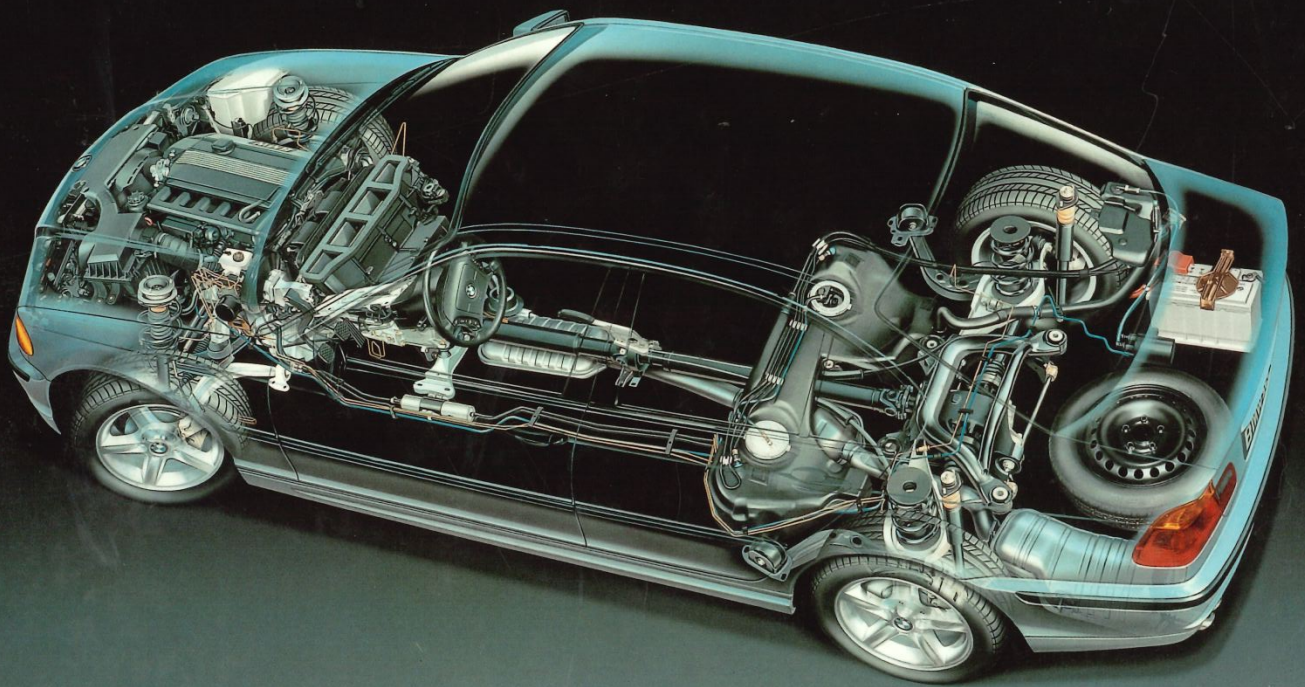


3rd Edition

Fundamentals of Machine Component Design



Robert C. Juvinall / Kurt M. Marshek

THIRD EDITION

Fundamentals of Machine Component Design

ROBERT C. JUVINALL

*Professor of Mechanical Engineering
University of Michigan*

KURT M. MARSHEK

*Professor of Mechanical Engineering
University of Texas at Austin*



JOHN WILEY & SONS, INC.

New York Chichester Weinheim Brisbane Singapore Toronto

To Arlene and Linda

ACQUISITIONS EDITOR
MARKETING MANAGER
SENIOR PRODUCTION EDITOR
ILLUSTRATIONS EDITOR
SENIOR DESIGNER
PRODUCTION SERVICES

Joseph Hayton
Katherine Hepburn
Robin Factor
Sigmund Malinowski
Kevin Murphy
Ingrao Associates

This book was set in Times Roman by The PRD Group, Inc. and printed and bound by R. R. Donnelley & Sons. The cover was printed by Phoenix Color.
Cover illustration: Courtesy BMW of North America, Inc.

