GEARCALC

User's manual

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December 13, 2006

Contents

Part I

General

Chapter 1

Elements of the KISSsoft user interface

KISSsoft has been developed for Windows. Regular Windows users will recognise common elements of the interface such as Menus, docking windows, dialog boxes, Tool tips, and status bars. As the development has heeded internationally recognised style guide lines the windows user will quickly become familiar with the operation of KISSsoft.

1.1 Menus, Context Menus and Toolbar

In the main menu using `File` the calculation files can be opened, saved, sent as e-mail, file properties examined and KISSsoft ended.

The project management (see 3) in KISSsoft is operated using the main menu `Project` as well as the project tree (see 1.2.2). Projects can be opened, closed and activated, or files either added or removed from a project, and project properties examined.

The single dock windows (see 1.2) of the user interface can be hidden or shown using the options in the main menu `View`. If the report facility or Help Viewer has been activated then the Action `Input Window` (see 1.3) can be used to return to the data entry tab for the calculation module.

The main menu options `Calculation`, `Report` and `Graphic` are only active if a calculation option is open. The Actions of these menus depend partly on the current calculation module. In the menu `Calculation` the current calculation can be carried out (see 4) and the module-specific settings changed.

In the main menu `Report` there is an Action to build and open a report. The report will always be produced for the current calculation. The Action `Drawing Data` shows the drawing data (see 5.3) of the selected element in the report viewer (see 1.4). Under `Settings` the text size, margins and scope of the reports can be changed. The actions to save, send, and print are only active if a report is open.

The graphic window (see 1.2.8) of a calculation module can be opened and closed in the main menu `Graphic`. The `3D-Export` option accesses a CAD interface (see 6) from KISSsoft. Under `Settings` the CAD-System can be chosen to which the selected element is to be exported.

Under `Extras` there is a licence tool (see 8.1), the configuration tool (see 8.2) as well as the database tool (see 8.3). From the main menu the 'Windows' calculator can be started and the Language (see 2.1) or unit system (see 2.2) changed. General program settings (see 7), such as formats for time and date, can be changed under `Settings`.

KISSsoft help `Help`, as with Windows convention, is the last entry at the end of the menu toolbar and can be used to open and navigate the KISSsoft manual. Under `Info` there is specific details of program version and support of KISSsoft.

In addition to the main menu, KISSsoft uses context menus in many places. Context menus offer access to Actions in a specific aspect or element of the software. Context menus are normally accessed using the right mouse button.

The toolbar allows quicker access to those Actions in the Menu system which are used more frequently. Note that there are Tool Tips which give information on the Actions in the toolbar as well as further explanation in the status bar (see 1.6).

1.2 Dock Window

As well as the menu bar, tool bar and status bar, the dock windows are important elements of the KISSsoft user interface. Dock windows are windows that are displayed either free-floating or arranged to the sides of the application. Dock windows can be arranged one over the other; a tab bar will be added in this case.

A dock window can be released by a double click on the title bar at the top. A window can be shifted by clicking and holding the mouse button while

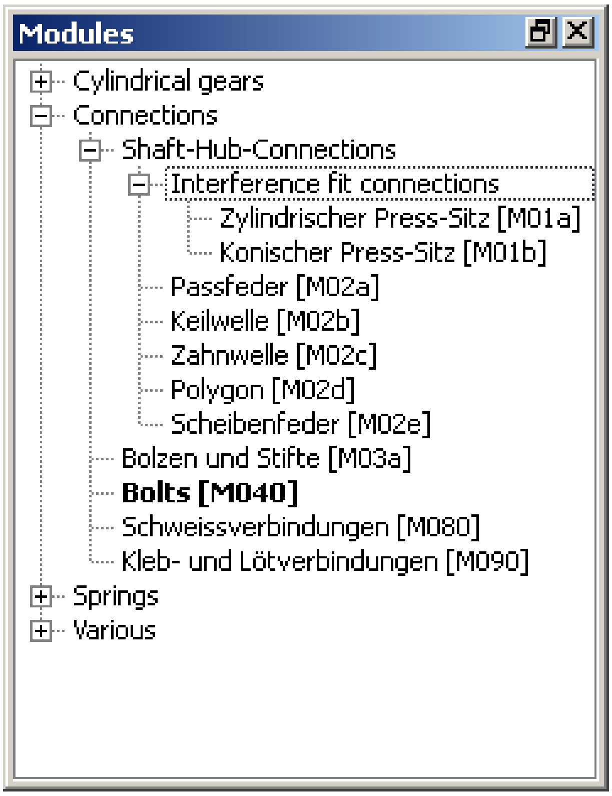


Figure 1.1: Calculation modules of KISSsoft

over the title bar and then moving the mouse. If the windows is close to the main window, the new position for the window will be indicated. Release the mouse button in order to set the window down in this position. The customised arrangement of the windows will be saved in the Registry (see 7.2). Dock windows can be hidden or shown using the menu `View` (see 1.1).

1.2.1 The Module Tree

All of the KISSsoft calculation modules are logically listed in the Module Tree. Calculation modules for which there is no current licence are greyed out. A calculation module can be opened by a double click of the left mouse button. The active calculation module is shown in bold print.

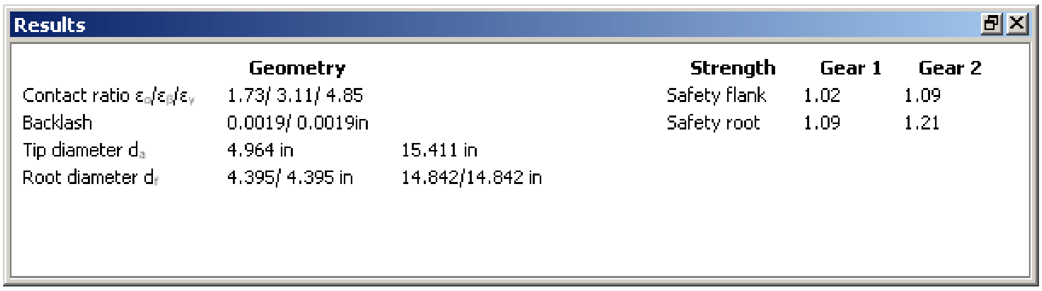


Figure 1.2: The KISSsoft results window

1.2.2 The Project Tree

The Project Tree gives a overview of opened projects and the files contained within, and also shows the active working project in bold print. The operation of the project management (see 3) is carried out from the main menu under `Project` as well as from a context menu (see 1.1).

1.2.3 The Explorer

The directory structure of the Explorer corresponds to the structure in the Windows-Explorer and offers the same functionality. The Explorer will be available from Release 02-2007.

1.2.4 The Results Windows

The KISSsoft Results Window shows the results of the latest calculation.

1.2.5 The Message Window

The Message Window information, warnings and errors occured during the latest calculation (see 4.2). A yellow exclamation mark in the Tab Message signals that messages exist that have not yet been read. Normally all messages will be shown in the Message Window and also in a message box. The display of information and warnings in a message box can be changed using `Extras` ⇒ `Settings` (see 7).

1.2.6 The Information Window

The Information Window shows information opened by the user via an Info-Button of the calculation module (see 1.3.1). Using a context menu (see 1.1) the information can be zoomed and printed.

1.2.7 Contents and Index

Contents and index of the manual are also available as dock window. If a list entry is selected using a double click, the Help Viewer (see 1.5) is opened and the required chapter is show.

1.2.8 Graphics Windows

Any number of graphics windows can be opened simultaneously in KISSsoft which can also be docked to the sides of the software. In this way all of the relevant graphics and diagrams for the calculation are in view at all times. Graphics windows have their own toolbar which can be used to save, print, or zoom the current graphic. Using the Action Lock in the toolbar, the current data in the window is frozen. The window is then prevented from updating by subsequent calculations. The lock capability enables the retention of results and therefore a direct comparison with the current settings of the calculation.

1.3 Input Window

The most significant region of the KISSsoft workspace is occupied by input for the calculation. In this region all the data for a given calculation must be defined. Depending upon the complexity of a calculation, the input window may be divided into several tabs. In most cases a single side is sufficient to carry out the calculation. Every input window uses the same control elements which will be described now in greater detail.

1.3.1 Value Input Field

As a rule, for each value input field there is the variable name, symbol, the editing field, and unit. If the editing field is greyed out then the variable can not be edited and will be determined by the calculation. Behind each input field there can be one or more of the following buttons:

 Setting the Check-Button fixes the entered value

 Setting the Radio-Button you select which of the values in a group will be calculated and which will be fixed

 The Size-Button calculates an appropriate suggestion for the value

 The Convert-Button recalculates the value from depending data

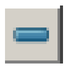
 The Plus-Button can be used to input further data related to the value

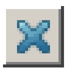
 The Info-Button shows appropriate information in the information window (see 1.2.6)

1.3.2 Tables

In some modules the data is displayed or entered in a table. Double clicking on the end tab to the left of a row selects a complete entry, while the data in a single cell can be edited by double clicking on the cell. Tables often have extra information as Tool Tips (see 1.6). The following buttons are as a rule provided with tables to input data:

 The Add-Button joins a new line to the table

 The Remove-Button removes a selected row from the table

 The Clear-Button deletes all entries in the table

1.3.3 Toggle Units

In KISSsoft the units of the value input field (see 1.3.1) and in the tables (see 1.3.2) can be changed. To do this, click on the unit with the right mouse button. A context menu is opened which contains all possible units for this value. If a different unit to that currently used is selected, then KISSsoft converts the value in the input field to the appropriate value.

In order to toggle the default unit between metric and imperial use the main menu option `Extras` ⇒ `System of Units`.

1.3.4 Enter formulae and angles

In some cases it is practical to define a value in terms of a small mathematical expression. A formula editor is opened by clicking on the edit filed using the right mouse button. A formula can be defined using the four basic operations $+$, $-$, $*$ and $/$. Additionally, all functions that are supported by the report generator can be used (see Tables 5.2). Confirm the formula with the Enter-Key (sometimes called 'Carriage Return'-Key) and the formula will be evaluated. The formula itself will be lost: if the formula editor is again opened the calculated value is seen and not the original formula.

For input fields which show an angle a dialog appears instead of the formula editor to input the value in Degrees, Minutes and Seconds.

1.4 Report Viewer

When a report is generated in KISSsoft a Report Viewer is opened for which entries in the Menu `Report` will be activated and the toolbar of the Report Viewer will be visible. The Report Viewer is a text editor which contains the usual functions to save and print a text file. The reports in KISSsoft can be saved in Rich Text Format (*.rtf), Portable Document Format (*.pdf), Microsoft Word Format (*.doc) and ANSII Text (*.txt).

Further functions of the Report Viewer are Undo/Redo, Copy, Cut and Paste with the usual Shortcuts. The view can be zoomed and the report edited and properties such as text type, size, etc. formatted. To change the default settings of the report, go to the main menu under `Reports` ⇒ `Settings`.

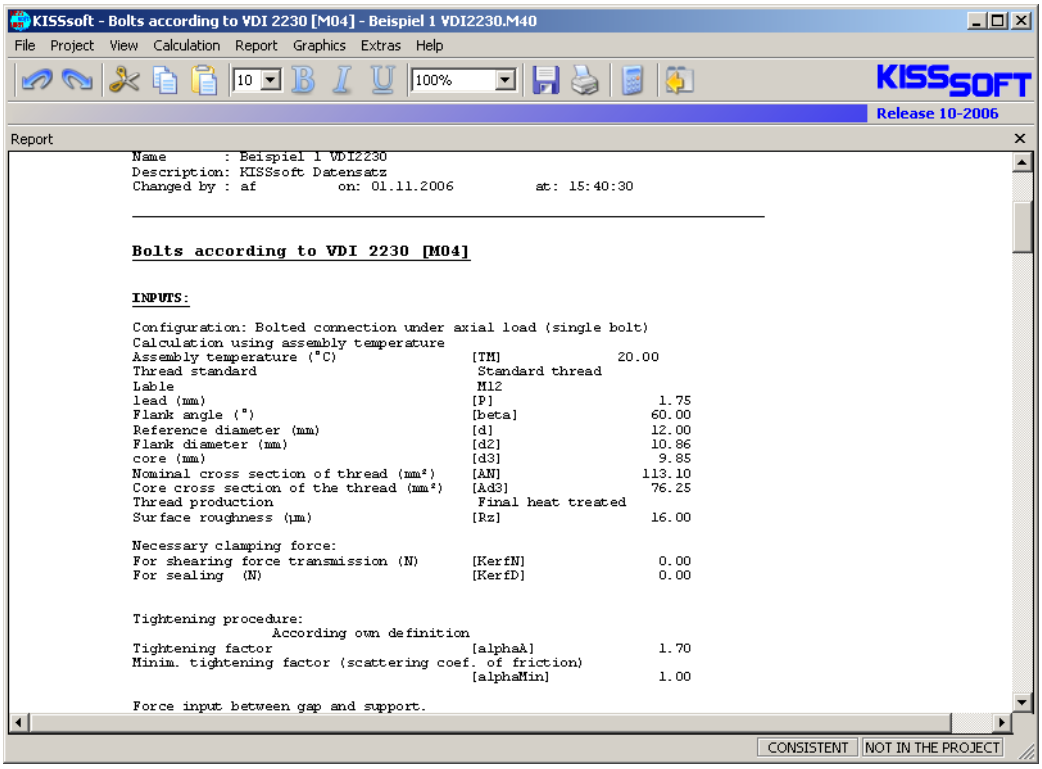


Figure 1.3: The KISSsoft Report Viewer

1.5 Help Viewer

The KISSsoft Manual is shown in HTML Format in the Help Viewer. Open the Manual using the contents or the index (see 1.2.7), or by pressing `F1` to open the Manual at a position showing information relevant to the current state of the program.

