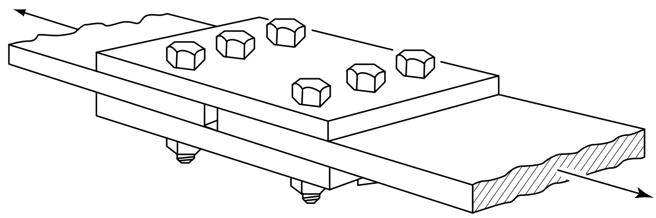


FIGURE 12.1 Bolted joint loaded in shear.

possible in nonstructural applications, the main requirement being the ability to control joint and hole geometry more closely than is possible in structural steel applications. So, let's start our review by looking at the classical friction-type and bearing-type joints. Then we'll return to the present structural steel viewpoint to see why that differs.

One additional comment about friction-type joints before we look at some details. Friction is never counted on in airframe joints subjected to shear loads, because the coefficient of friction is subject to too many variables. Such joints rely only on the shear strength of their fasteners [5].

12.1.2 Friction-Type Joints

The amount of friction between two mechanical parts is, of course, proportional to the normal force clamping the two parts together, and to the coefficient of friction at the interface. In a joint, the normal force is produced by the preload (axial tension load) in one or more bolts. The total friction force developed in the joint is called the "slip resistance" of the joint (RS). In Chapter 19 we'll see how to estimate the slip resistance of a joint. Suffice it to say at this point that it's generally a function of the clamping force between joint members and the inter-joint friction forces, which can be developed as a result.

12.1.2.1 Bolt Load in Friction-Type Joints

The bolt holes are always 1/16 in. or so larger than the bolt diameter in structural steel joints. Until slip occurs, therefore, there are no shearing forces on the bolt (see Figure 12.3). Under these conditions, the bolt is loaded by the pure axial tension created when the nuts were tightened. In other words, we're back to the original joint diagram—with zero external loads (Figure 12.4).

The elastic curve for the bolt will be that of a bolt under axial tension load—for example, the curve shown in Figure 3.1—even though the external load on the joint, in the present case, is perpendicular to the axis of the bolts.

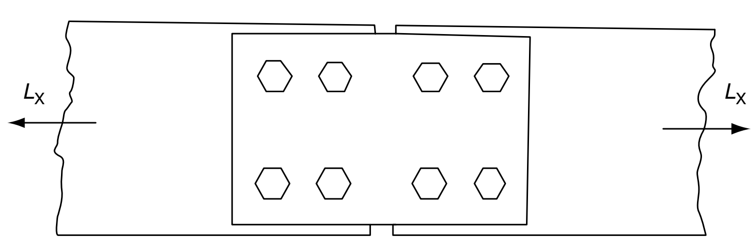


FIGURE 12.2 This is called an axial shear joint because the line of action of the external load passes through the centerline of the bolt pattern.

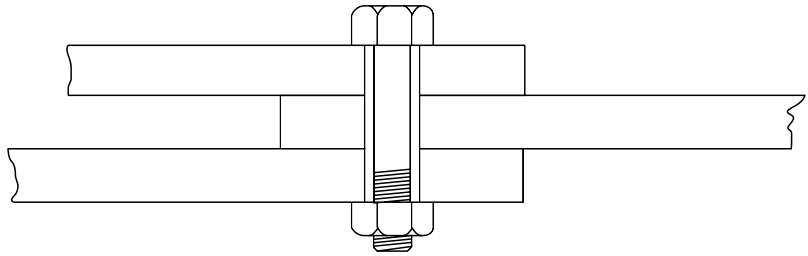


FIGURE 12.3 Any contact between bolts and joint members in a friction-type joint is accidental, so there is no shear stress on the bolts. The space between bolt and joint members is free.

The specifications recommend that the bolts in structural joints be set to a tension at least equal to the proof load of the bolt—i.e., approximately equal to the yield strength of the bolt. This is the minimum setting recommended. Through use of turn-of-nut procedures (described in Chapter 8), or tension-indicating fasteners (Chapter 9), most of the bolts are actually set well past this point; they are set into the plastic region of the elastic curve. This means, as we've seen, that every bolt is set to approximately the same tension. It follows, therefore, that until and unless slip occurs, all of the bolts in a friction-type joint are essentially loaded equally. This is not true in a bearing-type joint, as we'll see.

12.1.2.2 Stresses in Friction-Type Joints

As long as the joint doesn't slip, the tension in one set of plates is transferred to the others as if the joint were cut from a solid block. Lines of principal (tension) stress flow from one to the other without interruption (neglecting for the moment the local stress concentrations caused by the holes or by the clamping forces produced by individual bolts); see Figure 12.5.

12.1.3 Bearing-Type Joints

When the external loads rise high enough to slip a friction-type joint, the joint plates will move over each other until prevented from further motion by the bolts (Figure 12.6). This joint is now considered to be "in bearing"—or to be (for purposes of analysis) a "bearing-type joint." As mentioned earlier, some joints are designed to be in bearing from the start.

It's worth noting that the ultimate strength of all shear joints is determined by their strength in bearing—not by their frictional slip resistance. Friction-type joints, however, are often considered to have failed if they slip into bearing. Ultimate strength is rarely a good measure of the design strength or useful strength of a joint, any more than it is a good measure of the useful strength of a mechanical part.

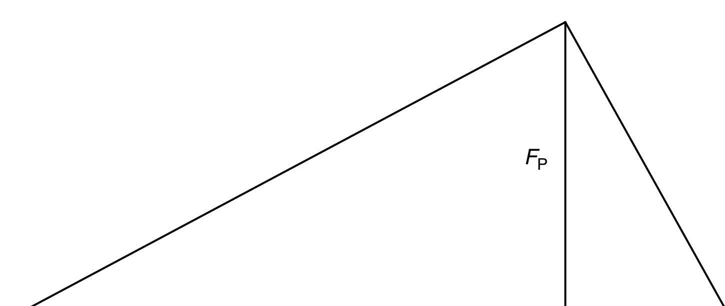


FIGURE 12.4 The only load on the bolts in a successful friction-type joint is the preload (F_P).

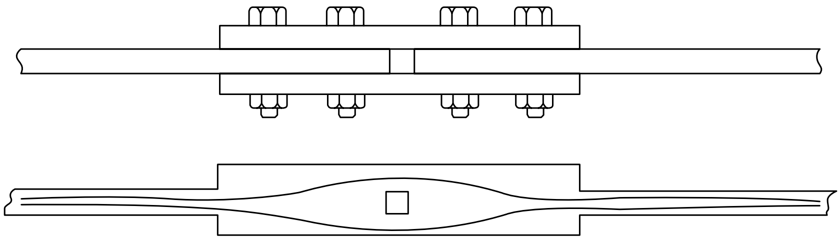


FIGURE 12.5 As long as there is no slip between joint members, a shear joint acts as if it were a solid block, with a smooth transfer of stress from one input member to the other.

12.1.3.1 Stresses in Bearing-Type Joints

The stress patterns in bearing-type joints are more complex than those in friction-type joints. The tension in one set of plates is transmitted to the others in concentrated bundles through the bolts (Figure 12.7). What's more, each row of bolts transmits a different amount of load—at least in so-called long joints (those having many rows of bolts) (Figure 12.8). The outermost fasteners always see the largest shear loads (see p. 93 in Ref. [1]; p. 12 in Ref. [3]).

We saw a similar phenomenon when studying the stresses in nut and bolt threads. Remember that the inboard (first engaged) threads saw the most load. The outboard ones saw the least, because the inboard had already transferred some of the bolt load to the nut. In the joint case, the outer rows of bolts transfer some of the load on one set of plates to the other plates, reducing the loads seen by both plates and bolts toward the center of the group of bolts.

As a result, the outer bolts in a joint see far more than the average stress level—some say as much as five times the average—while the inner ones see less than average. The outer bolts, therefore, usually are loaded plastically rather than elastically. This results in plastic flow, which helps to distribute the load more uniformly between the various rows of bolts. Before we leave the subject of the stresses in bearing-type joints, it's worth noting that since the bolts do bear on the joint, there are shear stress concentrations in the plates. These can cause local yielding of the plates (the bolt holes become slots), or the ends of the plate can tear out, as we'll see in our discussion of joint failures. These failures would not be seen in a friction-type joint unless it slipped.

12.2 FACTORS THAT AFFECT CLAMPING FORCE IN SHEAR JOINTS

As far as bolt preload and clamping force are concerned, shear joints are affected by all of the factors which affect those things in tensile joints.

Initial preloads will be scattered as a result of variations in geometry, lubricity, condition, etc. of the parts involved, as well as variations in tools, operators, procedures, and all the rest.

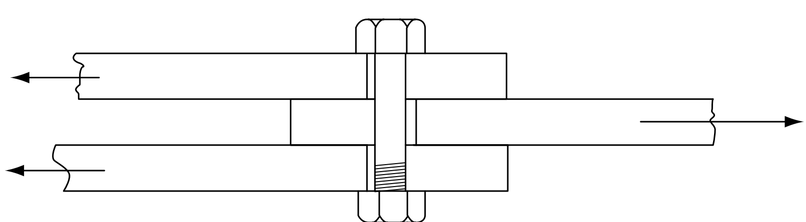


FIGURE 12.6 In a bearing-type joint the bolts act as shear pins. The spaces between bolt and joint members are offset.

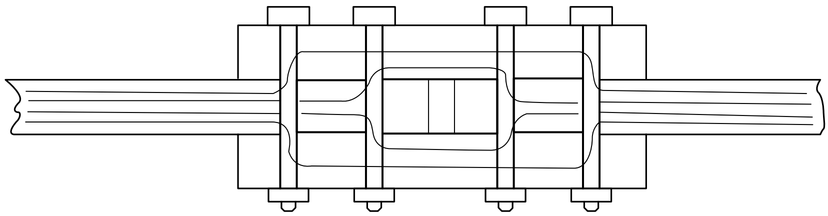


FIGURE 12.7 In a bearing-type joint the tension in one plate is transmitted to the other plate through the bolts, making for a more complex stress distribution than in a friction-type joint (compare with Figure 12.5).

The bolts and joint members embed in shear joints, just as they do in tensile joints. "Elastic interactions" occur as a group of bolts are tightened in a shear joint.

Preloads and clamping forces in a shear joint will be altered by differential expansion if the parts are made of different materials or are subjected to different temperatures.

As a result, we can use Equations 11.21 through 11.23 to estimate maximum bolt loads and minimum clamping forces; but with one important difference. The term $\Phi_{en}L_X$ should be omitted from Equation 11.21, and the term $(1-\Phi_{en})$ L_x should be omitted from Equation 11.22, because bolt loads and clamping forces will not be affected this way by external shear loads. Bolt-to-joint stiffness ratios have no influence on the way shear joints absorb loads, although they still affect the way such joints respond to temperature swings and the resulting differential expansion.

We can draw a joint diagram for such a joint. It would look exactly like the diagram of Figure 11.28 but without the external load effect shown at the bottom of that diagram. The minimum, residual clamping force on the joint in that diagram, Min per-bolt F_J, will be the clamp left after differential expansion (shown as "$-$th" in the diagram).

This is not to say, however, that shear joints don't respond to external loads. They do, but in ways that differ completely from the way their tensile cousins respond. Let's take a look.

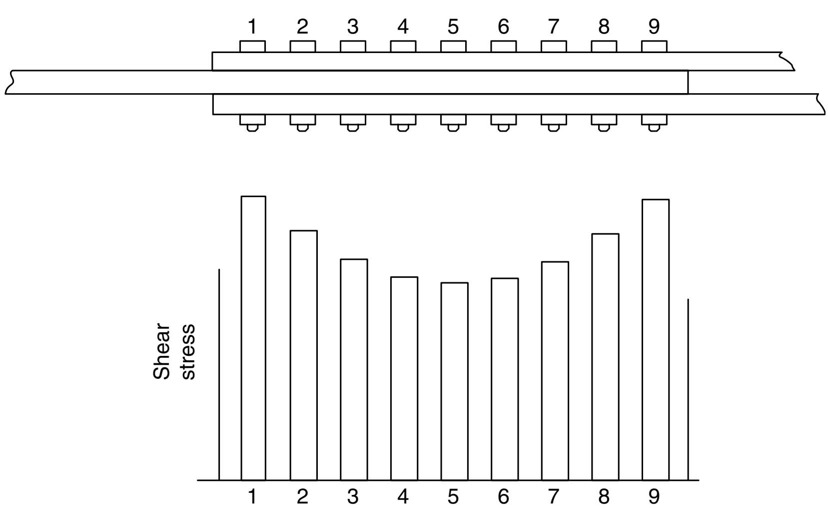


FIGURE 12.8 Shear stress in individual bolts varies substantially, especially in long joints. (Modified from Fisher, J.W. and Struik J.H.A., *Guide to Design Criteria for Bolted and Riveted Joints*, Wiley, New York, 1974.)

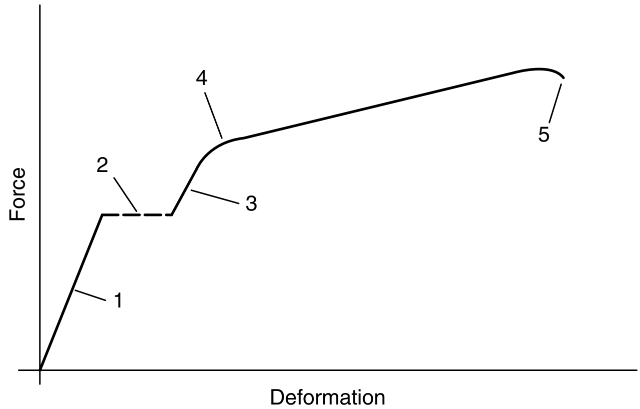


FIGURE 12.9 This chart illustrates the sequence by which a shear joint fails. See text for a discussion.

12.3 RESPONSE OF SHEAR JOINTS TO EXTERNAL LOADS

Figure 12.9 diagrams the way in which a shear joint responds to ever-increasing loads. The figure shows the overall deformation and displacement of joint members as a function of applied shear force. To start with (part 1 of Figure 12.9) there is linear, elastic deformation of joint members under relatively mild loads.

As applied force increases, the friction forces between joint members and the joint slips into bearing (part 2 in Figure 12.9).

Higher loads create more elastic deformation, this time of both bolts and joint members, as in part 3 of Figure 12.9. When these loads rise still further the parts start to deform plastically (part 4, Figure 12.9).

Finally—part 5 of Figure 12.9—something breaks. Either the bolts shear and the joint members are free to pull past each other, or the bolts tear out through the sides of the joint members.

Clamping forces and bolt stress are changed when shear loads are applied to the joint. Once the bolts have been brought into bearing, shear stress combines with the original tensile stresses created when we tightened the bolts to increase total stress in the bolts. Tensile stress doesn't change; but total stress does. This is true, at least, until the joint nears failure. Under extreme loads the parts have been much deformed, and bolts have probably been pulled sideways at an angle. Some of the shear load may now be seen as an increase in bolt tension. Some experiments indicate, however, that the bolts shed preload tension when abused this way. The bolts in shear joints fail, in shear, at about the same shear stress levels, regardless of how they were originally preloaded [2, p. 49].

None of this, however, can be usefully shown in a joint diagram. The diagram of Figure 10.4 still stands as our best representation of a joint loaded only in shear.

12.4 JOINTS LOADED IN BOTH SHEAR AND TENSION

There are joints that must support both shear and tensile loads in service. Such a joint is subject to all of the variables and potential problems faced by joints loaded solely in tension, as well as all of the problems faced by joints loaded only in shear. The full Equations 11.21 through 11.23 can be used to define the effect of the tensile load on bolt tension and clamp force. The tendency of the joint to slip into bearing can be based on the resulting Min per-bolt F_s.

The following equation can be used to determine how much shear stress the bolt can stand if subjected to a given tensile stress, or vice versa [1, p. 69; 3, p. 14; 2, p. 51].

$$\frac{S_T^2}{G^2} + T_T^2 \tag{12.1}$$

where

S_T = the ratio of shear stress in the shear plane(s) of the bolt to the ultimate tensile strength of the bolt

T_T = the ratio of the tensile stress in the bolt to the ultimate tensile strength of the bolt

G = the ratio of shear strength and tensile strength of the bolt (0.5–0.62 typically if computed on the thread stress area)

It is best to compute both S_T and T_T using the equivalent thread stress area formulas of Chapter 3 rather than the shank area.

If one plots the equation above for a given bolt, he will obtain an elliptic curve such as that sketched in Figure 12.10 [1, p. 54; 2, p. 51], which represents static loads on A325 or A354 BD bolts. Similar curves can also be drawn for dynamic loads.

Note that Equation 11.1 does not compute the allowable stress limits for a given bolt. It merely shows the relationship between tensile and shear stresses. However, a number of authors have plotted and published solutions of the equation for stress levels that would constitute failure—either plastic yield or fracture of the bolts in question. Use such curves with caution. Note that some are for static loads, some for cyclic loads; some assume that the

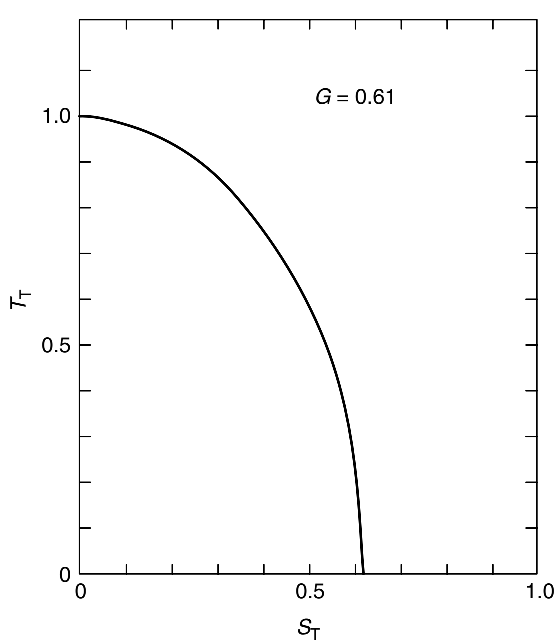


FIGURE 12.10 Elliptic curve used to relate the tensile capacity of a bolt to the shear stress imposed on a bolt, and vice versa. See Equation 12.1 for definition of the terms T_T, S_T, and G. This curve is for static loads on the thread stress area of ASTM A325 or A354 BD bolts. (Modified from Fisher, J.W. and Struik, J.H.A., *Guide to Design Criteria for Bolted and Riveted Joints*, Wiley, New York, 1974.)

shear planes will pass through the shank of the bolt, others assume that shear planes will pass through the threads, etc.

12.5 PRESENT DEFINITIONS—TYPES OF SHEAR JOINT

To this point in this chapter we've been using the descriptive but old-fashioned terms friction-type and bearing-type to analyze and discuss joints loaded in shear. The currently available "bolting spec" from the Research Council on Structural Connections (RCSC) however, no longer uses these terms, but instead defines three joint types as follows [4]:

1. Snug-tightened joints: those which resist shear loads only by shear bearing. They would be analyzed as bearing-type joints in the preceding discussion.
2. Pretensioned joints: those which resist shear loads only by shear bearing, but whose bolts are preloaded for reasons other than slip resistance. They would also be analyzed as bearing-type joints.
3. Slip-critical joints: those which resist shear loads with faying surface friction. They would obviously be analyzed as friction-type joints.

Snug-tightened joints are only used where slip will not affect the serviceability of the structure. They are not allowed for applications involving A490 bolts. Such joints may be exposed to static tensile loads, in or not in combination with shear loads, but may never be used if a joint will be subjected to varying tensile loads.

Pretensioned joints are required if called for by local code or spec, or if the joint will see significant load reversals. They are also required if the joint would be subjected to nonreversing fatigue loads, or if the A325 or F1852 bolts are subjected to fatigue loads or if A490 bolts are subjected to tension or combined shear and tension loads. The American Institute of Steel Construction (AISC) Load and Resistance Factor Design (LRFD) specification of 1999 says that bolts in bearing connections must be pretensioned if they are found in column splices with significant height to width ratios, or they are in members which brace the columns of tall buildings, or they are in connections in buildings having cranes of over five tone capacity, or in connections for supports of running machinery or other sources of shock and impact.

Slip-critical joints are required when slip could affect the serviceability, strength, performance, or stability of a structure, and when a joint will be subjected to load reversals and fatigue loads. They're also required if a joint member has oversized or slotted bolt holes, unless the line of action of a shear load is nearly perpendicular to the axis of the slots.

EXERCISES

1. Define a shear joint.
2. Define an axial shear joint.
3. Define faying surface.
4. Define friction-type joint.
5. Define bearing-type joint.
6. Draw a graph illustrating the way a shear joint fails.
7. Is bolt preload important in a shear joint? If so, why? If not, why not?
8. Should a slip-critical joint be analyzed as a friction-type or bearing-type joint?
9. Describe some of the applications in which slip-critical joints are mandated.
10. What is the difference between a pretensioned and a slip-critical joint?
11. Under what conditions is it acceptable to only snug-tighten the bolts in a joint?

REFERENCES

1. Fisher, J.W. and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, Wiley, New York, 1974.
2. Kulak, G.L., J.W. Fisher, and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, 2nd ed., Wiley, New York, 1987.
3. *High Strength Bolting for Structural Joints*, Booklet 2867, Bethlehem Steel Company, Bethlehem, PA.
4. *Specification for Structural Joints Using ASTM A325 or A490 Bolts*, June 30, 2004, Research Council on Structural Connections.
5. Barrett, R.T., Design of joints loaded in shear, *Handbook of Bolts and Bolted Joints*, Bickford J.H. and Nassar S. (Eds.), Marcel Dekker, New York, 1998.

13 Introduction to Joint Failure

One of our main goals in studying the design and behavior of bolted joints is to avoid joint failure by proper design and through the use of effective assembly techniques. Now that we have studied in some detail the assembly process and the effects of external loads on the joint, we are ready to turn our attention to the topic of joint failure.

As we learned in Chapter 1, the main function of a bolt is to clamp two or more joint members together. The main function of the assembly process is to introduce the tension in the bolts that produces the clamping force. Joint failure occurs when the bolts fail to perform their clamping function properly; for example, if they exert too high a force on the joint.

More commonly, as we will see, joint failure will occur if the bolts provide too little clamping force. In most such situations, the clamping force will probably be insufficient because of deficiencies in the assembly process. Remember that most of the factors included in the block diagram of Figure 6.28 would result in less than anticipated clamping force rather than in excessive force.

In other cases, however, insufficient clamping force can result from some form of instability in bolt tension. In Chapter 2, for example, we looked at some of the material properties which could lead to instability in service; and in Chapter 11, we considered changes caused by variations in temperature. We will be taking a look at other forms of instability in this and in the next few chapters.

We are going to start with an overview of the whole subject of joint failure, then go on to a closer study of the four principal types of failure: self-loosening, fatigue, corrosion, and leakage. But first, lets examine some of the ways in which a joint or bolt can fail.

13.1 MECHANICAL FAILURE OF BOLTS

Obviously, bolts will fail to exert sufficient clamping force on a joint if they are broken. They can break for a variety of reasons.

- Mechanical failure during assembly (the mechanic pulled too hard on the wrench! or the bolts weren't up to par)
- Mechanical failure at elevated temperatures (bolt strength dropped as temperature rose)
- Corrosion ate through the bolt
- Stress corrosion cracking
- Fatigue failure

We will look at corrosion, stress corrosion cracking (SCC), and fatigue in Chapters 15 and 16. These are relatively common causes of bolt failure. Mechanical failure during assembly or in service is much less common but is certainly not unknown. A few years ago, in fact, there was an increase in the number of elevated temperature/mechanical failures reported because of the fact that manufacturers of low-cost bolts had been using boron steel instead of medium carbon steel for Grade 8 fasteners, and boron steel loses strength more rapidly than carbon steel does as the temperature is raised.

There were also been recent reports that heads had snapped off low-cost bolts when they were tightened because of such things as improper heat treat (e.g., creating quench cracks), small fillet radii, or poor material. In a few cases, it was even found that suppliers had welded hexagonal heads onto threaded rod to manufacture the bolts.

If you use the proper material, maintain bolt quality, and dimension of the bolts to support the intended loads (see Chapter 3), your bolts should not break because of mechanical failure. They still may break because of corrosion, stress corrosion, or fatigue, however, as we will see in Chapters 15 and 16.

13.2 MISSING BOLTS

It's also obvious that bolts won't perform their clamping function properly if they're missing. Perhaps the most common reason for missing bolts is a phenomenon called self-loosening, which we will look at in detail in Chapter 14. Self-loosening is most commonly caused by vibration, but can also be caused by such things as temperature or pressure cycles. Anything, in fact, which puts reversing loads on the joint in a direction at right angles to the axis of the bolts may cause the bolts to loosen.

Self-loosening isn't the only cause of missing bolts, however. In a surprising number of situations, the bolts are missing because the mechanic didn't install them. In some cases, involving large, heavy equipment, some of the bolts were not installed because of hole misalignment or the like. In other cases, only carelessness was involved. Because most bolted joints are grossly overdesigned, most of them can get by with a few missing bolts, but obviously, we don't want to do this in critical situations.

13.3 LOOSE BOLTS

Bolts that aren't tight enough, i.e., bolts improperly preloaded, are probably the most common cause of joint misbehavior and failure at the present time. Broken and missing bolts could be considered an extreme form of "loose" as far as failure analysis is concerned. Any one of these three problems can lead to such failure modes as:

1. Joint leakage
2. Joint slip
3. Cramping of machine members (for example, bearings can get out of line)
4. Fatigue failure
5. Self-loosening

The relationship between loose bolts and self-loosening is a chicken-and-egg proposition. If the bolts are too loose to start with (were improperly tightened during assembly, for example), this will encourage self-loosening. Self-loosening, on the other hand, will progressively loosen the bolts. It's often difficult to tell, therefore, which was the cause and which the effect when one is analyzing a joint failure.

The relationship between loose bolts and fatigue requires some explanation, which you'll find in Chapter 15.

Gasketed joints can leak if their bolts are not properly tightened. This subject is discussed at length in Volume 2 of this edition.

13.4 BOLTS TOO TIGHT

It's less obvious and less common, but bolts that are too tight can also contribute to joint failure. Excessive bolt loads can crush gaskets, for example, or damage (gall) joint surfaces.

Excessive bolt loads can also encourage SCC, as we will see in Chapter 16, or can reduce fatigue life, as we will see in Chapter 15.

We learned a minute ago that insufficient preload (loose bolts) can encourage fatigue failure; now, we learn that too much can be a problem as well. Fatigue is one of those problems that can only be avoided by just the right amount of tension in the bolts, at least according to some experts. We will learn more about this in Chapter 15.

13.5 WHICH FAILURE MODES MUST WE WORRY ABOUT?

Which failure modes must the bolting engineer worry about? The answer depends on the job being performed by the bolted joints, on the consequences of failure, on the environment the bolts are working in, and usually on the industry. The petrochemical industry, for example, is primarily concerned with leakage from gasketed joints and with corrosion problems. Fatigue and vibration loosening are usually of little concern.

The automotive industry, on the other hand, would probably name self-loosening and corrosion as the two main problems, but leakage from head gaskets is sometimes a concern. Bolt fatigue is presumably a relatively minor issue.

The primary concerns of the structural steel industry are joint slip and corrosion. Fatigue is sometimes an issue, but usually of joint members rather than of the bolts. Self-loosening and leakage are never encountered. The aerospace industry would probably list fatigue first.

13.6 CONCEPT OF ESSENTIAL CONDITIONS

I think it's useful to recognize that each type of failure is set up by a limited number of "essential conditions," usually three or four in number. For example, to have a corrosion problem, you must have:

- An anode
- A cathode
- An electrolyte
- A metallic connection between anode and cathode

If you can eliminate any one of these essential conditions, you can completely eliminate the corrosion problem.

There are also essential conditions for the other types of failure we have discussed, as listed below.

SCC requires:

- A susceptible material
- Stress levels above a threshold
- An electrolyte
- An initial flaw

Hydrogen embrittlement requires the same conditions as SCC, but with hydrogen instead of an electrolyte.

Fatigue failure requires:

- Cyclic tensile stress
- A susceptible material
- Stress levels above an endurance limit
- An initial flaw

Mechanical failure requires:

- Stress levels exceeding the static strength of the bolts or threads

Self-loosening of the fastener requires:

- Cyclic loads at right angles to the bolt axis
- Relative motion (slip) between nut, bolt, and joint members

The fact that the essential conditions are limited makes it appear that it would be relatively easy to avoid joint failure. The problem, however, is that dozens—maybe even hundreds—of secondary conditions can establish the essential conditions required for a particular type of failure. We'll look at some of these secondary conditions when we study corrosion, fatigue, self-loosening, etc. in detail in subsequent chapters. To give you an example of the diversity of secondary conditions, however, a few years ago the Nuclear Regulatory Commission (NRC) became concerned about the growing number of reports they were receiving from nuclear operators concerning bolt problems on safety-related joints. No joint failures had been reported, only the failure of individual bolts. These failures included loose bolts, missing bolts, broken bolts, corroded bolts, etc. [1].

Studies were made to assess the extent of the problems and to reduce or prevent them. Tabulations were made of the factors, which had contributed to the potential problems reported. These factors are what I have called secondary conditions, above.

Remember that, earlier, we said that the only essential condition for mechanical failure was stress levels exceeding static strength. A total of 170 safety-related, mechanical failures of bolts were reported to the NRC over a 3 year period. The following secondary conditions were reported as possible causes for these failures:

- Bolt material not as specified
- Poor choice of material by the designer
- Improper heat treat (including quench cracks)
- Excessive preload
- Shear, bending, and torsion stress
- Creep damage
- Abnormal loads (water hammer, seismic shock, etc.)
- Poor fastener dimensions (e.g., poor thread fit)
- Elevated temperatures
- Construction procedures

It's easy to think of additional things which might cause mechanical failure of bolts. We've already discussed some of these things. The point is that the single "essential condition" we've defined for mechanical failure can be set up by a large number of problems behind the problem. In many cases, in fact, it is not at all obvious what conditions have led to the failure of the joint or what we could do to prevent a recurrence of the problem. In many cases, we can't afford to perform the basic metallurgical, chemical, or analytical work required to reach correct answers to the questions "How did it fail, and why?"

In many cases, furthermore, altering the conditions that lead to one problem can merely create conditions for another. For example, we know that more preload helps fight vibration or other forms of self-loosening. Excessive bolt stress, however, can encourage SCC. If we are concerned about both problems in a given application, then we're going to be forced to produce exactly the right amount of preload in those bolts.

13.7 IMPORTANCE OF CORRECT PRELOAD

This is not the only situation in which correct preload is useful, incidentally. A study of the essential conditions of the various failure modes shows that improper preload can be a contributing factor in almost every situation. Here's a tabulation of the relationship between failure mode and preload (which includes, in this discussion, its equivalents: bolt tension and clamping force).

13.7.1 Corrosion

Higher stress levels (higher preload) can make a material more anodic, more active in a corrosive environment. Insufficient preload can allow electrolytes to leak from pressure vessels and piping systems, exposing the bolts to corrosion attack.

13.7.2 Stress Corrosion Cracking

Excessive preload can raise stress levels above the SCC threshold. Insufficient preload can again allow leakage of corrosive materials.

13.7.3 Fatigue Failure

Excessive preloads can raise stress levels above endurance limits. Excessive preloads can mean an unnecessarily high mean stress in the material. Insufficient preload can increase the stress excursions seen by the parts.

13.7.4 Mechanical Failure

Excessive preload can add to subsequent service loads, thereby exceeding the strength of the fastener in service. Insufficient preload can expose the fastener to the full extent of the external load (see Chapter 9 on joint diagrams).

13.7.5 Self-Loosening of Fastener

Insufficient preload can allow transverse slip of the bolt and joint members, an essential condition for self-loosening. A major weapon against joint failure, therefore, is the correct clamping force on the joint. This, in turn of course, depends on the correct preload during assembly and then stability of bolt tension and clamping force in service.

13.7.6 Leakage

As mentioned earlier, insufficient preload can also lead to leakage of sometimes dangerous or expensive gases and fluids.

13.8 LOAD INTENSIFIERS

A large number of factors can contribute to the failure of bolted joints. In fact, this whole book could be called a discussion of such factors. It's worthwhile listing some of them here, however, to emphasize their effect on joint integrity.

Parts fail when subjected to loads which exceed their strength. That seems straightforward enough. Unfortunately it's often difficult or impossible to predict or control the loads. All sorts of things can make the actual loads worse than we thought they'd be. Other factors can increase the stress levels created by a given load; as far as the fastener is concerned, the load itself has increased.

In Chapter 11, for example, we saw that bolt loads are increased by prying action if the bolt and external load are not coaxial; and that the problem is magnified if joint members aren't stiff enough. Someone has discovered all this and opened our eyes to it, which is lucky, because this particular load intensifier is anything but obvious.

Table 13.1 lists some other things which can increase (intensify) the stress levels within the loaded bolt, and thereby increase the chances of failure. After each, I've listed the chapter in which you'll find more information.

The last two items in Table 13.1 require a brief explanation. "Poor fits" increase stress levels because they reduce the contact areas between nut and bolt threads, or between nut and joint members (e.g., if the bolt hole is oversize) and, therefore, increase contact pressures. The resulting stresses can exceed those anticipated by the designer.

Nonuniform preload in a group of bolts can cause a few of them to carry more than their share of the total load placed on the system.

13.9 FAILURE OF JOINT MEMBERS

We've said that a joint fails when the bolts fail to provide a suitable clamping force, and we've looked at several ways in which this may happen. What about failure—rupture—of the joint members themselves?

This is uncommon in joints loaded in tension. Sometimes you'll encounter failure in the neck of a piping flange, but the flanges themselves rarely crack. Automotive or other castings can crack. But these failures are seldom, in my limited experience at least, related to the bolting. Failure of joint members loaded in shear, however, is more common.

We took a brief look at some of the failure modes of joint members under noncyclic shear loads in Chapter 3, for example, in Figure 3.18. If the bolt holes are too closely spaced,

TABLE 13.1
Factors Which Increase the Stress Levels Produced in Bolts or Joint by a Given External Load

Factor	Reference Chapters
Prying action	11
Eccentric loads	19
Bending	3
Improper bolt/joint stiffness ratio	10,11,15
Shock or impact	10
Gaskets	5
Perpendicularity of threads and hole-to-joint surfaces	3
Poor fits (e.g., bolt to nut)	6
Nonuniform preload	6

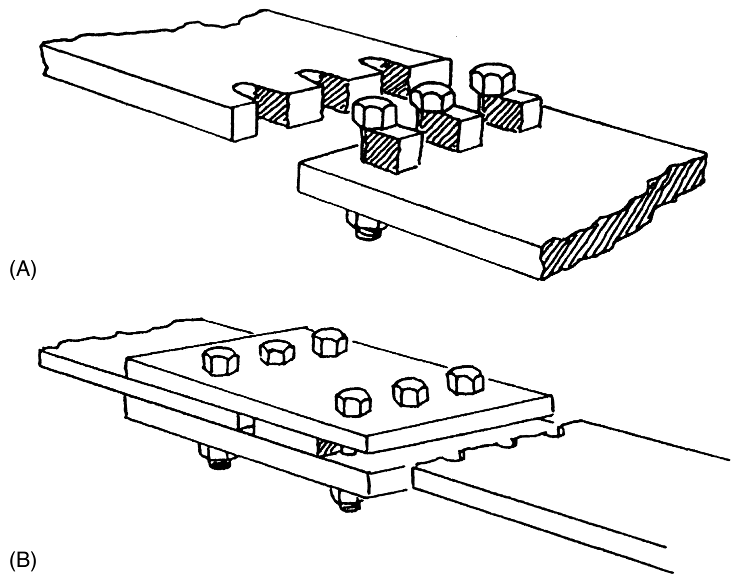


FIGURE 13.1 Some static failure modes of axial shear joints. (A) Tearout or marginal failure. (B) Failure through the "net section".

the joint can tear through the so-called net section, as in Figure 13.1B. If there are only a few bolts, and they're placed too near the edge of a plate, the bolts can pull their way through the plate, shearing it, as in Figure 13.1A.

It won't always be the plates which fail, however. In short joints with widely spaced holes located well back from the free edge, failure can consist of the simultaneous shearing of all bolts. It will be the bolts which fail in long joints, too, but in a different way.

Here, many rows of bolts are involved, and there is substantial frictional restraint between joint members, even if the joints are not slip-critical and the bolts have not been heavily preloaded.

The outermost bolts transfer the largest loads from plate to plate, and therefore see the largest loads (see Figure 12.8). As the loads on the joint increase, relative slip between joint members occurs first at the outer ends of the members. They're stretching. This act will distort the outer bolt holes and will eventually shear those bolts. Failure of these bolts can occur before the innermost bolts have suffered at all. Figure 13.2 shows such a joint at this point.

Under still higher loads, the remaining frictional restraint between joint members, in the center of the bolt pattern, will be overcome, and the rest of the bolts will shear [2].

All of this applies to joints under noncyclical loads. Fatigue loads lead to other types of failure, as we'll see in Chapter 15.

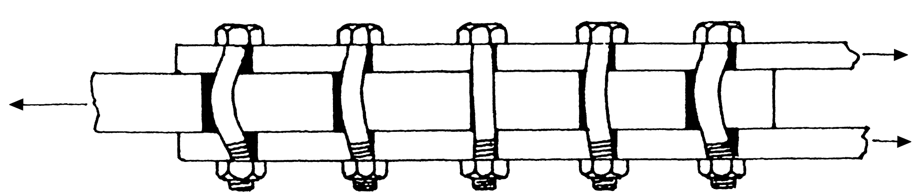


FIGURE 13.2 In long joints the bolts will often fail first, starting with the outermost ones because these see the greatest loads. (See Figure 12.8.)

13.10 GALLING

13.10.1 Discussion

We have now looked briefly at various ways in which a bolted joint can fail—in most cases because the bolts have failed to clamp the joint members together properly. Before going on to a detailed look at some of the more important failure mechanisms, I think we should take a brief look at another type of bolt failure which, while not affecting the performance of the joint in service, can be a real nuisance and expense. I'm referring to the galling of some bolts—especially larger ones—as they are tightened or removed during assembly or maintenance operations.

When the surfaces of male and female threads come in close contact under high contact stress, an atomic bond can form between them. If we try to loosen a bolt under these conditions the thread surfaces can tear and gouge each other. They are said to be galled. Galling is encouraged by such things as lack of lubrication, lack of oxide film on the metal, high contact pressure, and heat. Stainless steel bolts are particularly likely to gall.

Minor galling can cause minor damage to thread surfaces, but these can often be chased with a tap or die; the bolts can be reused. Major galling, however, can prevent removal of the bolt or nut. If this happens, the bolts have to be drilled out; the nuts have to be cut apart with a nut splitter, or, if they're too large for that, burned off with a torch.

There is no foolproof answer to galling. The following techniques, however, have worked for some people. These tips are listed in no particular order.

- Use coarse threads instead of fine; use a Class 2A fit.
- Use a good thread lubricant or antiseize compound. Here are some popular choices:
  - Moly disulfide works well if stresses are below 50% of yield and temperatures are below 750°F (400°C).
  - FelPro C670 lubricant is popular—again, at temperatures below 750°F (400°C).
  - Silver-based lubricants or antiseize compounds are especially effective.
  - Milk of magnesia has been found to be effective in high-temperature petrochemical and refinery operations.
  - Silicon grease works well on stainless steel bolts tightened into aluminum blocks.
  - Liquid dish detergent is said to work well if bolts are tightened into aluminum.
- Better Material Combinations:
  - Use stainless steel nuts on low-alloy bolts (for example, on A193 B7, B16, etc.).
  - Cold-drawn 316 stainless steel nuts work well on cold-drawn 316 bolts.
  - 400 series stainless steel nuts work well on 316 series bolts.
  - Grade 2 (pure) titanium fasteners work better than the higher-strength grades of titanium.
  - ARMCO Nitronic 60 bolts work well with Nitronic 50 nuts—but not the other way around, for some reason.
  - Carpenter Gall-Tough is a recently introduced austenitic stainless steel, which the manufacturer claims provides good galling protection.

13.10.2 Removing Galled Studs

If all else fails and the studs gall and you have to remove them, you might try some of the following tricks.

If galling is only minor, or if the studs have been exposed to high-temperature service for a long time, and you are sure that the original lubricants have dried out and are concerned about galling, you might try applying a good penetrating oil. These oils are usually slightly acidic. A product called Masteroil is popular.

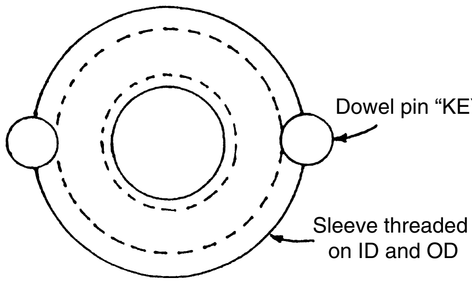


FIGURE 13.3 I galling forces us to drill out a bolt, the hole in the joint can sometimes be repaired by drilling and tapping a larger hole, then inserting a collar threaded on the ID to accept a new bolt of the original diameter. Dowel pins can be used as keys to prevent the collar from rotating.

Iodine, another mild acid, has worked well for some.

In some piping situations, people have removed the entire flange from the system and soaked it in a mild acid bath.

Note that, after removing a stud on which you have used these mild acids, you should clean and relubricate the studs as soon as possible to prevent unnecessary acid attack.

If the oils don't work, rapid heating and cooling of the flange or bolt—but not both—will sometimes work. Some people have gun-drilled a hole through a stud, then filled the hole with weldment to heat it suddenly.

If one of the problems is getting a good enough grip on a stud, apply penetrating oil and then weld a nut onto the end of the stud. Use the nut like the head of the bolt to apply more torque.

If the nut or stud has been threaded into a blind hole and you can reach the bottom of the hole (by drilling through the side of the flange, or block, or vessel, or whatever), drill such a hole, then tap it for a pipe fitting. Pump penetrating oil under relatively high pressure—2000 psi has been suggested—into the blind hole, wait for that oil to start to appear on the surface of the joint, then remove the bolt or stud.

If all else fails and the bolts must be destroyed, drill them out using a magnetic hold-down drill. EDM can also be used to drill them out, but the EDM process leaves a hard-surfaced hole, which is more difficult to retap. After removing the bolts or studs, drill the joint members for a collar, thread it on both ID and OD (the ID, of course, is tapped for the original bolts or studs), thread the collar into the drilled-out joint member, drill a couple of small holes, and insert some pins to retain the collar (as shown in Figure 13.3), then replace the bolts [3].

EXERCISES

1. Name at least three reasons why some bolts fail mechanically.
2. What is the most common cause of bolted joint misbehavior or failure?
3. Name at least two problems that can be caused by overtightening the bolts.
4. What do we mean by the term essential conditions?
5. Why do we often want to create the correct preload in the bolts when we assemble a joint?
6. Under what conditions is correct preload relatively unimportant?
7. Name at least three factors that can increase the stress in previously tightened bolts.
8. Mechanical failure of joint members is most often encountered in what type of joint?
9. Define galling.

REFERENCES

1. Bickford, J.H., Nuclear bolting issues and programs, *Pressure Vessel Components Design and Analysis—PVP*, Vol. 98-2, American Society of Mechanical Engineers, New York, 1985.
2. Kulak, G.L., J.W. Fisher, and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, 2nd ed., Wiley, New York, 1987, pp. 89ff.
3. From a conversation with Richard Potter of Bolting Services, Inc., Houston, TX, 1985.

14 Self-Loosening

14.1 THE PROBLEM

When we tighten a fastener, we pump energy into it: tension, torsion, and bending energy. The fastener is a stiff spring, and we stretch, twist, and bend it.

After we let go, this energy is held in the fastener by friction constraints in the threads or between contact faces of the nut and joint. If something overcomes or destroys these friction forces, the energy stored in the fastener will be released; the bolt will return to its original length with the inclined plane of the bolt threads pushing the inclined plane of the nut threads out of the way.

Subjecting the bolted joint to vibration will do this. Under certain circumstances, all preload in the fastener will be lost as a result. In fact, the fastener itself can shake loose and be lost. This can be a severe problem for any product that is bounced around or handled a lot—anything from a vehicle to a toy. Losing all preload or losing the fastener can, of course, lead to all sorts of other failures we would rather avoid. So it's useful to know what causes vibration loosening and some of the things we can do to minimize or prevent it.

Note that vibration loosening is a common cause of what we have called "clamping force instability." That force can be significantly reduced, or lost altogether, as a result of vibration.

It can also be lost by other forms of self-loosening, incidentally. Vibration may be the most common, but transverse slip, flexing of joint members, thermal cycles, and other things can also cause a joint to loosen. The self-loosening mechanism is the same in each case, however. We'll assume that vibration is the culprit and focus on it for now. For further information see the sources listed in Table 14.1.

14.2 HOW DOES A NUT SELF-LOOSEN?

We probably don't know why a fastener will self-loosen under vibration, shock, thermal cycles, or the like. A number of theories have been advanced, and their authors believe they know, but the theories vary [1,2,12,23–26,29]. They can't all be right—perhaps none are. A few years ago an ASME committee attempted to establish a working group to resolve the question. It was decided that a substantial amount of money would be required to finance the necessary research; but the attempts to attract financial support drew a blank. The project was, therefore, abandoned.

We'll examine one of the current theories in detail, and will briefly review one of the others. Before we do that, let's take a look at some basic factors which form a part, at least, of most or all of the existing theories.

Everyone agrees that a threaded fastener will not loosen unless the friction forces existing between male and female threads are either reduced or eliminated by some external mechanism acting on the bolt and joint. The disagreements concern the type of mechanism that does that. Before looking at these mechanisms, let's examine these all important friction forces we're trying to preserve.

TABLE 14.1
Web Sites or Names Providing Useful Information about Vibration-Resistant Fasteners

Spiralock	Nord-Lock
ESNA	Plate spring washer
SPS Technologies	NAS 3350
Stage8.com	NAS 3354
Loctite	Durlock
Detroit Tool Industries	Unbrako.com
MacLean	ThomasNet.com
Gripco Prevailing Torque Nuts	Longlok.com
Emhart.com	Omni-Lok Fasteners
LockBolt	Nylock.com
Huck Fasteners	Self-loosening of fasteners
Vibration-resistant fasteners	

They are created, of course, by the preload or tension in the bolt, which creates a "normal force" between male and female threads. The nut will turn with respect to the bolt only if some "antifriction force" (or torque) exceeds the thread-to-thread friction force.

Note that it is not necessary for the antifriction forces to be in the same direction as the forces which tend to loosen the nut. Let's place a block on a table as in Figure 14.1, for example. We want to move the block from point A to point B by exerting a force on it: force 1. The amount of force required, of course, is equal to that required to overcome the frictional force between block and table μW (where μ is the coefficient of friction and W is the weight of the block) plus the small additional force required to accelerate the block (to get it moving).

If someone else were to apply a second force to the block in any other direction—for example, force 2 in Figure 14.1—and if this second force were enough to overcome the frictional restraint between block and table, then we would only have to overcome the inertia of the block in order to move it from point A to point B.

As an example, it has been pointed out that it is easier to pull the cork from a wine bottle if one first rotates the cork to break the friction forces, then pulls the cork out. A straight pull must overcome both the friction forces and any suction forces, and is more difficult [1].

The point of all this is that vibration forces in direction 2 would allow some other low-level force to move a relatively heavy block across the surface of a table—if those direction 2 vibration forces were large enough to break most or all frictional restraint between block and table.

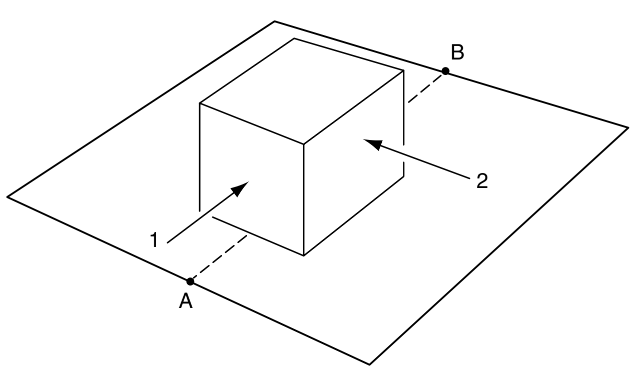


FIGURE 14.1 If force 2 is large enough to overcome any frictional force which exists between the block and the table, then only a very small force 1 would be required to move the block from point A to point B. Force 1 would only have to overcome the inertia of the block.

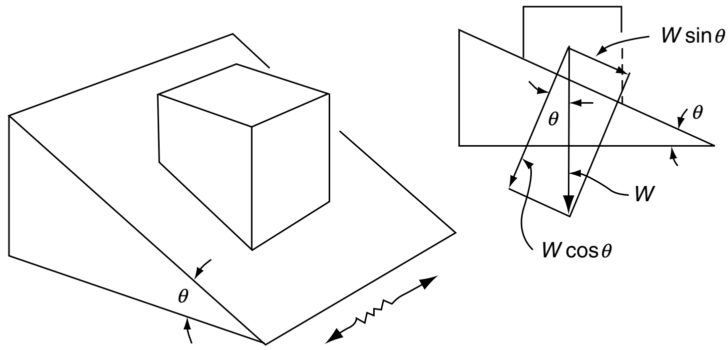


FIGURE 14.2 Shaking the inclined plane in the direction shown by the double arrows would destroy frictional constraints between block and plane and allow force $W \sin \theta$ to move the block to the foot of the plane. (W is the weight of the block.)

The fastener, of course, is not a block on a table. It can better be modeled by the inclined plane and block shown in Figure 14.2. There is now a small force which wants to move the block down the plane—a force equal to $W \sin \theta$, where W is the weight of the block and θ is the angle of the plane on which the block is resting. This tendency to slide, of course, can be overcome, or more than overcome, by frictional restraints ($\mu W \cos \theta$) between the block and plane. If we were to shake the plane vigorously in the direction shown by the double arrow, however, the block would gradually or rapidly walk its way to the bottom of the plane. This vibration in one direction would destroy the friction forces and allow $W \sin \theta$ to do its job [4].

In the fastener, of course, it is not the weight of the nut, but rather the tension stored in the bolt, which creates the force which pushes the inclined plane of the nut out of the way when the vibration forces are broken. This is suggested in Figure 14.3.

Note that the bolt in Figure 14.3 must be exerting a force on the nut—or the block in Figure 14.2 must be exerting a (gravitational) force on the inclined plane—if the reduction of friction force is to result in motion. The theories insist—and experiments appear to confirm—that some "off-torque" is required to create relative motion between nut and bolt, even under severe vibration. The source of this off-torque is generally considered to be the tension in the bolt acting against the inclined plane of the nut threads [1,12,26]. Other theories, however, have been proposed [25,33]. We'll look at one of these in a minute.

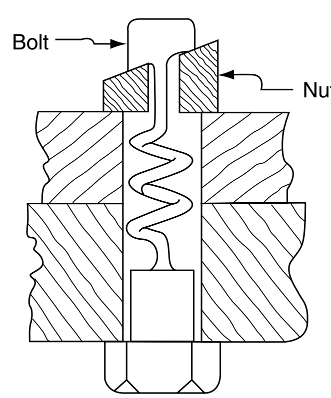


FIGURE 14.3 Schematic of a nut and bolt. Anything which breaks the friction forces between them (and between the nut and the joint) will allow the tension in the bolt to push the nut out of the way as the bolt attempts to return to its initial length.

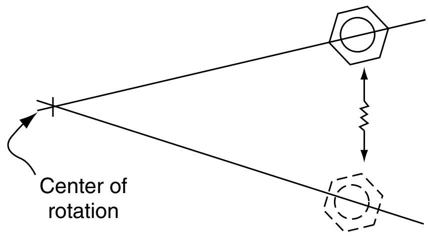


FIGURE 14.4 Vibration along the circular path will sometimes tighten, sometimes loosen, and sometimes do "neither" to a fastener.

One thing that has been agreed upon is that vibration is a far greater problem in a joint loaded in shear than in a joint loaded only in tension. Severe vibration parallel to the axis of the bolt, for example, might succeed in reducing preload by 30% or 40% over a long period of time. But it will usually not result in total loss of preload or in loss of the fastener. Severe transverse vibration, perpendicular to the axis of the bolt, can, and often does, cause complete loss of preload. The theory is that only transverse vibration destroys those frictional restraints in what are basically horizontal or transverse surfaces. I found this a little difficult to understand at first, because instinct tells me that if I tap the inclined plane in Figure 14.2 with a pencil—in any direction—or bang it up and down, the block will slip to the foot of the plane just as readily as it would if I shake the plane back and forth in the direction shown by the arrows. In this case, therefore, the direction of vibration would be unimportant.

In the bolt, of course, it's easy to see how one could break the frictional restraints by substantial transverse vibration. If hole clearances, etc. allow it, I could certainly slip the joint members with respect to the nut, for example. Lateral clearance of 0.0013–0.0114 in. allows relative slip between nut and bolt in a Class 2A/2B thread [30]. It's difficult to envision, however, how even severe axial vibration would cause slip or separation in these surfaces. Axial vibration would actually increase the contact forces and strains between parts during part of each cycle. In fact, at least one reference [3] says that axial vibration can sometimes tighten a bolt as well as loosen it. Severe axial vibration, however, can cause periodic dilation of the nut, creating the relative motion required to break the friction forces [11].

Some fasteners are subjected to neither pure transverse nor pure axial motion but to a combination, or to arc slip, as shown in Figure 14.4, where the vibratory motion is along a circular path rather than a straight one. Experiments have shown that this type of vibration will sometimes tighten, sometimes loosen—and sometimes do neither to—the fastener [2].

14.3 LOOSENING SEQUENCE

All agree that a fastener subjected to shock or vibration or thermal cycles will not lose all preload immediately, but will first undergo a relatively slow loss of preload. No one knows for sure why this progressive loss occurs, but it has been well documented [1,12]. Most seem to think that cyclic forces applied to the thread surfaces by vibration and the like cause additional embedment and the slow destruction—the breakdown—of contact surfaces. Only after sufficient preload has been lost by this process will the friction forces between thread surfaces be low enough to be overcome by subsequent load cycles. At this point the nut will loosen rapidly. Various patterns of loss have been seen, as suggested in Figures 14.5 and 14.6.

14.4 JUNKER'S THEORY OF SELF-LOOSENING

I mentioned earlier that several theories of vibration loosening have been proposed. The best known is probably that of Gerhard Junker [1]. His work has been published in many forms,

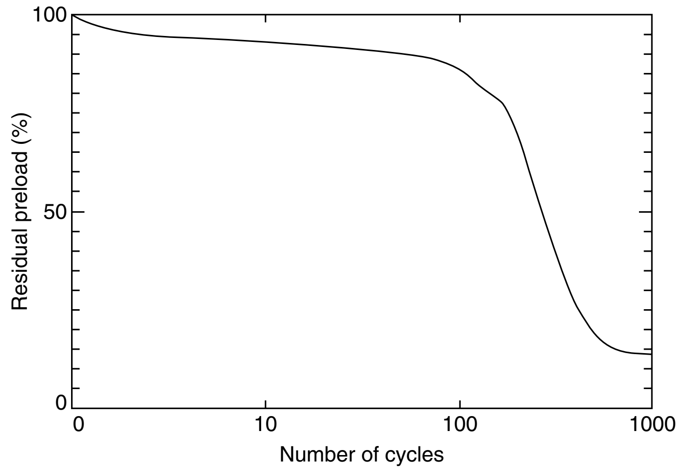


FIGURE 14.5 Vibration loosening starts with a slow and gradual relaxation of initial preload. Only when preload has fallen below a certain critical value does the nut actually start to back off. Loosening is rapid beyond this point. (Modified from Junker, G.H., *New Criteria for Self-Loosening of Fasteners under Vibration*, October 1973, Reprinted from *Trans. SAE*, 78, 1969.)

and has apparently been confirmed by experiments conducted on a ‘‘Junker machine,’’ which we’ll discuss later. Further confirmation comes from results obtained with special vibration-resistant fasteners whose designs have been based on his theories. His theory is, therefore, a reasonable starting point. Let’s look at it in detail.

14.4.1 The Equations

Junker’s theories are based, in part, on the so-called long-form torque–preload equation relating the torque applied to a fastener to the frictional and elastic reactions to that torque.

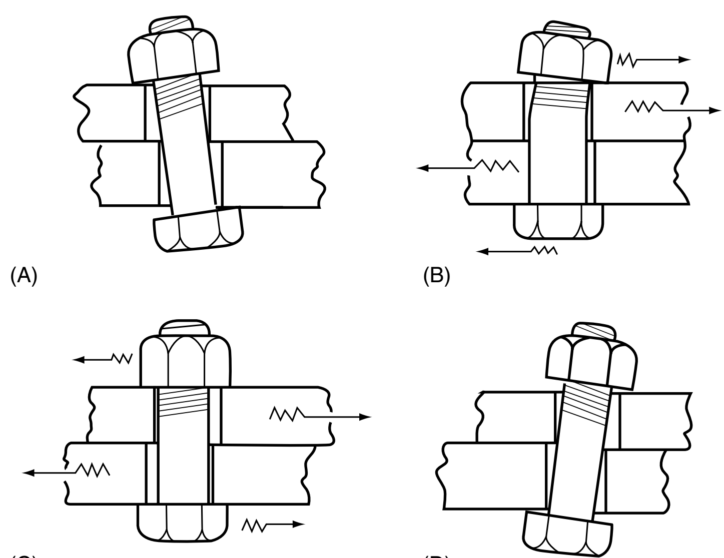


FIGURE 14.6 Illustration of the process by which fasteners loosen, according to Gerhardt Junker. See text for discussion.

The form of the equation I prefer, because I find it most descriptive, is a simplified version proposed by Nabil Motosh [21]:

$$T_{\text{in}} = F_{\text{P}}\left(\frac{P}{2\pi} + \frac{\mu_{\text{t}} r_{\text{t}}}{\cos\beta} + \mu_{\text{n}} r_{\text{n}}\right) \tag{14.1}$$

where

T_{in} = torque applied to the nut (in.-lb, N mm)
F_{P} = preload in bolt (lb, mm)
P = thread pitch (in., mm)
μ_{t}, μ_{n} = coefficient of friction in thread and nut surfaces, respectively
r_{t}, r_{n} = effective contact radii of thread and nut surfaces (in., mm)
β = half-angle of thread tooth (usually 30°)

If a "prevailing torque" fastener is being used, the long-form equation must be modified to include this torque—a reaction torque (T_{P}) which is not proportional to preload.

$$T_{\text{in}} = F_{\text{P}}\left(\frac{P}{2\pi} + \frac{\mu_{\text{t}} r_{\text{t}}}{\cos\beta} + \mu_{\text{n}} r_{\text{n}}\right) + T_{\text{P}} \tag{14.2}$$

where T_{P} = prevailing torque (in.-lb, N mm).

14.4.2 The Long-Form Equation in Practice

The long-form equation is believed to explain, correctly, the basic relationship between input and reaction torques in a threaded fastener. In practice, however, it is virtually useless as a means of predicting the exact relationship between applied torque and achieved preload in a given fastener, since we never know what values to assign to the parameters involved.

In spite of these limitations, the long-form equation can be used to explain behavior, even if it's not especially useful for numerical calculations.

14.4.3 The Equation When Applied Torque Is Absent

Junker uses the long-form equation with applied torque, $T_{\text{in}} = 0$. In other words, he's only concerned with the torque created by the elastic stretch of the bolt and the frictional reaction torques. Note that the elastic torque still wants to rotate the nut in a counterclockwise direction, as it did while the bolt was being tightened. The friction torque, however, has now reversed sign because it's opposing the counterclockwise elastic torque rather than an externally applied clockwise torque. The prevailing torque, if present, will also change sign. Taking all of this into account, and using Motosh, we can write for T_{OFF}, the net torque tending to loosen the fastener,

$$T_{\text{OFF}} = F_{\text{P}}\left(\frac{P}{2\pi} - \frac{\mu_{\text{t}} r_{\text{t}}}{\cos\beta} - \mu_{\text{n}} r_{\text{n}}\right) - T_{\text{P}} \tag{14.3}$$

How far would the coefficients of friction have to drop before there would be a net torque to loosen the bolt (assuming nominal geometry and no prevailing torque)?

As an example, in an ASTM A325 fastener, of 1 in. diameter with eight threads per inch tightened to proof load (F_{P} = 51,500 1bs), the coefficients of friction would have to drop to about 0.015 before there would be a net, internal off-torque to loosen the bolt. This is one-fifth to one-tenth of what we would expect the coefficients to be under normal conditions.

Some other factor must enter the picture if loosening is to occur. Junker attempts to explain this other factor. He suggests that (assuming that no locking or prevailing torque device is present)

1. The elastic stretch in a bolt will create a torque that attempts to loosen the nut, as suggested by the long-form equation.
2. If vibration is severe enough, transverse slip will occur between male and female threads and, simultaneously, between the joint surface and the face of the bolt head.
3. This slip momentarily overcomes all frictional restraint between parts, frees both ends of the bolt, and allows a portion of the elastic energy stored in the bolt to escape as the bolt, not the nut, rotates.
4. Whether or not any energy will escape depends on whether or not external forces on the system are great enough to overpower the friction or other forces which resist slip in the threads and between bolt and joint.
5. How much energy will be lost during each cycle depends on the "thread-slip distance" involved; the greater the thread clearance, the more energy will be lost each cycle before slip ends and friction forces are reestablished.
6. The amount of energy lost during each cycle also depends on the magnitude of the net "off-torque" on the nut during slip. According to the simplified Motosh equation, this would be, simply

$$T_{\mathrm{OFF}} = \frac{F_{\mathrm{P}}P}{2\pi} \tag{14.4}$$

since, in effect, $\mu = 0$ momentarily, and there is no T_{P}.

14.4.4 Why Slip Occurs?

An important part of Junker's theory is the explanation of why transverse vibration causes thread and nut or joint slip. In the 1969 paper [1], he suggests the following sequence of events:

1. As a result of the previous cycle, the nut and bolt are in the relative positions suggested by Figure 14.6A—where thread clearance has been magnified for clarity.
2. Now the top joint member starts to move toward the right; the lower member moves left.
3. At first, the nut and bolt remain in the relative position shown in Figure 14.6A, locked in that relationship by thread friction. As joint slip continues, however, the bolt bends toward the right because of the relative motion of joint members (Figure 14.6B).
4. After a while, bending forces overcome the friction forces between male and female threads. The bolt straightens up and moves toward the opposite side of the nut. This action alone is not sufficient to cause loosening since the other end of the bolt is still held by friction forces (Figure 14.6C).
5. If conditions are severe enough, however, the head of the bolt also slips against the adjacent joint surface—after the bolt threads have started to slip over the nut threads. Under these conditions both ends of the bolt are momentarily free and it will rotate slightly, losing a little of the stored potential energy.
6. As joint slip continues, the bolt is now cocked again to the right. During this period no further preload loss occurs because there is no further thread clearance in the necessary direction (Figure 14.6D).
7. The whole process then reverses, dumping a little more energy on the return stroke.

The loosening sequence described above has been supported by experiments made on a "Junker machine," shown in Figure 14.8 and discussed in Section 14.5. Careful measurements show the brief instant during which slip occurs, as well as those portions of the cycle during which the bolt is clamping the upper and lower joint members. To my knowledge, none of the other theories for vibration loosening have been demonstrated in so convincing a fashion. Confirming evidence of the Junker theory was revealed by experiments reported in 2005 [38]. The investigators found that greater hole clearance caused more rapid self-loosening. Of more interest, perhaps, they also found that high frequency vibration causes less self-loosening than does low frequency vibration, because the higher frequency gives the fastener less time to slip per cycle. These studies involved both mathematical analysis and experimental confirmation.

14.4.5 Other Reasons for Slip

Although Junker doesn't mention it, slip can presumably be caused by factors other than bending of the bolt. The bolts don't necessarily bend during a MIL-STD-1312 test, for example, but nuts loosen. Inertial forces presumably cause the slip there. There may be other situations where a combination of inertia and bending could do it. Anything which causes simultaneous slip of threads and head will satisfy Junker's theory.

14.4.6 Other Theories of Self-Loosening

There are many other theories on self-loosening, but brief summaries of them are not very useful and I'll skip them here. They are discussed briefly in the third edition of this text and at some considerable length in Ref. [28]. The most recent theories I'm aware of were presented by Nassar and Housari at the July 2005 Pressure Vessel and Piping (PVP) Conference [38] and by Wang et al. a year later at the PVP Conference in July, 2006 [33]. The authors describe a two stage self-loosening sequence. The first stage involves plastic deformation of the threads near their roots. Some bolt tension is lost during this process, but there is no relative motion between nut and bolt threads.

The second stage of loosening occurs when the nut, not the bolt, backs off. This action is caused by cyclic, reversing bending of the bolt under transverse loads. Alternate sides of the nut stick and slip at this point, creating an off-torque which loosens the nut. This theory was developed using finite element analysis, but was then confirmed by experiment.

14.5 TESTING FOR VIBRATION RESISTANCE

In a moment we will look at many ways to improve the vibration resistance of a bolted joint. Since we can't predict vibration loosening mathematically, the usefulness of such techniques must be determined experimentally, in a test machine.

Experts warn that it is not sufficient to conduct a laboratory test on a simulated joint, however. If possible you should always repeat the test on the actual joint: hopefully the one you are having trouble with—to be sure that you have really made a difference. Our knowledge of vibration loosening is entirely empirical, and there are many factors which can make a difference. Some experiments, in fact, have suggested that complex interactions between suspected factors, perhaps more than the factors themselves, determine the rate at which a given system will loosen, or that there probably are other factors which we have not been able to pin down as yet, which also make a difference [6]. You could easily be fooled by some of these unknown interactions and factors if you tested only a "test joint."

14.5.1 NAS Test

One popular way to test for vibration resistance is shown in Figure 14.7. The nut and bolt are tightened onto a small cylinder. This cylinder is placed in the slot of a test block. The cylinder

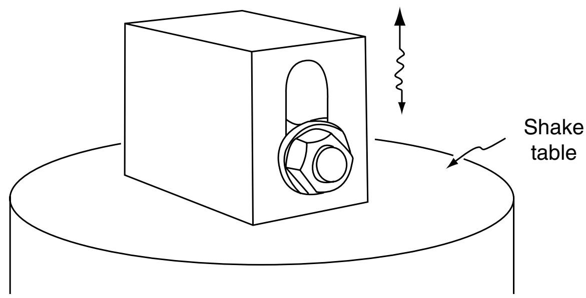


FIGURE 14.7 The NAS vibration test. The fastener under test is mounted in a cylinder which is free to bang up and down within the slot in the block as the block is vibrated vertically.

is longer than the block, and washers are used at both ends so that the cylinder is free to bang around in the slot without being able to fall out of it. The block is vibrated in a vertical direction, causing the fastener and its cylinder to bang back and forth between the top and bottom of the slot until the nut and bolt shake apart. Vibration frequency, amplitude, and time are measured to set a numerical value on the vibration resistance of the fastener system under test. This is sometimes called the ALMA test [3,5]. All pertinent dimensions are defined in the government specification MIL-STD-1312 and in National Aerospace standard NAS 3354. A test procedure is described in NAS 3350 [35]. In it the nuts are installed (tightened), removed, and reinstalled 3–5 times depending upon the fastener being tightened, before being subjected to the test. Counterclockwise rotation of the nut of more than 360° constitutes failure. This is a very severe test procedure. Many types of vibration-resistant fasteners which survive a controlled Junker test loosen rapidly during a NAS test. Sawa et al., for example, used both Junker and NAS fixtures to test twelve types of vibration-resistant washers and nuts. Only a nut with eccentric (out-of-round) threads, shown in Figure 14.12, survived the NAS tests. A nylon insert nut took several minutes to fall apart but eventually succumbed. All others tested fell apart in seconds. Several of the same fasteners, however, did well during a Junker test [34].

14.5.2 Junker Test

The test machine shown in Figure 14.8 is called the Junker machine. An eccentric cam generates a controllable amount of transverse displacement on the joint under test. Force cells

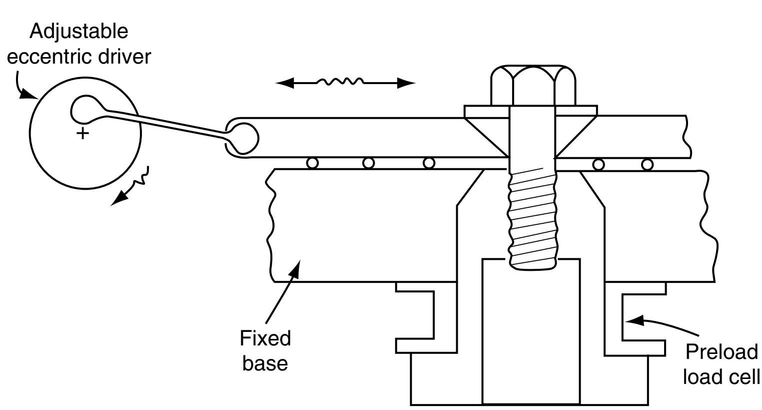


FIGURE 14.8 The Junker vibration test machine. The forces exerted on the test joint and the displacement of the test joint can both be measured with this device. (Modified from Finkelston, R.F. and Wallace, P.W., Advances in high performance fastening, Paper no. 800451, Presented at the Congress and Exposition of the SAE, Cobo Hall, Detroit, MI, February 29, 1980; from information published by SPS.)

measure the actual transverse forces exerted on the joint. One can now determine the relationship between residual preload in the fastener under test and external vibratory forces created by the test machine as a function of time. SPS used to make and sell Junker machines, but they no longer do that. The SPS Laboratory in Jenkintown, PA, however, has two Junker machines and SBS will test things for you [28,29].

The Junker is, I believe, the popular machine for testing the vibration resistance of fasteners, but it (and the NAS device) is far from the only fixtures which are used for this purpose. Hess describes half a dozen or so alternates, for example [28].

Theoretically, when you test a fastener for vibration resistance, you would like to subject it to the vibration frequencies and magnitudes you expect the joint to encounter in your application. Predicting the vibration environment a given product will see, however, is even more difficult than predicting external loads. You will rarely be able to find good data in the literature for your application or product, but will, instead, have to rely on your own field tests. Vibration frequencies and magnitudes will not be uniform, furthermore, but can vary from moment to moment as well as from user to user. Your only recourse is to provide a fastener system that is immune to the range of frequencies you expect it might encounter in practice and then to determine by trial and error whether or not you have been successful.

14.6 TO RESIST VIBRATION

In Chapter 13 we learned that self-loosening will occur only if two essential conditions are present: cyclic, transverse loads, and relative slip between thread and joint surfaces. Junker and several others suggest various ways in which these conditions might cause self-loosening. According to most theories, we can prevent self-loosening if we can eliminate one or both of these conditions. It's also obvious that we could prevent at least complete loss of preload if we could somehow fasten or lock the nut to the bolt, relying on mechanical or chemical means rather than on friction to guarantee the integrity of the fastener.

We're now about to examine a few of the many ways which have been proposed for doing these things. Most of our options will fall into one of the following categories.

1. Keep the friction forces in thread and joint surfaces from falling below the forces that are trying to loosen the nut.
2. Mechanically prevent slip between nut and bolt or nut and joint surfaces.
3. Provide a prevailing torque or locking action of some sort, which counters the back-off torque created by the inclined planes of the threads, and does so even after friction forces in the system have been overwhelmed by vibration (See Table 14.2).