This book is printed on acid-free paper. (∞)

Copyright © 2000 by John Wiley & Sons, Inc. All rights reserved.

No part of this publication may be reproduced, stored in a retrieval system or transmitted in any form or by any means, electronic, mechanical, photocopying, recording, scanning or otherwise, except as permitted under Sections 107 or 108 of the 1976 United States Copyright Act, without either the prior written permission of the Publisher, or authorization through payment of the appropriate per-copy fee to the Copyright Clearance Center, 222 Rosewood Drive, Danvers, MA 01923, (978) 750-8400, fax (978) 750-4470. Requests to the Publisher for permission should be addressed to the Permissions Department, John Wiley & Sons, Inc., 605 Third Avenue, New York, NY 10158-0012, (212) 850-6011, fax (212) 850-6008, E-Mail: PERMREQ@WILEY.COM. To order books please call 1(800)-225-5945.

Library of Congress Cataloging in Publication Data:

Juvinall, Robert C.

Fundamentals of machine component design / Robert C. Juvinall,
Kurt M. Marshek.—3rd ed.

p. cm.

ISBN 0-471-24448-1 (cloth : alk. paper)

1. Machine design. I. Marshek, Kurt M. II. Title.

TJ230.J88 1999

621.8'15—DC21

99-34061

CIP

Printed in the United States of America

20 19 18 17 16 15 14 13 12 11 10 9 8 7 6 5

roots without damage. When we consider *fatigue* loading of bolts (Section 10.11), this stress concentration is *very* important.

10.4.3 Combined Torsion and Axial Load

The combination of the stresses that have been discussed can be treated in the normal fashion presented in Sections 4.9 and 4.10, with the distortion energy theory used as a criterion for yielding. With threaded fasteners, it is normal for some yielding to occur at the thread roots during initial tightening.

10.4.4 Thread Bearing (Compressive) Stress, and Its Distribution Among the Threads in Contact

Figure 10.11 illustrates the “force flow” through a bolt and nut used to clamp two members together. Compression between the screw and nut threads exists at threads numbered 1, 2, and 3. This type of direct compression is often called *bearing*, and the area used for the P/A stress calculation

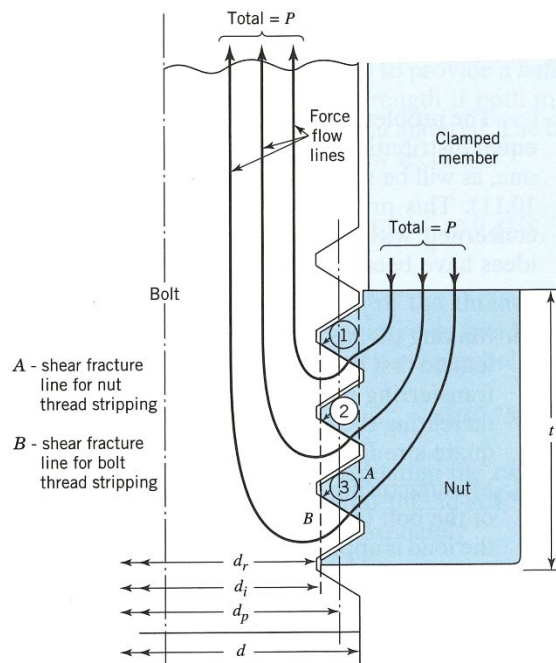


FIGURE 10.11
Force flow for a bolt in tension.

is the *projected* area that, for each thread, is $\pi(d^2 - d_i^2)/4$. The number of threads in contact is seen from the figure to be t/p . Thus

$$\sigma = \frac{4P}{\pi(d^2 - d_i^2)} \frac{p}{t} \quad (10.10)$$

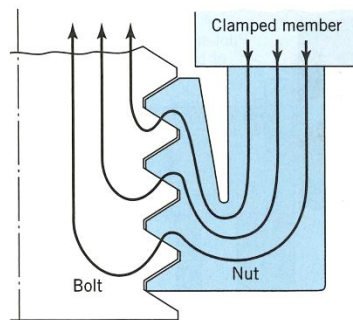
Diameter d_i is the minor diameter of the internal thread. For threaded fasteners this can be approximated by d_r , which is tabulated in Table 10.1. Exact values are given in ANSI B1.1-1974 and various handbooks, but these are not usually necessary because thread bearing stresses are seldom critical.

