BOLTCALC® Program

Software for the Analysis of Bolted Joints

BOLTCALC is produced by Bolt Science Limited
Bolt Science provides analytical solutions to bolting problems
www.boltscience.com
COPYRIGHTS

Copyright © 1996 - 2004 by Bolt Science Limited. All rights reserved. No part of this publication may be transmitted, transcribed, reproduced, stored in any retrieval system or translated into any language or computer language in any form or by any means, mechanical, electronic, magnetic, optical, chemical, manual or otherwise, without prior written consent from the Legal Department at Bolt Science Limited, 15 Isleworth Drive, Chorley, Lancashire PR7 2PU, United Kingdom.

The software described in this User Guide is provided under license and may be used or copied only in accordance with the terms of such licence.

Important Notice

Bolt Science provides this publication "as is" without any warranty of any kind, either express or implied, including but not limited to the implied warranties of merchantability or fitness for a particular purpose. Some jurisdictions do not allow a disclaimer of express or implied warranties in certain transactions; therefore, this statement may not apply to you. Bolt Science reserves the right to revise this publication and to make changes from time to time in the content hereof without obligation of Bolt Science to notify any person of such revision or changes.

Trademark References

BOLTCALC is a registered trademark of Bolt Science Limited in the United Kingdom and/or other countries.

All other products mentioned herein may be trademarks of their respective holders and are hereby recognised.

Bolt Science Limited
15 Isleworth Drive
Chorley
Lancashire
PR7 2PU
United Kingdom

URL: www.boltscience.com
Email: support@boltscience.com
# Contents

**Installing BOLTCALC**

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Introduction</td>
<td>1</td>
</tr>
<tr>
<td>System Requirements</td>
<td>1</td>
</tr>
<tr>
<td>Installing BOLTCALC</td>
<td>1</td>
</tr>
</tbody>
</table>

**Introduction**

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>About BOLTCALC</td>
<td>2</td>
</tr>
</tbody>
</table>

**Using BOLTCALC**

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>The Main Window</td>
<td>3</td>
</tr>
<tr>
<td>Types of Analysis</td>
<td>4</td>
</tr>
<tr>
<td>Bolt Size Estimate</td>
<td>4</td>
</tr>
<tr>
<td>Torque Analysis</td>
<td>4</td>
</tr>
<tr>
<td>Thread Stripping Analysis</td>
<td>4</td>
</tr>
<tr>
<td>Joint Analysis</td>
<td>4</td>
</tr>
</tbody>
</table>

**Initial Bolt Size Estimate**

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Introduction</td>
<td>5</td>
</tr>
<tr>
<td>Entering of Values</td>
<td>5</td>
</tr>
<tr>
<td>Bolt Size Estimate Results</td>
<td>6</td>
</tr>
</tbody>
</table>

**Torque Analysis**

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Introduction</td>
<td>7</td>
</tr>
<tr>
<td>Background to a Torque Analysis</td>
<td>8</td>
</tr>
<tr>
<td>Data Entry Form</td>
<td>9</td>
</tr>
<tr>
<td>Bolt Details</td>
<td>9</td>
</tr>
<tr>
<td>Fastener Thread Details</td>
<td>9</td>
</tr>
<tr>
<td>Outer Bearing Diameter of the Fastener</td>
<td>10</td>
</tr>
<tr>
<td>Countersunk Head Screws</td>
<td>10</td>
</tr>
<tr>
<td>Inner Bearing Diameter of the Fastener</td>
<td>10</td>
</tr>
<tr>
<td>Fastener Clearance Hole</td>
<td>10</td>
</tr>
<tr>
<td>Fastener Strength Grade Selection</td>
<td>11</td>
</tr>
<tr>
<td>Bolt Tightening Condition</td>
<td>11</td>
</tr>
<tr>
<td>Yield Factor Method</td>
<td>12</td>
</tr>
<tr>
<td>Defining a Tightening Torque</td>
<td>12</td>
</tr>
<tr>
<td>Defining a Bolt Preload</td>
<td>12</td>
</tr>
<tr>
<td>Torque – Angle Method</td>
<td>12</td>
</tr>
<tr>
<td>Friction Coefficient Databases</td>
<td>14</td>
</tr>
</tbody>
</table>
Thread Stripping Analysis

Introduction ........................................................................................................... 17
About Thread Stripping Failures ........................................................................... 17
Starting a Thread Stripping Analysis ..................................................................... 18
Thread Stripping Friction Form ............................................................................ 18
Thread Strength Data Entry Form ........................................................................ 19
  Thread Details Section ......................................................................................... 19
  External Thread Section .................................................................................... 20
  Internal Thread Section .................................................................................... 20
  Material Properties for the External Thread Section .......................................... 20
  Material Properties for the Internal Thread Section .......................................... 22
  Thread Engagement Details Section ................................................................. 23
  Fastener Chamfer Details Section .................................................................... 24
  Countersink Details Section ............................................................................. 24
  Tapping Drill Details Section ........................................................................... 24
  Bell Mouthing ................................................................................................... 25

Thread Stripping Analysis Results ..................................................................... 25
  Effective Length of the Thread Engagement ..................................................... 25
  Shear Area of the Internal Thread ..................................................................... 25
  Shear Area of the External Thread .................................................................... 25
  Internal to External Thread Strength Ratio ....................................................... 26
  Boss/Nut Dilation Factor ................................................................................... 26
  External Thread Bending Factor ........................................................................ 26
  Internal Thread Bending Factor ......................................................................... 26
  Direct Forces to Fail the Fastener ..................................................................... 26
  Fastener Failure Forces allowing for combined tension-torsion loading .......... 27
  Thread Stripping Forces .................................................................................... 27
  Factor of Safety - External Thread .................................................................... 27
  Factor of Safety - Internal Thread ...................................................................... 27
  Critical Length of Thread Engagement ............................................................ 27
  Notes related to the analysis ............................................................................. 27

Joint Analysis

Introduction ........................................................................................................... 28
Remarks Page ....................................................................................................... 28
Applied Forces ..................................................................................................... 28
  Axially Applied Force ....................................................................................... 29
  Shear Force ..................................................................................................... 29
  Residual Clamping Force ................................................................................ 29
  Lower Limit of the Dynamic Force .................................................................. 29
Bolt Details .......................................................................................................... 29
Fastener Fatigue Properties ................................................................................ 30
  The Effect of Joint Face Angularity ................................................................. 31
Modulus of Elasticity .......................................................................................... 31
  Modulus of Elasticity of Fastener Material ....................................................... 31
  Modulus of Elasticity for the Joint Material .................................................... 31
Clamped Parts Stiffness Details .......................................................................... 32
Multi-plate Analysis ............................................................................................ 32
Fastener Clearance Hole ..................................................................................... 32
Installing BOLTCALC

Introduction

This chapter provides information on installing and starting BOLTCALC. It presents the following topics:
♦ System Requirements
♦ Installing BOLTCALC

System Requirements

To install and run BOLTCALC, your Windows compatible PC must be equipped with the following:
♦ Microsoft Windows 95, 98, Me, Windows NT 4.0, 2000, or Windows XP.
♦ 32 MB of RAM for Windows 95, 98 and Me, 64 MB recommended.
♦ 64 MB of RAM for Windows NT, 2000, and XP; 128 MB is recommended.
♦ 10 MB of free space on your hard drive.
♦ SVGA monitor with at least 800 x 600 pixel resolution.

Installing BOLTCALC

The BOLTCALC installation program provides easy step by step instructions on every screen.

Before you install BOLTCALC

1. Close all other programs.
2. If you are installing BOLTCALC on Windows NT, 2000 or XP, log onto your computer with administrator privileges.

To install BOLTCALC from a CD

1. Insert the CD into your CD-ROM drive. The installation program should start automatically. If it does not, follow the instructions located on the sleeve of the CD.
2. Follow the instructions on each screen to install the software.
Introduction

About BOLTCALC

BOLTCALC is a program which is designed to assist the Engineer in the solution of problems related to the design and analysis of concentrically and shear loaded bolted joints. It is designed to be easy to use and makes extensive use of aids to make the selection and input of the bolt's properties and joint characteristics as easy as possible.

The importance of ensuring that fasteners securing an assembly are capable of sustaining all the applied forces is often critical in ensuring that the assembly performs satisfactory in service. When fasteners fail to maintain a minimum required clamp load, it is frequently other elements in the assembly that apparently fail. Examples of this are when gaskets leak because of insufficient clamp load to maintain a seal, or when brackets fail because of the load transfer, which can occur when bolts come loose. The assumptions made by an Engineer regarding how a bolted joint will perform is an important consideration in ensuring that a secured assembly performs satisfactory throughout its design life.

The program has a facility to allow a bolt size estimate to be performed prior to a full analysis. By using this facility, an estimate of the bolt size required for the application will be provided. Based upon this estimate a more detailed analysis can be completed to check whether the selected bolt would be adequate for the application. Data into the program is entered via the Data Entry Form. This tabbed form allows the user to enter and edit data related to the bolt and the joint.

The program has a database of standard values built into it to allow the easy selection of the most appropriate fastener. The program allows selection of metric fine as well as metric coarse threads; the user can enter non-standard fastener sizes as well. The program will also allow the use of imperial units (lbs. and inches) and the unified thread form.
Using BOLTCALC

The Main Window

When first started, the program shows the Tip for the Day form and once this is closed the main BOLTCALC window is displayed. (The program may look slightly different to the view shown due to differences between versions of Windows and the user display/monitor set-up.) At the top of the page is the main menu bar allowing the user to select File, Edit etc. Like other Windows programs, when an item is clicked on the main menu using the mouse, a sub-menu appears.

The program also includes speed buttons. These are specifically designed to provide fast access to menu choices. When the cursor passes over one of these buttons a hint appears to inform the user of the button's function. Additional information on the function of the button is also displayed at the bottom of the program on the help bar.

The Units menu entry allows the units of measure that the program uses to be changed. Clicking of the buttons marked 'Metric' or 'Imperial' can also change the units. The program will work in either metric or imperial units (lbs. and inches). When metric units are selected the program displays metric thread data, when imperial units are selected the program displays unified thread data. This user guide describes the features that are strictly applicable to metric units. In general, when imperial units are selected certain aspects of the data entry form, such as entering of thread information, are displayed differently.
Assistance on any differences can be found by looking at the program's help file.

Types of Analysis

The program provides options for completing four types of analysis:

- Completion of a Bolt Size Estimate.
- Completion of a Torque Analysis.
- Completion of a Thread Stripping Analysis.
- Completion of a full Joint Analysis.

Each type of analysis is covered in greater detail in the sections that follow. A summary of each of the analysis types is:

**Bolt Size Estimate**

This provides the engineer with a provisional estimate of the bolt size needed for a particular application. This option would be used when it is uncertain as to the bolt size required. It is based upon the preload needed to prevent the joint from opening (zero compression between the plates of the joint) or what preload is needed to meet other functional requirements such as to prevent shear slip. A full analysis should be subsequently completed to ensure that the approximation is correct and that other criteria, such as fatigue strength and bearing stress requirements, are met.

**Torque Analysis**

The Torque Analysis option allows the determination of what torque should be used for a particular thread size and strength grade or property class. It allows the torque to be determined based upon minimum, mean or maximum anticipated friction coefficients and prevailing torque conditions.

**Thread Stripping Analysis**

The thread stripping analysis provides a means of determining the forces required to strip the internal and external threads of a fastener. It also calculates the force required to fracture a external threaded fastener across the threaded section.

**Joint Analysis**

The Joint Analysis option is the main part of the BOLTCALC program. The analysis can include a torque analysis and a thread stripping analysis (if a tapped hole is being used). This option will determine if the bolt can adequately sustain the forces acting on it and whether or structural integrity problems can be anticipated within the joint (such as can be caused by excessive bearing stress).
Introduction

An Engineer is frequently faced with the problem of deciding what size of fastener is needed for a particular application and loading. The purpose of the initial bolt size estimate facility is to provide a prediction of the size of bolt required to sustain a particular loading.

This analysis option is selected by clicking Initial Bolt Size Estimate under the Analysis main menu entry. The form that appears is shown below:

Entering of Values

To allow an estimate of the required bolt size to be computed, it is first necessary to define the loads acting on the joint. By clicking on the appropriate button, additional information can be provided on the input values.

In order to estimate the bolt size, certain approximations have had to be made. Whether or not these or valid depends upon the particular application. In most situations, a bolt diameter slightly larger than is necessary would be provided. However, the size should be checked by defining specific values for your application on the data entry form.
Additional information and clarification about the values being requested on the form can be provided by clicking on the button opposite the edit boxes at the right. Once the data has been entered, clicking the OK button on the form allows the program to check for valid values and to display the results on the main form.

**Bolt Size Estimate Results**

After all the relevant data has been entered, the program will compute a bolt size estimate once the Ok button on the form is clicked. In general, the program will provide approximate bolt sizing based upon the stress area computed for a particular strength grade or property class of the bolt. Size options are provided for both fine and coarse threads.

Usually the bolt size estimate facility is used to assist the engineer in his/her judgement of the thread size needed. A full analysis should be completed to ensure that the approximation is correct and that other criteria, such as fatigue strength and bearing stress requirements, are met.
Introduction

The Torque Analysis option provides a means of determining the correct tightening torque and the resulting anticipated clamp force for a threaded fastener. The program accounts for both the tensile stress, due to elongation of the fastener, and torsional stress, due to the applied torque. It will account for both the frictional effects in the thread and between the nut face and clamped surface. Account can also be made for the effects of a reduced shank diameter (smaller than the thread size) and a prevailing torque. The program has also the facility to allow the tightening torque to be calculated from a specified clamp force - or vice versa.