TABLE 14.2
Relative Performance of Various Types of 3/8–16 Locking Fasteners

Type of Fastener	Percentage of Initial Load Retained
Serrated locking screw	85
Anaerobic adhesive	85
All-metal locking screw	50
Epoxy locking screw	45
Patch-type locking screw	30
Prevailing torque nut	30

It also helps to reduce the helix angle of threads to reduce the back-off torque component ($W \sin\theta$ in Figure 14.2). Let's look at some of the ways we can accomplish the three goals.

14.6.1 Maintaining Preload and Friction

14.6.1.1 Conventional Wisdom

The least expensive and simplest way to fight self-loosening is often by preventing loss of preload in the fastener. High preload or bolt tension provides a high normal force, which, in turn, creates frictional forces that discourage relative motion between nut and bolt. So we want to ensure proper control of preload during assembly and do whatever is possible to reduce or eliminate or compensate for the subsequent relaxation of preload caused by embedment, elastic interactions, and the like.

It is generally agreed that we want to tighten the fasteners to the threshold of yield if we need maximum vibration resistance [7,8]—always recognizing, however, that we may not be able to tighten them this much if external loads or safety factors make this much preload unwise. As far as vibration resistance alone is concerned, however, the more preload the better, as shown in Figure 14.9 [7].

We can also do things to modify the coefficient of friction of thread or other surfaces, avoiding lubricants, for example, and plating parts to increase the coefficient beyond that which we would get with as-received parts [9]. Nassar and Housari found that lubricating the threads of a bolt significantly increased its tendency to self-loosen. Underhead lubrication had much less effect [38].

Introducing some form of vibration damping can also help maintain friction, as mentioned earlier, because it reduces the rate at which preload will relax under vibration. Nylon inserts in the bolt or nut threads are said to be an effective way to accomplish this [3]. They presumably also prevent side slip between male and female threads.

Note that one way to reduce preload loss is to provide a low bolt-to-joint stiffness ratio. As the joint diagrams in Chapter 10 told us, a relatively "stiff joint" and "soft" bolt will reduce the amount of bolt preload lost for a given amount of bolt length change; in this case a given amount of embedment or hammering.

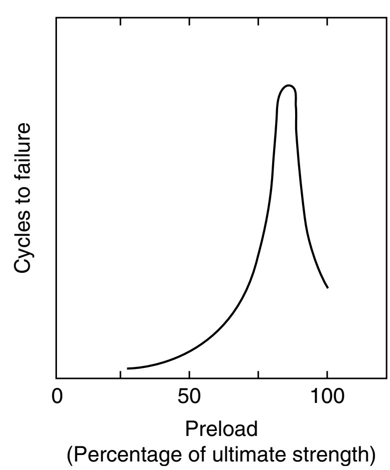


FIGURE 14.9 Vibration resistance of a Grade 5 fastener as a function of preload (expressed as a percentage of yield strength).

Anything else which reduces the amount and rate of relaxation will also be helpful. If possible, for example, you should avoid gaskets in a joint subjected to severe vibration.

14.6.2 Preventing Relative Slip between Surfaces

Providing and maintaining adequate bolt tension is probably the easiest and least expensive way to combat moderate vibration. In many cases, however, it is impossible to provide a large enough fastener preload to withstand the vibration present. The preloads required to resist vibration forces would yield or break the fasteners we have to work with—because of limitations on joint size, shape, cost, or the like. Under these circumstances, something else is required.

If you're designing the joint, one way to minimize slip is to orient the bolts and joints so that bolt axes are parallel to the expected direction of vibration. Remember that axial vibration is far less of a problem than transverse vibration.

In many cases a designer can shape the joint so that relative slip between joint members is prevented or at least minimized. Remember that there must be actual slip before we break the friction forces that resist vibration loosening. Joints such as those shown in Figure 14.10, therefore, can be very helpful in fighting vibration [19].

The fastener can sometimes be used as a dowel pin—in a tight hole—to reduce joint slip. Or an actual dowel pin can be added to the joint. Joint members can be tack-welded together. Or adhesives can be used between joint surfaces to minimize slip.

Experiments have indicated that the nuts of long, thin fasteners won't slip over joint surfaces under transverse vibration; instead, the fasteners bend. One knowledgeable source says that a length to diameter ratio of 6:1 provides optimum resistance to self-loosening [30]. Another authority says that if the length-to-diameter ratio is greater than 8:1, "you can't shake them loose" [10].

One way to reduce back-off torque on the nut would presumably be to reduce preload. Everyone who has studied the problem, however, says that we always want the maximum possible preload to reduce vibration loosening. The loss in friction forces which would result from less preload, in other words, overwhelms any advantages we would get by reducing the off-torque created by the inclined planes.

We can, however, reduce the off-torque by decreasing the helix angle. The only practical way to do this in most situations is to use a fine-pitch thread instead of a coarse-pitch thread. This can make a useful difference, as shown in Figure 14.11. Hess reports on an experiment in which fine pitch threads endured twice as many vibration cycles as coarse pitch threads [28].

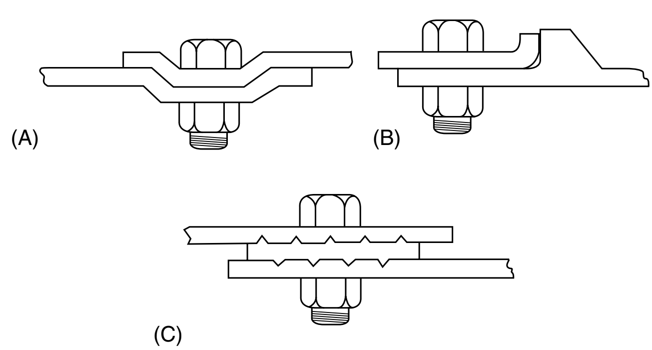


FIGURE 14.10 Joints can sometimes be designed to resist transverse slip. This can be an effective way to prevent self-loosening. A toothed "shear washer" has also been introduced in joint C.

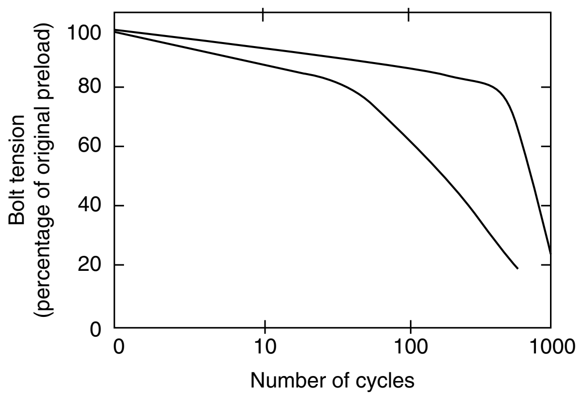


FIGURE 14.11 Reducing the helix angle of the threads by using a fine-pitch thread instead of a coarse-pitch thread can make a useful difference in the vibration resistance of a fastener. (Modified from Finkelston, R.F. and Wallace, P.W., Advances in high performance fastening, Paper no. 800451, Presented at the Congress and Exposition of the SAE, Cobo Hall, Detroit, MI, February 29, 1980.)

14.6.3 Countering Back-Off Torque

When all else fails—and it often does—the only thing we can do to fight vibration loosening is to provide another source of torque to counter the back-off torque produced by the inclined planes of the threads. Now, even if vibration totally destroys all friction forces, some other mechanism prevents that nut from being pushed out of the way by the bolt threads.

14.6.3.1 Prevailing Torque Fasteners

There are many different types of prevailing torque fasteners, a few of which are shown in Figure 14.12. In general they can be classed as (1) all metal nuts or bolts whose threads have been purposely distorted or modified to provide some interference with the mating part, (2) nuts or bolts with a plug or patch or insert of nonmetallic material—often nylon—in the threads to create interference, and (3) nuts with a collar or ring of nonmetallic material, again

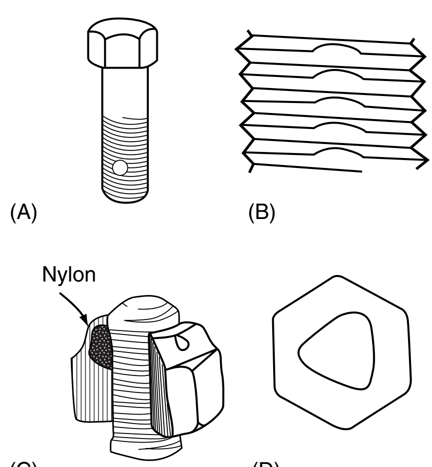


FIGURE 14.12 A selection of prevailing torque nuts and bolts. Those shown include (A) nylon pellet in bolt threads (Greer); (B) interference fit threads (SPS); (C) nylon locking collar in nut (ESNA); and (D) nuts with out-of-round holes.

TABLE 14.3
Typical Vibration Performance of Various Vibration-Resistant Nuts

Type of Nut	Relative Number of Cycles Required to Shake Nut from Bolt
Nut with locking ring of nylon	100
Beam-type self-locking nut—aircraft	53
Castellated nut and spring pin	38
Distorted thread nut—aircraft	19
Castellated nut and cotter key	18
Beam-type self-locking nut—commercial	4–17
Distorted thread nut—commercial	1–10
Castellated nut and lock wire	8
Plain nut and spring-type lock washer	5
Plain nut, with or without tooth-type lock washer	1

to create interference with the mating bolt and, in this case, as discussed earlier, to dampen the resonant frequency vibrations of the fastener. Visit ESNA and GRIPCO on the Web for details of currently available products. Tests conducted by several groups suggest that nuts with nylon inserts provide unusually good resistance to self-loosening. So do nuts with so-called eccentric or out-of-round holes [19,34]. See Tables 14.3 and 14.4.

Since the torque required to run down the nut on a prevailing torque fastener must be added to the torque required to achieve a desired preload, it's important to be able to predict the run-down torque. The Industrial Fasteners Institute in Cleveland, Ohio, has developed standards that specify acceptable run-down torques. They also specify the minimum torques required to disassemble the fasteners on the first and fifth removals. These torques, of course,

TABLE 14.4
Relative Resistance to Self-Loosening of M10×1.5, Class 4.8 Bolts, Tightened to 70% of Yield

Good resistance
- Eccentric nut
- Nord-Lock washer
- Belleville washer

Fair resistance
- Toothed washer
- Nylon insert nut

Poor resistance
- Spring washer
- Double nut
- Flanged nut

Source: From Sawa, T., Ishimura, M., and Yamanaka, H., Experimental Evaluation of Screw Thread Loosening in Bolted Joint with Some Parts for Preventing the Loosening Under Transverse Loading. Proceedings of PVP2006-ICPVT-11, ASME Pressure Vessel and Piping Division Conference, Vancouver, BC, Canada, July 23–27, 2006. With Permission.

Note: The tests were made in a Junker's machine. Results are listed in decreasing degree of resistance, the more resistant listed before the less resistant.

are those which would be acting to prevent "self-removal," and so are also of interest. Prevailing torque nuts are covered in IFI Standard 100/107. English and metric series bolts are covered in IFI 124 and 524, respectively [24].

Free-spinning lock nuts or bolts can be run down with normal (very little) torque. As they are tightened against the surface of the joint, however, they dig into it, or distort in some way to create an interference fit with the mating parts. A few of the many choices are sketched in Figure 14.13 [11,13–15,17,34]. The serrated head bolt shown in Figure 14.13B is reported to be especially effective. So is the Spiralock thread form of Figure 14.13A. The bolt threads here are conventional, but the root of the nut threads is a tiny ramp or inclined plane. As the nut is tightened, the tips of the male thread are forced into interference fit with the ramps. This eliminates all clearance between male and female threads. The inventor of this thread form, Harold (Ace) Holmes of Detroit, is a firm believer in the Junker theory of vibration loosening. He designed the Spiralock thread form to eliminate loosening by eliminating slip clearance—and the results seem to support Junker's theories.

The Spiralock thread form has been tested at MIT [26] and at Lawrence Livermore Laboratories [27]. In the resulting reports we learn that, although the nut is free spinning until seated, this thread form requires 20% more torque than a standard thread form to achieve a given preload. The extra torque, of course, is required to pull the male threads up the root ramps of the female threads. Another feature: the Spiralock thread form creates a more uniform distribution of load and stress than does a standard thread form. For example, Nayak of MIT says that only 18% of the tensile load in the bolt is transferred to the first engaged thread versus 34% in a standard thread form [26]. He also says that the Spiralock thread requires three times as much off-torque to start loosening the nut as does a standard form, a measure of the Spiralock's vibration resistance.

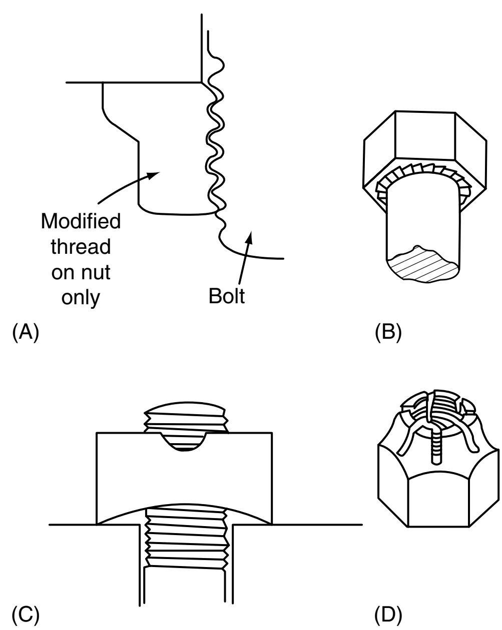


FIGURE 14.13 Free-spinning lock nuts and bolts which can be run down with normal torque, but which create an interference of some sort of final tightening. Those shown include (A) interference thread nut (Spiralock by Greer/Smyrna or Detroit Tool Industries); (B) serrated head bolt by SPS (serrations dig into joint surface and resist reverse rotation; serrated face nuts are also available); (C) spring head nut, which distorts inwardly to pinch bolt; and (D) spring arms on top of nut provide interference fit with bolt threads (called beam-type nut).

14.6.3.2 Nord-Lock Nuts and Washers

The nuts and washers shown in Figure 14.14 are made by the Swedish company Nord-Lock AB [36]. They also used to be called Disk-Lock products, but that name is not currently found on the Web so I assume those products are no longer available. The two piece Nord-Lock washer is, I think, more popular than the Nord-Lock nut but both work on the same principle. For example, mating surfaces on the two-piece washer have interposing, multitoothed cams; a series of short ramps whose angle exceeds the pitch angle of bolt threads. Sharp ridges on the other sides of the washer dig into nut and joint surfaces. If a nut starts to back off a bolt it must do so while dragging its half of the two piece washer with it. The cam surfaces on the nut's disk are forced to climb the cam surfaces on that portion of the washer which is gripping the joint. Since the angle of the ramped cam surfaces exceeds the lead angle of the threads, this relative motion between the two halves of the washer fights loosening of the fastener. In fact the tension in a bolt can actually be increased as the nut tries to loosen. These washers resist self-loosening during a Junker test more effectively than many other products [34]. They are available in ID's ranging from about 1/8 in. (3.4 mm) to about 5 in. (133.4 mm) and can be obtained from many different domestic and foreign suppliers as you'll see if you visit Nord-Lock on the Web.

Nord-Lock also makes a two-piece nut. The outer half is threaded, the inner half not. Opposing cam surfaces, again with a ramp angle exceeding the lead angle of the threads, force the nut to tighten if it attempts to rotate counterclockwise.

14.6.3.3 In General

When selecting a prevailing torque or locking fastener, you should consider these points (in consultation with potential suppliers):

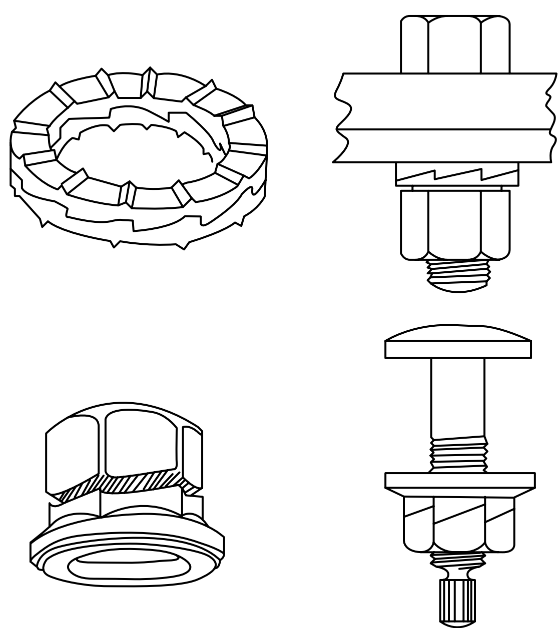


FIGURE 14.14 Nord-Lock washer and nut. The angle of the ramps shown in the drawing exceeds the lead angle of the bolt threads, so any counterclockwise rotation of one half of the two piece washer, or of the two piece nut, can actually increase the tension in the bolt.

- Operating temperature limits
- Mating thread accommodation
- Effect on mating parts (may damage or Brinell them)
- Reusability
- Type of installation tools required
- Effect on the mechanical properties of mating parts (for example, does it reduce the fatigue life of another part by providing a "softness" in the joint or by creating stress concentrations?)

14.6.4 Double Nuts

One popular way to fight self-loosening is to use a double nut; a thin nut in contact with the joint plus a standard nut on top of that. Conventional wisdom says that you should tighten the thin nut first, then tighten the top or "jam" nut onto the thin nut. Sawa et al., however, suggests a different procedure, which makes more sense to me. The inner nut is tightened, and then the outer nut is tightened, as in the conventional procedure. A wrench is then used to partially loosen the inner nut, presumably raising it into harder contact with the outer nut. Both nuts are then tightened together to complete the process [34].

14.6.5 Mechanically Locked Fasteners

Sometimes it's impossible to provide enough prevailing torque to prevent loosening under severe shock or vibration conditions; other times self-loosening would threaten safety and we want to be absolutely sure the fasteners won't come loose. In such situations we can consider the use of fasteners in which the nut and bolt are mechanically locked together. We'll find both new and old options in this category.

14.6.5.1 Lock Wires and Pins

The earliest attempts to prevent self-loosening of a threaded fastener probably involved lock wires, keys, and cotter pins; and these still find a lot of use. A couple of examples are shown in Figure 14.15. These can effectively prevent total loss of the nut—which may be extremely important—but they are not very effective in preventing substantial loss of preload within the fastener. It has been reported, for example, that 2° of rotation in the nut can reduce preload in a hard joint by 27%; 6° can reduce it 42% [1]. Most lock wires or cotter pins aren't intended to provide tight control of nut motion. Even if they save the nut, the loss of preload may lead to fatigue or another type of failure.

14.6.5.2 Welding

Nuts can be welded to bolts, at least if they're large enough. The normal procedure is to tack weld the nut to the end of the bolt. Another procedure is to tack weld both the nut and the head of the bolt to joint surfaces. Either procedure makes removal of the nut (for maintenance purposes, for example) very difficult and will probably make it necessary to replace them with new parts if they are removed. A related procedure, which preserves the parts, is to place a "keeper" over the tightened nut and then weld the keeper to the joint, as shown in Figure 14.16. This trick has been used to trap the large nuts used on the foundation studs of nuclear reactors; nuts which presumably will not be removed until the reactor is taken out of service.

14.6.5.3 Stage 8 Fastening System

The Stage 8 fastening system, shown in Figure 14.17, fights loosening in still another way. The bolt and nut are conventional except for snap-ring grooves, as shown. Loosening is resisted by

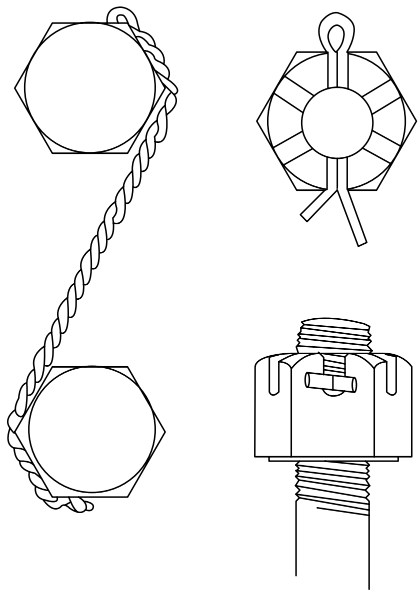


FIGURE 14.15 Lock wires and cotter pins are ancient ways to mechanically fasten the nut to the bolt. They're both still widely used, but are more awkward to install than more modern locking fasteners. The cotter pin is often used in conjunction with a castellated nut, as shown here.

a retaining arm which is slipped over the nut or bolt head. The far end of the arm butts up against an adjacent nut or other reaction surface to prevent counterclockwise rotation. The snap-ring prevents loss of the retaining arm. A wide variety of retaining arms, collars, rings, etc. are available, depending on the bolt pattern, the reaction surfaces available, etc.

14.6.5.4 Huck Lockbolt

Lest we forget, the so-called lockbolts which are described and illustrated (Figure 9.5) in Chapter 9 will also resist severe vibration. Remember that these bolts are held by collars swaged into annular grooves, or into threads and a key way instead of by nuts. Huck's Model C6L LockBolt is designed especially to fight vibration. (See it on their Web site.)

14.6.5.5 Honeybee Robotics

Honeybee Robotics has designed a vibration-resistant fastener under funding from NASA. Standard bolts are used with nuts whose threads have been modified. The nut threads have

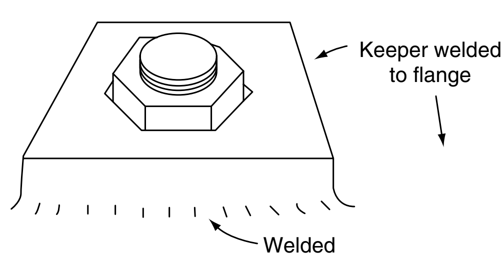


FIGURE 14.16 A square plate with a hexagonal hole in it is placed over the nut. Then the plate is welded to the top surface of the joint to retain the nut.

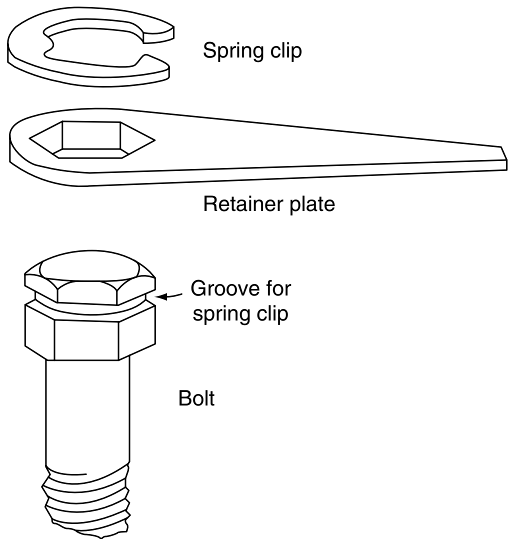


FIGURE 14.17 The Stage 8 fastening system provides a reaction or retaining arm to prevent reverse rotation of the nut or bolt. A variety of reaction arm configurations are available.

flat ends and an included angle which differs from the standard. When tightened, the crest of the nut threads digs into the roots of the bolt threads, giving the fastener significant resistance to vibration.

14.6.5.6 A-Lock Bolt and Nut

Shap, Inc. of Hacketstown, NJ, makes vibration-resistant bolts and nuts with unusual threads whose design has been patented (Figure 14.18). When tightened, these threads are ‘‘wedged into a locking position.’’ If wished, the fastener can later be loosened and is reusable.

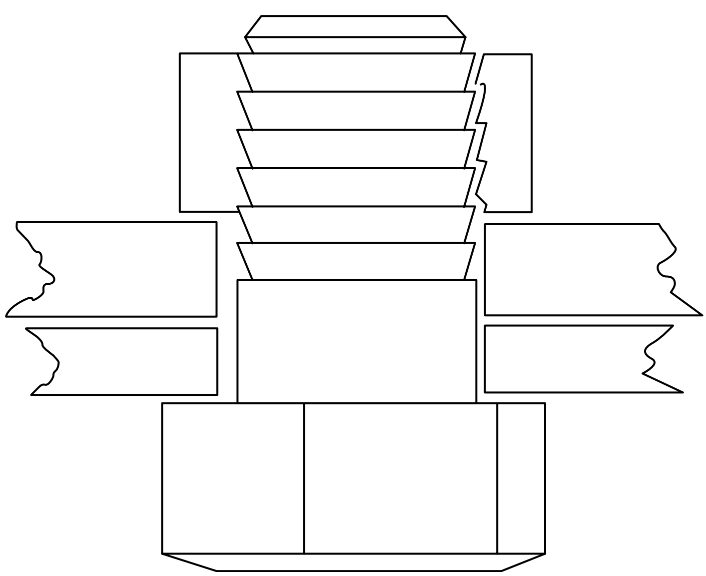


FIGURE 14.18 The Shap, Inc. A-lock thread form. When tightened the threads wedge together to resist self-loosening.

14.6.5.7 Omni-Lok Fasteners

These fasteners are locked by one or more pins that are inserted parallel to the axis of the threads. The OD of the pins extends above the root diameter of the external threads, and below the minor diameter of the internal threads, eliminating self-loosening. The pins can be made of a wide variety of materials, from high temperature alloys (up to 1200°F) to soft ductile materials, depending upon the application. Omni-Lok is one of several vibration-resistant fasteners made by Long-Lok Fasteners.

14.6.6 Chemically Bonded Fasteners

Chemically bonded fasteners provide the vibration resistance of mechanically locked ones, and are more popular. Part of the reason for this may be the fact that the chemicals can be used on small as well as large fasteners. Also contributing to their popularity, the chemicals cost little and can be used on standard fasteners. In many cases the chemicals are applied by the user before assembly; in other cases they're applied by the manufacturer. Several chemicals are used, including acrylics, but I believe that microencapsulated anaerobics are the most popular. Before taking a brief look at them, let's look at another, less common way to bond nuts to bolts.

14.6.6.1 Rust

The U.S. Marines have found an interesting way to fight vibration loosening in tank tread bolts. After they have assembled the tread, they drive the tank through surf. Saltwater corrosion effectively "welds" the nut and bolt together and welds them to the joint members. Petrochemical engineers often find rain helpful.

14.6.6.2 Anaerobic Adhesives

One very good way to resist off-torques is to "cement" the nut and bolt together. The most common way to do this is with an anaerobic adhesive, a material which is activated (hardens) when subjected to high pressure in the presence of metal and the absence of air [15,30]. It is applied to fastener threads much as a lubricant would be applied. It "glues" the threads together when they are tightened; it can, however, be overcome if you subsequently wish to take the joint apart. The material does no permanent damage to the threads.

A wide variety of anaerobic adhesives are available. Selection would be based on such things as the size of the gap to be filled, adhesive strength (off-torque requirements), size of the fastener, method of applications, etc. It's best to consult the manufacturers for details.

We learned earlier that it is important to know how much torque will be required to install a prevailing torque fastener, because that torque must be added to the torque required to achieve a desired preload. The same is true of fasteners coated with anaerobic adhesives or other chemicals. We're also interested, of course, in knowing how effective such coatings are in resisting vibration, which means we'd like to know how much breakaway torque is required to back off the nut and how much prevailing torque is involved if we try to loosen it further. The Industrial Fasteners Institute of Cleveland, Ohio, has developed standards listing acceptable values for each of these torques when the coating is applied by the manufacturer. Standard IFI-125 covers English series fasteners; IFI-525 covers metric series fasteners [24].

Locktite, the first company to provide anaerobic adhesives, says that the adhesive also fills the gap between male and female threads, preventing relative side slip and preventing moisture from corroding thread surfaces. This keeps the threads from seizing in service [30]. The adhesive also seals the bolt holes, preventing leakage of gas or liquids. Holes in blind flanges can, therefore, be drilled through; a significant cost saving. Anaerobic adhesives can be applied in the field, pre-applied by the user and allowed to dry before use; as long as the

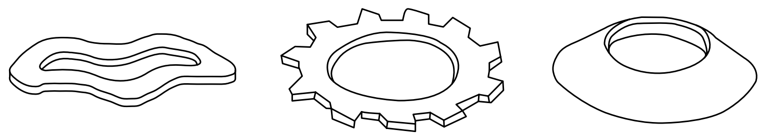


FIGURE 14.19 Washers used to fight self-loosening. The wave washer (A) is supposed to provide some spring tension in the bolt after the nut has loosened; but I doubt if it's of much help. The toothed washer (B) digs into both nut and joint surfaces and fights relative motion between them. The Belleville washer (C) is often used in stacks, on large fasteners in pressure vessel and similar applications. A stack provides a fairly high spring rate, and also allows us to use longer bolts whose lower spring rate will be more comparable to that of the Bellevilles. The lower spring rate also makes the bolt tension less susceptible to fluctuations in external load, thermal change, and the like, thereby fighting self-loosening by retaining bolt preload. Used singly especially on smaller bolts, the Belleville is also called a plate spring washer.

fasteners have not been previously exposed to certain oils or cleaning solutions which prevent the adhesive from curing. The total length of the thread must be wetted.

Properly applied anaerobics create a nut factor of 0.14–0.17 on steel, and so can serve as an assembly lubricant. They can be used at service temperatures up to 200°C [30].

14.6.7 Vibration-Resistant Washers

A wide variety of vibration-resistant or locking washers is also available, and the washers are very popular. A few of them are shown in Figure 14.19.

14.6.7.1 Washers That Maintain Tension in the Fastener

The wave and Belleville washers shown in Figure 14.19 are intended to push outward on the nut and so maintain some tension in the bolt if the nut loosens. The wave washer is typically used on small fasteners, the Belleville on large.

Obviously, almost all tension will have been lost in the bolt before the wave washer takes over. Its effectiveness must be questioned, to put it politely. One source says that wave, toothed, and fan washers are inadequate to prevent self-loosening in fastener of SAE Grade 5 (metric 8.8) or higher strength material [30].

The Belleville washer, however, has an impressive track record, at least in the pressure vessel world. They're usually used in stacks, four or more washers being piled on top of each other. The spring rate of the stack, while probably less than the spring rate of the bolt, is still high enough to provide significant clamping force on the joint. The stack also allows us to use a longer bolt, reducing bolt stiffness and making some loss of deflection less significant, as suggested in Figure 6.17.

Smaller Bellevilles, used singly, are also called "plate spring washers," and are quite effective in fighting self-loosening. They are available in sizes ranging down to about ¼ in. (6 mm) ID. At least one source can also provide them up to 39 in. (1003 mm) in size [36].

14.6.7.2 Toothed Washer

The toothed washer shown in Figure 14.19 is designed, I believe, to bite into both joint member and nut, preventing relative motion. They're widely used in appliances and in other applications where some self-loosening is, while annoying, of relatively little concern. As just mentioned, however, they should only be used with low strength fasteners.

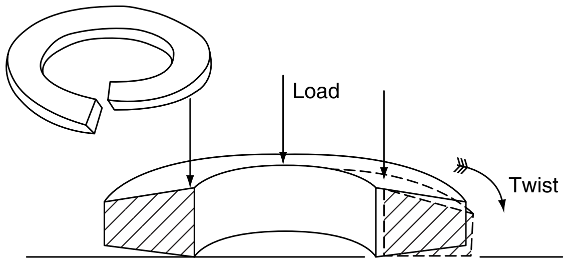


FIGURE 14.20 A helical spring lock washer would appear to be a fairly inefficient way to resist self-loosening; but recent research—described in the text—shows that this washer twists and rolls when it's fully loaded. Since that requires a clamping force that can equal 65% of the proof load of the bolt, this washer can, indeed, provide significant resistance to self-loosening.

14.6.7.3 Helical Spring Washer

At first glance the helical spring washer shown in Figure 14.20 would appear to be of as little value as the wave washer; unless the cut ends manage to bite into joint and nut and resist relative motion the way the toothed washer does. Research at the Lawrence Technological University in Southfield, Michigan, however, suggests that this device is more effective than it appears to be [22]. Dr. Clarence Chambers has shown that, while this washer is flattened by bolt tension equal to only 5% of its proof load, increasing bolt preload to 70% of proof will cause the trapezoidal cross section of the washer to roll and twist down on the outside diameter, which also grows. This complex action results in a washer spring rate which can approach 65% of the spring rate of the fastener. That spring rate will dominate the behavior of the fastener under load, and will reduce the amount of preload lost under a given applied load. Retaining preload, of course, is an effective way to resist self-loosening.

14.6.7.4 Nord-Lock Washer

To complete the list of washers: don't forget the Nord-Lock washer described earlier.

14.6.8 Comparison of Options

As you go through the literature on prevailing torque fasteners, lock washers, anaerobic adhesives, etc. you will find many claims and counterclaims about the efficiency and values of various techniques and products. I am not in a position to pass judgment on these claims—and some of them may be obsolete by the time you read this. Table 14.2 rates a number of different possibilities from the point of view of one manufacturer (who provides all of the methods tabulated) and may or may not be pertinent for your own applications [17]. Table 14.3 rates others from the point of view of a different manufacturer [19]. Table 14.4 is the most recent comparison I'm aware of and comes from Sawa et al. [34]. I suggest that you talk to many possible suppliers, giving them full information about your problems, before making a final selection. Even then, it would be best to test several possibilities in a Junker or other test machine before making a final selection.

EXERCISES

1. Self-loosening is common. Do we know why a fastener loosens or exactly how?
2. Which direction of vibration motion will be most likely to loosen a fastener?
3. With reference to Equation 14.1, explain the terms $F_P P/2\pi$, $F_P\ \mu_t r_t/\cos\beta$, and $F_P\ \mu_n r_n$.

4. Does the well-accepted Junker theory explain all self-loosening phenomena?
5. Describe the MIL-STD-1312 (NAS) test.
6. Name three generic ways to prevent or at least minimize self-loosening.
7. Define prevailing torque.
8. A proper length-to-diameter ratio for the bolt can reduce or eliminate self-loosening. What L/D ratios have been recommended?
9. Give the trade names of at least two special thread forms which have been designed to combat self-loosening.
10. What is an anaerobic adhesive?
11. How does a stack of Belleville washers fight self-loosening?

REFERENCES AND BIBLIOGRAPHY

1. Junker, G.H., *New Criteria for Self-Loosening of Fasteners under Vibration*, Reprinted October 1973 from Trans. SAE, vol. 78, 1969, by the Society of Automotive Engineers.
2. Sakai, T., Investigations of bolt loosening mechanism, *Bulletin of the JSME*, vol. 21, Paper No. 159-9, September 1978.
3. *Maintaining the Tightness of Threaded Fasteners*, ESNA Division, Amerace Corporation, Union, NJ, 1976.
4. Shigley, J.E., *Mechanical Engineering Design*, 3rd ed., McGraw-Hill, New York, 1977, p. 244.
5. Landt, R.C., Vibration loosening—causes and cures, Presented at Using Threaded Fasteners Seminar, University of Wisconsin-Extension, Madison, WI, April 24, 1979.
6. Walker, R.A., The factors which influence the vibration loosening of fasteners, Presented at 1973 Design Engineering Conference, Philadelphia Civic Center, April 10, 1973.
7. Smith, S.K., Fastener tension control—what it's all about, *Assembly Engineering*, November 1976.
8. Junker, G., *Principles of the Calculation of High Duty Bolted Connections—Interpretation of Guideline VDI 2230*, VDI Berichte no. 220, 1974. An Unbrako technical thesis, SPS, Jenkintown, PA.
9. McKewan, J., The effects of plating on torque-tension relationships and vibration resistance, Paper no. 800452, Congress and Exposition of the SAE, Cobo Hall, Detroit, MI, February 19, 1980.
10. Hardiman, R., Vibration loosening—causes and cures. Presented at Using Threaded Fasteners Seminar, University of Wisconsin-Extension, Madison, WI, May 9, 1978.
11. Aronson, R.B., How locknuts remove the "unknowns" in joint design, *Machine Design*, p. 166, October 1978.
12. *There's Nothing Quite Equal to Spiralock® Nuts*, Greer/Central, a division of the Microdot Corp., Troy, MI, 1980.
13. 1979 fastening and joining reference issue, *Machine Design*, Penton/IPC Publications, Cleveland, OH, November 15, 1979.
14. Through-hardened DURLOK 180® screws lock tight without gouging, Advertisement of SPS Technologies, Jenkintown. PC, published in *Machine Design*, April 1980.
15. *Loctite® Technology*, Loctite Corp., Newington, CT, 1979.
16. Ingenious locking concepts abound, *Product Engineering*, p. 39, March 1979.
17. Finkelston, R.F. and P.W. Wallace, Advances in high performance fastening, Paper no. 800451, Presented at the Congress and Exposition of the SAE, Cobo Hall, Detroit, MI, February 29, 1980.
18. Bonenberger, P.R., A basic approach to joint design, Presented at Using Threaded Fasteners Seminar, University of Wisconsin-Extension, Madison, WI, May 13, 1980.
19. Baubles, R.C. and G.J. McCormick, *Loosening of Fasteners by Vibration*, ESNA, Union, NJ, December 1966.
20. Landt, R.C., Comments made during presentation of a paper, Vibration loosening—causes and cures, University of Wisconsin-Extension, Madison, WI, Using Threaded Fasteners Seminar, May 1980.
21. Motosh, N., Development of design charts for bolts preloaded up to the plastic range. *Journal of Engineering for Industry*, August 1976.
22. Chambers, C., Spring rate of helical spring lock washers, Lawrence Technological University, Southfield, MI, 1991.

23. From information furnished by the MacLean-Fogg Co., Fasteners and Assemblies Division, Mundelein, IL, 1994.
24. Standards published by the Industrial Fasteners Institute of Cleveland, Ohio, as listed below:

    IFI 100/107, Prevailing-torque type steel hex and hex flange nuts, regular and light hex series, revised version published on August 15, 1993.
    IFI 124, Test procedure for the locking ability performance of nonmetallic element type prevailing-torque lock screws, 1987.
    IFI 125, Test procedure for the locking ability performance of chemical-coated lock screws, October 1973.
    IFI 524, Test procedure for the locking ability performance of metric nonmetallic locking element type prevailing-torque lock screws, 1982.
    IFI 525, Test procedure for the locking ability performance of metric chemical-coated lock screws, 1982.

25. Petras, J., Designing with locking fasteners, *Fastener Technology International*, pp. 48–49, December 1993.
26. Nayak, P.N., Spiralock: vibration resistance and stress distribution, prepared at MIT for the Chrysler Corporation, published by Spiralock, Detroit Tool Industries, Madison Heights, MI.
27. Harrel, A., III, Spiralock, finite element analysis, prepared for Lawrence Livermore Laboratories, published by Spiralock, Detroit Tool Industries, Madison Heights, MI.
28. Hess, D.P., Vibration and Shock Induced Loosening, *Handbook of Bolts and Bolted Joints*, Bickford, J. and Nassar, S. (Eds.), Marcel Dekker, NY, 1998.
29. Teleconference with members of the ASTM F16.96 Thread Class Project Task Group: Joseph Barron, Joseph Greenslade Charles Wilson and Bengt Blendulf, July 12, 2006.
30. *Loctite: Worldwide Design Handbook*, 2nd ed., Loctite North America, Hartford, CT, 1998.
31. Information obtained on their Web site about the A-Lok fastener made by Shap, Inc., Hackerstown, NJ.
32. Information obtained on the Long-Lok Web site about their Omni-Lok vibration resistant fastener.
33. Wang, Z. and Y. Jiang, Study of a Major Mechanism for Self Loosening of Bolted Joints, Proceedings of PVP2006-ICPVT-11, ASME Pressure Vessel and Piping Division Conference, July 23–27, 2006, Vancouver, BC, Canada.
34. Sawa, T., M. Ishimura, and H. Yamanaka, Experimental Evaluation of Screw Thread Loosening in Bolted Joint with Some Parts for Preventing the Loosening under Transverse Loading. Proceedings of PVP2006-ICPVT-11, ASME Pressure Vessel and Piping Division Conference, July 23–27, 2006, Vancouver, BC Canada.
35. National Aerospace Standards: NAS 3350, Nuts, Self Locking, 450°F and 800°F High Quality and NAS3354, Fixture-Vibration Test—Self-Locking Nuts.
36. From the 2006 Web site of the Swedish company Nord-Lock AB.
37. From the Web site of Key Belleville, Inc.
38. Nassar, S.A. and B.A. Housari Self-Loosening of Threaded Fasteners Due to Cyclic Transverse Loads, Paper PVP2005-71577, Proceedings of PVP2005, 2005 Pressure Vessel and Piping Division Conference, July 17–21, 2005, Denver, CO, USA.

15 Fatigue Failure

A metallic part subjected to cyclic tensile loads can suddenly and unexpectedly fail—even if those loads are well below the yield strength of the material. The part has failed in fatigue. Note that the failure occurs under tensile loads. I've heard that fatigue failure under cyclic compressive loads is possible—but is rare—so we'll ignore it.

Since failure only occurs under tensile loads, only the bolts (but not the joint members) in tension joints and only the joint members (but not the bolts) in shear joints can and do fail in fatigue. We'll devote most of this chapter to the fatigue failure of bolts; but will also look briefly at shear joints before we move on.

Fatigue failure of a single bolt means a reduction in clamping force. This in turn can increase the load excursions seen by the rest of the bolts, as we'll see later in this chapter, and that can encourage them to fail too. As a result, fatigue failure often means the complete loss of the joint.

15.1 FATIGUE PROCESS

15.1.1 Sequence of a Fatigue Failure

We learned in Chapter 13 that fatigue will be a potential problem only if four "essential conditions" are present: cyclic tensile loads, stress levels above a threshold value (called the endurance limit), a susceptible material, and an initial flaw in that material. If these conditions are all present, then a natural sequence of events can occur, and can lead to fatigue failure. These events are called

1. Crack initiation
2. Crack growth
3. Crack propagation
4. Final rupture

This sequence of events is shown in Figure 15.1. A tiny crack grows slowly, then more rapidly, until the bolt is destroyed, as shown in Figure 15.1. Let's examine this sequence of events one at a time.

15.1.1.1 Crack Initiation

Many things can produce that first fatal flaw which starts the fatigue process. A tool mark can do it. So can a scratch produced when the part is mishandled. Improper heat treatment can leave cracks. Corrosion can initiate them. Inclusions in the material can do it. It is probably safe to say, in fact, that no part is entirely free from tiny defects of this sort.

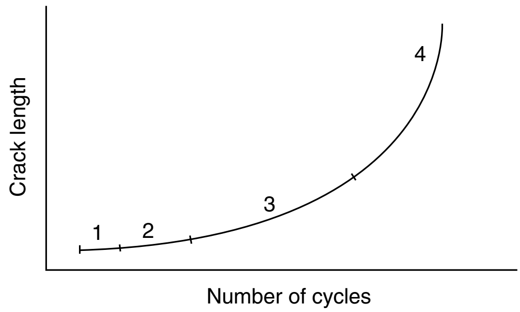


FIGURE 15.1 Fatigue failure occurs when a tiny crack in the bolt grows under cyclic tension loads until the crack is so large that the next cycle of load breaks the bolt. The stages of failure are (1) initiation, (2) growth, (3) propagation, and (4) rupture. (Modified from Keith, G., Standco Industries, Inc., Houston, Texas, Personal communication, 1979.)

15.1.1.2 Crack Growth

A tiny crack creates stress concentrations. When the part is subjected to cyclic tension loads, these stress concentrations yield and tear the material at the root of the crack. Since most of the bolt still remains undamaged to support the load, initial crack growth is fairly slow.

15.1.1.3 Crack Propagation

As the crack grows, stress levels at the end of the crack also increase, since less and less cross-section is left to support the loads. The crack grows more rapidly as stress levels increase.

15.1.1.4 Final Rupture

There comes a time when the crack has destroyed the bolt's capability to withstand additional tension cycles. Failure now occurs very rapidly. As far as the user is concerned, failure has been sudden and unexpected because, until this part of the fatigue process is reached, there is often no visible damage or change in the behavior of the bolt. Everything appears to be fine until suddenly, with a loud bang, the bolt breaks.

The number of cycles required to break the bolt this way is called its fatigue life. Apparently identical bolts in apparently identical applications can have, of course, substantially different fatigue lives, depending on the location and seriousness of those initial cracks as well as on apparently minor, but important, differences in such things as bolt and joint stiffness, initial preload, alloy content, heat treat, location and magnitude of external tension loads, etc. As a result, there is a lot of scatter in the fatigue life of the bolts used in a given application.

15.1.2 Types of Fatigue Failure

Fatigue failures are called high-cycle or low-cycle failures, depending on the number of load cycles required to break the part. High-cycle fatigue requires hundreds of thousands or even millions of cycles before rupture occurs. Low-cycle failure occurs in anything from one to a few ten thousand cycles. You can demonstrate low-cycle fatigue to yourself by bending a paper clip back and forth until it breaks.

The number of cycles required to break a bolt is determined by the magnitudes of mean and alternating stresses imposed on the bolt by external cyclic loads, as we'll see in a minute. Low-cycle failure occurs under very large loads, high-cycle failure under lesser loads.

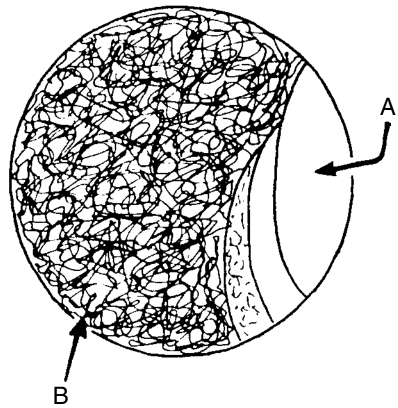


FIGURE 15.2 Break surface of a bolt which has failed in fatigue. (A) The surface is smooth and shiny in those regions which failed during crack initiation and growth. (B) It is rough in those regions which failed rapidly.

In many applications the bolt can see some of each—lots of relatively mild loads interrupted once in a while by a sudden shock or larger load (perhaps when the tractor hits a rock). In many cases it's difficult to know whether to characterize the failure as a low-cycle or a high-cycle failure. In most well-designed bolted joints, however, fatigue failure, if it occurs at all, will be high cycle.

15.1.3 Appearance of the Break

Close examination of the broken bolt can often tell you whether or not it failed in fatigue. That portion of the break surface which failed slowly, as the crack initiated and grew, will have a relatively smooth and shiny surface. That portion which failed during crack propagation will have a rougher surface; that portion which failed during final rupture will have a very rough surface. If the entire fastener fails suddenly during tightening or the like, the entire break surface will be rough; so these smooth "beach marks" seen on a fatigue surface can be used to distinguish fatigue breaks from breaks which occur under static load, see Figure 15.2.

You may find more than one crack in a bolt which has failed in fatigue. The initiation and growth of one crack may drastically increase loads in another region of the fastener, causing a second crack to grow and propagate there. Failure can occur in whichever one reduces the strength of the bolt more rapidly.

The most common places to find fatigue cracks and failures in bolts are in the regions of highest stress concentration. These are

where the head joins the shank of the bolt,
the thread run-out point,
the first thread or two of engagement in the nut, and
any place where there is a change in diameter of the body or shank.

15.2 WHAT DETERMINES FATIGUE LIFE?

In general, the higher the cyclic loads seen by the bolt, the sooner it will fail. Whether or not, or how rapidly, a fastener will fail depends on the mean stress level and the variation in stress level under cyclic loads.

There are techniques for estimating what the life of a given material or body will be. Accurate prediction, however, is possible only through actual experiments on the body of interest—in our case, on a bolt. Test results are usually presented in the form of *S–N* diagrams, where *S* stands for stress level and *N* for number of cycles of applied load. An examination of these diagrams gives considerable insight into the fatigue process.

15.2.1 *S–N* Diagrams

Figure 15.3 shows one possible form of the *S–N* diagram. Alternating tension and compression loads have been applied to the test specimen. Maximum compression stress equals maximum tension stress. Maximum amplitude of either stress is plotted on the vertical axis of the diagram. The number of cycles required to fail the test coupon is plotted on the horizontal axis. The curve shows the mean life of the test coupons.

Because the fatigue life of one test coupon may differ drastically from that of others, it is necessary to test many coupons before plotting the results shown in Figure 15.3. The statistical deviations in life can also be determined by such tests. A more complete picture of the tests, therefore, would be shown by a diagram such as that given in Figure 15.4. Note that many of the test specimens will fail at some number of cycles less than the mean. The remainder will fail at some number of cycles greater than the mean. If the lowest line in Figure 15.4 represents the minus two standard deviation data, then 95% of the test coupons will survive more cycles before failure than the number of cycles indicated by this line. Only 5% will last longer than the number of cycles indicated by the uppermost line.

Note that either Figure 15.3 or Figure 15.4 says that cycle life will be very short when applied alternating stress levels are very high. As alternating stresses are reduced, cycle life increases. Below some stress level, in fact, the curve becomes essentially parallel to the horizontal axis, and fatigue life becomes very large. This stress level is called the "endurance limit" of the material, or part, and is defined as the completely reversing stress level below which fatigue life will be infinite. There is some such limit for any material and any part. Unfortunately, endurance stress levels are usually only a small fraction of the static yield strength or static ultimate strength of a material or body.

Note, however, that only the change in stress must stay below the endurance limit. The total or mean stress can be considerably higher [5]. Incidentally, not all materials have an

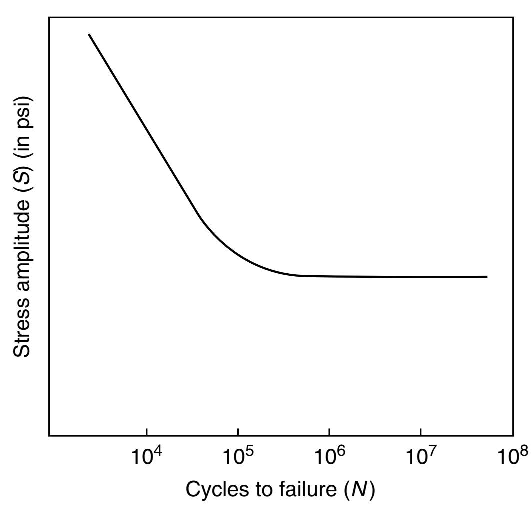


FIGURE 15.3 The mean life of a group of test coupons subjected to fully alternating stress cycles. When stresses are "full alternating," maximum tension stress equals maximum compression stress and the mean stress on the part is zero. (Modified from Wayne, D.M., Fatigue design considerations in bolted joints, presented at Using Threaded Fasteners Seminar, University of Wisconsin-Extension, Madison, WI, April, 1979.)

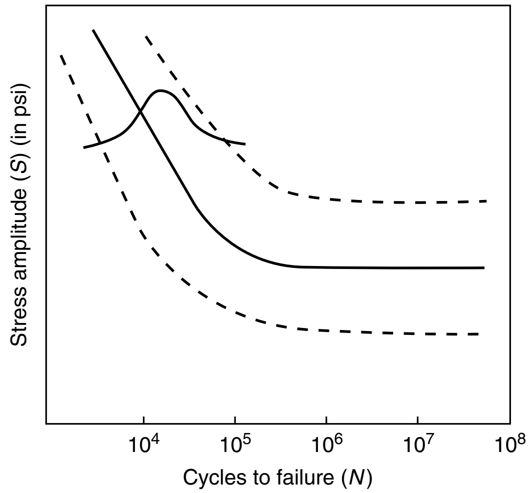


FIGURE 15.4 There will be considerable scatter in the life achieved in a group of test coupons as a result of a particular stress pattern. Rather than show just the mean life results as in Figure 15.3, therefore, it is sometimes useful to plot the statistical deviations as well. (Modified from Wayne, D.M., Fatigue design considerations in bolted joints, presented at Using Threaded Fasteners Seminar, University of Wisconsin-Extension, Madison, WI, April 1979.)

endurance limit. Aluminum alloys, for example, exhibit finite life even at very low cyclic stress levels [18].

We could now test another large number of test specimens, this time changing the mean tension while leaving the excursion (difference between maximum and minimum tension) the same as it was in the previous tests. This would result in a family of curves such as that shown in Figure 15.5. For clarity, only the mean curves are shown.

Although all the *S–N* data we have examined are based on tension (and compression) loading along the axis of the fastener, it is worth noting that if the fastener is subjected to some other form of stress as well as tension, its fatigue life will be adversely affected.

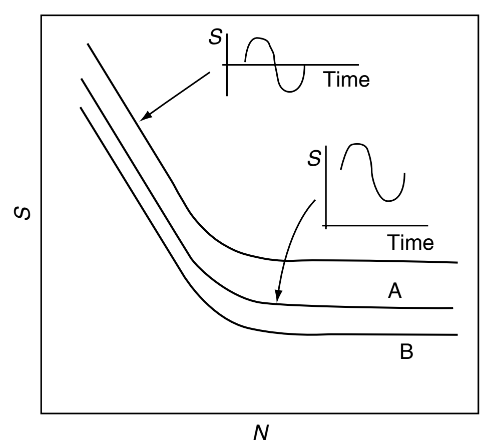


FIGURE 15.5 Increasing the mean stress will reduce the number of cycles to failure produced by a given magnitude of alternating stress. The uppermost curve here is repeated from Figure 15.3 for comparison. The mean stress associated with curve B is higher than that associated with curve A. (Modified from Wayne, D.M., Fatigue design considerations in bolted joints, presented at Using Threaded Fasteners Seminar, University of Wisconsin-Extension, Madison, WI, April 1979.)

Shear stress, for example, would rob a portion of the strength of the fastener, making it more susceptible to tension fatigue. Bending stress, which is often present, magnifies the tensile stress on one side of the bolt and can also be a significant problem in a fatigue situation.

15.2.2 Material versus "The Part"

If we were to test a bunch of coupons made from a different material or subject it to a different heat treatment, we would, in general, generate a set of curves that would be different from those shown in Figures 15.3 through 15.5. Fatigue life, in other words, is a function of material and heat treatment. It is also—and perhaps even more so—a function of the shape of the part being tested, just as the stress–strain performance of a body was different from the stress–strain performance of the material from which it is made (see Chapter 2).

The reason in both cases, of course, is that the shape of the body determines stress levels. These vary from point to point; the behavior of the body, therefore, varies from point to point. The gross behavior of the body is determined by the accumulation of its point-to-point behavior.

A bolt is a very poor shape when it comes to fatigue resistance. Although the average stress levels in the body may be well below the endurance limit of the material, stress levels in unavoidable stress concentration points such as thread roots, head-to-body fillets, etc. can be well over the endurance limit. As a result, the apparent endurance limit of the commercial fasteners can be as little as 10% of the endurance limit of the base material [3]. One source gives the endurance limit of a Grade 8 fastener, for example, as 18,000 psi [4], well below its proof strength of 120,000 psi.

Another reference says that the fatigue strength of a smooth test bar of steel is approximately half the ultimate tensile strength (UTS) of the steel, if the steel has a UTS under 200 ksi (which most bolt materials do) and if the test is conducted under fully reversing loads (defined as $R = 1$ as we'll see in Section 15.3 below). The reference goes on to say that the fatigue limit of the part under test will be less than half the UTS of the material if any of the following conditions are present:

Part is notched or threaded.
There are residual tensile stresses at the surface of the part.
Part has been electroplated.
Part is corroded.
There has been mechanical damage to the surface.

All of which says "bolt." The reference concludes that if you can't estimate the influence of these factors, then the only way to determine the fatigue strength of the part is to conduct *S–N* tests under service conditions [18]. We'll look at some actual fatigue strength data for fasteners in a minute. First, though, let's continue our review of fatigue in general.

15.2.3 Summary

In summary, the major factors which affect fatigue life are the following:

1. Choice of material
2. Shape of the part
3. Mean stress level
4. Magnitude of stress excursions or variations
5. Condition of the part

Of these, the shape of the part may be the most significant, magnitude of stress excursions the next most significant, and, within reason, material choice the least significant.

15.3 OTHER TYPES OF DIAGRAM

15.3.1 CONSTANT LIFE DIAGRAM

The *S–N* diagram is only one way to plot the results of a series of fatigue tests. Another more informative diagram is called a constant life diagram (Figure 15.6). Because of the amount of information on this diagram, it takes a little practice to read it. Let's take an example.

The curved lines marked 10^3 cycles, 10^4 cycles, etc. represent average, constant, fatigue lives under a variety of conditions.

They intersect the mean stress line at a common point, which is equal to the ultimate tensile strength (S_u) of the material being tested—100 ksi in the example in Figure 15.6. The data shown, incidentally, are for a group of unnotched, polished test specimens of 100 ksi material. Most, if not all, of the constant life diagrams you'll find are for polished, unnotched test specimens of this sort rather than for bolts or particular shapes.

Working now with the curved line in Figure 15.6 which represents 10^6 cycles, we learn that the average test coupon will have this life when it sees a maximum tensile stress of 80 ksi (vertical axis of the chart) and a minimum tensile stress of 30 ksi (horizontal axis). A variety of other combinations of maximum and minimum stress would also result in a coupon life of 10^6 cycles, but we'll focus on the 80⁄30 point. This point also represents a load ratio (R) of 0.375—the ratio between the minimum and maximum tensile stresses on the part. The mean stress on the part, at this point, is 55 ksi, which we determine merely by adding the maximum to the minimum and dividing by 2.

The total variation in stress on the part is 50 ksi; the difference between the maximum 80 ksi and the minimum 30 ksi. The alternating stress for this situation is half of 50 or 25 ksi, a fact which is perhaps best illustrated by Figure 15.7, described next.

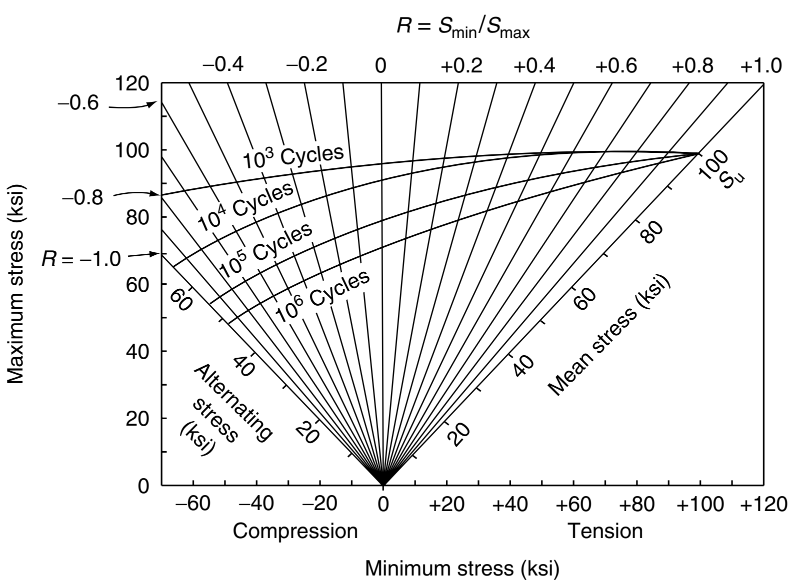


FIGURE 15.6 A constant life diagram, the most informative of all fatigue diagrams. See text for discussion.

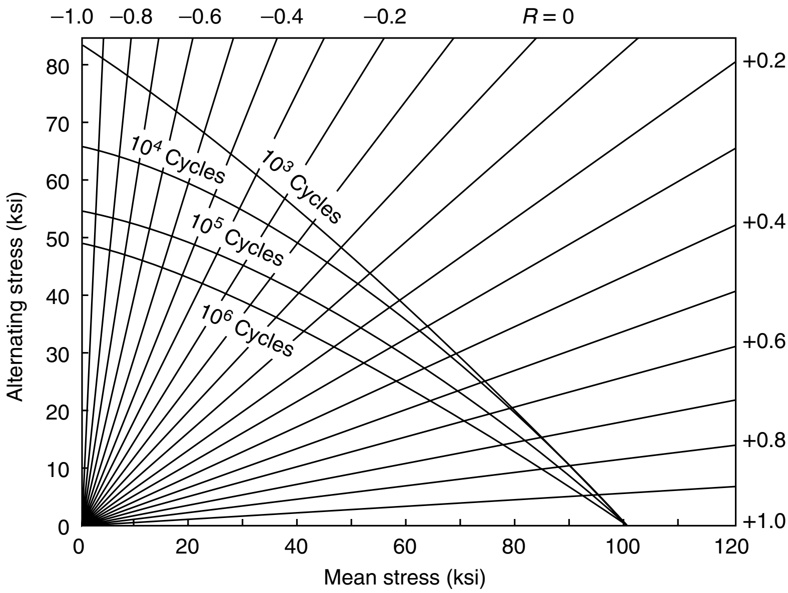


FIGURE 15.7 A simplified or modified constant life diagram; this one consists of only the center portion of the diagram shown in Figure 15.6.

15.3.2 Center Portion of Constant Life Diagram

Sometimes, only the center portion of the constant life diagram is given, as in Figure 15.7, with the alternating stress and mean stress lines now forming the axes of the diagram. The only information which is missing from this diagram, maximum and minimum applied stresses, can be computed from the plotted values for mean and alternating stress. The lines representing the various load ratios (R), which I have shown in Figure 15.7, are often omitted from this type of diagram.

The concept of load ratios, incidentally, is illustrated in Figure 15.8. In part A, the load is static and there would be no fatigue problems. In part B the load varies (fluctuates) slightly. The load fluctuations are progressively more severe in parts C, D, E, and F, with F being the worst situation from a fatigue point of view. This situation, where the maximum positive stress equals the maximum negative stress, is called a "completely reversing load" and represents a load ratio of -1.0. Note that a load ratio of -1.0 is represented by the vertical, alternating stress axis in the diagram of 15.7.

15.3.3 Approximate Constant Life Diagram

Note that the constant life lines labeled 10^3, 10^4, etc. in Figures 15.6 and 15.7 are nearly straight lines. This allows us to construct an approximate but conservative constant life diagram, as illustrated in Figure 15.9.

To do this, we first make a series of tests in which the mean stress is always zero. Only the magnitudes of the alternating stresses are varied. If we plotted the results on an S–N diagram, it would look like the diagram in the left side of Figure 15.9. As in all fatigue tests, we would have to test a number of specimens to get a true mean value—there will be a lot of scatter in individual results.

Having done this, we can now plot the average alternating stress for each mean life point of interest on the vertical axis of our modified constant life diagram, as shown on the right

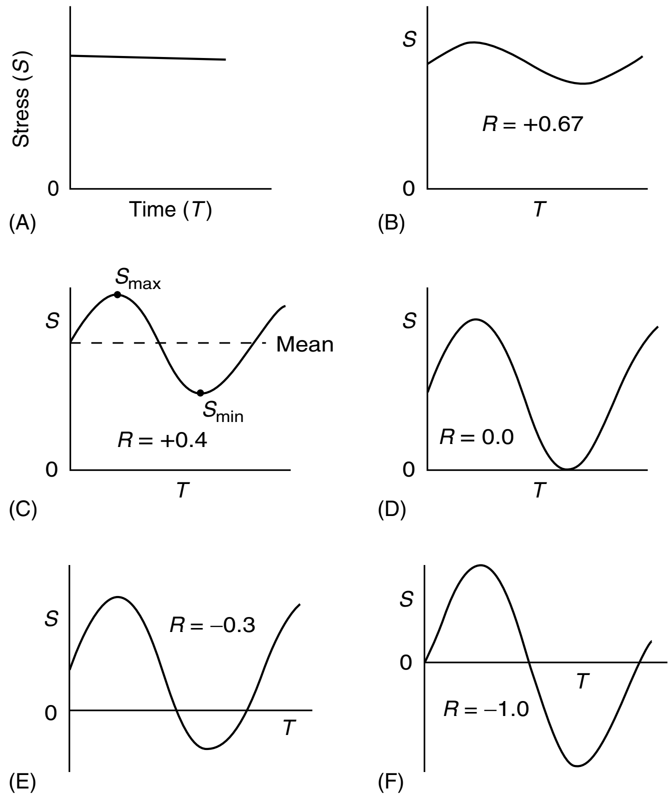


FIGURE 15.8 Graphical representation of a variety of fatigue loading conditions. In (A), the loads are static, and there will be no fatigue problem. (B) shows a slight cyclic fluctuation in load. (C) through (F) show progressively more severe loads. This in (F) is called a completely reversing load. This is the type of load used to determine an endurance limit.

of Figure 15.9. We next draw straight lines between these alternating stress points and the point on the horizontal (mean stress) axis which represents the ultimate tensile strength of the material (S_u). Since these straight lines will always lie below the actual, slightly curved lines, they are safe and conservative approximations of the actual lines. We don't have to make any tests at load ratios other than -1.0 to construct this diagram.

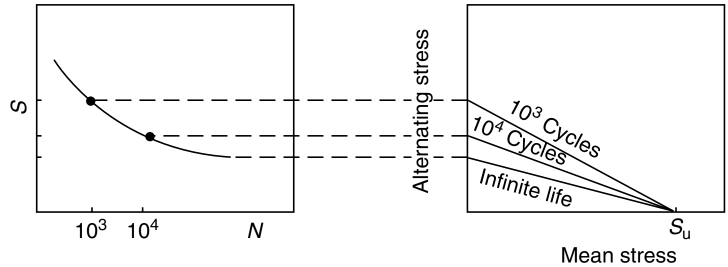


FIGURE 15.9 An approximate (conservative) constant life diagram is shown on the right. It can be constructed from the data used to draw the mean fatigue life line on an S–N diagram, as shown here.

15.3.4 Endurance Limit Diagram

The line representing infinite life in the diagram on the right side of Figure 15.9 would seem to define, fully, the conditions required for infinite product life. After all, fully alternating stress is the worst condition from a fatigue standpoint. This line, however, omits one factor; namely, that the safe "static" load which can be applied to a part is not its ultimate tensile strength, but is rather its yield strength. We must take this fact into account when predicting true "infinite life."

To do this, we construct the diagram shown in Figure 15.10. We start by repeating the infinite life line, but this time we lower it a little to represent the worst-case condition. Remember from Figure 15.4 that there will always be a considerable scatter in the life achieved in a group of test specimens. For the infinite life diagram, we want to plot the equivalent of the lower dashed line in Figure 15.4 rather than the mean line we used in Figure 15.9. Let's assume for the discussion that the worst-case line was 20% below the mean line.

There will also be some scatter, of course, in the yield and ultimate strengths of a material. We'll use the same 20% reduction for these values. We're now ready to construct our final infinite life diagram.

Instead of the original infinite life line, we now connect a point representing 80% of the mean infinite life alternating stress to 80% of the ultimate tensile strength—line A in Figure 15.10. We now connect points representing 80% of the yield strength of the material on both horizontal and vertical axes—line B in Figure 15.10.

The shaded region in the figure represents the true infinite life of the part. Any combination of mean and alternating stresses which fall within this region will never cause a fatigue failure. Note, however, that we're still dealing with data taken from tests on polished, unnotched test coupons rather than bolts since all of our data have been based on data obtained from the diagram of Figure 15.7. Had we instead conducted the original tests on actual bolts, then the final diagrams would have represented the infinite life conditions for the bolt itself.

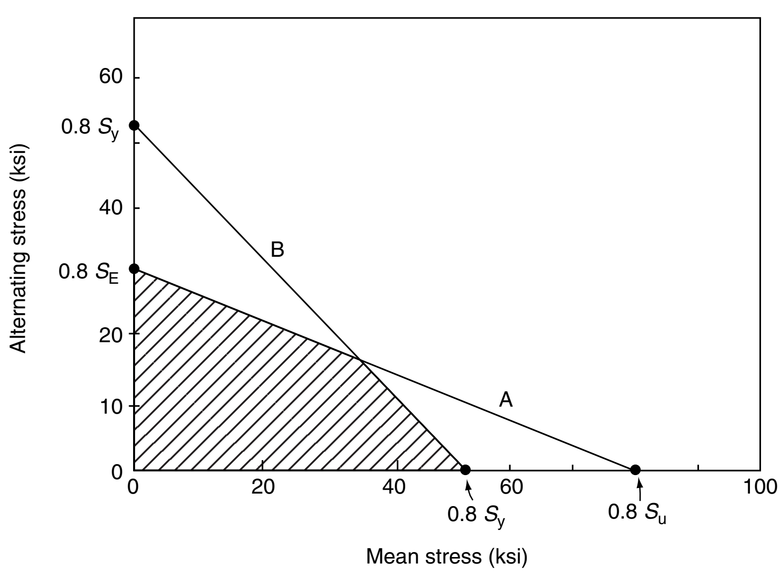


FIGURE 15.10 An infinite life diagram. Line A is a "worst-case" infinite life line, similar to the one shown on the right side of Figure 15.9. Line B connects points representing the worst-case yield strength (S_y) of the material on both axes. Any combination of mean and alternating stress which falls within the shaded region would never fail the part.

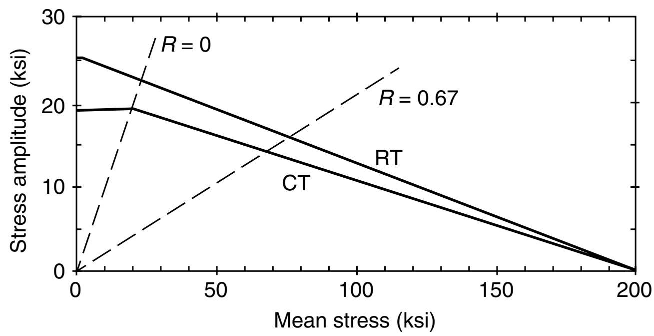


FIGURE 15.11 A constant life fatigue diagram for SAE Grade 8 bolts at 10 million cycles of life. This chart is based on a literature search conducted by the authors of Ref. [19]. Line RT represents fasteners having rolled threads; CT those with cut threads.

15.3.5 Fatigue Life Data for Fasteners

I have found specific fatigue life or endurance limit data for threaded fasteners hard to come by. I included what little I had until recently in Table 2.11. Both endurance limits and fatigue strengths (maximum stress excursions the bolts can stand for a given number of cycles) are included. Figures 15.11 and 15.12 summarize data which have been available for some time but are new to me. Figure 15.11 gives fatigue data for SAE Grade 8 bolts loaded 10×10^6 times [19] and Figure 15.12 gives data for ASTM A325 and A354-BD bolts used in structural steel applications [20]. The tests reported were stopped after 2×10^6 cycles, but the author says that studies have shown that if heavy structural steel joints survive that many cycles their life can be considered infinite.

Hess reports that the fatigue life of bolts tightened past yield was greater than the life of those tightened within the elastic range [28] presumably because higher preload means less variation in bolt stress under cyclic loads.

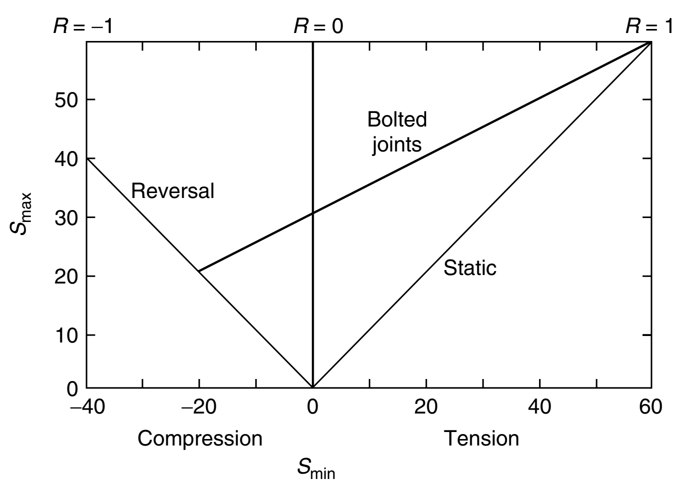


FIGURE 15.12 A modified Goodman diagram for structural steel bolts ASTM A325 and A354-BD subjected to 2 million load cycles. According to Ref. [20], a heavy structural steel which survives these many load cycles is considered to have infinite life.