Equation 10.10 gives an *average* value of bearing stress. The stress is *not* uniformly distributed because of factors such as bending of the threads as cantilever beams and manufacturing variations from the theoretical geometry. Furthermore, a study of Figure 10.11 reveals two important factors causing *thread 1* to carry more than its share of the load:

1. The load is shared among the three threads as *redundant load-carrying members*. The shortest (and stiffest) path is through thread 1. Thus it carries the greatest portion of the load (see Section 2.6).
2. The applied load causes the threaded portion of the bolt to be in *tension*, whereas the mating portion of the nut is in *compression*. The resulting deflections *slightly* increase bolt pitch and decrease nut pitch. This tends to relieve the pressure on threads 2 and 3.

The problem of overcoming this tendency, and obtaining a more nearly equal distribution of loads among the threads in contact, is an important one, as will be seen when considering the fatigue loading of bolts (Section 10.11). This problem continues to challenge the ingenuity of engineers concerned with fastener design and development. The three following ideas have been used.

1. Making the nut from a softer material than the bolt so that the highly loaded first thread will deflect (either elastically or plastically), thereby transferring more of the load to the other threads. This may require increasing the number of threads in contact in order to maintain adequate strength.
2. Manufacturing the nut threads with a *slightly* greater pitch than that of the bolt threads so that the two pitches are theoretically equal *after* the load is applied. The thread clearances and precision of manufacture must, of course, be such that the nut and bolt can be readily assembled.
3. Modifying the nut design as shown in Figure 10.12. Here, the nut loading puts the region of the top threads in *tension*, thus causing elastic changes in pitch that approximately match the changes in bolt pitch. Such special nuts are expensive and have been used only in critical applications involving fatigue loading.

**Figure 10.12**

A special nut provides more nearly equal distribution of load among threads in contact.

10.4.5 Thread Shear (“Stripping”) Stress and Nut Thickness Requirement

With reference to Figure 10.11, if the nut material is weaker than the bolt material in shear (and this is *usually* the case), a sufficient overload would “strip” the threads from the nut along cylindrical surface *A*. If the bolt material is weaker in shear, the failure surface would be *B*. From the thread geometry shown in Figure 10.2, the shear area is equal to $\pi d(0.75t)$, where d is the diameter of the shear fracture surface involved.

Let us now determine the nut thickness (or depth of engagement of a screw in a tapped hole) needed to provide a *balance* between bolt tensile strength and thread stripping strength if both members (bolt and nut or tapped hole) are made of the same material. The bolt tensile force required to yield the entire threaded cross section is

$$F = A_t S_y \approx \frac{\pi}{4} (0.9d)^2 S_y$$

where d is the major diameter of the thread. With reference to Figure 10.11, the bolt tensile load required to yield the entire thread-stripping failure surface of the nut based on a parabolic stress distribution is

$$F = \pi d(0.75t) S_{sy} \approx \pi d(0.75t)(0.58S_y)$$

where t is the nut thickness. Equating the two preceding expressions for F indicates that bolt tensile and thread-stripping strengths are balanced when the nut thickness is approximately

$$t = 0.47d \quad (\text{d})$$

Since nuts are usually softer than bolts in order to allow *slight* yielding of the top thread(s) and thus distribute the load more uniformly among the threads in contact (see point 1 at the end of the preceding subsection), the standard nut thickness is approximately $\frac{7}{16}d$.

The preceding problem was based on the use of a captive O-ring gasket (Figure 10.36). With an *ordinary* gasket (as in Figure 10.25, except not nearly so thick),

1. Flange rigidity and the spacing between successive bolts are more critical, for sealing effectiveness is controlled by the lowest clamping pressure, midway between bolts.
2. The stiffness of the clamped members, k_c , is much less, thus imposing much greater alternating stresses on the bolts.
3. Clamping force F_c must be maintained great enough for sealing when $F_{\max}$ is applied, whereas the captive O-ring gasket requires only that $F_c > 0$. (On the other hand, the captive O-ring gasket is more costly because of the need to machine an accurate groove, and the pliable material of the O-ring may be less resistant to heat, age, and chemical attack.)