1.6 Tool Tips and Status bar

Wherever it is appropriate in KISSsoft Tool Tips have been added which provide concise informative messages describing the program elements. Tool Tips appear automatically if the mouse is moved slowly over the program element.

More detailed information appears in the status bar for all Actions in the menus as soon as the mouse is moved over the menu item. In the right region of the status bar the current status of the calculation is shown. The second region from the right shows CONSISTENT when the results are current and INCONSISTENT when the calculation should be carried out again after one or more data edits (see 4.3). The area Project Members at the far right of the Status bar indicates whether the current calculation file belongs to the current working project (see 3).

Chapter 2

Setting Up KISSsoft

2.1 Language Settings

KISSsoft is available in five languages: German, English, French, Italian and Spanish. The choice of language will change the text in the user interface and the reports. It is also possible to operate KISSsoft in one language and produce reports in another.

2.1.1 Language of the User Interface

Normally KISSsoft starts using the language that is defined in *KISS.ini*-file in section *[SETUP]* in the line *DISPLAYLANGUAGE*. Here the value 0 is for German, 1 English, 2 French, 3 Italian and 4 Spanish.

The language of the user interface can be changed using the program under `Extras` ⇒ `Language`. This setting will be carried out in your personal Registry (see 7.2), not in KISSini (see 7.1).

2.1.2 Language of the Reports

The language of the reports is defined in the *KISS.ini*-file in section *[SETUP]* in the line *LANGUAGE*. Here the value 0 is for German, 1 English, 2 French, 3 Italian and 4 Spanish. A special case here is the value 11 which represents English with imperial units.

The language used for the reports can be changed using the program under `Protokolle` ⇒ `Settings`. This setting will be carried out in your personal Registry (see 7.2), not in KISSini (see 7.1).

2.1.3 Language for messages

Messages are either in the same language as the user interface or as in the reports. The setting for this is in the *KISS.ini*-file in section *[SETUP]* in the line *MESSAGELANG*. 0 represents the language of the messaging = language of report, while 1 represents the language of the messaging = language of user interface.

2.2 System of Units

KISSsoft recognises two unit systems: metric and imperial (US Customary Units). If the value in line *UNITS* in section *[SETUP]* of the *KISS.ini*-file is 0 then KISSsoft uses the metric system, while 1 will indicate that the imperial system should be used.

Using `Extras` ⇒ `System of Units` the unit system can be toggled. This setting will be recorded in your personal Registry (see 7.2), but not in the KISSini (see 7.1).

2.3 User Directory

If a calculation file or report needs to be opened or saved, KISSsoft will suggest your personal user directory as the location. This trait saves time by avoiding searching through the entire directory structure of the computer system. The user directory can be defined in the *KISS.ini*-file in section *[SETUP]* in the line *USERDIR* (see 7.1). By default this is the directory *USR* in the installation directory.

The user directory is ignored if an active working project has been chosen (see 3.3). In this case KISSsoft first suggests the project directory.

2.4 Definition of own Standard Files

If the same or similar calculations are often carried out, the same values must be given in or selected. KISSsoft makes this easier to achieve by means of default files. For each calculation module there exits an internal default set of data. A default file can be stored in which the data can be pre-defined and appears on opening of the associated module or loading of a new file.

To define a default file simply open a calculation module and give in the required data. The Action `File` ⇒ `Save as standard` will store these values in the default files.

Default files can be defined for single modules or for entire projects (see 3.5). If an active project is selected on saving, the default values from this project only will be saved. If there is no current project, the default values are applied generally. On loading a new file, a default file will first be sought in the active project. If it is not available, the general default file, internal preset settings for example, will be used.

2.5 Start Parameter

The call of KISSsoft from the prompt can be done using the following start parameters:

Parameter	Description
INI=`Filename`	The initialisation file *KISS.INI* is loaded from specified location. A file name (including directory) can be given.
START=`Module`	The given calculation module is started. The module identification is, for example, `M040` for the bolt calculation or `Z012` for the spur/helical calculation.
LOAD=`Filename`	The given calculation file will be loaded and the associated calculation module started. If a name is given without a path, the file will be loaded from a pre-defined directory location.

LANGUAGE=`Integer`	KISSsoft starts with given language for user interfaces and reports. (0: German, 1: English, 2: French, 3: Italian, 4: Spanish, 11: English with imperial units)
DEBUG=`Filename`	A file with debug information will be written which can be helpful in the identification of errors. It is recommended to give the file name complete with path in order to easily locate the log file.
`Filename`	The calculation module relating to the file is started and the file loaded. A link from KISSsoft with the corresponding file ending in Windows is also possible (→Start of KISSsoft by double-clicking on a calculation file).

Chapter 3

Project Management

KISSsoft has its own project management system which supports the user in helping to order multiple calculation modules and associated external files. The major part of the management system is the project tree (see 1.2.2). Here can be seen which projects are opened, i.e. active in the workspace, and all information about the files which belong to an individual project.

3.1 Create, open and close projects

A new project is created using `Project` ⇒ `New...`. This opens a Dialog in which the name of the project, the directory, descriptions and comments as well as the directory for the default files (see 2.4) can be entered. The new project is entered in the project tree navigator, and set as the active project.

If an existing project is opened (`Project` ⇒ `Open...`) this will likewise be set in the project tree navigator and marked as the active project.

The currently selected project is closed using the Action `Project` ⇒ `Close`. This Action can also be found in the context menu (see 1.1) of the project tree.

3.2 Add and Remove Files

Files can either be both added and removed using either the project properties (see 3.6) or the context menu (see 1.1). Not only calculation files from KISSsoft but also arbitrary external files can be added to the project.

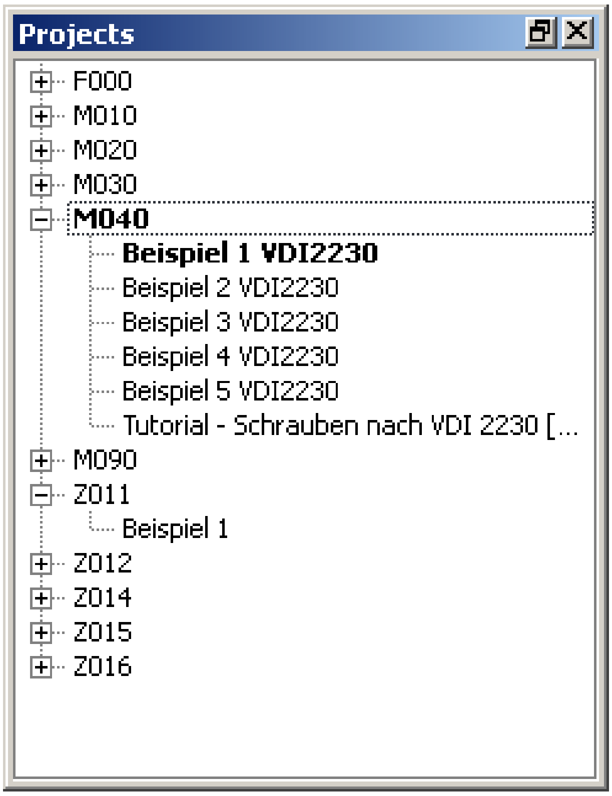


Figure 3.1: The Project Tree of KISSsoft

3.3 The Active Project

The project tree in the navigator shows all open projects, but the active project must not necessarily be defined. If the active project has been defined, it will be displayed in bold text. A project can be activated or deactivated using the Menu `Project` as well as the context menu.

The current calculation file must not necessarily belong to the active project. An indicator in the status bar (see 1.6) shows whether the current calculation file is a part of the active project.

3.4 File Storage

Files that belong to project do not have to be saved in the project directory. Files can therefore also belong to several projects simultaneously. If an active project has been defined, then KISSsoft proposes the active project directory for storage whenever a calculation file or a report is to be opened or saved. If no project is active, then the user directory (see 2.3) will be proposed as the storage point.

3.5 Projects and Default Files

On loading a new file, a default file will first be sought for the active project (see 2.4). If no file exists, then a general default file will be used. In the project properties (see 3.6) it can be seen whether a special default has been defined for a project.

3.6 Project Properties

The project properties for the selected project shown using the Action `Project` ⇒ `Properties`, or with the context menu (see 1.1).

Chapter 4

Calculations in KISSsoft

4.1 Current calculation of a Module

The current calculation of a module is carried out by the Action `Calculation` ⇒ `Run`. Additionally, the toolbar and function key `F5` can be used for quick and easy access to this Action.

A Module can have one or more calculations. In every case, the calculation of the visible tab will be carried out.

4.2 Messages

A calculation sends various types of messages to the input window: information, warnings and errors. Information and warnings should be heeded in order to ensure safe results. If an error occurs, the calculation is automatically stopped.

Normally all messages are written to a message box in the message window (see 1.2.4). The reporting of information and warnings in the message box can be changed (see 7) using `Extras` ⇒ `Settings`.

4.3 Consistency

The status of the calculation is consistent if it has been carried out without an error occurring. As soon as any data has been changed in the input window, the calculation becomes inconsistent i.e. the results of he calculation no longer match the current data set.

The current status of the calculation is indicated in the status bar (see 1.6).

Chapter 5

Results and Reports

5.1 Results of a calculation

If a calculation has been carried out then the results window (see 1.2.4) will show the results. If no results are shown then the calculation has encountered an error. In this case the MessageBox will notify the user of the error. An indicator in the status bar (see 1.6) shows whether the results are consistent i.e. whether the results apply to the current interface data set (see 4.3).

From Release 02-2007 it will be possible for the user to specify a template for the results in a similar way to the definition of report templates (see 5.5).

5.2 Calculation report

The Action `Report` $\Rightarrow$ `Generate` is used to write a report for the calculation. In addition, the toolbar and function keys `F6` provide quick and convenient access to this Action.

A module can have one or more reports. The report relevant to the currently selecvted tab will be generated.

As a rule, a report should only be generated if the calculation is consistent (see 4.3). If this is not the case the report will be written with the current status strongly indicated. This can be useful if it is only required to print a data set.

In generating a report, a RTF-File is produced with the designation of the module as a file name. The file will be stored in *tmp*-Directory, which is defined in the *KISS.ini*-File in section *[SETUP]* row *TMPDIR* (see 7.1).

The report will be shown in the KISSsoft report viewer as standard (see 1.4). From Release 02-2007 other editors, e.g. Windows Word, can also be selected. The report viewer can also be used to change, save, and print the report.

Important: If the user returns to the input window from the report viewer then the report is lost. In order to have the report for a longer period this must be saved with a user defined name!

5.3 Drawing data

Depending on the calculation module, the Action `Report` ⇒ `Drawing data` can be used to generate a report which can be used as a drawing ready for printing.

5.4 Report settings

Under `Report` ⇒ `Settings` the automatic generation of the reports can be adjusted. This Action will be available from Release 02-2007.

5.5 Report templates

KISSsoft has a template for each calculation module in which the form and content are already assigned. These templates can be changed using any Text Editor or with the KISSsoft Report Viewer. In this way every calculation can be formatted and output customised to a specific user requirements.

5.5.1 Storage und Designations

Every report template is stored in directory <*KISSDIR*>. User specified designations have the Structure ***MMMMlsz.rpt*** that summarise the following dimensions:

MMMM	Designation of module	e.g. *M040*
l	historical	allways = *l*
s	Language of report	*s* = *d*: german, *e*: english, *f*: french *i*: italian, *s*:spanish, *a*: english(imperial)
z	historical	always = *0*
.rpt	Designation templates of reports	Reports of calculations end on *.rtf*.