The importance of ensuring that fasteners securing an assembly maintain a minimum clamp load is often critical in ensuring that the assembly performs satisfactorily in service. When fasteners fail to maintain a minimum required clamp load, it is frequently other elements in the assembly that apparently fail. Examples of this are when gaskets leak because of insufficient clamp load to maintain a seal, or when brackets fail because of the load transfer that can occur when bolts come loose. The assumptions made by a Design Engineer regarding the magnitude of a fastener's clamp load is an important consideration in ensuring that an assembly will perform satisfactory throughout its design life. The importance of ensuring the fastener clamp load is adequate at the design stage is frequently underestimated.

The torque control method of tightening is the most popular way of ensuring that a fastener complies with an engineering specification. Most engineers recognise that the method is susceptible to inaccuracy. This being primarily due to variations in the coefficient of friction present in the threads and between the nut and joint surface.

At the design stage it is often necessary to be able to determine the clamp force which will be provided by a fastener. Frequently, because of the lack of any better information, the clamp force is determined by assuming that a certain value will be achieved. The value assumed is frequently a certain percentage of the fasteners proof load (normally a value of 75% is used). This program determines both the tensile and torsional stresses generated by the tightening process.

To facilitate ease of data entry, the program uses a number of databases containing information that is needed for an analysis. Specifically, the thread database contains thread dimensional data, the material database - information on bolt material specifications and thread and nut face friction databases for friction coefficients.
Background to a Torque Analysis

In general, the tightening torque can be considered to consist of four separate parts:

- The torque needed to extend the fastener.
- The torque needed to overcome thread friction
- The torque needed to overcome nut face friction.
- The prevailing torque, if present.

The torque needed to extend the fastener is that which generates the preload, typically it represents between 5% and 10% of the total tightening torque that is needed to be applied.

The torque needed to overcome thread friction is present because of the 'stickiness' of the internal and external thread surfaces. This torque does not increase the preload but does have the effect of creating a torsional shear stress in the threads. The thread friction torque is directly related to the value of the thread coefficient of friction. It is determined by the finish applied and the state of lubrication. Typically it represents about 40% of the total applied torque.

The torque needed to overcome nut face or under bolt head friction (depending on whether the nut or the bolt is rotated) represents typically 50% of the total applied torque. Again it is directly related to the friction coefficient under the bolt head or nut face that in turn depends upon the finish applied to the bolt, the lubrication condition and the joint material and surface condition.

The prevailing torque is the torque required to run a nut (or bolt) down a thread on certain types of fasteners that are designed to resist vibration loosening. This prevailing torque can be provided by an insert in the nut/bolt thread, by using nuts that have their threads locally distorted or by using micro-encapsulated adhesive applied to the threads. The effect of this torque is to increase the torsional stress acting in the threads.

The torsional stress, due to the applied torque, will be imposed upon the tensile stress due to the extension of the fastener. Failure of the fastener can be considered to occur when the combined effects of the tensile and torsional stresses reach a critical value. The Von-Mises distortion energy failure criterion is used by the program to determine an equivalent stress for the combined effects of torsional and tensile stresses.
Data Entry Form

The Data Entry Form consists of a number of pages that the user enters data into. To move between each page, the user should click the tabs at the top of the form marked Remarks, Bolt Details or Tightening Details.

**Bolt Details**

It is necessary to define to the program the specific details of the thread hole and bolt head or nut face details together with the material that is to be used to allow the appropriate tightening torque to be established.

Details of the bolt diameter and thread pitch can be entered directly into the program. Alternatively the user can select the fastener thread size from a database of standard threads by clicking on the appropriate button. The database also presents a range of other information about the thread. The program requires that only the fastener diameter and pitch be entered. The description of the thread is optional.

The thread size database form also displays additional information about the thread size, such as thread tolerances.

The program also enters standard data that relates to a standard fastener diameter into other entry boxes (such as the clearance hole). The user can change all these values. The user should check that any default value is appropriate for the specific application.

If a standard thread is being used then the program will automatically fill in standard values for the user if a thread from the database has been selected. If the user wishes to over-write any of these values, the program will allow it. For example if a reduced shank fastener is being used then the user can enter the appropriate diameter. The majority of fasteners have the shank diameter equal to the outside thread diameter. However some fasteners have the shank diameter reduced so that its resilience is improved. When the shank diameter is reduced to below the diameter of the stress area (the mean of the root and pitch diameters), then that area becomes the critical section as regards potential failure during tightening. The program will default to the outside diameter of the thread; this can be changed by the user typing in a smaller diameter if a reduced shank fastener is being used.
The program defaults are for standard thread forms such as the metric or unified threads. These threads have a flank angle of 60 degrees. Other, now redundant thread forms, have differing flank angles.

In this data entry box, the user should enter the minimum value of the outside diameter of the nut or bolt. There are also a number of check boxes that allow the selection of the fastener head type to be made, the Standard Hexagon Head check box acts as the default.

The program will enter a standard value if a standard diameter had been previously entered. Clicking on the Socket Head Cap Screw check box would select an appropriate diameter for this type of screw. By clicking the 'Other' check box, the user can enter his/her own value.

The user should check that any default value is appropriate for the specific application being considered. The standard values used by the program may not be appropriate or correct for the user's application.

If the fastener being used is a countersunk headed screw, then when the check box is clicked with this title, another window opens giving details about this head type. The effect of the countersunk head is to increase the frictional resistance because of the wedging action of the head.

Subsequently, other factors being equal, a higher tightening torque is needed with a countersunk head socket screw then with a standard head. If a nut is used (and the nut rather than the screw is tightened) than please enter details of the nut on the main form rather then using this form. (Details of the part that is being rotated should be entered is the appropriate torque/preload value is to be computed.)

The default head angle for the screw quoted by the program is the minimum angle for a standard countersunk head socket screw. The value can be changed to accommodate special designs. Countersunk head screws are usually used only in moderately loaded applications, they are not usually used in critical fastening applications.

The program, in the report listing will detail the effective friction diameter that the program has calculated and has used in the torque analysis. For countersunk headed screws, the effective friction diameter is generally larger than the actual head diameter. This is because the friction diameter has been adjusted to allow for the wedging effect of the head style.

In this box, the user should enter the minimum value of the inner diameter of the nut or bolt. In many applications, this is the same as the clearance hole, but not in all cases. The program's default is that the inner-bearing diameter is equal to the clearance hole diameter. By clicking the 'User defined Inner Bearing Diameter' check box, the user can enter his/her own value.

Again, the user should check that any default value is appropriate for the specific application.

In this box details of the diameter of the fastener clearance hole are entered. There are four check boxes in this section, with the 'Fine' button acting as default.

By clicking on the appropriate button the clearance hole related to the particular hole series defined in ISO 273:1979 (Fasteners - Clearance holes for bolts and screws) is selected if a standard bolt diameter has been previously selected. If the user wants to select a specific value not within this standard, by clicking on the 'User defined' check box the data entry box can be edited.

The majority of joints incorporating bolts in general mechanical engineering applications have clearance holes (that is, the hole diameter is larger than the bolt diameter). Clearance holes improve the ease in which the product can be assembled and, by allowing a reduction in the tolerances required, reduce
manufacturing costs. The program will allow the user to select the appropriate clearance hole based upon ISO 273. Alternatively, the user can enter the appropriate size if required. When the clearance hole is only slightly larger than the bolt diameter, care should be taken to ensure that interference is avoided between the edge of the hole and the underhead fillet of the bolt. A chamfer may be necessary. This is to avoid high-localised stresses that could lead to excessive preload loss from embedding.

Details of the yield strength of the fastener is required by the program. Information can be directly entered into the program or alternatively, the user can select an appropriate material by accessing the bolt material database that contains hundreds of bolt material specifications.

Property Class is the term used in the ISO metric fastener standards for strength grade. This terminology has been used by the various national standards when they adopted the ISO. The designation system used for the metric property class system is significant, in that it denotes minimum yield and tensile properties for the fastener. The designation system for bolts consists of two parts:

The first numeral of a two-digit symbol or the first two numerals of a three-digit symbol approximate 1/100 of the nominal tensile strength in N/mm² (or MPa).

The last numeral approximates 1/10 of the ratio expressed as a percentage between nominal yield stress (or the stress at the 0.2% permanent set limit) and nominal tensile stress.

The minimum tensile and yield strengths (or the 0.2% permanent set limit) are equal to or greater than, the nominal values. Hence a fastener with a property class of 8.8 has a minimum tensile strength of 800 N/mm² (or MPa) and a yield stress of 0.8x800=640 N/mm². The designation system for metric nuts is a single or double digit symbol. The numerals approximate 1/100 of the minimum tensile strength of the nut in N/mm².

For example, a nut of property class 8 has a minimum tensile strength of 800 N/mm². A bolt or screw of a particular property class should be assembled with the equivalent or higher property class of nut to ensure that thread stripping does not occur.

Designers can easily specify the highest strength grade for critical applications. However, they may do this without a full assessment of the associated risks. Research and experience has indicated that fasteners of hardness exceeding C39 on the Rockwell scale (such as grade 12.9 fasteners) have a high susceptibility to stress embrittlement. The higher the hardness of the fastener (which is directly related to the strength of the fastener for steel) the more critical becomes the choice of material and heat treatment.

Fasteners of property class 12.9 require careful control of the heat treatment operation and cautious monitoring of surface defects and the surface hardness. In addition, they are also more prone to stress corrosion cracking of non-plated as well as electro-plated finished fasteners.

**Bolt Tightening Condition**

The program allows for four ways in which the bolt preload can be defined, these are:

1. Yield Factor Method
2. Defining a Tightening Torque
3. Defining a bolt preload
4. Torque – Angle Method
Considering each in turn.

**Yield Factor Method**

The yield factor method requires the percentage of the yield strength that is wished to be used, to be specified. This is the normal way that the assembly tightening torque is determined and what preload this will result in. The program defaults to this method. The program defaults to 90% (a 0.9 yield factor) utilisation of the yield strength of the bolt material from the combined effects of tension and torsion. A yield factor of one would result in yield occurring of the bolt material. By double clicking on the Yield Factor button, the 0.9 yield factor default can be changed by the user. A 0.9 yield factor would result in approximately 75% of the yield being utilised in direct tension - dependent upon the friction conditions. The yield factor method is more consistent than assuming a proportion of yield in tension is to be used. This is because it allows torsional stresses to be included (such stresses vary with the friction value in the thread) in the determination of the appropriate value of the tightening torque. The 10% of yield strength remaining in the bolt material (when the default 0.9 yield factor is used) is usually sufficient to allow the additional stresses imposed on the bolt by external forces to be sustained.

**Defining a Tightening Torque**

By clicking on the button marked 'Tightening Torque', the user can enter an assembly tightening torque that the program will use to determine the bolt preload after considering the friction conditions specified. The bolt details need to be entered prior to using this form. This is to allow the program to determine what the tightening torque would be which would result in yield of the bolt material occurring. This is displayed on the form to assist the user in entering an appropriate value for the torque. The program will not allow a torque to be entered that would exceed the bolt's yield strength. The value suggested in the data entry box is that based upon a yield factor of 0.9 being used. Any torque value greater than zero, but below that which would cause yield, can be entered.

**Defining a Bolt Preload**

By clicking on the button marked 'Assembly Preload', the user can enter a bolt preload value that the program will use. The bolt details need to be entered prior to using this form. This is to allow the program to determine what the preload would be that would result in yield of the bolt material occurring. This is displayed on the form to assist the user in entering an appropriate value for the preload. The program will not allow a preload value to be entered that would exceed the bolt's yield strength. The value suggested in the data entry box is that based upon a yield factor of 0.9 being used. Any preload value greater than zero, but below that which would cause yield, can be entered.

**Torque – Angle Method**

The torque-angle tightening method allows a higher preload to be consistently achieved compared to torque tightening. The method elongates the bolt a pre-set amount (determined by the angle of turn – one full turn elongates the bolt and compresses the joint one pitch). By elongating the bolt in this manner it overcomes most of the frictional scatter associated with the torque tightening method. To eliminate frictional scatter the angle of turn must be such that the bolt is elongated past its yield point so that it is permanently stretched by a small amount. A bolt tightened repeatably using this method will break on tightening after a number of tightenings once the elongation limit is reached.

The method consists of applying a torque (known as the snug torque) to the fastener following by rotating it through a previously defined angle of rotation. The snug torque is typically between 40% to 60% of the yield torque and is used to ensure that the joints plates are pulled together so that they are in metal to metal contact. The subsequent angle of turn stretches the fastener so that the yield strength of the fastener is reached or exceeded. If a snug torque is not used then the angle of turn required for the bolt to reach yield will vary from joint to joint depending upon the flatness of the joint plates.
By tightening in this manner the variations in the preload that can result from frictional variations are largely removed. There is still a frictional effect since friction has the effect of increasing the torsional stress in the fastener and hence influencing the tension in the fastener when the yield point is reached.

The program will determine the maximum and minimum preload values based upon the scatter in the frictional conditions – hence it is necessary to define the upper and lower friction coefficients. The minimum preload will be determined based upon the minimum yield strength with the maximum friction condition. The maximum preload will be determined from using the maximum yield condition with the minimum friction condition. The minimum preload is of significance since most joints rely upon the fastener’s clamp force to prevent slippage or to prevent joint separation. The maximum preload is of significance in determining the maximum clamp force acting on the joint for bearing stress calculations.