10.13

Increasing Bolted-Joint Fatigue Strength

It may be helpful at this point to bring together a list of the various ways by which we can improve bolt (and screw) fatigue strength.

1. Modify stiffnesses to decrease the portion of the external load that increases bolt tension.
 - a. Increase k_c by using higher modulus materials, flat and smooth mating surfaces (without gaskets in between), and greater area and thickness of plates in compression. (Note in Figure 10.28 how increasing grip can increase A_c .)
 - b. Decrease k_b by securing the desired clamping force with smaller bolts of greater strength and by fully utilizing the material strength through more precise control of initial tensioning. Another effective means of reducing k_b is to reduce the shank area, as in Figures 7.10b, 7.11, and 10.13. Ideally, the shank should be reduced with a very large radius fillet so that stresses in this fillet are very nearly as high as stresses in the critical thread root. Reducing the bolt modulus of elasticity helps too, of course, if this can be done without losing material fatigue strength.
2. Modify the nut (or other female threaded member) to equalize the load carried by the several threads in contact (Figure 10.12), and be sure that the number of threads in contact is adequate.
3. Reduce the thread root stress concentration by using a larger root radius.
 - a. A U.S. government standard developed for military aircraft, MIL-B-7838, calls for modifying the basic profile of the external thread

given in Figure 10.2 by using a $0.144p$ thread root fillet radius. This form is used in some industrial fasteners and in most aerospace tension bolts up to 180-ksi tensile strength.

- b. Standard MIL-S-8879 specifies a fillet radius of $0.180p$, which is used with aerospace bolts of 180-ksi tensile strength and higher. (Obviously, these special bolt materials go beyond the SAE grade listing of Table 10.4. Ultrahigh-strength steel bolts have tensile strengths of 220 and 260 ksi.)
- c. Exotic aerospace bolts of columbium, tantalum, beryllium, and other highly notch-sensitive materials sometimes use fillet radii of $0.224p$, and even $0.268p$.

A limiting factor to the use of fillets with very large radii is the associated reduction in thread depth. However, when difficult bolt fatigue problems are encountered, it is well to consider going to a fillet radius of at least $0.144p$.

4. Use a material of highest practical proof strength in order to obtain maximum initial tension.
5. Use tightening procedures that ensure values of F_i as close as possible to $A_t S_p$.
6. Be sure that the threads are rolled rather than cut and that threads are rolled *after* heat treatment.³ The greater the strength, the more important it is to roll *after* hardening. This has been experimentally verified for tensile strengths as high as 300 ksi.
7. After reducing stress concentration and strengthening the thread as much as possible, be sure that the fillet radius under the bolt head is sufficient to avoid failures at this point. *Cold-roll* this fillet if necessary (see Section 8.14).
8. Minimize bolt bending (Figure 10.29).
9. Guard against partial loss of initial tension in service because threads loosen or materials take a permanent set. Retighten bolts as necessary. Also take steps to ensure proper tightening when bolts are replaced after being removed for servicing, and to replace bolts before they yield objectionably because of repeated retightening.

References

1. ANSI (American National Standards Institute) standards B1.1-1974, B1.5-1977, B1.9-1973, American Society of Mechanical Engineers, New York.
2. Chow, W. W., *Cost Reduction in Product Design*, Van Nostrand Reinhold, New York, 1978.
3. Finkelston, R. J., and F. R. Kull, "Preloading for Optimum Bolt Efficiency," *Assembly Engineering* (Aug. 1974).
4. Finkelston, R. J., and P. W. Wallace, "Advances in High-Performance Mechanical Fastening," Paper 800451. Society of Automotive Engineers, New York, Feb. 25, 1980.

³Ordinary heat-treated bolts are rolled *before* heat treatment to reduce cost.