Examples

Bolted joints calculation:	
M040LD0.RPT	german issue
M040USER.RPT	standard issue over interface, becomes file *M040USER.OUT*
Spur gear calculation:	
Z012LD0.RPT	spur gear pair, german issue
Z012USER.RPT	standard issue over interface, becomes file *Z010USER.OUT*
Z10GEAR1.RPT	print out over interface, contains only data of gear 1, becomes file *Z010GEAR1.OUT*
Z10GEAR2.RPT	issued over interface, contains only data of gear 2, becomes file *Z010GEAR2.OUT*
Z011LD0.RPT	Single gear, german issue
Z013LD0.RPT	Rack, german issue
Z014LD0.RPT	Planetary gear, german issue
Z015LD0.RPT	3 gears, german issue
Z016LD0.RPT	4 gears, german issue
English issue:	
*M040L**E**0.RPT*	Thread calculation, **English** issue
American issue:	
*M040L**A**0.RPT*	Thread calculation, **American** issue

5.5.2 Scope of Reports

The Scope, e.g. length, of the report can be defined in the Menu `Report` ⇒ `Settings` on a scale from 1 to 9 where 9 represents the complete data set and 1 for a short summary. In the report template there exists a digit at the beginning of each line between 1 and 9. This digit defines (independently of the previously mentioned setting) whether the line should be read or not.

Example: If a report length of length 5 (middle) has been chosen then all lines of the report template with 1, 2, 3, 4 or 5 at the beginning are read. Lines with 6, 7, 8 and 9 are not read.

5.5.3 Formatting

Report templates as well as completed reports are text files containing Microsoft Windows labels. Please process your reports only in Windows programs to avoid complications with symbols.

The following directions and key words are defined in the report format:

- Text that should be given out
- Comment that should not be given out
- Designations and formats of calculation variables.
- Conditional branches (*IF ELSE END*)
- Repeatitions (*FOR*-Loop)

5.5.3.1 Text formatting

KISSsoft reports are normally generated in RTF-Format. RTF recognises the following text formats:

Description	**Start**	**Ende**
Under Score	<UL>	</UL>
Strichen Through	<STRIKE>	</STRIKE>
Bold	<BF>	</BF>
Kursive	<IT>	</IT>
Tiefgestellt	_	
Font Size	<FONTSIZE=xx>	
Enlarge Font	<INCFONTSIZE>	<DECFONTSIZE>
Reduce Font	<DECFONTSIZE>	<INCFONTSIZE>
Page break	<NEWPAGE>	
Line break	 	
Text Color red	<RED>	<BLACK>
Text Color green	<GREEN>	<BLACK>
Text Color blue	<BLUE>	<BLACK>

Space	<SPACE>	
Figure einfügen	<IMAGE=name,WIDTH=xx, HEIGHT=yy,PARAM=xyz>	

5.5.3.2 Comments

Comment lines begin with //. Comments are ignored when generating a report.

Example

```
// I have changed the report text here on 13.12.95, hm
Tip diameter mm : %10.2f {sheave[0].da}
```

In this case, only the second line will be given out.

5.5.3.3 Calculation variables

No variables can be defined by the user (other than those used for *FOR*-Loop which can be named by the user and whose values can be entered; see Chapter 5.5.3.5).

Replacement character

The file type and format of a variable is given by a Replacement character:

- *%i* stands for a whole number
- *%f* stands for a floating point number
- $\%\nu_1.\nu_2 f$ stands for a formatted floating point number with ν_1 places in total (inc. digits and decimals) and ν_2 decimal places
- *%s* stands for a left-justified character string (Text)
- *%ns* stands for a right-justified character string in a n-symbol long field (n is a whole number).

The data types must match the data types used in the program. The value will be given out exactly in the position where the replacement character stands. The Syntax of the formatting corresponds to the *C/C++*-Standard.

Examples:

- *%10.2f* is a right-justified floating point number with 10 places in the field and 2 decimal places.
- *%i* is a whole, unformatted number.
- *%30s* is a right-justified character string in a 30 symbol long field (if the number 30 was to be removed, the string would be left-justified).

Counter-Example:

- *%8.2i* is an invalid format because a whole number has no decimal places.
- *%10f2* gives a floating point with 10 positions in its field, but the 2 decimal places are ignored and the number 2 is given as text. Floating point numbers are normally given to 6 decimal places.

Variables

The variable which should be actually given must be behind the replacement character in the same line. **The variable is marked in curly brackets.** If these brackets are removed then the variable name will be given as normal text.

Important: The number of the replacement characters must match the number of bracket pairings {}.

Example:

%f {*sheave[0].d*} gives a value for the variable *sheave[0].d* in the position *%f* as a floating point with 6 decimal places.

Basic Calculations – Output of Altered Variables

In the report, variables can be issued differently. They can be multiplied or divided as well as factors can be added or subtracted. This function is also valid in the arguments of the *IF*- or *FOR*-conditions.

Value of the variable multiplied	*%3.2f*	{*Var*2.0*}
Value of the variable divided	*%3.2f*	{*Var/2.0*}
Value of the variable added	*%3.2f*	{*Var+1.0*}
Value of the variable subtracted	*%3.2f*	{*Var-2*}

Similarly, the two functions grad and rad are available for conversion into degree or radiant respectively.

angle %3.2f {*grad(angle)*}

Variables can be combined with each other, like {*sheave[0].d-sheave[1].d*}. More than two variables can be used also. Values with signs have to be put in brackets, e.g. {*ZR[0].NL*(1e-6)*}.

You can use the functions you find in table 5.2.

5.5.3.4 Interrogation of Condition IF ELSE END

The interrogation of condition enables you to issue certain values or text only if a certain condition is fulfilled. The following conditions are supported:

Combination of Characters	**Meaning**
==	equal
>=	larger or equal
<=	smaller or equal
! =	unequal
<	smaller
>	larger

This condition has to be written as follows:

IF (Condition) {*Var*}
  Case 1
ELSE
  Case 2
END;

Example:

IF (%i==0) {*Zst.kXmnFlag*}
  Addendum modified   no
ELSE
  Addendum modified   yes
END;

If variable *Zst.kXmnFlag* is 0, the first text is issued, if it is not 0, the second. Any amount of lines can stand between *IF*, *ELSE* and *END*. Every branch beginning on *IF* has to be closed by *END;* (Please note the semicolon after *END*!). The key word *ELSE* is optional, it reverses the condition. Branches can be interlaced up to level 9.

Example of a Simple Branch

IF (%i==1) {*ZP[0].Fuss.ZFFmeth*}
  Calculation of the tooth form factor after method: B
END;

Function	Meaning
sin(angle)	Sinus of angle **in radians**
cos(angle)	Cosinus of angle **in radians**
tan(angle)	Tangens of angle **in radians**
asin(val)	Arcussinus of val, **returns radians**
acos(val)	Arcuscosinus of val, **returns radians**
atan(val)	Arcustangens of val, **returns radians**
abs(val)	$\|val\|$
exp(val)	e^{val}
log(val)	returns x in $e^x = val$
log10(val)	returns x in $10^x = val$
sqr(val)	val^2
sqrt(val)	returns $\sqrt{val}$
pow(x;y)	returns x^y
sgn(val)	returns $\begin{cases} 1 & \text{if } val > 0 \\ 0 & \text{if } val = 0 \\ -1 & \text{if } val < 0 \end{cases}$
sgn2(val)	returns $\begin{cases} 1 & \text{if } val \geq 0 \\ 0 & \text{if } val < 0 \end{cases}$
grad(angle)	Conversion from radians to degree
rad(angle)	Conversion from degree to radians
mm_in(val)	returns $val/25.4$
celsius_f(val)	returns $\frac{9}{5}val + 32$
min($\nu_1; \ldots, \nu_5$)	returns minimum of $\nu_1, \ldots, \nu_5$
max($\nu_1; \ldots, \nu_5$)	returns maximum of $\nu_1, \ldots, \nu_5$
and($\nu_1; \nu_2$)	binary and function
or($\nu_1; \nu_2$)	binary or function
xor($\nu_1; \nu_2$)	binary exclusive or function
AND($\nu_1; \ldots, \nu_5$)	logical and function
OR($\nu_1; \ldots; \nu_5$)	logical or function
NOT(val)	returns $\begin{cases} 0 & \text{if } val \neq 0 \\ 1 & \text{if } val = 0 \end{cases}$
LESS($\nu_1; \nu_2$)	returns $\begin{cases} 1 & \text{if } \nu_1 < \nu_2 \\ 0 & \text{if } \nu_1 \geq \nu_2 \end{cases}$
EQUAL($\nu_1; \nu_2$)	returns $\begin{cases} 1 & \text{if } \nu_1 = \nu_2 \\ 0 & \text{if } \nu_1 \neq \nu_2 \end{cases}$
GREATER($\nu_1; \nu_2$)	returns $\begin{cases} 1 & \text{if } \nu_1 > \nu_2 \\ 0 & \text{if } \nu_1 \leq \nu_2 \end{cases}$

Table 5.2: Possible functions in for calculations in the report.

If variable *ZP[0].Fuss.ZFFmeth* is 1, a text is issued, otherwise not.

Example of Interlacing Branches

IF (%f<=2.7) {z092k.vp}	
periodical manual lubrication	(Text 1)
ELSE	
IF (%f<12) {z092k.vp}	
Lubrication with droplets (2 to 6 droplets per minute)	(Text 2)
ELSE	
IF (%f<34) {z092k.vp}	
Lubrication with oil bath lubrication	(Text 3)
ELSE	
Lubrication with circulation system lubrication	(Text 4)
END;	
END;	
END;	

If variable *z092k.vp* is equal or smaller than 2.7, text 1 is issued. If not, the program checks whether *z092k.vp* is smaller than 12. If this is true, text 2 is issued. If it is not true, the program checks whether *z092k.vp* is smaller than 34. If this is true, text 3 is issued, otherwise text 4.

5.5.3.5 Loops FOR

In the KISSsoft report generator, *FOR*-loops can be entered, too. Within a *FOR*-loop a counting variable is counted up and down. You can employ up to 10 interlaced constructs.

A loop is constructed as follows:

FOR varname=%i TO %i BY %i DO {Initial value} {Final value} {Step}
// Access to variable with #varname oder $varname
...
END FOR;

- Instead of *%i* or *%f* there can also be fixed numbers (static *FOR*-Loop):

  FOR varname=0 TO 10 BY 1 DO
  ...
  END FOR;

- or intermingled:

  FOR varname=5 TO %i BY -1 DO {*Final value*}
  ...
  END FOR;

- Each *FOR*-Loop has to be paired with a closing *END FOR;* (inc. Semicolon). Each defined counter variable (*varname*) inside the loop can be addressed with *#varname*.

- You can choose negative steps (for example −1), but never can you choose 0. The step width must always be defined.

- The *#varname*-condition can be used for the definition of a variable. For example:

  Number of teeth: *%3.2f* {*ZR[#varname].z*}

- The *$varname*-condition can be used as a character for the issue of the variable value. 0 is A, 1 is B etc. For example:

  FOR quer=0 TO 3 BY 1 DO
  Cross section *$quer-$quer : %8.2f* {*Qu[#quer].sStatic*}
  END FOR;

Example of a Simple Loop

FOR i=0 TO 10 BY 1 DO
phase number *#i $i*
END FOR;

This is issued as:

phase number 0 A
phase number 1 B
phase number 2 C
phase number 3 D
phase number 4 E
phase number 5 F
phase number 6 G
phase number 7 H
phase number 8 I
phase number 9 J
phase number 10 K

Within a loop, you can use any counter variables for all functions, arrays included.

Chapter 6

Interfaces

Available from Release 02-2007.

Chapter 7

Program Settings

Program Settings available from Release 02-2007.

7.1 KISSini

7.2 Registry

Chapter 8

Additional KISSsoft Tools

From Release 02-2007 the following tools are available:

8.1 Licence Tool

8.2 Configuration Tool

8.3 Database Tool and Table Interface

Part II

GEARCALC

Chapter 9

GEARCALC in general

The GEARCALC windows version has several parts. First we have the GEARCALC wizard for the sizing of a new gear pair. Then we have three pages for the analysis of a gear pair. The input data of the wizard is independent of the data for analysis. Only if you accept the results from the wizard the data is transfered from the wizard to the analysis part of the software. All the graphics displayed are for the data for the analysis part of the software.

Usually you will start a new design in the GEARCALC wizard (see chapter 10). The wizard will guide you with several pages to get a design that suits your purpose. After accepting the result you can do further analysis on strength using the `AGMA 2001/2101` page (see chapter 12), you can do a lifetime analysis using a load spectrum on the page `Lifetime` (see chapter 13) or an analysis for scoring or wear on the `AGMA 925` page (see chapter 14).

If you want to modify the geometry afterwards you can either go through the wizard again. This can be done quickly because all the inputs are saved. Or you change the geometry directly on the `AGMA 2001/2101` page, if you know what you want to change.

For the analysis there are different reports for the three pages. So you can get a geometry and strength report, a report for the lifetime calculation and also a report for AGMA 925 calculation.