To complete the calculation the program requires the minimum and likely maximum yield values of the fastener. Since most standards only specify the minimum yield – the program will provide an estimate of the upper yield strength based upon the maximum tensile strength. Most standards only indirectly define the maximum tensile strength by specifying a maximum hardness value. For steel, a hardness value can be converted to a tensile strength by using such standards as DIN 50150 or SAE J417 the program already has data on the maximum tensile strength in the bolt material database. The program estimates the maximum yield strength by multiplying the minimum yield strength by the ratio of maximum tensile strength to minimum tensile strength.

The program will determine, if this option is selected, a snug torque value and an angle of turn. The program will determine the snug torque based upon 50% of the torque required to reach the minimum yield strength with the thread friction on the maximum condition. The angle of turn will be determined based upon the elongation needed so that the maximum yield strength of the bolt is reached. Because of uncertainties relating to the flatness of the individual joint plates, it is recommended that tests be completed to determine the snug torque and angle of turn. The test-measured preload should be within the upper and lower preload limits determined by the program. A specific preload value can be entered by selecting the Assembly Preload option on the Tightening Details page of the main data entry form.

When the user selects to enter snug torque and angle of turn values, the program will assume that the specified torque and angle values will be sufficient for the maximum yield strength to be reached. It is up to the user to experimentally verify that this is the case.

Selecting the torque-angle method of tightening can also be suitable for assessing the preload variability in yield controlled tightening methods. Such methods start with a snug torque specification (to eliminate non-linearities in the joint compression) and then use special tooling and electronics that allows the torque-angle gradient to be determined. The wrench or tightening tool indicating when the gradient of the torque-angle curve reaches a pre-determined value indicating that the yield point of the
bolt material as been reached. The upper and lower preloads from this method will be similar to that obtained from the torque-angle method.

Friction Coefficient Databases

Major influences on determining the appropriate tightening torque are the friction coefficients in the thread and under the face of the nut or bolt head. Establishing the value of these friction coefficients for particular material and finish conditions can be problematical unless test data is at hand.

The databases that are supplied with the program contain the results of tests conducted for that finish condition. The database lists the source of the information for reference purposes. Three values for the friction coefficients are listed; the minimum, mean and maximum values for that particular friction condition. One of the major problems in using torque control is the variability in the friction values that directly influences the resulting preload for a given applied torque value. Normally a minimum value of the friction coefficient should be selected since this gives the lowest tightening torque and hence ensures that the fastener will not be overtightened. The user can use the mean or maximum friction value but it should be borne in mind that this will lead to a higher tightening torque that may result in the fastener being over tightened.

Many test results or information from lubricant and finish suppliers list only the mean value of the friction coefficient. The database consists, in part, of test results that listed only the mean value of friction. Some tests define the minimum and maximum values, whenever the information is available the program lists the friction values based upon minimum and maximum values being 3 standard deviations away from the mean. In cases when only the test reported the mean result, to establish the minimum and maximum values the following equation was used:

Maximum Friction Value = 1.5 x Minimum Friction Value

The 1.5 factor was established based upon an analysis of the scatter associated with over 60 sets of tests conducted on different finish and lubrication conditions. The database states whether the minimum and maximum values were calculated from the mean or determined by testing.

Prevailing Torque Value

The prevailing torque is the torque required to run a nut (or bolt) down a thread on certain types of fasteners that are designed to resist vibration loosening. This prevailing torque can be provided by an insert in the nut/bolt thread, by using nuts that have their threads locally distorted or by using micro-encapsulated adhesive applied to the threads.

If a bolt diameter has been entered into the program, by clicking on the appropriate selection, the program will enter a lower bound value for the prevailing torque by default – check whether this is applicable to your application.

The characteristics of the majority of prevailing torque fasteners (nylon/polyester patch and distorted head types) are such that the magnitude of the prevailing torque reduces as the number of installations and removals increase. Typically the maximum prevailing torque listed is that for the first assembly. Standards such as ISO 2320 typically do not list a minimum value of prevailing torque but does list the fifth removal torque. The fifth removal torque, when other information is not available is taken as the

---

1 ISO 2320 Prevailing torque type steel hexagon nuts
minimum value of the prevailing torque. The difference between the
minimum and maximum prevailing torque can be quite substantial.

Bolts which have a prevailing torque are frequently used where there exists
a risk of vibration loosening. The prevailing torque counteracts the off
 torque which can be present when the bolt is subjected to vibratory loading.
During the tightening process, this prevailing torque has the effect of
increasing the torsional stress in the shank of the bolt. For the same state of
combined stress in the bolt, the higher the torsional stress in the bolt, the
lower will be the resulting preload. For the same frictional conditions, the
total tightening torque required to tighten the bolt so that so that a specified
combined stress exists in the shank of the bolt, does not significantly
increase with increasing prevailing torque. However the preload in the bolt
can be significantly reduced.

The use of threadlocking adhesives, such as structural cyanoacrylate and
anaerobic compounds, results in the bolt/nut exhibiting a prevailing torque
characteristic. This effect also occurs with adhesive contained within micro-
beads applied to the threads. The magnitude of the prevailing torque is
generally less than with proprietary prevailing torque fasteners, however the
use of threadlocking compounds does still affect the tightening torque
specification and the resulting preload.

The user can enter his/her own value of prevailing torque into the program.
By default, the minimum value is used but the maximum can easily be
selected. These values should only be used when more specific information
is not available from the fastener manufacturer or from test work.

**Stresses in the fastener**

When the fastener is tightened the shank sustains a direct stress, due to the
elongation strain, together with a torsional stress, due to the torque acting on
the threads. Values for both the tensile and torsional stresses in the thread
(or plain shank if a reduced shank fastener is being used) are presented in
the results.

To determine an equivalent stress to allow a comparison to be directly made
to a percentage of the yield strength, the program uses the Von-Mises failure
criterion. The Von-Mises (sometimes called the theory of constant energy of
distortion) theory is the most widely accepted for the elastic failure of
ductile materials and agrees closely to experimental evidence. The program
computes the direct and torsional stresses and combines these two values
together using the Von Mises criteria to establish an equivalent stress that
can be compared to the yield strength of the material or a percentage
thereof.

The tensile force in the fastener shank is determined by multiplying the
stress area (or shank area when applicable) by the tensile stress. The 90%
utilisation of the yield strength of the fastener is used as default by the
program to allow some reserve on strength to allow for the application of the
working load. If the working load is applied to the fastener and the yield
strength is exceeded, then upon release of this load a tension loss in the
fastener will result which can cause loosening and/or joint failure.
Generally, the fastener sustains a small proportion of the working load, the
majority is sustained by a reduction in the clamp force in the joint interface.

There is widespread misunderstanding by Engineers on the load transfer
mechanism involved when a bolted joint sustains an axially applied force.
Consider the case of a single bolt supporting a bracket. Before a nut is
tightened, the bolt would sustain the entire working load. However, once the
bolt is tightened, so that a tensile force (preload) is present within the bolt,
clamping the bracket to the support, the bolt would sustain an applied force
less than the working load. Typically, the bolt sustains less than 20% of the working load.

From a Designer's viewpoint, this illustrates one of the special features of a bolted joint in comparison to other components. The load-carrying mechanism of the joint is dependent upon the assembly operation itself.

**Considerations on Torque Tightening**

**Washers**

If loose fitting washers are used, inaccuracies can occur in the torque-tension relationship. The washer seating eccentrically relative to the bolt axis brings this about. The resulting high stress on one side of nut-washer interface, brought about by this eccentricity, can result in the deterioration of the bearing surface. This increases the surface friction as tightening progresses. Increasing friction can cause the relative motion to change to the other interface (that is, the washer to the joint). This changing of the interface can affect the bearing face torque by as much as 15%. To eliminate this potential problem give consideration to either using close fitting washers, or, if feasible to the use of flange type fasteners.

**Tolerance class of fasteners**

Fasteners are generally made to comply with standard tolerance classes. Variations in the effective diameter of the thread will occur as result of such tolerances. Specifying a tighter thread tolerance class will result in the tightening torque and clamp force scatter being reduced as a result of this factor. Generally speaking however, scatter due to tolerance variations is small compared with frictional variations. The Engineer is not usually at liberty to specify a fine thread class if he wishes to use standard fasteners. The majority of standard fasteners are made to the medium class of fit, 6H internal thread and 6g external, for metric fasteners, and, class 2A external and 2B internal for Unified fasteners.

**Re-use of plated fasteners**

On high tensile fasteners, the plating applied (especially zinc plating) can break down under the high interface pressures involved. Breakdown of the plating, which can result in effects such as galling, can have the consequence that the frictional coefficients are significantly higher than expected. Repeated use of plated fasteners increases the likelihood of this effect occurring.
Introduction

The thread stripping analysis provides a means of determining the forces required to strip the internal and external threads of a fastener. It also calculates the force required to fracture an external threaded fastener across the threaded section.

To precisely predict the force and mode of failure of a threaded assembly demands consideration of a large number of factors. Thread stripping is a complex phenomenon. The program considers the following factors when determining failure mode:

1. The effect of variation in the dimensions of the thread, such as major, pitch and minor diameters, has on fastener failure mode of both the internal and external threads.
2. Tensile and shear strength variations in the material for both the internal and external threads.
3. The effect of radial displacement of the nut (generally known as nut dilation) in reducing the shear strength of the threads. The tensile force in the fastener acts on the threads and a wedging action generates a radial displacement.
4. The effect which the bending of the threads, caused by the action of the fasteners tensile force, has on both internal and external thread shear strength.
5. The effect which production variations in the threaded assembly, such as slight hole taper or bellmouthing, can have on thread strength.

To assist and guide the Engineer, the program incorporates default values such as maximum and minimum thread dimensions based upon standard thread tolerances.

Following the guidelines presented in this manual in the use of the program, it is possible to allow, at the design stage, for anticipated variations in the forces required to cause failure. This variation is inherent due to dimensional and property differences between fasteners of the same size and property class.

About Thread Stripping Failures

Failure of a threaded fastener during assembly generally occurs in one of three modes.
1. Failure by tensile fracture through the shank or threaded section of the fastener.

2. Shear failure through the thread profile (thread stripping) of the external thread.

3. Shear failure through the thread profile (thread stripping) of the internally threaded part.

Thread stripping is a shear failure of an internal or external thread that results when the strength of the threaded material is exceeded by the applied forces acting on the thread. Thread stripping can be a problem in many designs where tapped holes are required in low tensile material. In general terms thread stripping of both the internal and external threads must be avoided if a reliable design is to be achieved. If the bolt breaks on tightening, it is obvious that a replacement is required. Thread stripping tends to be gradual in nature and it may go unnoticed at the time of assembly. It starts at the first engaged thread due to thread deformations causing it to carry the highest load and successively shears off subsequent threads. This may take a number of hours to complete and so the product may appear fine at the time of assembly. The risk is therefore present that threads that are partially failed, and hence defective, may enter service. This may have disastrous consequences on product reliability.

Because of the more widespread use of angle control and yield control tightening methods, bolt preloads, for a given size and strength of a fastener can be greater than traditionally was the case. This coupled with the widespread use of automatic and semi-automatic tightening procedures increases the likelihood that thread stripping will occur.

The strength of a nut or bolt thread cannot be viewed in isolation without considering the inter-dependence that both elements have on the strength of the assembly. One of the problems in predicting thread stripping strength is that, without considering such effects as thread bending, nut dilation or bellmouthing, an optimistic result occurs. The actual stripping strength being lower than that calculated.

**Starting a Thread Stripping Analysis**

A thread stripping analysis can be completed separately or completed as part of an overall joint analysis. The main form that data is entered on is identical in both cases. A separate analysis would be completed in cases when you are just interested in a parts thread stripping strength.

To start a separate thread stripping analysis click the menu entry Analysis Type - Thread Stripping Analysis. Alternatively there is a shortcut on the speedbar just under the menu that you can click to start the analysis.

**Thread Stripping Friction Form**

This form is only displayed if a separate thread stripping analysis is being completed. When a thread stripping analysis is completed as part of an overall joint analysis, the information that is entered on this form is entered under the Tightening Details page of the Joint Analysis Data Entry Form.

The information on the Thread Stripping Friction Form allows the user to enter the friction coefficient present in the threads. By allowing for thread friction the program will determine the reduced tensile force needed to
fracture the thread cross section as a result of the torsional stress induced by the tightening process.

If the bolt breaks on tightening, it is obvious that a replacement is required but not so with thread stripping that tends to be gradual in nature. Because a bolt, when it is being tightened, experiences both direct tensile stress and torsional stress, the direct tension force present in the bolt at failure will be lower than the direct force that would fail the bolt without torsion being present. By allowing for the torsion effect, the engineer can investigate the optimum length of engagement for torque tightening applications.

The thread friction value is determined by the finish applied and the state of lubrication. An appropriate value can be entered directly into the box, or if it is wished to select from a list, clicking on the button will display a form that has a large number of friction values for particular surface finishes displayed. The user can select an appropriate value.

**Thread Strength Data Entry Form**

This is the main form that thread stripping data is entered.