15.4 INFLUENCE OF PRELOAD AND JOINT STIFFNESS

15.4.1 Fatigue in a Linear Joint

As we saw in Chapter 10, the bolt will see a portion of any external tension load which is imposed on the joint. The magnitude of the mean load on the bolt depends on the preload in the bolt. The magnitude of the load excursion (ΔF_B) depends on

Magnitude of the external tension load
Bolt-to-joint stiffness ratio (K_B/K_J)

whether or not the external tension load exceeds the critical load required to separate the joint (which is determined by the magnitude of the initial preload).

The effect of the first two factors is summarized in Figure 15.13.

We could also use the triangular joint diagram to show the effects of very large external loads and insufficient preload. I think it's more instructive, however, to use the form of the alternative joint diagram given in Figures 10.18 and 10.19, in which we plotted the bolt load as a function of external load and could readily see what happens to the bolt load when the external load exceeds the critical level required for joint separation.

In Figure 15.14, for example, we apply external loads to two joints having the same initial preload and the same stiffness ratios as each other. Furthermore, the excursion (difference between maximum and minimum) of the external load is the same in both cases. Only the values of the maximum and minimum loads have changed. In Figure 15.14A, the maximum external load is less than that which would be required for joint separation. The resulting excursion in bolt load (ΔF_B) is relatively small.

In Figure 15.14B, however, the maximum external load exceeds the critical load. The bolt sees 100% of any external load that exceeds the critical level and so, under these circumstances, the excursion in bolt load is greatly increased.

The critical load depends on the initial preload. If we lower preload, we can get into trouble, as shown in Figure 15.15. Conversely, of course, raising the preload in the joint can get us out of trouble.

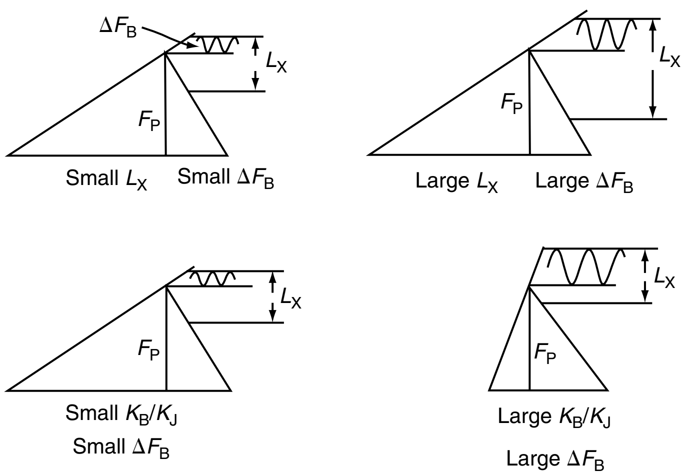


FIGURE 15.13 The load excursions (ΔF_B) in the bolt are increased with an increase in external load (L_X) or an increase in the bolt-to-joint stiffness ratio (K_B/K_J). Note that the initial preload is the same in each case.

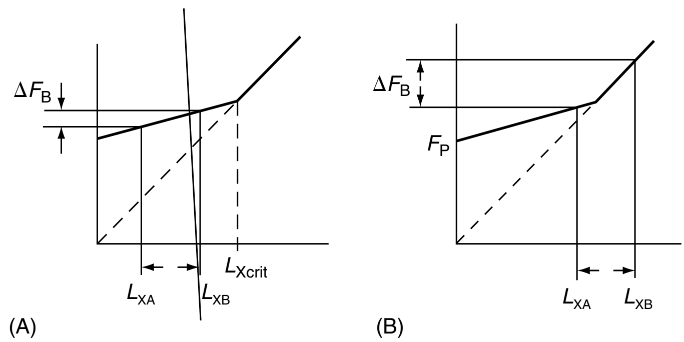


FIGURE 15.14 This bolt sees a far greater variation in tension (ΔF_B) if the external load exceeds the critical load required for joint separation (as in B) than it does when external loads are less than the critical value (A). Note that initial preload and joint stiffness ratio is the same in both cases. (Modified from Wayne D.M., Fatigue design considerations in bolted joints, presented at Using Threaded Fasteners Seminar, University of Wisconsin-Extension, Madison, WI, April 1979.)

15.4.2 Nonlinear Joints

The above analysis was based on the assumption that the joint will behave in a linear and fully elastic fashion. As we saw in Chapter 11, this is not always the case—in fact, it may very seldom be the case. If the external load, for example, is not applied along the axis of the bolt or the bolt is not located in the center of the joint, prying action can increase the load seen by the bolt. This can make a substantial difference in the load excursions produced in the bolt by a given cyclic external load, as suggested in Figure 15.16.

With reference to Figure 15.14, we can sometimes reduce the load excursions seen by the bolt by increasing the preload. This only works, however, if the new, higher preload raises the critical load required for joint separation above the maximum external load seen by the joint. If the maximum external load was already below the critical level, increasing the preload does not reduce the excursion seen by the bolt but merely increases the mean stress in the bolt, as shown in Figure 15.17.

Increasing the preload in an eccentric prying joint will also increase the mean tension seen by the bolt. This time, however, because of the nonlinear nature of the joint, it's likely that increasing the preload will also reduce the load excursion seen by the bolt, as suggested in Figure 15.18. Under these conditions, increasing the preload can result in a net gain in

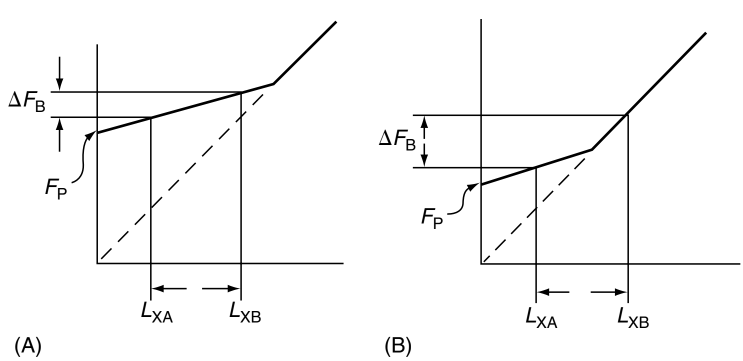


FIGURE 15.15 The maximum and minimum external loads are the same in both cases here. The maximum load, however, exceeds the critical load required for joint separation in B because of insufficient initial preload (F_P). Note that the joint stiffness ratio is the same in both cases.

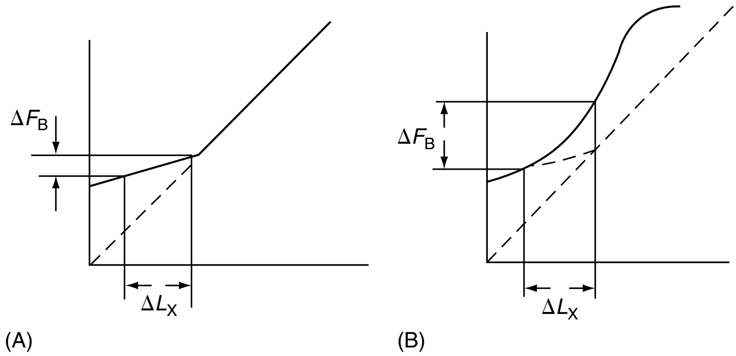


FIGURE 15.16 Comparison of the loads seen by a bolt in a linear concentric joint (A) and an eccentric joint in which the bolt is subjected to prying action (B). Note that the initial preload, the bolt–joint stiffness ratio, and the maximum–minimum external loads are the same in both cases. At least the apparent stiffness is the same. The fact that prying action occurs alters the stiffness of the joint.

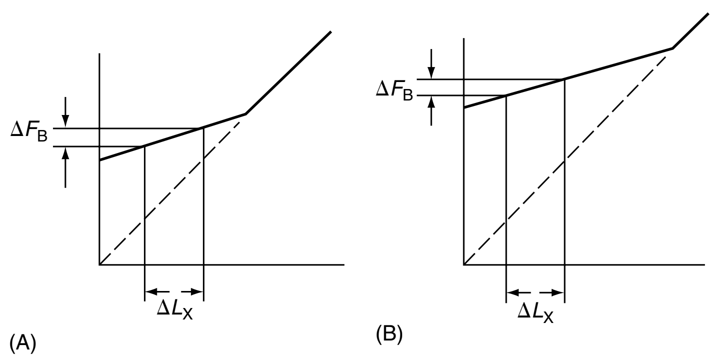


FIGURE 15.17 If the maximum external load (L_{XB}) is already below the external load required for joint separation, then raising bolt preload will not reduce the load excursion (ΔF_B) seen by the bolt; it will merely increase the mean load.

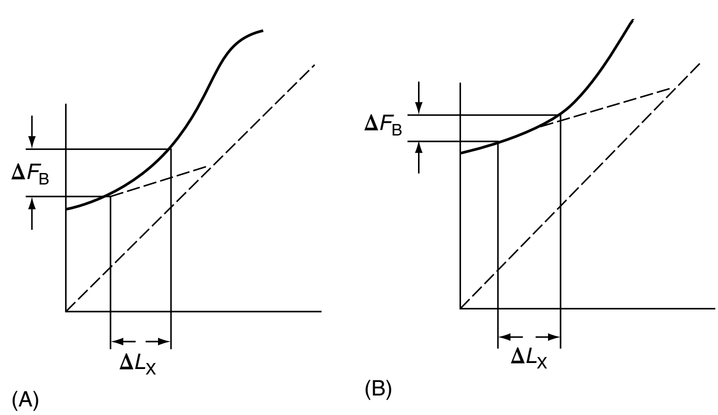


FIGURE 15.18 Increasing the preload in an eccentric nonlinear joint subjected to prying action will increase the mean load seen by the bolt, but will also reduce the load excursions it sees. Since large excursions are worse than large means, as far as fatigue life is concerned, this change can result in a net gain in fatigue life.

fatigue life. The increase in mean stress is detrimental but is more than offset by the reduction in excursion.

15.4.3 What Is the Optimum Preload?

We're now in position to answer the question "What preload should we use for maximum fatigue life?"

In general, greater load excursions reduce fatigue life more than higher mean load—but neither is helpful. As a result, we can conclude that:

1. A higher preload will help if it reduces bolt load excursions substantially. Higher preload will therefore always help if it raises the critical load required for joint separation above the maximum external load which will be seen by the joint. Using a bolt material having a higher UTS or using a bolt with a larger diameter (even though that means a stiffer bolt) will allow you to increase the initial preload and so reduce the alternating stresses on the bolts by preventing the partial opening of the joint under prying action [5,21]. Higher preload will also help if it reduces the amount of prying experienced by the joint.
2. A higher preload will reduce fatigue life if it makes no change in the load excursions seen by the bolt (see Figure 15.17).
3. A higher preload is probably neither good nor bad if it doesn't reduce the load excursions by very much. I suspect, in other words, that there are gray areas where results could go either way.

SPS has published data which suggest that higher preload, up to the yield point of the fastener, is always desirable in a fatigue situation because they also believe that prying is (almost) always present as well. Some of their results are shown in Figure 15.19 [5,22], which shows how prying action can adversely affect the fatigue life of a joint and how an increase in preload can improve fatigue behavior.

In another project SPS tested automotive connecting rod joints initially tightened well past yield. They subjected the joints to 10^6 load cycles without failure—even with extreme plastic deformation of the bolts. In other tests they obtained acceptable fatigue lives from 3/8–16 industrial bolts which had been yield tightened as much as three times. They report that yield tightening did decrease the fatigue life of the bolts slightly, and that the benefits of rolling the threads after, instead of before, heat treatment were reduced somewhat by yield tightening; but that, in general, yield tightening was beneficial [23].

Anyway, many people—including those at SPS—insist that higher preload will always improve fatigue behavior. Others, however, argue that since a higher mean will reduce fatigue life, though not as much as a higher excursion will, a higher preload can be helpful or may not be so [6], and only a careful analysis will answer the question.

15.4.4 Fatigue and the VDI Joint Design Equations

We last examined the VDI joint design equations in Chapter 11. Now it's time to see what they tell us about joint failure. At first glance the answer is, "not much." Equation 11.21, for example, tells us how to compute the maximum anticipated tensile stress to be seen by a bolt, as a function of assembly preloads, relaxation factors, thermal change, and the like. Most of the factors included in that equation won't fluctuate, and therefore will only affect the mean preload seen by the bolts. That can influence fatigue life but we're far more interested in fluctuations in load, if any.

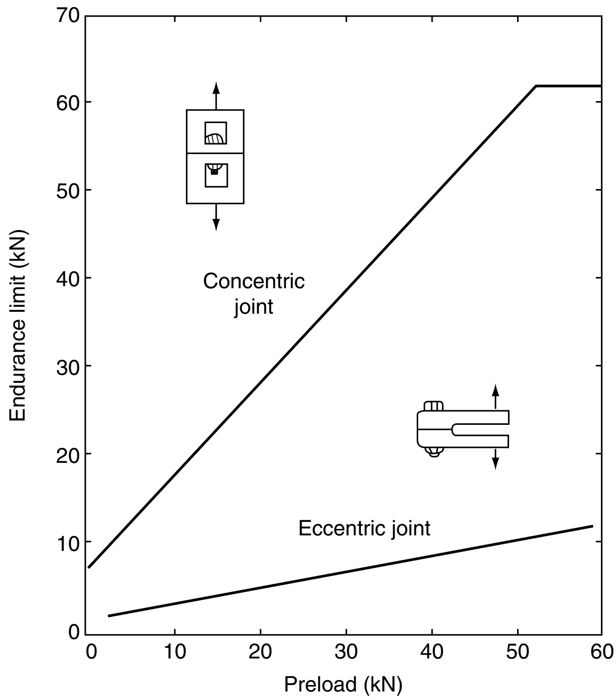


FIGURE 15.19 This chart, based on tests sponsored or conducted by SPS, shows that the endurance limit of a joint is increased if the initial preload applied to the joints is increased. It also shows that concentric loading results in much better fatigue performance than eccentric loading. The fasteners used in these tests were M10 $\times$ 65, Grade 12.9 bolts. The length of the concentric joint was 50.8 mm and its contact area was 38 mm^2. The contact area of the eccentric joint was 38 $\times$ 38 mm and the distance between the centerline of the bolts and the line of action of the applied load was 78.9 mm—a substantial eccentricity. (From Hood, A.C., Factors affecting the fatigue of fasteners, Attachment 7, presented at a meeting of the Bolting Technology Council, in Cleveland, OH, April 19, 1993.)

There is one term in Equation 11.21 which addresses a change in tension in the bolt caused by external load, namely, the ΔF_B term given as $\Phi_{en} L_X$ where Φ_{en} is the load factor (sometimes called the joint stiffness ratio) for a prying joint (with the axes of the bolt and the line of action of the tensile load both offset from the axis of gyration of the joint), and L_X is the external load placed on the joint. Other Φs also discussed in Chapter 11—for concentric or nonprying joints etc.—could be multiplied by L_X to compute ΔF_B for other situations, of course.

Now, the ΔF_B used in Equation 11.21 was described as the result of a static L_X exerted on the joint. If, however, the external load fluctuates between some L_X and zero ($R = 0$) as illustrated in Figure 15.14, then ΦL_X can be taken as the stress excursion seen by the bolt. If the external load fluctuates between an L_{Xmax} and an L_{Xmin} other than zero, then the difference between those values must be multiplied by the appropriate Φ to compute the excursion seen by the bolt.

The "stress amplitude" of this load excursion is now compared to the endurance limit of the bolt (if infinite life is desired) or to its fatigue strength for a desired number-of-cycles life. The stress amplitude is the difference between mean stress and maximum stress, as illustrated in Figure 15.20, or one-half the fluctuating L_X. In VDI terms we have, if we want infinite life,

$$\Delta F_{Ba} = (F_{Bmax} - F_{Bmin})/2 = \Phi L_{Xa}/2 \tag{15.1}$$

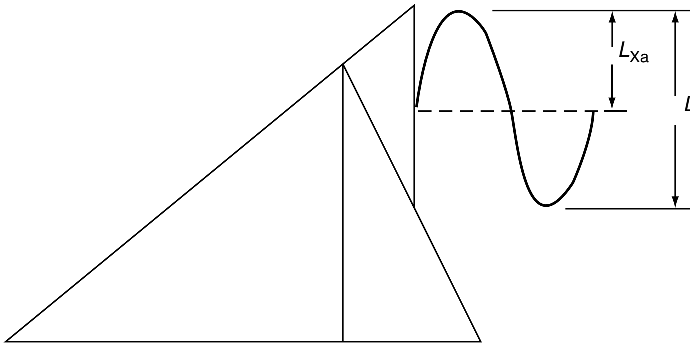


FIGURE 15.20 This joint diagram shows the relationship between the full excursion of the tensile–tensile (nonreversing) load L_X and the half excursion L_{Xa} called the stress amplitude in the German VDI Directive 2230. Most of the fatigue data I've seen have been based upon tests conducted under fully reversing loads ($R = -1$). But VDI here used half of a fluctuating tensile load in endurance limit calculations. Fatigue data are based on many different types of loading conditions and we're well advised to understand the basis of the data we're using for our own application.

and

$$\Delta F_{Ba} < \sigma A_r \tag{15.2}$$

where

A_r	= the root diameter area of the threads (in.^{2}, mm^2)
ΔF_{Ba}	= stress amplitude seen by the bolt (lb, N)
F_{Bmax} and F_{Bmin}	= maximum and minimum tensions in the bolt as a result of cyclic changes in the external load on the joint (lb, N)
L_{Xa}	= the fluctuating portion of the external joint on the joint (lb, N)
Φ	= load factor or joint stiffness ratio
σ	= the maximum stress in the outer fiber of the root of the first, load-bearing thread (psi, Pa).

You'll find the worst-case (eccentric prying) expression for σ in Equation 11.8.

Note that in most of this chapter the endurance limit has been based on tests in which a fully reversing load ($R = -1$) was applied to the parts. With these loading conditions the maximum stress seen by the bolts is indeed only half the change in stress, since the mean stress is zero. VDI Directive 2230 suggests that we compare the endurance limit to only half of a fluctuating stress, as shown Figure 15.21; even though the mean stress—thanks to preload—is certainly not zero. Their directive also contains a table of endurance limits for DIN steels which may be based on the same conditions. Certainly, most bolts don't see fully reversing stress cycles. But all of this should alert us to the fact that fatigue test data can be based on many different kinds of tests and should be applied with caution to our own applications if conditions differ.

In any event, the VDI 2230 aims at infinite life for every application, even when prying is present. A commendable goal, but, obviously, it won't always be economically practical to design for infinite life, especially under prying conditions. Of course, we'll always want to do it if practical and always must do it if joint failure would have serious consequences.

15.5 MINIMIZING FATIGUE PROBLEMS

You should realize that each of the things we can do to reduce or eliminate a fatigue problem is an attempt to overcome one or more of the four essential conditions without which failure

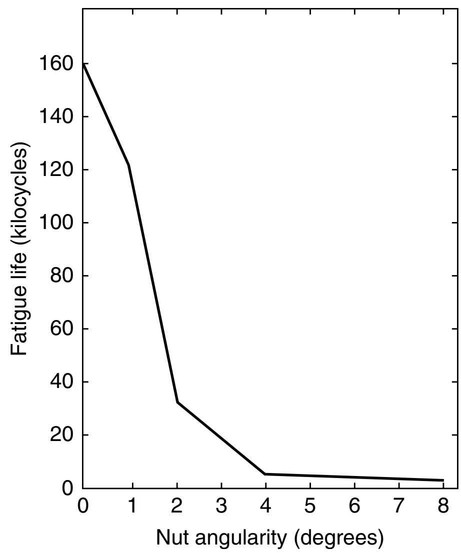


FIGURE 15.21 If the face of the nut is not exactly parallel to the surface of the joint, fatigue life can be seriously affected, as shown by this study made by Viglione [10]. Bolts were ⅜–24 MIL-B-7838 with 2 in. grip length.

would not occur. Remember that these conditions were cyclic tensile loads, stresses above an endurance limit, a susceptible material, and an initial flaw. We can rarely eliminate any one of these completely, but if we can at least reduce one or more of these factors we can usually improve the fatigue life of our bolted joints.

In general, most of the steps we can take are intended to reduce stress levels (including stress concentrations) or to reduce the load excursions seen by the bolt. Surface flaws and the susceptibility of the material will usually be the concern of the bolt manufacturer rather than the user. Let's look at some of our options.

15.5.1 Minimizing Stress Levels

The following are not listed in order of importance. They merely describe some of the many things which can be and are done to fasteners to limit stress concentrations and general stress levels. Some of them are relatively obvious; others are subtle. Many are incorporated in the so-called fatigue-resistant fasteners which are available from some manufacturers. In any event, here are some of the things which work.

15.5.1.1 Increased Thread Root Radius

Sharp, internal corners are natural places for fatigue cracks to start, so using threads with radiused roots can increase fatigue life. For example, going from a flat root thread to one with a radius equal to 0.268 times the pitch (a so-called 55% thread) increased the fatigue lives of various specimens from 80 to as much as 2,800% even though the change increased the tensile strength of the thread by only 1%–12% [22]. For illustrations of flat and rounded radius roots see Figure 3.2.

15.5.1.2 Rolled Threads

Rolling the threads instead of cutting them provides a smoother thread finish (fewer initial cracks). Rolling provides an unbroken flow of the grain of the material in the region of the threads, partially overcoming their notch effect, and it builds compressive stress into the surface of the bolt. This compressive "preload" must be overcome by tension force before the thread roots will be in net tension. A given tension load on the bolt, therefore, will result in a smaller tension excursion at this critical point (point to stress concentration).

Threads can be rolled either before or after heat treating. After is better but is also more difficult. Rolling before heat treating is possible on larger diameters.

In any event, one authority says that cold-rolled threads have double the fatigue strength of cut threads, although he doesn't specify whether the rolling occurs before or after heat treatment [24]. Others report that the higher the basic strength of the material, the greater the benefit of rolling after heat treat [25].

15.5.1.3 Fillets

A generous fillet between head and shank will reduce stress concentrations at this critical point. The exact shape of the fillet is also important; an elliptical fillet, for example, is better than a circular one [4].

Increasing the radius of a circular fillet will help. So will prestressing the fillet (akin to thread rolling) [7].

15.5.1.4 Perpendicularity

If the face of the nut, the underside of the bolt head, or the joint surfaces are not perpendicular to thread axes and bolt holes, the fatigue life of the bolt can be seriously affected. Figure 15.21, for example, shows the effect of a few degrees of nut angularity (lack of perpendicularity) on fatigue life. A 2° error reduces fatigue life by 79% [8].

15.5.1.5 Overlapping Stress Concentrations

Bolts normally see stress concentrations at thread run-out, first threads to engage the nut, and head-to-shank fillet. Anything which imposes additional load or concentration of load at these points is particularly damaging. Some such factors are shown in Figure 15.22. For best performance, for example, there should be at least two full bolt threads above and below the nut. Thread run-out should not coincide with the joint interface (where shear loads exist), etc.

15.5.1.6 Thread Run-Out

Thread run-out should be gradual rather than abrupt, as suggested in Figure 15.23.

15.5.1.7 Thread Stress Distribution

As we saw in Chapter 3, most of the tension in a conventional bolt is supported by the first two or three nut threads. Anything which increases the number of active threads will reduce stress concentrations and increase fatigue life. Some possibilities are suggested in Figures 3.9 through 3.11. The so-called tension nuts of Figure 3.9 create nearly uniform stress in all threads, as shown in Figure 3.10, for example.

Modifying the nut pitch so that it is slightly different from the pitch of the bolt threads can also make a substantial improvement in fatigue life. One authority [1] suggests that a nut with 11.85 threads per inch be used with a bolt having 12 threads per inch. He points out that this not only provides more uniform distribution of stress in the threads, but also reduces the stiffness of the bolt with respect to the joint by making the effective length of the bolt a little greater. Reducing the stiffness ratio helps, as we saw in Figure 15.13.

Another way to smooth stress distribution in the threads is to use a nut that is slightly softer than the bolt. The nut can now conform to the bolt more readily. Standard nuts are

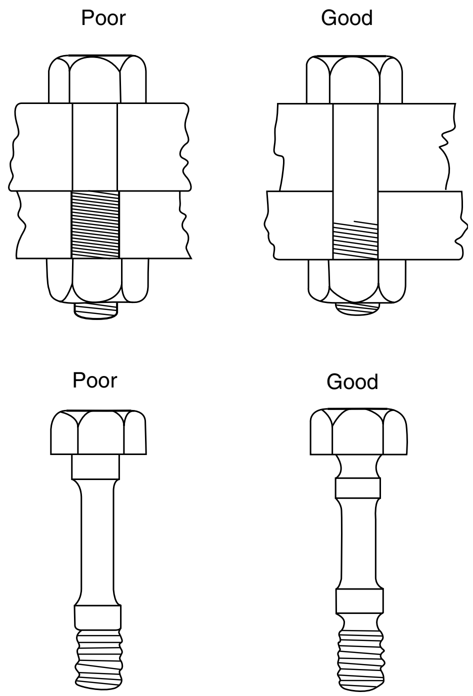


FIGURE 15.22 Joints should be designed so that maximum loads do not fall on stress concentration points of the fastener. Several points of good and bad practice are suggested.

softer than the bolts they're used with, for this reason; still softer nuts are possible if you can stand the loss in proof load capability.

A helical thread insert in a tapped hole also "conforms" to the male threads, because the insert is flexible; but the insert doesn't reduce static strength the way a soft nut will.

A jam nut improves thread stress distribution too, by preloading the threads in a direction opposite to that of the final load.

A final way to improve the distribution of stress in the threads is to taper them slightly, as shown in Figure 15.24. Tapering the lower threads of a nut at a 15° angle, until the first thread had been removed, for example, improved the fatigue life of one fastener by 20% [1]. You must

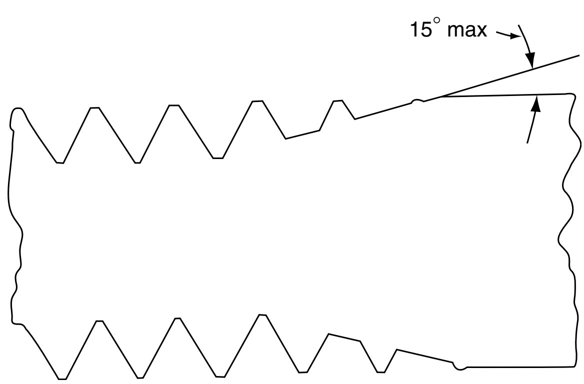


FIGURE 15.23 Thread run-out should be gradual—some people suggest a maximum of 15°—to maximize stress concentrations at this critical point in the fastener.

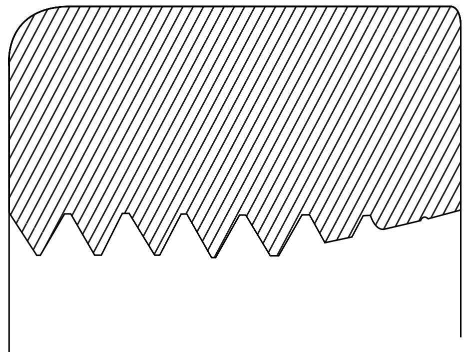


FIGURE 15.24 Tapering the input threads of the nut can distribute stresses more uniformly and increase fatigue life. The taper is 15° and is sufficient to just remove the first thread.

be sure to put such a nut on in the right direction, however, or it will increase stress concentrations and reduce fatigue life over that obtained with a conventional nut.

15.5.1.8 Bending

Bending increases the stress levels on one side of the fastener. This is one of the reasons why nut angularity hurts fatigue life. One way to reduce bending is to use a spherical washer.

15.5.1.9 Corrosion

Anything we do to minimize corrosion will reduce the possibilities of crack initiation and crack growth and will therefore extend fatigue life. This is confirmed by the fact that running the bolts in a hard vacuum results in an order-of-magnitude improvement in fatigue life [1] because it completely eliminates corrosion. Corrosion, as we'll see in Chapter 16, can be more rapid at points of high stress concentration. Since this is also the point at which fatigue failure is most apt to occur, fatigue and corrosion aid each other, and it is often difficult or impossible to tell which mechanism initiated or resulted in failure.

15.5.1.10 Flanged Head and Nut

At one time, the fastener shown in Figure 15.25 was being proposed as a fatigue-resistant, ISO standard configuration. To my knowledge, such a standard has not yet been published. The proposal is informative, however, because it shows that details of fastener geometry can have a significant effect on fatigue life. All of the refinements shown here are intended to reduce

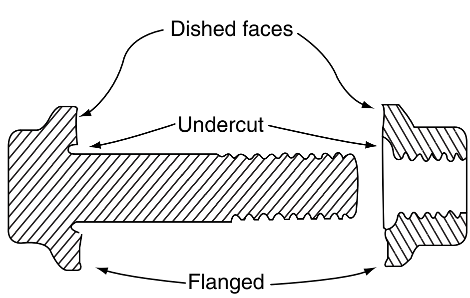


FIGURE 15.25 Flanged, dished, and undercut nut and bolt head improve stress distribution and therefore fatigue life. (Modified from Friesth, E.R., *Assembly Eng.*, 36, October, 1977.)

stress concentrations. In addition to the flanges, the design includes details we previously examined in Figures 3.9A and 15.24. Dishing the flanges slightly, incidentally, creates more uniform distribution of stress between flange and joint surface.

15.5.1.11 Surface Condition

Any surface treatment which reduces the number and size of incipient cracks can improve fatigue life substantially. Polished surfaces, for example, will make a big difference. Shot peening the surfaces also helps—not only because it smooths out beginning cracks, but also because it puts the surfaces in compressive stress (much as thread rolling does).

15.5.2 Reducing Load Excursions

Nothing can help extend the fatigue life of a bolt or joint more dramatically than a reduction in load excursions. We have discussed this at some length in an earlier section; I repeat it now simply because it is the single most important thing you can do. Your means of doing it include the following.

15.5.2.1 Prevent Prying

As we've seen, prying action greatly increases the load excursions seen by the bolts, and so should be avoided by proper design of the joint if at all possible. This, however, may mean economically unattractive, massive joint members.

15.5.2.2 Proper Selection of Preload

Correctly identify the maximum safe preload that your joint can stand, estimating fastener strength, joint strength, and external loads, analyzing them carefully with the help of a suitable joint diagram.

15.5.2.3 Control of Bolt-to-Joint Stiffness Ratios

Conventional wisdom says that we should try to minimize the bolt-to-joint stiffness ratio so that most of the excursion and external load will be seen by the joint and not by the bolt. Use long, thin bolts, for example, instead of short, stubby ones, even if it means using more bolts in a given joint. Eliminate gaskets wherever possible or use stiffer gaskets. (This may not, however, be helpful if you have leak problems!)

Against all this conventional wisdom, however, we have those who argue—as we learned earlier—that using stiffer bolts of a larger diameter will allow you to increase initial preloads and therefore reduce prying action. Again I think the stiff bolt versus soft bolt argument will be won only on a case-by-case basis.

15.5.2.4 Achieving the Correct Preload

Poor-quality tools and controls will increase the preload scatter and force you to work to a lower mean preload. Use the best you can afford, as discussed in Chapters 7 through 9.

15.6 PREDICTING FATIGUE LIFE OR ENDURANCE LIMIT

Techniques for theoretically predicting endurance limit or fatigue life of bolts are beyond the scope of this text. You will find some data, however, to Table 2.11. From these data you can see that the endurance limit of most bolts is significantly less than the endurance limit of the base materials. We've already learned that one expert [3] says that the endurance limit of bolts is only about one-tenth the endurance limit of the base materials. Others say that the cyclic loads imposed on a joint should be kept below 4% of the ultimate tensile strength of the fasteners if infinite life is desired [14]. A third source says that we can guesstimate the endurance limit of a bolt by experimentally determining the endurance limit of a polished, notch-free specimen of bolt material, then dividing that limit by a suitable stress concentration factor [15]. As an example, stress concentration factors for ½–13 × 6, SAE J429 Grade 2 fasteners in pure tension were found to range from 1.57 to 2.11 [16]. So here we have three experts saying that the endurance limit of a bolt is 1⁄10, 1⁄25, and ½ of the endurance limit of a test coupon. Take your pick! And accept this confirmation that fatigue test data are often scattered.

Here's a more carefully thought-out way to estimate the endurance limit of a bolt. An automative company estimates the endurance limit (S_n) by multiplying the endurance limit of a standard test specimen by a series of "correction factors" [17] using the following equation:

$$S_n^l = S_n(C_1 \times C_2 \times C_3) \tag{15.3}$$

where

S_n = the endurance limit of a standard test coupon (they say that this limit is one-half the ultimate tensile strength for wrought ferrous metals or 0.4 of the ultimate tensile for stainless steels)

C_1 = the loading factor (0.85 for axial loading, 0.58 for torsional loading)

C_2 = the size versus type of stress effect factor (0.85 for bending or torsional loads in fasteners 0.5–2 in. in diameter, 1.0 for axial loads of any diameter)

C_3 = the stress concentration factor (0.3 for rolled threads in quenched and tempered fasteners)

Other correction factors are added if the fastener is to be exposed to a corrosive environment or if the consequences of failure are great and they want to add a safety or reliability factor.

To guarantee that 98% of the fasteners will exceed the predicted life, for example, a reliability factor $C_4 = 0.8$ is included in Equation 15.3.

The multipliers need not all be less than 1.0, incidentally. If the fastener has been cold-worked, or surface-hardened and plated, correction factor C_5, greater than 1.0, is also included. The reference, however, doesn't suggest how much greater. The use of special thread, nut, and head geometry—as, for example, in Figures 15.23 through 15.25—might also allow use of a C_5 greater than 1.

I'm sure that Equation 15.3, and the proposed correction factors C_1 through C_5, is reasonable and appropriate for the fasteners used by the auto manufacturer who published this procedure for estimating endurance limits. The procedure would presumably work for other types of fastener in other industries as well; but it would be best to base your correction factors on fatigue tests or experiences of your own, rather than on data published by others.

15.7 FATIGUE OF SHEAR JOINT MEMBERS

As I mentioned at the beginning of this chapter, it's the bolts which fail in joints loaded in tension, but it's the joint members which fail under shear loads. Such failures—especially of symmetric butt splice joints—are described at length in the text by Kulak et al. [26] and so I'll only touch on a few highlights here. All of the following comments are derived from that text.

In a properly preloaded, slip-resistant shear joint, the fatigue failure will occur through the gross cross-section of the joint member (see Figure 3.18). If the joint is a bearing type, or is supposed to be slip resistant but was improperly preloaded and has slipped into bearing, then failure will occur through the net cross-section which intersects a line or group of holes. In general, bearing-type joints have less fatigue resistance than slip-resistant joints of comparable size. In fact, just increasing the slip resistance in a slip-resistant joint (presumably by increasing the coefficient of friction between faying surfaces or by increasing the initial clamping force) improved the fatigue behavior of the joints.

Kulak et al. summarize the results of fatigue tests conducted by many different workers. In most of these tests, the maximum applied stresses exceeded the yield strength of the net sections of the test specimens and often approached or sometimes exceeded the yield strength of the gross sections. These tests show that stress excursion is the dominant factor in determining crack growth. In fact, the fatigue strength of these structural steel members was relatively independent of the grade of steel tested, or of its strength. The materials tested had yield strengths ranging all the way from 36 to 120 ksi, but this variation in strength had negligible effect on fatigue life. (This is not to say that strength will never affect fatigue life of a metal part—a bolt, for example. It merely says that strength variation in the tested range did not affect the lives of structural steel members.) The joint members tested under reversing loads ($R < 0$) had better fatigue lives than those tested under cyclic tensile loads only ($R = 0$). This was true for both slip-resistant and bearing-type joints. The authors suggested that this result from the fact that crack growth is inhibited by compressive loads.

Kulak et al. point out that it is theoretically possible to predict fatigue life using the techniques of fracture mechanics, but to do so one must know the shape and size of the initial flaw—and the stress gradient. Since this is not practical for structural steel design, all of the fatigue data they report, such as the data shown in Figure 15.26, must be—and have

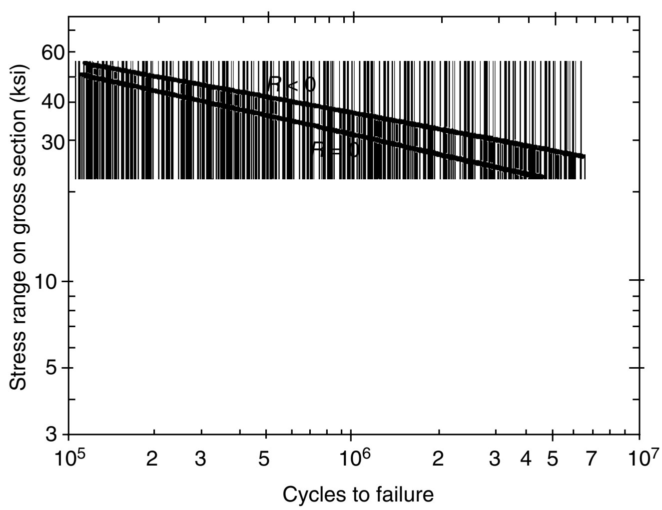


FIGURE 15.26 Mean *S–N* curves summarizing many tests of slip-resistant structural steel joints. The tests were conducted by many workers under a wide variety of conditions. Interestingly, they included steels having yield strengths ranging from 34 to 120 ksi, and show that yield strength has little influence the fatigue behavior of structural steel joint members. The upper line represents tests conducted with reversing loads fluctuating between tension and compression (i.e., $R < 0$). The lower line represents the mean *S–N* data for tests conducted under tensile loads varying from some maximum value to zero (i.e., $R = 0$). Note that these *S–N* curves, unlike those we studied earlier, are plotted on log-log scales, which converts the "curves" we saw earlier to straight lines. This is a very common way to present *S–N* data. I used the "curved" versions because I thought they were more informative for teaching purposes.

been—obtained by laboratory tests. I've been fortunate enough to see some of the machines used for these tests—at Lehigh University in Bethlehem, Pennsylvania, at the University of Toronto, and at the University of Texas at Austin—and they are very large and very impressive. The failure of a structural steel building is to be avoided at all costs, so a great deal of work has gone into the design, codification, and testing of this type of bolted joint. Those interested in this should attend meetings of—or at least follow the activities of—the Research Council on Bolted Joints, sometimes called the Bolting Council. They sponsored the preparation of the Kulak text, and they have sponsored many of the tests reported therein as well. Much of their knowledge is summarized in the AISC document describing the proper use in structural applications of ASTM A325 and A490 fasteners [27].

15.8 CASE HISTORIES

15.8.1 Transmission Towers

A midwestern power company installed 128, two-pole steel H-shaped towers to carry two 345-kV circuits in a horizontal configuration. The poles were fabricated in sections and the sections were connected by flanged, bolted joints. In some cases the flanges weren't pulled together completely, but were left open on one side.

The bolts used were 1¼ and 1½ in. in diameter, varied in length from 4½ to 10½ in., and most were made of ASTM A354 (Grade BD) or A325 steel.

Several years after the towers were erected, 15 bolt heads were found lying on the ground. An investigation showed that wind blowing on the towers had subjected the bolts to dynamic loads, which had led to fatigue failure. The fact that some joints were essentially loose on one side undoubtedly contributed to the problem, since preload was low and the joint very "springy," the critical load was low and the bolts had to absorb a larger share of any external load than intended. It was also felt that the bolts, being "high strength" (especially the A354), were too brittle to be acceptable in a fatigue situation.

The joints—there were 688 of them involving 15,000 bolts—were retightened, and many bolts were replaced. The problem was discovered before failure of any joint [12].

15.8.2 Gas Compressor Distance Piece

The studs originally used in this application were ASTM A193 B-7 with cut threads, used with standard A194, 2H heavy hex nuts, installed with a nominal torque of 385 lb-ft. Since molydisulfide lubricant was used, this torque produced bolt tension ranging from 25,000 to 45,000 lbs [1]. Studs were ⅛–7 × 7 in size.

The operator of this compressor reported daily failure of these studs when the equipment was first put into operation. An analysis of the failed bolts showed that they had failed in fatigue, and that bending stresses had contributed substantially to the problem. The following steps were taken:

1. Stud material was changed from B-7 (basically SAE 4140) to 4340, heat-treated to 37–43 R_C. This provided a minimum tensile strength of 160 ksi and a minimum yield of 145 ksi. The cleanliness of the 4340 was controlled.
2. Force washers were installed under one-quarter of the nuts to monitor installation torque and relaxation on a sample basis.
3. Standcote SC-1 PTFE coating (baked on) was used on all studs, nuts, and washers to reduce torque–preload scatter.
4. The threads were rolled after heat treating.

5. Nuts with washer faces were used. The faces were carefully machined perpendicular to thread axes, and seat correction spacers were added.
6. Preload was increased to the range 50,000–55,000 lbs.
7. The studs were inspected ultrasonically for fatigue cracks from time to time.

The results were dramatic. The first stud failure didn't occur until the studs had been in service for 6 months. The second and third failures occurred after the 7th and 8th months; and the fourth after 18 months. Nine additional bolts were found to be broken (out of 16) after 23 months. (All earlier failures had, of course, been replaced.)

No additional changes were made. All studs were subsequently replaced after each first failure, or after 18 months of service. This is considered an excellent service life for this very demanding application.

EXERCISES

1. What primary factors determine the number of load cycles required to break a bolt subject to fatigue loading?
2. Describe the appearance of the break surface of a bolt which has failed in fatigue.
3. Where on a bolt are you most apt to find fatigue cracks?
4. Why is a fatigue failure often worse than other types of bolt failure?
5. Define the endurance limit of a bolt.
6. Which is greater, the static yield strength or the endurance limit of the bolt?
7. When will an increase in preload increase fatigue life?
8. When will an increase in preload decrease fatigue life?
9. Which result can we usually expect?
10. Does a large bolt-to-joint stiffness ratio increase or decrease the possibility of fatigue failure of the bolt and why?
11. Name at least three bolt, thread, or bolt and thread configurations which can improve the fatigue resistance of a bolt.
12. Under what conditions can joint members suffer fatigue failure?

REFERENCES AND BIBLIOGRAPHY

1. Keith, G., Standco Industries, Inc., Houston, Texas, Personal communication, 1979.
2. Wayne, D.M., Fatigue design considerations in bolted joints, presented at Using Threaded Fasteners Seminar, University of Wisconsin-Extension, Madison, WI, April 1979.
3. Robert, F., SPS Laboratories, Jenkintown, PA, Personal communication, March 1980.
4. Thread forms and torque systems boost reliability of bolted joints, *Product Engineering*, December 1977.
5. Junker, G.H., *Principle of the Calculation of High-Duty Bolted Joints*, Unbrako-SPS, Jenkintown, PA.
6. Fazekas, G.A., On optimal bolt preload, *Journal of Engineering for the Industry—Transactions of the ASME*, pp. 779–782, August 1976.
7. Lingscheid, R.J., *Environmental Conditions for Advanced Fastener Systems*, Rocketdyne Div., North American Aviation, Inc., Society of Automotive Engineers, New York, September 1967.
8. High-strength bolted joints, *SPS Fastener Facts*, Standard Pressed Steel Co., Jenkintown, PA, Section IV-C-4.
9. Carl, C.O., *Fatigue Design*, Wiley, New York, 1970, p. 180.
10. Viglione, J., The effects of nut design on the fatigue life of internal wrenching bolts, Naval Air Engineering Center, Aeronautical Materials Lab., Report no. NAEC-AML-1910, AD 446 247, June 17, 1964.
11. Friesth, E.R., Modern metric hardware—simpler and better, *Assembly Engineering*, pp. 36ff, October 1977.

12. Ficken, L.A., Bolt failures on 345 kV pole type structures, Paper presented to the Transmission and Distribution Committee of the Edison Electric Institute, St. Petersburg, FL, January 18, 1980.
13. Bickford, J.H., Fatigue failure of threaded fasteners, *Journal of the Japanese Research Institute for Screw Threads and Fasteners*, vol. 14, no. 10, 1983.
14. Private conversion with Paul Bonenberger, General Motors Corp., Detroit, MI, 1981.
15. *Considerations for the Design of Bolted Joints*, Industrial Fastener Institute, Cleveland, OH, Undated.
16. Walker, R.A. and R.J. Finkelston, Effect of basic thread parameters on fatigue life, Society of Automotive Engineers, Paper no. 770851, October 1970.
17. *Fastener Engineering Handbook*, vol. 3, Ford Motor Co., Detroit, MI, June 1983.
18. Krams, W.E. and L. Raymond, Installation torque and fatigue performance of threaded fasteners, part HI of a series, *American Fastener Journal*, May/June 1991.
19. Krams, W.E. and L. Raymond, Installation torque and fatigue performance of threaded fasteners, last of a series, *American Fastener Journal*, September/October 1991.
20. Osgood, C.C., Saving weight in bolted joints, *Machine Design*, pp. 128ff, October 25, 1979.
21. Junker, G.H. and P.W. Wallace, The bolted joint: economy of design through improved analysis and assembly methods, *Proceedings of the Institute of Mechanical Engineering*, vol. 198B, no. 14, 1984.
22. Hood, A.C., Factors affecting the fatigue of fasteners, Attachment 7, Presented at a meeting of the Bolting Technology Council, in Cleveland, OH, April 19, 1993.
23. McNeill, W., A. Heston, and J. Shuetz, A study of factors entering into the calculations for connecting rod joints according to VDI 2230 guidelines, *VDI Berichte* (Germany), no. 766, 1989.
24. Eccles, W., MIED, Bolted joint design, *Engineering Designer* (UK), pp. 10ff, November 1984.
25. Nassar, S., Importance of initial tension on bolt fatigue load capacity, material passed out at a Bolted Joint Seminar presented by Lawrence Technological University, Southfield, Ml, May 1992.
26. Kulak, G.L., J.W. Fisher, and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, 2nd ed., Wiley, New York, 1987.
27. *Specification/or Structural Joints Using ASTMA325 or A490 Bolts*, prepared by the Research Council on Structural Connections, published by the American Institute of Steel Construction, Chicago, 1988.
28. Hess, D.P., Chapter 40, Vibration and Shock Induced Loosening, *Handbook of Bolts and Bolted Joints*, John H. Bickford and Sayed Nassar (Eds.), Marcel Dekker, NY 1998.

16 Corrosion

One of the most common problems we face when dealing with bolted joints is corrosion. It can take many forms, and can affect the stability of the clamping force and the useful life of the bolts or a joint in many ways.

For example, such mechanical failures as thread stripping and fatigue can be accelerated or made more likely by corrosion. Alternatively, initial buildup of rust can increase the tension in the bolt and the clamping force on the joint, because rust is a buildup, increasing dimensions.

Excessive corrosion, of course, can eventually lead to a reduction in preload as parts weaken, or to the total loss of clamping force through corrosion wastage, or, more unexpectedly and suddenly, through the mechanisms of hydrogen embrittlement or stress corrosion cracking (SCC).

Even if corrosion doesn't proceed far enough to affect the clamping force or life of the joint, it can cause problems. It can spoil the appearance of a product, or make assembly/disassembly difficult or impossible. So corrosion, which has been defined as "the deterioration of a material because of a reaction to its environment," [1] is a problem we can't ignore. Let's look at some of the factors that cause corrosion, and at some of the things we can do about it.

16.1 CORROSION MECHANISM

16.1.1 Galvanic Series

Every metal has a characteristic electrical potential, determined by its atomic structure and based on the ease with which the material can produce or absorb electrons. Those materials that will provide electrons more readily are called "anodic," those that absorb electrons more readily are called "cathodic." Anodes and cathodes are called electrodes. If properly interconnected they create "batteries."

No material is just an anode or just a cathode. Any material can serve either function, depending on the other materials to which it is connected. Steel, for example, is anodic in the presence of stainless steel or brass. It is cathodic in the presence of such materials as zinc or aluminum.

The relative anodic–cathodic potential of metals is defined by a table called the galvanic series. Materials listed toward the beginning of the table are anodic compared with those listed nearer the end of the table. The following list shows the relative anodic–cathodic relationship of many of the materials we will encounter in bolted joints [10].

Anodic end
(least noble—most likely to corrode)
Magnesium
Zinc

Aluminum 1100
Cadmium
Aluminum 2024-T4
Steel or iron
Cast iron
Chromium iron (active)
Nickel-resist cast iron
Types 304 and 316 stainless steel (active)
Tin
Nickel (active)
Inconel (active)
Hastelloy Alloy C (active)
Brasses
Copper
Bronzes
Monel nickel copper alloy
Nickel (passive)
Inconel (passive) types 304 and 316 stainless steels (passive)
Hastelloy Alloy C (passive)
Silver
Titanium
Graphite
Gold
Cathodic end
(most noble—least likely to corrode)

Note that some materials appear more than once in the table. Their electric potential depends on whether or not they are in an "active" or "passive" condition. Although passivity is not fully understood, it is believed to be caused by the presence of a very thin oxide layer on the surface of the material—a layer which, in effect, partially insulates the material and so reduces the ease with which it can give off electrons.

Although each metal will have a characteristic electric potential, this potential can vary, depending, as one example, on whether or not the metal is in an active or passive condition. There are other things that can alter the electric potential of a particular material. An increase in stress level, for example, can make a material more anodic. So can high temperature. Some materials are more anodic at grain boundaries, or in the vicinity of impurities [2].

Reducing the amount of oxygen in the electrolyte (the solution which forms the battery) will also make it easier for metals to give up electrons—it will make them more anodic.

Mechanisms such as these make it possible for a single body to act as both anode and cathode and so, alone, form its own battery. We'll take a detailed look in a moment.

16.1.2 Corrosion Cell

We learned in Chapter 13 that the four essential conditions for corrosion are an anode, a cathode, an electrolyte, and a metallic connection. A body will not corrode until it is immersed in, or wetted by, a solution of some sort, and provided with an electrical connection to another body having a different potential which is also immersed in, or wetted by, the same solution. The two bodies of different potential, electrically connected together, and in the presence of a liquid, form a miniature battery, as suggested in Figure 16.1. The anode in the battery will provide electrons, which flow to the cathode. In this process the anode is

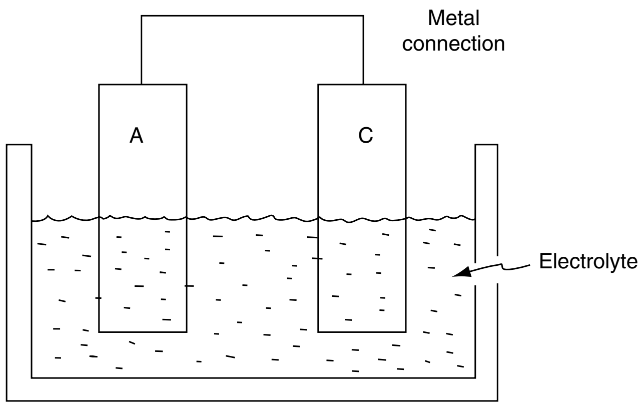


FIGURE 16.1 An electrical battery is formed whenever two metals having different electrical potentials are connected together by a piece of metal and by a liquid of some sort. Under these conditions the more anodic of the two materials will corrode.

gradually destroyed—in other words, it corrodes. The cathode, on the other hand, collects material which plates out on its surface. Remember:

Anodes away
Cathodes collect

Corrosion batteries can be formed in many different ways and by many different combinations of material. Some examples are given below. Corrosion chemists have distinguished among various classes of corrosion depending on the basic nature of the battery at work and the appearance of the results. Names given to various types of corrosion include: general corrosion or uniform attack, galvanic or two-metal corrosion, concentration cell corrosion, stress corrosion cracking, and pitting.

And there are others. In every case, however, the basic process is that described above—a relatively anodic material is connected to a relatively cathodic material in the presence of a solution, creating a miniature battery, destroying the anode as it produces electrons.

16.1.3 Types of Cells

To minimize corrosion problems we must either prevent the formation of batteries or reduce their size, and effectiveness. There are many ways to do this, as we'll see later. It will be helpful first, however, to look in detail at how some of these corrosion cells are formed in practice.

16.1.3.1 Two-Metal Corrosion

The most obvious way to encourage corrosion is to connect two different metals together, electrically, in the presence of a fluid. The farther apart the metals are on the galvanic series, the greater the potential difference between them, and the more apt the anode is to corrode. Using steel bolts on an aluminum tower that will be exposed to seawater, for example, is not a good idea, although it has been done (the tower collapsed!) [6].

In a less obvious fashion, dissimilar metals can be coupled through wet earth, puddles of rainwater, etc. It's surprising, in fact, how often we can inadvertently design batteries this way, or how often we must live with them because the design demands dissimilar materials.

Batteries involving two materials are relatively easy to spot and understand. There are many less obvious ways to create corrosion cells, however.

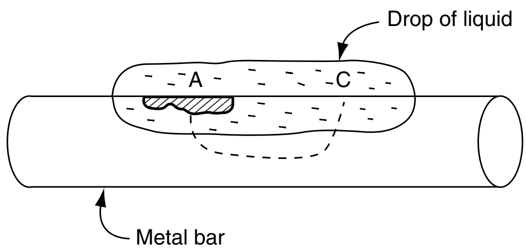


FIGURE 16.2 A single body can serve as both anode and cathode if a portion of it is protected by an oxide film (such as rust) while another portion is not. Variations in amount of oxide film can also make a difference. As a result, single bodies can and will rust when exposed to moisture.

16.1.3.2 Broken Oxide Film

Let's take a piece of steel that has rusted slightly. We will scrape the rust off one portion of the steel and wet the entire surface with water. That portion of the surface still protected by an oxide film (rust) will be cathodic with respect to that portion which is not so protected. There will, therefore, be an electrical potential difference between two adjacent portions of the surface (connected together inside the body), so the anode will corrode—forming rust on its surface, as suggested in Figure 16.2 [2].

Note that the amount of oxide film can also determine the relative anode–cathode relationship of various portions of the surface of a single body. As rust builds up on one portion of the body, it becomes less anodic and will eventually become cathodic with respect to a previously rusted area. In this way batteries form and re-form across the surface until the entire body has been destroyed.

16.1.3.3 Stress Corrosion Cracking

Stress makes a body more anodic. Stress concentrations at the root of a tiny crack, therefore, will make that portion of the body anodic with respect to adjacent portions, creating a tiny battery, which corrodes and enlarges the crack (Figure 16.3). This process often aids the

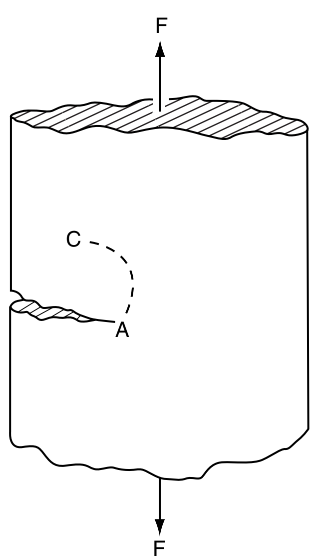


FIGURE 16.3 Stress concentrations make the tip of a crack more anodic than adjacent regions leading to stress corrosion cracking (SCC).

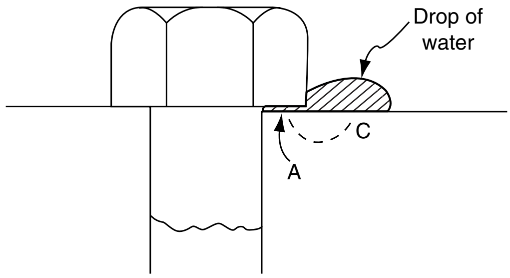


FIGURE 16.4 The oxygen content of water trapped in a crevice is less than that of water exposed to air. As a result, the crevice is anodic with respect to surrounding joint material.

growth of fatigue cracks—in fact, it's often difficult to tell whether or not a part has failed in fatigue or in corrosion cracking, or in a combination thereof. Hydrogen embrittlement may be a form of SCC, according to some experts [3].

Stress corrosion and hydrogen embrittlement are serious problems for bolting engineers. They're relatively common in bolts, and they lead to sudden and unexpected failure. As a result they deserve special attention. We'll take a closer look in Sections 16.2 and 16.3.

16.1.3.4 Crevice Corrosion

We saw earlier that reducing the oxygen content of the electrolyte will make it easier for adjacent metals to produce electrons. Water trapped under the head of the fastener, for example, as in Figure 16.4 will be oxygen starved. The oxygen trapped in a crevice cannot be replenished by the oxygen in nearby air. The oxygen-starved water becomes acidic. As a result, the crevice becomes more anodic than adjacent regions of the joint, and corrodes.

16.1.3.5 Fretting Corrosion

If two oxide-coated bodies are rubbed together, the oxide film can be mechanically removed from high spots between contacting surfaces, as shown in Figure 16.5. These exposed points will now be active (anodic) compared to nearby portions of the surface, which are still protected by an oxide film (more passive or cathodic). So the exposed regions will rust. Further, relative motion will knock off the next high spots, which will rust.

Over a period, this combination of electrochemical corrosion and mechanical motion will produce a very fine rust powder in the joint, called the products of fretting corrosion.

There are undoubtedly many other ways in which a corrosion cell can be formed, but these examples should suffice to convince you that there are many ways; they are not all

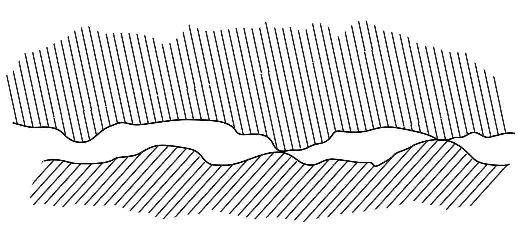


FIGURE 16.5 Mechanical action (fretting) removes corrosion products from surface high spots. Exposed areas will be anodic with respect to other areas still covered with rust or the like.

obvious, and it can be difficult to eliminate them all. We'll consider some ways of doing this in Section 16.3. First, however, let's take a closer look at one of the bolting engineer's main corrosion concerns—stress corrosion.

16.2 HYDROGEN EMBRITTLEMENT

16.2.1 Stress Cracking Failure Modes

The fastener industry—both suppliers and users—face two, major types of delayed failure by which an apparently good bolt or joint will suddenly and unexpectedly fail after hours or months or even years of satisfactory service. The two are fatigue failure, which we discussed in the last chapter, and stress cracking, which we'll consider now. Of the two, stress cracking is probably the more troublesome because it's more common. It is caused by a combination of tensile stress and corrosion, or tensile stress plus absorbed hydrogen. Unlike fatigue, stress cracking can cause failure even if the applied loads on the fastener are static, i.e., noncyclic.

Four different mechanisms have been identified that can cause stress cracking:

1. Hydrogen embrittlement
2. Stress embrittlement
3. Stress corrosion cracking
4. Hydrogen-assisted stress corrosion

We will discuss each of these but will concentrate on those which cause bolting engineers the most problems, namely, hydrogen embrittlement and SCC.

The time required for a bolt to fail under stress corrosion can be anything from a few hours after tension has been developed in the fastener to many years. As we'll see later, sophisticated ultrasonic techniques have been developed for detecting beginning stress corrosion cracks in in-service bolts, but these techniques are not readily available. Fasteners can be removed and early cracks may be found with magnetic particle or dye penetrant inspection, but stud or bolt removal and replacement is not always possible. As a result, many cracks go undetected until failure, which is sudden and unexpected.

Since failure is always complete—the bolt breaks—stress corrosion in its various forms can be a very serious problem.

Let's start by taking a close look at hydrogen embrittlement.

16.2.2 Hydrogen Embrittlement Mechanism of Failure

As far as the number of people affected are concerned, the most troublesome form of stress cracking is probably hydrogen embrittlement because it can and often does cause failure of common bolt materials being used in common applications. It's a frequent problem for the auto industry, for example, but also concerns the aerospace, structural steel, pressure vessel, and every other industry where bolt or joint failure has safety implications.

It can occur any time atomic hydrogen has been absorbed and retained by the fastener, and there are many ways that this can happen. For example, atomic hydrogen can be absorbed into the surface of the fastener during cleaning, descaling, pickling, or electroplating operations. If the fasteners are not baked properly as part of the plating process, the hydrogen will remain trapped by the plating.

Although electroplating is the most common source of entrapped hydrogen, the fastener can acquire it from other sources as well. Since hydrogen can be created at the cathode of a corrosion cell, embrittlement is sometimes caused when a sacrificial anode is used to protect a structure (a procedure described later in this chapter) [31]. Aluminum alloys coupled to steel will generate hydrogen at the steel electrode [33]. Lubricants can produce hydrogen if they break down during drilling, machining, or forming operations. Conversion coating operations such as phosphating or black oxiding can cause problems. Hydrogen present in the service environment can also be absorbed by the fastener. And these effects can be accumulative. Incidentally, highly stressed parts absorb more hydrogen than lower stressed ones [29].

The fact that properly plated fasteners can still acquire hydrogen and fail through embrittlement has caused a lot of liability problems for fastener suppliers, since most users automatically blame poor plating practices for any embrittlement failure. As the problems often don't become apparent until the user's product has been assembled and put in use, the liability claims can far exceed the cost of the supposedly faulty fasteners. Fortunately—at least if they're called upon—there are experts who can usually determine whether or not the failure was caused by improper manufacturing procedures or by service conditions [28,31].

The hydrogen absorbed by nonplated fasteners may or may not cause problems, because it can and often will diffuse out of the parts under certain conditions. Bolts given a phos-oil treatment, for example, shed the absorbed hydrogen if left on the shelf for 30 days or more at room temperature, or if baked for 34 h at 199°F (93°C). Plated fasteners, on the other hand, will not readily shed absorbed hydrogen if the plating is more than 0.06 ml (2.5 μm) in thickness [37]. In summary then—even unplated fasteners can and do fail, but electroplated ones are most likely to give us problems.

Failure occurs when the fastener is stressed above a threshold level—by preloading, for example. Curiously enough, the entrapped hydrogen will tend to migrate to points of stress concentration within the fastener. The pressure created by the hydrogen creates and extends a crack, which grows until the bolt breaks. That, at least, is one of several models for hydrogen embrittlement which have been proposed [10]. The main point for our purposes is that all of the proposed mechanisms start with the absorption of hydrogen by the base metal—usually during electroplating operations [3].

Note that although hydrogen embrittlement is usually included in a discussion of corrosion it is not really a corrosion failure. It usually occurs in corrosion-resistant (i.e., plated) bolts, however, and it's often difficult to distinguish this type of failure from others we're about to consider are corrosion related. So, I'll follow the conventional path and include it here.

16.2.3 Susceptible and Safe Materials

Traditionally, hydrogen embrittlement has been most commonly encountered in cadmium-plated, high-hardness steels. In general, common experience says that if the hardness of the fastener is less than 35 HRC, you'll probably have no problems; if it's above 40 HRC, problems are almost certain; in between you may or may not have a problem [10]. Socket head cap screws made of medium carbon alloy steel and hardened to over 45 HRC are so apt to fail if plated that plating is considered "risky business." Plated SAE J429 Grade 8 fasteners, hardened to 39 HRC, are susceptible, but Grades 5 (34 HRC) or less are said to be immune [37]. There is evidence, however, that lower-hardness, plated steels are not immune, but simply take longer to fail—years instead of hours—because crack growth is slowed by a decrease in strength and the toughness of the steel increases, so that the crack must grow over a larger distance before the bolt will fail [28].

Although we'll usually encounter hydrogen embrittlement in plated steels, it can and does occur in other fastener materials such as austenitic stainless steels [3], aluminum, and titanium [29]. In fact, embrittlement was a major problem when titanium fasteners were first introduced, but improved manufacturing procedures have made this type of failure rare. The alloy Ti-6A1-4V is said to be especially insensitive to embrittlement [35].

The company Weserchemie GmbH Brueder Mlody in Germany has recently introduced fasteners hardened to 40 HRC and above coated with a 0.05-mil (2 μm) layer of copper before being plated with zinc or nickel. The copper provides a barrier, which inhibits the absorption of hydrogen. The coating process, a combination of mechanical plating and chemical action, also "polishes" the surface of the fastener, and thereby reduces the reaction area for hydrogen [30].

Certain exotic, high-strength materials are also said to be relatively immune to hydrogen embrittlement. These include Inconel 718 and MP35N, for example. Unfortunately, their relatively high cost makes them unacceptable for many of the applications troubled by embrittlement.

16.2.4 Testing for Embrittlement

Several documents include test procedures designed to detect hydrogen embrittlement. Specification ASTM F606-90, section 7, for example, describes a procedure for the hydrogen embrittlement testing of commercial grade, through hardened fasteners having diameters ranging from ¼ to 1½ in. A sample group of fasteners is taken from a production run. The fasteners are inserted into a test fixture. A wedge with a 4°–6° taper is placed under the head of each fastener, which is then tightened to 75% of its minimum ultimate tensile strength. The fasteners are left under this stress for 48 h, after which a breakaway torque (in the tightening direction) is applied. If that torque is less than 90% of the initial torque a fastener is assumed to have relaxed and its test is aborted. Those which pass the breakaway torque test are examined under 20× magnification. If any cracks are found the lot of fasteners is rejected [27].

Although the torque test and search for cracks are specified as the F606-90 inspection criteria, if hydrogen embrittlement is present the heads of the fasteners will probably break off when the breakaway torque is applied, or will have popped off during the 48 h under the high stresses created by the tapered washer. And failure can be violent: the flying head can cause serious injury, such as loss of an eye or worse [27,28]. For example, if you were testing a ¾ in. diameter, 2 in. long, Grade 8 bolt loaded to 75% of its UTS or 113 ksi—and it broke—the potential energy stored in the bolt would be sufficient to project the fastener up onto the roof of a building six stories high. So—be careful when you conduct these tests!

Another test for hydrogen embrittlement is described in MIL-STD-1315-5A. The fasteners tested here are preloaded, then left under stress for 200 h. Magnetic particle inspection is then used to inspect them for cracks. If any are found, the sampled lot is rejected [38].

ASTM subcommittee E08.06 has proposed—and may by now has published—a test procedure in which fasteners hardened to 39 HRC (175 ksi UTS) or higher are left under load for 7 months before being inspected. Those hardened to less than 39 HRC are to be loaded for 14 months before inspection, confirming the earlier statement that softer fasteners can fail but will take longer to do so [31].

And there are still other recommendations. The Industrial Fastener Institute (IFI) recommends a 24-h test for fasteners in the 32–38 HRC range. The U.S. Navy requires tests of up to 4 years duration in some cases [31]. In each case visual or other crack detection methods are used at the end of the test if the fasteners haven't already failed. And failure can occur quite rapidly. One authority says that if a fastener breaks 1–48 h after initial preloading the problem is almost certain to be hydrogen embrittlement. If it breaks during assembly it's almost certain to be something else [31].

A current incentive to conduct a test of some sort is provided by the new Public Law 101-592, which requires that all fasteners having a minimum tensile strength of 150 ksi or more shall be recertified after they are electroplated. Failure to do so can lead to a 5 year jail term or a fine of $25,000 or both [27].

It would obviously be desirable to find an effective way of testing for embrittlement in a shorter time than those cited above. A rising step load test has recently been developed by Dr. Louis Raymond. Computer-controlled equipment is used to monitor the onset of crack growth with great precision. Dr. Raymond says that with this procedure dependable hydrogen embrittlement tests can be run in 24 h [28,31].

16.2.5 Fighting Hydrogen Embrittlement

There are several ways to combat hydrogen embrittlement. The most common is to try to prevent hydrogen from being absorbed by the fastener. In most situations this involves the use of correct plating procedures: baking the fasteners properly, keeping the baths clean, etc. The preparation and cleaning steps are special problems, with overloaded or underloaded barrels playing a major role [29]. As far as baking is concerned, a bake time of at least 3 h at a temperature of 350°F–400°F (178°C–204°C) within 4 h of plating used to be recommended for steels of hardness 32 HRC (150 ksi UTS) or higher. If the hardness was 40 HRC or more, a minimum of 23 h was recommended; above 50 HRC the recommendation was ‘‘don’t electroplate’’ [38]. It’s my understanding that the military, at least, now specify a minimum of 23.5 h for hardness of 32 HRC or more.

The requirement that baking start within 4 h of plating is based on the fact that if baking is delayed the absorbed hydrogen may already have concentrated and have started cracks. No amount of baking will eliminate these cracks. The baking, incidentally, doesn’t drive out the hydrogen; it merely forces it into ‘‘traps’’ where it loses its mobility, and therefore prevents it from concentrating [29].

A second, popular way to avoid entrapped hydrogen is to use fastener coatings, which do not involve electroplating. We’ll look at some of these in a minute, and will see that they include such things as mechanically plated cadmium or zinc as well as bonded coatings of molybdenum disulfide or Teflon. Ceramic and aluminum coatings are also finding some applications. Although these all reduce the chances for embrittlement they don’t always eliminate it, as we’ve seen—and many of the new coatings can be expensive.

A third way to avoid hydrogen embrittlement would be to minimize the stresses placed on the bolts. This would work—but it’s not a popular solution because high clamping force is usually desirable to prevent other types of bolt or joint failure. We’ll see that stress control is the most popular way to reduce stress corrosion problems—but only because it’s virtually our only option. With hydrogen embrittlement we have other, more attractive options such as those just cited.

A fourth—and also common—way to minimize embrittlement problems is to use a less susceptible bolt material. For the budget conscious this usually means using Grade 5 instead of Grade 8, or Grade 5 equivalents such as ASTM A325 or Metric 8.8. For those with critical applications, safety concerns, and money to spend, this means such high-class materials as Inconel 718 or MP35N.

16.3 STRESS CORROSION CRACKING

16.3.1 Mechanism of Failure

We learned earlier that stress tends to make a metal more anodic. The tip of a crack in a bolt under tension will, therefore, be more anodic than the materials surrounding it. If an electrolyte is added, we have produced a small battery, which will provide the energy needed to eat away the anode at the tip of the crack. The crack, therefore, will grow again until the bolt breaks.

Having said all that, I should confess that it is only one of several theories that attempt to explain the mechanism of SCC [3]. Although not universally accepted, it certainly illustrates the apparent process.

There is, I think, no dispute about the essential conditions required for stress corrosion, whatever the mechanism may be. These conditions are:

A susceptible material
Tensile stress above a threshold limit
An electrolyte

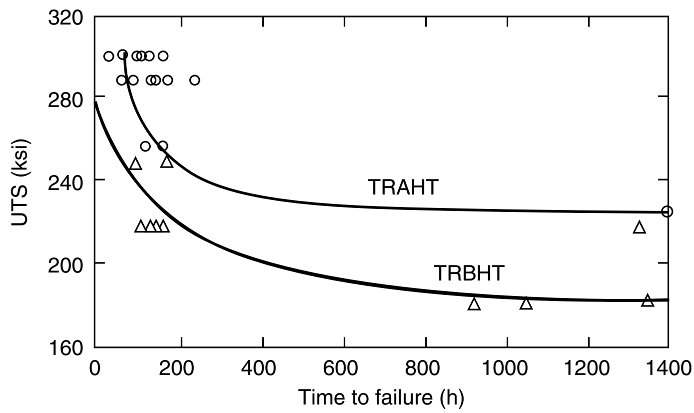


FIGURE 16.6 Rolling fastener threads after heat treatment (TRAHT) gives the fastener greater resistance to SCC than rolling them before heat treatment (TRBHT). The data shown here are based on tests of ¼–28 × 2 AIS1 H-11 bolts heat treated to a strength of 260 ksi UTS. (From Hood, A.C., *Met. Prog.*, September, 1967.)