AGMA 2001 is used if US customary units are selected while AGMA2101 (metric edition of AGMA 2001) is used if metric units are selected. The formula signs in this manual are given as in AGMA 2001 and in brackets behind you will find the symbols as used in the metric system.

AGMA2001 Symbol	Units	AGMA2101 Symbol	Units	Description	See
C	in	a	mm	Operating center distance	12.5
C_e	–	K_{Hme}	–	Mesh alignment correction factor	12.21.4
C_{ma}	–	K_{Hma}	–	Mesh alignment factor	12.21.3
C_{mc}	–	K_{Hmc}	–	Lead correction factor	12.21.1
C_{mf}	–	$K_{H\beta}$	–	Face load distribution factor	12.21
C_{mt}	–	$K_{H\alpha}$	–	Transverse load distribution factor	12.21.6
C_{pm}	–	K_{Hpm}	–	Pinion proportion modifier	12.21.2
C_H	–	Z_W	–	Hardness ratio factor for pitting resistance	
K_m	–	K_H	–	Load distribution factor	12.21
K_o	–	K_o	–	Overload factor	12.20
K_v	–	K_v	–	Dynamic factor	12.22
K_R	–	Y_Z	–	Reliability factor	11.2.6
F	in	b	mm	Net face width	12.7
L	hours	L	hours	Life	12.19
m_G	–	u	–	Gear ratio ≥ 1	
m_p	–	ϵ_α	–	Transverse contact ratio	
m_F	–	ϵ_β	–	Axial contact ratio	
N_P	–	z_1	–	Number of teeth in pinion	12.6
N_G	–	z_2	–	Number of teeth in gear	12.6
n_P	rpm	ω_1	rpm	Pinion speed	12.18
q	–	q	–	Number of contacts per revolution	12.25
P	hp	P	kW	Transmitted power	12.17
P_{nd}	1/in			Normal diametral pitch	12.2
		m_n	mm	Normal module	12.1
S_H	–	S_H	–	Safety factor – pitting	11.3
S_F	–	S_F	–	Safety factor – bending	11.3
s_{ac}	lb/in^2	σ_{HP}	N/mm^2	Allowable contact stress number	
s_{at}	lb/in^2	σ_{FP}	N/mm^2	Allowable bending stress number	
s_c	lb/in^2	σ_H	N/mm^2	Contact stress number	
s_t	lb/in^2	σ_F	N/mm^2	Bending stress number	
ψ	$^\circ$	β	$^\circ$	Helix angle at generating pitch diameter	12.4
ϕ_n	$^\circ$	α_n	$^\circ$	Normal pressure angle	12.3

Chapter 10

GEARCALC Wizard

10.1 GEARCALC/ page 1

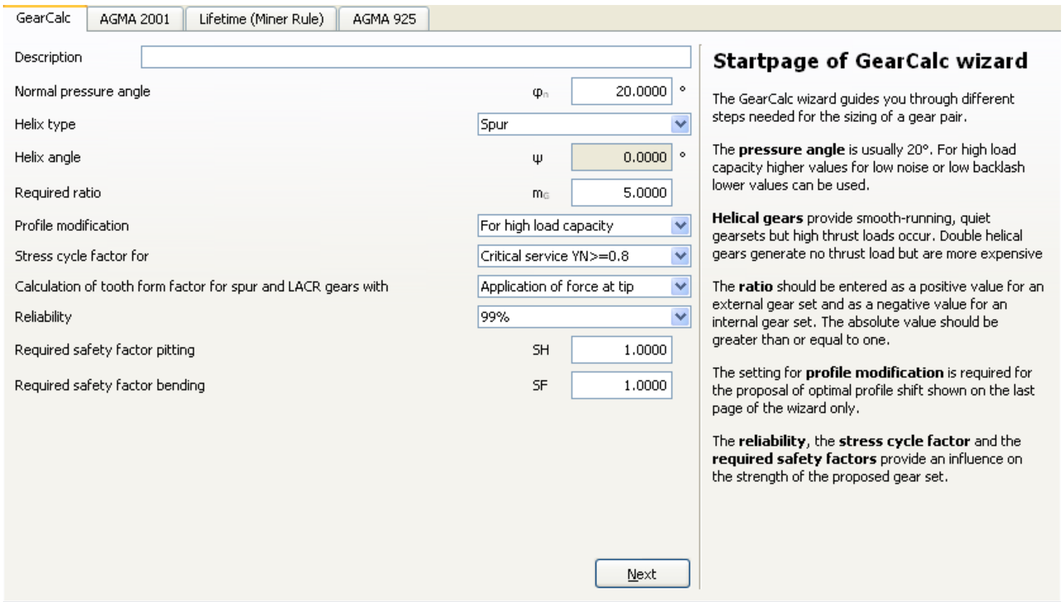


Figure 10.1: GEARCALC - Wizard page 1

10.1.1 Description

The 'Description' field allows the design to be labelled with a code or brief description for reference purposes and documentation.

10.1.2 Normal pressure angle

$\phi_n\{\alpha_n\}$ is the standard or generating pressure angle. For hobbed or rack-generated gears, it is the pressure angle of the tool. For helical gears, ϕ_n is measured on the generating pitch cylinder in the normal plane. ϕ_n is standardized to minimize tool inventory:

ϕ_n (deg.)	Application
14.5	Low Noise
17.5	
20	General Purpose
22.5	
25	High load Capacity

`Low pressure angle:` Requires more pinion teeth ($N_p\{z_1\}$) to avoid undercut. Gives larger topland for same addendum modification coefficient.

`High pressure angle:` Allows fewer pinion teeth without undercut. Gives smaller topland for same addendum modification coefficient.

10.1.3 Helix type

You can design spur, single–helical and double–helical gearsets.

Characteristics for spur gearsets are:

- Teeth are parallel to the gear axis.
- Theoretically, spur gears impose only radial loads on their bearings. In practice, misalignment of the gear mesh may cause small thrust loads.
- Spur gears are noisier than helical gears because they have fewer teeth in contact. Alternating one/two pair tooth contact causes mesh stiffness variation and vibration. Profile modification in the form of tip and root relief improves smoothness.
- Size for size, spur gears have less load capacity than helical gears.
- Although some aircraft gas-turbine spur gears run faster, most spur gears are limited to pitch line velocities less than 10000 fpm.
- Spur gears may be cut by hobbing, shaping or milling and finished by shaving or grinding.

Characteristics for helical gearsets are:

- Teeth are inclined to the gear axis in the form of a helical screw.
- Single helical gears impose both radial and thrust loads on their bearings. Helix angles are usually held to less than 20 degrees to limit thrust loads.
- Single helical gears are quieter than spur gears because they have more teeth in contact with smaller variations in mesh stiffness.
- Size for size, single helical gears have more load capacity than spur gears.
- Many industrial, single helical gearsets run at pitch line velocities up to 20,000 fpm. Special units have reached 40,000 fpm.
- Single helical gears are usually cut by hobbing or shaping and may be finished by shaving or grinding.

Characteristics for double–helical gearset are:

- Double–helical gears share all the advantages of single-helical gears while cancelling internally-generated thrust loads. This means smaller thrust bearings may be used (especially important to reduce power losses in high–speed units). Helix angles up to 35 degrees are typical.
- One member of a double–helical gearset must be free to float axially to share tooth loads between the two helices and to balance the internally generated thrust loads. However, external thrust loads on the floating shaft disturb the balance by unloading one helix while overloading the other helix. All shaft couplings generate large thrust loads if not properly aligned and lubricated. Elastomeric and steel-diaphragm couplings with high axial stiffness may be used to reduce external thrust loads.
- Because the two helices cannot be perfectly matched, the floating member will continualiy shift axially in response to unequal thrust loads. This shifting can cause axial vibration if tooth geometric errors are excessive.
- Double–helical gears allow larger F/d ratios than spur or single–helical gears because the floating member shifts axially and compensates for some of the alignment errors.

- Double–helical gears may be finished by grinding but this requires a large gap between the helices to allow runout of the grinding wheel. Most high–speed, double–helical gearsets are hobbed and shaved.

10.1.4 Helix angle

$\psi\{\beta\}$ is the standard or generating helix angle. The helix angle of a gear varies with the diameter at which it is specified. The standard helix angle is measured on the generating pitch cylinder.

For hobbed gears, the helix angle may be freely chosen because the hobbing machine can be adjusted to cut any helix angle. For pinion-shaped gears, the helix angle must correspond to the helical guides that are available for the gear-shaping machine.

ψ (deg.)	Application
0	spur
10-20	single helical
20-40	double helical

`Low helix angle:` provides low thrust loads but results in fewer teeth in contact (smaller face contact ratio, m_F and higher noise generation. For the full benefit of helical action, $m_F\{\epsilon_\beta\}$ should be at least 2.0. If $m_F < 1.0$ the gear is a low contact ratio (LACR) helical gear and is rated as a spur gear. Maximum bending strength is obtained with approximately 15 degree helix angles.

`High helix angle:` provides smooth-running, quiet gearsets but results in higher thrust loads unless double helical gears are used to cancel internally generated thrust loads.

10.1.5 Required ratio

The gear ratio $m_G\{u\}$of a gearset is defined as a number $|m_G| >= 1.0$ and is the ratio of the tooth numbers of the mating gears.

$m_G = N_G/N_p$

It is also the ratio of the speeds (high/low) of the mating gears:

$m_G =$ -n_p/n_G

For internal gearsets the gears rotate in the same direction instead of opposite directions. As convention the tooth number of the internal gear is set to a negative value. Therefore the ratio for an internal gear set is negative. For an internal gearset the difference of the tooth numbers $|N_G| - N_P$ should not be too small to avoid interference between the tips of pinion and gear teeth.

For the sizings in GEARCALC Wizard the ratio for internal gear sets has to be below $m_G < -2$.

For epicyclic gear trains, the overall gear ratio is:

$$\begin{aligned} m_{Go} &= |Z_G/Z_S| \qquad \text{for a star gear} \\ m_{Go} &= |Z_G/Z_S| + 1 \qquad \text{for a planetary} \end{aligned}$$

where:

Z_G = no. of teeth in internal gear

Z_S = no. of teeth in sun gear

Typical ranges for overall gear ratio:

m_{Go}	Application
1-5	offset gears
3-6	star gear epicyclic
4-7	planetary epicyclic

For gear ratios larger than those shown in the table, it is generally more economical to use multiple stages of gearing rather than a single gearset.

`Star gear Epicyclic Ratios:`

planet/sun gear ratio for m_{Go} >= 3:

m_G = (m_{Go}-1)/2

planet/sun gear ratio for m_{Go} < 3:

m_G = 2/(m_{Go}-1) planet is the pinion

internal/planet gear ratio:

m_G = (2*m_{Go})/(m_{Go}-1)

Note: star gears cannot have m_{Go} = 1. A reasonable minimum ratio is m_{Go} = 1.2.

`Planetary Epicyclic Ratios:`

planet/sun gear ratio for m_{Go} >= 4:

$m_G = (m_{Go}$-2)/2 sun is the pinion

planet/sun gear ratio for $m_{Go} < 4$:

$m_G = 2/(m_{Go}$-2) planet is the pinion

internal/planet gear ratio:

$m_G = (2^*(m_{Go}$-1))/(m_{Go}-2)

Note: planetary gears cannot have $m_{Go} = 2$. A reasonable minimum ratio is $m_{Go} = 2.2$.

10.1.6 Profile modification

You can make corrections to the theoretical involute (profile modification). The type of profile modification has an impact on the calculation of the scoring safety. The Distribution factor (or Force Distribution factor) XGam is calculated differently depending on the type of profile modification. There is a significant difference between cases with and without profile correction. The difference between profile correction 'for high load capacity' gears and thise 'for smooth meshing' however is not so important. The calculation procedure requires that the C_a (of the profile correction) is sized according to the applied forces, but does not indicate an exact value.

10.1.7 Stress cycle factor

The stress cycle factor can be determined dependent upon the expected application. The choice of critical service ($Y_N \geq 0.8$) or general applications ($Y_N \geq 0.9$) can be set from the drop-down list.

10.1.8 Calculation of tooth form factor

The point of force to be assumed by the calculation of tooth form factor for spur and LACR gears is defined here. The drop down list allows the definition of force applied at tip or at the high point of single tooth contact (HPSTC).

10.1.9 Reliability and The Reliability Factor

The reliability factor K_R accounts for the statistical distribution of fatigue failures found in materials testing. The required design life and reliability varies considerably with the gear application. Some gears are expendable, and a high risk of failure and a short design life are acceptable. Other applications such as marine gears or gears for power generation, require high reliability and very long life. Special cases such as manned space vehicles demand very high reliability combined with a short design life.