![Thread Strength Data Entry Form](image)

The form is split into several sections to assist in data entry. Covering each section in turn:

**Thread Details Section**

This comprises data entry boxes for a Thread Description, Bolt Diameter and Thread Pitch. If the Access Thread Database button is pressed then a standard thread size can be selected from a database - if this is used - all three data entry boxes will be filled in automatically. If the user is using a special size of fastener - the details can be entered directly into the relevant box (this applies to the other data entry boxes as well such as for the external and internal thread sizes).

**Thread Description**

This can be optionally used to record details about the thread.

**Bolt Diameter**

This is the diameter commonly used to describe the thread. For example, for a M8x1.25 fastener the nominal thread diameter is 8mm.

**Thread Pitch**
This is the distance from the top of one thread crest to the next.

In this section details of the external (bolt) thread are entered. If the thread database was previously used to select a thread size then values will have been entered into the data entry boxes based upon a 6g tolerance class if the thread was metric or a 2A tolerance class if the thread was inch based. These tolerance classes are the standard values used on the majority of threaded fasteners. The user can enter his/her own values if these are incorrect for the thread being used.

**Major Diameter of the External Thread**

The major diameter of an external thread is the diameter of an imaginary cylinder parallel to the crests of the thread. Both maximum and minimum diameters are requested to be entered.

**Pitch Diameter of the External Thread**

The pitch or effective thread diameter of the external thread is the diameter that has equal metal and space widths. Put more simply, it is the mean diameter of the thread.

**Minor Diameter of the External Thread**

The minor or root diameter is the diameter of a cylinder that just touches the roots of the thread.

In this section, details of the strength of the threaded fastener are entered. The program will automatically fill in data values if the user selects the appropriate grade by clicking on the Bolt Material Database button. The materials database contains information of fastener strength grades appropriate to metric or inch based fasteners. If the user desires, values can be entered directly into the data boxes if material values are wished to be entered that are not covered in the database or to over-write values presented by the database.

**Property Class** is the term used in the ISO metric fastener standards for strength grade. This terminology has been used by the various national standards when they adopted the ISO. The designation system used for the metric property class system is significant, in that it denotes minimum yield and tensile properties for the fastener. The designation system for bolts consists of two parts:

The first numeral of a two-digit symbol or the first two numerals of a three-digit symbol approximate 1/100 of the nominal tensile strength in N/mm² (or MPa).

The last numeral approximates 1/10 of the ratio expressed as a percentage between nominal yield stress (or the stress at the 0.2% permanent set limit) and nominal tensile stress.
The minimum tensile and yield strengths (or the 0.2% permanent set limit) are equal to or greater than, the nominal values. Hence a fastener with a property class of 8.8 has a minimum tensile strength of 800 N/mm² and a yield stress of 0.8x800=640 N/mm². The designation system for metric nuts is a single or double digit symbol. The numerals approximate 1/100 of the minimum tensile strength of the nut in MPa.

When using both metric and inch based fastener standards, the user should ensure that the strength grade that is selected is appropriate to the size and type of fastener being specified.

Designers can easily specify the highest strength grade for critical applications. However, they may do this without a full assessment of the associated risks. Research and experience has indicated that fasteners of hardness exceeding C39 on the Rockwell scale (such as grade 12.9 fasteners) have a high susceptibility to stress embrittlement. The higher the hardness of the fastener (which is directly related to the strength of the fastener for steel) the more critical becomes the choice of material and heat treatment.

Fasteners of property class 12.9 require careful control of the heat treatment operation and cautious monitoring of surface defects and the surface hardness. In addition, they are also more prone to stress corrosion cracking of non-plated as well as electro-plated finished fasteners.

Covering each of the data entry boxes in turn:

**Minimum External Strength**

This is the tensile strength of the external thread - it is the ultimate stress that the thread can sustain calculated on the basis of ultimate load on original unstrained dimensions. The values quoted in standards are the minimum strength required for the bolt material. Thus a bolt of property class 8.8 has a minimum tensile strength of 800 N/mm².

**Maximum External Strength**

It is unusual for a specification to define directly the maximum fastener strength. The maximum strength is of importance since if the strength was too high relative to the nut then thread stripping could occur rather than the more desirable tensile fracture of the bolt thread.

Measurements of the tensile strength of bolts indicate that the bolt's actual strength is significantly greater than the minimum specified. Tests have indicated that the actual strength is as much as 30% above the minimum specified strength for low strength fasteners; 15% above for standard structural fasteners decreasing to 10% for high strength fasteners (Class 10.9).

The majority of standards control upper tensile strength indirectly by specifying a maximum hardness value. In the majority of cases the value presented by the program is based upon a conversion of this hardness value to a tensile strength. The user can modify this value if required by simply over-typing it. The program uses this value to determine the upper bound for the tensile fracture load of the fastener.

**Ratio of Tensile to Shear Stress**

The shear strength of a material is not the same, in general terms, as the tensile strength. For steel, tests have indicated that the ratio of shear strength to tensile strength is approximately in the order of 0.6. That is, the shear strength is 0.6 of the tensile strength for steel. Based upon torsional shear tests on steel bolts, the ratio of shear strength to tensile strength is approximately 0.61 for standard grade fasteners (Class 8.8) decreasing to 0.58 for higher strength fasteners (Class 10.9). For a given fastener property
grade, the program will enter an appropriate value of the ratio that the user can modify if desired or if specific test data is available.

**Minimum Shear Stress**

It is the minimum shear stress that in part determines the thread stripping of the fastener. This value is the minimum tensile strength multiplied by the ratio of tensile to shear strength. For a given fastener property grade, the program will enter an appropriate value of the minimum shear strength that the user can modify if desired or if specific test data is available.

**Material Properties for the Internal Thread Section**

A number of buttons are presented in this section to allow the user to select the most appropriate material. This is done to allow the program to present guideline values for the ratio of shear to tensile stress.

**Description for the Material**

This is a text field and the user can enter a suitable description for the material being used. A brief description of the material is entered by the program once the user selects an appropriate material button. This field is non essential and is not used in any of the calculations.

**Minimum Tensile Strength**

The user enters the tensile strength for the material that is being used for the internal thread.

**Ratio of Tensile to Shear Strength**

If a material button has been selected then the program will enter a guideline value for this ratio. Multiplying this ratio by the tensile strength will give the shear strength of the material. For steel, typically this ratio is 0.6 (higher for lower tensile steels and slightly lower for high tensile), that is the shear strength of steel is typically 60% of the tensile strength.

The ratio of shear strength to tensile strength is significantly higher for grey cast iron than it is for steel. Tests have indicated that the ratio reduces from 1.4 at a tensile strength of 110 N/mm² to 1.0 at a tensile strength of 250 N/mm².

The ratio of shear strength to tensile strength is lower for nodular and malleable cast irons than it is for grey cast iron. Research indicates that the ratio is approximately 0.9 for these types of cast iron.

The user can change any guideline value entered by the program.

Experimenters have used one of three methods of determining the shear strength of materials for determining thread stripping strength:

1. The torsional test involves determining the torque required to fail a specimen and then using the 'plastic torsion equation' to establish the shear stress at failure.
2. A punch test that involves using a punch and die to punch out slugs from a test block.
3. A double shear test involving placing a specimen in a fixture and using a test machine to shear the specimen

**Minimum Shear Strength**

The shear strength of the material is used to determine the stripping strength of the thread. The program will enter a value if the tensile strength and ratio of shear to tensile strength values have been previously entered into the program.
In this section, details relating to the engagement of both threads are entered.

**Length of Thread Engagement**

The nominal length of engagement of the external thread into the internal thread is entered into this box. For a bolt being used with a nut, the nominal length is the height of the nut if the bolt is intended to protrude through it. In a blind tapped hole (the hole does not pass through the material) the length of thread engagement is the distance the external thread engages with the internal thread.

For short to medium lengths of thread engagement the thread stripping length is proportional to the length of engagement. For thread engagements greater than about 1.2 times the thread diameter (for steel), length of engagement is no longer proportional to stripping strength. For aluminium, tests have indicated that this limit of proportionality is higher at approximately 1.5 times the thread diameter. This is the limit to the length of thread engagement beyond which increasing the length will no longer proportionally increase the stripping strength. Presently there is no way to accurately predict the stripping strength for long thread engagement lengths. It is recommended that the results from the program be used cautiously when the length of thread engagement is greater than 1.5 times the nominal diameter of the thread.

Due to elastic deformation within the threads the pitch of the bolt thread is shortened and the pitch of mating internal thread increased when the bolt is tightened. These small deformations within both threads must balance each other. The result of this is that the first few engaged threads have to take significantly higher than average loads. Also, for a given length of thread engagement, a fine thread pitch will give a higher loading on the first thread. For the long lengths of thread engagement, the first few threads have to excessively deform before the bottom threads can carry any appreciable load. This can result in the first threads starting to shear before the full load capability of the bottom threads is realised. Successive threads can then sheared resulting in the assembly failing. Since it is a progressive failure it may not be initially noticed and may take several hours before final failure occurs. Also pitch errors on long lengths of thread engagement may also make it difficult to assemble the two threads together.

**Is the tapped thread into a nut or a boss?**

This is to indicate whether the internal thread is tapped into a nut or a boss. The default offered by the program is 'No'. If the user selects 'Yes' then the edit box 'Diameter of the boss or width across flats of the nut' is activated. The user enters the boss diameter if the tapped thread is into a boss on a casting, plate or similar, or the across flats dimension if a nut is being used. The program includes this feature to allow for the phenomenon known as nut dilation. The tensile force present in the fastener during tightening acts on the vee threads to produce a wedging action that results in a radial displacement. This radial displacement is generally known as nut dilation and occurs in threaded bosses as well as conventional nuts. Theoretical and practical studies of this phenomenon indicate that the top face of the nut contracts in a radial direction while its bearing surface expands. The net effect of this dilation is to reduce the shear area of both the internal and external thread.

The stripping strength of an assembly can be improved by increasing the width across flats of the nut, or boss diameter, up to about 1.9 times the nominal thread diameter. This increases the stiffness locally around the internal thread and reduces radial expansion.

The degree of dilation also depends upon the value of the thread coefficient of friction. Low values of friction facilitate nut dilation by easing the
movement of the thread flanks over each other. This phenomenon is also more significant when the nut is tightened rather than the bolt. The lower value of sliding friction allows increased nut dilation.

If the external thread does not pass fully through the internally threaded part than when determining the effective length of engagement for the threads allowance must be made for any end chamfer present. Bolts and screws typically have an end chamfer to aid starting the thread. This typically extends from half to one and a half threads from the end of the screw. The program uses a default chamfer length equal to the thread pitch, if a standard thread size was previously selected. If a nut is being used or if the internal thread is tapped all the way through a plate then the chamfer at the start of the external thread has no effect on the thread engagement length.

To facilitate starting, tapped holes are frequently countersunk from half to one and a half threads. If the hole is countersunk then the effective length of thread engagement is less then the tapping depth. Tests have indicated that for the depth of the countersink, the countersinking contributes about 40% of the strength of an equivalent equal depth without the countersinking. The program defaults assume that one side of the hole is countersunk. This can be changed to either no countersinking present or alternatively countersinking being present on both sides of the hole. The program offers a default size of countersinking for the size of thread being used; this can be changed if required by the user. The default value used by the program for the angle of countersinking is 90 degrees. This to can be changed by the user.

This section allows details of the tapping hole to be entered. The program defaults that the tapping drill diameter will not be used in the thread stripping calculations. In this case, the minor diameter specified on the Thread Size Details page will be used. In tapped holes, the minor diameter and hence thread height is dictated by the diameter of the tapping drill. If the user selects that the tapping drill diameter is to be used then the Tapping Drill Diameter edit box is activated. If a standard thread size has been previously selected, then the program will show a standard tapping drill diameter for this size of thread. The drill diameter entered is based upon the ISO 2306 standard ('Drills for use prior to tapping screw threads'). In general terms the drill diameter shown will be a standard size and will be less than the internal thread maximum minor diameter.

The user can change the size of the tapping drill used if required. To reduce the risk of failure, the Design Engineer is often cautious and specifies high percentages of thread height in tapped holes. From a production standpoint these higher percentages of thread height result in higher tapping torques, increased tap breakage and, as such, are not favoured. For short lengths of thread engagement, the minor diameter size - resulting from the tapping drill - has a significant effect on assembly strength. Tapping costs are likely to be lower if the lowest possible thread height is used. Studies have shown that for threaded assemblies of usual proportions, tap-drill size is relatively unimportant so long as the percentage of thread height is greater than 60%. Typical radial engagement with the external thread based upon the ISO 2306 standard is typically 81.5%.

The effect of a low proportion of thread height is to reduce the shear area of the external thread. However, for very low thread heights, the shear plane through the internal threads need not be parallel to the thread axis. Such failure modes are difficult to predict and can be easily eliminated by maintaining a reasonable percentage thread height (greater than 50% of the thread height).

The exact size of holes produced by a twist drill can be difficult to predict. There are a large number of factors that influence the exact hole size produced by a drill of a stated size. For example, whether drill bushings are
used, the condition, rigidity and accuracy of the drilling spindle and
centrality of the drill web are important factors in determining hole drill
accuracy.

Based upon extensive test work, the drilled hole size is always larger than
the stated drill size. The mean oversize being 0.05mm for 1.6mm drills
increasing to 0.15 for 24mm drills. This is important since the minor
diameter of the tapped hole will be larger than anticipated. An increase in
this diameter will result in a reduction in the shear area of the external
thread and hence the increased likelihood of threads stripping. Thus in
critical cases it is advisable to control the hole size precisely by reaming or
other measures. The program does not allow for holes being oversized
relative to the drill diameter entered - it is up to the user to specifically state
the hole size.