As far as susceptible material is concerned, all metallic bolting materials are susceptible to some extent, though some of them much less than others. As far as the electrolyte is concerned, we do not need to immerse the part or constantly drip on it, as is usually required for general corrosion wastage. Only a tiny amount of electrolyte is required for SCC. 1 once heard a "dirty fingerprint" had caused a failure. The story may have been apocryphal, but the message was valid. Such normally benign electrolytes as humid air can cause SCC. We'll look at some other possibilities in Section 16.5.

The importance of stress above a threshold level is more involved, and is illustrated in Figure 16.6. It shows the time to failure of a number of AISI H-11 bolts subjected to a variety of tensile loads [40]. The higher the load, the shorter the life of the bolts, at least if the loads exceeded 3500 lbs. Bolt life was essentially infinite, however, if the applied load was 3500 lbs or less. This load defined the threshold stress level for that material in that environment.

The threshold stress level for a given material can sometimes be predicted, using the techniques of fracture mechanics and a material property called K_{ISCC}. Let's take a detailed look at this important concept.

16.3.2 The Concept of K_{ISCC}

Experts in failure analysis and fracture mechanics have given us a linear elastic fracture mechanics (LEFM) equation with which we can estimate the amount of stress that can safely be applied to a part which might otherwise fail through SCC. The equation is [17]:

$$K_{ISCC} = C\sigma\sqrt{\pi a} \tag{16.1}$$

where

K_{ISCC} = the threshold stress intensity factor for SCC (ksi [in.$^{1/2}$])
C = the shape factor (1.5 has been used for threads) [17]
σ = nominal stress (ksi)
a = crack depth (in.)

Note that Equation 16.1 does not explain the mechanism of SCC. It just provides a means for characterizing the tendency of the body—in this case a bolt—to crack, given an existing flaw and various levels of applied stress. The equation tells us that if the product $C\sigma(\pi a)^{1/2}$ exceeds a certain critical value K_{ISCC}, then a crack (the initial flaw, for example) will grow and the part will break. The initial flaw can be a tool mark, a corrosion pit, a crack caused by heat treatment, etc.

Even though the equation doesn't give us new knowledge about the SCC process, it has physical meaning—the all-important threshold stress intensity factor, K_{ISCC}, is as much of a material property as yield strength or coefficient of expansion. It can (and must) be determined experimentally for a given material, in a given condition (heat treat, etc.) in a given environment. If we know K_{ISCC} for out bolting application, then we can relate an anticipated flaw size (e.g., crack depth) to an allowable (safe) stress or preload in the fastener.

Many factors affect K_{ISCC}. We'll look at some of these next. It's important to realize, however, that the number of variables involved is so large that experimental results tend to be scattered.

Different investigators get different results because one or more variables—often unidentified—vary between one set of experiments and another. Even local variations within a bolt can affect SCC properties [19]. As with vibration and fatigue problems, therefore, we're dealing with conflicting data, and if avoiding failure is essential, we must set limits on stress or preload in accordance with the worst-case results, which have been reported.

If the calculations are based on the worst-case (i.e., lowest) value of K_{ISCC}, the results are often very conservative. If an occasional failure is acceptable and the consequences of failure are of no great concern, then applied stresses can be higher than suggested by Equation 16.1 [17].

Unfortunately, there is no common source of information for K_{ISCC} for bolting materials. The literature is scattered, and sometimes contradictory [10]. In critical applications, you should make your own tests, using bolts and conditions which closely reflect your own application.

16.3.3 Factors Affecting K_{ISCC}

16.3.3.1 Bolt Material

As already mentioned, K_{ISCC} is a material property, like the modulus of elasticity or the coefficient of thermal expansion. It's sometimes called a measure of the strength of a material in a corrosive environment [34]. Like most (all?) other material properties, however, it's not a constant, but is a function of a number of variables.

16.3.3.2 The Environment

As mentioned earlier, K_{ISCC} is not a single-valued material property, and therefore there is no single threshold stress level. These things depend very much on the environment [37]. And, unfortunately, K_{ISCC} data for many corrosive environments are just plain not available [32].

16.3.3.3 Thread-Forming Procedure

The way the threads are formed also affects the resistance of the bolt to SCC. Threads rolled after heat treat have greater resistance than those rolled before heat treat, as suggested by Figure 18.6 [40].

16.3.3.4 Bolt Strength or Hardness

The hardness of the bolt—which relates to its strength—affects the stress corrosion behavior of the bolt and, therefore, its K_{ISCC} value. Figure 16.7, for example, shows K_{ISCC} versus yield strength results (small circles) for a variety of tests made on a variety of low-alloy quenched

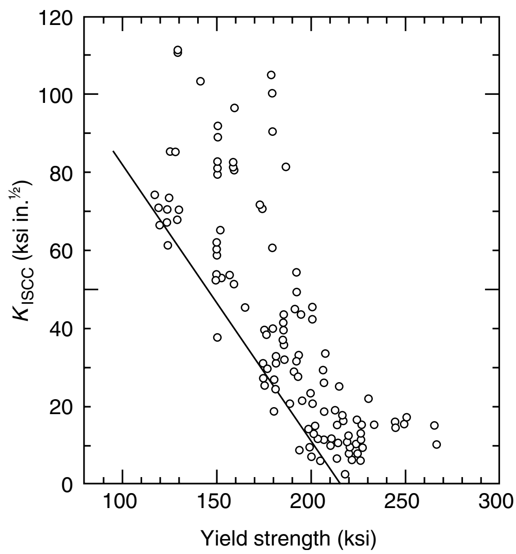


FIGURE 16.7 K_{ISCC} versus yield strength for low-alloy quenched and tempered (LAQT) steel in humid air and chloride-containing aqueous environments. Circles show results of test made by a variety of investigators under a variety of conditions. The solid line defines the so-called lower-bound relationship between K_{ISCC} and yield strength. Assuming a higher K_{ISCC} for a given yield strength would introduce some probability of SCC failure.

and tempered (LAQT) steels. Examples of such materials would be ASTM A193 B7 or B16, SAE J 429 Grade 8, AISI 4340, ASTM A490 or A307 or A540, etc. The environments involved included humid air, seawater, aqueous solutions of sodium chloride, and distilled water [17].

The solid line on the graph shows the worst-case relationship between K_{ISCC} and hardness. If our bolts had a yield strength of 150 ksi, for example, then, using Figure 16.7, we could safely assume a K_{ISCC} value of 50 ksi (in.^{2}) or less. From this we could determine an acceptable preload. Let's take an example.

Let's assume that our bolts are A540, Grade B21s, 7⁄16 in. in diameter, with a yield strength of 140 ksi and a K_{ISCC} of 55 ksi (in.)$^{1/2}$. For the shape factor (C) we'll use 1.5 [17].

We next have to make some assumptions about the depth of the cracks or flaws we can expect to find in the bolts before we introduce them to the SCC environment. This is a tough problem, of course, with no absolute answers, but we'll assume that the maximum flaw size (a) is 20 mils. Now we are prepared to answer the question "How much preload can I develop in this bolt without risking an SCC failure?" We start by solving Equation 16.1 for nominal stress.

$$\sigma = K_{ISCC}/C\sqrt{\pi a}$$

$$\sigma = 55/1.5\sqrt{\pi(0.02)}$$

$$\sigma = 146 \text{ ksi}$$

If our bolts have a 7⁄16–16 UN thread, then the tensile stress area will be 0.114 in.^{2}. We can now compute the safe tension which can be developed in this bolt when we preload it.

$$F_p = \sigma As$$

$$F_p + 146(0.114) = 16.6 \text{ kips}$$

This calculation is correct, however, only if—and this is a big if—the environment with which we are involved is similar to those used during the tests which created the data on which Figure 16.7 is based. Those tests were made in humid air or salt water environments or equivalent. The K_{ISCC} value and safe preload would be reduced substantially if our environment included more aggressive electrolytes, such as a strong acid or hydrogen sulfide, as we'll see later.

If we wish, we could now go on and use the short-form torque–tension equation $\tau = KDF_p$ to compute the assembly torque.

Note that Figure 16.7 teaches us that the higher the yield strength of our bolts, the less the preload we can introduce, safely, at assembly. This suggests that "stronger" bolts can, in fact, be less dependable than "weaker" bolts in an SCC situation, at least as far as LAQT steels are concerned.

Fortunately, not all "strong" bolts are prone to SCC. Socket head screws, for example, with a hardness of 39–44 HRC, are resistant to SCC if they are made with a low-alloy steel having sufficient alloying content to gain a high as-quenched hardness and, therefore, a lower yield-to-ultimate-strength ratio (i.e., good ductility). A generous chromium content in the steel helps too [10]. We'll look at some other exceptions in a minute.

16.3.3.5 Type of Electrolyte

The aggressiveness of the electrolyte has a significant impact on the probability of an SCC failure. Petrochemical plants manufacturing acids, for example, have had SCC failures in A193 B7 bolts tempered to values as low as 22 HRC (80 ksi yield). In a humid air environment, the same bolt material can be tempered safely to 38 HRC. (See also Figure 16.13).

As mentioned earlier, even a small amount of electrolyte can cause problems. In recent years, for example, the nuclear power industry has been concerned about a number of SCC failures in A193 B7 and other LAQT steels, where the electrolyte was formed of a combination of humid air and molydisulfide thread lubricant. The molydisulfide decomposes (hydrolyzes) at modestly elevated temperatures to form corrosive hydrogen sulfide [21].

Moly isn't the only lubricant that can cause such problems. In one study, sulfur-based, copper-based, and lead-based lubricants also contributed to the cracking of such materials as 17-4PH, cold-worked 304, and even annealed 304 stainless steels, as well as Inconel and Inconel-X. Only graphite-based lubricants led to crack-free behavior [22].

It's interesting to note, I think, that some environments which cause no corrosion in unstressed parts can cause SCC when tensile stress is present. Conversely, other electrolytes that lead to rapid general corrosion may not cause SCC [10]. This, apparently, is one of the mysteries, which tends to refute the "mechanism" discussion I gave in Section 16.1.

16.3.3.6 Temperature

Cracking susceptibility is also a function of temperature. For example, the resistance of a 132 ksi yield steel in aqueous hydrogen sulfide solution was halved when the temperature was raised from room temperature to 300°F (150°C) [19]. The reverse can be true, however, if the bolts are entirely immersed in the electrolyte. Here, the higher temperatures may drive off some of the oxygen present in the electrolyte, slowing the rate at which SCC cracks will develop.

16.3.3.7 Bolt Diameter and Thread Pitch

Apparently there's a relationship between the depth of the threads on a bolt and its sensitivity to SCC. The presence of a thread makes it difficult to choose an equivalent crack depth, a. Furthermore, the shape factor, C, is affected by such things as thread engagement [17]. So, although the thread itself is not a "crack," it affects SCC sensitivity.

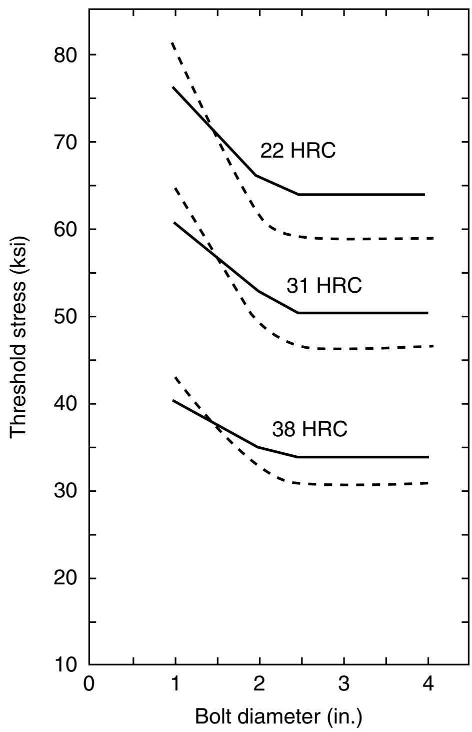


FIGURE 16.8 Threshold stress as a function of bolt diameter for LAQT steels with UNC threads. Initial crack depth is assumed to equal the thread depth plus 0.1 in. in each case (solid line) or twice thread depth (dotted line).

As a result, larger-diameter bolts (deeper threads) of a given material (given K_{ISCC}) have lower threshold stress (σ) levels than smaller bolts, at least up to a point (see Figure 16.8). For the same reason (thread depth) bolts with fine-pitch threads are less sensitive than those with coarse threads [17,20].

16.3.4 Combating SCC

To fight SCC, we must try to find a way to eliminate or minimize one of the three essential conditions: (1) a susceptible material; (2) a stress level above a threshold limit; or (3) the presence of an electrolyte. Here are some of the steps commonly taken to accomplish these things.

16.3.4.1 Susceptibility of the Material

Although every metallic bolting material is susceptible to SCC under certain conditions, most of them can be made resistant to it if properly heat-treated, except in the most aggressive of environments. For example, although carbon steel and LAQT fasteners can have SCC problems at all strength levels, they are usually safe to use unless hardened to an ultimate strength in excess of 160 ksi (40 HRC or higher). They should also be used with some caution in a hardness range 35–39 HRC. Below 35 HRC, they are generally considered immune to SCC; all of this, again, is for normal environments (humid air, aqueous chloride, etc.). As an example, steels with yield strengths below 100 ksi are highly resistant to SCC [19]. ASTM A325 bolts are considered safe from SCC because their hardness (strength) is not high enough to cause a problem—at least in the sort of environments they are most likely to encounter in structural steel and similar applications.

If the environment involves hydrogen sulfide, then such materials as A193, B7M, and A320 L7M should be considered. They have higher threshold stress levels than the more common B7 or L7 grades; in fact, they are intended specifically to resist SCC and have carefully limited hardnesses.

Austenitic stainless steels (such as A193 B8 and AISI 316) give better SCC service than martensitic (such as AISI 410, 17-4PH, or ASTM A449) stainless steels because the martensitic materials have a propensity for pit formation and crevice corrosion, which apparently encourage the entry of hydrogen into the material [19].

As mentioned earlier, socket head screw materials give exceptionally good service. Many aerospace materials are essentially immune to stress corrosion under normal conditions. MP35N, for example, has excellent resistance to SCC; so do some titanium alloys, although these may be susceptible to SCC at elevated temperatures unless properly processed [24].

In general, however, SCC failure of aerospace bolts is limited to alloy steels. These can still be used successfully, but only after being coated with combinations of cadmium and nickel or other inorganic materials [26]. More about coating in Section 16.4.

Aluminum 7075-T73 (a proprietary Alcoa heat treatment) is fairly impervious to SCC and is stronger than 2024-T4 aluminum, but there is a significant cost differential as well [24].

NASA's George C. Marshall Space Flight Center has published extensive lists of resistant and susceptible materials to be used or are proposed for use in space vehicles, other flight hardware, ground support equipment, and test facilities. This means exposure to seacoast or mild industrial environments according to their report [37]. The materials they evaluated include those used in structural or joint applications as well as for bolting. Their list of alloys having high resistance to SCC includes: carbon (1000 series) and low-alloy (4130, 4340, etc.) steels having ultimate tensile strengths below 180 ksi and Custom 455 stainless steel in condition HI000 and above. The list also included the following materials in all conditions (i.e., any hardness): A286 stainless steel, Inconel 718, Inconel X-750, Rene 41, Unitemp 212, Waspaloy, MP35N, and several titanium alloys including Ti-6A1-4V.

NASA's list also includes many materials used for joints, structures, or other purposes as well as for bolting. This includes a number of stainless steels, wrought and cast aluminum alloys, copper alloys, beryllium, and magnesium.

Another long list of resistant materials can be found in Hood (1967) [40]. In addition to the materials just cited, they list many titanium alloys including Ti-7Al-12Zr, Ti-8AI-lMo-lV, and Ti-5AI-5Sn-5Zr. They also list resistant joint materials such as 7075-T6 aluminum, type 321 stainless steel, titanium TU6A1-4V, and aluminum alloys 2219-287 and 2014-T6.

Materials to avoid include carbon and alloy steels with hardnesses over 40 HRC and high-strength maraging steels such as Vascomax 250 or Marage 300. Materials such as these are so sensitive to SCC that a tiny flaw can be fatal. NASA's list of susceptible materials included carbon, H-11, and alloy steels having UTS above 200 ksi and various maraging steels aged at 900°F.

16.3.4.2 Eliminating the Electrolyte

One common way to eliminate the electrolyte is to coat the bolts to prevent electrolyte from contacting them. Materials such as aluminum, ceramics, and graphite, for example, can be very effective against SCC. You'll find more details in the discussion of fastener coatings. Other than coating them, it is difficult to isolate the bolts, fully, from environments which can produce electrolytes sufficiently aggressive to cause SCC. As mentioned several times, humid air can do it. A dirty fingerprint may cause a problem. Bolts completely embedded in concrete, and therefore apparently isolated from corrosive liquids, have failed by SCC because the concrete leached chlorines, which formed the electrolyte [20]. Normal thread lubricants, as already mentioned, can also lead to SCC problems. Joint sealants (chemical gaskets) have also been identified as the source of leachable sulfur, fluorine, and chlorine materials, which led to cracks [25]—although there is some debate about these findings. But properly applied coatings will, in effect, eliminate the electrolyte.

Although complete elimination of possible electrolytes is very difficult, without coating protection, reducing the exposure to electrolytes can extend SCC life. Tightening gasketed joints so that they don't leak electrolytes onto the bolts is an obvious step for a bolting engineer to take. This, of course, can also reduce other types of corrosive attack on the bolts as well as SCC.

16.3.4.3 Keeping Stress Levels below a Threshold Limit

Although many of the charts in this chapter imply that only preloads affect a bolt's resistance to SCC, in fact, any source of stress can contribute to the problem. Preload is usually a major factor, but we mustn't forget residual manufacturing stresses, bending stress, the stresses created by hole interference or press fits, etc. In spite of this complexity, probably the most common way to combat SCC is to keep stress in the fasteners below a threshold limit defined (or at least computed) by the K_{ISCC} value. As already discussed, the acceptable stress limit will be a function of K_{ISCC} the crack shape factor, and the size of the initial flaws which must be tolerated. With K_{ISCC} data in hand, we can compute acceptable maximum stress levels from a rearrangement of Equation 16.1.

Figure 16.9 gives the resulting data, as a function of bolt hardness, for LAQT steels used in aqueous or mildly chlorine environments, assuming initial crack depths of one and two

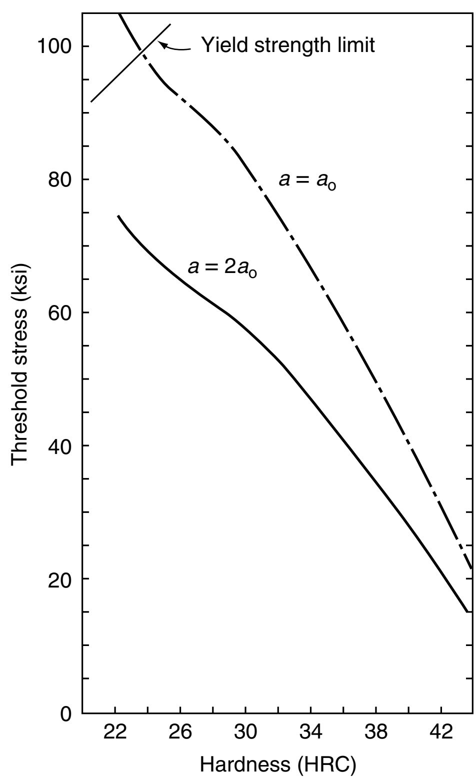


FIGURE 16.9 Plot of threshold stress limits versus hardness for 1½ in. diameter bolts of LAQT steel, used in humid air or chloride-containing aqueous environments. The K_{ISCC} values used to compute threshold stresses were taken from the solid line of Figure 16.7. Crack depths equal to the thread depth (dashed line above) and twice that (solid line) were assumed.

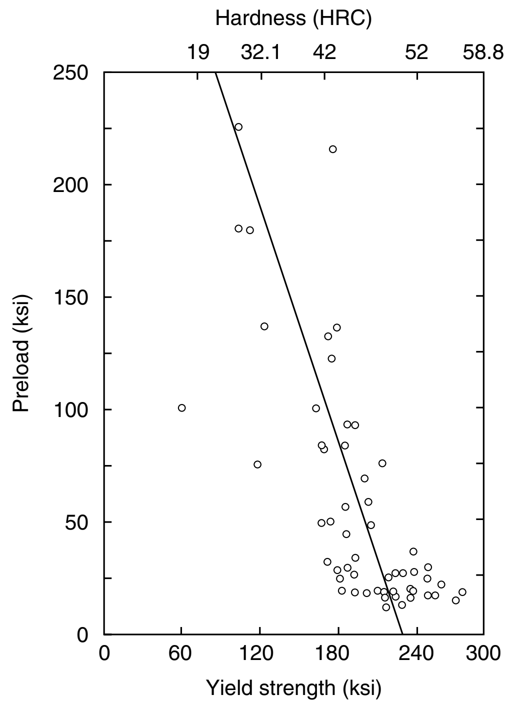


FIGURE 16.10 Another plot of acceptable preload (which equates to threshold stress) for an LAQT steel with a UNC thread. The differences between the recommendations of this plot and that of Figure 16.9 are discussed in the text.

times the thread depth [17]. Note that this author reported that these plotted stress limits, suggested by use of the worst-case (lower-bound) stress intensity factor K_{ISCC} were very conservative. This seems to be confirmed by the theoretical calculations of acceptable preload stress versus hardness, which resulted in the plot shown in Figure 16.10, one of a series of 36 such plots (for a variety of bolting materials, threads, etc.) reported in Czajkowski (1984) [20]. He suggests that a preload stress as high as 68 ksi would be acceptable for a 4 in. diameter LAQT 4340 bolt hardened to 42 HRC. Figure 16.9 suggests a maximum preload stress of 32.5 ksi for a 11 in. diameter bolt of the same material hardened to 42 HRC, in spite of the fact that larger-diameter bolts are supposed to be more sensitive to SCC than smaller ones. Both references were using the same K_{ISCC} data; but Czajkowski [20] assumed different crack depths and crack shape factors than Chung [17].

There's another difference here as well. The solid line in Figure 16.10 is a linear regression line representing the average results of the stress versus hardness tests. The line in Figure 16.9 is based on the lower-bound, worst-case/MSCC values shown in Figure 16.7. Although the people who developed Figure 16.10 don't say so, they imply that maximum preload (i.e., threshold stress) decisions can be based on the average response of the material, rather than on the worst case. So, how we use the data can be another variable.

Note that this plot shows threshold stress (maximum safe preloads) as a function of both hardness, as in Figure 16.9, and yield strength. We'll use yield strength in the remaining SCC plots. This is a common practice. We should keep in mind, however, that yield strength alone doesn't determine SCC behavior. Heat treatment, microstructure, and the composition of the material also play a role [19].

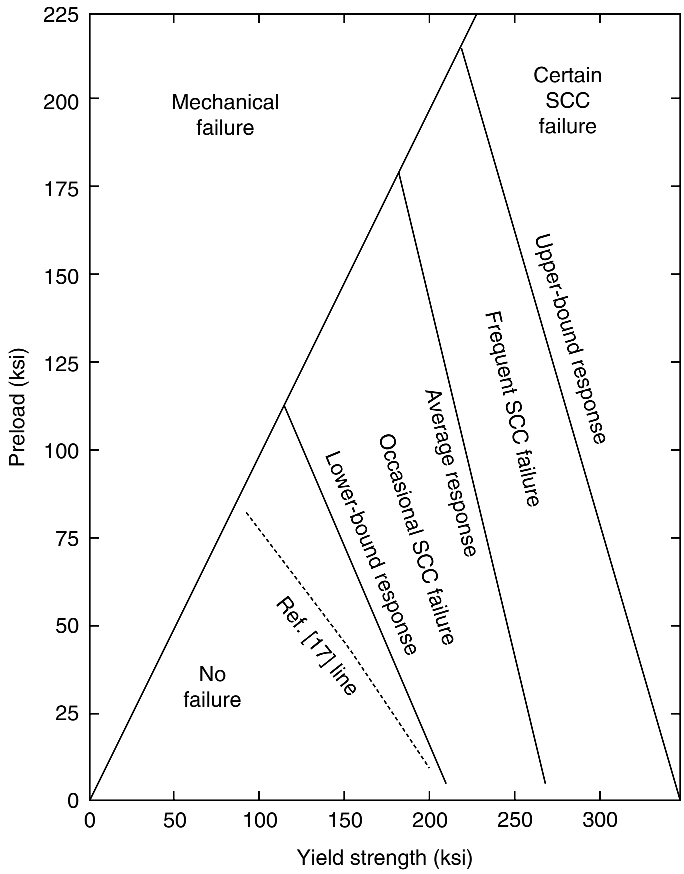


FIGURE 16.11 A more complete presentation of the data plotted in Figure 16.10 [20] plus some comparison data from Ref. [17]. See text for details. (From Chung, Y., Threshold preload levels for avoiding stress corrosion cracking in high strength bolts, Technical Report no. 0284-03 EV, Bechtel Group, San Francisco, April 1984. Czajkowski, C.J., Bolting applications, NUREG/CR-3604 BNL-NUREG-51735, U.S. Nuclear Regulatory Commission, Washington, DC, May 1984.)

Figure 16.11 repeats the regression line (called here the average response line) of Figure 16.10. This time, however, I've also included lower-bound and upper-bound response lines that fully encompass the individual test points "spotted" throughout Figure 16.10. In Figure 16.11 I've also replotted the lower-bound threshold stress versus yield strength line for a 4 in. diameter bolt, from Chung [17]. As you can see, there's reasonable agreement between the lower-bound lines of the two references. Our choice of lower-bound or average data will presumably be based on the consequences of bolt failure in our own application.

Another aspect of Figure 16.11 is worth noting. In addition to the lines derived from K_{ISCC} data and calculations, I've plotted a yield line representing all the points at which the recommended preload (threshold stress) equals the yield strength of the fasteners. It's important to realize that Equation 16.1 will recommend preloads in excess of yield, since the K_{ISCC} values on which it depends are blind to the yield strength of the material. Your calculations, therefore, must always be checked against yield.

Note, too, that the type of bolting material and the character of the electrolyte also play roles in the selection of the maximum or threshold preload for a given situation. Figure 16.12 shows the threshold stress versus yield strength for six different bolting materials. (The response

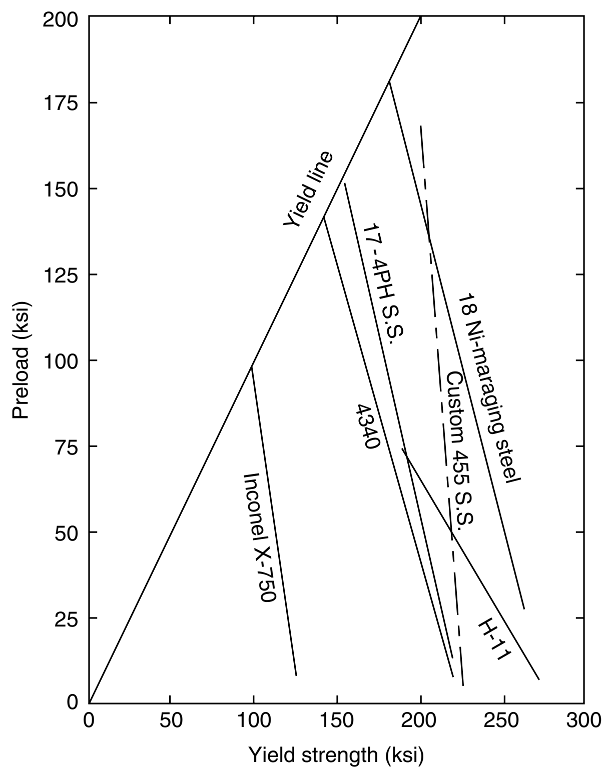


FIGURE 16.12 The average response, yield strength versus threshold stress (maximum safe preload) for six different bolting materials. The data are for 4 in. diameter bolts in aqueous environments. The response of LAQT steels would be essentially identical to that of the 4340 steel plotted here.

of LAQT steels in general is also shown here, by implication, since it's essentially identical to that of 4340 steel.) Figure 16.12, like Figure 16.11, is for 4 in. diameter bolts in a variety of aqueous and chloride environments.

As you can see from Figure 16.12, which is taken from various plots in Ref. [20], different materials exhibit different degrees of sensitivity to SCC in aqueous environments. I'm a little surprised by the relatively high standing of maraging steel in this group, since it generally gets low marks in the literature on SCC. These are all linear regression (average response) lines, however, which may explain things. The lower-bound line for the maraging steel, not shown in the figure, is well to the left of the average 4340 line.

Figure 16.13 shows the threshold stress versus yield strength response of miscellaneous low-alloy steels to three different environments: aqueous, including humid air, hydrogen gas, and hydrogen sulfide. As you can see, the preloads which can be applied safely to a low-alloy steel bolt in humid air are many times greater than those which can be applied if the bolt is exposed to hydrogen or H_2S. All of which emphasizes that knowing an experimentally determined K_{ISCC} doesn't eliminate the uncertainties for our predictions of SCC life, safe preload levels, etc. Reported data are too scattered, and the assumptions we must make on C and a and the many other factors which affect SCC are too uncertain to guarantee our predictions. As in so many aspects of bolting, your own tests and prior experience should count for more than the data and conclusions published by others.

If the tests and better data aren't available, try to keep preloads and working loads in the bolts as low as possible, consistent with any need to fight leaks or vibration loosening or other

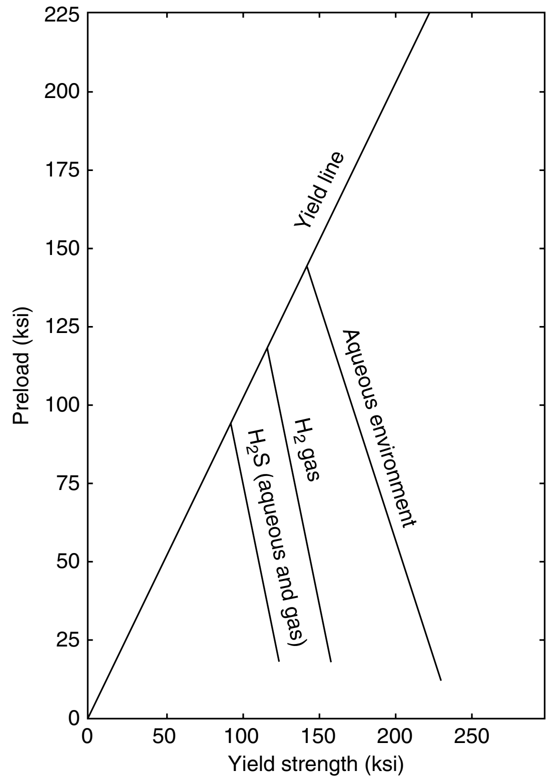


FIGURE 16.13 The average response, yield strength versus threshold stress lines, for LAQT steels in three different environments: aqueous or humid air, hydrogen gas, and hydrogen sulphide.

problems suggesting high tension. The IFI suggests that you should "be alert" if bolts are to be tightened to 50% of yield or higher; so you might try to keep them below that [10].

Another approach is to use very soft bolts—below 22 HRC, for example—to maximize K_{ISCC}, then tighten them to yield. This maximizes the clamping force available from the most SCC-resistant bolts, and has worked for petrochemical plants dealing with very aggressive electrolytes.

One final way to reduce stresses in the fastener is to shot-peen or pressure-roll the fastener in production to build up a compressive stress on the thread and other surfaces. This reduces the tensile stress when the fastener is placed under load.

Figure 16.14 gives us a final look at the relationship between SCC resistance and tensile stress in the fastener. I include these data because they relate to high-strength H-11 steel instead of to the LAQT steels covered in the previous several figures. The corrosive environment here is still the same—a sodium chloride solution. Note that even such a high-strength material as H-11 still has a threshold stress level below which it exhibits long—probably infinite—life in this environment [40].

16.3.5 Surface Coatings or Treatment

The K_{ISCC} value for a fastener can be affected by surface coatings or treatment. Figure 16.15 shows the effect various plating have on low-alloy steel fasteners, for example [34]. A coat of nickel under conventional cadmium led to satisfactory SCC resistance in H-11 bolts heat treated to a UTS of 260 ksi and then loaded to 90% of the proportional limit [40]. The same

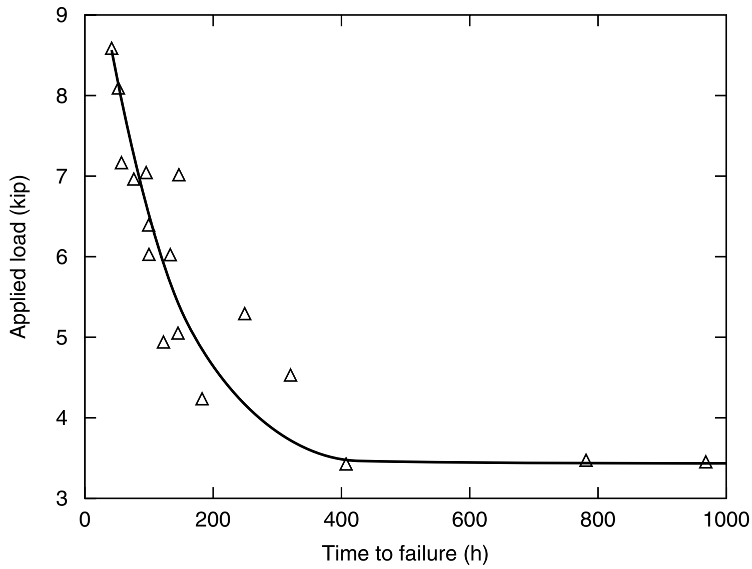


FIGURE 16.14 SCC is similar to fatigue failure in many ways. For example, SCC won't occur if tensile stress within the bolt is kept below a threshold value, which is a function of the environment. The tests reported here involved exposure to 31% NaCl bath. The fasteners were immersed for 10 min, then dried for 50 min. This cycle was repeated until failure. The ¼–28 × 2 uncoated AISI H-11 bolts, whose threads had been rolled after heat treatment, had infinite SCC life if subjected to a tensile force of 3,500 lbs or less in this environment. The 3,500 lbs load would correspond to an average tensile stress of 96,154 psi or about 37% of the 260 ksi UTS of these bolts. (From Irving, R.R., *Metalworking News*, 6ff, December 18, 1989.)

reference says that nickel and nickel–cadmium coatings in general result in a significant improvement in SCC resistance, as does electroplated cadmium with a chromate conversion coating. Vapor-deposited cadmium and a zinc chromate primer give less but some protection. But plating or surface treatment is not an automatic cure. Gold plating makes

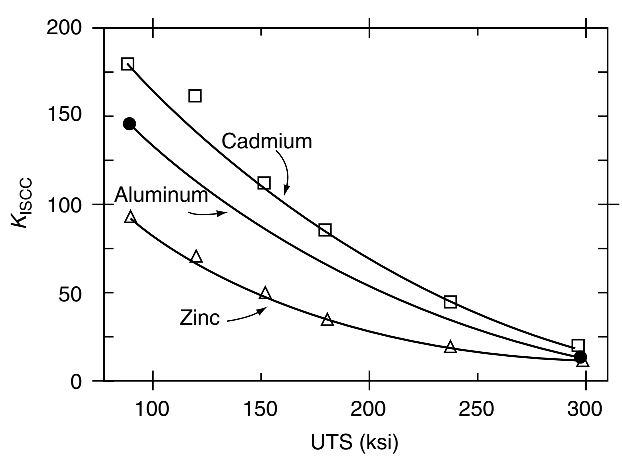


FIGURE 16.15 K_{ISCC} versus tensile strength for low-alloy steel fasteners electroplated with cadmium, aluminum, or zinc. As the chart shows, some platings are more beneficial than others in fighting SCC, but none help very much if the bolts are too hard. (From Raymond, L., *Am. Fast. J.*, 12ff, March/April, 1993.)

little difference [40]. Nor can you eliminate SCC by chrome plating 440 C stainless steel, for example, or by anodizing 2024-T3 aluminum. And carburizing a low-strength carbon steel to a surface hardness corresponding to a UTS of 200 ksi can make that usually resistant material susceptible [37].

16.3.6 Detecting Early SCC Cracks

As mentioned earlier, SCC cracks can usually be detected by magnetic particle or dye-penetrant techniques if bolts or studs are removed from the joint for inspection. It would obviously be desirable, however, to inspect them in place.

Conventional ultrasonic flaw detection equipment can be used on short studs (perhaps up to a foot in length), especially if the cracks are relatively large and are oriented more or less at right angles to the axis of the bolts. A crack that has penetrated through one-quarter or one-half the diameter of the bolt, for example, would be relatively easy to spot (but might also be propagating so rapidly that it would be detected too late).

More sophisticated ultrasonic techniques have been developed for detecting cracks with depths of 50 mils or larger in the threaded regions of bolts and studs which are up to 112 in. (285 cm) in length [18]. A cylindrically guided wave technique was used. The investigation was sponsored by the Electric Power Research Institute of California.

16.4 OTHER TYPES OF STRESS CRACKING

Hydrogen embrittlement and SCC are our main concerns. We should, however, take a brief look at the other two stress cracking mechanisms before we finish the discussion.

16.4.1 Stress Embrittlement

Stress embrittlement is the same as hydrogen embrittlement except that it starts with a chemical reaction between a noncoated fastener and the atmosphere, for example, a reaction between a high-carbon steel and hydrogen sulfide. The hydrogen is introduced by the environment rather than by a plating process.

High-carbon and high-strength steels in general are the most susceptible to stress embrittlement [10]. High-strength martensitic stainless steels are also very susceptible, whereas austenitic and ferritic stainless materials will rarely cause problems [24].

Other bolting materials very resistant to stress embrittlement include ASTM A193/A193M Grade B7M bolts and A1942M nuts. The bolts, however, should not be hardened above 99 HRB; the nuts should be 22 HCR or less [10].

16.4.2 Hydrogen-Assisted Cracking

Hydrogen-assisted cracking is basically hydrogen embrittlement or stress embrittlement combined with SCC. A corrosion cracking process, in other words, is aided by a buildup of hydrogen pressure, with the hydrogen coming either from a plating process or from a chemical reaction with the environment.

Hydrogen-assisted cracking, for example, was encountered a few years ago in some low-carbon martensite bolts tempered to 40 HRC and higher (metric Class 12.8). The bolts, used in automotive rear suspension applications, began to fail unexpectedly about 3 years after they were installed. The failure mechanism was identified as hydrogen-assisted cracking, which had been delayed but not prevented by the fact that the fasteners also had a decarburized (i.e., softer) surface [23].

16.5 MINIMIZING CORROSION PROBLEMS

16.5.1 In General

To fight corrosion we must find a way to reduce or eliminate one or more of the four essential conditions: the anode, the cathode, the electrolyte, or the metallic connection between anode and cathode. Anything we can do to reduce the efficiency of this corrosion "battery" will be helpful.

Note that we're now talking about reducing corrosion in general. We've already considered the steps required to combat the special case of stress corrosion in its various forms.

16.5.2 Detailed Techniques

Some of the ways in which we can accomplish our basic goals of destroying the corrosion battery or of minimizing its effectiveness are fairly obvious. If we can keep a bolted joint dry, for example—perhaps providing a roof, or drainage holes—we remove the electrolyte. Simple cures of this sort, of course, aren't always possible. Some of the other things we can do are far from obvious. Here are some of the things commonly suggested.

1. Use bolts made of materials classified as corrosion-resistant; such as austenitic stainless steel, titanium, Inconel, and MP35N. See Table 16.1 for details.
2. When designing fastener, joint, and structure, select materials as close together as possible in the galvanic series, minimizing electrical potential differences. The best solution of all, of course, is to use identical materials (although we would still be faced with differences in potential created by differences in stress level, temperature, oxide film, etc.).
3. Since the anode is destroyed, it is desirable to have a large anode and a small cathode. It now becomes relatively easy for the anode to supply the electrons demanded by the cathode; only a relatively small percentage of the anode is destroyed in a given time. By comparison, it would be very bad to have small aluminum fasteners (small anode) holding together a large steel structure (large cathode). The anode would be destroyed very rapidly by the demands of the cathode. In general, therefore, the fasteners should be the most noble, most cathodic element in the joint.
4. Break the metallic circuit connecting anode to cathode by electrically insulating one or the other with paint or other coatings, or with spacers and the like. If you use coatings, you must keep them in good repair, however. A small break in the material coating a large anode will produce a small anode—that portion of the body which is no longer coated. This can lead to very rapid corrosion at that point. Coatings, in general, are such an effective way to fight corrosion they deserve special treatment. We'll take a detailed look at them in Section 16.6.
5. Introduce a third electrode—a sacrificial anode—to reverse the flow of current in the battery. As an example, let's assume that you must use steel bolts to clamp brass joint members together, and are troubled by rapid corrosion of the bolts. If blocks of aluminum are placed near the bolts, they will act as a sacrificial anode. Both steel and brass now become cathodes, absorbing electrons from the aluminum, which is rapidly destroyed. The sacrificial aluminum anode is replaced from time to time to protect the steel bolts. As we'll see, some types of coating provide galvanic protection of this sort, too.
6. In some critical applications, an actual battery—such as an automobile battery—is physically connected, in the reverse direction, between the natural anode and cathode. Since the potential of the battery exceeds that of the corrosion battery, the current is reversed and that part which would have been an anode is protected.

TABLE 16.1
Corrosion-Resistant Materials

Material	Notes
Steel, coated	UTS 80–125 ksi Low carbon, medium carbon, and low-alloy steels can be made more resistant to atmospheric corrosion by coating or by plating them. Examples of such materials: A193 except the B8 series; A325, A490, SAE J429 materials, metric materials 4.6–12.9.
Austenitic stainless steels	UTS 75–120 ksi Most common of the stainless steels and more corrosion-resistant than the three other types listed below. Nonmagnetic. Can't be heat-treated but can be cold-worked. Good high and low temperature properties: 321 can be used up to 800°F (816°C), for example. Examples: A193 B8 series, A320 B8 series, any of the 300 or 18–8 series materials such as 303, 304, 316, 347, etc. [21].
Ferritic stainless steels	UTS 70 ksi Can't be heat-treated or cold-worked. Magnetic. Examples: 430 and 430F [21].
Martensitic stainless steels	UTS 70–180 ksi Heat treatable, magnetic. Can experience stress corrosion if not properly treated. Examples: 410, 416, 431 [21].
Precipitation hardening stainless steels	Typical UTS 135 ksi Heat treatable. More ductile than martensitic stainless steels. Examples: 630, 17-4PH, Custom 455, PH-1308 Mo, ASTM A453-B17B, AISI 660 [21,46].
Nickel-based alloys	
Nickel–copper	UTS 70–80 ksi Can be cold-worked, but not heat-treated. Example: Monel [21].
Nickel–copper–aluminum	UTS 130 ksi Can be heat-treated and also cold-worked. Good low-temperature material. Example: K-monel [21].
Titanium	UTS 135–200 ksi Good corrosion resistance. Low coefficient of expansion. Has a tendency to gall more readily than some other corrosion-resistant materials. Expensive. Example: Ti6A1-4V [21].
Superalloys	UTS 145–286 ksi High-strength materials with excellent properties at high and low temperatures. Primarily used in aerospace applications. Expensive. Some, such as MP35N, are virtually immune to marine environments and stress corrosion cracking. Examples: H-11, Inconel, MP35N, A286, Nimonic 80A. MP35N, Inconel 718 and A286 are especially recommended for cryogenic applications [33,46].
Nonferrous materials	There are many nonferrous fastener materials which can provide outstanding corrosion resistance in applications which would rapidly destroy more common bolt materials. The main drawback to these materials is a general lack of strength, but that can sometimes be made up by using fasteners of a larger diameter and/or by using more fasteners. Here are a few of the many materials available:
Silicon bronze	UTS 70–80 ksi
Aluminum	UTS 13–55 ksi
Nylon	UTS 11 ksi

Source: The references cited at the end of this chapter.

7. Seal crevices and the like to prevent accumulation of oxygen-starved moisture. Sealing materials can include paint, putty, nonwicking plastic washers (such as nylon), and the like.

8. Minimize stresses or stress concentrations by providing fillets, polishing or shot peening surfaces, designing for uniform distribution of external loads, preloading bolts uniformly, using conical washers to minimize bending stress, etc.—all the things we have talked about in previous chapters for minimizing stress variations, fatigue damage, and the like.
9. If you have some control over the electrolyte, you can sometimes add inhibitors which reduce its capacity for transporting electrons (ions). We use such materials to protect the radiators in automobiles, for example.
10. Use materials that resist the electrolyte. The IFI publishes a long table showing how much resistance such materials as nylon, brass, and stainless steel have to various chemicals and solutions [4]. Type 304 stainless steel, for example, has excellent resistance to such materials as turpentine, sulfur, fresh water, or wine; but it has poor resistance to sulfuric acid or zinc chloride. Some very general information on corrosion-resistant materials is given in Table 16.1 but you'll find the very long and detailed IFI presentation more helpful, I'm sure.
11. If all else fails, or is impractical for your application, you might consider replacing the bolts periodically, before they fail. This can be a less expensive solution to a corrosion problem than a more technical response. No need to decide exactly what type of corrosion you're facing; no need to search for that perfect bolt material or coating; just throw them away once in awhile. Fresh anodes! It's a valid response.

16.6 FASTENER COATINGS

16.6.1 In General

One of the most common ways to combat corrosion in bolts is to make the bolts of a corrosion-resistant material, the type of material usually being a function of the industry you are in. Petrochemical people, for example, favor various kinds of stainless steel. Aerospace favors Inconel and titanium. Automotive users favor such things as aluminum and plastics. Marine users like silicon bronze, Monel, and titanium.

In spite of the popularity of corrosion-resistant base materials, a more popular way to protect bolts is to coat them with a protective layer of some sort. One source, for example, says that 90% of all carbon steel bolts are coated with something or other [10].

Coatings can resist corrosion in one of three ways [15].

1. They can provide barrier protection, isolating the bolt from the corrosive environment, and breaking the metallic circuit which connects the anode to the cathode.
2. They can provide "passivation" or "inhibition," slowing down the corrosion, making the battery less effective.
3. They can provide galvanic or sacrificial protection, reversing the direction of current in the battery to protect the more important electrode (in this case the bolt).

We'll look at coating examples in a minute. Note first, however, that barrier protection—involving such things as paint, cad plating, etc.—requires a perfect coating. A small break in the coating can create a tine anode, which will erode more rapidly than would the bolt as a whole. Sacrificial coatings on the other hand—such as aluminum or zinc—do not have to be perfect. As long as some of what remains of the coating is near the material to be protected—and is immersed in the electrolyte—the coating will provide some protection.

Coatings in general have an enormous impact on fastener performance. Before we look specifically at their role in the war against corrosion, it's worth noting that they also provide other important features.

Certain coatings, for example, will have a very desirable effect on the coefficient of friction, reducing the drag between male and female threads and between nut and workpiece. Perhaps more important, good coatings also reduce the amount of variation or scatter in friction, improving the accuracy of the torque–tension relationship (see Table 7.1).

A less important, but also popular, use for coatings is to change the appearance of the fastener—matching colors to hide the fasteners, or preventing rust for appearance sake rather than for any structural reasons, or even to make the fasteners stand out as design accents.

Our main concern at the moment, however, is with corrosion protection.

Coatings are often divided into three groups: organic, metallic, and composite. Let's look at some examples.

16.6.2 Organic Coatings

Organic materials are derived from plant or animal matter and contain compounds of carbon. They can provide as much as twice the corrosion protection of such things as cadmium or zinc plating and have the additional advantage that they can be provided in a wide variety of colors. Like the other types of coatings we will consider later, organics can be applied by many different methods—dip/spin, spraying, painting, etc. The resulting layers are not always uniform in thickness; materials applied by these techniques tend to build up in the cracks and crevices of a fastener, as shown in Figure 16.16.

One of the advantages of organic coatings is that they eliminate the hydrogen embrittlement problems sometimes caused by the electroplating process. Also, they involve no heavy metals, such as zinc or cadmium, which are of concern to environmentalists. Organics such as the fluorocarbons, furthermore, can provide more resistance to salt spray than can some of the more popular metallic coatings such as cadmium or zinc.

Typical organic coatings are described below.

16.6.2.1 Paints

A few years ago, alkyd and phenolic paints were very popular, but better, more lubricious materials such as the fluorocarbons and other polymers (to be discussed below) have generally replaced them [9]. Zinc-rich paint is still popular in structural steel work, but it is generally the entire joint, including the bolts, which is painted rather than the bolts alone.

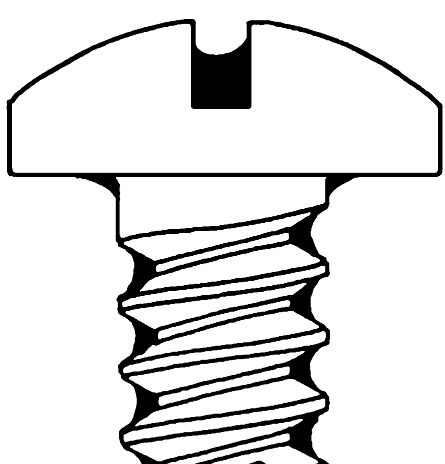


FIGURE 16.16 Organic coatings tend to build up in thread roots and other crevices as shown here (exaggerated).

16.6.2.2 Phos-Oil Coatings

Zinc phosphate and manganese phosphate are mild acids. If fasteners are placed in a solution of one of these materials and then tumbled, their surfaces will become slightly porous, a "chemical conversion coating" has been created. Such surfaces provide an excellent base for the retention of oils, waxes, or other organic lubricants [10]. Such coatings as phosphate plus oil, phosphate plus paint plus oil, phosphate plus zinc-rich paint, etc. have all been very popular [15].

16.6.2.3 Solid-Film Organic Coatings

A bonded, solid-film lubricant provides a coating which might be described as "a thin layer of slippery paint." The films generally consist of an air-dried or oven-dried resin binder in which are embedded tiny particles of one or more lubricating or corrosion-resistant materials. These can include such things as molybdenum disulfide, graphite, or polytetrafluoroethylene (PTFE)—at least as far as the organic coatings are concerned [11]. We'll consider composite solid-film coatings later. (A composite coating has more than one active lubricating and corrosion-resistant component.)

Solid-film coatings are available in a wide variety of proprietary formulations. For example, the fluorocarbons (which are used on carbon steel, stainless steel, and aluminum fasteners) are sold under such trade names as Teflon-S, Stalgard, Xylan, Emralon, and Everlube. In general, the fluorocarbons can be used in applications which involve temperatures ranging from −450°F (−268°C) to +400°F (+204°C).

16.6.3 Inorganic or Metallic Coatings

Inorganic materials are any materials not containing plant or animal matter, hence inanimate. The class can include such things as ceramic coatings, but we're going to concentrate on the more common metallic materials.

Metallic coatings can be applied to fasteners by a variety of processes, including electroplating, hot dipping, vacuum deposition, and the so-called mechanical plating techniques.

If electroplating is used, the inorganics tend to build up on the sharp edges of fastener surfaces rather than in the cracks and crevices, as shown in Figure 16.17.

Let's look at some examples.

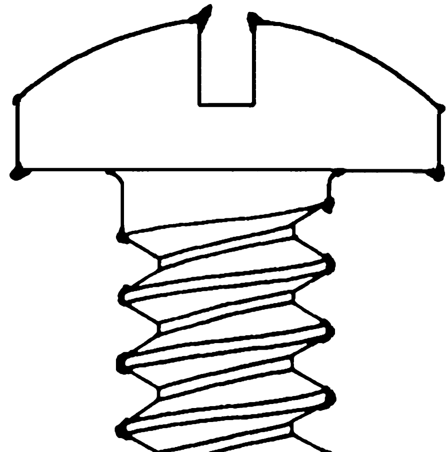


FIGURE 16.17 Metallic coatings build up on the tips of threads and other sharp edges and corners.

16.6.3.1 Electroplated Coatings

Cadmium and zinc are the two most common electrodeposited coatings, although more expensive materials, such as nickel, chromium, and silver, can also be applied this way and are used in special applications.

Cadmium protects fasteners more effectively than zinc does in marine environments; but zinc is a better choice in most industrial environments (a combination of cadmium and zinc does better than either in both environments) [13].

A few years ago, cadmium came under attack from the general public because a cyanide rinse used in the plating process can create a dangerous effluent [7]. The final coating was not dangerous, but the process was considered environmentally unsound, and so steps were taken to find substitute coatings.

This search is probably still going on. One popular solution to the problem is to use zinc instead of cadmium. One of the attractions of zinc, incidentally, is that it is a cheaper coating. But zinc is less lubricious than cadmium, so a given preload requires a higher assembly torque. Zinc also tends to double the scatter in the torque–tension relationship, which has created a number of problems in automated assembly operations. Zinc, furthermore, can provide significantly less corrosion protection than cadmium in certain environments.

Another disadvantage is that zinc will develop a dull, white corrosion product called "white rust" unless protected by a clear or colored chromate coating [10]. One advantage of zinc is that it gives galvanic protection as well as barrier protection. Cadmium provides only barrier protection. Techniques have been developed to process cadmium's cyanide rinse effluent more effectively. Although these increase the cost of cadmium plating, the lubricity and corrosion problems encountered when zinc is substituted for cadmium have led to a rebirth of interest in cadmium. The search for other substitutes was partially successful, however, so cadmium will probably never regain its previous popularity.

We'll look at some of the coatings selected as cadmium substitutes in a minute. But first, let's continue our survey of coating types.

16.6.3.2 Hot-Dip Coatings

In general, two materials are applied by hot-dip techniques: aluminum and zinc. Fasteners coated with aluminum are said to be aluminized. Zinc-coated fasteners have been galvanized.

These are both low-cost coatings and are generally used on relatively inexpensive, high-strength fasteners; ASTM A325 structural steel bolts are often galvanized, for example.

The hot-dip process is difficult to control, so the resulting coatings tend to vary quite a bit in thickness. Threads are generally undercut or overcut to provide room for the coating—and then should be recut with a tap or die after the coating has been applied. Although this process is relatively common, it can result in a significant reduction in the stripping strength of the threads, a problem that is generally avoided if the fasteners are mechanically galvanized (see below) rather than hot-dip galvanized.

Hot-dip-galvanized fasteners can have more corrosion resistance than mechanically galvanized ones, however, because the coating thickness is greater. On the other hand, because of the difficulties of controlling the hot-dip process, corrosion resistance can vary more for the hot-dip products. Galvanizing in general, incidentally, tends to give greater corrosion protection than electroplating, again because of the greater thickness.

16.6.3.3 Mechanical Plating

Fasteners are said to be mechanically plated when a ductile metal such as cadmium, zinc, or tin is cold-welded onto the metal substrate by mechanical energy. Glass beads are usually used to do the welding.

Fasteners that have been mechanically plated with zinc are said to be mechanically galvanized. As already mentioned, coating thicknesses are much more uniform than they are with the hot-dip galvanizing process, so it is not necessary to chase the coated threads with a die.

One big advantage of mechanical plating, as opposed to electroplating, is that no baths are involved, eliminating hydrogen embrittlement and detempering concerns.

In one recently developed process, combinations of aluminum and zinc can be uniformly deposited on fastener surfaces without buildup in thread roots, etc. These combinations are said to give the durability of aluminum coatings plus the galvanic protection of zinc (aluminum alone does not always give sufficient galvanic protection because it will form oxide films which partially or wholly isolate it from the electrolyte and other metals).

The mixed aluminum–zinc coating thickness run about half a mil, making overlapping unnecessary, and, therefore, preserving thread strength. The coatings are applied at room temperature so that detemper and hydrogen embrittlement are not problems. Costs are comparable to those associated with galvanizing.

A phosphate coating is usually applied on top of the aluminum–zinc coat for better corrosion resistance and lubricity. The resulting coating is said to provide better corrosion resistance than galvanizing [13].

16.6.3.4 Miscellaneous Coating Processes

A large number of other coating processes are available. High-strength steels and titanium, for example, are sometimes coated with aluminum in a process called ion vapor deposition (IVD). The resulting coating is said to have excellent resistance to SCC and to be usable to temperatures as high as 950°F (510°C) [7].

In another process, a nonporous layer of high-purity nickel is actually alloyed with the surfaces of carbon or alloy steel fasteners. The process gives good resistance to severe acids and alkalines and creates no buildup. Additional coats of cadmium or zinc are sometimes added for additional corrosion protection and lubricity [8].

16.6.4 Composite Coatings

It's not entirely clear to me when a multiple-component coating should be classed as organic or metallic rather than composite, but, in general, composite coatings consist of a wide variety of combinations of active organic or inorganic materials or both applied in separate layers and in various mixtures. As one example, using a dip/spin process, one might apply zinc to a fastener for corrosion protection, cover this with an organic paint for color, and finish up with a PTFE coating for lubricity [16]. Another available combination is inorganic aluminum coated with phosphate coated with a chromate. Aluminum with an inorganic ceramic binder is another offering [15]. In fact, various combinations of aluminum and inorganic coatings are said to have replaced cadmium across the board in airframe applications, and are also recommended for electrical connections. Such coatings are said to give torque–tension characteristics virtually identical to those of cadmium [14].

Note that stainless steel fasteners are often used in aerospace applications, which usually involve aluminum structural members. A combination of stainless steel and aluminum leads to galvanic attack, which can be prevented by coating the stainless steel fasteners with an aluminum/inorganic coating [14].

The number of composite (and other) coatings available is nearly endless, but the preceding should be sufficient introduction. Further details on a number of coatings are given in Table 16.2. I don't mean to recommend those listed by including them nor do I mean not to recommend others by not listing them. The table merely lists typical offerings

TABLE 16.2
Fastener Coatings

Type	Trade Name[a]	Manufacturer[b]	Corrosion Resistance (hr)[c]	Temp. Range (°F)	Ref. and Notes
Aluminum in ceramic binder	SermaGard	SermaGard/Teleflex	1500	+1200	[e]
Alum. filled, corrosion inhibitive, organic binder	Alumazite 'Z'	Tiodize Company	—	+500	8
Alum., inorganic binder and chromates	SermaGard	SermaGard/Teleflex	Many thousand	+900	[e]
Ion vapor deposition alum.	IVD aluminum	SPS Technologies	1000	+925	[e]
Cadmium, plain	—	3M	50–320[d]	—	8
Cadmium plus chromate	—	3M	200–2000[d]	—	8
Cad-tin (50/50)	—	3M	50–320[d]	—	8
Cad-tin-chromate	—	3M	200–2000+[d]	—	8
Cad-zinc(25/75)-chromate	—	3M	2100–6700[d]	—	8
Cad-zinc(50/50)-chromate	—	3M	5900–6720[d]	—	8
Fluorocarbon (PTFE)	Teflon	DuPont	—	−450 to + 400	7
Fluorocarbon + resin binder	Reflon-S	DuPont	1100–>3000	−450 to + 400	7,44
PTFE plus binding resin	Emralon 305	Acheson Colloids Co.	500	+500	7
PTFE in phenolic binder	Everlube 6108	E/M Corporation	500	−100 to + 400	11
Fluoropolymers with high-temp. organic polymers plus corrosion inhibitors	Xylan	Whitford Corporation	96–2000[d]	—	7
Graphite plus binder	Dag	Acheson Colloids Co.	Poor	+850	7
Graphite plus binder	Pepcoat	G* Chemical Company	1200	−350 to + 650	[e]
Molydisulfide + epoxy binder	Ecoalube 642	E/M Corporation	500	−365 to + 500	11
Molydis. + silicone binder	Everlube WL-135	E/M Corporation	100	−365 to + 1300	11
Molydis. in bonding resin	Lube-Lok 2109	Electrofilm Inc.	—	−459 to + 450	[e]

MoS_2/graphite/phenolic resin	Electrolube E-40	Electrofilm Inc.	—	−459 to + 450	[e]
Molydis. plus binder	Molydag	Acheson Colloids Co.	—	+650	7
Surface alloy of nickel	Sanbond	AMCA International	—	+600	8
Surf. alloy Ni-Cad-chromate	Sanbond-Cad	AMCA International	2000	+ 600	8
Surface alloy Ni + zinc	Sanbond-Z	AMCA International	600	+600	8
Silver + indium + binder	Lube-Loc 4253	Electrofilm Inc.	—	−459 to +450	[e]
Zinc, plain	—	3M	36–192[d]	+500	8
Zinc + clear chromate	—	3M	24–192[d]	—	8
Zinc plus yellow chromate	—	3M	100–650[d]	—	8
Zn-tin(75/25)-chromate	—	3M	400–1000[d]	—	8
Zn-tin(50/50)-chromate	—	3M	600–1300[d]	—	8
Alloyed coatings of electro-deposited zinc and nickel	Ni-Alloy	Deveco Corporation			
Yellow bronze chromate over zinc plating	Macrocor 250	Nucor Fastener	250	0	41
Thermosetting epoxy coating containing aluminum flake over zinc plating	Magnigard Silver 17	Magni Industries, Inc.	1000–10,000	—	42
Phenoxy topcoat over organic, zinc-rich film over zinc phosphated steel	Magnigard-Black	Magni Industries, Inc.	100–1000	—	42
Aqueous coating dispersion containing chromium, proprietary organics, and zinc flake	Dracromet 320	Metal Coatings International	[f]	700	43

[a] Most, if not all, of these trade names are registered.
[b] Addresses for these manufacturers are given in Table 18.2.
[c] Usually tested per ASTM B117.
[d] Depending on the thickness of the coating.
[e] From the manufacturer's literature.
[f] Said to have "three times the resistance of zinc chromate coatings."

TABLE 16.3
Fastener Coating Suppliers

Acheson Colloids Company Port Huron, Michigan (Dag, Molydag, Emralon)
AMCA International New Bedford, Massachusetts (Sanbond)
Deveco Corporation Addison, Illinois
DuPont Wilmington, Delaware (Teflon)
Elco Industries Rockford, Illinois
Electrofilm Inc. Valencia, California (Lub-Lok, Electrolube)
E/M Corporation West Lafayette, Indiana (Ecoalube, Everlube)
Fel-Pro Skokie, Illinois (N5000, C5A)
G* Chemical Corporation Wayne, New Jersey (Pepcoat)
MacDermid Inc. Waterbury, Connecticut (mechanical Al plus Zn)
Magni Industries, Inc. (Magnigard) Birmingham, Michigan
Metal Coatings International (Dacrotizing, Dacrosealing) Chardon, Ohio
Never Seez Compound Corp. Broadview, Illinois (Never-Seez)
Nucor Fastener Corp. St. Joe, Indiana
Serma Gard Division of Teleflex Incorporated Limerick, Pennsylvania (Sermatel, SermaGard)
SPS Technologies Jenkintown, Pennsylvania (IVD Aluminum)
3M Plating Systems Dept. Commercial Chemicals Division St. Paul, Minnesota (mics. plates)
Tiodize Company Huntington Beach, California (Alumazite)
Whitford Corporation West Chester, Pennsylvania (Xylan)

that are available, in most cases, from many sources (addresses for the suppliers are listed in Table 16.3).

The buildup of composites is shown in Figure 16.18. There's some buildup in thread roots, but much less than is (sometimes) the case with organic coatings. There's also some buildup on sharp edges but, unlike the edge buildup of metallic plating, the composite material accumulated on edges will break off easily.

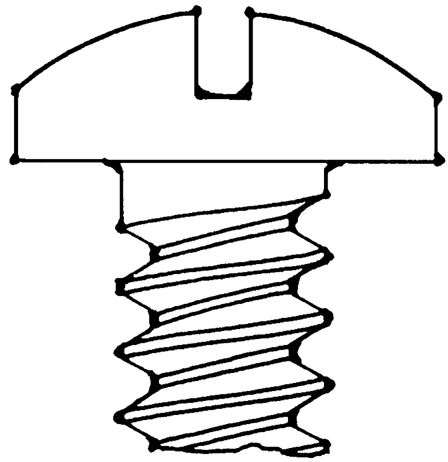


FIGURE 16.18 Composite coatings build up in crevices and on sharp edges. The crevice buildup, however, is usually less than that encountered with organic coatings; and the edge buildup breaks off easily.

16.6.5 Rating Corrosion Resistance

One of the columns in Table 16.2 gives the resistance of various coatings to salt spray. Many different types (e.g., concentrations) of salt spray are possible; I believe that most, if not all, of the data given in Table 16.2 came from use of the ASTM B-117 procedure. Resistance is given in hours of successful exposure (before excessive damage has occurred).

Obviously resistance to salt spray doesn't completely define the corrosion resistance of a coating. You may be primarily interested in resistance to something else, like a specific acid or fuel or sulfur or wine. Salt spray is commonly used to rate coatings, however, and so I've used it here. Suppliers can presumably tell you how well their coatings will stand up to other types of environment, although the information they'll give you is often fairly general (good, fair, poor, etc.).

ASTM Standard B-117 is not the only possible way to test for corrosion, of course. Many manufacturers have developed special procedures appropriate for the environments seen by their products. Another published standard is the German DIN 50018, called the "Kesternich test." These tests, which are also specified in the Factory Mutual Standard FM 4470, are conducted in a special cabinet and simulate the effects of acid rain [45].

Predicting corrosion results is not easy. As is so often the case with bolted joints, laboratory tests can rarely predict field results. If tests must be made in a laboratory environment, try to duplicate field conditions as accurately as possible. Test actual joints, not chunks of metal with bolts stuck in them. Better yet, find a way to conduct tests in the field, on the actual job sites, if possible. And remember that past experience, recorded in maintenance records or the like, may be a more accurate way to evaluate a corrosion-resistant material or coating than a quick and dirty lab test.

16.6.6 Substitutes for Cadmium Plate

As mentioned earlier, many people have been actively seeking acceptable substitutes for cadmium plating. Let's take a brief look at some of the resulting choices.

Ideally, a perfect substitute would equal or exceed cadmium plate in corrosion resistance, lubricity, and cost. Substitutes that avoided the hydrogen embrittlement problems sometimes associated with cadmium plate would be especially attractive. Although none of the substitutes found so far match all of cadmium's characteristics, they're close enough to be acceptable.

Here are some of the coatings which have been adopted as substitutes. (See Table 16.3 for trade names and sources.)

- Solid-film organics. Fluorocarbons or molybdenum disulfide in resin binders has been used by some of the automotive companies [8].
- Electrodeposited, alloyed coatings of zinc plus tin have been found to be a good substitute for cadmium when magnesium auto blocks are involved (e.g., European auto manufacturers) [8].
- Aluminum with inorganic binders (e.g., ceramics) provides torque–tension relationships virtually identical to cadmium and have replaced cadmium in most airframe applications [14].
- IVD aluminum is being used in place of cadmium in many high-performance applications [8].
- Tin flash over electroplated zinc. The zinc provides galvanic protection; the tin prevents galling.
- Yellow bronze chromate over zinc. This coating has been adopted by the Defense Industrial Supply Center (DISC) as a substitute for cadmium, especially on Grade 8 fasteners [411].

EXERCISES

1. What are the essential conditions for corrosion?
2. If corrosion occurs, which element is eaten away, the anode or the cathode?
3. Will an uncoated ASTM A325 bolt act as an anode or a cathode?
4. What combination of factors can lead to stress cracking of a bolt?
5. What is the most common source of absorbed hydrogen?
6. Which is more apt to fail by hydrogen embrittlement, an SAE J429 Grade 5 or a Grade 8 fastener and why?
7. Name at least three ways to reduce the possibility of hydrogen embrittlement.
8. What are the essential conditions for stress corrosion cracking (SCC)?
9. Name at least three ways to reduce the possibility of SCC.
10. Which is better, a relatively small anode or a large one and why?
11. Is coating a fastener always helpful in fighting corrosion resistance, even if there are minor breaks in the coating?
12. Name some substitutes for electroplated coatings.

REFERENCES AND BIBLIOGRAPHY

1. Myers, J.R., *Fundamentals and Forms of Corrosion*, University of Wisconsin-Extension, Madison, WI, 1980.
2. Neill, W.K., Corrosion prevention—basic theory and practical design pointers, Presented at the Using Threaded Fasteners seminar, University of Wisconsin-Extension, Madison, WI, May 9, 1978.
3. Latanision, R., O.H. Gastine, and C.R. Compeau, Stress corrosion cracking and hydrogen embrittlement—differences and similarities, Paper presented at Symposium on Environment Sensitive Fracture of Engineering Materials, October 1965.
4. Fastener Standards, Industrial Fastener Institute, Cleveland, OH, 1988, pp. B40–B43.
5. Brochure, Standco/Bolt Division of Standco Industries, Inc., Houston, TX.
6. Aimone, M.A., Fundamentals and the forms of corrosion, Presented at the Using Threaded Fasteners seminar, University of Wisconsin-Extension, Madison, WI, May 1980.

7. Emrich, M., Corrosion protection for fasteners, part 1, *Assembly Engineering*, pp. 36–40, October 1982.
8. Emrich, M., Corrosion protection for fasteners, part 2, *Assembly Engineering*, pp. 17–21, November 1982.
9. 1984 fastening and joining reference issue, *Machine Design*, pp. 50, 52–53, November 15, 1984.
10. *Fastener Standards*, Industrial Fastener Institute, Cleveland, OH, 1988, pp. B30–B49.
11. Gresham, R.M., Bonded solid film lubricants for fastener coatings, *Fastener Technology International*, April/May 1987.
12. Taylor, E., SermeTel W aluminum coating for aerospace fasteners, SPS Laboratory Report no. 5392, SPS Technologies, Jenkintown, PA, March 24, 1972, reprinted October 30, 1980.
13. From a speech given by an employee of MacDermid, Inc., Waterbury, CT, in a meeting of the Research Council on Structural Connections, Miami, FL, June 8, 1988.
14. From a speech given by A.E. Simmons, Jr., of the SermaGard Division of Teleflex, Inc. (USA) at the 1988 Annual Meeting of the Industrial Fastener Institute, Laguna Niguel, CA, February 22, 1988.
15. From a speech given by T. Dorsett of Metal Coatings International Inc. at the 1988 Annual Meeting of the Industrial Fasteners Institute, Laguna Niguel, CA, February 22, 1988.
16. From a speech given by Roger Kelly of the Man-Gill Chemical Company at the 1988 Annual Meeting of the Industrial Fasteners Institute, Laguna Niguel, CA, February 22, 1988.
17. Chung, Y., Threshold preload levels for avoiding stress corrosion cracking in high strength bolts, Technical Report no. 0284-03 EV, Bechtel Group, San Francisco, April 1984.
18. Light, G.M., H.R. Joshi, and S.-N. Liu, Detection of stress corrosion cracks in reactor pressure vessel and primary coolant system anchor studs, Southwest Research Institute, San Antonio, TX, about 1985.
19. Rungta, R. and B.S. Majumdar, Stress corrosion cracking of alternative bolting alloys, Report on research project 2058-12, Battelle Memorial Institute, Columbus, OH, December 1985.
20. Czajkowski, C.J., Bolting applications, NUREG/CR-3604 BNL-NUREG-51735, U.S. Nuclear Regulatory Commission, Washington, DC, May 1984.
21. Czajkowski, C.J., Testing of nuclear grade lubricants and their effect on A540 B24 and A193 B7 bolting materials, Report NRC FIN A-3011, Brookhaven National Laboratories, Upton, NY, March 1984.
22. Rowland, M.C and T.C. Rose, Tests on thread lubricants for use in nuclear reactors, Report APED-4422, R63APE 29, General Electric Co., San Jose, CA, December 1963.
23. Hughel, T.J., Delayed fracture of Class 12.8 bolts in automotive rear suspensions, *Fastener Technology*, pp. 36–38, April 1982.
24. Blake, A., *What Every Engineer Should Know about Threaded Fasteners*, Marcel-Dekker, New York, 1986.
25. Degradation of threaded fasteners in the reactor coolant pressure boundary of PWR plants, IE bulletin no. 82-02, U.S. Nuclear Regulatory Commission, Office of Inspection and Enforcement, Washington, DC, June 2, 1982.
26. Taylor, E., Stress corrosion crack protection from coatings on high strength H-11 steel aerospace bolts, Special Technical Publication 518, ASTM, Philadelphia, 1972.
27. Greenslade, J., Hydrogen embrittlement testing requirements for bolts per ASTM F606-90, section 7, *American Fastener Journal*, pp. 8ff, January/February 1993.
28. Raymond, L., Hydrogen embrittlement of fasteners: problems and solutions, *American Fastener Journal*, pp. 85ff, May/June 1993.
29. Grobin, A.W., Jr., Hydrogen embrittlement problems, *ASTM Standardization News*, March 1990.
30. Mlody, M., New method for the prevention of hydrogen embrittlement, *Fastener Technology International*, pp. 74ff, February 1994.
31. Raymond, L., Test methods for hydrogen embrittlement of fasteners: meaning and use, *American Fastener Journal*, May/June 1994.
32. Raymond, L., Titanium alloys, *American Fastener Journal*, pp. 8ff, May/June 1992.
33. Raymond, L., Aluminum alloys, *American Fastener Journal*, pp. 40ff, January/February 1993.
34. Raymond, L., Copper alloys/steels, *American Fastener Journal*, pp. 12ff, March/April 1993.
35. Faller, K. and W.A. Edmonds, The incentives for titanium alloy fasteners in the Navy/marine environment, *American Fastener Journal*, pp. 44ff, September/October 1992.