Reliability R	Application	Failure Frequency
0.9	Expendable gears. Motor vehicles.	1 in 10
0.99	Usual gear design	1 in 100
0.999	Critical gears. Aerospace vehicles	1 in 1000
0.9999	Seldom used.	1 in 10000

10.1.10 Required safety factors

An extra margin of safety can be specified by assigning $S_F > 1.0$ for the bending stress and $S_H > 1.0$ for the pitting. Since pitting fatigue is slowly progressive, and pitted gear teeth usually generate noise which warns the gearbox operator that a problem exists, pitting failures are not usually catastrophic. Bending fatigue frequently occurs without warning and the resulting damage may be catastrophic.

The safety factors should be chosen with regard to the uncertainties in the load and material data and the consequences of a failure. Small safety factors can be used where the loads and material data are known with certainty and there are small economic risks and no risk to human life. However, if the loads and material data are not known with certainty and there are large economic risks or risks to human life, larger safety factors should be used. The bending fatigue safety factor is frequently chosen greater than the pitting safety factor ($S_F > S_H$) since bending fatigue may be catastrophic. However, S_F should not be too large because it leads to coarse-pitch teeth which may be noisy and prone to scoring failures.

Choosing a safety factor is a design decision that is the engineer's responsibility. It must be carefully selected accounting for the uncertainties in:

- `External Loads`
  - Static or dynamic?

  - Load variation (time history)
  - Transient overloads
  - Loads from test data or service records?
- `Component Geometry`
  - Dimensional tolerances
  - Variation in fabrication
  - Surface finish, notches, stress concentrations
  - Damage during assembly (or incorrect assembly)
  - Quality assurance/inspection techniques
- `Material Properties`
  - Handbook values or test data for strengths?
  - Material procurement control
  - Heat treatment control
  - Quality assurance/inspection techniques
- `Design Analysis`
  - Is gear rating verified with computer programs AGMA2001 and Scoring? Will gears be tested before going into service?
- `Service Conditions`
  - Environment: thermal, chemical, etc.
  - Installation procedures
  - Operation procedures
  - Maintenance procedures

Consider the need to conserve material, weight, space or costs. Most importantly, consider:

- `Consequences of Failure`
  - Nature of failure modes
  - Risk to human life
  - Economic costs
  - Environmental impact

10.2 GEARCALC/ page 2

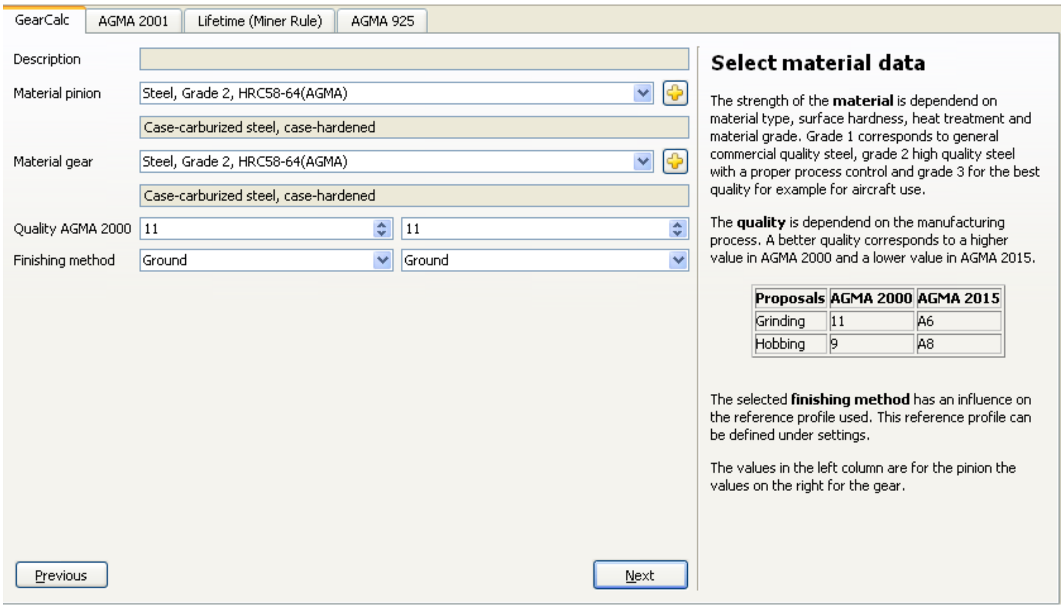


Figure 10.2: GEARCALC - Wizard page 2

10.2.1 Material selection

The material of the gears can be selected from the material database. The strength is dependend of material type, treatment and quality.

10.2.1.1 Material treatment

There are different possibilities for heat treatment: through hardened, nitrided, induction hardened and case hardened materials:

- **Through hardened:** annealed, normalized or quenched and tempered. Carbon content ranges from 0.30 to 0.50%. Alloy content ranges from plain carbon steels (e.g. MSI 1040) for tiny gears, to Cr-Ni-Mo alloys (e.g.AISI 4340) for large gears. The best metallurgical properties are obtained with quenched and tempered steels. Hardness ranges from HB = 180 for lightly-loaded gearsets, to the limit of machinability (approximateby HB = 360) for highly-loaded gears.

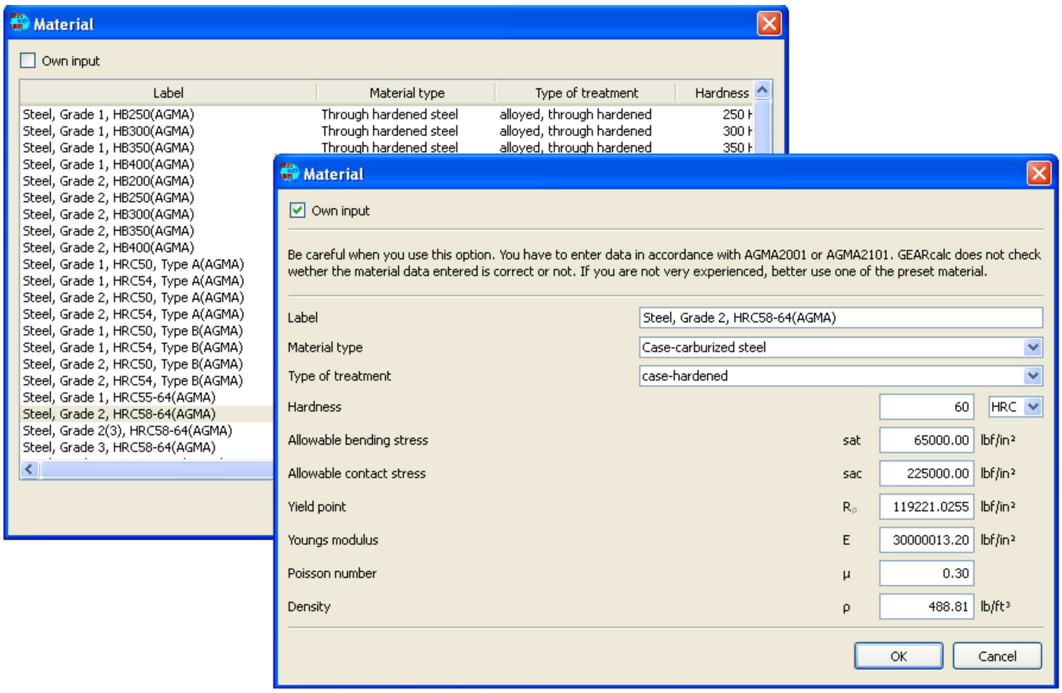


Figure 10.3: GEARCALC - Material

Good tooth accuracy (typically $Q = 10$ acc. AGMA2000) can be obtained by hobbing the teeth after heat treatment, eliminating heat treatment distortion from the generated tooth forms. Hardenability must be adequate to obtain the required hardness at the root diameter.

- **Nitrided gears** are quenched and tempered to obtain the desired core properties, then the teeth are cut and finished, followed by the nitriding process. fle gears are placed in an ammonia gas atmosphere where nitrogen is absorbed into the surface bayers of the gear teeth and forms hard fron nitrides. Because nitriding is performed at the relatively low, temperature of 950-1050 °F, and there is no quench, the distortion due to heat treatment is small. Surface hardness ranges from HB = 432 for alloys such as AISI 4340 to HB = 654 for Nitralloy 135M and 2.5% chrome alloys. The practical limit on case depth is about 0.025 in, which limits the application of nitriding to pitches finer than approximately $P_{nd} = 8$.

- **Induction hardened** gear teeth are heated by electromagnetic induction from a coil or inductor and are immediately quenched. Because only the surface layers of the gear teeth are hardened, heat treat distortion is minimized. Very tight controls of every step of the process are necessary for satisfactory results, and it is best for high-volume production where the process can be optimized. Several gears from each production run must be destructively inspected for case depth to ensure that the induction hardening is properly controlled. Carbon content of induction hardened gears is usually 0.40 or 0.50%. Plain carbon steels (e.g. AISI 1050) may be used for small gears, while alloys such as AISI 4350 may be used for large gears.
  Note: ANSI/AGMA 2001-D04, Figure 18, allows interpolation of Y_N for through-hardened gears. However, no guidance is given for flame/induction hardened gears. In lieu of guidance from AGMA, for flame/induction hardened gears the same Y_N curve for carburized and flame/induction hardened gears.

- **Carburized gears** are first cut, then heated in a carbon atmosphere (usually gas carburizing) which causes carbon to diffuse into the surface layers of the gear teeth. The gears are either quenched from the carburizing temperature or cooled, reheated and quenched later. Most gears are tempered at 300-400 °F after carburizing and quenching. Carbon content of carburizing steels range from 0.15 to 0.25%. Low alloy steels (e.g. AISI 8620) are used for small gears and moderate loads while high alloy steels (e.g. AISI 4820) are used for large gears and high loads. Minimum surface hardness ranges from HB = 615 to HB = 654. Because carburized gears are subjected to a drastic quench from a high temperature the distortion is large, and grinding is usually required to obtain acceptable accuracy.

10.2.1.2 Material quality

Material quality strongly influences pitting resistance and bending strength. For high quality material, the following metallurgical variables must be carefully controlled:

- Chemical coposition
- Hardenability
- Toughness
- Surface and core hardness
- Surface and core microstructure
- Cleanliness/inclusions
- Surface defects (flanks and root flllets)
- Grain size and structure
- Residual stress pattern
- Internal defects, seams or voids
- Microcracks
- Carbide network
- Retained austenite
- Intergranular oxidation
- Decarburization

There are three basic grades of material:

Grade 1: Commercial quality typical of that obtained from experienced gear manufacturers doing good work. Modest level of control of the metallurgical variables.

Grade 2: High quality typical of aircraft quality steel with cleanliness certified per AMS 2301 or ASTM A534. Close control of critical metallurgical variables.

Grade 3: Premium quality typical of premium aircraft quality with cleanliness certified per AMS 2300 or .ASTM A535. Absolute control of all metallurgical variables.

10.2.1.3 Own input of material data

Using the plus button  next to the material list the material values can be entered directly by the user. You have to be careful choosing the values since they are not checked by the software. Important for the calculation are the allowable stress numbers $s_{ac}\{\sigma_{Hlim}\}$ and $s_{at}\{\sigma_{Flim}\}$. The youngs module is needed for the hertzian stress and the yield point for the static strength. The hardness value is only used for documentation.

10.2.2 Quality according to AGMA 2000/AGMA 2015

The quality for both the pinion and gear can be defined independently. The actual quality achieved is dependent upon the manufacturing process used.

10.2.3 Finishing method

1. Finish cut:
   Many gears are not shaved or ground. Accuracy and surface roughness of as-cut gear teeth depend on the condition of the cutting machine, the accuracy and rigidity of the fixtures which hold the gear, the quality of the gear blank, and the quality of the cutter. For gears that are cut only, the most accurate are through hardened gears whose teeth are cut after the gear blanks are heat treated. Carburized gears are cut and then heat treated and usually must be finished by grinding to remove the distortion due to the heat treatment. Nitrided and induction hardened gears usually are not ground because they have low distortion due to heat treatment. The shallow case depth of nitrided gears makes

grinding risky. Sometimes nitrided gears are shaved or ground before nitriding to obtain good surface finish and accuracy.

2. Shaving:
   A finishing process which uses a pinion-shaped shaving cutter with hardened steel helical teeth that have radial gashes which act as cutting edges. The shaving cutter is run in tight mesh with the gear to be shaved with the axes of cutter and gear skewed. Axial sliding removes small amounts of material. Shaving is frequently used as a final finishing operation on through hardened gears, and sometimes as a finishing operation before nitriding. It can be applied to both external and internal, spur and helical gears. Shaving can produce profile modification (e.g. tip and root relief) and lengthwise (helix) modification. Shaved gears are usually cut with a protuberance cutter followed by shaving of the tooth flanks only.