Bell Mouthing

A complicating factor that can occur when a drilled hole is tapped, is bell
mouthing. This is a slight taper on the hole that is usually encountered on
most drilled holes to some degree. This taper extends normally for about
half the diameter from the start of the hole but can extend for a full
diameter. The cause of this tapering is torsional and transverse flexibility of
the drill together with instability of the drill point during entry into the
material. Bellmouthing can be minimised by the use of close fitting, well
aligned and rigid drill bushes together with accurate drill sharpening. As
default, the program uses a default value of the length of bellmouthing of
half the thread nominal thread diameter. This can be adjusted by the user, by
entering a value of 0 into the Length of Bellmouthing box; the program will
effectively ignore bellmouthing effects.

Holes exhibiting bell mouthing will, when tapped, experience a variable
thread height along the length of the hole. This variation can be significant
on short lengths of engagement and fine pitches. The net effect of
bellmouthing is to reduce the shear area of the external threads. The finer
the thread the more pronounced is the effect of bellmouthing. Tests have
indicated that the maximum degree of bellmouthing is approximately 1.03
times the minor diameter. The program uses 1.03 as the default value of the
bellmouthing ratio. The program allows for the bellmouthing effect by
determining the shear area based upon the mean diameter over the length of
bellmouthing.

Thread Stripping Analysis Results

This section explains the results related to the thread stripping analysis.

Effective Length of the Thread Engagement

This is the length of thread engagement that is effective in resisting the shear
forces that are applied to the thread. It is the thread engagement length
entered by the user minus:

- The height of the countersink, single or double sided if present that is
  ineffective in resisting the shear forces acting on the threads.
- The height of the chamfer, if present, on the external thread that is
  ineffective in resisting the shear forces acting on the threads.

Shear Area of the Internal Thread

This is the shear area of the internal thread in the unstrained condition. It is
equal to the area of intersection of a cylinder equal to the major diameter of
the external thread acting on the mating internal thread profile. The critical
dimensions for this area are the length of thread engagement and the
minimum major diameter of the external thread.

Shear Area of the External Thread

This is the shear area of the external thread in the unstrained condition. It is
equal the area of intersection of a cylinder of diameter of the nut minor
diameter acting on the mating external thread profile. The calculated area
takes into account bell mouthing, if present in the internal thread. The
critical dimensions for this area are the length of thread engagement and the maximum internal thread minor diameter.

Note that the overall shear area may be less than the shear area per mm times the thread engagement length if bell mouthing is present in the hole because this effect has been accounted for by the program.

This ratio is the nominal ratio of the uncorrected shear strength of the internal thread compared to the uncorrected shear strength of the external thread. This ratio is used to allow for the effect that the ratio of the strength of the male and female threads has on the bending of the threads and the subsequent strength of the weaker member. The relative strengths of the mating threads affect the localised distortion of the thread form. At small values of this ratio, when the external thread is significantly stronger than the internal thread there will be less distortion of the internal thread near its failure load. This in turn constrains the bending of the nut threads and limits nut dilation resulting in the threads shearing neatly if stripping does occur. Thus a high strength external thread will result in a higher strength internal thread than if similar strength threads are used. This is because the external thread prevents localised distortion of the internal thread and so effectively reduces the tendency of the shear plane diameter to be reduced. This phenomenon is also present when a high strength internal thread is used with a low strength external thread.

As the thread shear strength ratio increases and approaches unity, thread bending does occur increasing the wedging action leading to greater nut dilation. If the value of this ratio is approximately unity, it can be difficult to determine whether it was the nut or bolt thread that sheared first.

The wedging action from the vee threads can adversely affect the location of the thread stripping shear plane. The tensile force present in the fastener during tightening acts on the vee threads to produce a wedging action that results in a radial displacement. This radial displacement is generally known as nut dilation and occurs in threaded bosses as well as conventional nuts. Theoretical and practical studies of this phenomenon indicate that the top face of the nut contracts in a radial direction while its bearing surface expands. The net effect of this dilation is to reduce the shear area of both the internal and external thread.

The stripping strength of an assembly can be improved by increasing the width across flats of the nut, or boss diameter, up to about 1.9 times the nominal thread diameter. This increases the stiffness locally around the internal thread and reduces radial expansion. The program allows for this effect by determining a strength reduction factor based upon the width across flats, or boss diameter, of the internally threaded part.

This factor is to allow for the strength increase, or decrease, to the external thread afforded by the relative thread strengths. A factor less than unity indicates that the shear strength of the external thread is reduced as a result of the deformation of the internal thread.

This factor is to allow for the strength increase, or decrease, to the internal thread afforded by the relative thread strengths. At low values of the strength ratio, the external thread does not deform greatly and maintains approximately the thread flank angle and so reduces any boss/nut dilation that may be present.

This is the direct tensile force that will fail the fastener through tensile fracture of the threaded section. Minimum and maximum values are quoted to allow for the variation in the thread dimensions and minimum and maximum tensile strengths.
When the threaded fastener is being tightened it experiences both tension and torsion. Tension is experienced as a result of the fastener being stretched, torsion as a result of a torque acting due to friction and inclined plane forces in the thread. The effect of these combined tension and torsional stresses is that the fastener will fail at a lower force then if only a directly applied force is applied without any torsion. Higher the thread friction value, higher will be the induced torsional stress and lower will be the direct force that coupled with the torsion will result in fastener failure.

The program states the minimum forces needed to strip the internal and external threads assuming a direct tensile load is applied. For most cases it is more realistic to use the combined tension-torsion loading since most fastener are torque tightened.

The program determines the factor of safety of the internal thread based upon the maximum force required to fail the fastener thread due to the combined effects of tension and the internal thread stripping force. A value below 1 indicates that the internal thread could strip before tensile fracture of the fastener during the tightening operation.

This is the length of thread engagement that would result in the fastener failing due to tensile fracture at the same force as the thread would be stripping. Care should be taken when using this length since the program assumes that effects such as bell mouthing will be reduced linearly as the length of thread engagement reduces. It is up to the user to adjust these values accordingly. The final design lengths should always be run through the program to check the validity of the thread engagement length when coupled with bell mouthing, chamfers and countersink lengths. If the fastener is highly loaded in service then the engineer may decide to base the design on direct forces to fail the fastener rather than allowing for the reduction that occurs if torsional effects are considered from the tightening process.

To assist the user in understanding the significance of the results, the program provides some additional notes at the end of the results section. The key note being that if the bolt breaks on tightening, it is obvious that a replacement is required. Thread stripping tends to be gradual in nature. If the thread stripping mode can occur, assemblies may enter into service which are partially failed, this may have disastrous consequences. Hence, the potential of thread stripping of both the internal and external threads must be avoided if a reliable design is to be achieved.
Joint Analysis

Introduction

The joint analysis feature is the main part of the program. This feature allows an analysis to be completed on a single bolt. This analysis can include both a torque analysis and a thread stripping analysis if these are relevant to the joint design. The Data Entry Form consists of several pages, the user can move between the pages by clicking the tabs of the top of the form.

Remarks Page

This page consists of several boxes that the user can enter details about the analysis. Entering text into any of these boxes is optional, if values are entered these are included in the programs output and will be saved in the data file that the program saves. In the recommendations/notes section text can be entered regarding the results of the analysis or any additional notes that you may wish to include. Unlike the other sections the only practical limit on the amount of text that can be entered is limited by the resources (disk space) on the computer that you are using. The information contained in this section is saved in a text file of the same name as the main data file.

Applied Forces

The program completes an analysis of a bolted joint by considering the forces acting on a single bolt. If a joint consisting of several bolts is being analysed, then the forces applied to the joint must first be appropriately
divided between the bolts prior to these forces being entered into the program.

**Axially Applied Force**

The axial force is that which is applied along the same axis as the bolt. A tensile force has the effect of relieving the bolt's clamp force and increasing the possibility of a gap occurring at the joint face, or slippage of the separate parts of the joint occurring. The program assumes that this force is concentric with the bolt axis. That is, there is no tendency for the bolt to be subjected to a bending moment because of the application of this force.

**Shear Force**

If the joint is loaded transversely, i.e. in shear, then usually a minimum level of clamp force is needed to prevent the joint from slipping. Most joints in mechanical applications are designed to resist shear by the bolts clamping the joint together with sufficient tightness so that the shear force is transmitted by friction between the joint. The bolts do not "feel" the shear force unless loosening or slippage occurs. If the joint slips then additional embedding will occur, further reducing the bolt's preload, that can quickly lead to loosening or failure.

The program requests the force required to prevent any shear movement from occurring. For friction grip joints the force required to prevent shear movement is usually greater than the shear force applied. This is because the coefficient of friction is usually less than unity. Clicking on the button adjacent to the edit box reveals a form that assists the user in entering details of the required shear force. By entering the number of shear planes present (default 1), the coefficient of friction present between the surfaces (default 0.2) and the shear force applied, the form will calculate the clamp force required to prevent shear movement.

The use of dowel pins and other design features can resist shear forces acting on the joint. If no such feature is present then the bolt must provide sufficient clamp force to prevent any possible movement. The program will allow for single, double or multiple shear planes.

**Residual Clamping Force**

In many joints, a minimum level of clamp force must be provided in order that joint integrity is maintained. For example, gaskets must have, typically, a minimum clamp force in order that leakage does not occur. Gasket manufacturers can usually provide details of the minimum clamp force required for the type of gasket used.

If a residual clamping force of zero is specified then there is no reserve for the joint faces to be clamped together when the axial force is applied. That is, if only an axial force is applied the clamp force present between the joints faces may be reduced to zero. This may be satisfactory for some joints, however joints containing gaskets usually require a preload to be always present in order to prevent leakage occurring. Having a high residual clamp force also helps to ensure that the bolt will not loosen when dynamic forces are applied to the joint.

**Lower Limit of the Dynamic Force**

If the applied axial force varies in magnitude, then the bolt may fail due to fatigue. The upper value of the dynamic force should be entered in the axially applied force entry box and the lower value in the Lower Limit of the dynamic force entry box. The dynamic force can alternate between a maximum positive value (whose value should be entered in the axial force box) and a negative value. If a negative value is present, (i.e. a force that is tending to increase the clamp force present between the clamped members) then the lower limit of the dynamic force would be entered as a negative value. Further details are provided by clicking on the button at the right of the form.

**Bolt Details**

The Bolt Details page of the joint analysis data form is very similar to the one displayed on the Torque Analysis data form. There is however an
additional feature, the user can specify whether the fastener head or the nut is tightened. On the Torque Data Entry form, the calculations are completed on the basis of the fastener head sizes displayed on this form. In a joint analysis, if the fastener head bearing diameter is different to the nut, then the results will be different if the fastener head or the nut is rotated. Because the bearing diameters also affect the bearing stress, specifying sizes for both can be important. The appropriate fastener head and nut type will also be displayed on the joint drawing.

**Fastener Fatigue Properties**

The program will calculate the alternating stress present in the bolt threads based upon the forces acting and the joint characteristics. In the results, the program compares the calculated alternating stress to the fatigue endurance limit. The program allows a major factor that determines the fatigue strength of the fastener, that is, whether or not the threads have been rolled. Most standard threads for fasteners are rolled and are heat treated before rolling.

The thread (in all except the most unusual of fasteners) represents the weakest link, with regard to fatigue, for the fastener. Rolled threads have higher fatigue endurance strength than machine cut threads. Rolling the threads after heat treatment is also more beneficial then before. Select the appropriate check box for the application.

If the 'User Defined Properties' check box is selected, then a form appears for the user to enter a specific value for the fatigue endurance strength.

All materials have a tendency to fail under repeated loading at a stress level considerably less than the static strength of the material. This characteristic of materials is known as fatigue and it is a common cause of failure in many products, including fasteners.

Fatigue failure is the result of repeated plastic deformation. The thread represents a severe notch and subsequently at the root, localised stresses beyond yield are sustained. It has long been known that:

1. The surface condition of the part subjected to fatigue is important. Specifically, cold working, such as involved with thread rolling, induces residual compressive stresses in the thread that have a beneficial effect on fatigue life. Rolling the thread after heat treatment is more beneficial than thread rolling before heat treatment because these compressive stresses are better retained. Machine cut threads relative to rolled threads compare badly in regard to fatigue because such threads do not have such beneficial compressive stresses.

2. As the diameter of a specimen increases, the fatigue endurance limit decreases. Thus as the bolt diameter increases the endurance strength decreases.

The program will calculate the alternating stress in the thread of the bolt based upon the forces entered and the characteristics of the joint. The fatigue strength of the bolt material will be calculated by the program if the defaults are accepted. The program uses lower bound empirically derived values for the fatigue endurance strength. If the user has specific information on the fatigue endurance strength of the fastener being analysed then this should be used in preference to the program calculated value. The user should satisfy himself/herself that the fatigue endurance estimate being proposed by the program is appropriate for the specific application.

The defaults used by the program assume that steel bolts of grade 8.8, 10.9 or 12.9 are used and the threads are rolled. Based upon experimental work, if the threads are rolled before heat treatment there is little dependence upon
the preload. For threads that are rolled after heat treatment it has been found that there is a dependence upon preload. The program determined endurance limit includes the preload dependence when applicable as well as size dependency. (It has been found that the endurance strength reduces with increasing diameter.) The range of validity of the program determined endurance values is between 0.2 and 0.8 of the yield force of the bolt. For materials other than steel and for special thread profiles the user is required to enter a fatigue endurance limit.