36. Notes prepared for a bolting seminar by Raymond Tootsky of McDonnell Douglas Corp., Long Beach, CA, December 1991.
37. Greenslade, J., Zinc plating socket head cap screws is a risky business, *American Fastener Journal*, pp. 33ff, March/April 1993.
38. Greenslade, J., Military hydrogen embrittlement requirements, *American Fastener Journal*, pp. 9ff, February/March 1989.
39. Raymond, L., Use of a hard plate in hydrogen embrittlement testing is not everything it's cracked up to be, *American Fastener Journal*, pp. 49ff, March/April 1994.
40. Hood, A.C., Preventing stress-corrosion cracking in threaded fasteners, *Metal Progress*, September 1967.
41. Irving, R.R., DISC gives its approval to new coating, *Metalworking News*, pp. 6ff, December 18, 1989.
42. From a brochure published by Magni Industries, Inc., Birmingham, MI, 1993.
42. From information provided by John Dutton of Metal Coatings International, Chardon, OH, September 1993.
43. From a brochure provided by Prosper Engineering, Ltd., Irvine, Ayshire, UK, 1993.
44. Willis, D.P. Jr., Engineered coatings for fasteners, *Product Finishing*, May 1987.

17 Selecting Preload for an Existing Joint

We have reviewed all of the main topics which concern the beginner who wants to learn about the design and behavior of bolted joints. Even though this was only an introduction, it has been a long and sometimes complex story. It's often difficult for the novice to decide how much of this is pertinent for a given application. In most situations, he or she can't afford—or won't need—to address all of the issues we've discussed. We'll end our studies, therefore, with several chapters that show how to put it all together; how to focus on the factors of importance for a given application; how to identify those which can be ignored; and how to estimate the combined impact of the chosen factors in order to make better design or assembly decisions.

This first chapter will help you decide how much preload you should use in an existing joint, or "what torque?" as the question is usually stated. To answer this question we must briefly review many of the topics we've covered. Hopefully this summary will help you decide which of the many factors we've looked at will affect your results. And it will give you a reasonable way, I think, to make acceptable assembly decisions without having all the hard and fast data we'd like to have. Such data cost money—often a lot of money—and are rarely available to those who must deal with existing joints. In fact, in many (most?) situations you won't even need most of the procedures to be described in this chapter. You'll be able to use the simple ways to select preload described in Section 17.2. If you've had problems with a joint, however, or have failure or economic concerns, then you'll need something practical but a little more elaborate; you'll need the procedures that make up the bulk of this chapter. For safety-related joints you may want to start with these procedures, but use the information in previous chapters and the many references to go well beyond them. So—here's how we would start our search for a "better torque" if we had had problems or wanted to do a better job with an existing joint.

In picking preload or torque for such a joint we should answer two questions: "How much initial, assembly clamping force do we want in this joint, considering the service loads and conditions the joint will face?" and "How much clamping force—and scatter in clamping force—can we expect from the assembly torques, tools, procedures, etc. we plan to use?" Let's see how we might answer these questions in an economically acceptable way.

17.1 HOW MUCH CLAMPING FORCE DO WE WANT?

17.1.1 Factors to Consider

We start by considering the in-service clamping force needs of the joint. What loads and service conditions must the clamping force between joint members resist? Note carefully that this is the in-service clamping force we're talking about, not just that theoretically created by the initial assembly preloads. This initial clamp must be high enough to compensate for all of the mechanisms which may reduce the clamping force to the in-service level, including:

Embedment relaxation
Elastic interactions
Creep of metal parts, gaskets, etc.
External tensile loads
Hole interference
Resistance of joint members to being pulled together
Prevailing torque
Differential thermal expansion

We must start by deciding how much clamp the joint will need in service, then try to estimate how much each of the above factors will have robbed from the initial clamp. Only then can we decide how much initial clamp—and therefore initial, assembly preload—we need. OK—here's a check list of the main factors, which the in-service clamping force might have to resist.

17.1.1.1 Joint Slip

This is a key one because slip can cause a lot of problems. It can cause unfortunate stress concentrations in a slip-distorted structure. It can cause fretting corrosion or fatigue of joint members. It can cause self-loosening, or misalign and cramp bearings, etc. It's relatively easy to calculate the clamping force to reduce slip, however, if we know the magnitude of the shear loads imposed on the joint.

Let's assume that a shear load of L_X is to be imposed on this joint. To avoid slip the frictional forces created by the clamping force must exceed the external load as follows:

$$F \geq L_X \mu \tag{17.1}$$

where
F = clamping force on the joint (lb, N)
L_X = external load (lb, N)
μ = coefficient of friction (typically 0.15–0.30)

17.1.1.2 Self-Loosening

We learned in Chapter 14 that self-loosening will occur when transverse loads cause slip between joint members and thread surfaces. Although it is often difficult to quantify the vibratory or other forces creating self-loosening, the joint slip equation above could be used to estimate the point at which the joint members will slip if the external loads are known.

If external loads cannot be estimated and the application is likely to involve vibration or other forms of self-loosening, then a good practice would be to plan for the maximum clamping force the parts can stand. The more the better, up to, but probably not exceeding, yield.

17.1.1.3 Pressure Loads

The influence of pressure loads on bolted joints is complex and is discussed in detail in Volume 2. One way to quantify the required initial and in-service clamping forces is to use the equations of the boiler and pressure vessel code. The equations found in the Code at present are

$$W_{M1} = \frac{\pi G^2}{4} P + 2\pi G b m P \quad (17.2)$$

$$W_{M2} = \pi b G y \quad (17.3)$$

where
W_{M1} = tension in the bolts in service (lb, N)
W_{M2} = initial tension in the bolts at assembly (lb, N)
G = diameter of gasket (in., mm)
P = contained pressure (psi, Pa)
b = effective width of the gasket (in., mm)
m = gasket maintenance factor
y = initial gasket stress at assembly

These equations assume that the entire pressure load will be seen by the bolts (i.e., ignoring the implications of the joint diagram), but these equations or their equivalent could presumably be used for an approximate (and conservative) estimate of the clamping force required in most gasketed joint situations.

17.1.1.4 Joint Separation

In some applications, a noncritical foundation bolt is an example; gravity holds the joint in place and it is sufficient for the bolts merely to maintain alignment. In this situation, clamping force could be as low as zero without risk of joint failure. Even here, however, some clamp would be useful merely to retain the nuts. This little clamp would also be acceptable in those structural steel joints, which are not slip critical. In most tension joints, however, it's as important to avoid separation as it is to avoid joint slip—maybe even more important. Separation can lead to such horrors as gross leakage and low-cycle fatigue of bolts. To avoid separation we must be sure that the initial preloads are high enough to compensate for all of the clamp loss factors listed a minute ago and still leave some residual in-service clamp. A margin of safety, if you will.

17.1.1.5 Fatigue

Although joint separation can reduce the fatigue life of bolts by a substantial amount it is not, as some handbooks imply, the only cause of fatigue failure. Too little clamping force or, less commonly, too high a mean bolt tension can also cause problems. This time, however, we have no simple equation to compute the amount of in-service bolt tension or the related interface clamping force required. A fairly complex analysis is required, as described in Chapter 15 and in the references cited at the end of that chapter.

17.1.2 Placing an Upper Limit on the Clamping Force

When determining the amount of clamping force required to combat separation, self-loosening, slip, or a leak, we are interested in establishing the essential minimum of force. In each of those situations, additional clamping force is usually desirable (for added safety) or is at least acceptable. You might remember that another early theme we addressed was "We always want the maximum clamping force the parts can stand." There is, however, always some upper limit on that clamping force. If that weren't the case, we could simply tighten them a lot more to avoid failures. Instead, we must define an upper limit for our application.

In fact, this is one of two key issues we first addressed in Chapters 10 and 11. Remember our attempts to identify the maximum tension which will be seen by some of the bolts and the minimum clamping force we can expect in some joints. Well, in Section 17.1.1 we discussed the minimum clamping force requirements for a joint; now we're about to address the equally important issue of maximum tension or stress in the bolts because this will usually—though not always—be the thing which places an upper limit on the amount of clamp we want. We might like more, but not if it means broken or threatened bolts. So, here are some bolt factors that limit clamping force.

17.1.2.1 Yield Strength of the Bolt

There is a good deal of debate about this in the bolting world at the present time, but most people feel that it is unwise to tighten bolts past yield in most applications. There are many exceptions, with structural steel being the most obvious. As we saw in Chapter 8, torque–turn equipment, which tightens the fastener past yield, or to yield, is popular in automotive and similar applications. In general, however, we usually won't want to lighten them past yield during initial assembly. Bolt yield, then, is one easy-to-estimate, "worst-case," upper limit on the in-service clamp force.

17.1.2.2 Thread-Stripping Strength

Obviously, we will never want to tighten the fasteners past the point at which their threads will strip. This then provides another, simplistic, worst-case upper limit. If we didn't consider this limit when selecting the bolts, we should do so now (unless one of the limiting factors listed below will obviously dominate our decision).

17.1.2.3 Design-Allowable Bolt Stress and Assembly Stress Limits

We always want to identify any limits placed on bolt stress by codes, company policies, standard practices, personal bias, or the like. Both structural steel and pressure vessel codes, for example, define maximum design allowable stresses for bolts. It's necessary, however, to distinguish between a maximum design stress and the maximum stress which may be allowed in the fastener during assembly. This will differ from the maximum design allowables if a design safety factor is involved. In the structural steel world, for example, bolts are frequently tightened well past yield, even though design allowables are only 35%–58% of yield. Pressure vessel bolts are commonly tightened to twice the design allowable, as explained in Chapter 19. Aerospace, auto, and other industries may also impose more stringent limits on design stresses than on actual stresses to force the designer to use more or larger bolts than he might otherwise select (and, therefore, to introduce safety factors in the design).

17.1.2.4 Torsional Stress Factor

If the bolts are to be tightened by turning the nut or the head, then they will experience some torsional stress as well as tensile stress during assembly. If tightened to yield, they will yield under a combination of tensile and torsional stress. If we plan to tighten them to or near yield, it's pertinent to reduce the maximum tensile stresses allowed at assembly by a torquing factor, which makes room for the torsional stress. If as-received steel-on-steel bolts are used, then a reduction in the allowable tensile stress of 10% is probably reasonable. If the fasteners are to be lubricated, you might use 5%. The true amount of strength absorbed by torsion will be determined by all of the variables, which affect the torque–preload relationships, so the torsional effect on the tensile capacity of the bolts may be much greater than 5% or 10% (and is as difficult to predict as the specific torque–tension relationship for a given set of parts).

Anyway, 5%–10% will probably be acceptable in most applications. Most people don't tighten bolts to the yield point anyway.

17.1.2.5 Shear Stress Allowance

If the bolts will also be exposed to shear stress we must take that into account in defining maximum assembly preloads and the resulting in-service clamping force, since shear stress will reduce the amount of bolt strength capacity available for the tensile stress.

The shear reduction is explained and illustrated in Chapter 12 (Equation 12.1).

17.1.2.6 Stress Cracking

As we learned in Chapter 16, stress cracking is encouraged by excessive tension in the bolts and we are told to be alert if service loads exceed 50% of yield, at least for low-alloy quenched and tempered steels. See the many bolt stress versus yield strength curves in Chapter 16 for a more complete definition of this important upper limit.

17.1.2.7 Combined Loads

As we try to determine the maximum stress the parts can stand, to avoid damage to the parts, stress corrosion cracking (SCC), or other problems, we must evaluate our estimates in light of the anticipated service loads on the bolts. Most of these service loads would add to the bolt stresses introduced during assembly (the preloads) even as they reduce interface clamping force.

The joint diagrams of Chapters 10 and 11 can be used to add external loads to preloads. The effects of a temperature change can be estimated using the procedure described in Chapter 11. The bolts must be able to withstand worst-case combinations of these preloads and service loads.

All of the limiting factors discussed so far deal with bolt strength, which, in turn, limits the clamping force available for the joints the bolts are used in. There are also a few joint factors that can limit clamping force. Here are some of the most common ones.

17.1.2.8 Damage to Joint Members

Too much tension in a bolt can cause its head and nut to embed themselves into the surfaces of the joint, not just by a normal amount of a few mils, but enough to cause visible damage to the joint surfaces. The VDI Directive 2230, for example, says that the "boundary surface pressure" (P_G) of joint materials is usually slightly greater than the yield strength of the material, and they recommend that bolt tension not exceed the value suggested by this equation.

$$P_G A_P \geq 0.9 \text{ Max } F_B \tag{17.4}$$

where

$A_P =$ contact area (e.g., between nut face and joint) (in.^{2}, mm^2)
$F_B =$ tension in the bolt (lb, N)
$P_G =$ boundary surface pressure (psi, N/mm^2)

The 0.9 is added as a safety factor.

17.1.2.9 Distortion of Joint Members

Joint members can sometimes be distorted by excessive bolt loads. For example, the outer ends of raised face flanges can be pulled toward each other—perhaps bent—by too much preload. This can unload the ID of a gasket, opening up a leak path, for example.

17.1.2.10 Gasket Crush

Excessive preload can so compress a gasket that it will not be able to recover when internal pressure or a thermal cycle partially unloads it. Contact the gasket manufacturer for upper limits. Note that these will be a function of service temperatures.

17.1.3 Summarizing Clamping Force Limits

Let's try to summarize the range of clamping force we want. Not too high—that might cause joint problems or, more commonly, bolt problems. Not too low—that might cause joint slip or separation or leakage.

Figure 17.1 illustrates one possible scenario based on some of the factors we've just discussed. Each factor is described as a percentage of the ultimate strength of the bolt, to give us a common, vertical axis. The figure shows that 100% of ultimate would break the bolts. It suggests that bolt tension above about 80% of ultimate would take the bolts past yield, and assumes that we don't want to do that in our application: this region is considered "unusable."

The chart suggests that we subtract about 5% from the tensile yield strength of the bolts to account for torsional stress introduced when torque is applied. It suggests that we might have reduced the maximum allowable bolt stress still further to accommodate a code or other

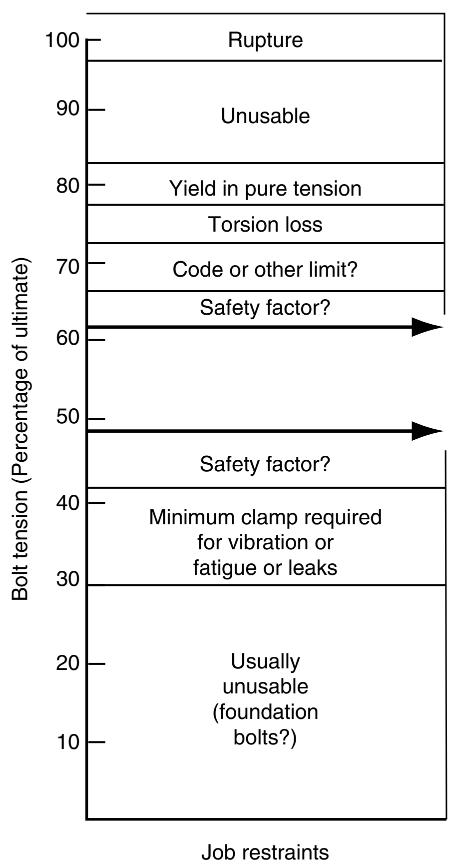


FIGURE 17.1 Chart summarizing the clamp load decisions made for a hypothetical joint. See text for a detailed discussion.

specification limit. And it further reduces the limit by a 5% safety factor to acknowledge that our data are limited or that we haven't included such factors as external loads or differential expansion, which can add to bolt tension. The final result is a suggested upper limit on desired clamp force equal to 62% of the ultimate strength of the bolts.

The chart also defines an acceptable lower limit on clamping force; suggesting that we want a clamp equivalent to at least 30% of ultimate to make sure the nuts don't fall off and that, more seriously, we want a clamp equivalent to as much as 42% of ultimate to prevent that failure mode (or modes) we're most concerned about, whether that be self-loosening, leakage, slip, fatigue, or combination of these. Since the exact amount of load or vibration to be seen by a joint is often hard to calculate, and since more is generally better, we throw in a larger safety factor on the low end than we did at the high. The result: a lower limit on desired clamp which is equivalent to 48% of the bolt's ultimate strength (or about 60% of yield if the bolt yields at 80% of ultimate).

All of which is very broad-brush and seat-of-the-pants, but that's the way most assembly preload or torque decisions are made in the field. The principal advantage of the procedure is that it forces us to consider separately, and assign at least gut-feel values to, those factors we think might affect our joint. The chart shown in Figure 17.1, incidentally, can also be constructed in terms of yield strength or torque, for single bolts or the whole group.

OK—we have now roughly defined the range of clamping force we'd like to see in our joint, a range defined in part by limitations in the strength of the bolts. Now we want to pick a target preload to specify for assembly purposes. If dealing with a problem joint we'll also want to estimate the range of clamping force we can expect to achieve during assembly, taking our choice of preload plus our assembly tools and procedures into account. It will often be necessary, too, to estimate how much that range might be increased by postassembly and in-service conditions. With all this information in hand we can then compare the desired range of clamping force to the anticipated range. If they agree, we're done. If they don't agree we'll have to take steps to correct our assembly procedures—perhaps pick a better torque—to make them agree.

Before proceeding, let's acknowledge, once again, that it usually won't be necessary for us to go through the full procedure I'll describe in a minute. If service conditions are reasonable and we have no safety concerns, we can choose a preload or torque by some of the very simple ways described next. We'll want to go beyond these things only if we're concerned about results, or—more commonly—have had problems with this joint in the past.

17.2 SIMPLE WAYS TO SELECT ASSEMBLY PRELOADS

In most situations, as the bulk of prior experience tells us, we can pretend we never read this book, and can pick a preload by one of several simple, time-proven methods that have worked on most joints in the past. And we don't want to forget this point because, if we do, we can waste time and money solving problems which don't exist. So, before considering a procedure to use when it is necessary to "do it right," let's look quickly at some of the simpler ways that will often be sufficient.

17.2.1 Best Guide: Past Experience

If you've had previous experience assembling this joint, and the results have been acceptable, don't change your procedures or tools. As we've seen, bolting involves more variables than we can cope with. We can never predict results with perfect accuracy; so prior experience, if satisfactory, is better than all of our theories. Leave it alone! Note that the tools and procedures you've used have, by default, "selected" a preload range for you. It worked. It's acceptable.

Note that you won't usually be able to identify a successful preload per se in situations such as those we're discussing. You'll know what torque the mechanics have used, or which size of tool, or which procedure.

Note that even if we know exactly which torque has been used on this joint we don't know the preload created by that torque. A given torque—any torque—will create a range of preload in a group of bolts. Variations in friction can scatter the preload by as much as $\pm 30\%$. Elastic interactions can leave residual preloads scattered by maximum/minimum ratios of 20:1 or more. Embedment, tool error, operator problems, etc. can also enter the picture as we have seen. In spite of all this, in spite of the fact that we haven't measured, or even been aware of, such factors, if the tools, procedures, torques, etc. we've used in the past have worked—we should keep using them.

17.2.2 Second Best: Ask the Designer

If you've had no prior experience with this joint, or have had some problems, your best source of information is the man who designed the joint. As we've seen, many factors must be considered in the design of a successful joint. It's a rare customer or user or assembler who knows enough about the materials and configurations of—or loads on—a joint to duplicate the designer's expectations when it comes to specifying the correct clamp force. If the selection of preload matters, the designer should know what the desired range of values should be. Presumably he will have gone through a design procedure similar to that discussed in Chapter 18.

There will be many cases, however, in which the designer has done no such thing—but even here he may have been guided by the past experience of other customers, or by general experience with this type of product. Even if he hasn't made an analysis, therefore, he may be able to recommend a preload—again, specified as a torque in most situations. Note that the designer's recommendations can sometimes be found in the operating or maintenance manuals which came with the equipment. If not—give the designer a call!

17.2.3 Unimportant Joint: No Prior Experience

In another common situation, you'll have no prior experience with this joint, and no way to reach the designer (or he can't help) and so must pick the preload (probably a torque) yourself. If the joint is a common one, and you have no real concern about the consequences of failure, then the normal procedure is to pick a torque from a suitable table. You might use that in Appendix F of this book, for example. Picking a torque (and a tool and a procedure) determines the range of preload you'll achieve at assembly, even if you won't be able to quantify this range with any precision. Most joints are overdesigned; most joints will behave themselves when assembled with a torque selected this way.

Again, picking a torque will, in effect, pick a range of preload, as suggested in Section 17.1.1.

17.2.4 When More Care Is Indicated

If your past experience with the joint could be better and you're a little concerned about the consequences of failure, you might want to compute an appropriate torque instead of picking one from a table. One way to do this:

Determine the yield strength (S_y) of the bolt material at the operating temperature of the joint (data on yield strengths will be found in Chapter 2).

Pick a target percentage of yield (P) from Table 17.1. Determine the tensile stress area of the bolt (A_s) (see Appendix F or H). Estimate the lubricity of the fastener by picking a nut

TABLE 17.1
Typical Target Preloads as a Percentage of the Yield Strength of the Bolts

Percentage of Yield	Applications
25	Unimportant non-gasketed joints exposed to static loads, foundation and anchor bolts under static load, also joints where there have been serious stress corrosion cracking problems
40	Gasketed joints in routine service, including those covered by the ASME Code, which have not given problems
50–60	Average non-gasketed joint, with normal safety or performance concerns, where past experience does not suggest higher or lower preloads; a good place to start a search for the optimum preload when some trial-and-error is acceptable
	Probably the maximum acceptable preload for gasketed joints designed to ASMK Code rules (although there will be a few exceptions)
70–75	Upper limit for non-gasketed joints with which you've had low preload problems in the past (leaks, self-loosening, fatigue, etc.) and where torque control will be used at assembly
85–95	Joints that have had consistent low preload problems in the past, where the need to avoid failure dictates the use of special techniques such as stretch or ultrasonic control, and where service loads (or ignorance of service loads) make it unwise to take the fasteners any closer to the yield point
100	Structural steel bolts tightened by turn-of-nut procedures; also high-performance or problem joints facing self-loosening, fatigue, or other low preload problems, but where service loads can be predicted with sufficient accuracy to guarantee that the fasteners won't be ratcheted to rupture later on; sometimes used in gasketed joints in automotive applications, for example, requires special assembly techniques

factor (K) from Table 7.1 in Chapter 7. Now use the following version of the short-form, torque–preload equation to compute an assembly torque (T) in in.-lb.

$$T = KDPS_yA_s \tag{17.5}$$

D is the nominal diameter of the bolt in inches.

Why is a computed torque better than a torque taken from a table? The computation takes specific job constraints into consideration; a table is based on normal assumptions which may or may not be valid for your application. In the procedure above, for example, we made application-specific decisions based on:

- Percentage of yield strength as a preload (not torque) target
- Lubricant we're planning to use
- Operating temperature of the joint

In addition, or instead, you could base the desired preload on such things as the fatigue endurance limit of the bolts or on the limitations suggested by SCC concerns instead of basing it on a percentage of yield. But the end result at this level of the preload selection process is still a torque, with all the subsequent uncertainties of the torque–tension relationship. Neither this process nor a torque table should be used on critical joints. As one more confirmation of the uncertainties involved here, Hayman and Brown report on a series of tests made to determine the nut factor (K) to be used for five different lubricants. They started with values found in Novak and Patel [3], then made tests to confirm or refute these data. Fasteners used in their experiments were defined as "a new ⅝ in., new ¾ in., old ¾ in., and new ⅞ in." Results were well scattered and I don't have all of their data. The nut factor for all bolts coated with a thick layer of a nickel-based antiseize compound all fell within the maximum/minimum range reported in Novak and Patel [3], but the maximum K of that range is essentially double the minimum K. This would mean that twice as much torque would be required to create a given preload in one bolt as in others. Furthermore, the scatter in results with thinly lubricated bolts was greater than the scatter found with unlubricated bolts. A thick coat of lubricant reduced the scatter to a narrower range than that found on the unlubricated bolts, but this was only for new bolts. The nut factor for thickly lubricated old studs varied more than it did for unlubricated new studs. So beware: all we have here, after using Equation 17.5, is a specified torque and with the unspecified but unavoidable (and perhaps large) range of the preload it will create.

17.2.5 If Improvements Are Required

If you've used a torque based on past experience, or picked from a table, or computed with the short-form equation, and have had nuisance problems, and can't get any help from the people who designed the equipment, you might want to try a different torque. Eureka! Seriously, trying a different torque is valid. Bolting is an empirical art at present; experiments are often the most cost-effective way to make improvements. They may, in fact, be your only choice. (Again, you shouldn't use this or any other simple preload selection technique if the joint is critically important and failure would be disastrous.)

What torque should you try? If the problems you've experienced included a leak, self-loosening, joint slip, joint separation under load or fatigue, more preload is indicated. Try increasing it (e.g., try increasing assembly torques) by 10% or so.

If the problem has involved stress corrosion, stripped threads, crushed gaskets, or excessive flange rotation, try 10% less preload or torque the next time.

In either case, keep good records so that you can benefit fully from your experiments and ultimately find the optimum preload, torque, procedure, etc.

Note that picking a better torque probably won't reduce the range of preload created in the bolts; it will merely move the average or mean preload to a higher or lower level. But that will frequently solve a problem.

17.2.6 Selecting Preload for Critical Joints

The procedures described above can be used in most applications. Obviously they can't be applied—except perhaps as a first step—to critical or safety-related joints. A good design engineer must then be called upon to select a preload and an assembly procedure (to control the range). Either he—or a well-trained bolting engineer—might be able to use the more rigorous procedures described in Section 17.4, hopefully with the help of well-founded data on the factors, which can affect results in a given application.

Let's assume that we've used the quick-and-dirty procedures described above to pick a target preload, which we'll use to specify an assembly torque. We know that this preload will result in a range of residual preload, and that range will be further modified by service conditions to give us a still wider in-service range. We're very interested in estimating the size and limits of this range, because that range in preload will determine whether or not we have created at least the minimum clamping force we decided we needed in Section 17.1. Let's see how we might estimate the in-service range of clamp force.

17.3 ESTIMATING THE IN-SERVICE CLAMPING FORCE

Placing exact limits on the in-service clamping force will usually be impossible. There are too many variables, and we'll rarely have good data with which to define their exact effect on the overall results. We can, however, often use the procedure I'm about to describe, plus general knowledge of typical values, to give us some idea as to the possible range. The answers we get will be good enough for most applications. The procedure we'll follow is this.

We'll list the variables, or groups of variables, which we think will affect the preload achieved at assembly and the stability of that preload in service. Most, if not all of the possible assembly problem groups are listed in Figure 6.28. It would also be useful to consult the chapter on the assembly technique (torque or turn, etc.) you're planning to use. Or you might identify possible variables by looking at the table of contents or the index to this book. In most situations, incidentally, you'll be primarily concerned with a half-dozen or so key factors. You don't have to consider every little thing which could affect the outcome.

Next we'll estimate the contribution which each key variable, or group, might make to the preload scatter experienced during assembly, or might contribute to the subsequent instability. We'll deal with the key variables one at a time, or one group at a time, as we make these estimates.

Then we'll use the chart shown in Figure 17.2 to combine these estimates and so arrive at final answers to our questions.

The procedure is simple and is easy to use, in part because it's heavily based on assumptions, estimates, and typical values. In most situations this is all we'll have. Accurate data will be rare, and only in the most critical situations will we be able to afford the time and money to

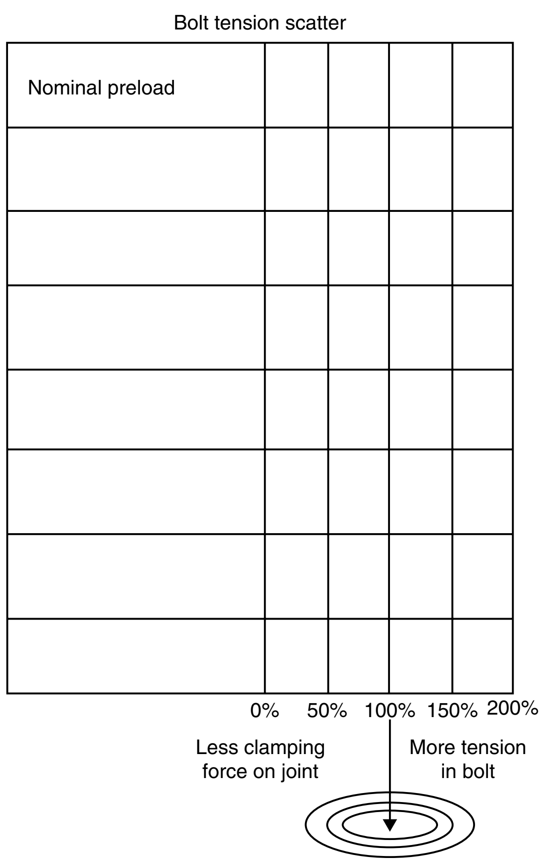


FIGURE 17.2 Chart used to summarize the results of scatter in the major variables, which affect the amount of bolt tension and interface clamping force expected during assembly and when the joint is put into service.

make the experiments or analyses required to obtain hard data. Nevertheless, the proposed process will take us well beyond the usual procedure for estimating in-service bolt loads or clamping force on the joint.

What is the usual procedure? Most people seem to base their estimates of assembly accuracy—or in-service preload scatter—entirely on the process used to tighten individual bolts. If torque control is used at assembly, for example, it's usually assumed that final assembly preloads and in-service bolt tensions will be scattered $\pm 30\%$ around a mean or target value of tension. But the group of variables which create the uncertainty in the torque–tension relationship for single bolts is only one of the many groups of variables which can affect assembly or in-service results. Elastic interactions, relaxation effects, thermal effects, and many other things can also play a significant role. The procedure I'm about to describe allows us to consider all or most of the factors that will make a significant difference in a given application.

As mentioned, Figure 17.2 is the worksheet that we'll use to estimate the combined effect of the factors we think will affect the outcome in a particular application. Let's work an example to see how it's used.

17.3.1 Basic Assumptions

Let's assume that we'll use a torque wrench to tighten the bolts on a straightforward, non-gasketed joint. We'll also assume that we've used the procedure described in Section 17.1 to define an acceptable range or preload (48%–62% of the bolt's ultimate tensile strength (UTS) as in Figure 17.1). Now, after reviewing Figure 6.28 and Chapters 6 and 7 (on torque-controlled assembly), let's further assume that we've decided that five factors, each of which involves a different group of variables, will determine assembly results and subsequent changes of bolt tension in service. At the end of the example we'll briefly review some of the other factors we might have considered.

We've picked the five factors listed below. We could have picked more—or less. There's nothing magic about five. The following list, however, includes several different types of problem or variables; some assembly, some in-service; some introducing plus or minus scatter, others biased in only one direction; some uncertain, others inescapable; etc. So this is a good list with which to study the procedure.

Tool accuracy
Operator accuracy
Control accuracy
Short-term relaxation
External loads

17.3.2 Combining the Scatter Effects

Now let's use the worksheet to combine these assembly and service factors. Then we'll be able to compare anticipated results with the desired results established in Section 17.1.

We start with the concept that there is an ideal in-service tension for this joint. If we could always introduce that tension at assembly, and if subsequent load and environmental factors never altered it, we'd have achieved perfection. We would have "hit the bull's eye" 100% of the time. We represent this ideal tension by a vertical line drawn from the top of our worksheet to the center of the target at the bottom. As a first cut, we'll assume that this tension is at the midpoint of the 48%–62% range established earlier—i.e., our target, in-service bolt tension corresponds to 55% of the UTS of the bolt. This ideal bolt tension, of course, will create an ideal clamping force between joint members.

Now we introduce our first factor of concern—the accuracy of the tools to be used at assembly. Note that anything which reduces the preload or service tension in the bolts also reduces the all-important clamping force on the joint, making it less than our ideal or target value. Anything which increases the preload or tension means more tensile stress in the bolt than is ideal. Let's see what our first variable, tool accuracy, does to us.

By tool accuracy I mean the accuracy with which the tool produces the thing it's supposed to produce. In this example, the tool produces torque and we'll assume that it does this with an accuracy of $\pm 5\%$. (We'll consider the accuracy with which that torque creates preload in a minute.) We'll assume that this 5% (and the other percentages to follow) are ''three sigma'' values, and represent the worst-case errors we can expect to encounter in nearly all such assemblies. The worksheet could, instead, be based on one standard deviation (68% of the population), or on two sigma values (95%), to be less conservative. But we'll assume three sigma (99%) for now.

In the real world, would we really consider tool accuracy to be a key variable? Sure. It's not uncommon to believe (sometimes correctly) that a more accurate tool can make a significant difference. For our example, let's pretend that a torque wrench salesman has heard that we've had vibration loosening problems, and he says that our problems will be solved if we use his 5% wrench instead of the 15% multiplier arrangement we've been using. We can evaluate his claims as we make our analysis, and can decide if a better, more accurate, torque tool would be likely, or unlikely, to reduce our problems.

We said that we expected the torque accuracy of this tool to be $\pm 5\%$. Since torque is linearly related to preload (see the short-form equation in Chapter 7), this means that tool inaccuracies, worst case, will introduce as much as 5% more tension in some of our bolts than we'd like to see, or create, in other assemblies, as much as 5% less clamping force in the joints. These are both undesirable results, and therefore are of concern to us. We don't want to break the bolts (too much tension) and we don't want the clamping force to be too low (remember that we always want the maximum clamping force those parts can stand). So, as we plot tool accuracy on our worksheet, as in Figure 17.3, we label plus errors as ''more tension in the bolt'' and negative errors as ''less clamping force on the joint.'' The accumulated three sigma scatter at this point is, of course, just $\pm 5\%$.

Now we introduce the second factor of concern: operator accuracy. (By ''operators'' I mean the mechanics or assemblies or others who operate the tools, use the wrenches, and tighten the bolts.) This factor defines the amount of scatter introduced by operator errors, operator carelessness, poor accessibility (which makes it difficult for the operator to do a perfect or ideal job), etc. Let's assume that this factor will contribute, alone, $\pm 10\%$ of scatter.

Do operators contribute this much uncertainty to bolted joint results? They certainly do. Some people claim, in fact, that all in-service problems result from improper assembly or maintenance practices: the result of operators who don't care or lack skill. We'll assume that we're dealing with well-trained mechanics who know that bolting is important. They'll contribute $\pm 10\%$ to the scatter primarily because accessibility is poor and working conditions are difficult. Even with a perfectly accurate torque wrench they won't always be able to apply the ideal or target torque to each nut.

Note that this contribution to the scatter will be especially difficult to quantify. Our procedure, however, makes our guess more realistic by isolating it from the other variables we must deal with. Anyway, let's assume $\pm 10\%$ for the example.

We don't just add this $\pm 10\%$ to the tool's $\pm 5\%$. Probability theory tells us that it's unlikely that we'll experience a worst-case tool error and a worst-case operator error in the same assembly. We may see that once in a while, but in 99% of the assemblies (three sigma again) the combined effect of these two errors will be less. We can compute the probable

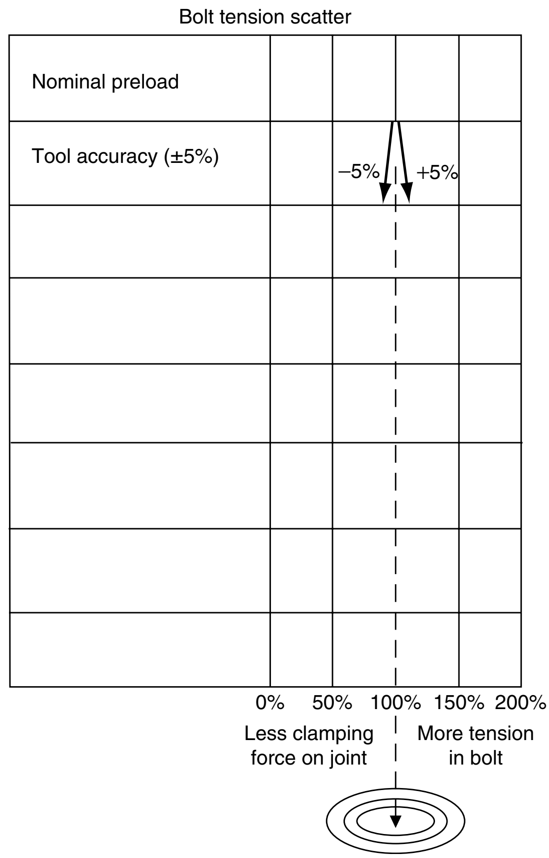


FIGURE 17.3 The first variable, tool accuracy, is assumed to contribute ±5% of scatter to the bolt tension and clamping force.

combined error by taking the square root of the sum of the squares of the two error percentages, as follows:

$$\pm V_{\mathrm{T}} = \pm[V_{\mathrm{TL}}^2 + V_{\mathrm{OP}}^2]^{1/2} \tag{17.6}$$

where

V_{T} = the total three sigma scatter
V_{TL} = the three sigma scatter contributed by the tool
V_{OP} = the three sigma scatter contributed by the operator

So the combined effect of these two errors is only ±11%, as shown in Figure 17.4.

Now we add the third factor, control accuracy. This is the accuracy with which the selected control variable—torque in this example—produces the thing we're interested in—which is always bolt tension. Table 17.2, incidentally, lists typical control accuracies for a variety of assembly tools and procedures. In our example, we'll use the conventional wisdom and say that the torque–tension scatter will be ±30%. We could introduce elastic interactions as an additional variable affecting the bolt tension achieved at assembly, but won't for this first example. We'll assume a fairly rigid joint with metal-to-metal contact and insignificant interactions.

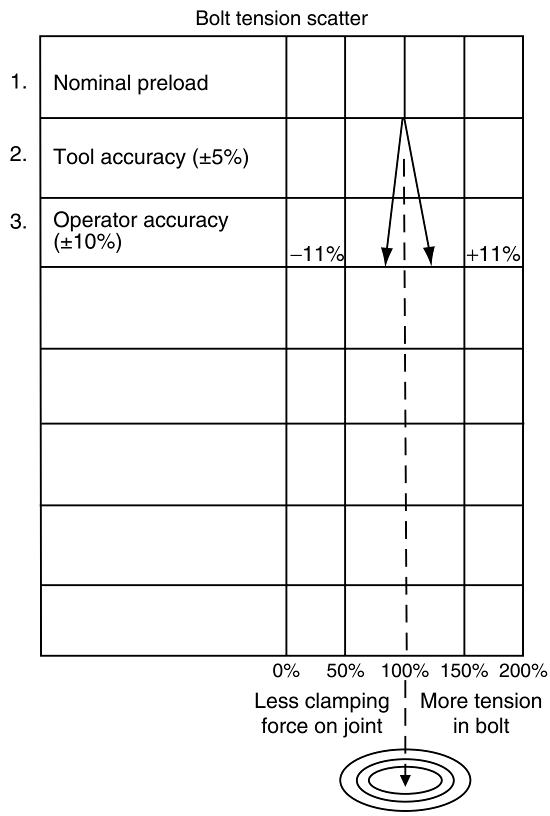


FIGURE 17.4 The second variable studied in the example given in the text is operator accuracy, and is assumed to contribute $\pm 10\%$ of scatter. The accumulated scatter at this point is $\pm 11\%$ (see text for that calculation).

TABLE 17.2
Preload Scatter Reported for a Variety of Bolting Tools or Procedures

Torque control with hand wrench	± 30
Stall torque air tool	± 35
Click type torque wrench	± 60 to 80
Torque wrench plus multiplier	-70 to $+150$
Turn-of-nut (structural steel)	± 15
Computer-controlled air tool to yield point	± 3 to 10
Rockwell International's LRM (torque–angle) system	± 3 to 10
Strain-gaged load washers	± 15
Strain-gaged bolts	± 1
Swaged lockbolts	± 5
Air-powered impact wrench	-100 to $+150$
Hydraulic tensioners with vernier gage readout	± 20
Operator feel	± 35
Bolt stretch (μm)	± 3 to 15
Ultrasonic control	± 1 to 10

Source: All values come from the author's own experiences; Shigley, J.R. and Mischke, C.R. (Eds.), in *Standard Handbook of Machine Design*, McGraw-Hill, New York, 1986, 23.23.

Note: All values are in percentage.

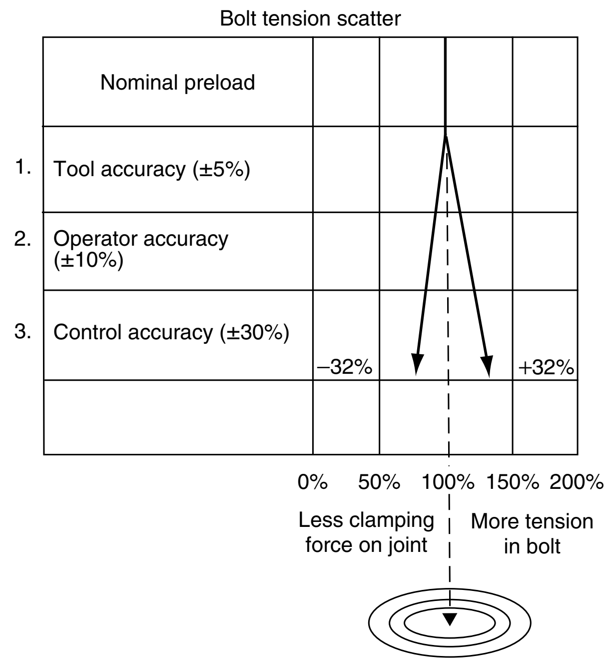


FIGURE 17.5 Control accuracy (torque versus tension) scatter of $\pm30\%$ raised the accumulated scatter to $\pm32\%$.

We take the square root of the sum of the squares again, taking three variables into account this time, and now find a cumulative scatter of $\pm32\%$ (Figure 17.5).

Our next variable, short-term relaxation, is handled the same way; but now we have to compute positive and negative errors since they're different. Let's assume embedment relaxation of -10%. Relaxation will never increase bolt tension, so the plus scatter is zero. Cumulative errors now stand at, worst case, +32% and -34%. This tells us that, over a long period and many assembly operations (i.e., a large statistical sample), we could expect to see, worst case, 32% more tension in some bolts than the ideal hoped for, and that, in other cases (not simultaneously!), we'd see 34% less than the ideal clamping force in some of the joints (Figure 17.6).

This summarizes the situation at the end of assembly. Note that the $\pm30\%$ torque–tension variable has dominated the results so far (justifying those who often assume that it's the only variable!). It wouldn't have dominated results if we had included elastic interactions in our analysis, or had assigned a large scatter to the operators.

The final variable we've selected involves an in-service condition. We'll handle it in the same way, however. When we put the joint into service it sees an external tensile load. This load will both increase the tension in the bolts and decrease the clamping force on the joint (remember the joint diagrams in Chapters 12 and 13). If we assume a 5:1 joint-to-bolt stiffness ratio and an external tensile load equal to 25% of preload, we might see a change of $+7\%$ in bolt tension and -20% in clamping force on the joint when the load is applied.

This factor differs from the others we've considered in several ways. First, it simultaneously affects both bolt tension and clamping force. Most of the previous variables would affect both, but not in the same assembly. Second, the external load may be relatively easy to quantify, compared at least to such factors as operator accuracy or torque–tension scatter. Third, this effect will probably be unavoidable. Tools and operators may perform perfectly,

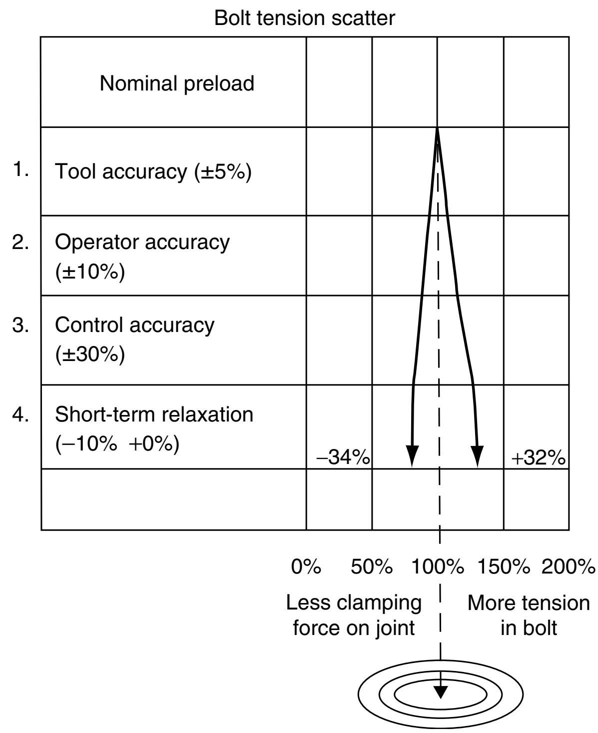


FIGURE 17.6 Short-term relaxation can decrease the clamping force but not increase bolt tension. The accumulated scatter is now +32% and −34%.

in some assemblies at least, but all assemblies have been designed to carry an external load. When we combine this certain variation with the earlier "probabilistic" ones, therefore, we're adding apples to oranges and will undoubtedly distress our neighborhood expert on probability. But I think the procedure is useful.

We're trying to estimate—roughly, simply, inexpensively—the combined effect of many variables. Most of the data we're using are soft, to say the least. This latest violation of probability mathematics (if that's what it is) won't, in my opinion, make things much worse.

Adding the first service factor of concern, the effect of tensile loads, doesn't change the picture much, thanks to the smoothing effect of the square root of the sum of squares equation. The accumulative worst-case errors now stand at +34% and −39%, as in Figure 17.7.

Let's pause at this point and see what those errors suggest. Plus 34% means that some of our bolts will experience 134% of ideal or optimum tension because of the combined assembly and service factors considered so far. Minus 39% tells us that, worst case, we can only count on 61% (100%−39%) of the desired clamping force in some of those joints. The ratio, therefore, between the tension the bolts must be able to support without breaking and the clamping force we can count on in 99% (three sigma) of our assemblies is 134%/61%; in other words, the ratio is greater than 2:1.

Let's assume that past experience has suggested to the designer that a 1⅛ in. diameter bolt will probably be required for this joint. This means, presumably, that a 1⅛ in. bolt can withstand the maximum 134% of ideal stress experienced by bolts in this application. But we can only count on 61% of the ideal clamping force. We could get that much clamp from a perfectly tightened ¾ in. bolt.

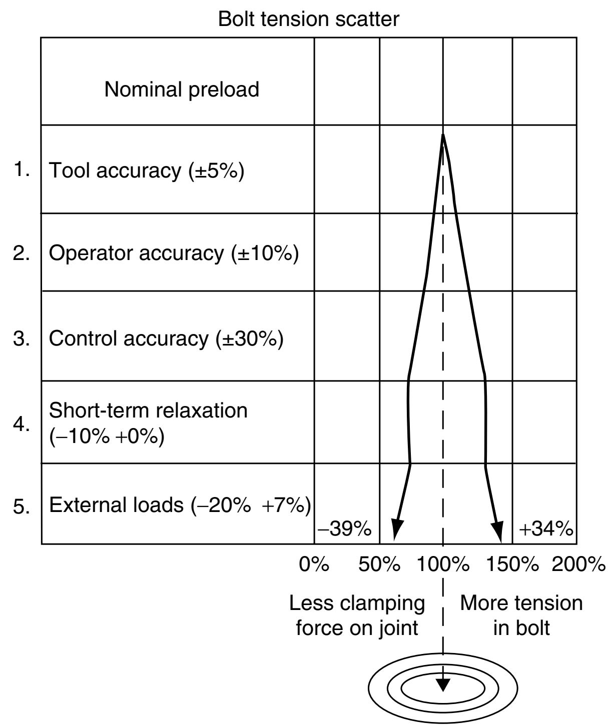


FIGURE 17.7 An external (tensile) load will simultaneously increase bolt tension by 7% and reduce clamping force by 20% if the bolt/joint stiffness ratio is 5:1 and the external load is 25% of preload. Note that the accumulated scatter can now be defined as −39% to +34% of the desired or target tension, or it can be defined as 61%–134% of the desired tension.

This demonstrates a phenomenon that we've mentioned several times. Bolted joints are usually overdesigned to compensate for uncertainties in assembly results or in-service conditions. Designers know that they have to use lots of big bolts to avoid problems. If the assembly and service variations could be reduced, less overdesign would be required. Smaller bolts, lighter joint members, and smaller tools could do the job now done by the heavy joints and large bolts. All of this would mean economic advantages—which would be partially offset by the added costs of better control at assembly. At present the benefits offset the costs only in applications where weight, size, efficiency, etc. are also significant factors.

Even though we've considered only five variables, the ratio between the force we can count on and the force the parts must support is over 2:1. Many additional variables will have to be included in some applications, often increasing the ratio well beyond 2:1. Table 17.3 shows how the ratio affects the amount of overdesign required in a joint.

In Section 17.5 we'll try to answer the question, "Which variables should we include in our analysis?" The answer will have a large impact on the results we get, and on the estimated ratio between the maximum bolt tension and the minimum per-bolt clamping force in the joint. The ratio between "must support" and "can count on" will often be much greater than 2:1. Before deciding which variables to include, however, let's see how we might use the resulting estimate.

17.4 RELATING DESIRED TO ANTICIPATED BOLT TENSIONS

We estimated the range of bolt tension we wanted in Section 17.1 and the range we can expect above. Now we will combine our two analyses. To do this we turn the target of our last worksheet on its side and hold it up against the "desired tension" summary of Figure 17.1.

TABLE 17.3
Joint Overdesign Factor

Ratio between the Amount of Tension the Bolts must Support and the Amount of Clamping Force We Can Count On (Worse Case)	That Ratio Would Require a Bolt of the Diameter Shown below (a Perfectly Tightened 3/4 in. Diameter Bolt Could Provide the Same Clamping Force)
2:1	1⅛ in.
5:1	1¾ in.
10:1	2½ in.
16:1	3 in.

The results are shown in Figure 17.8. In doing this we have to convert the scatter percentages of Figure 17.7 to percentages of ultimate strength. The target or ideal preload was 55% of ultimate. On the downside, then, we'd expect a worst-case 61% of 55% or 34% of ultimate. On the upper end, we'd expect 134% of 55% or 74% of ultimate.

We have said that we want bolt tension to lie between 48% and 62% of UTS. The assembly/behavior analysis suggests the actual range will be 34%–74%, exceeding both the high and low ends of the desired range. How can we respond to this?

Remember that the anticipated range was based on a preselected type of assembly tool, a torque wrench in our example. Note that we were using a better torque wrench with ±5% scatter instead of ±15%. This undoubtedly got us closer to the desired results, but many assemblies will still lie outside the desired range. Going to a still better torque tool (±2% is also available) obviously wouldn't provide sufficient improvement.

The scatter in anticipated results could be narrowed considerably by going to a computerized torque–turn system, and reduced even further by ultrasonic control of assembly. A second analysis would show whether or not such refinements would make the

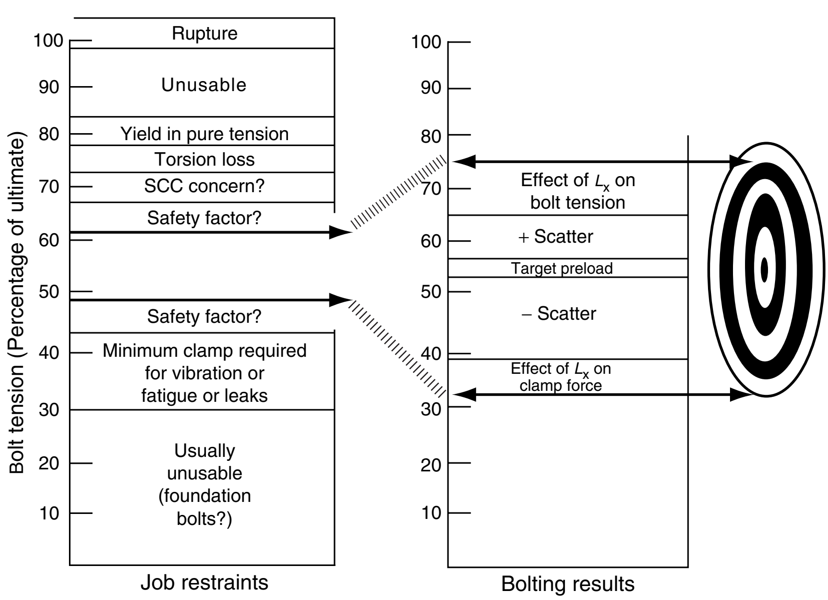


FIGURE 17.8 Chart combining the estimates of Figures 17.1 and 17.7. The range in bolt tension desired is shown on the left; the range expected is summarized on the right.

anticipated range fall within the desired range. I'm sure, in this example, that they would. But this much improvement in assembly accuracy may be economically unattractive for you. Fortunately, there are other options.

If the consequences of failure aren't too great, you can shrug your shoulders and accept the fact that some assembled joints will fail and will require field repair or maintenance, for example. The ranges we've estimated are three sigma ranges—covering the worst case with 99% probability. Most of the preloads introduced at assembly will be nearer the ideal and not very accurate ones at that. We may luck out! So, let some fail. Keep good records. Build on your experience to modify your original selection of the target preload and assembly procedures to minimize failure, even if you can't eliminate it.

Note that your estimates of the bolt tension desired may involve as much uncertainty as your estimates of expected tension. In our example, we estimated maximum and minimum tensions desired—then a couple of safety factors. Maybe that was unnecessarily conservative. So, all is not lost if "expected" exceeds "desired."

If the analysis is correct, however, and it matters, another option would be to use a different fastener. A stronger one would increase ultimate strength, raising the upper end of the desired range. The target preload could now be shifted upward as well. The lower end of the desired range would fall (would become a smaller percentage of the higher ultimate strength of the new bolt). So a stronger bolt will open up the desired range.

The desired range can also be opened up by other fastener-related changes. You may have selected a relatively high minimum tension to fight vibration loosening. Perhaps you can use an anaerobic adhesive or a prevailing torque fastener to reduce minimum tension requirements.

The maximum desired (acceptable) tension might be increased by choosing a different fastener material, if SCC is the concern, etc. There are many other possibilities of this sort which can be used to make the bolts and the joint better able to tolerate a wide variation in assembly and behavioral results. Think "overdesign!"

In some situations you'll find it easy to fit the anticipated range into the required range. In this case you could ask yourself whether or not less accurate assembly procedures would be acceptable. Presumably that would reduce assembly costs, and there's no economic sense in keeping the range of anticipated results under tighter control than required by the joint. Such results might also indicate excessive overdesign.

All of this will sound too casual to the purists reading this book. But we're dealing with a chaotic number of variables. A rigorously accurate analysis may be impossible and is often going to be prohibitively expensive. It will rarely be worth the effort. But the estimating procedure I've described, even though crude, will take you well beyond the accuracy implied by codes and other documents which merely suggest that you "tighten them correctly." And in some cases, the accuracy of our rough estimates can be quite good.

This accuracy, of course, will depend on the accuracy with which we estimate the scatter of the individual key variables. Sometimes an experiment can be made to refine our guess for one or a couple of the number of variables you pin down this way, the more accurate your estimate, of course. But you'll never—or rarely—be able to pin them all down.

In any event, I think that you'll find the analysis summarized in Figure 17.8 to be an efficient and relatively easy way to get an overview of the many different aspects of bolted joints, which we have discussed in this book.

17.5 WHICH VARIABLES TO INCLUDE IN THE ANALYSIS

17.5.1 In General

Which variables should we include when making an analysis of this sort to pick preload or torque for an existing joint? The answer to this question will obviously have a major impact on the results we get, and the answer depends upon what it is we are trying to do. Why are we trying to pick preload or torque or other control variables for this particular joint? Picking preload or a torque is something the designer should have done—if a particular value is important. If we're doing it, it can only be because we think a choice is more important than he did, or his choice wasn't communicated to us, or because—most commonly—we've been having troubles with this joint (whether or not the designer specified a preload).

17.5.2 Possible Factors to Include

The VDI analysis, which we looked at in Chapter 13 and will return to in Section 17.7, includes a long list of factors with which to start our own lists when picking a torque or other type of preload control. In their procedure they consider:

Magnitude of the external load
Stiffness ratio (''load factor'')
Eccentricity, if present
Embedment
Bending stress in the bolts
Tool scatter (e.g., torque-to-preload variations)
The effect of torsional stress on the tensile stress capacity of the bolt
The fatigue endurance limit of the bolts
The bearing stresses between nut or bolt head and joint surface
The clamping loads required to combat self-loosening, fatigue, or leakage

As far as I know, however, the VDI Directive 2230 doesn't include any of the following effects:

Differential expansion
Gasket creep
Elastic interactions
Hole interference
Resistance of joint members to being pulled together (weight effect)
Operator errors caused by lack of training, carelessness, or accessibility problems
Tool calibration errors
Safety factors

17.5.3 Which Should We Include?

Almost all of these factors are present and influence results when we tighten a group of bolts. Only differential expansion, gasket creep, hole interference, and the weight effect are application specific and may be absent. And we won't care about the fatigue endurance limit if the applied loads are noncyclical. But the others, including such major factors as tool scatter and elastic interactions (and often operator problems), are essentially always there. If so—why leave any of them out?

One big reason most of these factors have always been left out is that we have been generally unaware that they exist. Another reason: even if we know they exist we have had little or no data with which to support specific values. The VDI Directive is the only publicly available document I'm aware of which includes things like embedment relaxation and tool scatter in a formal design procedure, and gives tables of typical values. Gasket manufacturers publish some gasket creep data, but as we learned in the last chapter it is usually limited to one thickness of a given material tested only at room temperature. In this book I've included whatever further data I could find on other variables, but, considering the number of variables involved in bolting, the data will often not apply to your application.

How can we leave them out and still design or assemble most joints successfully? Easy: we have traditionally overdesigned bolted joints—and included healthy "safety factors"—to cover our ignorance. The overdesign gives us room to include the factors now, whenever we want a clearer understanding of why a joint has caused problems and want to pick a better torque or equivalent. If this is a safety-related joint, however, in an industry where designers do consider many of these factors, we should only use the analysis to probe our problems. Only the designer should pick a better torque.

Even though we'll usually include factors the designer overlooked, or use different assumptions than he did about things like the stiffness ratio and the magnitude of external loads, it's still useful to make the analysis described earlier—or the more rigorous one to be described in Section 17.7. Either analysis is relatively easy to make, so we can repeat them, using several different sets of factors, and using the typical data found in this book if we have no application-specific data of our own. Such analyses can often reveal design flaws or suggest root causes for a troublesome problem.

But don't overdo it. Start by including only the three or four factors you believe will have the most impact on the behavior of your joint, based on previous experience with that joint whenever possible. Include a safety factor or tolerance to cover uncertainties. Revise your list—or the values you're using—if the results don't seem reasonable, or don't agree with prior experience. Add or subtract percentages from some of the variables to see if results are more realistic. Try adding or subtracting 10% from the target torque, as suggested earlier in this chapter, and see what effect that has on your estimated results. Play with this stuff: don't let it dominate your life. We know too little to lock it in concrete. Remember: past experience is a better guide than any theory or procedure or equation. If it ain't broke—don't fix it!

17.6 ASTM F16.96 SUBCOMMITTEE ON BOLTING TECHNOLOGY

I have a final suggestion for those of your who are frustrated by the fact that bolted joint assembly is still dominated by uncertainty, trial-and-error, and chance; and by the fact that most standards organizations have ignored this. Slow but steady progress has been made on bolting materials, configurations, and design procedures (finite-element codes, for example) but very little progress has been made in assembly technology. As a result a group of us, representing nearly 100 organizations, formed the Bolting Technology Council (BTC), which was for many years sponsored by the Materials Properties Council, then of New York (current Ohio address in Appendix C). The goal of the BTC was to sponsor research, provide recommendations, act as a clearinghouse for information, and, in general, further the technology of bolted joint assembly and (because it affects assembly) of the behavior of bolted joints in service. A couple of years ago the ASTM agreed to sponsor this group, and the BTC has become Committee F16.96 on Bolting Technology. Anyone interested in participating in this important effort should contact the ASTM (see also Appendix C).

17.7 A MORE RIGOROUS PROCEDURE

The procedure we've looked at is very useful for making seat-of-the-pants decisions about preload and other matters, using our best guess about which factors might most influence the results—and our guesses about the possible magnitude of each effect. We use rough statistics to make the results less intimidating than they might be if we assumed the worst case for each effect. We ignored many factors we might have included and assumed typical values for such difficult-to-estimate factors as the magnitude of the external load or the joint stiffness ratio. In spite of all this, and although we certainly wouldn't want to use it to make final decisions on safety-related or critical joints, I think that the procedure is valid for most situations.

17.7.1 Experiments Required for True Accuracy

When truly accurate answers are required, we'd have to make carefully controlled experiments on the actual joint. Finite-element analysis might help, but at the present time experiments are the only way to get the accurate application-specific data we need about such things as elastic interactions, stiffness ratios, and the response of the joint to service loads. Just picking a torque or other assembly control parameter isn't enough, either. We'd also have to impose strict quality standards on such things as assembly procedures, bolt quality, and lubricity to improve the chances that results of our experiments accurately and consistently reflect production assembly results. The auto industry, for one, does all of this now and has for years.

We could use Equations 11.21 through 11.23 to utilize the experimentally collected data. Those equations are repeated below, with some examples, for your convenience.

17.7.2 The Equations

Whether or not we have experimental data with which to make our analysis, we will often want a more rigorous way to analyze the possibilities than the seat-of-the-pants method described earlier. Here's a set of equations roughly based on those found in the VDI Directive 2230, but including differential expansion and elastic interactions. The equations can easily be extended to include more factors, or reduced to include less, to fit your own applications. The discussion in Section 17.5 is just as pertinent for this rigorous analysis as for the simpler one described earlier.

Maximum bolt load (extending Equation 10.15):

$$\text{Max } F_B = (1+s)F_{Pa} - \Delta F_m - \Delta F_{EI} + \Phi_{en} L_X \pm \Phi_{en} K_J''(\Delta L_J - \Delta L_B) \tag{11.21}$$

Minimum per-bolt clamping force on the joint (extending Equation 10.16):

$$\text{Min } F_J = (1-s)F_{Pa} - \Delta F_m - \Delta F_{EI} - (1-\Phi_{en})L_X \pm \Phi_{en} K_J''(\Delta L_J - \Delta L_B) \tag{11.22}$$

Total minimum clamping force on a joint containing N bolts:

$$\text{Min total } F_J = N \times \text{per-bolt Min } F_J \tag{11.23}$$

where

F_{Pa} = the average or target assembly preload (lb, N)
ΔF_m = the change in preload created by embedment relaxation (lb, N); Equation 10.5 showed us that $\Delta F_m = e_m F_{Pa}$
e_m = percentage of average, initial preload (F_{Pa}) lost as a result of embedment, expressed as a decimal
ΔF_{EI} = the reduction in average, initial, assembly preload caused by elastic interactions (lb, N); Equation 10.6 told us that $\Delta F_{EI} = e_{EI} F_{Pa}$
e_{EI} = the percentage of average, initial preload (F_{Pa}) lost as a result of elastic interactions, expressed as a decimal
ΔL_B = the change in length of the grip length portion of a loose bolt created by a change of Δt (°F, °C) in temperature (in., mm); see Equation 11.14

ΔL_J = the change in thickness of the joint members, before assembly, if exposed to the same Δt (in., mm); see Equation 10.15
s = half the anticipated scatter in preload during assembly, expressed as a decimal fraction of the average preload; see Equation 10.1
K_J'' = the stiffness of a joint in which both the axes of the bolts and the line of application of a tensile force are offset from the axis of gyration of the joint, and in which the tensile load is applied along loading planes located within the joint members (lb/in., N/mm); K'' is the reciprocal of the resilience of such a joint (see Figure 11.15)
Φ_{en} = the load factor for the joint whose stiffness is K_J''; see Figure 11.16
N = number of bolts in the joint
L_X = external tensile load (lb, N)

Again, these equations can be rewritten using any appropriate combination of Φ and K_J.

17.7.3 Minimum Clamping Force—Some Examples

17.7.3.1 First Example—Using Worst-Case Values

Let's feed these equations some numbers to see how important it is to use the best available numbers. Let's start with Equation 11.22 and compute the anticipated minimum clamping force on the joint. For a first round we'll assume the parametric values used in the example given in Chapter 11. To repeat, these were:

Average assembly preload (F_{Pa}) = 1,430 lbs
Tool scatter (s): we assumed a full 30% below target preload, so $s = 0.3$
Embedment: assumed a typical loss of 10%, so $e_m = 0.1$
Elastic interactions: assumed an average loss of 18% for a two-piece, metal-on-metal joint, using results obtained in George Bibel's experiments, so $e_{EI} = 0.18$. (Note that Bibel tells us that the average loss would be 30% if a sheet gasket were included in the joint, or 46% if the gasket is a spiral wound type (see Section 10.1.2).
External load effect: the tensile load reduced the per-bolt clamping force on the joint by 269 lbs, based on the assumption that the external load was equal to 25% of the preload we were trying to compute (talk about winging it!) and on the eccentrically and internally loaded load factor Φ_{en} computed in Chapter 11. Instead, of course, we could use any of the other Φ_s listed in Chapter 11.
Differential expansion loss: a per-bolt clamping force loss of 208 lbs based on the assumptions that Inconel bolts were being used in a mild steel joint and that the operating temperature would be 400°F.

These assumptions give us the results shown in Figure 17.9, a minimum, per-bolt clamping force on the joint of 123 lbs. The ratio between average initial preload and residual clamping force is about 12:1. A ratio of 15:1 was calculated in Chapter 11 but was based on maximum, not average, preload and also included a gasket creep factor which I've omitted here.

Whether the ratio is 12:1 or 15:1, this is not a pretty picture! And it doesn't include many factors that might have made it worse, such as the gasket creep, which was included in the Chapter 11 example, or hole interference or the weight effect, both described in Chapter 8. But how valid is this calculation?

It purports to define the minimum tension we can expect to see in some of the bolts. To do that, however, we would have to assume a worst-case elastic interaction loss of 36% instead of an average 18%. I tried that—and computed a residual, minimum, per-bolt clamping force of

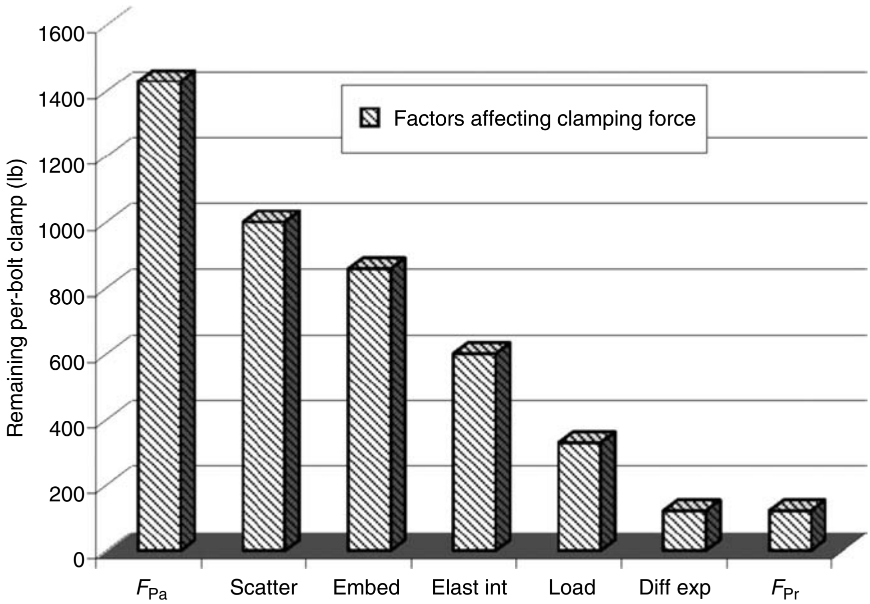


FIGURE 17.9 Chart showing the large difference expected between the average, initial, per-bolt clamping force applied to a joint during assembly (F_{Pa}) and the residual, in-service clamping force (F_{Pr}) when we assume worst-case tool scatter and embedment losses in bolt tension, plus an average elastic interaction loss, plus the unavoidable losses in clamping force caused by the external tensile load and differential expansion between bolts and joints. This is a plot of the calculations made in Chapter 11.

minus 135 lbs. Since my goal was to teach you how to use Equation 11.22, not to define and analyze an actual example, I used 18%. But it means that the computed 123 lbs is an "almost worst case" answer.

Equation 11.23 says that we can compute the minimum anticipated clamping force on a joint containing N bolts by multiplying 123 by N, which we did in Chapter 11. But this assumes that every one of the N bolts in some joints will only contribute an almost worst case amount of clamping force to the joint. I think that that's highly unlikely. I think that Equation 11.22 can reasonably predict the worst-case clamp contributed by an occasional bolt, but not by each bolt in a given joint.