3. Grinding:
   Gear teeth may be ground by either the form-grinding or generating-grinding method. Either method is capable of producing the highest accuracies of any finishing method. Both spur and helical gears can be ground. Most grinders finish only external gears; some can grind internal gears. Some gear grinders can produce profile and helix modification. Grinding is used where high accuracy is required and most often used for finishing carburized gears to remove the distortions due to heat treatment. The strongest gear teeth are cut with a protuberance cutter and ground on the tooth flanks only, leaving the root fillets unground.

Comparison of tooth finishing methods:

Gear Tooth Finishing Method	Accuracy Quality No. Q_n	Surface Roughness μin (rms)	Brinell Hardness Limit HB
Milling	< 6	64-125	360
Shaping	6-10	32-125	360
Hobbing	7-11	30-80	360
Shaving	10-13	10-40	360
Grinding	11-15	10-40	None

The finishing method has an influence on the selected tool addendum according to the `GEARCALC` setting (see 11.1.5).

10.3 GEARCALC/ page 3

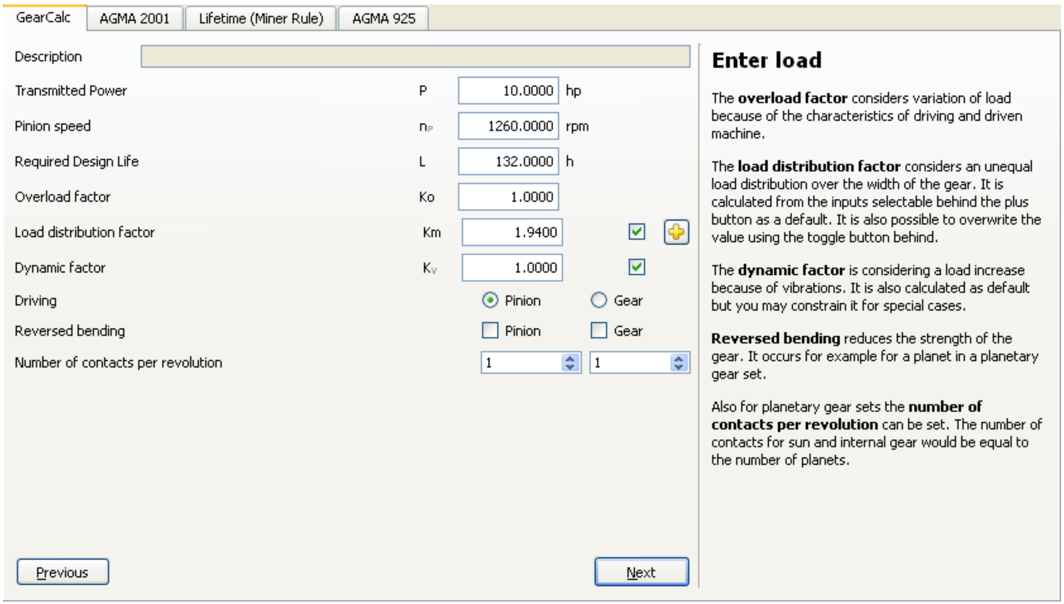


Figure 10.4: GEARCALC - Wizard page 3

10.3.1 Transmitted power

P is the power transmitted per gear mesh. For multiple power paths load-sharing must be considered:

`Branched offsets`: If the pinion meshes with two or more gears (or the gear meshes with two or more pinions), use the power of the more highly-loaded branch.

`Epicyclic Gearboxes`: The degree of load sharing depends on the number of planets, accuracy of the gears and mountings, provisions for self-aligning, and compliance of the gears and mountings.

10.3.2 Pinion speed

The pinion is defined as the smaller of a pair of gears. For planetary sun/planet gearsets, the sun is the pinion for $m_{Go} >= 4$ and the planet is the pinion for $m_{Go} < 4$. For star sun/planet gearsets, the sun is the pinion for $m_{Go} >= 3$ and the planet is the pinion for $m_{Go} < 3$. For planet/internal gearsets, the planet is always the pinion since it is smaller than the internal gear. Epicyclic gearsets are analyzed using relative speeds. The pinion and gear speeds are in proportion to the gear ratio:

$m_G = -n_P/n_G =$ pinion speed/gear speed

10.3.3 Required Design life

A gearset's design life L is determined by the particular application. Some gears such as hand tools are considered expendable, and a short life is acceptable, while others such as marine gears must be designed for long life. Some applications have variable loads where the maximum loads occur for only a fraction of the total duty cycle. In these cases, the maximum load usually does the most fatigue damage, and the gearset can be designed for the number of hours at which the maximum load occurs.

Typical design lives:		
Application	No. Cycles	Design Life, L(hr)
Vehicle	$10^7 - 10^8$	3000
Aerospace	$10^6 - 10^9$	4000
Industrial	10^{10}	50000
Marine	10^{10}	150000
Petrochemical	$10^{10} - 10^{11}$	200000

The number of load cycles per gear is calculated from the required life (L), the speed (n) and the number of contacts per revolution (q):

$$N = 60 \cdot L \cdot n \cdot q$$

10.3.4 Overload factor

The overload factor K_o makes allowance for the externally applied loads which are in excess of the nominal tangential load, W_t. Overload factors can only be established after considerable field experience is gained in a particular application. For an overload factor of unity, this rating mehtod includes the capacity to sustain a limited number of up to 200% momentary overload cycles (typically less than four starts 8 hours, with a peak not exceeding one second duration). Higher or more frequent momentary overloads shall be considered separately. In determining the overload factor, consideration should be given to the fact that many prime movers and driven equipment, individually or in combination, develop momentarypeak torques appreciably greater than those determined by the nominal ratings of either the prime mover or the driven equipment. There are many possible sources of overload which should be considered. Some of these are: system vibrations, acceleration torques, overspeeds, varitions in system operation, split path load sharing among multiple prime movers, and changes in process load conditions.

Examples of operating characteristics of driving machines:

- Uniform – Electric motor, steam turbine, gas turbine.
- Light shock – Multi-cylinder internal combustion engine with many cylinders.
- Medium shock – Multi-cylinder internal combustion engine with few cylinders.
- Heavy shock – Single-cylinder internal combustion engine.

Examples of operating characteristics of driving machines:

- Uniform – Generator, centrifugal compressor, pure liquid mixer.
- Light shock – Lobe-type blower, variable density liquid mixer.
- Medium shock – Machine tool main drive, multi-cylinder compressor or pump, liquid + solid mixer.
- Heavy shock – Ore crusher, rolling mill, power shovel, single-cylinder compressor or pump, punch press.

Operating Characteristics of Driving Machine	Operating Characteristics of Driven Machine			
	uniform	light shock	medium shock	heavy shock
uniform	1.00	1.25	1.50	1.75
light shock	1.10	1.35	1.60	1.85
medium shock	1.25	1.50	1.75	2.00
heavy shock	1.50	1.75	2.00	2.25

10.3.5 Load distribution factor

The factor allows for the variation in contact brought about by differing manufacturing processes, operating conditions and mounting error on assembly. The load distribution factor K_m can either be defined directly or calculated by the empirical method of AGMA 2001/2101. This empirical method is recommended for normal, relatively stiff gear designs which meet the following requirements:

1. Net face width to pinion pitch diameter ratios less than or equal to 2.0. (For double helical gears the gap is not included in the face width).

2. The gear elements are mounted between bearings, i.e., not overhung.

3. Face widths up to 40 inches.

4. Tooth contact extends across the full face width of the narrowest member when loaded.

The input values used for the empirical method for the load distribution factor calculation can be found by pressing the plus button  beside the field:

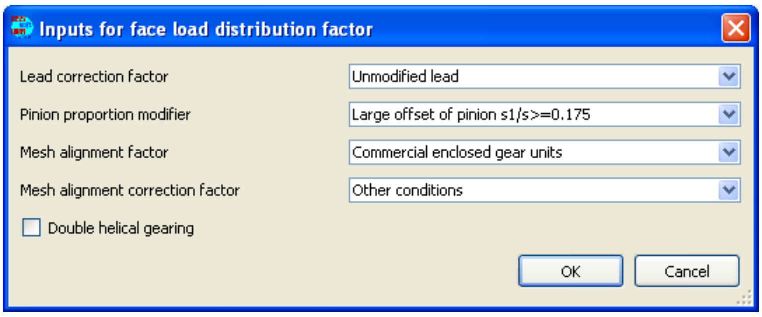


Figure 10.5: GEARCALC - Face load distribution factor

`Lead Correction Factor)`

The nominal setting 'Unmodified lead' should be used when the machining quality is not known. An option 'Lead properly modified by crowning or lead correction' exists to define a well defined lead modification possible using high quality grinding machines.

Lead modification (helix correction) is the tailoring of the lengthwise shape of the gear teeth to compensate for the deflection of the gear teeth due to load, thermal or other effects. Certain gear grinding machines have the capability to grind the helical lead to almost any specified curve. Many high-speed gears are through-hardened, hobbed and shaved. Usually the gear member is shaved to improve the surface finish, profiles and spacing, but the helix lead is not changed significantly. The pinion and gear are then installed in the housing and a contact pattern is obtained by rolling the gears together under a light load with marking compound applied to the gear teeth. Based on the contact pattern obtained from this test, the pinion is shaved to match the lead of the gear. The process is repeated until the desired no-load contact pattern is obtained.

`Pinion proportion modifier )`

This setting allows consideration of the degree of alignment change as the pinion is offset under a defelction of the bearings. The C_{pm} value alters the pinion proportion factor, C_{pf}, based on the location of the pinion relative to its bearing center line.

`Mesh alignment factor )`

The mesh alignment factor C_{ma} accounts for the misalignment of the axes of rotation of the pitch cylinders of the mating gear elements from all causes other than elastic deformation. The factor is dependend on the face width and the follwing options:

- Open – This type of gearing is used in such applications as rotary grinding mills, kilns, dryers, lifting hoists and winches. These gears are frequently of low accuracy because their large size limits the practicable manufacturing methods. The gear shafts are usually supported by separate pedestal bearings with the gears covered by sheet metal shields. The gear mesh alignnent is dependent on the skill and care exercised in the mounting and alignment of the shaft bearings.

- Commercial – This classification pertains to low speed, enclosed gear units, which employ gears that are through-hardened and hobbed or shaped, or hobbed or shaped and surface hardened and which are not subsequently finished by shaving or grinding.

- Precision – This classification pertains to low or high speed, enclosed gear units, which employ gears which are finished by shaving or grinding.

- Extra Precision – This classification pertains to high speed, enclosed gear units, which employ gears which are finished by grinding to high levels of accuracy. The lead and profiles of the gear teeth are usually modified to compensate for load deflections and to improve the meshing characteristics.

`Mesh alignment correction factor`

This selection can be used to account for improved corrective action after manufacturing for a better contact condition.

Some gearsets are adjusted to compensate for the no-load shaft alignment error by means of adjustable bearings and/or by re-working the bearings or their housings to improve the alignment of the gear mesh. Lapping is a finishing process used by some gear manufacturers to make small corrections in the gear tooth accuracy and gear mesh alignment. Lapping is done by either running the gear in mesh with a gear-shaped lapping tool or by running the two mating gears together while an abrasive lapping compound is added to the gear mesh to promote removal of the high points of the gear tooth working surface.

`Double Helical`
For double-helical gears, the mesh alignment factor is calculated based on one helix (one half of the net face width).

NOTES:

It usually is not possible to obtain a perfectly uniform distribution of load across the entire face width of an industrial gearset. Misalignment between the mating gear teeth causes the load and stress distribution to be non-uniform along the tooth length. The load distribution factor is used to account for the effects of the non-uniform loading. It is defined as the ratio of the maximum load intensity along the face width to the nominal load intensity, i.e.,

$K_m = C_m =$ Maximum Load Intensity/(W_t/F)

Variations in the load distribution can be influenced by:

`Design Factors`
Ratio of face width to pinion diameter

Bearing arrangement and spacing
Internal bearing clearance
Geometry and symmetry of gear blanks
Material hardness of gear teeth

`Manufacturing Accuracy`
Gear housing machining errors (shaft axes not parallel)
Tooth errors (lead, profile, spacing & runout)
Gear blank and shaft errors (runout, unbalance)
Eccentricity between bearing bores and outside diameter

`Elastic Deflection of:`
Gear tooth (bending)
Gear tooth (hertzian)
Pinion shaft (bending and torsional)
Bearings (oil film or rolling elements)
Housing

`Thermal Distortion of:`
Gear teeth, gear blank, shafts, and housing

`Centrifugal Effects`
Centrifugal forces may cause misalignment for high-speed gears

`External Effects`
Misalignment with coupled machines
Gear tipping from external loads on shafts
External thrust from shaft couplings

10.3.6 Dynamic factor

The dynamic factor accounts for internally generated gear tooth loads which are induced by non-uniform meshing action (transmission error) of gear teeth. If the actual dynamic tooth loads are known from a comprehensive dynamic analysis, or are determined experimentally, the dynamic factor may be calculated from:

$$K_v = (W_d + W_t)/W_t$$

where W_t = Nominal transmitted tangential load and
W_d = Incremental dynamic tooth load due to the dynamic response of the gear pair to the transmission error excitation, not including the transmitted tangential loads.