It is well known that it is good design practice to ensure that the joint faces, which the nut or the bolt head seat on, are at right angles to the bolt axis. Angle errors on the joint face do have an influence on the static strength of bolted joints, however for small angles, this is usually not significant. What is less well known is that even small angularity errors can have a catastrophic effect on fatigue life. An influencing factor relating the angularity error to the fatigue endurance strength is the amount of clearance in the thread. A large thread clearance allows the nut to rock slightly on the bolt thread and so reducing the bending stress that will be imposed on the bolt thread. The length of thread engagement also as an influence on this effect. Tests results indicate that although a large thread clearance will assist in minimising the effect of angularity, it will not eliminate it.

By clicking on the button 'Allow for Joint Face Angularity' on the fatigue properties section of the Data Entry form will display a form giving details of the information requiring to be entered to allow for an angularity effect. The program requires that you first enter the thread details into the program so that the thread clearances can be determined. The calculation is based upon 32 fatigue tests completed on nuts having deliberate angle errors on their bearing faces. Equations have been fitted to the results allowing an estimate to be provided for the fatigue endurance strength given a particular angle, thread clearance condition and length of engagement. The results should be used with caution because of the limited test results that are available.

So called self aligning nuts and washers are available with spherical bearing surfaces to compensate for the effects of joint face angularity. Such products need to be used with care since at small angles they will not completely self align because of the effect of friction.

**Modulus of Elasticity**

The Modulus of Elasticity or Young's Modulus of the fastener is a property that is used in determining the bolt stiffness and hence from this the amount of the applied force the bolt actually sustains. A default value for steel is used. The user can enter a value as appropriate for the particular bolt material that is being used.

The Modulus of Elasticity, or Young's Modulus is a factor in determining the joint stiffness and hence from this the amount of the applied force the bolt actually sustains. A default value for steel is used. By clicking on the 'Aluminium' check box a value suitable for this material will be entered. By clicking on the 'Other' check box, the user can enter a value as appropriate.

---

2 From the publication 'Effect of Manufacturing Technology and Basic Thread Parameters on the Strength of Threaded Connexions' by A. I. Yakusev translated from the Russian by S. H. Taylor, published by Pergamon Press.
Clamped Parts Stiffness Details

In this section, details of the stiffness of the joint are entered. There are also three check boxes, with the 'Plate' box acting as default. If the joint consists of different materials, or the cross sectional area is not constant then a multi-plate analysis can be performed. You may also wish to select this option if elongated holes are being used in the joint. Multi-plate analysis is discussed in a separate section.

If the joint surface dimensions exceed the outer diameter of the bolt bearing area, the compressive stress induced by the bolt into the joint will decrease the further you move away from the bolt. Due to this, the stiffness of the joint can be problematical to calculate. The program determines the joint stiffness by using an equivalent area approach based upon research conducted into the subject. (The default used by the program is for a plate, i.e. the clamped parts consist of plates that are wide in comparison to their thickness.)

If a boss or sleeve is present in the design, when the user clicks on the 'Boss or Sleeve' box a form appears into which the user is prompted to enter the boss diameter. If the boss is non-circular in cross section, enter the diameter of the boss that would be enclosed within its area.

The stiffness of the clamped parts is determined assuming that the parts lie snugly on each other and their surfaces are flat. It is also assumed that the joint contains no gasket and that the clamp length to bolt diameter ratio is no greater than eight. If these assumptions cannot be made or if the joint contains a large number of thin sheets clamped together (which as the effect of reducing the joint stiffness) then the program cannot accurately determine the stiffness of the clamped parts. In such cases the stiffness of the clamped parts should be determined experimentally or based upon the characteristics of existing joint designs. In such cases, enter a user defined stiffness value by clicking on the button marked 'User Defined Stiffness' and a form will appear allowing a value to be entered.

Multi-plate Analysis

A separate form is used to allow a user to enter details about the joint, when several layers of differing materials are used. The program allows up to 20 different layers to be entered, each can be a different material, hole size etc.

Two types of joint plate are considered, plate or boss, a plate having effectively limitless width and a boss having a specific diameter. It is important to list the joint items in the order that they are present in the joint. That is, the first entry relates to the top surface of the joint, the second to the one directly underneath and so on through the joint.

Two types of elongated holes can also be considered. Type A has fully rounded ends and is the traditional elongated hole, type B with a radiused end is used often when the hole is punched since it leads to a longer tool life.

For each joint material, the modulus of elasticity and the compressive yield strength is requested. For the modulus of elasticity, clicking on the appropriate button, steel or aluminium, will allow typical values to be entered. Other wise, the user can directly enter an appropriate value. The compressive yield strength of the joint material is requested so that it can be subsequently checked against the bearing stress calculated by the program.

Fastener Clearance Hole

To allow a bolt to enter the joint there is generally a clearance between the bolt and the hole. There are four check boxes in this section, with the 'Fine'
button acting as default, fine meaning a small amount of clearance between the bolt and the hole as compared with coarse.

By clicking on the appropriate button the clearance hole related to the particular hole series defined in ISO 273:1979 (Fasteners - Clearance holes for bolts and screws) is selected if a standard bolt diameter has been previously selected. If the user wants to select a specific value not within this standard, by clicking on the 'User defined' check box the data entry box can be edited.

The majority of joints incorporating bolts in general mechanical engineering applications have clearance holes (that is, the hole diameter is larger than the bolt diameter). Clearance holes improve the ease in which the product can be assembled and, by allowing a reduction in the tolerances required, reduce manufacturing costs. Transmission of any shear forces, which may be acting on the joint, is achieved, not by interference of the bolt shank to the hole wall, but by friction grip between the joint surfaces. The amount of friction grip generated is directly proportional to the amount of preload produced by the bolts.

The program will allow the user to select the appropriate clearance hole based upon ISO 273. Alternatively, the user can enter the appropriate size if required. When the clearance hole is only slightly larger than the bolt diameter, care should be taken to ensure that interference is avoided between the edge of the hole and the underhead fillet of the bolt. A chamfer may be necessary. This is to avoid high-localised stresses that could lead to excessive preload loss from embedding.

**Load Introduction Level**

In this section, details of the location of the load introduction level are entered. There are two check boxes in this section, with the '0.5' button acting as default.

The applied force sustained by the joint can be considered to act at some location within the joint. The factor \( n \) is the proportion, as a decimal fraction, of the clamped length, which lies between the load introduction levels. (See diagram on the form that appears when the 'Help - Load Introduction' button is pressed.)

The 0.5 default infers that the load introduction is at the midpoints within the plates comprising the joint. The smaller the value of \( n \), the smaller will be the proportion of applied force sustained by the bolt. Each design will be different and should be individually assessed. The value of \( n \) can be reduced by the introduction of design features such as raised bosses. If the 'User entered value' is clicked, a user determined value can be entered into the edit box.

**Embedding Details**

In this section, details of the amount of embedding sustained by the joint are entered. There are two check boxes in this section, with the 'Program Calculated' box acting as default. If the user clicks the 'User entered value' button then a form appears allowing a user determined value to be entered into the program. This form contains further information about embedding.

Clicking on the button in this section will provide further details about the amounts of embedding used by the program to calculate preload loss. Rougher the surface of the joint – the larger is the amount of loss from embedding. The form allows the user to select the appropriate value from the list of surface finishes identified.

Embedding is localised plastic deformation that occurs under the bolt head and nut, in the thread and between the joint layers. This occurs as a result of
high pressures deforming the contact points between the joint materials resulting in plastic deformation and preload loss.

The plastic deformation sustained can be estimated by the program based upon published experimental results. Alternatively, the user can enter the amount of embedding, if such information is available, for the joint under analysis.

If the bearing pressure under the bolt head or nut exceeds the limiting surface pressure for the joint material then embedding probably will significantly exceed the calculated value.

Research indicates that the joint material has no significant influence on embedding so long as the material's compressive yield strength is sufficient to withstand the bearing stress. The surface roughness of the clamped parts also has only a small influence on the amount of embedding. This is thought to be due to the smoothing of the joint surface that occurs from the tightening process.

**Limiting Surface Pressure for the Joint Material**

In this section, details of the value of the limiting surface pressure for the joint material are entered. Very high surface pressures can exist under the nut face (or bolt head). If such pressures exceed the compressive yield strength of the joint material, plastic deformation will occur. Such deformation will result in bolt preload loss.

Typically, the compressive yield strength of steel is similar to the tensile yield strength, however usually it is slightly higher. For example, the bearing stress for aluminium as been found to be approximately 1.3 times the tensile yield strength.

Standard fasteners tend to have a relatively small bearing area and accordingly, high bearing stresses often result. The use of flanged head fasteners or washers will increase the bearing area and reduce the bearing stress. The default value used by the program (the value can be changed to any user-determined value) is typical for mild steel.

Plain washers are frequently used to reduce the bearing stress of the nut onto the joint material. However loose fitting washers can cause problems because of the washer being eccentric to the bolt axis. (The edge of the washer hole being against one side of the bolt shank.) This eccentricity can result in a high bearing force concentration that in turn can lead to frictional changes occurring between the nut-washer interface. This can cause relative motion to be transferred from the nut-washer interface to the washer joint interface. Such a change in relative motion can result in the bearing face torque changing by 15%. To assist in minimising this effect, consideration should be given to reducing the clearance hole in the washer so that any eccentricity would be reduced.

If loose washers must be fitted, it is necessary to ensure that they are of a sufficient hardness so that they are capable of sustaining the high bearing stress to prevent excessive embedding from occurring.

**Tightening Factor**

The tightening factor is a measure of the scatter in a bolt's clamp force because of the tightening method used to tighten the fastener. It is defined as the maximum bolt preload divided by the minimum value anticipated for that tightening method.
The method used to tighten a joint consisting of bolts plays a large influence in determining the size and number of bolts required for a particular application. The effect that the tightening method has on bolt sizing is largely underestimated. If several bolts are tightened by any single method, then variation in the bolt's preload can be anticipated. The ratio of maximum bolt preload to minimum is known as the tightening factor. Research completed into the subject over the last thirty years has established guide values for the tightening factor for the most common tightening techniques.

Determining the appropriate tightening factor is of crucial importance to a successful joint design being completed. The basis of the methodology is that if a successful joint is to be designed then it should be designed on the basis of the lowest anticipated preload as against designing it based upon the mean or highest value. There are two ways that the program provides to allow an estimate of an appropriate tightening factor to be established.

- Selecting a tightening factor from a table based upon the tightening method proposed to be used in the assembly. The values have been experimentally determined. If non-torque control methods are proposed to be used then this is the appropriate method to use to select the tightening factor.

- Allowing the program to determine the tightening factor based upon the frictional scatter present, the variation in the prevailing torque value, the torque specification and the accuracy of the tightening tool proposed to be used. This method is appropriate as an alternative to selecting a value from a table when torque control is being used. It allows the preload variation to be assessed based upon a tighter control of friction values and of the torque specification.

The more accurate the tightening method, the smaller the size of bolts that are needed for a particular application. To give an example, if, to prevent leakage from a joint containing a gasket, a total clamp force of 10000 N is needed. Then if a torque wrench is used that has a tightening factor of 1.6 then the bolts must be sized on the basis of 1.6 x 10000 = 16000 N. (To guarantee the 10000 N needed, bolts would have to be sized capable of sustaining a 16000 N preload - since this would be the maximum value that would result.) However if an impact power wrench was to be used, tightening factor 3, then the bolts need to be sized on the basis of 3 x 10000 = 30000 N. Almost twice the number of bolts being needed or the same number of a larger diameter.

Clicking the button marked 'Select the Tightening Factor Directly From a Table' allows a form to be displayed that provides guide values for the Engineer to select.

Clicking the button marked 'Determine the Tightening Factor Based Upon Frictional Scatter' displays a form that allows the user to enter data that will allow the program to determine an appropriate value for the tightening factor directly. This form is discussed in more detail in the section presented below.

This form allows the user to enter details that have a major influence on a bolt's preload when the torque tightening method is being used. Covering each of the sections of the form in turn:

**Tightening Tool Accuracy**

Torque tightening tools have a limited accuracy. If the tool is set to a specific torque value then there will be slight variations in the ability of the tool to deliver that torque value. For example, if the tool has an accuracy of 5% and a torque value of say 100 Nm is set, then the bolt could sustain a
torque between 95 Nm and 105 Nm (5% of 100 Nm is 5 Nm). This variation in the torque value directly influences the preload scatter.

Another factor that influences tightening tool accuracy is the stiffness of the joint. A 'hard' joint has a high stiffness (does not compress significantly under the loading applied by the bolt). A hard joint is defined (according to ISO 5393) as one whose torque value is achieved after the bolt has turned approximately 30 degrees of rotation from the snug level. The 'snug' state is when the plates comprising the joint are in metal to metal contact - no gaps present. A 'soft' joint in comparison has low stiffness and whose final torque is reached after approximately 720 degrees of bolt rotation. Most joints fall between these two extremes. The reason why the hardness of the joint can be of importance is that with many tightening tools the accuracy is influenced by the joint stiffness. A soft joint absorbs more energy during the tightening process that can lead to limited accuracy. This applies to impact and impulse nutrunners. One way to improve the tightening accuracy is to tighten using a power tool to below the specified torque value and then use a calibrated torque wrench to tighten the bolts to the final torque value.