17.7.3.2 Second Example—Using Statistically Combined Values

In the quick-and-dirty procedure we discussed earlier in this chapter, we used the square root of the sum of the squares of the variances to reduce the combined effect of tool scatter, embedment, and elastic interactions, to get a more cheerful—and more realistic—picture. This introduces the assumption that no one bolt will simultaneously be exposed to the worst-case minimum assembly preload, maximum embedment, and maximum interaction loss. The results, shown in Figure 17.10, are based, therefore, on the square root of the sum of squares of the following assumptions:

Preload scatter: ±30%
Embedment: –10% max
Elastic interactions: –36% max

The load and differential expansion effects remain unchanged at 269 and 208 lbs, respectively. Combining the three factors tabulated above suggests a three sigma maximum loss of only

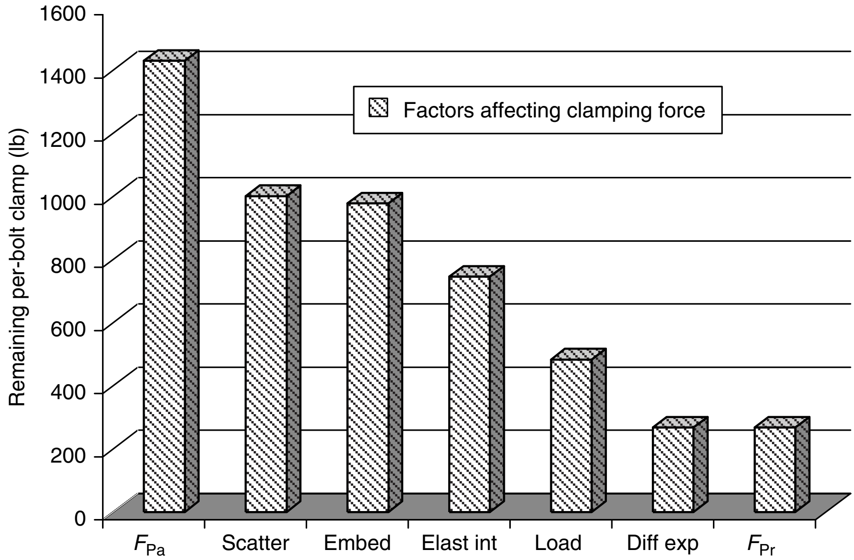


FIGURE 17.10 Chart showing the anticipated difference between initial and residual clamping force when we assume that worst-case effects will never (or rarely) occur simultaneously, that the expected deviations in individual factors can be combined by the square root of squares Equation 17.6.

47.9% of average assembly preload (from which we further subtract the load and differential losses). The resulting per-bolt clamping force is now 268 lbs, more than twice what it was under our earlier assumptions. The preload-to-residual-clamp ratio is about 5:1, not something to celebrate, but a definite improvement.

But again, I think it would be very conservative to assume that each of the N bolts in a given joint ended up with 47.9% less tension than anticipated by the target preload. What we really need is a series of experiments, a resulting list of combined losses and the mean and scatter for the combined data. Then we could use Equation 11.23 with more confidence. Such data, unfortunately, do not exist.

17.7.3.3 Third Example—Using Average Values

Could we resolve this by feeding only average data into Equation 11.22? That would suggest the following:

Tool scatter: 0% average loss!
Embedment: 5% average
Interaction loss: 18% average
Load and differential expansion losses unchanged

These assumptions would result in a big improvement in predicted clamping force, thanks to the fact that the average of ±30% tool scatter factor is 0%. Although the ±30% figure appears again and again in the literature, I think that it's unwise to assume that the plus scatter will equal the minus scatter on a given joint. We know that more factors lead to less preload than to more preload. Therefore, although your own experience may well lead you to use a different value, I'm going to use −10% for the average, single-joint, per-bolt, tool scatter.

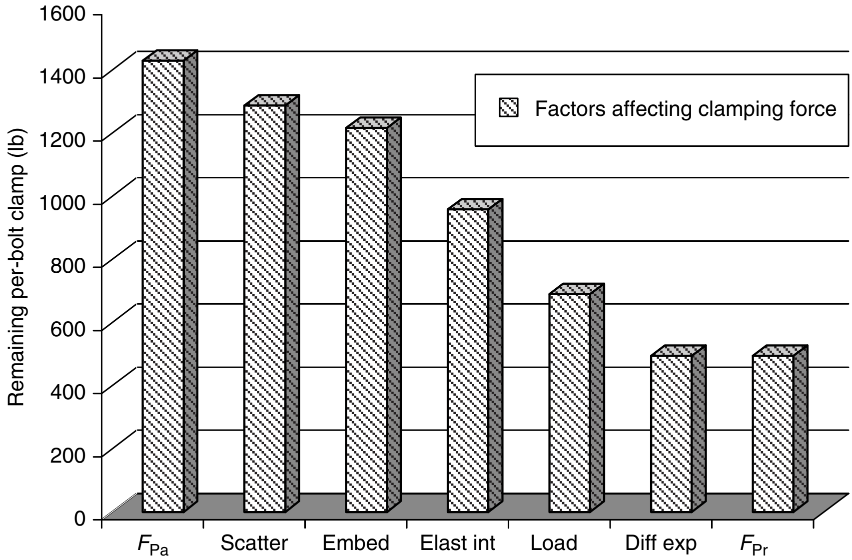


FIGURE 17.11 Chart showing the difference between initial and average residual per-bolt clamping force when we assume an average deviation for the various loss factors. This is probably a more realistic view of the results than those shown in Figures 17.9 and 17.10, especially if our goal is to compute the total clamping force applied to the joint by N bolts. See the text for a detailed discussion.

The results are shown in Figure 17.11, an average worst-case (if there is such a thing) per-bolt, in-service, clamping force of 495 lbs, and a preload-to-clamp-force ratio of only 2.9:1. I would feel comfortable multiplying this number by N to estimate the total force on the joint. To repeat, however, every answer we get depends upon which factors we've included. Only experiments on the actual joint will give us a truly accurate answer.

17.7.3.4 Fourth Example—Using Feedback Control Values

If we use strain gage, ultrasonic, or other feedback control of final assembly preload, we can reduce the combined effects of tool scatter, embedment, and elastic interactions to $\pm10\%$. The results arc shown in Figure 17.12, a ratio between target preload and final, per-bolt clamping force of only 1.8 to 1. Since the load effect and differential expansion losses can be accurately predicted if we know joint materials and geometry—and, the tough part, the magnitude of the loads we will see in service—this would also eliminate the need for joint experiments. The feedback gives us the experimental data we need as we assemble each joint. No more need to estimate assembly loss factors or even to decide which to include in our analysis. But this type of control is expensive and we'll usually have to settle for estimates, which we'll use with either the quick-and-dirty procedures described first in this chapter or in a more rigorous analysis such as that described in this section.

17.7.4 Maximum Bolt Tension

We have focused this more rigorous discussion entirely on the use of Equations 11.22 and 11.23. We should never accept the assembly preloads suggested by use of these equations without also using Equation 11.21 to estimate the worst-case stresses our choice would create in some of the bolts. These were calculated, correctly for the worst-case example, in Chapter 11, and so I won't repeat it here. Note that it's perfectly reasonable to use worst-case deviations when estimating maximum bolt loads, because we don't want any bolt to break

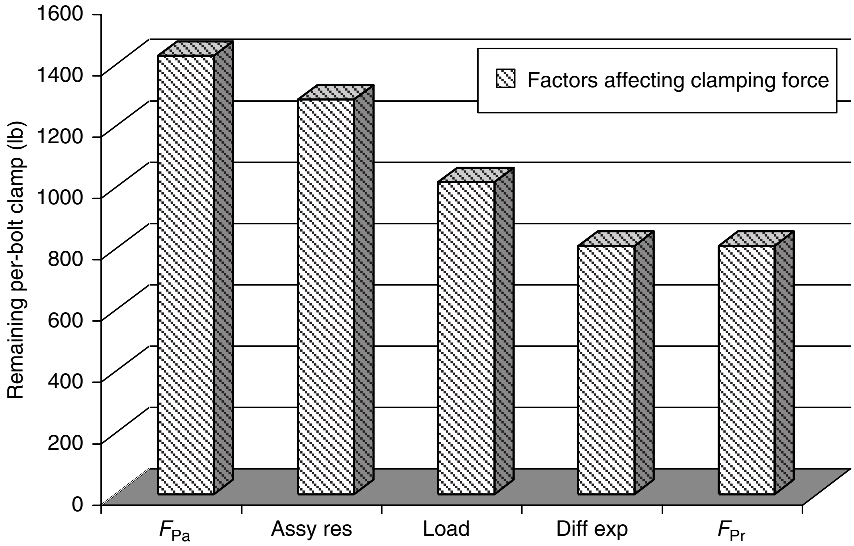


FIGURE 17.12 Chart showing the anticipated difference between initial and residual clamping force when feedback control is used during assembly to correct for tool scatter and compensate for embedment and elastic interaction loss, giving a practical, worst-case deviation of assembly preload of only 10% of average assembly preload.

and we don't multiply the results by N or something. Even here, however, our inputs to the equation can distort the results and penalize our design. If we're planning to use torque is it reasonable to assume a worst-case +30% scatter, for example, or will that be reduced when we lubricate the bolts?

Regardless of which procedure you use, I think that each of those we've examined in this chapter has its place and can be useful—can help you to estimate, however roughly, the assembly and in-service clamping forces in your joints, and thereby can help you to analyze and solve many bolting problems. They can certainly give you the insight you need to use good engineering judgment when picking a better torque—or when picking a more appropriate assembly control variable. They've also given us a chance to review many of the topics we've considered in this book.

17.8 NASA'S SPACE SHUTTLE PRELOAD SELECTION PROCEDURE

Selecting correct preload is always important when joint safety is involved: when failure of the joint would lead to loss of life or serious equipment malfunction. It's interesting, therefore, to look at a procedure which NASA has developed for selecting preload for bolted joints found on the space shuttle. This procedure confirms many of the things already said in this chapter but gives us a slightly different view of the preload selection process and its ramifications.

17.8.1 Calculating Maximum and Minimum Preloads

Four different procedures are given for identifying the possible maximum and minimum preloads created during assembly.

1. The first uses modifications of the prevailing torque versions of the long-form or short-form, torque–preload equations (Equations 7.3 and 7.5). These are solved by

inputting typical friction coefficients or nut factors. These typical values must be those obtained from prior experience or, presumably, technical reports, involving the same materials, surface finishes, and lubricants. Uncertainties are compensated for by tool scatter factors. Unlike the equations in Chapter 7, the NASA equations also include a term for any anticipated differential expansion between joint members and bolts ($\pm F_{th}$). Here are the resulting long-form equations, using the symbols and terms used in the present text rather than those used by NASA.

$$F_{P\max} = [(1+s)T_{\max}]/[r_t(\tan\alpha + \mu_t/\cos\beta) + \mu_n r_n] + F_{th}{}^{pos} \tag{17.7}$$

$$F_{P\min} = [(1-s)(T_{\min} - T_P)]/[r_t(\tan\alpha + \mu_t/\cos\beta) + \mu_n r_n] + F_{th}{}^{neg} - F_{em} \tag{17.8}$$

where

s = tool scatter as a decimal (See Table 17.4)
T_{max} = maximum limit of a specified, assembly torque range (in.-lb)
P = thread pitch (in.)
μ_t = coefficient of friction between male and female threads
r_t = effective radius of thread forces (in.)
α = thread lead angle (for UN threads $\tan^{-1} = [1/(\pi n E)]$
β = thread half angle
E = basic pitch diameter of external threads
μ_n = coefficient of friction between nut face or bolt head and joint member
R_e = effective radius of torqued element (nut or head) (in.)
F_{th} = increase or decrease of the force on the bolt as a result of differential thermal expansion (lb)
T_P = prevailing torque (in.-lb)
F_{em} = loss of preload due to embedment (lb)
N = threads/in.

NASA modifies the short-form, torque–preload equation in similar fashion. Here, for example is their expression for F_{Pmin}.

TABLE 17.4
Preload Control Scatter Factors

Control Means Used	Anticipated Scatter
Torque measurement of unlubricated bolts	±35%
Torque measurement of cad-plated bolts	±30%
Torque measurement of lubricated bolts	±25%
Hydraulic tensioner	±15%
Preload indicating washers	±10%
Ultrasonic measurement device	±10%
Bolt elongation measurement	±5%
Instrumented bolts	±5%

Source: From "Criteria for Preloaded Bolts," NASA Space Shuttle Program Document NSTS 08307, Revision A, Lyndon Johnson Space Flight Center, Houston, Texas, July 6, 1998.

Note: NASA says that these estimates should only be used on bolts of diameter ≤¾ in. The scatter factor (uncertainty) for bolts of larger diameter must be determined by application-specific tests.

$$F_{\mathrm{Pmin}} = (1 - s)(T_{\min} - T_{\mathrm{P}})/KD + F_{\mathrm{th}}^{\mathrm{neg}} - F_{\mathrm{em}} \tag{17.9}$$

where
K = the nut factor
D = nominal diameter of the bolt

NASA says that the embedment loss can be 5% of F_{Pmax} if this is a metal-to-metal joint. The embedment loss of preload in gasketed joints or joints involving nonmetals must be determined by an application-specific test.

2. The second procedure uses basically the same equations but this time uses friction coefficients or a nut factor which have been determined by application-specific testing on an existing joint or in a test where everything must be the same as in the actual joint: materials, finishes, dimensions, tools used etc. Because he is now using actual coefficients the designer no longer needs to include the $(1+s)$ or $(1-s)$ terms in the equations.
3. In the third recognized procedure maximum and minimum preloads are measured after tightening a group of bolts to yield, in an application-specific joint.
4. In the fourth and final recognized procedure preload is controlled by means of a torque wrench or tightening the bolts to yield. Now the designer must place a maximum and minimum tolerance band on the acceptable preload. The calculations also include a tool scatter factor, plus differential expansion and embedment factors, as in this example:

$$F_{\mathrm{Pmin}} = (1 - s)(F_{\mathrm{P}} - \mathrm{TOL}) + F_{\mathrm{th}}^{\mathrm{neg}} - F_{\mathrm{em}} \tag{17.10}$$

17.8.2 Confirming the Preload Calculations

The NASA procedure does not end with the calculations of maximum and minimum preloads. The designer must now consider the impact of shear and bending loads on the fastener; he must satisfy himself that the maximum expected loads on the joint, added to preloads and including safety factors, will not exceed the tensile strength of the bolt or the shear strength of the threads; he must make sure that the joint will not separate under the anticipated joint separation load and under minimum anticipated preload conditions. He must also consider fatigue loading and fracture criteria.

17.8.3 Discussion

This NASA procedure covers all of the variables and uncertainties described earlier in this chapter, which is reassuring. But it all starts, mysteriously, by picking a torque and then using that to compute preloads. Furthermore, a procedure for selecting this torque is not defined or mentioned in their document. In the procedures you'll find throughout this book, by contrast, the designer first defines an acceptable assembly clamping force (and therefore desired preloads). He then worries about the effects of service conditions and loads on the clamping force before finalizing his selection of assembly preload. Only after that does he decide how much torque, or other control parameters, should be used to create that initial clamp. It's obvious, then, that a NASA designer must make some important design decisions before addressing the material in document NSTS 08307. Their success with the space shuttle program proves this suggestion.

EXERCISES

1. Name at least four factors which can reduce in-service clamping force.
2. Name at least three factors which can be prevented or at least reduced by sufficient in-service clamping force.
3. Name at least four factors which determine the allowable upper limit of the in-service clamping force.
4. At approximately what percentage of the ultimate tensile stress (UTS) does a common bolt (e.g., ASTM A490 or SAE J429 GR 8) yield in pure tension?
5. Name two simple ways to pick a preload.
6. What's the easiest way to pick a preload for joints which are not safety related or economically important?
7. Compute a reasonable torque for an as-received, unlubricated ½–32 × 4 SAE J429 GR 5 bolt in a nonsafety-related normal application.
8. Compare your torque to that suggested in Appendix E.
9. When estimating bolt tension as in Figure 17.4 why do we go to the trouble of using the square root of the sum of the squares? Why not just add the error variables arithmetically?
10. What would be the cumulative scatter of the four variables we used in the Figures 17.2 through 17.7 if we combined them arithmetically?
11. How does this compare with the scatter computed in the text using the square root of the sum of squares?
12. What do we mean by an overdesigned joint?
13. Who should pick the preload for a safety-related joint?
14. In most situations what is your best guide to a correct torque?
15. Why should you keep records of the torques and procedures used in your applications?

REFERENCES AND BIBLIOGRAPHY

1. Hayman, E. and W. Brown, "Assembly of Bolted Joints" presented at a PVRC Bolted Flanged Connection Workshop in Safety Harbor, Florida, February 06–07, 2002.
2. "Criteria for Preloaded Bolts," NASA Space Shuttle Program Document NSTS 08307, Revision A, Lyndon Johnson Space Flight Center, Houston, Texas, July 6, 1998.
3. Novak, G.J. and T. Patel, Thread lubricants, *Handbook of Bolts and Bolted Joints*, J.H. Bickford and S. Nassar (Eds.), Marcel Dekker, NY, 1998.
4. Shigley, J.R. and C.R. Mischke (Eds.), *Standard Handbook of Machine Design*, McGraw-Hill, New York, 1986, p. 23.23.

18 Design of Joints Loaded in Tension

This is the first of two chapters on the design of bolted joints. In this first chapter we are going to discuss, in general, the design of joints loaded in tension, first examining a typical procedure, then the more rigorous, VDI procedure. Then, in Chapter 19, we'll take a brief look at the design of a joint loaded in shear. Those interested in the design of gasketed joints should refer to volume 2 of this text.

To repeat one of the earliest points we covered, the most common purpose of the bolted joint is to clamp two or more things together. When we design a bolted joint, therefore, we are usually designing a clamp and will be concerned about such things as the strength of the clamp, the integrity or reliability of the clamp, the stability of the clamp in service, and the life of the clamp.

The specific factors a given designer must consider will depend, of course, on the application of his particular joint. Designers of joints for nuclear power plants or aircraft will obviously have to consider many more factors and make better decisions than will designers of less critical joints.

18.1 A MAJOR GOAL: RELIABLE JOINTS

To the extent that he can afford it, every designer will be interested in reliable joints. This will mean different things in different applications, of course, and will require joint sizes and configurations appropriate for specific service conditions. In spite of the fact that reliability can mean many different things, I think that the following checklist will be a useful place to start our summary of the design process. It tabulates the main issues which determine reliability. Few applications will involve all of the points listed here, but each topic is a valid concern somewhere.

When you start a new design, review this list. Check the items as you read them, to force yourself to focus, if only briefly, on each issue. Double check, or make a separate list of those items which you feel will affect reliability in your particular application.

18.1.1 Checklist for Reliable Bolted Joints

1. I recognize that a bolt is a clamp; that an unreliable joint is synonymous with an inadequate clamp.
2. I realize that reliability requires the following:
   i. A joint that is strong enough (rigid enough) to provide sufficient structural integrity to prevent slip, separation, vibration, misalignment, wear, etc. of the interconnected parts of the product or system
   ii. Enough bolts

   iii. Adequate bolt diameter
   iv. Appropriate material strength
   v. Sturdy joint members
3. A joint (clamp) which is also stable under service conditions, i.e., won't degrade or weaken. (Factors such as corrosion, vibration, fatigue, and elevated temperatures can change the clamp.) Stability requires the things tabulated below.
4. Materials which are stable in the service environment:
   i. Won't corrode excessively
   ii. Have an acceptable resistance to stress corrosion cracking
   iii. Have an acceptable resistance to fatigue
   iv. Won't lose too much strength at operating temperatures
   v. Won't relax too much at elevated temperatures (because of stress relaxation, creep, or reduction in the modulus of elasticity)
5. Bolt and joint geometry (shape which encourages stability of the clamp):
   i. Properly designed to minimize stress concentrations (e.g., bolt head-to-shank fillets and thread run-out details which reduce the chances of fatigue failure; another example—joint shear planes which coincide with thread run-out encourage failure of the bolt in shear).
   ii. Bolt-to-joint stiffness ratios which direct external loads or loads created by differential expansion to the components (bolts or joint members) best able to support them. (Usually means stiff joint members, flexible bolts.)
6. An in-service clamping force which is able to enforce stability. (The in-service clamping force will be determined by the preloads introduced at assembly, but modified by service loads and/or thermal effects.)
7. Enough clamp:
   i. Clamping force must be high enough to minimize self-loosening of the bolts under vibration, thermal cycles, joint flexing, etc.
   ii. It must be high enough to minimize load changes in the bolts (thereby improving fatigue life).
   iii. It must be high enough to prevent joint slip (which encourages self-loosening, wear, unexpected stress concentrations in joint members and connected parts, etc.).
   iv. It must be high enough to prevent leaks, a problem in itself, but which also lead to corrosion and further degradation of the clamp.
8. But not too much:
   i. Clamping force must also be low enough to avoid excessive stress in bolts; this could lead to stress corrosion cracking, for example, or tensile failure under unexpected loads or thermal effects.
   ii. Clamping force must be low enough to avoid damaging or distorting joint members (e.g., excessive flange rotation).
   iii. Clamping force must be low enough to avoid crushing a gasket (which can also cause leakage).

18.2 TYPICAL DESIGN STEPS

The checklist above helps us set our sights. Now, how do we proceed?

Most bolted joints in this world probably are, and will continue to be, designed by gut feel based on past experience with similar joints. This is perfectly acceptable. A complete design analysis can be very expensive, thanks to the large number of variables and factors involved. Analysis is usually justified if one is starting from scratch with no prior experience in a particular application, and if the consequences of failure are severe; or if the joint is to be mass-produced and overdesign would be uneconomical.

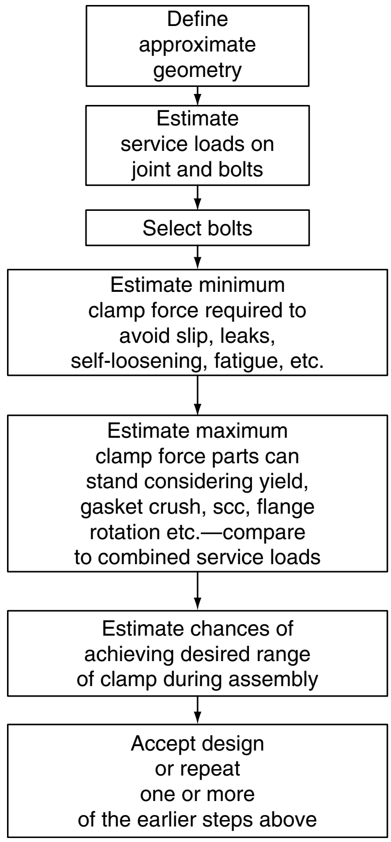


FIGURE 18.1 Block diagram that summarizes the activities involved in the design of bolted joints.

Even with critical joints, past experience will be involved and pertinent, of course, again because of the large number of variables and the inevitable uncertainty in each which will always make the outcome less than certain.

No matter what the specific application for a bolted joint, certain common design steps will usually be included in the procedure. The amount of attention devoted to each step will depend on the importance of the joint, but the step will be there in some form or other in most situations. The steps I'm about to describe are summarized in Figure 18.1.

18.2.1 Initial Definitions and Specifications

The obvious first step is to define the purpose of the joint and to rough out some preliminary specifications concerning the use of the equipment or system in which the joint is located, defining such things as operating speed, temperature, desired life, estimated cost targets, etc.

18.2.2 Preliminary Design

Preliminary geometric layouts are usually next. These roughly define the size and shape of the various parts involved, including the joint members.

18.2.3 Load Estimates

Once we know how big this thing is, roughly how much it weighs, and the intended use, we can start to guesstimate the possible loads on the bolted joints. Except for obviously critical joints, this is an oft-neglected step, but for a good reason. It's often very difficult to estimate service loads. If you're serious about the analytical design of bolted joints, however, you can't ignore this step.

As far as the joints are concerned, these loads will presumably include such things as weight, inertial affects, thermal affects (see Chapter 11), pressure, shock, etc. Both static and dynamic loads must be estimated. Load intensifiers (prying, eccentricity, etc.) should be acknowledged if present (see Chapters 11 and 19).

18.2.4 Review Preliminary Layouts: Define the Bolts

The load estimates will, of course, affect the size and shape of parts, and vice versa. In fact, in most situations, the geometric layout and load-estimating steps will be performed simultaneously. In any event, with approximate joint geometry and loads established, we can now make a preliminary selection of bolt size and number. Important parameters to be defined at this point include nominal diameter, grip length, number of threads per inch, tensile stress area, the bolt material, and the strength of the material.

At this point it would also be useful to estimate the stiffness of the bolt using Equation 5.10 and then go on to estimate the stiffness of the joint using the procedure described in Section 5.2. We'll need these estimates to predict the effects of external loads on the joint, the danger of fatigue failure, etc.

It's also useful, when selecting the bolts, to estimate their static strength to set an upper bound on the stress or tension which they can support. We will rarely design the joint to impose such stresses on the bolt, but it is useful to define this upper limit. At this point, the designer should also estimate the stripping strength of the threads of bolt unless an off-the-shelf bolt is to be used with an off-the-shelf and recommended nut. Stripping strength should always be estimated if the length of thread engagement is abnormally short or if the bolts are to be tightened into tapped holes in a soft material such as aluminum.

18.2.5 Clamping Force Required

18.2.5.1 Minimum Clamp

We have now roughed out the joint configuration, picked the bolts, and estimated the maximum tensile strength (and, therefore, clamping force) available from these bolts. Our next step (which many designers will place first!) is to estimate the minimum amount of clamping force the joint must have to avoid failure in this application. We'll design and assemble for more than this minimum, to be safe, but start by asking, "What's the least required here?"

If the joint must only face static or slowly moving tensile loads, then it will probably be sufficient to design for clamping forces which are somewhat greater than the maximum anticipated tensile load. If some overdesign is acceptable, the designer can ignore such things as the joint diagram and assume that the bolts will see any external tensile load in its entirety.

The amount by which the assembly and in-service clamping forces selected by the designer should exceed the external load will depend on such things as the accuracy with which the bolts are to be tightened, the accuracy with which service loads can be estimated, the consequences of failure, etc. As far as assembly accuracy is concerned, the less accurate the tool, the greater the design clamping force. The larger design value, of course, means that the joint will have enough clamping force even if grossly undertorqued during assembly. This is one of the factors which cause most joints to be overdesigned—heavier than they need to be to compensate for assembly uncertainties. As an example, many people want the nominal clamping force to be three to four times the anticipated service loads.

In critical situations, the designer must carefully estimate the service loads on the joint and the required resistance to those loads when selecting a clamping force. The goal, of course, is to eliminate any chance of bolted joint failure. The analysis, therefore, can be defined in terms of failure modes, as follows. The designer need only concern himself with one or a couple of these possibilities in most applications. Few joints are threatened by all possible modes of failure. The possibilities were listed in the last chapter and included joint slip or separation, self-loosening, and fatigue. We'll take a closer look at this in Section 18.4.

18.2.5.2 Maximum Clamp

As mentioned several times in this and previous chapters, it's not enough to define a minimum clamping force. We must also satisfy ourselves that the joint—and especially the bolts—are never exposed to too much stress. As we saw in the last chapter, the clamping force is usually limited by a bolt strength factor; including the bolt's fatigue strength, thread-stripping strength, or susceptibility to stress corrosion cracking. Sometimes, however, it's the crushing strength of a gasket or the bearing strength of the joint member which determines the upper limit. See the last chapter, or Section 18.4, for more specifics.

18.3 JOINT DESIGN IN THE REAL WORLD

Figure 18.1 implies that joint design is a well-organized, step-by-step logical procedure. It can be and it has been reduced to this by codified procedures in several industries. But, in most situations, I suspect it's governed more by impulse and intuition than logic. The designer starts at this point in the process which most interests or concerns him, leapfrogs to his next concern, backs up to repeat an earlier step, balances one concern against another, and circles and cycles until a joint is born.

Design is, after all, a creative process, and too many rules can be counterproductive.

The complex technology of the bolted joint, however, makes rules, shopping lists, guidelines, etc. often helpful and sometimes essential. If the joint is important the several topics listed in Figure 18.1 should be addressed by the designer, even if he chooses to do so in a different order. For truly critical joints, where the consequences of failure are severe, the designer will want to go well past the relatively simplistic approach described in Section 18.2. Perhaps he'll resort to a finite-element analysis. In other situations he'll want to adhere rigorously to a codified procedure, many of which are mandated by law. Or, he may want to use the procedure published some years ago by the German engineering society VDI. This is the most detailed, publicly available, general-purpose, joint design procedure I've encountered. We first used a version of it in Chapters 10 and 11 when we were studying the behavior of a joint under tensile loads. We returned to it in the last chapter; using it to define a more rigorous procedure for selecting preload for an existing joint. Now we're going to use it for the purpose for which it was intended, to design a joint.

18.4 VDI JOINT DESIGN PROCEDURE

I'm basing the following discussion primarily on the G.H. Junker paper [1] listed in the references at the end of this chapter. Although my copy isn't dated, I know that this paper predates the first edition of this book, and therefore is probably about 25 years old. The VDI procedure was first published in 1977 [2] and was modified later. The most recent version I'm aware of was published in 1986 [3]. I'm sure that the procedure described by Junker is still valid, however, and we'll follow it step by step, with only a few modifications to accommodate some personal judgments or new factors such as elastic interactions.

18.4.1 Terms and Units

We'll use the following terms and units while following the VDI procedure:

A_P = contact area between bolt head, or nut face, and the joint (in.^{2}, mm^2).
A_r = root diameter area of the threads (in.^{2}, mm^2); see Table 3.3.
A_S = tensile stress area of the threads (in.^{2}, mm^2); see Appendix F.
F_B = tension in bolt, in general (lb, N).
ΔF_B = change in bolt tension caused by external load, L_X (lb, N).
Max F_B = maximum estimated bolt tension, e.g., as a result of Max F_P plus the effects of external load, L_X etc. (lb, N); equals the maximum per-bolt clamping force on the joint under the present assumptions.
Min F_B = minimum estimated bolt tension and per-bolt clamping force (lb, N).
F_{Krqd} = the minimum preload (or clamping force) required to prevent separation of an eccentrically loaded joint (lb, N).
F_P = preload in general (lb, N).
ΔF_P = loss of preload during or immediately following assembly because of embedment and elastic interaction effects (lb, N).
Max F_P = maximum anticipated per-bolt preload during assembly (lb, N).
Min F_P = minimum anticipated per-bolt preload during assembly (lb, N).
F_{Pa} = "target" preload used to compute the torque or other control parameter to be used at assembly (lb, N).
F_{Prqd} = minimum preload (or per-bolt clamping force) required to prevent slip, separation, or leakage of a concentrically loaded joint (lb, N).
F_J = per-bolt clamping force on the joint, generally assumed to equal the existing per-bolt preload (lb, N).
ΔF_J = the change in per-bolt clamping force created by external load, L_X (lb, N).
Max F_J = maximum bolt preload and clamping force created during assembly; before the joint is put in service (lb, N).
Min F_J = minimum bolt preload and clamping force created during assembly, before the joint is put in service (lb, N).
F_y = the tensile force required to yield the bolt (lb, N).
K_J' = stiffness of a concentric joint loaded at internal loading planes (lb/in., N/mm).
ΔL_B = increase or decrease in the grip length section of the bolt because of a temperature change (in., mm).
ΔL_J = increase or decrease in thickness of the joint because of a temperature change (in., mm).
L_S = external shear load (lb, N).
L_X = external tensile load applied to joint (lb, N); also equals the maximum external load experienced during a load cycle if the load varies.
L_{Xmin} = minimum external load experienced during a load cycle if the load varies (lb, N).
P_G = maximum allowable pressure which can be exerted on the joint by the bolt head or nut without damaging the joint; Pa is usually a little greater than the yield strength of the joint material (psi, MPa).
R_G = radius of gyration of the joint (in., mm).
a, s, u = important dimensions of an eccentrically loaded joint, illustrated in Figure 11.12 (in., mm).
s = scatter in preload caused by the assembly tools and procedures; e.g., if torque is used for control, with a resulting scatter of $\pm 30\%$, then $s = 0.30$. A few values, based on a VDI table, are given in Table 18.1.

TABLE 18.1
Preload Scatter Factors

Factor Used by VDP[a] α_A	Factor Used in This Book $\pm s$ (%)	Type of Control Used during Assay
1	0	Yield control[b]
1.2	10	Elongation control
1.35–1.86	15–30	Torque control[c]
1.5–2.64	20–45	Airstall power wrench
1.67–4.0	25–60	Impact power wrenches[d]

[a] I computed these values using the data tabulated in Figure 7 of Ref. [5] and shown in the next column. I believe that the Ref. [5] numbers are based on Table 17 of Ref. [2] (which is in German).
[b] VDI, influenced by Junker and SPS, say that yield control is nearly perfect: that any scatter will be caused only by variations in the yield points of individual bolts. This view can, of course, be disputed.
[c] VDI says that the lower values can be achieved by using an experiment on the actual bolts to pick the torque. The possible reduction in scatter caused by a good lubricant is not considered.
[d] VDI says that the higher values are for soft connections or rough surface finishes.

α_A = the scatter in preload caused by the assembly tools and procedures as defined by VDI. $\alpha_A = \text{Max } F_P/\text{Min } F_P$. If we use a $\pm30\%$ torquing procedure, $\alpha_A = 1.30/0.70 = 1.86$. See Table 18.1 for other VDI values.
Φ_{Kn} = load factor for a concentric joint, loaded internally at loading planes. $\Phi_{Kn} = \Delta F_B/L_X$.
μ = coefficient of friction between joint surfaces.
σ_A = the bolt stress at its endurance limit (psi, MPa).
α_y = the yield stress of the bolt (psi, MPa).

18.4.2 Design Goals

We have two main goals.

1. An in-service clamping force great enough to prevent slip, separation, or leakage (partial unloading).
2. Bolts strong enough to survive and support the maximum assembly preload plus the maximum service loads, including those caused by tensile and shear forces and thermal effects.

18.4.3 General Procedure

We'll assume that we have roughed out the configuration and size of this joint and its bolts, starting basically as we did in the procedure described in Section 18.2. We have also estimated the type and magnitude of the external loads to be placed on this joint; and have decided how much clamping force will be required to combat anticipated failure modes and service conditions. Now we can use the VDI procedure to analyze, refine, and correct our initial decision, as follows.

1. We'll make a preliminary estimate of the assembly preload requirements, taking things like tool scatter and relaxation effects into account. We'll assume that preload equals the per-bolt clamping force, too (i.e., no hole interference or weight effect).
2. Then we'll revise our estimate of the assembly preload requirements to add the effects of the external loads on the joint. As we take this step we'll introduce a value for the minimum clamp load required in service.
3. Next we'll add the effects of some other types of load the joint might see, if we think that they might be present. These include:
   - Shear loads
   - Eccentric loads
   - Bending loads on the bolts or on the joint
   - Load changes caused by differential expansion
   - Dynamic (i.e., fatigue) loads
   - Combinations of the above
4. Finally we'll check our results against several limiting conditions to make sure that neither the bolts nor joint members are overstressed.

We'll do all this to determine

- Maximum tension seen by some bolts
- Minimum clamp force seen by some joints
- Specifications for the optimum bolt for this joint

18.4.4 Estimating Assembly Preloads: Preliminary Estimate of Minimum and Maximum Assembly Preloads

VDI says that we should use the following expressions for preliminary estimates of the minimum and maximum preloads to be created during assembly

$$\text{Min } F_{\text{P}} = F_{\text{P}} + \Delta F_{\text{P}} \tag{18.1}$$

We express this minimum as the sum of an as-yet unidentified preload F_{P} and the reduction in that preload which will be caused by relaxation effects during assembly. We do this in order not to overlook the fact that the assembly preload must be great enough to compensate for the relaxation.

The ΔF_{P} term gives us our first problem, or rather, our first need for some engineering judgment. VDI equates ΔF_{P} to embedment relaxation, but we now know that elastic interactions will also be present and can cause far more loss than will embedment. In my opinion, therefore, we should start by including these interactions. If the results suggest an unacceptable spread between maximum bolt tension and minimum clamping force, then we must consider using an assembly procedure that compensates for relaxation.

The expression we use for a preliminary estimate of the maximum assembly preload is simply α_{A} times the minimum preload or

$$\text{Max } F_{\text{P}} = \alpha_{\text{A}}(F_{\text{P}} + \Delta F_{\text{P}}) \tag{18.2}$$

It's important to note at this point that VDI assumes that these max and min assembly preloads are equal to and opposite to the max and min clamping forces which the bolt will create on the joint. Things like hole interference and weight effect, in other words, are ignored (as they should be for most joints).

18.4.5 Adding the Effects of the External Load

We have not yet fully defined the max and min preloads we want to create during assembly, because we haven't taken another important clamp load loss factor into account. We do so now. The per-bolt clamping force on the joint will be reduced, by the external load, by an amount equal to

$$\Delta F_{\mathrm{J}} = (1 - \Phi_{\mathrm{Kn}})L_{\mathrm{X}} \tag{18.3}$$

The external load will also add to the tension in the bolt, as follows:

$$\Delta F_{\mathrm{B}} = \Phi_{\mathrm{Kn}} L_{\mathrm{X}} \tag{18.4}$$

We're going to ignore ΔF_{B} for the moment and concentrate on ΔF_{J}, the loss in clamp load caused by the external load. We must compensate for this loss in our selection of assembly preload.

The joint must be able to function, in service, with the minimum clamping force which remains in the joint after relaxation and external load effects have done their worst. I'm going to follow the VDI procedure and call this remaining clamping force the "force that's required" to prevent the joint from misbehaving or failing, or F_{Prqd}. Junker calls F_{Prqd} "the decisive factor in the design equation." This is the clamping force which must be able to resist slip, separation, or leakage. I said in an earlier chapter that the purpose of the bolted joint was to generate a force: this is that force.

We can now rewrite the expression for the minimum and maximum preloads we want to create during assembly—taking tool scatter, relaxation, and load effects into account, as illustrated in Figure 18.2

$$\text{Min } F = \Delta F_{\mathrm{J}} + F_{\mathrm{Prqd}} + \Delta F_{\mathrm{P}} \tag{18.5}$$

and

$$\text{Max } F_{\mathrm{J}} = \alpha_{\mathrm{A}}\left[\Delta F_{\mathrm{J}} + F_{\mathrm{Prqd}} + \Delta F_{\mathrm{P}}\right] \tag{18.6a}$$

We can rewrite that last expression in terms of the external load or

$$\text{Max } F_{\mathrm{J}} = \alpha_{\mathrm{A}}\left[(1 - \Phi_{\mathrm{Kn}})L_{\mathrm{X}} + F_{\mathrm{Prqd}} + \Delta F_{\mathrm{P}}\right] \tag{18.6b}$$

Now, VDI says that we can use Equation 18.6 to size the bolts. They do this by using the relationship

$$\text{Max } F_{\mathrm{J}} < 0.9F_{\mathrm{y}} \tag{18.7}$$

Why $0.9F_{\mathrm{y}}$? VDI assumes that the bolts will be tightened by turning the nut, whether or not torque is selected as the control variable, and that some of the input energy will be turned into torsional stress, reducing the bolt's capacity to support tensile stress by 10%.

Note that these equations for max and min joint clamping force, F_{J}, are the values we want to or expect to introduce at assembly, before the external load is actually applied. They are, therefore, still equal and opposite to the preloads created when we tighten the bolts. The Max F_{J}

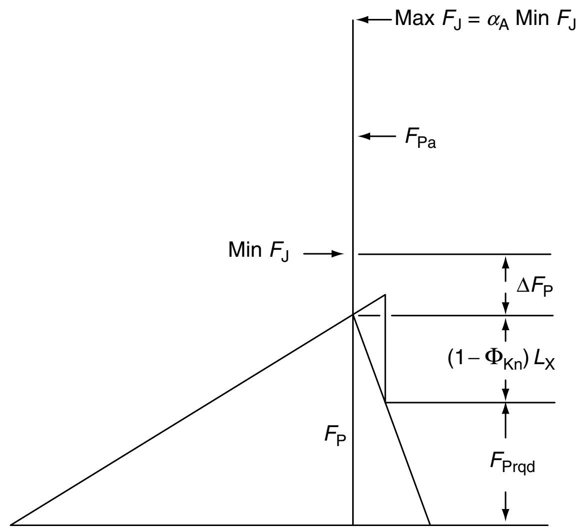


FIGURE 18.2 This joint diagram illustrates Equations 18.5 and 18.6. We start by applying an assembly preload equal to F_P plus ΔF_P to the joint. Elastic interactions and embedment reduce this to just F_P. This is equal and opposite to the clamping force on the joint, F_J. When the joint is placed in service the external tensile load will further reduce the clamping force by a factor equal to $(1 - \Phi_{Kn})L_X$, leaving a residual clamping force called the "force required to combat joint failure," F_{Preq}. The minimum required assembly preload, Min F_J, is the sum of F_{Preq} plus the two loss effects. The maximum assembly preload is α_A times that. The target preload, F_{Pa}, to be used to compute an assembly torque lies halfway between the max and min F_J's.

does not define the maximum tension the bolts will see when actually placed in service. We'll get to that in Equation 18.9.

18.4.6 Is the Required Force Good Enough?

At the beginning of this design procedure we defined the failure modes we fear, and the minimum clamping force we think will be required to prevent them. We can now enter that clamping force, F_{Prqd}, in Equation 18.6. Is the value we picked appropriate?

In answering this question we are, in effect, repeating the "how much clamping force do we need?" question raised in Sections 17.2 and 17.4. If you skipped those, you might want to review them now. Or review the various chapters devoted to fatigue, self-loosening, and the like.

At this point in the procedure, however, VDI gives us the following suggestions, with which to evaluate our estimate. To combat

Separation: The residual clamping force, F_{Prqd} must be greater than zero, and the more greater the better! Limited by the forces/stresses the parts can stand.

Transverse slip:

$$F_{Prqd} > F_S/\mu \tag{18.8}$$

Leakage: F_{Prqd} must be large enough to create the residual gasket stress currently defined by the ASME Code's *m* factor—or the residual gasket stress defined by the new PVRC gasket factors.

Fatigue: Clamp load F_{Prqd} must be as large as possible—presumably limited by how much force or stress the parts can stand. If this is an eccentric joint, subjected to cyclic loads, F_{Prqd} will be replaced by F_{Krd} as discussed under Section 18.4.7.6 below.

This ends Mr. Junker's list of possible clamp force requirements. We might want to add other possibilities, again based on the discussion in Section 17.4. For example:

Creep relaxation: If this is a gasketed joint we'll certainly want to increase the assembly preload to compensate for the subsequent loss due to gasket creep by adding a term covering this loss to those within the brackets in Equation 18.6. We might call this term ΔF_{Cr}. We want to do this—but sometimes may not be able to. Sometime the loss is so great the initial preload required to compensate for it would crush the gasket, and we have to compensate by retightening the bolts after creep has occurred. Nevertheless, it's a factor we mustn't forget.

Differential thermal expansion: A change in temperature can cause a simultaneous increase or decrease in bolt tension and clamping force. If a decrease is indicated we'll want to add a term covering that loss to Equation 18.6 as well. The expression for this term would be

$$\Delta F_{\mathrm{th}} = \pm(1 - \Phi_{\mathrm{Kn}})K_{\mathrm{J}}'(\Delta L_{\mathrm{J}} - \Delta L_{\mathrm{B}})$$

If an increase is indicated we might be able to subtract the change from the terms within the brackets in Equation 18.6. We'd certainly want to include the increase on the left side of Equation 18.12, below.

18.4.7 Further Considerations

VDI adds some further considerations: things to check before we specify the size and material of the bolts to be used in our newly designed joint.

18.4.7.1 Static Strength of the Bolt

We don't want the stress in the bolts to exceed their yield strength, so:

$$\text{Max } F_{\mathrm{B}} = \text{Max } F_{\mathrm{J}} + \Delta F_{\mathrm{B}} < F_{\mathrm{y}} \tag{18.9a}$$

or

$$\text{Max } F_{\mathrm{R}} = \text{Max } F_{\mathrm{J}} + \Phi_{\mathrm{Kn}} L_{\mathrm{X}} \leq F_{\mathrm{y}} \tag{18.9b}$$

These equations define the true, maximum, in-service tension the bolts will see.

VDI says that another way of expressing this is by the following (Equation 18.7 justifies this alternate way to define the maximum bolt load):

$$\Delta F_{\mathrm{B}} \leq 0.1\sigma_{\mathrm{y}} A_{\mathrm{S}} \tag{18.10}$$

18.4.7.2 Fatigue

If the joint is loaded concentrically, and the load cycles between L_{X} and zero, we can express the allowable limits on the excursion in load mathematically by

$$\Delta F_B/2 = \Phi_{Kn} L_X/2 \leq \sigma_A A_r \tag{18.11}$$

We could, of course, substitute a specific fatigue life stress for the endurance limit stress if something less than infinite life were acceptable for the application.

If the load cycles between L_X and an L_{Xmin} which is greater than zero, then

$$\Phi_{Kn}(L_X - L_{Xmin})/2 \leq \sigma_A A_r \tag{18.12}$$

Or, for a push–pull load:

$$\Phi_{Kn}\left(L_{X\,tensile} - L_{X\,compressive}\right)/2 \leq \sigma_A A_r \tag{18.13}$$

See Table 2.11 for some endurance limit data if you have none of your own, or see the discussion in Section 15.6.

The presence of fatigue loading will usually have a major impact on the size of the bolt chosen, its material, and its design. As a partial response we might want to use a fatigue-resistant fastener or take some of the other steps described in Chapter 17.

18.4.7.3 Bearing Stress

VDI says that the bearing stress under the head of the bolt or between nut and joint should not exceed the boundary surface pressure of the joint material, or

$$0.9 \text{ Max } F_J \leq P_G A_P \tag{18.14}$$

where

P_G = boundary surface pressure (psi, MPa) and is usually slightly higher than the yield strength of the joint material

A_P = contact area under head or nut (in.^{2}, mm^2)

Why Max F_J instead of Max F_B here? As we say in Equation 18.9, Max $F_B >$ Max F_J. The tensile stress in the bolt increases under an external load, but the clamping force between joint members, and therefore the contact stress between bolt and joint, decreases because of this same load. Equation 18.14 acknowledges this by basing the limit on Max F_J, the highest bolt load created during assembly.

18.4.7.4 Shear Stress

If the bolt will also be loaded in shear, its tensile capacity must be reduced as illustrated in Figure 12.10. Now we must compare Max F_B not to the yield strength of the bolt, but to remaining tensile capacity of the bolt.

18.4.7.5 Bending Stress

In critical situations we might want to estimate any bending stresses to be seen by the bolt; perhaps using the VDI expression given in Equation 11.8. Tensile or combined stress limits would then be based upon the loads seen by the more highly stressed convex side of the bolt rather than on the average tensile stress.

18.4.7.6 Eccentric Loading

With eccentric loading and the resulting prying action, things get a little more complicated. The required force, which we have previously called F_{Prqd}, is now called F_{Krqd} and is the force required to prevent even partial separation of the joint. The equation for this force is

$$F_{\mathrm{Krqd}} = (a - s)u / \left(R_{\mathrm{G}}{}^{2} + su\right) \tag{18.15}$$

There are potential problems here. We're told that a can be statically indeterminate in some joints (e.g., for an automotive conrod) [3] and that s and R_G are not easy to quantify [1]. See Figure 11.12 for an illustration defining a, s, and u.

If the eccentric joint is subjected to cyclical, fatigue loading, then we must calculate a fatigue strength based upon the maximum fiber stress. We use Equation 11.8 to compute the change in fiber stress and use that change instead of ΔF_B in Equation 18.11.

18.4.8 Revised Bolt Specifications

At the beginning of this design procedure we tentatively selected a bolt size and material. The analysis which followed was intended to confirm or refute that choice. VDI tells us that we can now use Equations 18.6 through 18.15 to revise our choice of size and material, if revisions are indicated. Note that the new selection will be based, not just on gut feel or past experience, but on an impressive list of design considerations, including the following:

1. Yield strengths of the bolt and the joint material
2. The external loads to be placed on the joint (magnitude, static or dynamic, tensile or shear, or a combination)
3. Maximum fiber stress in the bolt under head or nut contact stress on joint
4. Expected relaxation (embedment and elastic interactions)
5. Tool scatter (which means that we have decided upon the type of tool and assembly procedure to be used)
6. The joint stiffness ratio or load factor, with a consideration of how the joint is going to be loaded (internally or externally; eccentrically or concentrically)
7. The amount of bolt strength which will be absorbed by torsional stress
8. The clamping force required to prevent slip, separation, or leakage

18.5 AN EXAMPLE

Let's try an example to see how we might use the VDI equations.

18.5.1 Inputs

We must first decide whether we want to assign maximum, minimum, or average values to the various design parameters, in effect facing the same decisions we dealt with in Section 17.7.3. My preference at that time was for average values, and that's what we'll use here. If this were a safety-related joint we'd also want to run the calculations for worst-case values. The VDI Directive 2230, incidentally, does not specifically address this issue, but that document contains several tables of data defining possible inputs; and I'm sure that these are all average (i.e., typical) values. Only tool scatter is treated as a statistical variable by VDI; input loads may also be treated as variable in fatigue situations, but always with the implication that the max or min loads are known. In any event—here are the average inputs we're going to use in this example.

Let's assume that we have roughed out the design of our product and have need for a joint which will be subjected to a concentric, per-bolt tensile load which will cycle between 3,000 and 4,000 lbs. We'd like the minimum clamping force to be at least 1,000 lbs per bolt, to prevent joint separation. Based on past experience we've decided to use ½–13 UNC SAE J429 Grade 8 bolts. Joint material will be a low-carbon steel. We've computed the load factor, using the procedure in Chapter 10, and estimate it to be 0.2. The equipment will be assembled using manual torque wrenches, and it will be used at room temperatures only. Here, then, are our inputs:

A_S	$= 0.1419$ in.^{2} (from Appendix E)
ΔF_P	$= 28\%$ of F_P (10% from embedment plus 18% average relaxation through elastic interaction)
L_X	$= 4{,}000$ lbs
$L_{X\min}$	$= 3{,}000$ lbs
s	$= \pm 30\%$
Φ_{Kn}	$= 0.2$
σ_A	$= 18$ ksi (from Table 2.11)
σ_y of bolts	$= 130$ ksi (from Table 2.1)
σ_y of joint material	$= 34$ ksi (from Table 2.16)

18.5.2 Calculations

We start with some preliminary computations

$$\alpha_A = (1 + s)/(1 - s) = 1.86$$

$$F_y = \sigma_y A_s = 130{,}000(0.1419) = 18{,}447 \text{ lbs}$$

ΔF_P: To compute ΔF_P we must assume or estimate an F_P; let's say that will be 50% of the yield strength of the bolt, or

$$F_P = 0.5F_y = 0.5(18{,}447) = 9{,}224 \text{ lbs}$$

Therefore $$F_P = 0.28F_P = 2{,}583 \text{ lbs}$$

$$\Delta F_J = (1 - \Phi_{Kn})\ L_X = 0.8(4{,}000) = 3{,}200 \text{ lbs [from Equation 18.3]}$$

18.5.2.1 Maximum and Minimum Assembly Preloads

Now we're ready to compute min and max assembly preloads (Equations 18.5 and 18.6)

$$\text{Min } F_J = \left(\Delta F_J + \Delta F_{Prqd} + \Delta F_P\right) = (3{,}200 + 1{,}000 + 2{,}583) = 6{,}783 \text{ lbs}$$

$$\text{Max } F_J = \alpha_A \text{Min } F_J = 1.86(6{,}783) = 12{,}616 \text{ lbs}$$

18.5.2.2 Static Strength of the Bolts

Are the bolts strong enough? From Equation 18.7

$$0.9F_y = 0.9(18{,}447) = 16{,}602 \text{ lbs}$$

This is greater than Max F_J, so the latter seems OK, but a second test is required.

From Equations 18.3 and 18.6:

$$\Delta F_B = \Phi_{Kn} L_X = 0.2(4{,}000) = 800 \text{ lbs}$$

$$\text{Max } F_J + \Delta F_B = 12{,}616 + 800 = 13{,}416 \text{ lbs}$$

That's only 73% of the yield strength (F_y) of the bolt, leaving a 13% safety factor. Our choice of a ½ in Grade 8 bolt may not be optimum, but it certainly appears to be acceptable.

18.5.2.3 Fatigue Strength

Next we analyze the fatigue situation, using Equation 18.9

$$\Phi_{Kn}(L_X - L_{Xmin})/2 \leq \sigma_A$$

$$0.2(4000 - 3000)/2 = 100 \text{ psi}$$

which is well below the endurance limit of 18 ksi, so our choice of bolt still seems acceptable.

18.5.2.4 Contact Stress

Finally, we'll check the contact stress between bolt head and joint. We obtain a sample of the bolt and measure the across-flats distance of its hex head to be 0.75 in. The bolt, of course, has a nominal diameter of 0.5 in. We use the familiar $\pi(D^2/4)$ to compute the areas described by these diameters; then we compute the difference between them.
Contact area $A_P = A_{flats} - A_{diam}$

$$A_P = 0.442 - 0.196 = 0.246 \text{ in.}^2$$

The yield strength of the joint material is 30 ksi, so we'll call that the maximum allowable contact pressure, P_G [from Equation 18.14]

$$A_P P_G = 0.246(34{,}000) = 8{,}364 \text{ lbs}$$

$$0.9 \text{ Max } F_J = 0.9(12{,}616) = 11{,}354 \text{ lbs}$$

$A_P P_G$ is less than 0.9 Max F_J and so the contact stress exceeds the limits imposed by Equation 18.14. We could respond by specifying less preload, but that should be avoided if possible because it would mean a minimum clamping force of less than 1,000 lbs per bolt. We could also specify a stronger joint material, but that sounds expensive. (I'm assuming that our other joint stress computations say that the material is acceptable.) The simplest thing to do is to use washers under the bolt head and nut to reduce contact stress.

A quick calculation shows that a 1 in. diameter washer would raise $A_P P_G$ to 20,040 lbs, well above the 0.9 Max F_J value of 11,354 lbs.

No gasket or temperature change or shear loads or eccentric loads are involved here, so we're done. As you can see, the procedure is easy to use. But estimating the values to be used for the input data—loads and things—can require some effort and some engineering judgment.

18.6 OTHER FACTORS TO CONSIDER WHEN DESIGNING A JOINT

This concludes the discussion of the VDI procedure. There are, however, many other factors we'll often want to consider when designing a joint. These include the following, in no particular order.

18.6.1 Thread Strength

We want the body of the bolt to break before the threads strip. That will be the case if we've selected standard bolts and nuts, but we'll want to use the thread strength equations of

Chapter 4 to check length of engagement etc. if we're planning to use the bolts in tapped holes, especially in soft materials. For example, one rule of thumb I've heard suggests that the length of engagement should be at least two times the bolt diameter if a steel bolt is to be used in a hole tapped in an aluminum joint.

18.6.2 Flexible Bolts

Although some say use of stronger bolts with the resulting higher preload is more important than flexibility, conventional wisdom says that we'll usually prefer flexible bolts. They are much better energy storage devices and are, therefore, less sensitive to thermal change, vibration, embedment or other relaxation loss, etc. Although there are no absolutes in bolting, previous chapters have included statements like this: "If the bolt's length to diameter ratio is 8:1 or more, it will never self-loosen" or "We always want the joint-to-bolt stiffness ratio to be 10:1 or greater." In general, we want to avoid such poor energy storage parts as short, stiff bolts; composite or soft joint materials; etc.

18.6.3 Accessibility

We want a minimum of 60° in which to swing a wrench, and would like to be able to see the bolts. Remember the guy who wrote that "the preload in most bolts in this world is directly proportional to their accessibility." Try to put yourself in the mechanic's shoes when locating those bolts.

18.6.4 Shear versus Tensile Loads

In general, shear joints are subjected to fewer failure modes than are tensile joints. If you have a choice, therefore, you'd be well advised to connect your parts with well-designed shear joints rather than with tension joints. See Chapter 19 for more on shear joint design.

18.6.5 Load Magnifiers

We want to avoid things which magnify the loads seen by the bolts. The most common sources of this problem are prying, as illustrated in Figure 11.4, and eccentric shear loads, which will be discussed in Chapter 19. (See Figure 19.8.)

18.6.6 Minimizing Embedment

We can minimize embedment relaxation by chamfering holes, by insisting on flat and parallel joint surfaces, by specifying that holes should be drilled perpendicular to joint surfaces, or by specifying hard washers.

18.6.7 Differential Expansion

Differential expansion can create a significant and simultaneous increase—or decrease—in both bolt tension and the clamping force of the joint. We can at least reduce the effects by using bolts and joint members made of materials having similar coefficients of thermal expansion. It helps if bolts and joint members are exposed to the same changes in temperature, though this is often difficult to achieve. Using bolts with a generous length-to-diameter ratio, perhaps with the help of Belleville springs or cylindrical collars, can be a big help, too.

18.6.8 Other Stresses in Joint Members

The general-purpose VDI procedure doesn't cover all of the stresses a joint might be exposed to, of course, so we must be careful not to overlook them. Pressure vessel and piping designers, for example, estimate the stresses in flange fillets. Studies have been made of ways to combat bending stresses placed on certain types of aerospace joints [4]. And the PVRC is trying to quantify the effects of bending moments on gasketed, flanged joints. Excess stress can cause shear joints to tear out, etc. These are only a few examples, but presumably they'll give you the message, "Don't base your designs solely on the VDI procedure."

18.6.9 Locking Devices

If the joints are to be exposed to extreme shock or vibration, then we should consider using one of the locking devices or fasteners described in Chapter 14. It would also help to design the joints so that the axes of the bolts are more or less parallel to the vibratory or shock loads.

18.6.10 Hole Interference

We want to avoid inadvertent interference between the bolts and clearance holes in joint members, by careful dimensioning of hole locations, sizes, etc. because hole interference can absorb a substantial portion of assembly preload. If interference is desired or unavoidable we should specify that the bolts be forced through the holes by other means before being tightened.

18.6.11 Safety Factors

Many of the values we'll assign to design parameters will be subject to considerable variation, based on an outright guess, or both. The safety and reliability of our designs, therefore, can be enhanced by the judicial use of safety factors, applied either to the individual values we use or to final results. For example, if the analysis suggests that a ½ in. diameter bolt will do the job we might well decide to use a ⅝ in. or ¾ in. one to cover the uncertainties.

18.6.12 Selecting a Torque to Be Used at Assembly

VDI doesn't tell us, specifically, how to pick an assembly torque, but they give us the necessary information to do this. The target preload would be the midpoint between Max F_J and Min F_J as defined in Equations 18.5 and 18.6. Averaging these gives us

$$F_{Pa} = (1 + \alpha_A)(\text{Min } F_J)/2 \tag{18.16}$$

We'd then use a suitable nut factor (K, from Table 7.1) and the short-form, torque-preload equation to compute a torque value

$$T = KDF_{Pa}$$

where
- T = torque (in.-lb, mm-N)
- K = nut factor (dimensionless); typically 0.2 for as-received steel
- D = nominal diameter of the fastener (in., mm)
- F_{Pa} = preload to be used at assembly (lb, N)

Continuing the example of Section 18.5:

$$F_{Pa} = (1 + 1.86)(6783)/2 = 9700 \text{ lbs}$$

$$T = 0.2(0.5)9700 = 970 \text{ in.-lb}$$

And that's it! If we do all of the things described in this section and in Section 18.4, we'll end up with better-designed bolted joints than most of those which are already out there. Who could ask for anything more?

EXERCISES

This chapter is basically a recapitulation of most of the previous chapters, so the answers to the following questions can be found throughout the book and/or in this chapter.

1. What is the primary role of the bolts in a bolted joint?
2. Why do many designers of safety related joints, and most codes, favor joints that could support four or more times the anticipated service loads?
3. You have roughed out the design of a joint and believe that four, ⅝–12 × 3 SAE J429 Grade 5 bolts will provide sufficient clamping force if tightened to 50% of yield. How much preload will this create?
4. The bolts will be lubricated with moly paste, for better control of preload, and will be tightened with a torque wrench. What torque will you specify?
5. What range of preload would you expect to encounter when that torque is applied to a production quantity of this joint; assuming perfect tools and operators?
6. What might that range be with imperfect (i.e. normal) tools and operators?
7. Does the estimated maximum preload threaten your bolts?
8. You guess, since this is a conventional or typical joint, that the joint to bolt stiffness ratio is 5:1. You believe that the tensile loads applied to this joint in service will range from 3000 to 4000 lbs. What range of bolt tension (preload) and clamping force do you now anticipate for your joint in production?
9. As a percentage of yield what's the minimum and maximum tensile stress to be seen by a production quantity of these bolts?
10. With the estimated minimum in-service tensile stress in mind, what possible bolt or joint failure modes must you consider, and under what service conditions?
11. What are some obvious things you can do if you are concerned about one or more of these failure modes?

REFERENCES

1. Junker, G.H., Principle of the calculation of high-duty bolted joints: interpretation of Directive VDI 2230, published by SPS Technologies, Jenkintown, PA, no date given (but probably about 1978).
2. VDI 2230: Systematic Calculation of High Duty Bolted Joints, translated from the 1977 German text by Language Services, Knoxville, TN, reproduced by the U.S. Department of Energy, Office of Scientific and Technical Information, Oak Ridge, TN, no date given.
3. McNeill, W., A. Heston, and J. Schuetz, A study of factors entering into the calculation for connecting rod joints according to the VDI 2230 guidelines, VDI Berichte No. 766, 1989.
4. Baumann, T.R., Designing safer prestressed joints, *Machine Design*, pp. 39ff, April 25, 1991.
5. Eccles, W., Bolted joint design, *Engineering Designer* (UK), pp. 10ff, November 1984.

BIBLIOGRAPHY

You might find the following works useful, in addition to the references cited at the end of this and the next chapter.

A. General (Includes Auto, Aero, and Miscellaneous Joints)

Blake, A., *Design of Mechanical Joints*, Marcel Dekker, New York, 1985.

Joint design, Prepared by the Technical Information Staff of the Industrial Fastener Institute, *Machine Design*, p. 12, September 11, 1969.

Junker, G.H., Principle of the calculation of high-duty bolted joints, SPS, Jenkins town, PA.

Osgood, C.C., How elasticity influences bolted joint design—part 1, *Machine Design*, p. 92, February 24, 1972.

Osgood, C.C., How elasticity influences bolted joint design—part 2, *Machine Design*, p. 104, March 9, 1972.

Osgood, C.C., *Fatigue Design*, Wiley-Interscience, New York, 1970.

Shigley, J.E., *Mechanical Engineering Design*, McGraw-Hill, New York, 1977.

Shigley, J.E. and Charles R.M., (Eds.), *Standard Handbook of Machine Design*, McGraw-Hill, New York, 1986.

Systematische Berechnung hochbeanspruchter Schraubenverbindungen, Verein Deutscher Ingenieure (VDI), Berlin, West Germany, October 1977.

B. Pressure Vessel and Piping Joints

ASME Boiler and Pressure Vessel Code, ASME, New York, 1983. Section III—Division 1, Subsection NB-Design of Class 1 components (those contained in the reactor coolant boundary). See especially sections NB3231 and Appendix E.

Section III—Division 1, Subsection NC—Class 2 components (pressure retaining). See especially Appendix XI.

Section III—Division 1, Subsection NF (component supports). See especially sections NF 3324.6 and NF 3332.5.

Section VIII—Division 1 (pressure vessels). See especially Appendix 2 and Appendix Y.

Section VIII—Division 2 (design of pressure vessels "by analysis" instead of "by rules"). See especially Appendix 3.

Blach, A.E., A. Bazergui, and R. Baldur, Bolted flanged connections with full faced gaskets, *WRC Bulletin* 314, Welding Research Council, New York, May 1986.

Petrie, E.C., The ring joint; its relative merit and application, *Heating*, *Piping and Air Conditioning*, vol. 9, no. 4, April 1937.

Reddy, M.D. and B.S. Nau, Factors affecting the design of large flanged joints and joint seals for chemical plant, British Hydromechanics Research Association (BHRA), Cranfield, Bedford, England, December 1977.

Rodabaugh, E.C., F.M. O'Hara, Jr., and S.E. Moore, FLANGE, a computer program for analysis of flanged joints with ring-type gaskets, U.S. Department of Commerce, National Technical Information Service, Springfield VA, January 1976.

Rodabaugh, E.C. and S.E. Moore, Evaluation of the bolting and flanges of ANSI B16.5 flanged joints—ASME Part A Design Rules, ORNL/Sub/2913–3, NRC-5, PP 115–7h, Oak Ridge National Laboratory, Oak Ridge, TN, September 30, 1976.

Wesstrom, D.B. and S.E. Bergh, Effect of internal pressure on stresses and strains in bolted-flanged connections, *ASME Transactions*, p. 121, 1951.

Webjorn, J., The bolted joint, a series of problems, Dissertation No. 130, Institute of Technology, Linkoping, Sweden, 1985.

C. Structural Steel Joints

Bibliography on Bolted and Riveted Joints, American Society of Civil Engineers, New York, 1967.

Frank, K.H. and J.A. Yura, An experimental study of bolted shear connections, Report FHWA/RD-81–148, U.S. Department of Transportation, Federal Highway Administration, Published by the National Technical Information Service, Springfield, VA, December 1981.

Kulak, G.L., John W.F., and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, 2nd ed., Wiley, New York, 1987.

Manual of Steel Construction, 8th ed., American Institute of Steel Construction (AISC), Chicago, 1980.

Specification for structural steel joints using ASTM A325 or A490 bolts, American Institute of Steel Construction (AISC), Chicago, November 13, 1985. (Note: This specification is now available in two versions, one describing allowable stress design procedures, the other load and resistance factor design procedures.)

Merritt, F.S., (Ed.), *Structural Steel Designers' Handbook*, McGraw-Hill, New York, 1972.

19 Design of Joints Loaded in Shear

19.1 AN OVERVIEW

Eighteen chapters ago I described two kinds of bolted joints: those loaded in tension and those loaded in shear. With the exception of Chapter 12, the discussion since then has been focused on tension joints because they're more common, their behavior is more complex, and analyzing them is more difficult. In this last chapter, however, we're going to take another look at the shear joint. To be specific, we'll study the design of such joints and will see the ways in which the design process is the same as that for tension joints, and the many ways in which the two differ.

There are many different types of shear joint, but most can be defined as either a lap joint or a butt joint, as shown in Figure 19.1. Historically, joints of either type were further classified as "friction type" or "bearing type." The structural steel industry has now abandoned the friction and bearing classifications, as we'll see, but the distinction is still handy for a preliminary review of shear joint design, and so I'll continue to use it.

Shear joints are most commonly encountered in structures, such as airframes, buildings, and bridges. Most of the bolted joints found in structures, in fact, are shear joints. In part this is because of the way loads are applied to structures, but I suspect that part of the reason is that shear joints are more forgiving of assembly errors or preload scatter; they can operate successfully under a much wider range of clamping force than can many tension joints. One of the main reasons for this is that shear loads don't change bolt tension or clamping force the way tension loads do.

One new problem we do have to be concerned about, however, when dealing with shear joints, is the possible mechanical failure of the joint members themselves. Tension joint failure can usually be blamed on the bolts; either they have created the wrong clamping force or they have themselves failed. Improper clamp can cause a shear joint to loosen under vibration, but most shear joint failures involve the rupture of the joint members.

We're going to start our study of shear joint design with our old friends, the VDI equations. We'll see what they have to teach us about shear joints, and will find that it's useful but not enough. So we'll go on to look at the way the bolts and joint members see and resist shear loads. All of this will be pertinent for the design of shear joints in general. As we go along we'll take an occasional look at some of the codified design procedures, which have been developed by the structural steel industry. Those procedures are described and explained in detail in the definitive text, *Guide to the Design of Bolted and Riveted Joints* by Kulak et al. [7]. This complete text is now available, free, on the Research Council on Structural Connections (RCSC) Web site. Structural steel designers should rely on this text and on the bolt specs also written by the RCSC [8,13] rather than on my text. The bolt spec is also available on the RCSC and AISC Web sites. I'm not going to repeat the procedures described in those documents here. But some comments are certainly in order.

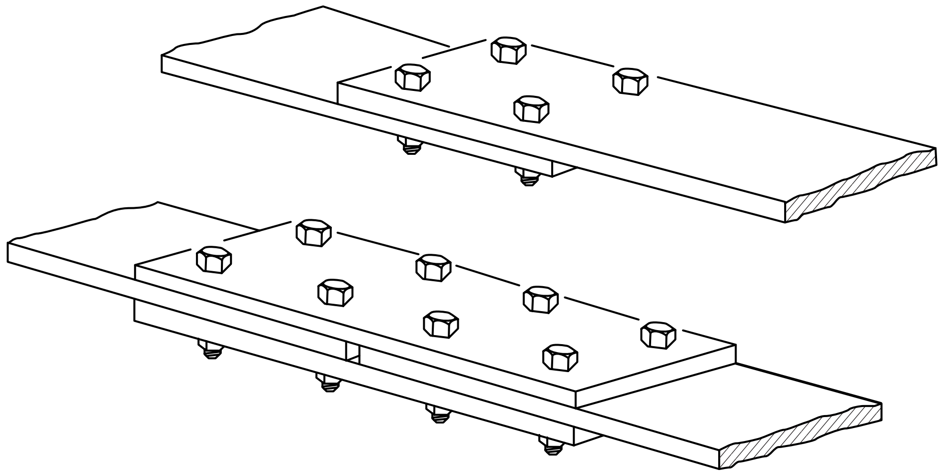


FIGURE 19.1 Two basic types of shear joint. The upper is called a lap joint; two joint members are bolted to each other. The lower is called a butt joint; the joint members are connected by upper and lower splice plates.

19.2 THE VDI PROCEDURE APPLIED TO SHEAR JOINTS

Let's return, for a final time, to the VDI joint design equations first encountered in Chapter 18 [11]. These raise two important issues we must address when designing any joint: the minimum clamping force we can expect to see in the joint and the maximum tension the bolts will have to support. Since clamping force is assumed to equal bolt tension, these equations are expressed in terms of clamp force (F_J) but really define bolt preload limits. I repeat those equations here, for your convenience.

$$\text{Min } F_J = \Delta F_J + F_{Prqd} + \Delta F_P \tag{18.5}$$

Since $\Delta F_J = (1 - \Phi_{Kn})L_X$, this can also be expressed as

$$\text{Min } F_J = (1 - \Phi_{Kn})L_X + F_{Prqd} + \Delta F_P$$

We also have

$$\text{Max } F_J = \alpha_A[(1 - \Phi_{Kn})L_X + F_{Prqd} + \Delta F_P] \tag{18.6}$$

where

ΔF_P = loss of preload during or soon after assembly (lb, N)
Max F_J = maximum anticipated bolt tension created during assembly; equals the clamping force that bolt applies to the joint before the joint is put in service (lb, N)
Min F_J = minimum anticipated bolt tension and clamping force created during assembly, before the joint is put in service (lb, N)
F_{Prqd} = minimum preload (and per-bolt clamping force) required to prevent joint failure (lb, N)
L_X = external tensile load applied to the joint (lb, N)
α_A = the scatter in preload caused by the assembly tools and procedures
α_A = Max F_P/Min F_P
Φ_{Kn} = load factor for a concentric joint, loaded internally at loading planes

$\Phi_{Kn} = \Delta F_B/L_x$ where $\Delta F_B =$ the change in tension created in the bolts by the external load This is the load factor we used when discussing tension joints in general

$\Delta F_J =$ the change in clamping force (or bolt tension) created by an external tensile load on the joint (lb, N)

Let's look at each of these terms and see how they apply to shear joints.

ΔF_J: Since there is no tensile load on a joint loaded only in shear, $\Delta F_J = 0$.

ΔF_P: When designing a tensile joint we had to consider four ways in which initial preload might be lost during or after assembly, giving us this expression:

$$\Delta F_P = \Delta F_{em} + \Delta F_{El} + \Delta F_{CR} \pm \Delta F_{TH} \tag{18.1}$$

where the subscripts

em = embedment relaxation

El = elastic interaction loss

CR = creep loss

TH = gain or loss because of thermally induced differential expansion.

Each of these effects can cause a loss of assembly preload in a shear joint as well as in a tension joint, but the effects are usually smaller, with the exception of embedment. Let's look at each effect as it applies to a shear joint.

Embedment: We can expect to see a typical embedment loss, perhaps 5%–10%.

Elastic interactions: Bibel tells us that the average elastic interaction loss in an ungasketed, metal-on-metal joint is 18%, well below the values for gasketed joints [10]. This figure, however, is based on limited tests on 24 in. diameter, raised-face, pressure vessel joints, where the bolts are unsupported by metal-to-metal contact. I would expect to see less loss in most shear joints: but that's just a guess.