If the factor is calculated according AGMA, the Transmission Accuracy Grade A_{nu} is used. A_{nu} is calculated following formula (21) in AGMA2001, page 15. Therefore A_{nu} is not always identical but close to the gear quality.

CAUTION: This factor has been redefined as the reciprocal of that used in previous AGMA standards. It is now greater than 1.0. In earlier AGMA standards it was less than 1.0.

10.3.7 Driving

GEARCALC needs to know whether pinion or gear is driving when determining the optimum addenda modification for maximum scoring resistance. The driving member influences load-sharing between successive pairs of teeth and load distribution along the path of contact. This in turn influences the flash temperature and scoring resistance.

10.3.8 Reversed bending

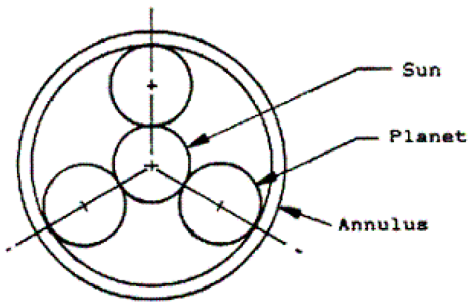
Usually a pair of gears rotate in one direction without torque reversals and the gear teeth are loaded on one side only. For this case, the gear teeth are subjected to one-way bending or uni-directional loading.

Some gears are loaded on both sides of the teeth and are subjected to reverse bending. Examples are:

- idler gears
- planet gears (planetary or star gear systems)
- gearsets which have fully reversed torque loads

10.3.9 Number of contacts per revolution

For a single pinion in mesh with a single gear, each member has one contact per revolution. Some gears have more than one cycle of load contact per revolution. An epicyclic gearset (planetary or star gear) is shown below:

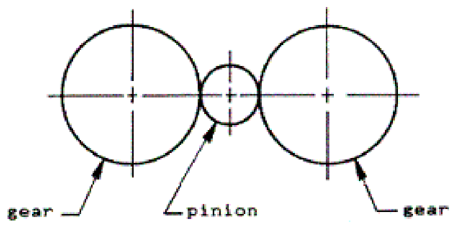


Sun The gear has Q contacts/rev, where Q = number of planets. For the example shown, the sun gear has 3 contacts/rev.

Planet The planet gear has 1 contact/rev because the loads from the sun gear and ring gear occur on opposite sides of the planet gear teeth. The reverse bending that occurs on the planet gear teeth is accounted for with the flag for reversed bending (see chapter 10.3.8).

Annulus (planetary gear train) The internal gear has Q contacts per revolution, where Q = number of planets. Although the internal gear in a planetary gearset is fixed, it is analyzed as if it were rotating at the planet carrier speed.

Annulus (star gear train) – the internal gear has Q contacts per revolution of the internal gear where Q = number of planets. An example of a split-power-train (branched) gearset is shown below:



In this example, if the pinion is the driver or is driven, it has 2 contacts/rev. If the pinion is an idler, it has 1 contact per revolution and reversed bending. The mating gears each have 1 contact/rev.

10.4 GEARCALC/ page 4

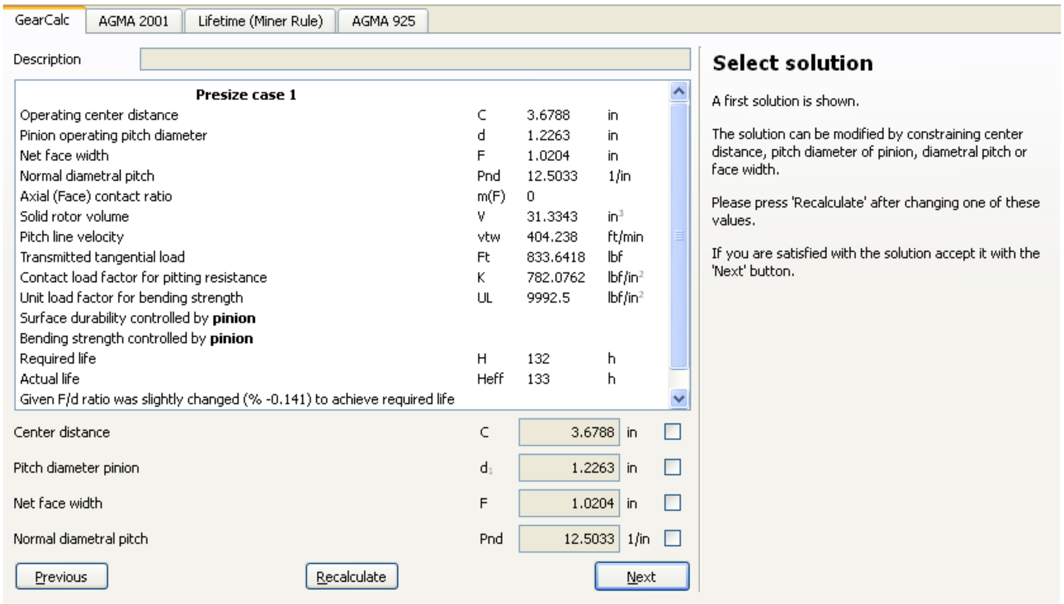


Figure 10.6: GEARCALC - Wizard page 4

10.4.1 Center distance

The standard center distance $C\{a\}$ is dependent upon ratio, tooth pitch, and pressure angle. Standard Center Distance:

A pair of gears may operate on modified or standard center distance. The standard center distance is given by:

$$C_{STD} = (N_G + N_P)/(2 * P_{nd} \cdot \cos \psi_s)$$

For gears that operate on standard centers:

C $= C_{STD}$

Modified Center Distance:

For gears that operate on modified centers, the center distance modification is:

ΔC=C-C_{STD}

10.4.2 Pitch diameter pinion

The value for the pitch diameter of the pinion is normally calculated and entered here. The value can be directly entered by checking the box by the side of the field.

10.4.3 Net face width

The net contacting face width $F\{b\}$ excludes any face width that is non-contacting because of chamfers or radii at the ends of the teeth. For double-helical gears the net face width equals the total face width minus the gap between the helices.

10.4.4 Normal diametral pitch

The normal diametral pitch is shown if US customary units are selected as a default (see 1.3.3). For metric units also the normal module can be shown instead.

The normal diametral pitch defines the size of a tooth. It is π divided by the normal pitch $P_{nd} = \pi/p$. So the tooth thickness increases with a decreasing normal diametral pitch. The value can be directly entered by checking the box by the side of the field. So you have the possibility to select a standard value.

10.4.5 Normal module

The normal module m_n is only shown if metric units are selected (see 1.3.3). For US customary units you see the normal diametral pitch instead.

The normal diametral pitch defines the size of a tooth. It is the normal pitch divided by π $m_n = p/\pi$. So the tooth thickness increases with an increasing module. The value can be directly entered by checking the box by the side of the field. So you have the possibility to select a standard value (with are normally given in millimeters).

10.5 GEARCALC/ page 5

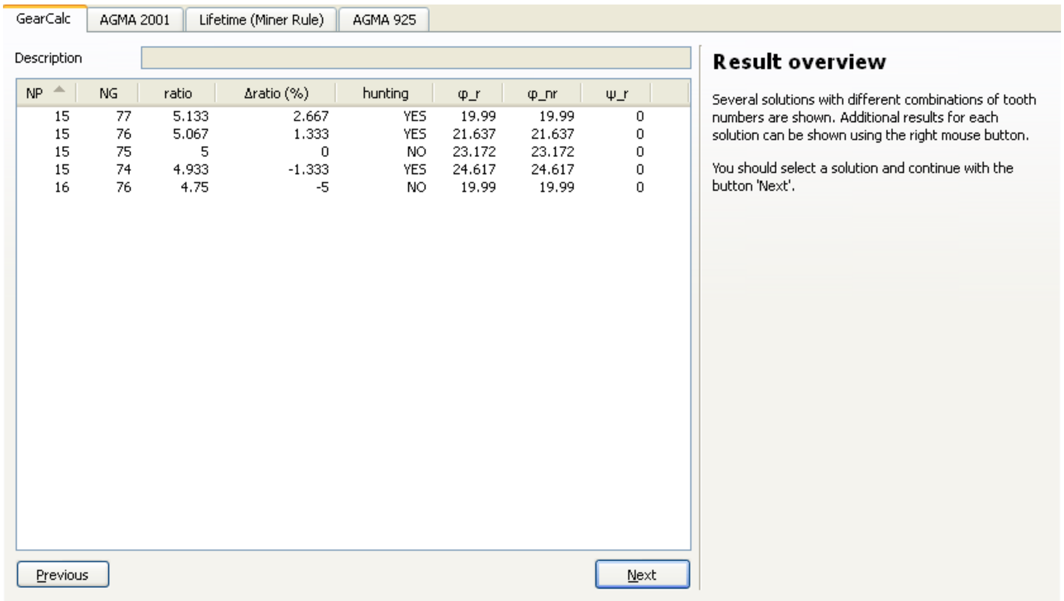


Figure 10.7: GEARCALC - Wizard page 5

10.5.1 Result overview

This page is a tabluated form showing all solutions for the design, manufaturing and operating conditions defined previously. An appropriate solution must be chosen to progress to the next page. Click the table on the row containing required option details to continue.

10.6 GEARCALC/ page 6

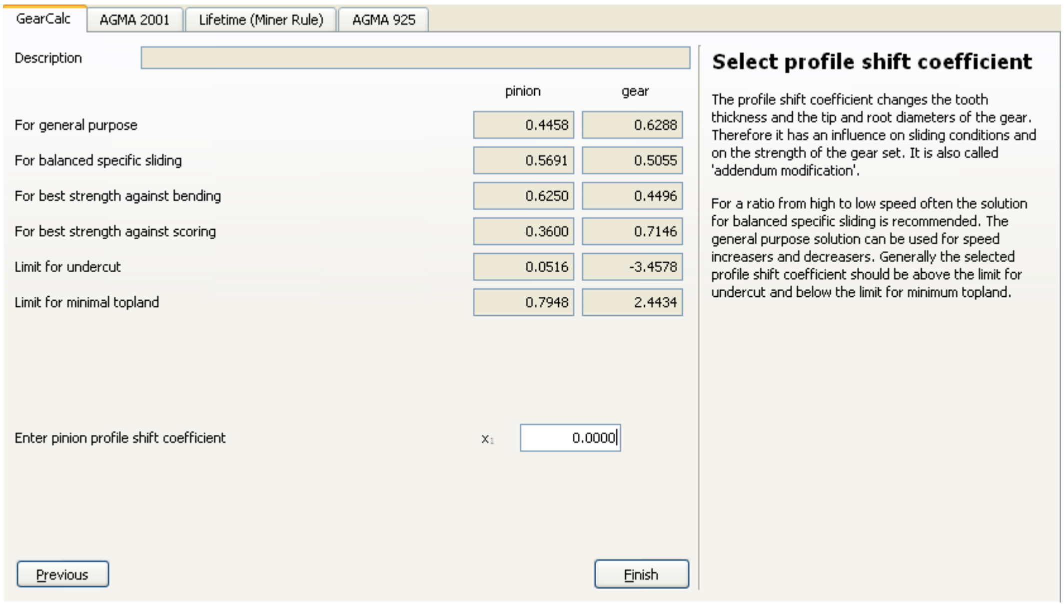


Figure 10.8: GEARCALC - Wizard page 6

This page allows the selections of a profile shift coefficient. Several proposals are made by the software.

10.6.1 Proposals for profile shift factors

- **General purpose** The profile shift factor is calculated according to a formula by Robert Errichello:

$$x_1 = \frac{\Sigma x}{u+1} + \frac{u-1}{3u} \qquad \text{for speed reducers}$$

$$x_1 = \frac{\Sigma x}{u+1} \qquad \text{for speed increasers}$$

- **Balanced specific sliding** The specific sliding at the beginning and the end of the contact has the same values on the root of the gears.

- **Best strength against bending** Choose x for the best bending strength

- **Best strength against scoring** Choose x for the best scoring resistance
- **Limit for undercut** The profile shift factor should normally not be less than this value for the undercut boundary.
- **Limit for minimal topland** The profile shift factor should not be bigger than this value. If you choose a bigger value the addendum has to be shortened to avoid a pointed tip..

10.6.2 Enter pinion profile shift factor

This field allows the user to enter the appropriate profile shift coefficient setting based on the above proposals.