Thread and Head Friction Variation

Even if the tightening tool can perfectly accurately tighten the bolt to a specified torque, then significant preload variations can still result from variations in the friction values under the bolt head and in the thread. The form allows a user to enter details about the friction variation to be entered and so accounted for in the calculations. The larger the difference between the maximum and minimum friction coefficients -the larger will be the preload scatter.

Two distinct friction values are required to be entered into the program. The friction coefficient in the bolt threads and the friction coefficient under the bolt head or nut face. Differing values can be entered for both.

The thread friction value is determined by the finish applied and the state of lubrication. The friction coefficient under the bolt head or nut face depends upon the finish applied to the bolt, the lubrication condition and the joint material and surface condition. The program provides help in selecting the appropriate friction value for a particular finish. By clicking on the button marked 'Select Finish/Material Details from a database provided with the program. The friction values corresponding to the proposed finish can be selected from the list provided.

As previously mentioned the preload scatter is directly influenced by maximum and minimum friction coefficients. Entering the friction details on this form allows this scatter to be calculated. However the friction values used by the program to determine the appropriate tightening torque to be specified and the preload resulting from the tightening torque is based upon the minimum value entered. Such minimum values will result in a lower bound value for the tightening torque. This will reduce the possibility of bolt failure occurring when the bolt is initially tightened with prevailing friction conditions lower than anticipated. The effect that friction variation has on bolt preload variation is taken into account by the tightening factor. The lowest friction value results in the highest bolt preload value - this will reduce if the friction increases. Using the higher value of preload allows the bearing stress to be checked. Whether or not there is sufficient clamp force present is checked by comparing the applied forces, factored up by the tightening factor, to the calculated preload value.

Prevailing Torque Value

The prevailing torque is the torque required to run a nut (or bolt) down a thread on certain types of fasteners that are designed to resist vibration
loosening. This prevailing torque can be provided by an insert in the nut/bolt thread, by using nuts that have their threads locally distorted or by using micro-encapsulated adhesive applied to the threads.

If a bolt diameter has been entered into the program, by clicking on the appropriate selection, the program will enter lower and upper bound values for the prevailing torque – check whether these are applicable to your application.

The characteristics of the majority of prevailing torque fasteners (nylon/polyester patch and distorted head types) are such that the magnitude of the prevailing torque reduces as the number of installations and removals increase.

Because significant variations in the prevailing torque can be present then this will result in subsequent variations in the preload scatter. By entering maximum and minimum prevailing torque values onto the form, the program will include this influence when calculating the preload scatter.

Bolts which have a prevailing torque are frequently used where there exists a risk of vibration loosening. The prevailing torque counteracts the off torque that can be present when the bolt is subjected to vibratory loading. During the tightening process, this prevailing torque has the effect of increasing the torsional stress in the shank of the bolt. For the same state of combined stress in the bolt, the higher the torsional stress in the bolt, the lower will be the resulting preload. For the same frictional conditions, the total tightening torque required to tighten the bolt so that so that a specified combined stress exists in the shank of the bolt, does not significantly increase with increasing prevailing torque. However the preload in the bolt can be significantly reduced.

The use of threadlocking adhesives, such as structural cyanoacrylate and anaerobic compounds, results in the bolt/nut exhibiting a prevailing torque characteristic. This effect also occurs with adhesive contained within micro-beads applied to the threads. The magnitude of the prevailing torque is generally less than with proprietary prevailing torque fasteners, however the use of threadlocking compounds does still affect the tightening torque specification and the resulting preload.

The values quoted by the program should only be used when more specific information is not available from the fastener manufacturers or from test work.

**Joint Analysis Results**

Once data has been entered on the Data Entry Form then, on clicking the Calculate button, the results will be displayed on the main form. The summary of the results will be displayed at the end of results listing. The summary of the results can alternatively be viewed by clicking the option from View on the main menu, or by clicking the appropriate speed button. The results present factors of safety for five possible failure modes:

1. Failure of the bolt to provide sufficient clamp force.
2. The bolt being overloaded by the applied force.
3. Fatigue failure of the bolt.
4. Excessive bearing stress under the bolt head or nut face.
5. Thread stripping failure if a tapped hole is used.

Considering each of these modes in turn:
The main factor as to whether a bolted joint will sustain the applied forces is whether the bolts will generate sufficient clamp force. For any application there is a minimum clamping force required to prevent joint failure. If the bolt is unable to generate sufficient clamp force, after taking into account any likely scatter in its value that is likely to occur, then it can be considered to have failed to meet its functional requirement. The clamping force is required to prevent joint movement being caused as a result of axial and/or shear forces. It is also required in some applications to allow gasket compression to happen to prevent leakage occurring. The analysis takes both axial and shear forces, acting individually or simultaneously, into account, together with any residual clamp force that may be needed to maintain a functional requirement such as gasket sealing. In many instances, when a gap in the joint or slippage occurs, then failure by the bolt loosening or fatigue will occur. When fatigue failure occurs, the cause is frequently insufficient preload rather than poor fatigue strength.

The decompression point is a term used to denote the condition at which there is zero pressure at the joint interface as a result of forces applied to the joint. If the applied force is increased beyond the decompression point, a gap will form at the interface. Analytically, a criterion of joint failure is often taken as when the applied force on the joint reaches the decompression point. This is because forces acting on the bolt(s) can dramatically increase at this point. Loading beyond this point can also result in fretting at the interface that will lead to bolt tension loss that will subsequently lower the decompression point. This process can continue until bolt failure does occur. The failure can be by fatigue or other mechanism but the underlying cause was loading of the joint beyond the decompression point. It is for this reason that it is frequently taken as a failure criteria in analysis work.

If a very high axial force is applied to a joint there is the possibility that the bolt will sustain additional loading that will cause its yield strength to be exceeded. If this does occur then either the bolt will fail due to direct tensile failure, or, when the load is removed, will sustain a plastic deformation that will result in preload loss that could cause the bolt to loosen. The program checks for this possible failure mode and determines a safety factor. For the majority of joints, failure by direct overloading is unlikely because the bolt usually sustains only a small proportion of any force applied to the joint.

All materials have a tendency to fail under repeated loading at a stress level considerably less than the static strength of the material. This characteristic of materials is known as fatigue and it is a common cause of failure in many products, including bolts. The program will calculate the alternating stress in the thread of the bolt, based upon the forces entered and the characteristics of the joint. The program will calculate the fatigue strength of the bolt material, based upon the user's selection as to whether the bolt thread was rolled or machine cut. The program uses lower bound, empirically derived values for the fatigue endurance strength. If the user has specific information on the fatigue endurance strength of the bolt being analysed, then this can be used in preference to the program value.

If the bearing stress under the nut face exceeds the compressive yield strength of the joint material, plastic deformation will occur. Typically the compressive yield strength is higher than the tensile yield strength. This can be due to work hardening of the joint surfaces during the tightening process. Usually a conservative estimate is to assume that the compressive yield strength is equal to the tensile yield strength. The preload loss from embedding determined by the program assumes that the compressive yield strength is not exceeded. If it is, then such preload loss can increase uncontrollably. The program checks that the bearing stress is within acceptable limits and if it is not, recommends a number of design alternatives. Specifically, either that flanged fasteners or hardened washers should be used to reduce the bearing pressure.
5. Thread Stripping

If the bolt is secured into a tapped hole then the program will present results pertaining to the likelihood of thread stripping occurring. If a nut is presented then this analysis is not included. Standard nuts have been sized on the basis that the bolt thread will always sustain tensile fracture before thread stripping will occur. The appropriate grade of nut should always be used that matches with the grade or property class of bolt that is being used.
The BOLTCALC Databases

Introduction

To assist the user with data input and to provide relevant information that the user may not have access to, the program uses several databases. The databases are accessed by the BOLTCALC program on a read only basis so that records cannot be changed. A separate program, dbEditor, is provided if you wish to change, add or delete values from the databases. Since all standard thread sizes and common bolt material specifications are included in these databases, you may not have need to add to or change the databases. This section is included for those users who may wish to do so.

The databases themselves are in dbase IV format and can be read into a spreadsheet program such as Microsoft Excel. However, if you do make changes to a database and save it from Excel then BOLTCALC will report an error because Excel saves all fields as string variables. (Some of the fields are numeric however Excel saves these as string.) The databases can also be loaded into a full database program such as Microsoft Access. Access will save the files with the correct field assignments. You may wish to use such a program instead of the database editor supplied. Not all the fields in each database are used by the program in any calculations completed - some of the fields are displayed for information purposes only.

The databases that the program uses are:

- mthd.dbf Metric thread database
- ithd.dbf Inch thread database
- mprops.dbf Metric bolt property database
- iprops.dbf Inch bolt property database
- threadu.dbf Thread friction coefficient database
- headu.dbf Head friction coefficient database

The fields that each record is made from are presented in the following sections:

Thread Friction Database

This database (threadu.dbf) lists the coefficient of friction for the thread for various finish conditions and materials. Each record consists of a number of fields, the name and meaning of each field is presented below:

DETAILS Details about the fastener finish condition.
NOTES Notes relating to the record such as whether the minimum and maximum are calculated or test determined results.
THDUMIN  The minimum thread coefficient of friction for the finish condition.

THDUMAX  The maximum thread coefficient of friction for the finish condition.

THDUMEAN  The mean or average thread coefficient of friction for the finish condition.

**Nut Face Friction Database**

This database (headu.dbf) lists the coefficient of friction for the nut face or under the bolt head for various finish conditions and materials. Each record consists of a number of fields, the name and meaning of each field is presented below:

- DETAILS  Details about the fastener finish condition.
- NOTES  Notes relating to the record such as whether the minimum and maximum are calculated or test determined results.
- HEADUMIN  The minimum head coefficient of friction for the finish condition.
- HEADUMAX  The maximum head coefficient of friction for the finish condition.
- HEADUMEAN  The mean or average head coefficient of friction for the finish condition.

**Material Properties Databases**

The two material property databases (mprops.dbf for metric thread materials and iprops.dbf for inch based thread materials) have the same structure in terms of the fields that make up each record. The names and meaning of each field are:

- MATERIAL  The type of material the fastener is made from i.e. steel, stainless, nonferrous.
- ORG  The standards organisation who is responsible for the standard i.e. ISO, SAE or ASTM.
- GRADE  The strength grade or property class.
- STANDARD  The standard that the fastener is made to such as ISO 898
- SIZE  The nominal size of the product.
- MAT_NOTES  Notes on the material.
- PROOF  Proof load stress.
- YIELD  The yield stress or 0.2% non-proportional limit specified in the standard - this is a key value used by the program.
- MINTENSILE  Minimum specified tensile strength - this is a key value used by the program for a thread stripping analysis.
- MAXTENSILE  Maximum specified tensile strength.
- MINHARD  Minimum core hardness.
- MAXHARD  Maximum core hardness.
- UNITSHARD  Hardness units used such as Rockwell.
- MINELONG  Minimum elongation as a percentage after fracture.
MINAREARED Minimum reduction of area as a percentage after fracture.
GRADEIDENT Grade identification marking on the bolt head.
MARKNOTES Notes on grade identification.
MANFMARK Whether manufacturers marking is required.
IMPACTSTH Impact strength.
TEMPERTEMP Tempering temperature in C.
E Modulus of Elasticity -this is a key value used by the program.
DENSITY Material density.
POISSON Poisson's ratio.
EXPANSION Coefficient of thermal expansion.
MINSHEAR Minimum shear strength.
MAXSHEAR Maximum shear strength.
MAXTENSION Maximum tensile strength (same as MaxTensile if the standard specifies a value) - this is a key value used by the program in the thread stripping analysis.
G Shear Modulus.