Creep: Some structural steel joints are given a thin coat of paint, to control interface friction and to provide some corrosion protection, and so we might see some creep loss after assembly. I would expect the loss to be negligible, however, certainly nothing like the major loss created by a gasket.

Thermal: In most of the (few) structural steel joints I've studied the bolts and joint members have both been made of steel, presumably with similar coefficients of expansion. Unlike, say, a pressure vessel joint, both bolts and joint members would experience the same change in temperature at the same time. And the temperature change would be modest, again unlike pressure vessel applications, where temperatures of 1000°F or more are not uncommon. All of which suggests that differential expansion can be ignored in structural steel joints.

Airframe structures, on the other hand, often involve several materials, including aluminum joint members and bolts of ferrous metals or exotic alloys. Temperature changes, however, are still modest. Grip lengths, furthermore, tend to be small in a structure which must be light enough to fly. I'm sure that airframe designers take thermal change into account, but I doubt if differential expansion is large enough to be a concern. All of which suggests that in most shear joints, ΔF_P is probably going to be less than 30% of initial preload, perhaps much less.

α_A: Shear joint bolts will be subject to the same preload scatter as tension joint bolts. If a torque wrench is used on unlubricated bolts, for example, scatter might be $\pm$30%. Following VDI's lead, this would create an α_A of $(1+0.3)/(1-0.3)$ or 1.86. There are some factors that make tool scatter different when we're dealing with a shear joint, however. First of all, preload control is less important in most shear joints, so scatter doesn't matter as much. Second, in structural steel work, at least, the bolts are often tightened well past yield, on purpose, and this reduces scatter to $\pm 5\%$ or so [1] and an α_A of only 1.11. (VDI says that yield control will result in an $\alpha_A = 1.0$, as you'll see in Table 18.1, but I think that that's overoptimistic.)

F_{Prqd}: This is the force required to guarantee satisfactory operation of the joint, and it will dominate the selection of the bolts when we design a shear joint. We'll consider it in some detail in the next section.

Final equations: As a result of all this we end up with the following VDI equations when we tackle a typical shear joint.

$$\text{Min } F_J = 1.3F_{\text{Prqd}} \tag{19.1}$$

$$\text{Max } F_J = \alpha_A \text{Min } F_J \tag{19.2}$$

The 1.3 comes from my conclusion that we need to create an initial clamping force 30% greater than the in-service force required to accommodate elastic interactions and embedment. I suspect that that's conservative, but I'll use it for this example.

Do we now have what we need to design a shear joint? Hardly. The VDI equations allow us to define the bolts required in a joint. This information helps us size the joint members and pick joint materials, but it isn't sufficient to configure a joint. Most configurations will be based on past experience or custom in a particular application, and shear joints are no exception to the rule. Most structural steel joints are designed "to code," and I'm sure that most airframe joints are equally well defined by past experience. Both are supported, furthermore, by theoretical studies, exhaustive tests, and by analysis of past failures. The VDI equations give us a first cut only at one important input: bolt tensions.

As Equations 19.1 and 19.2 show us, furthermore, we can't even define the bolts until we determine F_{Prqd}. Just as the leakage properties of a gasket determined F_{Prqd} and therefore dominated the design in pressure vessel applications, so does something other than the bolt dominate the design of a shear joint. We'll take a close look, therefore, at F_{Prqd}. First, however, let's look at the way a shear joint resists being torn apart by applied shear loads.

19.3 HOW SHEAR JOINTS RESIST SHEAR LOADS

19.3.1 In General

Shear joints resist applied loads in two ways. First, the interface clamping force generated by the bolts creates friction forces which resist joint slip. Second, the bolts act as shear pins to prevent slip. Because of this, structural steel joints used to be divided into two categories: friction type and bearing type, and shear joints in other industries were presumably seen in the same light. As discussed at length in Chapter 12, however, the structural steel industry no longer considers friction type and bearing type to be valid models for building or bridge joints. Instead, they now classify shear joints as "slip-critical" or "not slip-critical." To some extent their concern would apply to shear joints in other industries as well. At the risk of some redundancy, here's what they say.

19.3.2 Concept of Slip-Critical Joints

Why are friction-type and bearing-type joints no longer considered valid models for structural steel joints, and what has replaced them?

Steel erectors must work with large, cumbersome, relatively crude components (compared to those found on an assembly line, for example). Holes in joint members must be oversized or slotted to compensate for misalignment and to make it possible to insert the bolts. In virtually every joint, whether it's intended to develop its strength-through-friction or strength-through-bearing, some bolts are "in bearing" from the start. Pure friction-type, no-hole-bolt-contact joints are almost nonexistent.

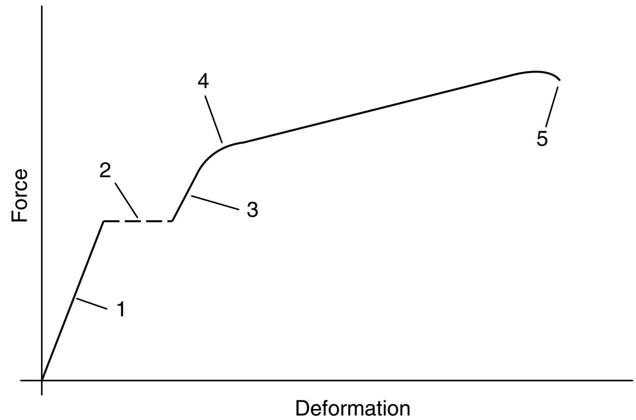


FIGURE 19.2 This graph illustrates the way in which a joint subjected to ever increasing shear loads will fail. First (1), joint members, splice plates, and perhaps bolts will deform elastically; assuming that all are locked together by interface friction. Second (2), joint members and splice plates slip past one another until the bolts are brought into bearing. Next (3), there's additional elastic deformation of the parts. This is followed (4) by plastic yielding of bolts and joint members. Finally (5), something breaks.

Every joint loaded in shear, furthermore, derives some of its initial strength from friction between the joint members, even if the bolts were only snugged. Furthermore, every joint under unidirectional axial shear loads fails by first slipping into bearing. Final failure of such a joint is by shear of bolts and joint members, and the loads required for such failure always exceed the loads required to overcome any friction forces between faying surfaces, even if the bolts were tightened past yield. This is illustrated in Figure 19.2 (Ref. [3], p. 89; Ref. [1], p. 94; Ref. [7]).

Tests suggest, furthermore, that the shear load at which the joint finally fails is independent of the initial preload in the bolts and of the coefficient of friction between faying surfaces (Ref. [8], p. 34).

As a result of all this, structural steel designers currently use the shear and bearing strengths of bolt and joint members, rather than the friction forces, to estimate the ultimate strength of all joints. They design for friction-dependent, no-slip behavior only in applications which the engineer of record considers to be slip-critical.

One such situation, for example, could be a joint where the bolt holes are slots and the shear loads on the joint are parallel to the axis of the slots. Since the slots would allow significant motion, joint slip could result in dangerous geometric distortions of the structure, leading to failure even if the joints themselves did not fail in shear or bearing (Ref. [8], p. 38).

Joints subjected to significant load reversals are also considered slip-critical, as are some joints under fatigue loading. Again, it's up to the engineer of record to specify which joints are slip-critical, and should, as a result, be heavily preloaded. (Structural joints loaded in tension are also heavily preloaded, to make sure that each bolt carries a share of the load.)

So, joints whose integrity depends on resistance to slip are still designed, even though they are no longer referred to as friction-type joints. And, to repeat an earlier suggestion, shear joints in manufactured products, where close hole tolerances and alignment are economically feasible, could presumably be designed for friction or bearing strength.

The concept of slip-critical joints was first codified in the November 13, 1985 version of the AISC Specification for Structural Steel Joints Using A325 and A490 Bolts. This was one of several significant differences between the 1985 edition of this specification and earlier editions. After completing the 1985 specification, which defines allowable stress design procedures, the authors (The Research Council for Structural Connections) developed an optional, alternate load and resistance factor design specification—again for A325 and A490 bolts. This so-called LRFD specification was approved for publication on June 8, 1988. People involved in structural steel work should obtain and study these two documents. They are significant departures from previous editions of this important specification. Slip-critical joints, and other contemporary topics, are also covered in Kulak et al. [7], an updated version of Fisher and Struik [1].

Although the definition of slip-critical comes from the structural steel industry, the concept is valid for most if not all shear joints. If any amount of slip will place the structure at risk, the designer must specify enough bolting to create sufficient friction to prevent slip. If some slip is acceptable, he can assume and design for strength-through-bearing. There's also a third option, which combines zero slip with strength-through-bearing. Airframe designers frequently specify interference fit bolt holes. The bolts are forced through the holes by an insertion tool of some sort before the nuts are tightened. Slip is prevented by zero bolt-to-hole clearance. It's difficult to create a known amount of interface clamping force under these conditions, however, as illustrated in Figures 6.6 through 6.8, the joints must be designed so to resist applied loads in bearing. Interference fit holes are possible in structural steel, too, using special bolts which have ribs running parallel to the axis of the bolts, but designers in other industries are more likely to take advantage of this combination of zero slip and bearing strength. One of the attractions of using interference fit holes, I'm told, is that it increases the fatigue resistance of joint members.

When we apply the VDI equations to shear joints we see that F_{Prqd}, the clamping force required to prevent joint failure, can be a key issue. In order to quantify F_{Prqd} we need to relate bolt tension to joint strength. The relationship will depend, obviously, on whether we're designing for strength-through-friction or strength-through-bearing. We're going to look at some joint strength details in the next two sections of this chapter and will see that they do indeed define the F_{Prqd} requirements.

19.4 STRENGTH OF FRICTION-TYPE JOINTS

19.4.1 In General

Frictional resistance to joint slip is created by the normal force which clamps the joint members together. and this force is created by the bolts. In the structural steel world, this frictional force is called the "slip resistance" of the joint (R_s) and we'll use the same terminology.

The slip resistance is also a function of the number of slip surfaces (M) involved in the joint (see Figure 19.3). Three surfaces produce three times as much slip resistance as one

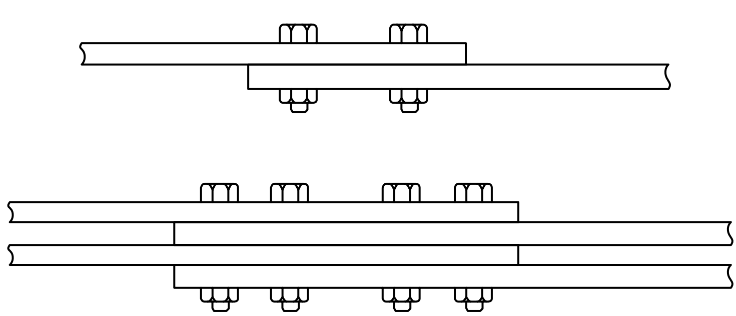


FIGURE 19.3 Shear joints having one and three slip surfaces.

surface, and so on. The equation for slip resistance, therefore, is (see p. 71 in Ref. [1] or p. 75 in Ref. [7])

$$R_S = \mu_S F_P N M \tag{19.3}$$

where
R_S = slip resistance of the joint (lb, N)
μ_S = slip coefficient of the joint
F_P = preload per bolt (lb, N)
N = number of bolts holding the joint together
M = number of slip surfaces (see Figure 19.3)

This means that F_P in Equation 19.3 is our friend F_{Prqd}. The slip resistance of the joint, of course, R_S, must be greater than the maximum external load which the joint will have to resist in service. By definition F_{Prqd} is the minimum bolt load required for a successful joint, which, in this case means to prevent slip.

19.4.2 Allowable Stress Procedure

We can use Equations 19.1 through 19.3 to design a friction-type shear joint. We'll look at an example in a minute. But first: I said in the introduction to this chapter that we'd take an occasional look at the codified design procedures developed by and for the structural steel industry. If we were designing a shear joint for a bridge or building we wouldn't use Equations 19.1 through 19.3. Instead we'd use equations found in Kulak et al. [7] or in the AISC bolt specifications [8,13]. One of these specifications presents what is called the allowable stress procedure [8]. Using that document, we find that the allowable slip load which can be placed on a slip-critical joint is defined as

$$L_{Xmax} = F_s A_b N M \tag{19.4}$$

where
L_{Xmax} = the maximum load which can be placed on the joint (lb, N)
A_b = nominal body area of a bolt (in.^{2}, mm^2)
N = number of bolts on one side of the joint
M = number of slip planes
F_s = allowable slip load per unit of bolt area (psi, MPa)

A table of allowed values of F_s is given in the reference. These values range from a low of 10 ksi for use with joints having slip coefficients of 0.33, and which contain A325 bolts mounted in long slots parallel to the direction of the applied load, to a maximum of 34 ksi for joints having slip coefficients of 0.50, and which contain A490 bolts mounted in standard holes.

This equation does not include any reference to bolt preload, which might seem to be a strange omission. But it's only one of many provisions in the bolt specification of that document. In Table 4 of that document we'll find specifications for the minimum tensions which must be created in bolts of various sizes during assembly, and elsewhere in the same document we'll even find instructions concerning the way in which the bolts should be tightened. Only when we combine these and other instructions from Ref. [8] and specifications with Equation 19.4 will we see a complete picture and have an acceptable, allowable stress design.

19.4.3 Other Factors to Consider

Equations 19.1 through 19.3 seem to define everything required to prevent the failure of a general-purpose, friction-type joint. But, as usual, there some other factors to be considered—including some substantial uncertainties.

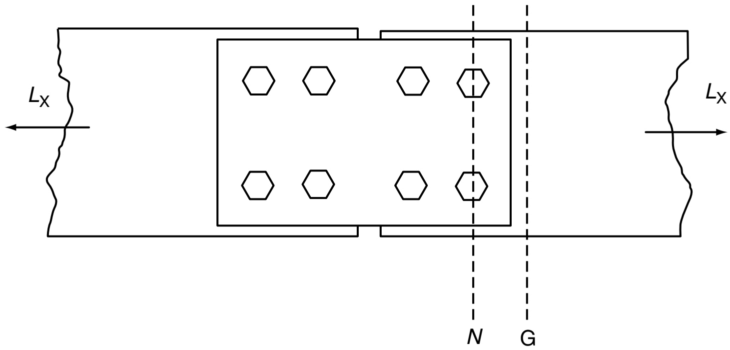


FIGURE 19.4 A plane passing through a row of bolt holes defines what is called the net section of the joint. A plane passing through an uninterrupted section of the joint plate defines the gross section. See also Figure 13.1. Note that an external load L_X whose line of action passes through the geometric center of a symmetrical bolt pattern such as this is called an axial shear load.

One factor is the strength of the joint member itself. It's certainly possible to design a joint in which the friction forces will exceed the tensile strength of the joint. That's also easy to avoid. Cyclic loads on a friction joint, however, can cause it to suffer a fatigue failure through the gross cross section as illustrated in Figure 19.4, and that's much less easy to predict. Structural steel and airframe industries spend a lot of time and money doing joint fatigue tests, and other industries which must deal with cyclic shear loads will have to do the same. Standard procedures for designing structural steel joints subject to fatigue loading can be found in Refs. [7], [8], and [13].

As far as uncertainties are concerned, it will often be difficult to predict the applied load. In critical situations we may have to build, instrument, and test a prototype or model. Finite-element analysis is also becoming more popular as a way to do this, but the FEA model should be confirmed by physical tests whenever possible. As far as structural steel is concerned, the cited references deal with these uncertainties in a couple of different ways. More about this in Section 19.7.

In most situations it will also be difficult—perhaps impossible is a better word—to predict or control the coefficient of friction between joint members. Laboratory tests can give us a clue, but real-world conditions will affect the friction between joint members as much as they do between nut and bolt or between nut and joint member. The $\pm 30\%$ scatter in preload we expect to see when we tighten a group of as-received bolts with a given torque will probably be duplicated by a $\pm 30\%$ variation in the slip resistance in a group of joints clamped together by a given amount of preload. To my knowledge, only the structural steel industry has published slip coefficient data. Let's take a quick look at their findings. They won't apply directly to shear joints in other industries, but they will illustrate the concern and raise issues which must be faced whenever we deal with shear joints.

19.4.4 Slip Coefficients in Structural Steel

The slip resistance of the joint is proportional to the coefficient of friction in the joint, the so-called slip coefficient, so surface treatment of the joint members is very important. A number of different treatments have been studied experimentally. The current practice is to allow the use of any coating which has been tested by procedures specified in Ref. [8], and which has been so certified.

Note that the treatment of the faying surfaces and coating, and not just the type of coating, affects the slip coefficient of the joint. For example, a hot-dip galvanized coating has a friction coefficient of 0.18 (average). If that same coating is wire-brushed or grit-blasted, however, the average coefficient becomes 0.4 (Ref. [7], p. 212).

Note, too, that generic classifications for coatings are not possible. The actual slip coefficient of an inorganic, zinc-rich paint, for example, will vary from one paint manufacturer to another, and sometimes even from lot to lot. Hence the current requirement for coating certification.

In general, faying surfaces should be clean and dry at assembly. Loose scale, dirt, etc. should be removed by wire brushing, but clean mill scale should not be removed unless the joint is going to be grit-blasted, or the equivalent, to roughen the surfaces (which should never be polished or buffed or smoothed).

If a coating is required for corrosion protection, it should be one that has been thoroughly tested. The four listed below, for example, have at one time or other been tested by the Research Council on Structural Connections. Less expensive vinyl washes and paints are a recent addition to the list (Ref. [7], p. 206).

1. Metallized aluminum
2. Metallized zinc
3. Hot-dip galvanized
4. Inorganic zinc-rich paint

Miscellaneous paints, platings, etc. are not recommended for friction-type joints unless previous tests show that they create an adequate slip coefficient for the joint.

The slip coefficients given above are average figures. In critical situations it's going to be necessary to use minimum figures or (as is the codified practice in structural steel, allowable stress procedures) provide enough safety factor in the bolting requirements to cover such contingencies. Several tables in Chapter 12 in Kulak et al. [7] list the results of tests on a variety of joints. Surfaces were prepared and coated in many different ways. Of most importance for the present discussion, each of these tables lists the average coefficient and the standard deviation. Since the text deals exclusively with structural steel joints, the treatments and coatings are limited to those found in or proposed for that industry. As mentioned above, coatings include hot-dip galvanizing, zinc-rich paint, vinyl coatings, and metallized surfaces which have been sprayed with such materials as zinc or aluminum. Some data are also given for uncoated joint members whose surfaces are coated with clean mill scale. Surface treatment—before coating—involves such rough procedures as sandblasting or wire brushing, which, as mentioned earlier, made a big difference but which would not be appropriate in the shear joints of many other industries.

Although most of this data won't apply directly to joints in other industries, a sampling may be of interest. You'll find these in Table 19.1. Note that a standard deviation equal to 10% of the average slip coefficient is not hard to find. This would suggest a three sigma deviation of $\pm 30\%$, confirming my statements above. Standard deviations greater than 10%, unfortunately, are also easy to find in the Ref. [7] data.

In any event, Equations 19.1 through 19.3 give us much of what we need to know in order to design a shear joint which derives its strength from friction. Here's an example.

19.4.5 An Example

We'll use the joint shown in Figure 19.5 as an example. Input data include:

Number of rows of bolts; one on each side of the joint = 1
Number of bolts in each row, $N = 2$
Bolts: ⅜–20 UN
Bolt material: ASTM A325

TABLE 19.1
Slip Coefficients

Joint Preparation	Type of Coating	Coating Thickness (mils)	Average Coefficient	Standard Deviation
Clean mill scale	None	NA	0.33	0.07
Grit-blasted	Zinc spray	0.6–1.0	0.42	0.04
Grit-blasted	Aluminum spray	1.6–2.2	0.74	0.08
Sandblasted	Zinc dust paint	0.8	0.39	0.02
Sandblasted	Zinc silicate paint	1.0	0.53	0.01
Sandblasted	Vinyl wash	0.3–0.5	0.27	0.01
Sandblasted	Vinyl wash; exposed 2 months	0.3–0.5	0.27	0.05
Acid pickling Bath	Hot-dip galvanized	Not given	0.21	0.08
Acid pickling Bath	Hot-dip galvanized	2.4–5.0	0.23	0.023

Source: All data taken from Kulak, G.L., Fisher J.W., and Struik, J.H.A. in *Guide to Design Criteria for Bolted and Riveted Joints*, Wiley, New York, 1987.

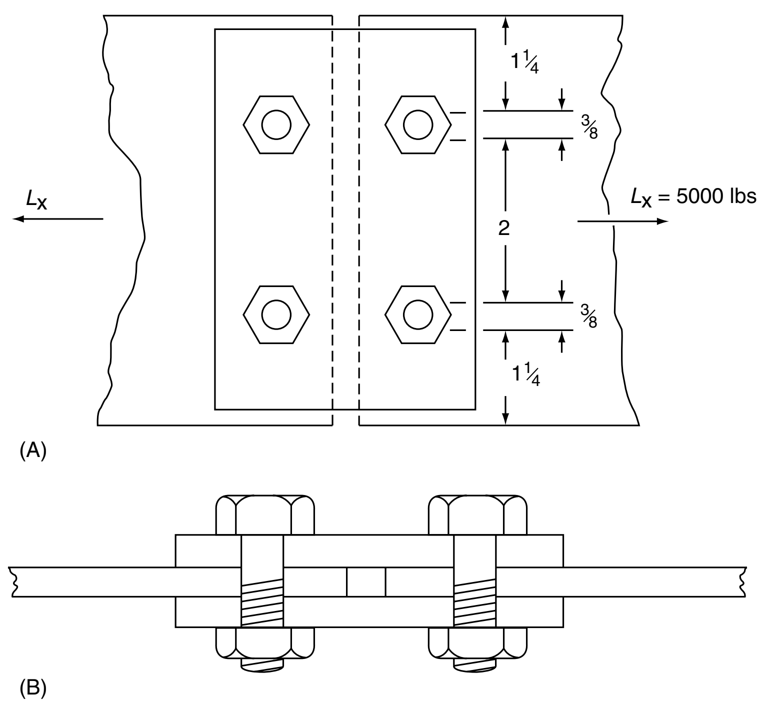


FIGURE 19.5 The joint shown here is used as a design example in the text. Load L_X is also an axial shear load because its line of action passes through the centroid of the joint. Note there will be no tendency for the bolt group to rotate under such a load. The joint and splice plates in this joint are 5 in. thick; the bolts have a nominal diameter of ⅜ in. There are two shear planes here. One passes through the bodies of the bolts, the other through the threads.

Cross-sectional areas of the bolt:

Body = $A_B = \pi D^2/4 = 0.110$ in.^{2}
Tensile stress area of threads = $A_s = 0.0836$ in.^{2} (see Appendix E)
Yield strength of bolt material = $\sigma_y = 92$ ksi (see Table 5.1)
Shear strength of bolt material = $\sigma_s = 79$ ksi (see Table 5.2)
Joint material: A36
Number of slip surfaces (or shear planes) = $M = 2$
Average tensile strength of joint material = $\sigma_{tj} = 70$ ksi (see Table 2.16)
Average shear strength of joint material = $\sigma_{sj} = 48$ ksi (Table 2.16)
Thickness of joint and splice plates: ¼ in.
Grip length = $L_G = 3 \times$ ¼ = ¾ in.
Other joint dimensions are shown in Figure 19.5
Load on joint = $L_X = 5{,}000$ lbs
Joint surfaces: clean mill scale

19.4.5.1 Minimum Preload Required to Prevent Slip

First we'll use Equation 19.3 to determine the minimum preload required to prevent slip. Then we'll return to the VDI equations to see what information they can add. So first, from Equation 19.3, recognizing that the F_P in this equation is our first cut at the VDI F_{Prqd}:

$$R_s = \mu_S F_{Prqd} NM$$

The frictional forces must be large enough to resist the 5,000 lbs load, so $R_s = 5{,}000$. The average slip coefficient for clean mill scale is 0.33 (Table 19.1). So we have:

$$5{,}000 = 0.33(F_{Prqd})(2)(2)$$

$$F_{Prqd} = 3{,}788 \text{ lbs}$$

Are the bolts capable of producing this much clamping force? The preload they could create if tightened to yield would be:

$$F_{Py} = A_S \sigma_y = 0.0836(92{,}000) = 7{,}691 \text{ lbs}$$

so everything seems OK so far.

Now, from VDI Equations 19.1 and 19.2

Min $F_J = 1.3F_{Prqd} = 1.3(3{,}788) = 4{,}924$ lbs and
Max $F_J = \alpha_A(4{,}924)$

What value shall we use for α_A? If we plan to use torque control at assembly we'll have to use 1.86, but that would mean a Max F_J greater than the yield strength of the bolts, and it would be impossible to develop that much preload in these ¾ in. A325 bolts. If we accept the VDI equations, therefore, we'd have to redesign the joint to use larger or more bolts. Or we could specify joint surface treatment giving a larger slip coefficient. There are other ways of looking at this joint, however. Let's see what happens if we specify that the bolts shall be tightened to or past their yield point, a common structural steel practice. Now we use the VDI equations "backward" and set Max F_J equal to bolt tension at yield. The act of yielding will reduce preload scatter to $\pm 5\%$ [1] so

$$\alpha_A = \frac{1.05}{0.95} = 1.11$$

Now, from Equation 19.2

$$7{,}691 = 1.11 \text{ Min } F_J$$
$$\text{Min } F_J = 6{,}929 \text{ lbs}$$

and, from Equation 19.1

$$6{,}929 = 1.3F_{\text{Prqd}}$$
$$F_{\text{Prqd}} = 5{,}330 \text{ lbs}$$

This value of F_{Prqd} is greater than the F_{Prqd} determined earlier from Equation 19.3. Which is correct? The modified VDI Equations 19.1 and 19.2 take into account such factors as tool scatter and elastic interaction loss, which are ignored in Equation 19.3. This equation allows us to compute the minimum preload required to resist the 5,000 lbs load, but doesn't tell us how to achieve that preload. VDI tells us we need more preload than we think we do to compensate for preload loss and assembly uncertainties.

Another possible concern here would be the fact that we've assumed an average slip coefficient for clean mill scale. What if the coefficient is, say, two standard deviations less than average? (See Table 19.1.)

Now

$$\mu_s = 0.33 - 2(0.07) = 0.19$$

From Equation 19.3

$$F_{\text{Prqd}} = \frac{5{,}000}{0.19(2)(2)} = 6{,}579 \text{ lbs}$$

Plugging this value into Equations 19.1 and 19.2 would again suggest a Max F_J above the tensile capacity of the bolt. One way or another, the cold, hard, realistic approach of VDI will force us to use larger or more numerous bolts, or higher friction joint surfaces, to guarantee that the joint won't slip, even if we decide to use yield control.

19.4.5.2 Alternate Using the Allowable Stress Procedure

Let's repeat the example, using Equation 19.4 to compute the maximum slip load which could be placed on this joint per the AISC bolt specifications [8].

$$L_{\text{Xmax}} = F_s A_b N M$$

Table 3 in the specification [8] says that $F_s = 17$ ksi if the slip coefficient is 0.33, the bolts are A325s, and holes are standard (i.e., neither slotted nor oversize). So

$$L_{\text{Xmax}} = 17{,}000(0.110)(2)(2) = 7{,}480 \text{ lbs}$$

Since the anticipated load is only 5,000 lbs, our design is acceptable. Do we now use Equation 19.3 to find the preload required to support this load? For this example we'd have to. Normally we'd use Table 4 in Ref. [8] to determine the preload requirements, but that table doesn't include any bolts less than ½ in. in diameter.

If we were using larger bolts, however, and were designing a structural steel or similar joint, we'd be foolish not to use the AISC specifications. The design equations and allowed stress or load in these documents are based upon many decades of analysis, test, and experience. Although such factors as tool scatter and elastic interaction are invisible in the design equations, they were always present in practice and are covered by implication in the allowable load figures or in the specified procedures for the handling, installation, and tightening of the bolts. If you are designing structural steel joints, therefore, you should follow the AISC design procedures and ignore the complications created by use of the modified VDI equations. If you're designing any other kind of slip-resistant shear joint, however, you'd be wise to use Equations 19.1 through 19.3 to get a more detailed look at a proposed design.

In any event, the joint illustrated in Figure 19.5 would appear to be an acceptable slip-resistant joint, but only if the bolts are properly tightened and an acceptable slip coefficient can be guaranteed.

19.5 STRENGTH OF BEARING-TYPE JOINTS

Several factors determine the load-carrying capability of a bearing-type shear joint. These include:

- The shear strength of the bolts
- The tensile strength of the joint members (called plates in structural steel)
- The bearing stress created in the plates by the bolts
- The tearout strength of the plates

Let's examine each of these.

19.5.1 Shear Strength of Bolts

19.5.1.1 Distribution of Load among the Bolts

The general shear strength of the bolts in a shear joint can be expressed as:

$$R_B = A_{blt} N \sigma_S \tag{19.5}$$

where

R_B = force required to shear all of the bolts on one side of a joint (lb, N)
A_{blt} = total cross-sectional area of the bolt which must be sheared (in.^{2}, mm^2)
N = number of bolts on one side of the joint
σ_S = shear strength of the bolt material (psi, MPa)

These terms will become clearer when we look at an example in a minute. But first, some general comments about shear loads on bolts. Equation 19.5 implies that every bolt will bear an equal share of the load in a shear joint, and, for simplicity, we'll make this assumption when selecting the number, size, and type of bolts to be used in our joint. But, in fact, the bolts will not share the load equally, a fact which was illustrated in Figure 12.8. Those bolts closest to the leading and trailing edges of the joint will see far more load than those nearer the center of the bolt pattern. Unfortunately, not even the bolts in a given row will share the load absorbed by that row equally. The placement of the bolts in their holes, minor variations in hole and bolt diameter, and the way the load is applied to the joint will all affect the loads seen by individual bolts. The situation, in fact, is statically indeterminate.

The answer to these uncertainties is our old friend the safety factor. We decide how much shear stress the average bolt will see, and then use a bolt capable of supporting many times that stress. All of this applies to what we call an "axial shear load" in which the line of action of the applied load passes through the centroid of the bolt group as it does in Figures 19.4 and 19.5. The safety factor must be increased still further if the applied load is eccentric. Just as prying significantly magnified the loads seen by some of the bolts in a tension joint, so does eccentricity magnify them in a shear joint. We'll take a look at that in Section 19.6.

For now, however, we're going to adopt the procedures long used by the designers of bearing-type joints and assume that each bolt in the joint sees the same load.

19.5.1.2 Shear Strength Calculations

Once again we'll use the joint shown in Figure 19.5 for our example. The design data given at the beginning of the earlier example, Section 19.4.4., still apply.

The shear stress within a single bolt will be:

$$\sigma = \frac{F}{A_{\text{blt}}} \tag{19.6}$$

where

- σ = shear stress (psi, MPa)
- F = shearing force applied to that one bolt (lb, N)
- A_{blt} = total cross-sectional shear area of that one bolt (in.^{2}, mm^2)

If there are N bolts on one side of the joint, then

$$F = L_X/N \tag{19.7}$$

For example, in Figure 19.5 upper and lower splice plates hold two joint members together. There are two bolts to the left of the gap between joint members and two to the right of the gap, so $N = 2$.

The total shear area of a bolt depends upon how many shear planes pass through it and whether they pass through the body of the bolt, the threaded region, or both.

$$A_{\text{blt}} = n_b A_B + n_s A_s \tag{19.8}$$

where

- n_b = number of shear planes which pass through the body
- n_s = number of shear planes which pass through the threads
- A_B = cross-sectional area of the body (in.^{2}, mm^2)
- A_s = tensile stress area of the threads (in., mm) (see Appendix F)

In Figure 19.5 two shear planes pass through each bolt, one plane through the body and one through the threads, so

$A_{\text{blt}} = 1(A_B) + 1(A_S)$. These are 3/8–20 UN bolts, so $A_B = 0.110$ in.^{2} and $A_s = 0.0836$ in.^{2}

$$A_{\text{blt}} = 0.110 + 0.0836 = 0.194 \text{ in.}^2$$

We can now compute the stress within the bolt and compare it to an allowable stress. The load on this joint is 5,000 lbs. From Equations 19.6 and 19.7

$$\sigma = \frac{5{,}000}{2(0.194)} = 12{,}887 \text{ psi (90 MPa)}$$

The allowable shear stress for Grade A325 bolts used in structural steel applications is 21 ksi (145 MPa) [8], so we're probably OK. The allowable stress in other industries might be higher, closer perhaps to the 79 ksi shear strength of A325 bolt material (Table 5.2). In fact, as an alternative we can compute the total shear strength of the bolts in the joint and compare it to the applied load. From Equation 19.5

$$R_B = A_{blt} N \sigma_S$$

$$R_B = (0.194)(2)(79{,}000) = 30{,}652 \text{ lbs}$$

So the joint shown in Figure 19.5 should be able to support the 5,000 lbs load with a 6:1 safety factor. But we still have several other factors to consider.

19.5.2 Tensile Strength of Joint Plates

We compute the tensile strength of the joint with respect to the net section which includes the most bolts. (All rows may not have the same number of bolts.) In our example, of course, there are only two bolts per row. With reference to the dimensions shown in Figure 19.5, and to the fact that the joint plates are 0.250 in. thick, the area of the net section is

$$A_J = 0.250(1.25 + 2 + 1.25) = 1.125 \text{ in.}^2$$

Since the tensile strength of A36 is 70 ksi, the tensile strength of this joint will be

$$R_T = A_J \sigma_u = 1.125(70{,}000) = 78{,}750 \text{ lbs}$$

Obviously this is not a concern in our example, where the applied load is only 5,000 lbs.

19.5.3 Bearing Stress

The bearing stress the bolts create in the joint plates is found from

$$\sigma_B = \frac{L_X}{NDL_G} \tag{19.9}$$

where
- σ_B = bearing stress (psi, MPa)
- L_X = load applied to joint (lb, N)
- N = number of bolts on one side of the joint
- D = nominal diameter of the bolts (in., mm)
- L_G = grip length of the joint (in., mm)

The area in bearing is shown in Figure 19.6. In our example:

$$\sigma_B = \frac{5{,}000}{2(0.375)0.750} = 8{,}889 \text{ psi}$$

Is that an acceptable amount? We're concerned here, for example, with the possibility that the bolt holes will be elongated by the bearing load. According to Ref. [8], bearing stresses of 1.0 to 1.2 times the ultimate tensile strength of the plate material are allowed in structural steel if the bolts are A325 or A490. The exact amount allowed depends upon whether or not the holes are slotted, upon the number of bolts, and upon the distance between the row of bolt

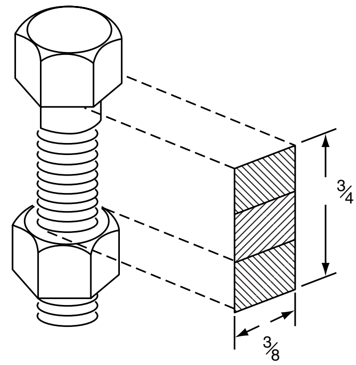


FIGURE 19.6 We compute the bearing stresses created by the bolt on the joint plates using the crosshatched area shown here, which is equal to the nominal diameter of the bolt times the joint's grip length. In the joint shown in Figure 19.5, therefore, the bearing area would be $0.375 \times 0.750 = 0.281$ in.^{2}

holes and the edge of the plate. If we accept 1.0 times the minimum ultimate strength of A36 we can accept a bearing stress of up to 58,000 psi, well below the amount computed above.

19.5.4 Tearout Strength

To compute the tearout strength we must first compute the minimum plate area which would have to be sheared. This area is illustrated in Figure 19.7. Note that there will be two such areas per bolt, or four areas in the present example. The pieces torn out of the plate are sometimes wedge shaped, with larger shear areas, but we're only concerned with the minimum force required. So

$$A_{sh} = 2N_R D L_{eg} \tag{19.10}$$

where

A_{sh} = total area which must be sheared if tearout is to occur (in.^{2}, mm^2)
N_R = number of bolts in the row nearest the edge
D = distance to edge of plate (in., mm)
L_{eg} = distance from bolt hole to edge of plate (in., mm)

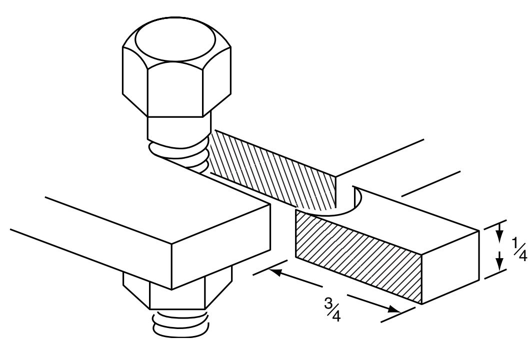


FIGURE 19.7 If tearout is to occur each bolt must shear at least twice the amount of plate area shown crosshatched here. This area is equal to the thickness of the joint plate times the distance between the centerline of the bolt hole and the edge of the plate. In the joint shown in Figure 19.5, therefore, the total area to be sheared by the two bolts would be $4 \times 0.25 \times 0.75$ in.^{2} $= 0.75$ in.^{2} (See also Figure 13.1.)

In our example

$$A_{sh} = (2)(2)(0.375)(0.750) = 1.125 \text{ in.}^2$$

And the tearout strength would be

$$R_{TO} = A_{sh}\sigma_s \tag{19.11}$$

where
R_{TO} = tearout strength of joint (lb, N)
σ_s = shear strength of plate material (psi, MPa)

In our example:

$$R_{TO} = (1.125)(48{,}000) = 54{,}000 \text{ lbs}$$

Note that computation of tearout strength becomes far more complicated if there are several rows of bolts on each side of the joint. In this situation tearout would involve every row—and would be highly unlikely if not impossible.

19.5.5 Summary

We have now decided that the following loads would be required to pull this joint apart.

By shearing the bolts: 30,652 lbs
Through tensile failure of the joint member: 78,750 lbs
By tearout of the bolts: 54,000 lbs

This means that the shear strength of the bolts determines the strength of this joint. Since the applied load is only a sixth of the shear strength, joint failure will not occur.

We also estimated the bearing stress the bolts will create on the joint and found it to be much less than that we can allow. So the design of our bearing-type joint is confirmed.

19.5.6 Clamping Force Required by a Bearing-Type Joint

OK so far, but remember that one of our goals is to determine how much F_{Prqd} is required to prevent joint failure, so that we could solve VDI Equations 19.1 and 19.2. What do Equations 19.5 through 19.11 tell us about F_{Prqd}? They tell us that no specific F_{Prqd} is required. The bearing-type shear joint will theoretically resist shear loads even if all the bolts are dead loose. In practice, we'll tighten them at least enough to retain the nuts—and therefore to retain the bolts. In fact, we may tighten them more to resist self-loosening (which, as we saw in Chapter 14, is caused by transverse slip). Self-loosening requires cyclic loads, however, and they can also cause fatigue failure of joint members. If we're concerned about self-loosening we should probably design the joint to resist shear loads through friction rather than in bearing.

19.6 ECCENTRICALLY LOADED SHEAR JOINTS

19.6.1 Rotation about an Instant Center

Axial shear load means that the centerline of the external load passes through the centroid of the group of fasteners holding the shear joint together, as suggested in Figures 19.4 and 19.5.

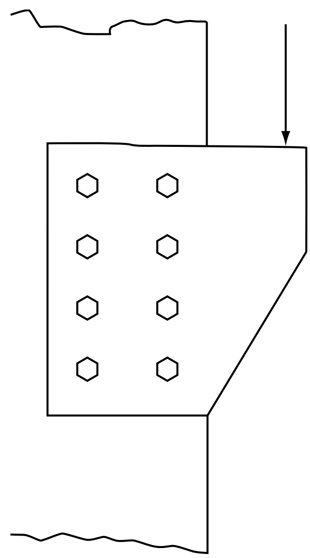


FIGURE 19.8 A load on a shear joint is said to be eccentric when its resultant does not pass through the centroid of the bolt pattern.

In many cases shear joints are subjected to external loads that don't pass through the centroid of the fastener group, as in Figure 19.8. Such joints are called eccentrically loaded shear joints. The external load is applied to the bolts through some sort of leverage determined by the geometry of the joint; so eccentric load is the shear joint equivalent of prying load in a tension joint (see Chapter 11). In both cases leverage makes a big difference in the load seen by some of the fasteners [1,5–7].

Under an eccentric load, the entire group of bolts will tend to rotate about an instant center that is determined by the bolt pattern and by the direction of the applied load (Figure 19.9). Computing the exact load on each bolt under these conditions is time consuming and involves eccentricity and safety factors that must be determined by experiment. Such factors are listed in structural design handbooks, which also list precomputed solutions for a number of standard joints (see also Refs. [7] and [9]).

For our purposes, it is sufficient to say that those bolts located farthest from the theoretical center of rotation will carry the greatest load and are most apt to fail.

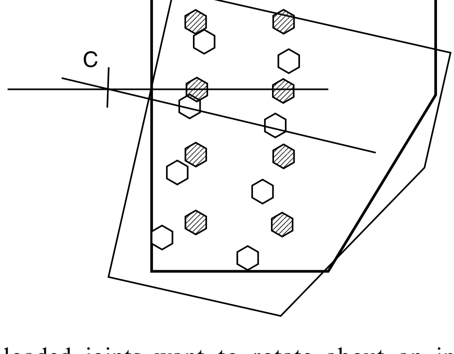


FIGURE 19.9 Eccentrically loaded joints want to rotate about an instant center as shown here, magnifying the shear loads on the bolts.

19.6.2 Rotation about the Centroid of the Bolt Group

Although the joint will actually attempt to rotate about an instant center, it is easier to estimate the shear loads on the bolts if we assume that it tries to rotate about the centroid of the bolt group. This historically earlier and simpler approach results in conservative estimates of shear loads; i.e., it overestimates them, and so is safe to use. Here's an example of this procedure.

19.6.2.1 Find the Centroid of the Bolt Group

We can usually find the centroid of a bolt group by inspection, as in Figure 19.8. If the bolts are not in neat rows, however, as in Figure 19.10, we can use the following equations to locate it [9,12]:

$$\overline{X} = \frac{\sum A_n x_n}{\sum A_n} \tag{19.12}$$

$$\overline{Y} = \frac{\sum A_n y_n}{\sum A_n} \tag{19.13}$$

where

A_n = cross-sectional area of the body of bolt n (in.^{2}, mm^2)

x_n = distance of bolt n from an arbitrarily located y axis (in., mm)

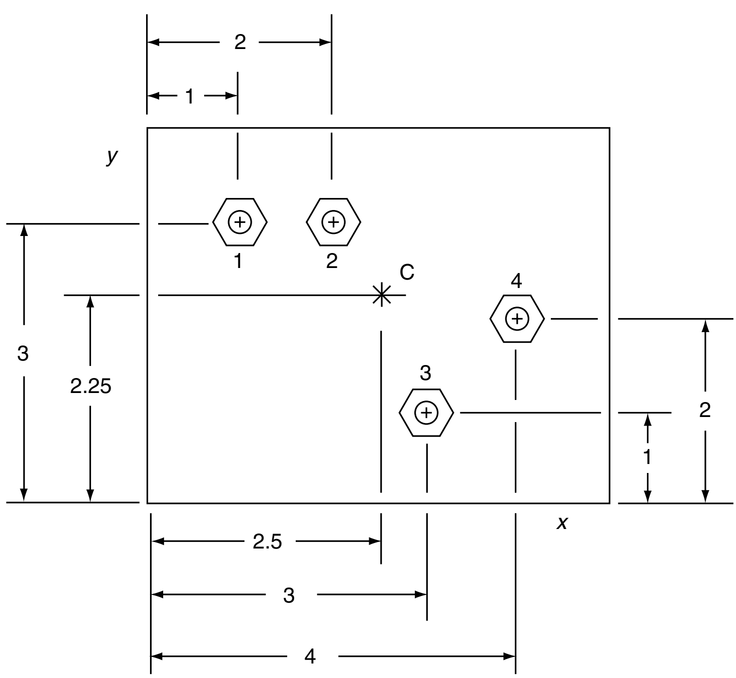


FIGURE 19.10 This joint is used as an example in the text, to compute the location of the centroid (C) of the bolt group. The bolts are ¼ in. in diameter. The x and y axes are arbitrarily located along the left and bottom sides of the plate.

y_n = distance of bolt n from an arbitrarily located x axis (in., mm)
X = distance of the centroid from the y axis (in., mm)
Y = distance of the centroid from the x axis (in., mm)

With reference to Figure 19.10, let's assume that the nominal diameter of the bolts is ¼ in. and that we've arbitrarily located x and y axes along the bottom and left edge of the splice plate as shown. The distance of bolt 1 from the y axis is x_1; its distance from the x axis is y_1, and so on.

$$A_n = \frac{\Pi(0.25^2)}{4} = 0.0491 \text{ in.}^2$$

$$\overline{X} = \frac{A_1x_1 + A_2x_2 + A_3x_3 + A_4x_4}{A_1 + A_2 + A_3 + A_4}$$

Since each bolt has the same cross-sectional area in this example:

$$\overline{X} = \frac{0.0491(1+2+3+4)}{4(0.0491)} = 2.5 \text{ in.}$$

Similarly,

$$\overline{Y} = \frac{0.0491(1+2+3+4)}{4(0.0491)} = 2.5 \text{ in.}$$

The centroid so located is labeled C in Figure 19.10.

19.6.2.2 Estimating the Shear Stress on the Most Remote Bolt

Now, with reference to Figures 19.11 and 19.12 we're going to estimate the shear load on an eccentrically loaded fastener. Let's assume that the joint shown is a bearing type. First, we compute the primary shear load on each fastener.

$$P_{\mathrm{P}} = L_{\mathrm{X}}/N \tag{19.14}$$

where
P_{P} = the primary shear force on each bolt (lb, N)
L_{X} = the load applied to the joint (lb, N)
N = the number of bolts in the group

In this example

$P_{\mathrm{P}} = 15{,}000/6 = 2{,}500$ lbs

Next we must compute the secondary shear load on the fastener. To do this we first compute the distance between the centroid and the most distant bolt (see Figure 19.11).

$$r_{\mathrm{d}} = \frac{2}{\sin 45^\circ} = \frac{2}{0.707} = 2.828 \text{ in.}$$

And the reaction moment on the joint

$$M = L_{\mathrm{X}}U \tag{19.15}$$

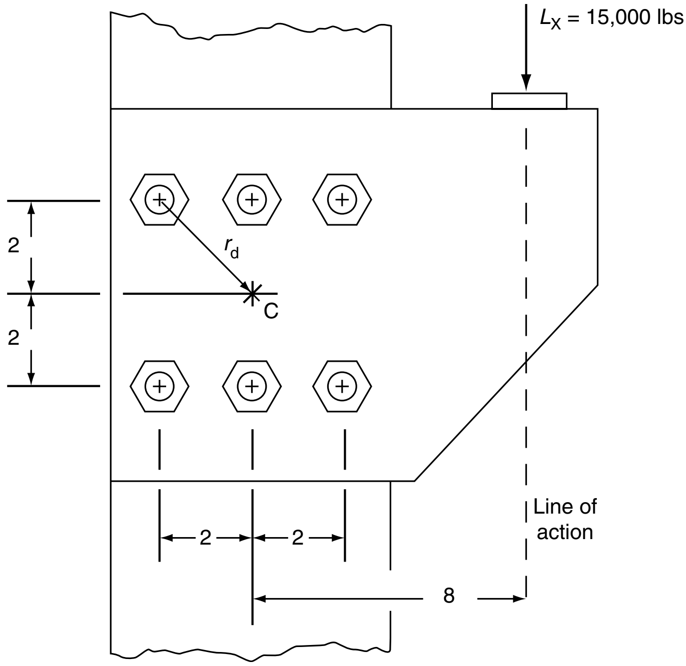


FIGURE 19.11 The joint shown here is used as an example in the text to compute the effects of an eccentric load on a shear joint. Note that the line of action of the applied force, L_X, is 8 in. from a parallel line passing through the centroid (C) of the bolt group.

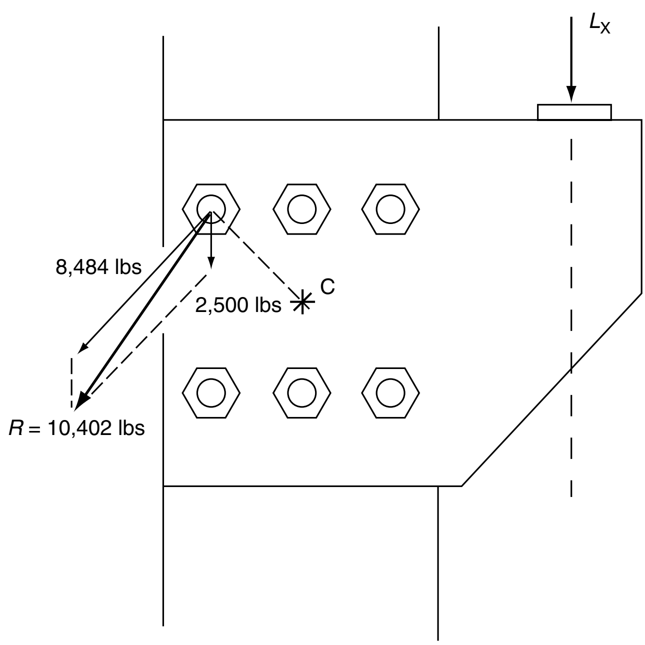


FIGURE 19.12 The shear forces acting on the bolt most distant from the centroid are shown here, along with the resultant force, R. The forces consist of a primary shear force of 2,500 lbs and a secondary shear force of 8,484 lbs created by the fact that the load is eccentric. The resultant total force on the bolt is 10,402 lbs, well in excess of the 2,500 lbs the bolt would have to support if the same external load had a line of action which passed through the centroid of the joint. The same force would be exerted on each of the bolts in the four corners of the group, but each force would be in a different direction.

where

M = reaction moment (in.-lb, mm-N)
L_X = load on the joint (lb, N) = 15,000 lbs
U = perpendicular distance between the centroid and the line of action of L_X (in., mm) = 8 in.

In our example:

$$M = 15{,}000(8) = 120{,}000 \text{ in.-lb}$$

Next we estimate the reaction force on the most distant bolt

$$P_d = \frac{Mr_d}{\sum r_n^2} \tag{19.16}$$

where

P_d = reaction force on most distant bolt (lb, N)
r_n = distance of bolt n from the centroid (in., mm)

From Figure 19.11 we see that four bolts are each at a distance 2.828 in. from the centroid, and two bolts are 2 in. from it, so

$$P_d = \frac{120{,}000(2.828)}{4(2.828^2) + 2(2^2)}$$

$$P_d = 8{,}484 \text{ lbs}$$

The line of action of the force P_d will be perpendicular to r_d, as shown in Figure 19.12. Combining the P_s and P_d vectors we get the resultant shear force (R) exerted on the most distant bolt by the eccentric load L_X. In our example the resultant is 10,402 lbs. This same force would be seen by each of the bolts in the four corners of the bolt pattern, but each force would be in a different direction. We now use Equation 19.6 to estimate the shear stress in the most distant bolts.

$$\sigma_s = \frac{R}{A_{blt}}$$

If these are 5/8–20 UN, A325 bolts, and there's only one shear plane which passes through the body of the bolt, then:

$$A_{blt} = \frac{\Pi(0.625^2)}{4} = 0.307 \text{ in.}^2$$

and

$$\sigma_s = \frac{10{,}402}{0.307} = 33{,}883 \text{ psi}$$

This is less than the average shear strength of an A325 bolt, which is 79 ksi, but it exceeds the AISC allowable stress limit of 21 ksi [8]. Note that the stress would be well under the allowable if the bolt were exposed only to primary shear force. Eccentricity has magnified the loads seen by the most distant bolts.

If we were to analyze this joint assuming rotation about an instant center, we might conclude that the worst-case shear stress in the most distant bolt was within the allowable limit, but, on the basis of our conservative rotation-about-the-centroid procedure we have decided that the farthest bolts will be overstressed. We'd have to go to a ⅞ in. bolt to bring stress within the allowable limit (or go to a higher-strength bolt material, with less increase in size).

This would not end the analysis, of course. We must still check the bearing stress created by the bolts on the joint plates. Tearout and tensile failure would be unlikely here, but in some designs would have to be checked. We could also design this joint to be slip resistant by determining the resultant forces on each bolt, combining these to compute the total force to be resisted, then choosing bolts and a slip coefficient to provide sufficient slip resistance. I'll leave that as an exercise for the reader.

19.7 ALLOWABLE STRESS VERSUS LOAD AND RESISTANCE FACTOR DESIGN

The structural steel examples I've given in this chapter have all been based upon what that industry calls the allowable stress design procedure [7,8]. Until recently this was the only procedure employed by that industry.

Any safe design procedure must take into account the fact that estimates of the strength of a structure are not absolute, that the actual strength will vary because of inevitable variations in dimensions, material properties, and the like. Any safe procedure must also take into account the fact that the exact loads a structure must support are also unknown, thanks to variations in field or service environments, or design assumptions or errors which result in unexpected stresses or stress concentrations. The allowable stress design procedure is based on the worst-case assumption that a structure will have minimum strength but must support the maximum possible load. Uncertainties are covered by a variety of safety factors. The designer, for example, must assume that the strength of the bolts is less than the minimum strength specified by ASTM.

As discussed in Chapter 12 an alternate procedure has been developed by the industry, led by the Research Council on Structural Connections. This is called the "load and resistance factor" procedure [7,13]. Anticipated loads and anticipated strength (resistance) are modified by factors that reflect the probabilistic uncertainties in those estimates. The general expression for relating structural strength to anticipated loads is [7]

$$\Phi R = \alpha D + \gamma (L + I) \tag{19.17}$$

where

Φ = resistance factor reflecting the uncertainty in strength
R = the average strength (resistance)
α, γ = load factors reflecting the probability of an increase in load
D = the anticipated dead load on the structure
L = the anticipated live load on a structure
I = the anticipated impact load on a structure

The load and resistance factors would, of course, be a function of the application, of the type of joint involved. Those used by the structural steel industry would presumably not be appropriate for the designer of airframes or autos, so I'm not going to go into more detail here. Those involved with structural steel design should consult the cited references. I will add, however, that the two AISC specifications [8,13] will lead to similar results. Load and resistance factor design is an alternate procedure; it is not a mandated replacement for the allowable stress procedure. Designers are free to use the one they prefer.

EXERCISES

1. Name the two ways in which joints loaded in shear resist loads.
2. Which kind is slip-critical?
3. Which kind requires control of bolt preload during assembly?
4. What are typical coefficients of friction for slip-critical joints?
5. Approximately how much force would it take to shear one 3⁄4–28 × 6 ASTM A325 bolt?
6. If the code you're working with mandates a 4:1 safety factor what's the design maximum force allowed for that bolt?
7. How much more shear load could you allow in your design if you specified A490 bolts instead of A325?
8. How much more shear load could you allow if you specified 7⁄8–28 A490 bolts instead of 3⁄4–28 A325s?
9. What obvious other factor would you have to consider before making this change and allowing the increased load it implies?
10. You're concerned about the slip resistance of a slip-critical joint you're designing. You had hoped to avoid the expense of specifying a slip-resistant coating for the joint members, but now decide that you must do so. You specify zinc spray over a grit-blasted joint surface instead of using the joint in the as-received condition. By what percentage do you expect this will increase the shear load that this joint can support?

REFERENCES

1. Fisher, J.W. and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, Wiley, New York, 1974.
2. Munse, W.H., Final report on riveted and bolted structural joints, Division of Highways, Federal Highway Administration, NTIS, PB-201297, 1C34022427.
3. Carter, J.W., J.C. McCally, and L.T. Wyly, Comparative test of a structural joint connected with high-strength bolts and a structural joint connected with rivets and high strength bolts, *AREA Bulletin*, vol. 56, Proceedings, pp. 217–267.
4. *High Strength Bolting for Structural Joints*, Booklet 2867, Bethlehem Steel Company, Bethlehem, PA.
5. Fazekas, G.A., On eccentrically loaded fasteners, *Transactions of ASME*, pp. 776–778, August 1976.
6. Fazekas, G.A., Eccentric limit load on fasteners, *Transactions of ASME*, pp. 773–775, August 1976.
7. Kulak, G.L., J.W. Fisher, and J.H.A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, 2nd ed., Wiley, New York, 1987.
8. *Specification for Structural Joints Using A325 and A490 Bolts: Allowable Stress Design*, American Institute of Steel Construction, Chicago, November 13, 1985.
9. Section 23, Bolted and Riveted Joints, *Standard Handbook of Machine Design*, J.E. Shigley and C.R. Mischke (Eds.), McGraw-Hill Book Company, New York, 1986, pp. 23.12–23.16.
10. Bibel, G.D. and D.L. Goddard, Preload variations of torqued fasteners, a comparison of friction and elastic interaction effect, *Fastener Technology International*, pp. 31ff, June/July 1994.
11. *Systematic Calculation of High Duty Bolted Joints*, VDI 223, Society for Design and Development, Committee on Bolted Joints, Verein Deutscher Ingenieure, VDI Richtlinien, Dusseldorf, Germany, October 1977; translated by Language Services, Knoxville, TN; translation published by the U.S. Department of Energy as ONRL-tr-5055.
12. Shigley, J.E., *Mechanical Engineering Design*, 3rd ed., McGraw-Hill, New York, 1977, p. 258.
13. *Specification for Structural Joints Using A325 and A490 Bolts: Load and Resistance Factor Design*, American Institute of Steel Construction, Chicago, June 8, 1988.

Appendix A
Units and Symbol Log

Symbol	Uses	English Units	Metric Units
A	Area	in.^{2}	mm^2
a	Slope of the S_G–T_P line on a log–log plot: one of three new PVRC gasket constants	psi	MPa
a	Distance between centerline of an eccentric joint and the point of application of an external tension load (or resultant of several loads)	in.	mm
a	Experimentally derived exponent used to compute T_P	None	
a	Depth of crack in K_{ISCC} calculations	in.	mm
A_B	Cross-sectional area of the body of a fastener	in.^{2}	mm^2
A_{blt}	Total, cross-sectional shear area of a bolt (sum of the shear areas through body and through threads)	in.^{2}	mm^2
A_C	Cross-sectional area of the equivalent cylinder used to compute joint stiffness	in.^{2}	mm^2
A_G, A_g	Gasket contact area	in.^{2}	mm^2
A_i	Area used to compute the hydrostatic end load on a gasketed joint	in.^{2}	mm^2
A_J	Cross-sectional area of the joint	in.^{2}	mm^2
A_p	Contact area between bolt head or nut face and joint	in.^{2}	mm^2
A_r	Cross-sectional area of the minimum minor diameter of bolt threads	in.^{2}	mm^2
A_S	Effective cross-sectional area of the threaded section of a fastener (an assumed area based on the mean of pitch and minor diameters)	in.^{2}	mm^2
A_{sh}	Area which must be sheared to ''tear out'' a shear joint	in.^{2}	mm^2
A_{TS}	Cross-sectional area of shear of a thread	in.^{2}	mm^2
A_{tot}	Total loaded cross-sectional area of a fastener in shear	in.^{2}	mm^2
A_v	Cross-sectional area of a pressure vessel	in.^{2}	mm^2
b	Effective width of gasket per ASME Code rules	in.	mm
C	Shape factor in K_{ISCC} calculations	None	
c	A tightness factor defining an acceptable, minimum leak rate	None	
C_1–C_5	Correction factors for fatigue calculations	None	
D	Nominal diameter of fastener	in.	mm
D	The anticipated dead load on a shear joint	lb	N
d	Diameter of hole through or into fastener	in.	mm
D_B	Diameter of bolt head or washer (diameter of contact with joint members)	in.	mm
D_H	Diameter of bolt hole	in.	mm
D_J	Diameter of a cylindrical joint, or length of one side of a square joint, or length of short side of rectangular joint	in.	mm
d_m	Minor diameter of male thread	in.	mm
D_r	Root diameter of the threads	in.	mm
D_{smin}	Minimum nominal diameter	in.	mm

continued

Appendix A: Units and Symbol Log (continued)

Symbol	Uses	English Units	Metric Units
E	Young's modulus (modulus of elasticity)	psi	GPa
e	Assembly efficiency	None	
E_A	Average loss of preload because of elastic interactions	lb	N
E_B	Modulus of elasticity of the bolt material	psi	GPa
e_{EI}	Percentage of preload lost because of elastic interactions, expressed as a decimal	None	
e_m	Percentage of preload lost because of embedment, expressed as a decimal	None	
E_m	Young's modulus for alternate joint material	psi	GPa
E_{min}	Minimum pitch diameter	in.	mm
E_{nmax}	Maximum pitch diameter of internal thread (nut)	in.	mm
E_p	Basic (nominal) pitch diameter of fastener	in.	mm
E_{Smin}	Minimum pitch diameter of external thread (bolt)	in.	mm
F	Force	lb	N
ΔF	Change in force	lb	N
f	Frequency ($\Delta f =$ change in frequency)	Hz	Hz
F_B	Tension force in bolt	lb	N
ΔF_B	Change in bolt tension	lb	N
F_{CL}	Clamping force	lb	N
ΔF_{EI}	Loss of preload caused by elastic interactions	lb	N
F_J	Per-bolt clamping force on the joint	lb	N
F_j	Resonant ultrasonic frequency in a bolt	Hz	Hz
F_J	Compression force in joint	lb	N
ΔF_J	Change in per-bolt clamping force created by the external load	lb	N
F_{Krqd}	Minimum preload (or clamping force) required to prevent separation of an eccentrically loaded joint	lb	N
ΔF_m	Loss of preload caused by embedment	lb	N
F_P	Preload	lb	N
F_{Pa}	Average, initial assembly preload; also called the "target" preload	lb	N
F_{Prqd}	Minimum preload required to prevent slip, separation, or leakage of a concentrically loaded joint	lb	N
F_S	Force required to shear a fastener	lb	N
F_T	Tension created in bolt by differential thermal expansion	lb	N
ΔF_{th}, ΔF_{TH}	Change in preload caused by a change in temperature	lb	N
F_y	Force required to create a 0.2% permanent set in the bolt (i.e., to yield it)	lb	N
f_z	Amount by which a fastener embeds	in.	N
G	Diameter of a pressure vessel to the midpoint of the gasket	in.	mm
G	Grip length	in.	mm
G	Ratio between shear strength and tensile strength	None	
G	Shear modulus	psi	MPa
G_b	Intercept of the S_G–T_P line with the S_G axis: one of the three new PVRC gasket constants	psi	MPa
G_o	Outer diameter of a gasket	in.	mm
G_S	Intercept of the gasket's unloading–reloading line with the S_G axis: one of the three new PVRC gasket constants	psi	MPa
H	Height of basic thread triangle ($H = 0.86603 - n$, if $\beta = 60°$)	in.	mm
H_D	End force exerted by internal pressure (e.g., in a pressure vessel) on a flange; reaches the flange through the wall of the vessel (or pipe) and through the hub of the flange	lb	N
H_G	Reaction force exerted by a gasket on the flange	lb	N
H_T	Force on a flange created by pressure exerted by contained fluid on that portion of the flange which lies between the ID of the gasket and the ID of the pipe or vessel	lb	N

Appendix A: Units and Symbol Log (continued)

Symbol	Uses	English Units	Metric Units
I	The anticipated impact load on a shear joint	lb	N
I	Moment of inertia	in.^{4}	mm^4
K	Nut factor	None	
K	Spring constant or stiffness	lb/in.	N/mm
K_B, K_b	Spring constant of bolt	lb/in.	N/mm
K_F	Spring constant of flange	lb/in.	N/mm
K_G	Spring constant of gasket	lb/in.	N/mm
K_{ISCC}	Threshold stress intensity factor for SCC	psi-in.$^{1/2}$	Pa-mm$^{1/2}$
kip	Force in thousands of pounds	Is an English unit	No equivalent
K_J	Spring constant of joint	lb/in.	N/mm
K_J'	Stiffness of a concentric joint loaded at internal loading planes	lb/in.	N/mm
K_J''	Stiffness of a joint in which both the axes of the bolts and the line of application of a tensile force are offset from the axis of gyration of the joint, and in which the tensile load is applied along loading planes within the joint members	lb/in.	N/mm
K_N	Spring constant of nut	lb/in.	N/mm
K_{nmax}	Maximum minor diameter of nut threads	in.	mm
ksi	English unit for thousands of pounds per square inch	Is an English unit	No equivalent
K_{Smin}	Minimum minor diameter of bolt thread	in.	mm
K_T	Spring constant of a group of springs in series	lb/in.	N/mm
K_T	Stiffness of a nut-bolt-washer system	lb/in.	N/mm
K_{TG}	Stiffness of a gasketed joint	lb/in.	N/mm
K_{tor}	Torsional spring constant, defined as $T_{tor} = K_{tor} \times \Delta L$	lb	N
K_w	Spring constant of washer	lb/in.	N/mm
L	Length (ΔL = change in length)	in.	mm
L	The anticipated live load on a shear joint	lb	N
$\Delta L'$	Bolt stretch after application of external load	in.	mm
l	Length	in.	mm
L_B	Length of fastener (e.g., body plus threads)	in.	mm
ΔL, ΔL_B	Bolt stretch (same as ΔL_C)	in.	mm
L_{be}	Effective length of body of fastener (includes a portion of the head of the fastener)	in.	mm
L_C	Total length of fastener, including head	in.	mm
ΔL_C	Combined or total change in length of bolt	in.	mm
L_E	Effective length of fastener (total length under load)	in.	mm
L_e	Minimum length of thread engagement required to develop maximum strength	in.	mm
L_e, L_{eff}	Effective length of the bolt	in.	mm
L_{eg}	Distance from bolt hole to tearout edge of joint plate in a shear joint	in.	mm
L_G	Grip length	in.	mm
L_J	Length of joint members involved in thermal expansion	in.	mm
ΔL_J	Increase or decrease in the thickness of a joint because of a temperature change	in.	mm
L_S	External shear load	lb	N
L_X	External tensile or shear load applied to joint. Also the maximum tensile load experienced during a load cycle	lb	N
L_{Xmin}	Minimum tensile load experienced during a load cycle	lb	N
L_{max}	Maximum external load a fastener can support	lb	N
L_{RM}	Mass leak rate	lb/h in.	mg/s mm
L_{RM}^*	Reference mass leak rate	lb/h in.	mg/s mm
L_{ro}	Length of thread run-out per ANSI or other fastener specifications ($L_T + L_{ro} = L_t$)	in.	mm

continued

Appendix A: Units and Symbol Log (continued)

Symbol	Uses	English Units	Metric Units
L_{se}	Effective length of threads of fastener (exposed threads plus a portion, e.g., half, of the threads within the nut)	in.	mm
L_T	Minimum thread length per ANSI or other fastener specifications	in.	mm
L_t	Total length of threads on fastener	in.	mm
ΔL_W	Change in distance between the washers of a bolt as it is loaded	in.	mm
L_X	External load on joint	lb	N
L_{XC}	External compression load	lb	N
L_{Xcrit}	Critical external load (load required to free the joint completely)	lb	N
M	Reaction moment on an eccentrically loaded shear joint	in.-lb	mm-N
M	Number of slip surfaces in a shear joint	None	
m	Gasket factor; ratio of residual (under load) clamping pressure on a gasket required to prevent a leak, to the contained pressure	None	
M_b	Bending moment acting on fastener	in.-lb	N-mm
MV	Material velocity	in./sec	mm/s
N	Number of bolts in the joint; also the number of threads per inch	None	
N	Full width of a gasket	in.	mm
N	Number of bolts on one side of a shear joint	None	
N	Number of cycles; also number of bolts in a shear joint	None	
n	Number of tests in a statistical sample	None	
n	Number of threads per inch	None	
n	Fraction of joint thickness which lies between the loading planes: also number of bolts in a joint	None	
n_b	Number of shear planes which pass through the body of a bolt	None	
N_R	Number of bolts in the row nearest the edge of a shear joint	None	
n_s	Number of shear planes which pass through bolt threads	None	
P	Percentage, expressed as a decimal (e.g., $65\% = 0.65$)	None	
P	Number of shear planes in a bolt in a bearing-type shear joint	None	
P	Pressure	psi	Pa
P^*	Atmospheric pressure	psi	Pa
P, P_i, p	Pitch of threads	in.	mm
PE_B	Potential energy stored in the bolt	in.-lb	mm-N
P_G	Maximum allowable pressure which can be exerted on the joint by the nut or bolt head	psi	MPa
P_p	Primary shear load on the fasteners in an eccentrically loaded shear joint	lb	N
Q	Prying force	lb	N
R	Radius of curvature of bent bolt	in.	mm
R	Average strength (resistance) of a shear joint	lb	N
R_B	Combined shear strength of a shear joint of all of the bolts in a shear joint	lb	N
R_B	Total strength (in shear) of bearing-type shear joint	lb	N
R_G	Radius of gyration of cross-sectional area of joint	in.	mm
R_S	Slip resistance of shear joint	lb	N
r_B	Resilience of bolt (reciprocal of bolt stiffness K_B)	in./lb	mm/N
r_J	Resilience of concentric joint (reciprocal of joint stiffness K_J)	in./lb	mm/N
r_J'	Resilience of eccentric joint in which external load and bolt are coaxial	in./lb	mm/N
r_J''	Resilience of eccentric joint in which external load and bolt lie along different but parallel axes	in./lb	mm/N
r_n	Effective radius of contact between nut and joint or washer	in.	mm
r_t	Effective radius of contact between nut and bolt threads	in.	mm

Appendix A: Units and Symbol Log (continued)

Symbol	Uses	English Units	Metric Units
R_T	Tensile strength of the joint members in a shear joint	lb	N
R_{TO}	Tearout strength of a shear joint	lb	N
r_s	Resilience of the screw (i.e. bolt)	in./lb	mm/N
S	Sample standard deviation	Any	Any
S	Stress		
S_1; $S_t{}'$ max	Maximum principal tensile stress, under combined tension and torsion, at root diameter of bolt thread	psi	Pa
S_2; $S_t{}'$ min	Minimum principal tensile stress, under combined tension and torsion, at root diameter of bolt thread	psi	Pa
s	Scatter in preload caused by assembly tools and/or procedures	None	
s	Distance between centerline of eccentric joint and centerline of bolts	in.	mm
S_a	Allowable stress at room temperature (per ASME Code)	psi	MPa
S_B	Maximum fiber stress created by pure bending	psi	Pa
S_b	Allowable stress at service temperature (per ASME Code)	psi	MPa
SCC	Stress corrosion cracking	None	
SF	Stress factor	None	
S_H	Contained pressure	psi	MPa
S_{m1}, S_{m2}	Design in-service stresses for a gasketed joint	psi	MPa
S_{mo}	Gasket design stress: the largest of three possibilities; $2P$, S_{m1}, or S_{m2}	psi	MPa
S_{ns}	Unit shear strength of nut material	psi	Pa
S_s	Direct torsional stress for combined tensile and shear	psi	Pa
S_{ss}	Unit shear strength of bolt material	psi	Pa
$S_S{}'$	Maximum shear stress at root diameter of bolt thread under combined tension and torsion	psi	Pa
S_{st}	Unit tensile strength of bolt material	psi	Pa
S_t	Direct tensile stress component of combined tensile and shear stresses (shear resulting from torsion stress)	psi	Pa
S_U	Ultimate shear strength	psi	Pa
S_y	Yield strength	psi	Pa
S_{YP}	Value of tensile stress component (S_t) of total stress when bolt yields under combined tension and torsion	psi	Pa
S_T	Ratio of shear stress in shear plane of bolt to its ultimate tensile strength	None	
S_U	Ultimate shear strength	psi	GPa
S_{ya}	Design seating stress for a gasketed joint	psi	MPa
T	Time (ΔT = change in time or transit time)	sec	sec
T	Thickness of joint	in.	mm
T_1, T_2	Thickness of joint members 1 and 2	in.	mm
ΔT	Compression of joint members before application of an external load	in.	mm
$\Delta T'$	Compression of joint members after application of external load	in.	mm
Δt	Change in temperature	°F	°C
nT	Distance between loading planes in a joint of thickness T	in.	mm
T_1, T_{tor}	Wrench torque which gets converted to frictionally generated torsion from nut to bolt	lb-in.	N-mm
$\Delta T_\in$	Change in transit time created by bolt strain	sec	sec
TF	Temperature factor	per °F	per °C
T_H	Height of head of bolt	in.	mm
T_{in}	Torque applied to nut	lb-in.	N-mm
T_J	Thickness of joint	in.	mm
T_N	Thickness of nut	in.	mm
T_{os}	Threshold or offset torque used in LRM tool control	lb-in.	N-mm

continued

Appendix A: Units and Symbol Log (continued)

Symbol	Uses	English Units	Metric Units
T_P	Prevailing torque	lb-in.	N-mm
TP	Tightness parameter	None	
T_{pmin}	Minimum acceptable tightness parameter for a gasketed joint	None	
T_{pn}	Minimum acceptable tightness parameter for a gasketed joint taking hydrotest pressure and service temperature into account	None	
T_r	Tightness ratio (Log T_{pn}/Log T_{pmin})	None	
T_T	Ratio between tensile stress in bolt and its ultimate tensile strength	None	
T_{tf}	Torque required to overcome thread friction	lb-in.	N-mm
u	Distance from centerline of eccentric joint to edge nearest the point of application of an external load	in.	mm
v	Velocity of ultrasonic wave in bolt, in general	in./sec	cm/sec
v_0	Velocity of ultrasonic wave in unstressed bolt	in./sec	cm/sec
v_t	Velocity of ultrasonic wave in stressed bolt	in./sec	cm/sec
W	Symbol used for bolt tension ASME Code	lb	N
W_{bn}	Work done in bending bolt	ft-lb, in.-lb	N-M, N-mm
W_{in}	Input work done in tightening nut	ft-lb, in.-lb	N-M, N-mm
W_{jc}	Work done in compressing joint	ft-lb, in.-lb	N-M, N-mm
W_{M1}	Maintenance bolt load (ASME Code)	lb	N
W_{M2}	Gasket seating bolt load (ASME Code)	lb	N
W_{nc}	Work done in compressing nut	ft-lb, in.-lb	N-M, N-mm
W_{nf}	Work done against nut to joint friction	ft-lb, in.-lb	N-M, N-mm
W_{ten}	Work done in stretching bolt	ft-lb	N-M
W_{tf}	Work done against thread friction	ft-lb, in.-lb	N-M, N-mm
W_{tor}	Work done in twisting bolt	ft-lb, in.-lb	N-M, N-mm
W_{mo}	Design bolt load	lb	N
y	Recommended initial seating stress on a gasket (ASME Code)	psi	Pa
Z_p	Polar section modulus of bolt	$in.^3$	mm^3
α	Y intercept of linear regression line	Any	Any
α_A	Scatter in preload caused by assembly tools and/or procedures as defined by VDI	None	
β	Slope of linear regression line	Any (always a ratio)	Any
β	Half-angle of thread root (30° for UN or ISO metric threads)	deg	deg
Δ	"Change in," e.g., ΔL is "change in length"	None	
$\in$	Strain	Dimensionless	
θ	Turn of nut	deg	rad
$\Delta\theta$	Turn of nut created by a change in preload (ΔF_P)	deg	rad
θ_0	Starting turn used in LRM tool control	rad	rad
θ_G	Relative angle of turn between nut and "ground"	deg or rad	deg or rad
θ_{in}	Input turn of nut	deg or rad	deg or rad
θ_R	Relative angle of turn between nut and bolt threads	deg or rad	deg or rad
θ_{tw}	Angle of twist of bolt under torsion	rad	rad
μ	Coefficient of friction	None	
μ_n	Coefficient of friction between nut and joint surfaces or washer	None	
μ_S	Slip coefficient (of friction) of a shear joint	None	
μ_t	Coefficient of friction between male and female thread surfaces	None	
μ, λ, l, m	Second- and third-order elastic constants		
π	3.14159	None	
ρ	Density	$slugs/in.^3$	g/mm^3
ρ	Linear coefficient of expansion	in./in./°F	mm/mm/°C
ρ_1	Coefficient of thermal expansion of the bolt material	in./in./°F	mm/mm/°C
ρ_2	Coefficient of thermal expansion of the joint material	in./in./°F	mm/mm/°C
σ	Stress	$lb/in.^2$	Pa
σ	Standard deviation of a population	Any	Any

Appendix A: Units and Symbol Log (continued)

Symbol	Uses	English Units	Metric Units
σ_A	Fatigue endurance limit stress in the bolt	psi	MPa
σ_{av}	Average stress level in threaded section of fastener	lb/in.^{2}, psi	N/mm^2
σ_B	Maximum fiber stress caused by bending bolt	psi	Pa
σ_B	Bearing stress	psi	MPa
σ_P	Proof strength of fastener expressed as stress	psi	Pa
σ_S	Ultimate shear strength of bolt	psi	N/mm^2
σ_{SA}	Change in bolt stress caused by an external load	psi	MPa
σ_{SAb}	Change in the outer fibre of a bolt caused by bending the bolt	psi	MPa
σ_{ult}	Ultimate tensile strength	psi	MPa
σ_y	Stress required to yield the bolt	psi	MPa
Φ	Resistance factor reflecting uncertainty in strength of a shear joint	None	
Φ_e, Φ_{ek}	Load factor for an eccentrically loaded tensile joint; loaded at the joint surfaces	None	
Φ_{en}	Load factor for an eccentrically loaded tensile joint; loaded internally along loading planes	None	
Φ_k	Load factor for concentrically loaded tensile joint; loaded at the joint surfaces	None	
Φ_{kn}, Φ_n	Load factor for concentrically loaded tensile joint; loaded internally along loading planes	None	

Appendix B Glossary of Fastener and Bolted Joint Terms

Accuracy See *Preload accuracy*.

AISC The American Institute of Steel Construction.

Allowable stress The maximum stress a designer can assume that the parts will stand. It is always less than the minimum strength of the material. For example, the ASME Boiler and Pressure Vessel Code typically specifies an allowable stress that is one-quarter of the service temperature yield strength of the material. This introduces a four-to-one safety factor into the design process and is intended to compensate for uncertainties in estimates of strength, service loads, etc.

Allowable stress design A design procedure developed for the AISC by the Research Council on Structural Connections. Purposely underestimates the strengths of bolts and joint materials to introduce safety factors into the design of structural steel joints. It is an alternative to the more recently defined load and resistance factor design procedure.

Angularity The underfaces of the nut and the bolt head should be exactly perpendicular to the thread or shank axes. If the angle between the face and the axis is, for example, 86° or 94°, the fastener is said to have an angularity of 4° (sometimes called *Perpendicularity*).

Anode That electrode in a battery or corrosion cell which produces electrons. It is the electrode which is destroyed (corrodes).

Area stress or tensile stress See *Stress area*.

ASME The American Society of Mechanical Engineers.

ASME Code See *Boiler and Pressure Vessel Code*.

Barrier protection The coating on a fastener is said to provide barrier protection if it merely isolates the fastener from the environment. Paint, for example, provides barrier protection.

Body See Figure B.1.

Boiler and Pressure Vessel Code A large and complex document, maintained and published by the American Society of Mechanical Engineers (ASME—see Appendix C for address). The code describes design rules, material properties, inspection techniques, fabrication techniques, etc. for boilers and pressure vessels. The recommendations of the code have been adopted by most states, and have influenced similar codes in many countries.

Bolt Officially, a threaded fastener designed to be used with a nut. In this book, the word is often used interchangeably with *Threaded fastener* for convenience.

Bolt gage An ultrasonic instrument, manufactured by Bidwell Industrial Group (see Appendix C for address) and used to measure the stress or strain in bolts.

Bolt, parts of See Figure B.1.

Brittle A bolt is said to be brittle if it will break when stretched only a small amount past its yield point (compare *Ductile*).

Cathode That electrode in a battery or corrosion cell which attracts electrons.

Clamping force The equal and opposite forces which exist at the interface between two joint members. The clamping force is created by tightening the bolts, but is not always equal to the combined

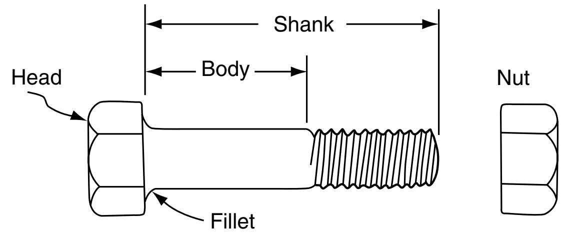


FIGURE B.1 Parts of a bolt.

tension in the bolts. Hole interference problems, for example, can create a difference between clamping force and bolt loads.

Constant life diagram A plot of experimentally derived fatigue-life data; perhaps the most complex and complete of the popular charts used to represent such data. See Figure 15.6 for an example.

Code The Boiler and Pressure Vessel Code published by the American Society of Mechanical Engineers (ASME).

Corrosion cell A natural "battery" formed when two metals having different electrical potentials (an *Anode* and a *Cathode*) are connected together in the presence of a liquid (the *Electrolyte*).

Creep The slow, plastic deformation of a body under heavy loads. Time-dependent plasticity.

Dimensions of bolt See Figure B.2.

DTI Direct tension indicator. A fastener used primarily in the structural steel industry, designed to indicate that a certain minimum amount of tension has been developed in the fastener during assembly. See Figures 9.5 and 9.6 for examples.

Ductile If a bolt can be stretched well past its yield point before breaking, it is said to be *ductile* (see also *Brittle*).

Eccentric load The *external load* on a fastener or groups of fasteners is said to be eccentric if the resultant of that load does not pass through the centroid of the group of fasteners (*eccentric shear load*) or does not coincide with the bolt axis (*eccentric tensile load*).

Effective length of a bolt The *grip length* plus some portion of the bolt (often one-half of the thickness of the nuts) which lies within the nut(s) plus some portion (often one-half the thickness) of the head. Used in stiffness and stretch calculations (see Figure 5.3).

Effective radius of nut, bolt head, or threads Distance between the geometric center of the part and the circle of points through which the resultant contact forces between mating parts passes. Must be determined by integration.

Elastic interactions When a bolt is tightened it partially compresses the joint members "in its own neighborhood." When nearby bolts are tightened later, they further compress the joint in this region. This allows the first bolt to relax a little (lose a little preload). Tightening bolts on the opposite side of the joint, however, might increase preload in some of the earlier bolts tightened on the near side. These shifts and changes in the elastic energy stored in individual bolts, during assembly, are called elastic interactions. Details can be found in Chapter 6.

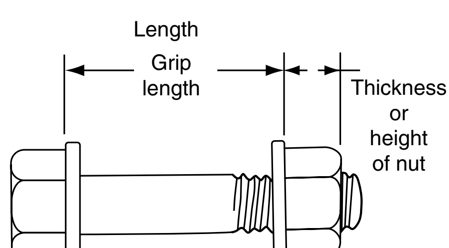


FIGURE B.2 Bolt dimensions.

Electrode The two metallic bodies in a battery or *Corrosion cell* which give up electrons (the *Anode*) or which attract them (the *Cathode*).

Electrolyte The liquid with which the *Electrodes* of a battery or *Corrosion cell* are wetted.

Embedment Localized plastic deformation in heavily loaded fasteners allows one part to sink into, or smooth the surface of, a softer or more heavily loaded second part. Nuts embed themselves in joint surfaces. Bolt threads embed themselves in nut threads, etc.

Endurance limit That completely reversing stress limit below which a bolt or joint member will have an essentially infinite life under cyclic fatigue loads. Note that the mean stress on the bolts here is zero.

Equation, long form An equation which relates the torque applied to a bolt to the preload created in it, and involves fastener geometry and the coefficient of friction between mating surfaces. A theoretical equation based on rigid body mechanics and the assumption that the geometry of the fastener is perfectly described by blueprint dimensions (see Equations 7.2 and 7.3).

Equation, short form An empirical equation which relates the torque applied to the bolt to the preload created in it, and which depends mainly on an experimentally derived factor called the *Nut factor* (see Equation 7.4).

Essential conditions Each type of failure to which bolted joints are subject is set up by three or four conditions. The conditions vary, depending on the mode of failure, but never number more than four. Eliminating any one of the essential conditions for a particular type of failure can prevent that type of failure. See Chapter 15 for specifics.

Extensometer Any instrument which measures the change in length of a part as the part is loaded.

External load Forces exerted on fastener and/or joint members by such external factors as weight, wind, inertia, vibration, temperature expansion, pressure, etc. Does not equal the *Working load* in the fastener.

Failure of the bolt Term implying that the bolt has broken or the threads have stripped. There can be many reasons for this.

Failure of the joint Failure of a bolted joint to behave as intended by the designer. Failure can be caused or accompanied by broken or lost bolts, but can also mean joint slip or leakage from a gasketed joint even if all bolts still remain whole and in place. Common reasons for joint failure include vibration loosening, poor assembly practices, improper design, unexpected service loads or conditions, etc.

Fastener dimensions See Figure B.2.

Fillet Transition region between bolt head and shank, or between other changes in diameter (see Figure B.1).

Flange rotation Angular distortion of a flange under the influence of bolt and reaction forces. Measured with respect to the center of the cross section of the flange (see Figure B.3).

Galling An extreme form of adhesive wear, in which large chunks of one part stick to the mating part (during sliding contact).

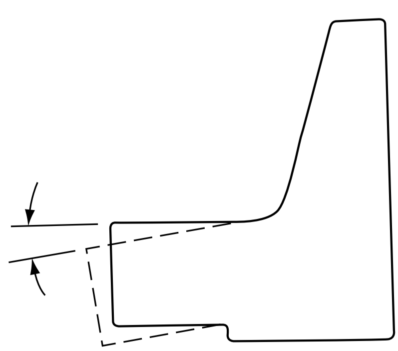


FIGURE B.3 Flange rotation.

Galvanic protection The coating on a fastener is said to provide galvanic protection if it is more anodic than the fastener and will, therefore, be destroyed instead of the fastener. Zinc plate (galvanizing) provides galvanic protection to steel fasteners, for example.

Gasket factors Experimentally derived "constants" used to define the behavior of a gasket or the assembly and in-service conditions required for acceptable behavior. The term "gasket factor" comes from the *Boiler and Pressure Vessel Code*, which contains a tabulation of m and y factors defining the recommended *Gasket stress* in-service and at assembly—for design purposes only. (Actual assembly and in-service stresses will usually be greater.) New factors, called G_b, a, and G_s, have recently been proposed for the Code. These factors are not design recommendations, but instead, define the behavior of the gasket.

Gasket stress The contact stress exerted on the gasket by the joint members.

Grip length Combined thickness of all the things clamped together by the bolt and nut, including washers, gaskets, and joint members (see *Dimensions of bolt*).

Head of bolt See Figure B.1.

Height of head or nut See *Thickness of head or nut* under *Dimensions of bolt*.

Hydrogen embrittlement A common and troublesome form of *Stress cracking*. Several theories have been proposed to explain hydrogen embrittlement, but, at present, the exact mechanism is still unknown. What is known, however, is the fact that if hydrogen is trapped in a bolt by poor electroplating practices, it can encourage stress cracking. Bolts can fail, suddenly and unexpectedly, under normal loads. See Chapter 16 for a more complete discussion.

Impact wrench An air- or electric-powered wrench in which multiple blows from tiny hammers are used to produce output torque to tighten fasteners.

Inclusions Small pieces of nonmetallic impurities trapped within the base metal of, for example, a bolt.

Infinite life diagram A simple plot experimentally derived fatigue-life data, showing the conditions required for infinite life. See Figure 15.10 for an example.

Initial preload The tension created in a single bolt as it is tightened. Will usually be modified by subsequent assembly operations (see *Elastic interactions*) or by in-service loads and conditions.

Joint diagrams Mathematical diagrams which illustrate the forces on and deflections of fasteners and joint members (see Chapters 10 and 11).

Junker machine A test machine, first proposed by Gerhard Junker, for testing the vibration resistance of fasteners (see Figure 14.11).

Length, effective See *Effective length of a bolt*.

Length of bolt See Figure B.2.

Load and resistance factor design A design procedure developed for the AISC by the Research Council on Structural Connections. Assigns uncertainties in the strength of (i.e., resistance of) and in the service loads to be placed on a shear joint to estimate the probable strength of the joint. It is a recently defined alternative to the *Allowable stress design* procedure.

Load factor (Φ) The ratio between an increase in bolt tension and the external load which has caused the increase (i.e., $\Phi = \Delta F_p/L_X$).

Load factors (α, γ) Factors reflecting the probability of an increase in load in a shear joint. Used in load and resistant factor design.

Lockbolt A fastener which bears a superficial resemblance to a bolt, but which engages a collar (instead of a nut) with annular grooves (instead of threads). The collar is swaged over the grooves on the male fastener to develop preload (see Figure 9.7).

Lock nut A nut which provides extra resistance to vibration loosening (beyond that produced by proper *Preload*), either by providing some form of *Prevailing torque*, or, in free-spinning lock nuts, by deforming, cramping, or biting into mating parts when fully tightened.

Material velocity The velocity of sound in a body (e.g., a bolt). A term used in the ultrasonic measurement of bolt stress or strain.

Mean value The average value of a number of data points. Computed by dividing the sum of all data by the number of data points.

Monitor, torque See *Torque monitor*.

Nominal diameter The "catalog diameter" of a fastener. Usually roughly equal to the diameter of the body, or the outer diameter of the threads.

Nonlinear behavior A fastener or joint system is said to exhibit nonlinear behavior when the relationship between the *External load* on the joint and deformation of the parts is nonlinear, or when the relationship between increasing *Preload* and deformation is nonlinear (see Chapter 13).

Nut factor An experimental constant used to evaluate or describe the ratio between the torque applied to a fastener and the *Preload* achieved as a result (see Equation 7.4).

Perpendicularity See *Angularity*.

Pitch The nominal distance between two adjacent thread roots or crests (see *Thread nomenclature*).

Preload The tension created in a threaded fastener when the nut is first tightened. Often used interchangeably, but incorrectly, with *Working load* or bolt force or bolt tension (see also *Clamping force*).

Preload accuracy A measure of the precision with which a given tool or procedure creates preload in a bolt when the bolt is tightened. A common torque wrench, for example, is said to produce preload with an accuracy of $\pm30\%$. The mean preload, however, may not be that which the designer intended, or may not be what he should have intended. Accuracy as used here, in other words, is synonymous with *Scatter*.

Preload, initial See *Initial Preload*.

Preload, residual See *Residual preload*.

Prevailing torque Torque required to run a nut down against the joint when some obstruction, such as a plastic insert in the threads, or a noncircular thread, or other, has been introduced to help the fastener resist vibration loosening. Prevailing torque, unlike normal torque on a nut or bolt, is not proportional to the *Preload* in the fastener.

Proof load The maximum, safe, static, tensile load which can be placed on a fastener without yielding it. Sometimes given as a force (in lb or N) sometimes as a stress (in psi or MPa).

Prying The magnification of an *External load* by a pseudolever action when that load is an *Eccentric tensile load*.

Radius, effective See *Effective radius*.

Raised-face flange A flange which contacts its mating joint member only in the region in which the gasket is located. The flanges do not contact each other at the bolt circle. Figure B.3 shows a raised-face flange.

Relaxation The loss of tension, and therefore *Clamping force*, in a bolt and joint as a result of *Embedment*, vibration loosening, gasket creep, differential thermal expansion, etc.

Residual preload The tension which remains in an unloaded bolted joint after *Relaxation*.

Resistance factor Probabilistic factor representing the uncertainties in the designer's estimate of the strength of a shear joint. Used in *Load and resistance factor design*.

Rolled thread A thread formed by plastically deforming the surface of the blank rather than by cutting operations. Increases fatigue life and thread strength, but is not possible (or perhaps economical) on larger sizes.

Rotation of flange See *Flange rotation*.

Sacrificial coating See *Galvanic protection*.

Scatter Data points or calculations are said to be scattered when they are not all the same. A "lot of scatter in preload" means wide variation in the preloads found in individual bolts.

Screw Threaded fastener designed to be used in a tapped or untapped (e.g., wood screw) hole, but not with a nut.

Self-loosening The process by which a supposedly tightened fastener becomes loose, as a result of vibration, thermal cycles, shock, or anything else which cause transverse slip between joint members and between male and female threads. *Vibration loosening* is a common, but special, case of self-loosening.

Shank That portion of a bolt which lies under the head (see *Bolts, parts of*).

Shear joint A joint which is subjected primarily to loads acting more or less perpendicular to the axes of the bolts.

Slug wrench A box wrench with an anvil on the end of the handle. Torque is produced by striking the anvil with a sledge hammer. Called a flogging wrench in England.

Sonic velocity See *Material velocity*.

Spherical washer A washer whose upper surface is semispherical. Used with a nut whose contact face is also semispherical. Reduces bending stress in a bolt or stud, by allowing some self-alignment and some compensation for nonparallel joint surfaces or *Angularity*.

Spring constant The ratio between the forces exerted on a spring (or a bolt) and the deflection thereof. Has the dimensions of force per unit change in length (e.g., lb/in.). Also called *Stiffness*.

Standard deviation A statistical term used to quantify the *Scatter* in a set of data points. If the standard deviation is small, most of the data points are ‘‘nearly equal.’’ A large deviation means less agreement.

Stiffness See *Spring constant*.

Strength of bolt An ambiguous term which can mean *Ultimate strength* or *Proof load* or *Endurance limit* or *Yield strength*.

Stress area The effective cross-sectional area of the threaded section of a fastener. Used to compute average stress levels in that section. Based on the mean of pitch and minor diameters (see *Thread nomenclature*).

Stress corrosion cracking (SCC) A common form of *Stress cracking* in which an *Electrolyte* encourages the growth of a crack in a highly stressed bolt. Only a tiny quantity of electrolyte need be present, at the tip or face of the crack.

Stress cracking A family of failure modes, each of which involves high stress and chemical action. The family includes *Hydrogen embrittlement*, *Stress corrosion cracking*, stress embrittlement, and hydrogen-assisted stress corrosion. See Chapter 16 for details.

Stress factor A calibration constant used in ultrasonic measurement of bolt stress or strain. It is the ratio between the change in ultrasonic transit time caused by the change in length of the fastener, under load, to the total change in transit time (which is also affected by a change in the stress level).

Stress relaxation The slow decrease in stress level within a part (e.g., a bolt) which is heavily loaded under constant deflection conditions. A ‘‘cousin’’ to creep, which is a slow change in geometry under constant stress conditions.

Stud A headless threaded fastener, threaded on both ends, with an unthreaded body in the middle section, or threaded from end to end. Used with two nuts, or with one nut and a tapped hole.

Temperature factor A calibration constant used in ultrasonic measurement of bolt stress or strain. Accounts for the effects of thermal expansion and the temperature-induced change in the velocity of sound.

Tensile strength See *Ultimate strength*.

Tensile stress area See *Stress area*.

Tension, bolt Tension (tensile stress) created in the bolt by assembly preloads and/or such things as thermal expansion, service loads, etc.

Tension joint A joint which is primarily subjected to loads acting more or less parallel to the axes of the bolts.

Tensioner A hydraulic tool used to tighten a fastener by stretching it rather than by applying a substantial torque to the nut. After the tension has stretched the bolt or stud, the nut is run down against the joint with a modest torque, and the tensioner is disengaged from the fastener. The nut holds the stretch produced by the tensioner.

Thickness of nut or of bolt head See *Dimensions of bolt* (also called *Height*).

Thread form The cross-sectional shape of the threads, defining thread angle, root, and crest profiles, etc.

Thread length Length of that portion of the fastener which contains threads cut or rolled to full depth (see Figure B.2).

Thread nomenclature See Figure B.4.

Thread run-out That portion of the threads which are not cut or rolled full depth, but which provide the transition between full-depth threads and the body or head (see Figure B.1). Officially called thread washout or vanish, although the term run-out is more popular. (Run-out is officially reserved for rotational eccentricity, as defined by total indicator readings or the like.)

Threaded fastener Studs, bolts, and screws of all sorts, with associated nuts. One of the most interesting, complex, useful—and frustrating—components yet devised.

Tightness A measure of the mass leak rate from a gasketed joint.

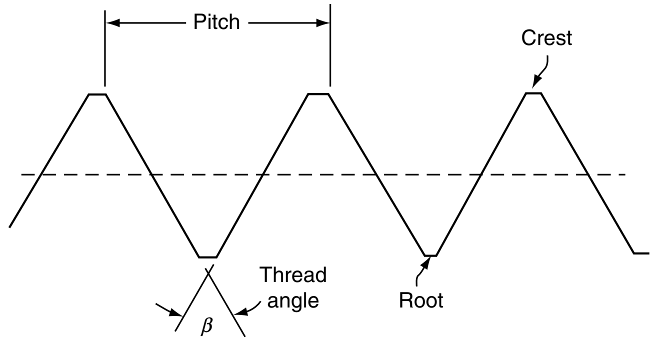


FIGURE B.4 Thread nomenclature.

Tightness, acceptable Wholly leak-free joints are impossible, at least if the contained fluid is a gas, so it has been proposed that the design of a gasketed joint should start with the selection of an "acceptable" leak rate. The designer would dimension bolts and joint members so that the actual leak rate would never exceed this. Three standard levels of tightness have been proposed as well.

Tightness parameter A dimensionless parameter which defines the mass leakage of a gasket as a function of contained pressure and a contained fluid constant.

Torque The twisting moment, product of force and wrench length, applied to a nut or bolt (for example).

Torque monitor A torque tool control system which monitors the amount of torque being developed by the tool during use, but does not control the tool or the torque produced.

Torque multiplier A gearbox used to multiply the torque produced by a small hand wrench (usually a *Torque wrench*). The output of the multiplier drives the nut or bolt with a torque that is higher, and a speed that is lower, than input torque and speed. There is no torque gage or readout on the multiplier.

Torque pack A geared wrench which multiplies input torque and provides a read-out of output torque. In effect, a combination of a *Torque wrench* and a *Torque multiplier*.

Torque wrench A manual wrench which incorporates a gage or measuring apparatus of some sort to measure and display the amount of torque being delivered to the nut or bolt. All wrenches produce torque. Only a torque wrench tells how much torque.

Transducer A device which converts one form of energy into another. An ultrasonic transducer, for example, converts electrical energy into acoustic energy (at ultrasonic frequencies) and vice versa.

Turn-of-nut Sometimes used to describe the general rotation of the nut (or bolt head) as the fastener is tightened. More often used to define a particular tightening procedure in which a fastener is first tightened with a preselected torque, and is then tightened further by giving the nut an additional, measured, turn such as "three flats" (180°).

Ultimate strength The maximum tensile strength a bolt or material can support prior to rupture. Always found in the plastic region of the stress–strain or force–elongation curve, and so is not a design strength. Also called *Tensile strength* and ultimate tensile strength.

Ultrasonic extensometer An electronic instrument which measures the change in length of a fastener ultrasonically as, or before and after, the fastener is tightened (see also *Extensometer*).

Washer, tension indicating See *DTI*.

Width across flats A principal dimension of nuts, or of bolt heads (see Figure B.5).

Work hardening The slight increase in hardness and strength produced when a body is loaded past its yield point. Also called strain hardening.

Working load The tension in a bolt in use; tension produced by a combination of *Residual preload* and a portion (usually) of any *External load*. The *Joint diagram* is usually used to predict the approximate working load a fastener will see in service.

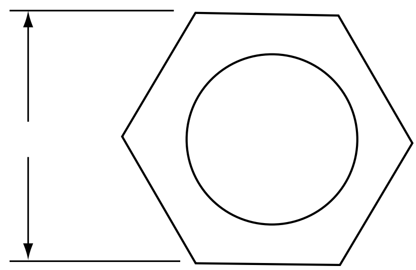


FIGURE B.5 Width across flats.

Yield strength That stress level which will create a permanent deformation of 0.2% or 0.5% or some other small, preselected, amount in a body. Approximately equal to the elastic and proportional limits of the material; a little higher than the proof strength of a bolt (see *Proof load*).

Appendix C
Sources of Bolting Information and Standards

AISC	American Institute of Steel Construction, Inc. 400 North Michigan Ave. Chicago, IL 60611 http://www.aisc.com
AISI	American Iron and Steel Institute 1000 16th Street N.W. Washington, DC 20036 http://www.steel.org
AMS	Obtain Aeronautical Material Specifications from SAE, the Society of Automotive Engineers (see below)
ANSI	Obtain American National Standards Institute standards from the ASME (see below) or from www.asme.org
ASME	American Society of Mechanical Engineers United Engineering Center 345 East 47th Street New York, NY 10017 www.asme.org
ASTM	ASTM International 100 Barr Harbor Drive Post Office Box C700 West Conshohocken, PA 19428–2959 www.astm.org
BSI	British Standards Institution BSI Library and Bookshop BSI House 389 Chiswick High Road London W4 4AL http://www.bsi-global.com
FF QQ	Obtain Federal Specifications from: Commanding Officer

GGG	DAPS Philadelphia 5801 Tabor Avenue Philadelphia, PA 19120 http://assist.daps.dla.mil Assist-Quicksearch
IFI	Industrial Fasteners Institute 1505 East Ohio Building Cleveland, OH 44114 http://www.industrial-fasteners.org
MIL	MIL SPECS DOD www.nssn.org Also see FF etc. above
NAS	National Standards Association See ANSI above
PVRC	Pressure Vessel Research Council
WRC	c/o Welding Research Council http://www.forengineers.org or mprager@forengineers.org
SAE	SAE International 400 Commonwealth Drive Warrendale, PA 15096 www.sae.org

Also see Table 14.1 which lists the web sites of organizations providing information and/or products about self-loosening and vibration resistant fasteners.

Appendix D
English and Metric Conversion Factors

Conversions for	To Obtain	Multiply Number of	By
Length, diameter, etc.	mm	in.	25.4
	in.	mm	0.0394
	m	ft	0.305
	ft	m	3.28
Force	N	kgf	9.81
	kgf	N	0.102
	N	lb	4.448
	lb	N	0.225
	kgf	lb	0.454
	lb	kgf	2.203
	g	lb	454
	lb	g	0.0022
	oz	g	0.0352
	g	oz	28.3
Torque	g-cm	lb-in.	0.000868
	lb-in.	g-cm	1150
	g-cm	oz-in.	0.0139
	oz-in.	g-cm	71.88
	N-m	lb-ft	1.36
	lb-ft	N-m	0.738
	kgf-m	lb-ft	0.138
	lb-ft	kgf-m	7.23
Pressure or stress	MPa	kgf/mm^2	9.804
	kgf/mm^2	MPa	0.102
	N/mm^2	MPa	1
	ksi	MPa	0.145
	MPa	ksi	6.895
	ksi	kgf/mm^2	1.42
	kgf/mm^2	ksi	0.704
	MPa	psi	6.895×10^{-3}
	N/m^2	Pa	1
	ksi	N/mm^2	0.145
	N/mm^2	ksi	6.895

Appendix E
Tensile Stress Areas for English and Metric Threads with Estimated "Typical" Preloads and Torques for As-Received Steel Fasteners

E.1 GENERAL INSTRUCTIONS

As suggested in Chapter 7, selecting a torque value for a given fastener can be a complex problem. Tables of recommended torque values, however, are common and popular. They're safe to use in noncritical applications or such tasks as selecting a tool of the appropriate size. The following table is based on the following assumptions:

1. The fasteners are commercial grade and made of steel.
2. The nut factor (K) is 0.2; i.e., the fasteners are used in as-received condition and are neither cleaned nor lubricated.
3. The fasteners will be tightened by applying torque to the nut, not to the head.
4. The fasteners are tightened to an average stress (in the threaded section) of 25,000 psi (or its metric equivalent of 172.4 MPa).

E.2 TORQUES FOR DIFFERENT LUBRICANTS OR STRESS LEVELS

If you're using a lubricant, or want an average stress different than 25 ksi, you can compute the new torque value from the equation

$$T = \frac{K}{0.2} \times \frac{\sigma}{25{,}000} \times T_{\mathrm{T}}$$

where

T = corrected torque (in.-lb)

K = nut factor for your application

σ = average stress in thread region desired in your application (psi only; see below for metric)

T_{T} = torque value given in this table (in.-lb)

Example: I want to lubricate a 1/4–20 screw with moly, and tighten it to 60 ksi stress in the threads. What torque should I use?

The nut factor, K, for the moly I'm using is, typically, 0.137 (see Table 7.1). So my new torque T will be

$$T = \frac{0.137}{0.2} \times \frac{60{,}000}{25{,}000} \times 39.75 = 65.3 \text{ in.-lb}$$

A similar equation is used for metric units:

$$T = \frac{K}{0.2} \times \frac{\sigma}{172.4} \times T_T$$

where
- T = corrected torque (N-m)
- K = nut factor (Table 7.1)
- σ = average thread stress desired (MPa)
- T_T = torque from this table (N-m)

E.3 PRELOADS AND STRESSES FOR DIFFERENT LUBRICANTS OR TORQUES

In a similar fashion, you can compute the preload achieved with a different lubricant and/or torque as follows:

English or metric:

$$F_P = \frac{0.2}{K} \times \frac{T}{T_T} \times F_{PT}$$

where
- F_P = corrected preload (kip, kN)
- K = nut factor for your application
- T = torque for your application (in.-lb, N-m)
- T_T = torque from this table (in.-lb, N-m)
- F_{PT} = preload from this table (in.-lb, N-m)

If you wish to compute the new stress produced by this new preload, divide the new preload by the tensile stress area given in the table.

Example: I want to compute the preload produced in a 1/4–20 screw by a torque of 65.3 in.-lb, if I use moly lube. Then I want to compute the resulting stress level in the fastener.

$$F_P \frac{0.2}{0.137} \times \frac{65.3}{39.75} \times 0.795 = \ 1.907 \text{ kip}$$

The new stress will be

$$\sigma = \frac{1.907}{0.0318} = 60 \text{ ksi}$$

which agrees with our first example.

E.4 TORQUE UNITS

Note that the torques listed in the table are in inch-pounds, not foot-pounds because the short-term torque equation

$$T = KDF$$

used to derive the table computes torque in inch-pounds unless the nominal diameter D is measured in feet. I've seen many errors in the use of this equation where people dealing with large bolts (and used to working in foot-pounds) automatically expressed D in inches and then took the answer as foot-pounds. So, to avoid this I've used inch-pounds in the table.

If you wish to convert the torque values I've given to foot-pounds, I think it's reasonable to divide the value in the table by 10, even though division by 12 would be theoretically proper. But tables of "recommended torques" can never do more than approximate the right torque for your application, so you'll probably get as "good" an answer dividing by 10 as you would be dividing by 12.

In any event, use tables such as this with caution!

English Tensile Stress Areas

Size	Series	Tensile Stress Area A_S (in.^{2})	Preload (F_p) at 25 ksi (kip)	Torque to Achieve 25 ksi (in.-lb)
0–80	UNF	0.00180	0.045	0.54
1–64	UNC	0.00263	0.0658	0.96
1–72	UNF	0.00278	0.0695	1.01
2–56	UNC	0.00370	0.0925	1.59
2–64	UNF	0.00394	0.0985	1.69
3–48	UNC	0.00487	0.122	2.42
3–56	UNF	0.00523	0.131	2.59
4–40	UNC	0.00604	0.151	3.38
4–48	UNF	0.00661	0.165	3.70
5–40	UNC	0.00796	0.199	4.98
5–44	UNF	0.00830	0.208	5.2
6–32	UNC	0.00909	0.227	6.27
6–40	UNF	0.01015	0.254	7.01
8–32	UNC	0.0140	0.35	11.5
8–36	UNF	0.01474	0.369	12.1
10–24	UNC	0.0175	0.438	16.6
10–32	UNF	0.0200	0.5	19.0
12–24	UNC	0.0242	0.605	26.1
12–28	UNF	0.0258	0.645	27.9
12–32	UNEF	0.0270	0.675	29.2
1/4–20	UNC	0.0318	0.795	39.75
1/4–28	UNF	0.0364	0.91	45.5
1/4–32	UNEF	0.0379	0.948	47.4
5/16–18	UNC	0.0524	1.310	82.0
5/16–20	UN	0.0547	1.367	85.6
5/16–24	UNF	0.0580	1.450	90.8
5/16–28	UN	0.0606	1.515	94.8
5/16–32	UNEF	0.0625	1.563	97.8
3/8–16	UNC	0.0775	1.938	145.4
3/8–20	UN	0.0836	2.090	156.8
3/8–24	UNF	0.0878	2.195	164.6

continued

English Tensile Stress Areas (continued)

Size	Series	Tensile Stress Area A_S (in.^{2})	Preload (F_p) at 25 ksi (kip)	Torque to Achieve 25 ksi (in.-lb)
3/8–28	UN	0.0909	2.273	170.5
3/8–32	UNEF	0.0932	2.330	174.8
7/16–14	UNC	0.1063	2.658	232.8
7/16–16	UN	0.1114	2.785	244
7/16–20	UNF	0.1187	2.968	260
7/16–28	UNEF	0.1274	3.185	279
7/16–32	UN	0.1301	3.253	285
1/2–13	UNC	0.1419	3.548	354.8
1/2–16	UN	0.151	3.775	378
1/2–20	UNF	0.1599	3.998	399.8
1/2–28	UNEF	0.170	4.250	425
1/2–32	UN	0.173	4.325	433
9/16–12	UNC	0.182	4.550	512
9/16–16	UN	0.198	4.950	557
9/16–18	UNF	0.203	5.075	571
9/16–20	UN	0.207	5.175	583
9/16–24	UNEF	0.214	5.350	602
9/16–28	UN	0.219	5.475	616
9/16–32	UN	0.222	5.550	625
5/8–11	UNC	0.226	5.650	706
5/8–12	UN	0.232	5.800	725
5/8–16	UN	0.250	6.25	781
5/8–18	UNF	0.256	6.400	800
5/8–20	UN	0.261	6.525	816
5/8–24	UNEF	0.268	6.700	838
5/8–28	UN	0.274	6.850	856
5/8–32	UN	0.278	6.950	869
13/16–12	UN	0.289	7.225	994
11/16–16	UN	0.308	7.700	1,060
11/16–20	UN	0.320	8.000	1,101
11/16–24	UNEF	0.329	8.225	1,132
11/16–28	UN	0.335	8.375	1,152
11/16–32	UN	0.339	8.475	1,166
3/4–10	UNC	0.334	8.350	1,253
3/4–12	UN	0.351	8.775	1,316
3/4–16	UNF	0.373	9.325	1,399
3/4–20	UNEF	0.386	9.650	1,448
3/4–28	UN	0.402	10.05	1,508
3/4–32	UN	0.407	10.18	1,527
13/16–12	UN	0.420	10.50	1,707
13/16–16	UN	0.444	11.10	1,805
13/16–20	UNEF	0.458	11.45	1,862
13/16–28	UN	0.475	11.88	1,932
13/16–32	UN	0.480	12.00	1,951
7/8–9	UNC	0.462	11.55	2,021
7/8–12	UN	0.495	12.38	2,167
7/8–14	UNF	0.509	12.73	2,228
7/8–16	UN	0.521	13.03	2,280
7/8–20	UNEF	0.536	13.40	2,345
7/8–28	UN	0.554	13.85	2,424
7/8–32	UN	0.560	14.00	2,450
15/16–12	UN	0.576	14.40	2,701
15/16–16	UN	0.604	15.10	2,833
15/16–20	UNEF	0.620	15.50	2,908
15/16–28	UN	0.640	16.00	3,002

English Tensile Stress Areas (continued)

Size	Series	Tensile Stress Area A_S (in.^{2})	Preload (F_p) at 25 ksi (kip)	Torque to Achieve 25 ksi (in.-lb)
15⁄16–32	UN	0.646	16.15	3,030
1–8	UNC	0.606	15.15	3,030
1–12	UNF	0.663	16.58	3,316
1–16	UN	0.693	17.33	3,466
1–20	UNEF	0.711	17.78	3,556
1–28	UN	0.732	18.30	3,660
1–32	UN	0.738	18.45	3,690
1 1⁄16–8	UN	0.695	17.38	3,693
1 1⁄16–12	UN	0.756	18.90	4,016
1 1⁄16–16	UN	0.788	19.70	4,186
1 1⁄16–18	UNEF	0.799	19.98	4,246
1 1⁄16–20	UN	0.807	20.18	4,288
1 1⁄16–28	UN	0.830	20.75	4,409
1 1⁄8–7	UNC	0.763	19.08	4,293
1 1⁄8–8	UN	0.790	19.75	4,444
1 1⁄8–12	UNF	0.856	21.40	4,815
1 1⁄8–16	UN	0.889	22.23	5,002
1 1⁄8–18	UNEF	0.901	22.53	5,069
1 1⁄8–20	UN	0.910	22.75	5,119
1 1⁄8–28	UN	0.933	23.33	5,249
1 3⁄16–8	UN	0.892	22.30	5,296
1 3⁄16–12	UN	0.961	24.03	5,707
1 3⁄16–16	UN	0.997	24.93	5,921
1 3⁄16–18	UNEF	1.009	25.23	5,992
1 3⁄16–20	UN	1.018	25.45	6,044
1 3⁄16–28	UN	1.044	26.10	6,199
1 1⁄4–7	UNC	0.969	24.23	6,058
1 1⁄4–8	UN	1.000	25.00	6,250
1 1⁄4–12	UNF	1.073	26.83	6,708
1 1⁄4–16	UN	1.111	27.78	6,945
1 1⁄4–18	UNEF	1.123	28.08	7,020
1 1⁄4–20	UN	1.133	28.33	7,083
1 1⁄4–28	UN	1.160	29.00	7,250
1 5⁄16–8	UN	1.114	27.85	7,311
1 5⁄16–12	UN	1.191	29.78	7,817
1 5⁄16–16	UN	1.230	30.75	8,072
1 5⁄16–18	UNEF	1.244	31.10	8,164
1 5⁄16–20	UN	1.254	31.35	8,229
1 5⁄16–28	UN	1.282	32.05	8,413
1 3⁄8–6	UNC	1.155	28.88	7,942
1 3⁄8–8	UN	1.233	30.83	8,478
1 3⁄8–12	UNF	1.315	32.88	9,042
1 3⁄8–16	UN	1.356	33.90	9,323
1 3⁄8–18	UNEF	1.370	34.25	9,419
1 3⁄8–20	UN	1.382	34.55	9,501
1 3⁄8–28	UN	1.411	35.28	9,702
1 7⁄16–6	UN	1.277	31.93	9,180
1 7⁄16–8	UN	1.360	34.00	9,775
1 7⁄16–12	UN	1.445	36.13	10,387
1 7⁄16–16	UN	1.488	37.20	10,695
1 7⁄16–18	UNEF	1.503	37.58	10,804
1 7⁄16–20	UN	1.51	37.75	10,853
1 7⁄16–28	UN	1.55	38.75	11,141
1 1⁄2–6	UNC	1.405	35.13	10,539

continued

English Tensile Stress Areas (continued)

Size	Series	Tensile Stress Area A_S (in.^{2})	Preload (F_p) at 25 ksi (kip)	Torque to Achieve 25 ksi (in.-lb)
1½–8	UN	1.492	37.30	11,190
1½–12	UNF	1.581	39.53	11,859
1½–16	UN	1.63	40.75	12,225
1½–18	UNEF	1.64	41.00	12,300
1½–20	UN	1.65	41.25	12,375
1½–28	UN	1.69	42.25	12,675
1 9/16–6	UN	1.54	38.50	12,031
1 9/16–8	UN	1.63	40.75	12,734
1 9/16–12	UN	1.72	43.00	13,438
1 9/16–16	UN	1.77	44.25	13,828
1 9/16–18	UNEF	1.79	44.75	13,984
1 9/16–20	UN	1.80	45.00	14,063
1⅝–6	UN	1.68	42.00	13,650
1⅝–8	UN	1.78	44.50	14,463
1⅝–12	UN	1.87	46.75	15,194
1⅝–16	UN	1.92	48.00	15,600
1⅝–18	UNEF	1.94	48.50	15,763
1⅝–20	UN	1.95	48.75	15,844
1 11/16–6	UN	1.83	45.75	15,441
1 11/16–8	UN	1.93	48.25	16,284
1 11/16–12	UN	2.03	50.75	17,128
1 11/16–16	UN	2.08	52.00	17,550
1 11/16–18	UNEF	2.10	52.50	17,719
1 11/16–20	UN	2.11	52.75	17,803
1¾–5	UNC	1.90	47.50	16,625
1¾–6	UN	1.98	49.50	17,325
1¾–8	UN	2.08	52.00	18,200
1¾–12	UN	2.19	54.75	19,163
1¾–16	UN	2.24	56.00	19,600
1¾–20	UN	2.27	56.75	19,863
1 13/16–6	UN	2.14	53.50	19,394
1 13/16–8	UN	2.25	56.25	20,391
1 13/16–12	UN	2.35	58.75	21,297
1 13/16–16	UN	2.41	60.25	21,841
1 13/16–20	UN	2.44	61.00	22,113
1⅞–6	UN	2.30	57.50	21,563
1⅞–8	UN	2.41	60.25	22,594
1⅞–12	UN	2.53	63.25	23,719
1⅞–16	UN	2.58	64.50	24,188
1⅞–20	UN	2.62	65.50	24,563
1 15/16–6	UN	2.47	61.75	23,928
1 15/16–8	UN	2.59	64.75	25,091
1 15/16–12	UN	2.71	67.75	26,253
1 15/16–16	UN	2.77	69.25	26,834
1 15/16–20	UN	2.80	70.00	27,125
2–4½	UNC	2.50	62.50	25,000
2–6	UN	2.65	66.25	26,500
2–8	UN	2.77	69.25	27,700
2–12	UN	2.89	72.25	28,900
2–16	UN	2.95	73.75	29,500
2–20	UN	2.99	74.75	29,900
2⅛–6	UN	3.03	75.75	32,194
2⅛–8	UN	3.15	78.75	33,469
2⅛–12	UN	3.28	82.00	34,850

English Tensile Stress Areas (continued)

Size	Series	Tensile Stress Area A_S (in.^{2})	Preload (F_p) at 25 ksi (kip)	Torque to Achieve 25 ksi (in.-lb)
2 1/8–16	UN	3.35	83.75	35,594
2 1/8–20	UN	3.39	84.75	36,019
2 1/4–4 1/2	UNC	3.25	81.25	36,563
2 1/4–6	UN	3.42	85.50	38,475
2 1/4–8	UN	3.56	89.00	40,050
2 1/4–12	UN	3.69	92.25	41,513
2 1/4–16	UN	3.76	94.00	42,300
2 1/4–20	UN	3.81	95.25	42,863
2 3/8–6	UN	3.85	96.25	45,719
2 3/8–8	UN	3.99	99.75	47,381
2 3/8–12	UN	4.13	103.3	49,068
2 3/8–16	UN	4.21	105.3	50,018
2 3/8–20	UN	4.25	106.3	50,493
2 1/2–4	UNC	4.00	100.0	50,000
2 1/2–6	UN	4.29	107.3	53,650
2 1/2–8	UN	4.44	111.0	55,500
2 1/2–12	UN	4.60	115.0	57,500
2 1/2–16	UN	4.67	116.8	58,400
2 1/2–20	UN	4.72	118.0	59,000
2 5/8–4	UN	4.45	111.3	58,433
2 5/8–6	UN	4.76	119.0	62,475
2 5/8–8	UN	4.92	123.0	64,575
2 5/8–12	UN	5.08	127.0	66,675
2 5/8–16	UN	5.16	129.0	67,725
2 5/8–20	UN	5.21	130.3	68,408
2 3/4–4	UNC	4.93	123.3	67,815
2 3/4–6	UN	5.26	131.5	72,325
2 3/4–8	UN	5.43	135.8	74,690
2 3/4–12	UN	5.59	139.8	76,890
2 3/4–16	UN	5.68	142.0	78,100
2 3/4–20	UN	5.73	143.25	78,788
2 7/8–4	UN	5.44	136	78,200
2 7/8–6	UN	5.78	144.5	83,088
2 7/8–8	UN	5.95	148.8	85,560
2 7/8–12	UN	6.13	153.3	88,148
2 7/8–16	UN	6.22	155.5	89,413
2 7/8–20	UN	6.27	156.8	90,160
3–4	UNC	5.97	149.3	89,580
3–6	UN	6.33	158.3	94,980
3–8	UN	6.51	162.8	97,680
3–12	UN	6.69	167.3	100,380
3–16	UN	6.78	169.5	101,700
3–20	UN	6.84	171.0	102,600
3 1/8–4	UN	6.52	163	101,875
3 1/8–6	UN	6.89	172.3	107,688
3 1/8–8	UN	7.08	177.0	110,625
3 1/8–12	UN	7.28	182.0	113,750
3 1/8–16	UN	7.37	184.3	115,188
3 1/4–4	UNC	7.10	177.5	115,375
3 1/4–6	UN	7.49	187.3	121,745
3 1/4–8	UN	7.69	192.3	124,995
3 1/4–12	UN	7.89	197.3	128,245
3 1/4–16	UN	7.99	199.8	129,870
3 3/8–4	UN	7.70	193	130,275

continued

English Tensile Stress Areas (continued)

Size	Series	Tensile Stress Area A_S (in.^{2})	Preload (F_p) at 25 ksi (kip)	Torque to Achieve 25 ksi (in.-lb)
3⅜–6	UN	8.11	202.8	136,890
3⅜–8	UN	8.31	207.8	140,265
3⅜–12	UN	8.52	213.0	143,780
3⅜–16	UN	8.63	215.8	145,665
3½–4	UNC	8.33	208.3	145,800
3½–6	UN	8.75	218.8	153,200
3½–8	UN	8.96	224.0	156,800
3½–12	UN	9.18	229.5	160,650
3½–16	UN	9.29	232.3	162,610
3⅝–4	UN	9.00	225	163,125
3⅝–6	UN	9.42	235.5	170,738
3⅝–8	UN	9.64	241.0	174,725
3⅝–12	UN	9.86	246.5	178,713
3⅝–16	UN	9.98	249.5	180,888
3¾–4	UNC	9.66	241.5	181,125
3¾–6	UN	10.11	252.8	189,600
3¾–8	UN	10.34	258.5	193,875
3¾–12	UN	10.57	264.3	198,225
3¾–16	UN	10.69	267.3	200,475
3⅞–4	UN	10.36	259	200,725
3⅞–6	UN	10.83	270.8	209,870
3⅞–8	UN	11.06	276.5	214,288
3⅞–12	UN	11.30	282.5	218,938
3⅞–16	UN	11.43	285.8	221,495
4–4	UNC	11.08	277.0	221,600
4–6	UN	11.57	289.3	231,440
4–8	UN	11.81	295.3	236,240
4–12	UN	12.06	301.5	241,200
4–16	UN	12.19	304.8	243,840
4⅛–4	UN	11.83	296	244,200
4⅛–6	UN	12.33	308.3	254,348
4⅛–8	UN	12.59	315	259,875
4⅛–12	UN	12.84	321.0	264,825
4⅛–16	UN	12.97	324.3	267,547
4¼–4	UN	12.61	315.3	268,005
4¼–6	UN	13.12	328.0	278,800
4¼–8	UN	13.38	335	284,750
4¼–12	UN	13.65	341.3	290,105
4¼–16	UN	13.78	344.5	292,825
4⅜–4	UN	13.41	335	293,125
4⅜–6	UN	13.94	348.5	304,938
4⅜–8	UN	14.21	355	310,625
4⅜–12	UN	14.48	362	316,750
4⅜–16	UN	14.62	365.5	319,813
4½–4	UN	14.23	355.8	320,220
4½–6	UN	14.78	369.5	332,550
4½–8	UN	15.1	378	340,200
4½–12	UN	15.3	382.5	344,250
4½–16	UN	15.5	387.5	349,200
4⅝–4	UN	15.1	378	349,650
4⅝–6	UN	15.6	390	360,750
4⅝–8	UN	15.9	398	368,150
4⅝–12	UN	16.2	405	374,625
4⅝–16	UN	16.4	410	379,250

English Tensile Stress Areas (continued)

Size	Series	Tensile Stress Area A_S (in.^{2})	Preload (F_p) at 25 ksi (kip)	Torque to Achieve 25 ksi (in.-lb)
4¾–4	UN	15.9	397.5	377,625
4¾–6	UN	16.5	412.5	391,875
4¾–8	UN	16.8	420	399,000
4¾–12	UN	17.1	427.5	406,125
4¾–16	UN	17.3	432.5	410,875
4⅞–4	UN	16.8	420	409,500
4⅞–6	UN	17.5	437.5	426,563
4⅞–8	UN	17.7	443	431,925
4⅞–12	UN	18.0	450	438,750
4⅞–16	UN	18.2	455	443,625
5–4	UN	17.8	445	445,000
5–6	UN	18.4	460	460,000
5–8	UN	18.7	468	468,000
5–12	UN	19.0	475	475,000
5–16	UN	19.2	480	480,000
5⅛–4	UN	18.7	468	479,700
5⅛–6	UN	19.3	482.5	494,563
5⅛–8	UN	19.7	493	505,325
5⅛–12	UN	20.0	500	512,500
5⅛–16	UN	20.1	502.5	515,063
5¼–4	UN	19.7	492.5	517,125
5¼–6	UN	20.3	507.5	532,875
5¼–8	UN	20.7	518	543,900
5¼–12	UN	21.0	525	551,250
5¼–16	UN	21.1	527.5	553,875
5⅜–4	UN	20.7	518	556,850
5⅜–6	UN	21.3	532.5	572,438
5⅜–8	UN	21.7	543	583,725
5⅜–12	UN	22.0	550	591,250
5⅜–16	UN	22.2	555	596,625
5½–4	UN	21.7	542.5	596,750
5½–6	UN	22.4	560	616,000
5½–8	UN	22.7	568	624,800
5½–12	UN	23.1	577.5	635,250
5½–16	UN	23.2	580	638,000
5⅝–4	UN	22.7	568	639,000
5⅝–6	UN	23.4	585	658,125
5⅝–8	UN	23.8	595	669,375
5⅝–12	UN	24.1	602.5	677,813
5⅝–16	UN	24.3	607.5	683,438
5¾–4	UN	23.8	595	684,250
5¾–6	UN	24.5	612.5	704,375
5¾–8	UN	24.9	623	716,450
5¾–12	UN	25.2	630	724,500
5¾–16	UN	25.4	635	730,250
5⅞–4	UN	24.9	623	732,025
5⅞–6	UN	25.6	640	752,000
5⅞–8	UN	26.0	650	763,750
5⅞–12	UN	26.4	660	775,500
5⅞–16	UN	26.5	662.5	778,438
6–4	UN	26	650	780,000
6–6	UN	26.8	670	804,000
6–8	UN	27.1	678	813,600
6–12	UN	27.5	687.5	825,600
6–16	UN	27.7	692.5	831,000

Metric Tensile Stress Areas

Size	Series	Tensile Stress Area A_S (mm^2)	Preload (F_p) at 172.4 MPa (kN)	Torque to Achieve 172.4 MPa (N-m)
M1.6 × 0.35		1.27	0.219	0.070
M2 × 0.4		2.07	0.357	0.1428
M2.5 × 0.45		3.39	0.584	0.292
M3 × 0.5		5.03	0.867	0.5202
M3.5 × 0.6		6.78	1.169	0.8184
M4 × 0.7		8.78	1.514	1.211
M5 × 0.8		14.2	2.448	2.448
M6 × 1		20.1	3.465	4.158
M6.3 × 1		22.6	3.896	4.908
M8 × 1.25		36.6	6.310	10.1
M10 × 1.5		58.0	9.999	20.0
M12 × 1.75		84.3	14.533	34.9
M14 × 2		115	19.826	55.5
M16 × 2		157	27.067	86.6
M20 × 2.5		245	42.24	168.96
M24 × 3		353	60.96	292.6
M30 × 3.5		561	96.72	580.4
M36 × 4		817	140.85	1,014
M42 × 4.5		1,120	191.71	1,613
M48 × 5		1,470	253.4	2,433
M56 × 5.5		2,030	349.97	3,920
M64 × 6		2,680	462.03	5,914
M72 × 6		3,460	596.5	8,597
M80 × 6		4,340	748.22	11,968
M90 × 6		5,590	963.72	17,352
M100 × 6		6,990	1,205.08	24,100

Appendix F
Basic Head, Thread, and Nut Lengths

Most of the bolts described in these tables are dimensioned as indicated in Figure F.1. An additional dimension, thread run-out length (L_{ro}), is given for some types of bolt (see Figure F.2). Some portion of the run-out length, perhaps one-half for the shortest bolts, 90% or so for the longest, should be added to nominal thread length in stiffness or stretch calculations.

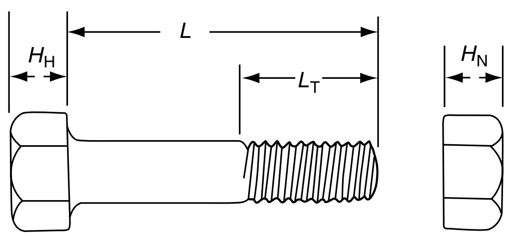


FIGURE F.1 Bolt dimensions. Note that body length L_B equals basic length L minus thread length L_T.

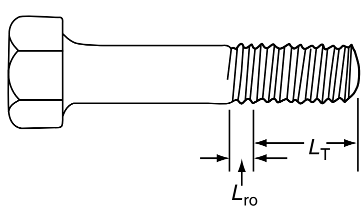


FIGURE F.2 Thread run-out length L_{ro}.

Type: Square Bolts
Reference: ANSI B18.2.1-1972
Dimensions are in: Inches

		Thread Length L_T	
Nominal Diameter D	**Head Height H_H**	**Bolts 6 in. and Shorter**	**Bolts Over 6 in. in Length**
1/4	11/64	0.750	1.000
5/16	13/64	0.875	1.125
3/8	1/4	1.000	1.250
7/16	19/64	1.125	1.375
1/2	21/64	1.250	1.500
5/8	27/64	1.500	1.750
3/4	1/2	1.750	2.000
7/8	19/32	2.000	2.250
1	21/32	2.250	2.500
1 1/8	3/4	2.500	2.750
1 1/4	27/32	2.750	3.000
1 3/8	29/32	3.000	3.250
1 1/2	1	3.250	3.500

Type: Hex Bolts
Reference: ANSI B18.2.1-1972
Dimensions are in: Inches

		Thread Length L_T	
Nominal Diameter D	**Head Height H_H**	**Bolts 6 in. and Shorter**	**Bolts Over 6 in. in Length**
1/4	11/64	0.750	1.000
5/16	7/32	0.875	1.125
3/8	1/4	1.000	1.250
7/16	19/64	1.125	1.375
1/2	11/32	1.250	1.500
5/8	27/64	1.500	1.750
3/4	1/2	1.750	2.000
7/8	37/64	2.000	2.250
1	43/64	2.250	2.500
1 1/8	3/4	2.500	2.750
1 1/4	27/32	2.750	3.000
1 3/8	29/32	3.000	3.250
1 1/2	1	3.250	3.500
1 3/4	1 5/32	3.750	4.000
2	1 11/32	4.250	4.500
2 1/4	1 1/2	4.750	5.000
2 1/2	1 21/32	5.250	5.500
2 3/4	1 13/16	5.750	6.000
3	2	6.250	6.500
3 1/4	2 3/16	6.750	7.000
3 1/2	2 5/16	7.250	7.500
3 3/4	2 1/2	7.750	8.000
4	2 11/16	8.250	8.500

Type: Heavy Hex Bolts
Reference: ANSI B18.2.1-1972
Dimensions are in: Inches

Nominal Diameter D	Head Height H_H	Thread Length L_T	
		Bolts 6 in. and Shorter	Bolts Over 6 in. in Length
1/2	11/32	1.250	1.500
5/8	27/64	1.500	1.750
3/4	1/2	1.750	2.000
7/8	37/64	2.000	2.250
1	43/64	2.250	2.500
1 1/8	3/4	2.500	2.750
1 1/4	27/32	2.750	3.000
1 3/8	29/32	3.000	3.250
1 1/2	1	3.250	3.500
1 3/4	1 5/32	3.750	4.000
2	1 11/32	4.250	4.500
2 1/4	1 1/2	4.750	5.000
2 1/2	1 21/32	5.250	5.500
2 3/4	1 13/16	5.750	6.000
3	2	6.250	6.500

Type: Hex Cap Screws (Finished Hex Bolts)
Reference: ANSI B18.2.1-1972
Dimensions are in: Inches

Nominal Diameter D	Head Height H_H	Thread Length L_T			
		L_T for Bolts ≤ 6 in.	L_T for Bolts >6 in.	L_{ro} for Bolts ≤ 6 in.	L_{ro} for Bolts >6 in.
1/4	5/32	0.750	1.000	0.400	0.650
5/16	13/64	0.875	1.125	0.417	0.667
3/8	15/64	1.000	1.250	0.438	0.688
7/16	9/32	1.125	1.375	0.464	0.714
1/2	5/16	1.250	1.500	0.481	0.731
9/16	23/64	1.375	1.625	0.750	0.750
5/8	25/64	1.500	1.750	0.773	0.773
3/4	15/32	1.750	2.000	0.800	0.800
7/8	35/64	2.000	2.250	0.833	0.833
1	39/64	2.250	2.500	0.875	0.875
1 1/8	11/16	2.500	2.750	0.929	0.929
1 1/4	25/32	2.750	3.000	0.929	0.929
1 3/8	27/32	3.000	3.250	1.000	1.000
1 1/2	15/16	3.250	3.500	1.000	1.000
1 3/4	1 3/32	3.750	4.000	1.100	1.100
2	1 7/32	4.250	4.500	1.167	1.167
2 1/4	1 3/8	4.750	5.000	1.167	1.167
2 1/2	1 17/32	5.250	5.500	1.250	1.250
2 3/4	1 11/16	5.750	6.000	1.250	1.250
3	1 7/8	6.250	6.500	1.250	1.250

Type: Heavy Hex Screws
Reference: ANSI B18.2.1-1972
Dimensions are in: Inches

Nominal Diameter D	Head Height H_H	Thread Length L_T			
		L_T for Bolts $\leq$6 in.	L_T for Bolts $>$6 in.	L_{ro} for Bolts $\leq$6 in.	L_{ro} for Bolts $>$6 in.
1/2	5/16	1.250	1.500	0.481	0.731
5/8	25/64	1.500	1.750	0.773	0.773
3/4	15/32	1.750	2.000	0.800	0.800
7/8	35/64	2.000	2.250	0.833	0.833
1	39/64	2.250	2.500	0.875	0.875
1 1/8	11/16	2.500	2.750	0.929	0.929
1 1/4	25/32	2.750	3.000	0.929	0.929
1 3/8	27/32	3.000	3.250	1.000	1.000
1 1/2	15/16	3.250	3.500	1.000	1.000
1 3/4	1 3/32	3.750	4.000	1.100	1.100
2	1 7/32	4.250	4.500	1.167	1.167
2 1/4	1 3/8	4.750	5.000	1.167	1.167
2 1/2	1 17/32	5.250	5.500	1.250	1.250
2 3/4	1 11/16	5.750	6.000	1.250	1.250
3	1 7/8	6.250	6.500	1.250	1.250

Type: Heavy Hex Structural Bolts
Reference: ANSI B18.2.1-1972
Dimensions are in: Inches

Nominal Diameter D	Head Height H_H	Thread Length L_T	
		L_T for Any L	L_{ro} for Any L
1/2	5/16	1.00	0.19
5/8	25/64	1.25	0.22
3/4	15/32	1.38	0.25
7/8	35/64	1.50	0.28
1	39/64	1.75	0.31
1 1/8	11/16	2.00	0.34
1 1/4	25/32	2.000	0.38
1 3/8	27/32	2.25	0.44
1 1/2	15/16	2.25	0.44

Nuts
Reference: ANSI B18.2.2-1972
Dimensions are in: Inches

	Height of Nut (H_N)				
Nominal Diameter *D*	**Square**	**Heavy Square**	**Hex**	**Thick Hex**	**Heavy Hex**
1/4	7/32	1/4	7/32	9/32	15/64
5/16	17/64	5/16	17/64	2 1/64	19/64
3/8	21/64	3/8	21/64	13/32	23/64
7/16	3/8	7/16	3/8	29/64	27/64
1/2	7/16	1/2	7/16	9/16	31/64
9/16			31/64	39/64	35/64
5/8	35/64	5/8	35/64	23/32	39/64
3/4	21/32	3/4	41/64	13/16	47/64
7/8	49/64	7/8	3/4	29/32	55/64
1	7/8	1	55/64	1	63/64
1 1/8	1	1 1/8	31/32	1 5/32	1 7/64
1 1/4	1 3/32	1 1/4	1 1/16	1 1/4	1 7/32
1 3/8	1 13/64	1 3/8	1 11/64	1 3/8	1 11/32
1 1/2	1 5/16	1 1/2	1 9/32	1 1/2	1 15/32
1 5/8					1 19/32
1 3/4					1 23/32
1 7/8					1 27/32
2					1 31/32
2 1/4					2 13/64
2 1/2					2 29/64
2 3/4					2 45/64
3					2 61/64
3 1/4					3 3/16
3 1/2					3 7/16
3 3/4					3 11/16
4					3 15/16

Bolts
Type: Metric Hex Bolts
Reference: ANSI B18.2.3.5M-1979
Dimensions are in: mm

		Thread Length L_T		
Nominal Diameter *D*	**Head Height H_H**	**$\leq$125**	**$>$125 $\leq$ 200**	**$>$200**
5	3.35–3.58	16	22	35
6	3.55–4.38	18	24	37
8	5.10–5.68	22	28	41
10	6.17–6.85	26	32	45
12	7.24–7.95	30	36	49
14	8.51–9.25	34	40	53

continued

(continued)

Nominal Diameter D	Head Height H_H	Thread Length L_T		
		≤125	>125 ≤ 200	>200
16	9.68–10.75	38	44	57
20	12.12–13.4	46	52	65
24	14.56–15.9	54	60	73
30	17.92–19.75	66	72	85
36	21.72–23.55	78	84	97
42	25.03–27.05	90	96	109
48	28.93–31.07	102	108	121
56	33.8–36.2		124	137
64	38.68–41.32		140	153
72	43.55–46.45		156	169
80	48.42–51.58		172	185
90	54.26–57.74		192	205
100	60.1–63.9		212	225

Type: Metric Hex Cap Screws
Reference: ANSI B18.2.3.1M-1979
Dimensions are in: mm

Nominal Diameter D	Head Height H_H	Thread Length L_T			
		≤125	>125 ≤ 200	>200	Thread Runout
5	3.35–3.65	16	22	35	4.0
6	3.85–4.15	18	24	37	5.0
8	5.10–5.50	22	28	41	6.2
10	6.17–6.63	26	32	45	7.5
12	7.24–7.76	30	36	49	8.8
14	8.51–9.09	34	40	53	10.0
16	9.68–10.32	38	44	57	10.0
20	12.12–12.88	46	52	65	12.5
24	14.56–15.44	54	60	73	15.0
30	17.92–19.48	66	72	85	17.5
36	21.62–23.38	78	84	97	20.0
42	25.03–26.97	90	96	109	22.5
48	28.93–31.07	102	108	121	25.0
56	33.80–36.20		124	137	27.5
64	38.68–41.32		140	153	30.0
72	43.55–46.45		156	169	30.0
80	48.42–51.58		172	185	30.0
90	54.62–57.74		192	205	30.0
100	60.10–63.90		212	225	30.0

Type: Metric Hex Flange Screws
Reference: ANSI B18.2.3.4M-1979
Dimensions are in: mm

		Thread Length L_T			
Nominal Diameter D	**Head Height H_H**	**≤ 125**	**$>125 \leq 200$**	**>200**	**Thread Runout**
5	5.4	16	22	35	4.0
6	6.6	18	24	37	5.0
8	8.1	22	28	41	6.2
10	9.2	26	32	45	7.5
12	11.5	30	36	49	8.8
14	12.8	34	40	53	10.0
16	14.4	38	44	57	10.0
20	17.1	46	52	65	12.5

Type: Metric Heavy Hex Structural Bolts
Reference: ANSI B18.2.3.7M-1979
Dimensions are in: mm

		Thread Length L_T		
Nominal Diameter D	**Head Height H_H**	**≤ 100**	**>100**	**Thread Runout**
16	9.25–10.75	31	38	6.0
20	11.60–13.40	36	43	7.5
22	13.10–14.90	38	45	7.5
24	14.10–15.90	41	48	9.0
27	16.10–17.90	44	51	9.0
30	17.65–19.75	49	56	10.5
36	21.45–23.55	56	63	12.0

Type: Metric Heavy Hex Bolts
Reference: ANSI B18.2.3, 6M-1979
Size Range: M12–M36
Head Heights, Thread Lengths: Same as for Metric Hex Bolts (ANSI B18.2.3.5M-1979)

Type: Metric Heavy Hex Screws
Reference: ANSI B18.2.3.3M-1979
Size Range: M12–M36
Head Heights, Thread Lengths, Thread Run-Out: Same as for Metric Hex Cap Screws (ANSI B18.2.3.1M-1979)

Type: Metric Formed Hex Screws
Reference: ANSI B18.2.3.2M-1979
Size Range: M5–M24
Head Heights, Thread Lengths, Thread Run-Out: Same as for Metric Hex Cap Screws (ANSI B18.2.3.1M-1979)

Type: Socket Head Cap Screws (Metric Series)
Reference: ANSI B18.3.1-1978
Size Range: M1.6–M48

Table 3 on p. 8 plus Table 3A on p. 9 of the Reference Gives Data from Which Thread Length Can Be Computed (Many Different, Standard, Grip, and Body Lengths)

Type: Metric Nuts–Various
Reference: British Standards and ISO R272
Dimensions are in: mm

Nominal Diameter *D*	ISO Recommendation R272 (1968)	Black Hex (Typical) BS4190-1967	Precision Hex BS3692-1967	Hi-Strength Hex BS4395-1969
5	4	4	3.70–4	
6	5	5	4.7–5	
8	6.5	6.5	6.14–6.5	
10	8	8	7.64–8	
12	10	10	9.64–10	10.45–11.55
14	11		10.57–11	
16	13	13	12.57–13	14.45–15.55
20	16	16	15.57–16	17.45–18.55
24	19	19	18.48–19	21.35–22.65
30	24	24	23.48–24	25.35–26.65
36	29	29	28.48–29	30.20–31.80
42	34	34	33.38–34	
48	38	38	37.38–38	
56	45	45	44.38–45	
64	51	51	50.26–51	
72	58			
90	72			
100	80			

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Mechanical Engineering

Redesigned for increased accessibility, this fourth edition of the bestselling Introduction to the **Design and Behavior of Bolted Joints: Non-Gasketed Joints**, describes the design, behavior, misbehavior, failure modes, and analysis of the bolts and bolted joints that play such a large, even ubiquitous, role in the myriad machines and structures that form our world. The book provides numerous tables, charts, graphs, and appendices, giving you all the information and data required to design and use non-gasketed bolted joints. Now leaner and meaner, this new edition is better suited to the practicing engineer.

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