Thread Databases

The two thread databases (mthd.dbf for metric thread details and ithd.dbf for inch based thread details) have the same structure in terms of the fields that make up each record. (The ithd.dbf database has additional fields that are noted at the bottom of this section.) The names and meaning of each field are:
DESCRIBE Description of the thread including tolerance.
THREAD Description of the thread size and pitch.
THDSERIES Fine, Coarse or constant pitch.
DIAMETER Nominal Thread Diameter.
PITCH Thread Pitch.
TOLCLASS Tolerance Class for both the external and internal thread.
DMAX_ET Major Diameter External Thread Maximum Size.
DMIN_ET Major Diameter External Thread Minimum Size.
D2MAX_ET Pitch Diameter External Thread Maximum Size.
D2MIN_ET Pitch Diameter External Thread Minimum Size.
D1MAX_ET Minor Diameter External Thread Maximum Size.
D1MIN_ET Minor Diameter External Thread Minimum Size.
DMIN_IT Major Diameter Internal Thread Minimum Size.
D2MAX_IT Pitch Diameter Internal Thread Maximum Size.
D2MIN_IT Pitch Diameter Internal Thread Minimum Size.
D1MAX_IT Minor Diameter Internal Thread Maximum Size.
D1MIN_IT Minor Diameter Internal Thread Minimum Size.
TAPDRILL Tapping drill size.
<table>
<thead>
<tr>
<th>Field</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>HOLEFINE</td>
<td>Clearance hole diameter Fine series (if metric thread then size is based upon ISO 273:1979).</td>
</tr>
<tr>
<td>HOLEMED</td>
<td>Clearance hole diameter Medium series (if metric thread then size is based upon ISO 273:1979).</td>
</tr>
<tr>
<td>HOLECRSE</td>
<td>Clearance hole diameter Coarse series (if metric thread then size is based upon ISO 273:1979).</td>
</tr>
<tr>
<td>HEADPLN</td>
<td>Washer face diameter of standard hex nut or bolt head - smallest size used.</td>
</tr>
<tr>
<td>HEADFLAN</td>
<td>Washer face diameter of flanged hex nut or bolt head - smallest size used.</td>
</tr>
<tr>
<td>SOCKETHD</td>
<td>Minimum head diameter of socket head cap screw.</td>
</tr>
<tr>
<td>PTNOMETL</td>
<td>Maximum prevailing torque for nonmetallic resistant element screws.</td>
</tr>
<tr>
<td>PTCHEM</td>
<td>Maximum prevailing torque for preapplied chemical coating.</td>
</tr>
<tr>
<td>PTMET59</td>
<td>Maximum prevailing torque for steel hexagon nuts classes 5 and 9 (metric database) or class A (inch database).</td>
</tr>
<tr>
<td>PTMET10</td>
<td>Maximum prevailing torque for steel hexagon nuts class 10 (metric database) or class B (inch database).</td>
</tr>
<tr>
<td>PTNOMETLM</td>
<td>Minimum prevailing torque (fifth removal torque) for nonmetallic resistant element screws.</td>
</tr>
<tr>
<td>PTCHEMM</td>
<td>Minimum prevailing torque for preapplied chemical coating.</td>
</tr>
<tr>
<td>PTMET59M</td>
<td>Minimum prevailing torque (fifth removal torque) for steel hexagon nuts classes 5 and 9 (metric database) or class A (inch database).</td>
</tr>
<tr>
<td>PTMET10M</td>
<td>Minimum prevailing torque (fifth removal torque) for steel hexagon nuts class 10 (metric database) or class B (inch database).</td>
</tr>
<tr>
<td>HEADCSK</td>
<td>Head diameter of a countersunk head cap screw.</td>
</tr>
</tbody>
</table>

The inch thread database (ithd.dbf) has the following two fields, in addition to the above, to allow the properties of class C nuts to be included.

<table>
<thead>
<tr>
<th>Field</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>PTMETC</td>
<td>Maximum prevailing torque (in Lb-ft) for steel hexagon nuts class C.</td>
</tr>
<tr>
<td>PTMETCM</td>
<td>Minimum prevailing torque (fifth removal torque in Lb-ft) for steel hexagon nuts class C.</td>
</tr>
</tbody>
</table>

**Torque Range Databases**

The torque range databases mtorque.dbf and itorque.dbf (for metric and inch based torque ranges respectively) are used to assist the user when the company uses specific torque ranges to reduce the number of torque ranges that are available on the factory floor. The databases are used when user specified torque values are used. The torque units used are Nm for metric sizes and lb-ft for inch based sizes. The two databases have the same structure, specifically:

<table>
<thead>
<tr>
<th>Field</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>MEANTORQ</td>
<td>Mean torque value specified (information only).</td>
</tr>
<tr>
<td>MINTORQ</td>
<td>Minimum specified torque value.</td>
</tr>
</tbody>
</table>
MAXTORQ       Maximum specified torque value.
DRIVESIZE     The size of the drive tool for the torque values
               (information only).
NOTES         Notes about the torque range (information only).
Glossary of Terms

**bearing stress**
The stress under the bolt head or nut face. It is the total force in the bolt divided by the area between the inner diameter of bolt head or nut face and the outside diameter.

**bell mouthing**
This is a slight taper on the hole, which is usually encountered on most drilled holes to some degree. The cause of this tapering is torsional and transverse flexibility of the drill together with instability of the drill point during entry into the material.

**chamfer**
A tapered surface that is usually present at the starting end of an externally threaded fastener to aid starting the thread.

**countersink**
A tapered section at a start of a hole. Internal threads are often countersunk to aid mating with the external thread.

**countersunk**
A tapered section at a start of a hole. Internal threads are often countersunk to aid mating with the external thread.

**decompression point**
The decompression point is a term used to denote the condition at which there is zero pressure at the joint interface as a result of forces applied to the joint. If the applied force is increased beyond the decompression point, a gap will form at the interface. Analytically, a criterion of joint failure is often taken as when the applied force on the joint reaches the decompression point. This is because forces acting on the bolt(s) can dramatically increase at this point. Loading beyond this point can also result in fretting at the interface that will lead to bolt tension loss that will subsequently lower the decompression point. This process can continue until bolt failure occurs. The failure can be by fatigue or other mechanism but the underlying cause was loading of the joint beyond the decompression point. It is for this reason that it is frequently taken as a failure criteria in analysis work.

**external thread**
A screw thread that is formed on an external cylinder, such as on bolts, screws, studs etc.

**fatigue**
The tendency for materials to fail under repeated loading at a stress level considerably less than the static strength of the material. This characteristic of materials is known as fatigue and it is a common cause of failure in many products, including bolts.

**friction**
A dimensionless number representing the ratio of the friction force to normal force. Typically for threaded connections it is about 0.12 but can vary significantly depending upon the materials used and whether a lubricant has been used.

**friction**
Mechanical resistance to the relative movement of two surfaces.

**hard joint**
A 'hard' joint has a high stiffness (does not compress significantly under the loading applied by the bolt). A hard joint is defined (according to ISO 5393) as one whose torque value is achieved after the bolt has turned approximately 30 degrees of rotation from the snug level. The 'snug' state is
when the plates comprising the joint are in metal to metal contact - no gaps present.

A screw thread which is formed in holes, such as in nuts.

This ratio is the nominal ratio of the uncorrected shear strength of the internal thread compared to the uncorrected shear strength of the external thread.

The nominal length of engagement of the external thread into the internal thread.

The major diameter of an external thread is the diameter of an imaginary cylinder parallel to the crests of the thread. Both maximum and minimum diameters are requested to be entered. If a standard ISO metric thread size is being used than the program will present values appropriate for a 6H/6g nut/bolt tolerance class. These values can be over written if required.

The minor or root diameter is the diameter of a cylinder that just touches the roots of the thread.

This is the diameter commonly used to describe the thread. For example, for a M8x1.25 fastener the nominal thread diameter is 8mm.

Under load, the wedging action of the threads causes dilation of the nut resulting in an increase in the minor diameter of the nut and reducing the effective shear areas of both the external and internal threads.

This is the distance from the top of one thread crest to the next. If the outside diameter entered previously is the same as that for a standard metric thread, the pitch for this thread will be shown. For example if 8mm was previously entered for the nominal diameter then on entering this field a value of 1.25mm would be shown.

The pitch or effective thread diameter of the external thread is the diameter that has equal metal and space widths. Put more simply, it is the mean diameter of the thread.

The clamp generated by a bolt or other threaded fastener when initially tightened.

The prevailing torque is the torque required to run a nut (or bolt) down a thread on certain types of fasteners that are designed to resist vibration loosening. This prevailing torque can be provided by an insert in the nut/bolt thread, by using nuts that have their threads locally distorted or by using micro-encapsulated adhesive applied to the threads.

A designation system which defines the strength of a bolt or nut. For metric fasteners, property classes are designated by numbers where increasing numbers generally represent increasing tensile strengths. The designation symbol for bolts consists of two parts:

1. The first numeral of a two digit symbol or the first two numerals of a three digit symbol approximates 1/100 of the minimum tensile strength in MPa.

2. The last numeral approximates 1/10 of the ratio expressed as a percentage between minimum yield stress and minimum tensile stress.

Hence a fastener with a property class of 8.8 has a minimum tensile strength of 800 MPa and a yield stress of 0.8 x 800 = 640 MPa.

The designation system for metric nuts is a single or double digit symbol. The numerals approximate 1/100 of the minimum tensile strength in MPa. For example a nut of property class 8 has a minimum tensile strength of 800 MPa. A bolt or screw of a particular property class should be assembled
with the equivalent or higher property class of nut to ensure that thread stripping does not occur.

| **ratio of shear strength to tensile strength** | This is the shear strength for the material divided by the tensile strength. For steel, typically this ratio is 0.6 (higher for lower tensile steels and slightly lower for high tensile), that is the shear strength of steel is typically 60% of the tensile strength. |
| **shear area of the external thread** | This is the shear area of the external thread in the unstrained condition. It is equal the area of intersection of a cylinder of diameter of the nut minor diameter acting on the mating external thread profile. The calculated area takes into account bell mouthing, if present in the internal thread. The critical dimensions for this area are the length of thread engagement and the maximum internal thread minor diameter. |
| **shear area of the internal thread** | This is the shear area of the internal thread in the unstrained condition. It is equal to the area of intersection of a cylinder equal to the major diameter of the external thread acting on the mating internal thread profile. The critical dimensions for this area are the length of thread engagement and the minimum major diameter of the external thread. |
| **shear strength** | The maximum stress applied by nearly co-linear equal and opposite forces that can be sustained by a material before fracture occurs. If the shear strength is exceeded in the threads of a fastener, shearing (stripping) of the threads occurs. For metric fasteners the units used are megapascals (Mpa or N/mm²), for inch based fasteners the units used are lbf/in². |
| **soft joint** | A 'soft' joint has low stiffness and whose final torque is reached after approximately 720 degrees of bolt rotation (according to ISO 5393). |
| **stress area** | The effective cross sectional area of a thread when subjected to a tensile force. It is based upon a diameter which is the mean of the pitch (or effective) and the minor (or root) diameters of the thread. The use of this diameter stems from the work of E. M. Slaughter in the 1930's. He completed carefully controlled tests using various sizes of standard threads and compared their strength with machined bars made from the same bar of material. He found that this mean diameter give results that agreed with the tensile test results to within about 3%. The error on the minor and pitch diameters was about 15%. Tests completed subsequent to these by other investigators have also shown that the stress diameter is a reasonable approximation to a threads tensile strength. (Reference: 'Tests on Thread Sections Show Exact Strengthening Effect of Threads.' by E. M. Slaughter, Metal Progress, vol 23, March 1933 pp. 18-20) |
| **tapping drill** | The size of the drill that is used to produce the hole to allow an internal thread to be formed. The minor diameter of the internal thread is the diameter of the drill used to create the hole. |
| **thread stripping** | Thread stripping is a shear failure of an internal or external thread that results when the shear strength of the threaded material is exceeded by the applied forces acting on the thread. |
| **tightening factor** | The tightening factor is a measure of the scatter in a bolt's clamp force because of the tightening method used to tighten the fastener. It is defined as the maximum bolt preload divided by the minimum value anticipated for that tightening method. |
| **tolerance class** | A combination of tolerance grade and a fundamental deviation which is given to an internal or external thread. A tolerance class for an internal thread when combined with the tolerance class for an external thread gives the class of fit for the mating threads. |
| **torque control** | Specifying how tight (the amount of preload present) a bolt or threaded fastener should be by specifying a torque that should be applied. |
About Thread Stripping Failures ........................................ 17
Bell Mouthing.......................... 25–27
Bolt Diameter........................ 5, 9–10, 14, 19, 30–33, 37
Boss/Nut Dilation Factor......... 26
Countersink Details ................. 24
Critical Length of Thread
Engagement............................ 27
Description for the Material .... 22
Direct Forces to Fail the Fastener .................................... 26
Effective Length of the Thread
Engagement............................ 25
External Thread Bending Factor ..................................... 26
External Thread Section ............ 20
Factor of Safety - External
Thread..................................... 27
Factor of Safety - Internal Thread .................................. 27
Fastener Camfer ...................... 24
Fastener Camfer Details ........... 24
Fastener Failure Forces allowing
for combined tension-torsion
loading..................................... 27
Internal Thread Bending Factor .................................... 26
Internal Thread Section ........... 20, 22
Internal to External Thread
Strength Ratio.......................... 26
Joint Analysis......................... 4, 18, 28, 37
Length of Thread Engagement...
........................................ 23–24, 25–27
Major Diameter of the External
Thread..................................... 20, 25
Major Diameter of the Internal
Thread..................................... 20
Material Properties for the
External Thread Section .......... 20
Material Properties for the
Internal Thread Section ........... 22
Maximum External Strength ... 21
Minimum External Strength .... 21
Minimum Shear Strength .. 22, 42
Minimum Shear Stress .......... 22
Minimum Tensile Strength11, 21–22
Minor Diameter of the External
Thread..................................... 20
Minor Diameter of the Internal
Thread..................................... 20
Pitch Diameter of the External
Thread..................................... 20
Pitch Diameter of the Internal
Thread..................................... 20
Preload Scatter from Torque and
Frictional Variations Form .. 35
Prevailing Torque Value14, 35, 36
Ratio of Tensile to Shear Stress
........................................ 21
Shear Area of the External
Thread..................................... 24–25
Shear Area of the Internal Thread ................................ 25
Starting a Thread Stripping
Analysis.................................... 18
Tapping Drill Details ............... 24
Thread and Head Friction
Variation.................................. 36
Thread Description.................. 19
Thread Details Section ............ 19
Thread Engagement Details
Section.................................. 23
Thread Pitch........... 9, 19, 23–24, 42
Thread Strength Data Entry Form
........................................ 19
Thread Stripping Analysis4, 17–18, 25, 28, 41–42
Thread Stripping Analysis
Results.................................... 25
Thread Stripping Forces ........... 27
Thread Stripping Friction Form
........................................ 18
Tightening Factor.................... 34, 36
Tightening Tool Accuracy...... 35