Chapter 11

Calculation Settings

11.1 GEARCALC

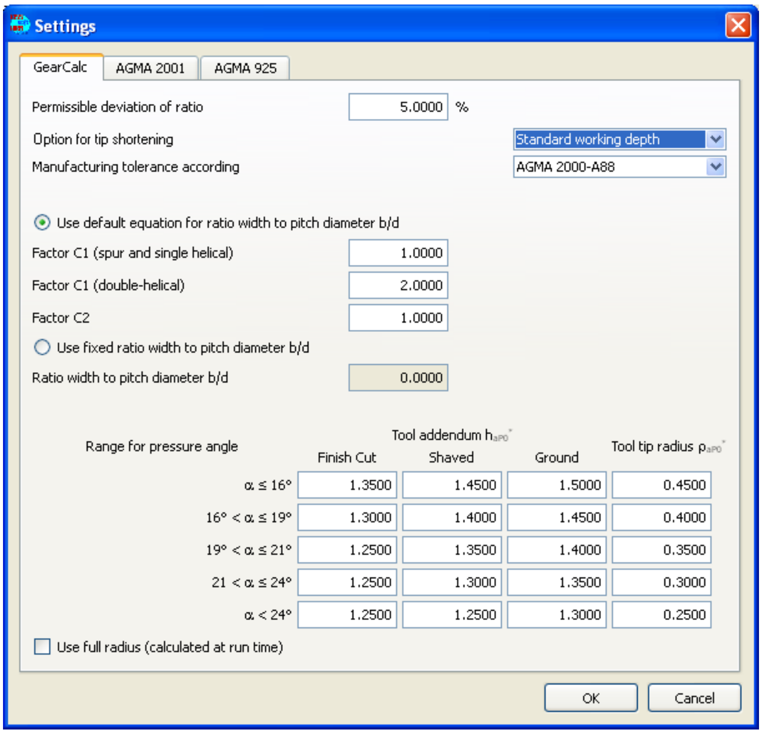


Figure 11.1: GEARCALC settings - page GEARCALC

11.1.1 Permissible deviation of ratio

There are often several designs which will achieve the required criteria but be outside the exact ratio. A permissible deviation as a percentage of the nominal ratio can be entered to allow the assessment of such designs.

11.1.2 Tip shortening

The sum of profile shift factors not equal to zero will decrease the tip clearance for external gear sets. To avoid this decrease of tip clearance a tip shortening is often made. For internal gear sets the sum of profile shift factors not equal zero will result in an increase of tip clearance. Therefore no automatic tip shortening is made for internal gear sets.

There is a choice of three tip treatment methods from drop down list:

`Full length teeth` The addenda of the gear and pinion are calculated without tip shortening:

$$h_{a1} = \frac{1 + x_1}{P_{nd}}$$
$$h_{a2} = \frac{1 + x_2}{P_{nd}}$$

CAUTION : Option may leave insufficient tip-to-root clearance if the operating center distance is much larger than the standard center distance.

`Standard working depth`

$$h_{a1} = \frac{1 + x_1 - k_s/2}{P_{nd}}$$
$$h_{a2} = \frac{1 + x_2 - k_s/2}{P_{nd}}$$

CAUTION : Option may leave insufficient tip-to-root clearance if the operating center distance is much larger than the standard center distance.

`Standard tip-to-root-clearance` This represents the safest calculation option but the contact ratio is reduced:

$$h_{a1} = \frac{1 + x_1 - k_s}{P_{nd}}$$
$$h_{a2} = \frac{1 + x_2 - k_s}{P_{nd}}$$

11.1.3 Manufacturing tolerance

The tolerance method can be defined for the calculation. A choice of AGMA 2000-A88 or AGMA 2015-1-A01 is available from the drop down menu. The scale runs from 15(best) to 3(worst) according to AGMA 2000 or from 2 (best) to 11(worst) according AGMA 2015. In ISO 1328 also the low numbers are for better quality like in AGMA 2015.

11.1.4 Calculate ratio face width to pitch diameter

There are two alternatives for establishing the ratio face width to pitch diameter m_a which ar toggled using the radio buttons;

The upper option activates the three cells directly under the radio button. Then factors C_1 and C_2 can be entered to define the ratio as follows;

$$m_a = (m_G/(m_G + C_2)) \cdot C_1$$

where:

$C_1 = 1.0$	for spur/helical gears
$C_1 = 2.0$	for double helical
$0 \leq C_2 \leq 1.0$	depending on user preference
$C_2 = 1.0$	suggested for general purposes

The lower button allows the direct input of the ratio of face width, $F\{b\}$, to pitch diameter, d:;

$$m_a = F\{b\}/d$$

This option activates the cell directly under the radio button. The cells for factors C1 and C2 will be de-activated, and the cell for the definition of width to pitch diameter can be accessed to enter a fixed value.

11.1.5 Tool addendum

The user can specify an addendum h^*_{aP0} of the tool for three given machining processes (finish cutting, shiving, and grinding) for a specified range of pressure angle designs. The tool addendum is measured from the datum line with $s_n = \pi/2/P_{nd}$. An associated radius, ρ^*_{aP0} can also be specified at this point. The tool addendum form is defined as follows:

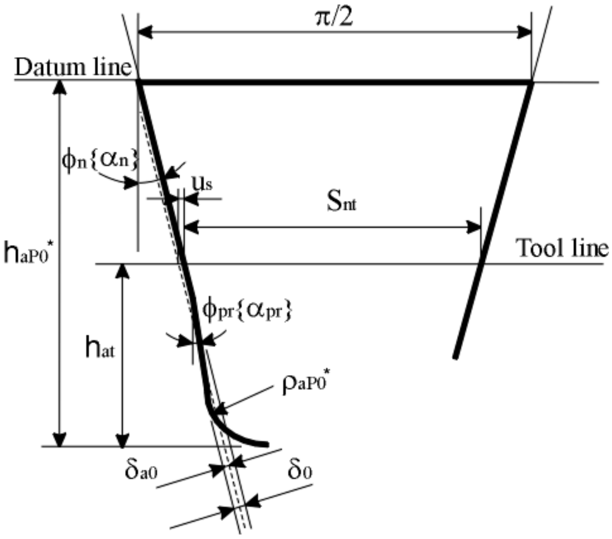


Figure 11.2: This figure shows a normal plane view of a rack-type generating tool (hob, rack cutter or generating grinding wheel).

11.1.6 Use full radius (calculated at run time)

This option implies that a radius is to be determined during the calculation (at run time) which will be the largest possible fitting to the defined tooth form tip.

11.2 AGMA 2001/2101

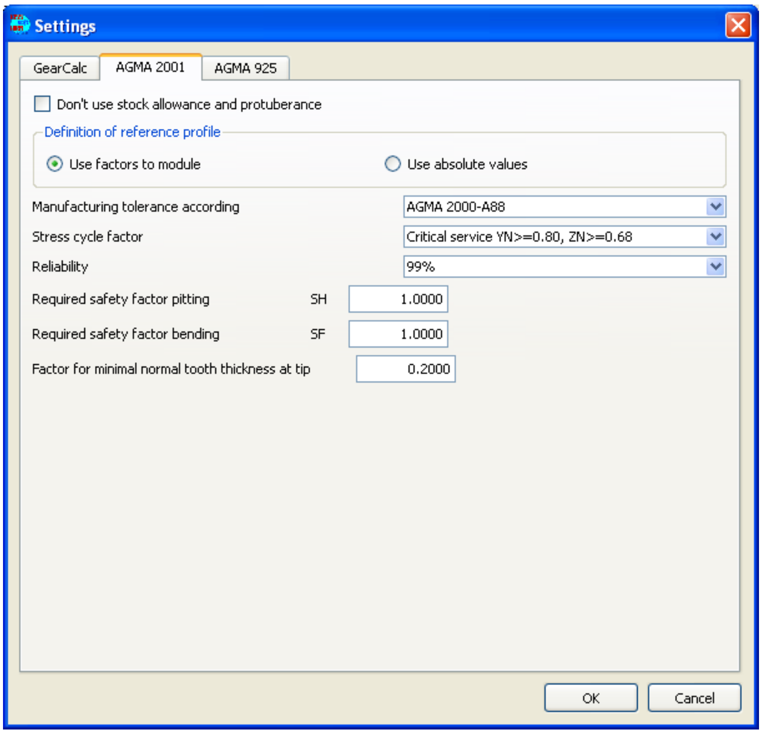


Figure 11.3: GEARCALC settings - page AGMA 2001

11.2.1 Don't use stock allowance and protuberance

The standard calculation procedure will use both stock allowance and protruberence defined on the tool profile. The check box on the 'General' tab-sheet will prevent this during the calculation.

11.2.2 Definition of reference profile

The reference profile dimensions such as addendum and dedendum can be defined in dimensionless multiples of module instead of mm or inch values using this setting. Normally the reference profile is given in factors of module (or $1/P_{nd}$).

11.2.3 Manufacturing tolerance according to standard

The tolerance method can be defined for the calculation. A choice of AGMA 2000-A88 or AGMA 2015-1-A01 is available from the drop down menu. The scale runs from 15(best) to 3(worst) according to AGMA 2000 or from 2 (best) to 11(worst) according AGMA 2015. In ISO 1328 also the low numbers are for better quality like in AGMA 2015.

11.2.4 Stress cycle factors

Three options are available to define stress cycle factors, Y_N for bending strength and Z_N for pitting resistance, based upon the application. For critical service $Y_N \geq 0.8$ is used while $Y_N \geq 0.9$ is used for general applications The option $Y_N \geq 1.0$ and $Z_N \geq 1.0$ is not recommended by AGMA and could be used for optimum contitions.
Note: Y_N for flanc/induction hardened steel (see chapter 10.2.1.1)

11.2.5 Calculation of tooth form factor

This options allows consideration of the tooth form which may concentrate loading on a specific area of the tooth. Consideration of loading expected at the tip or at HPSTC can be specified. This setting has only an influence on spur and LACR gears.

11.2.6 Reliability

The reliability factor,(K_R), accounts for the statistical distribution of fatigue failures found in materials testing. The required design life and reliability varies considerably with the gear application. Some gears are expendable, and a high risk of failure and a short design life are acceptable. Other applications such as marine gears or gears for power generation, require high reliability and very long life. Special cases such as manned space vehicles demand very high reliability combined with a short design life.

Reliability R	Application	Failure Frequency
0.9	Expendable gears. Motor vehicles.	1 in 10
0.99	Usual gear design	1 in 100
0.999	Critical gears. Aerospace vehicles	1 in 1000
0.9999	Seldom used.	1 in 10000

11.3 Choosing Bending/Pitting safety factors

An extra margin of safety can be specified by assigning $S_F > 1.0$ and/or $SH > 1.0$. Since pitting fatigue is slowly progressive, and pitted gear teeth usually generate noise which warns the gearbox operator that a problem exists, pitting failures are not usually catastrophic.Bending fatigue frequently occurs without warning and the resulting damage may be catastrophic.

The safety factors should be chosen with regard to the uncertainties in the load and material data and the consequences of a failure. Small safety factors can be used where the loads and material data are known with certainty and there are small economic risks and no risk to human life. However, if the loads and material data are not known with certainty and there are large economic risks or risks to human life, larger safety factors should be used. The bending fatigue safety factor is frequently chosen greater than the pitting safety factor ($S_F > S_H$) since bending fatigue may be catastrophic. However, S_F should not be too large because it leads to coarse-pitch teeth which may be noisy and prone to scoring failures.

Choosing a safety factor is a design decision that is the responsibility of the engineer. It must be carefully selected accounting for the uncertainties in:

- `External Loads`
  - Static or dynamic?
  - Load variation (time history)
  - Transient overloads
  - Loads from test data or service records?
- `Component Geometry`
  - Dimensional tolerances
  - Variation in fabrication
  - Surface finish, notches, stress concentrations
  - Damage during assembly (or incorrect assembly)
  - Quality assurance/inspection techniques
- `Material Properties`

  - Handbook values or test data for strengths?
  - Material procurement control
  - Heat treatment control
  - Quality assurance/inspection techniques

- `Design Analysis`
  - Is gear rating verified with computer programs AGMA2001 and Scoring? Will gears be tested before going into service?

- `Service Conditions`
  - Environment: thermal, chemical, etc.
  - Installation procedures
  - Operation procedures
  - Maintenance procedures

Consider the need to conserve material, weight, space or costs. Most importantly, consider:

- `Consequences of Failure`
  - Nature of failure modes
  - Risk to human life
  - Economic costs
  - Environmental impact

11.3.1 Factor for minimal normal tooth thickness at tip

This is the multiple of normal module which must exist at the tip. This factor is used to warn against pointed tip designs.

11.4 AGMA 925

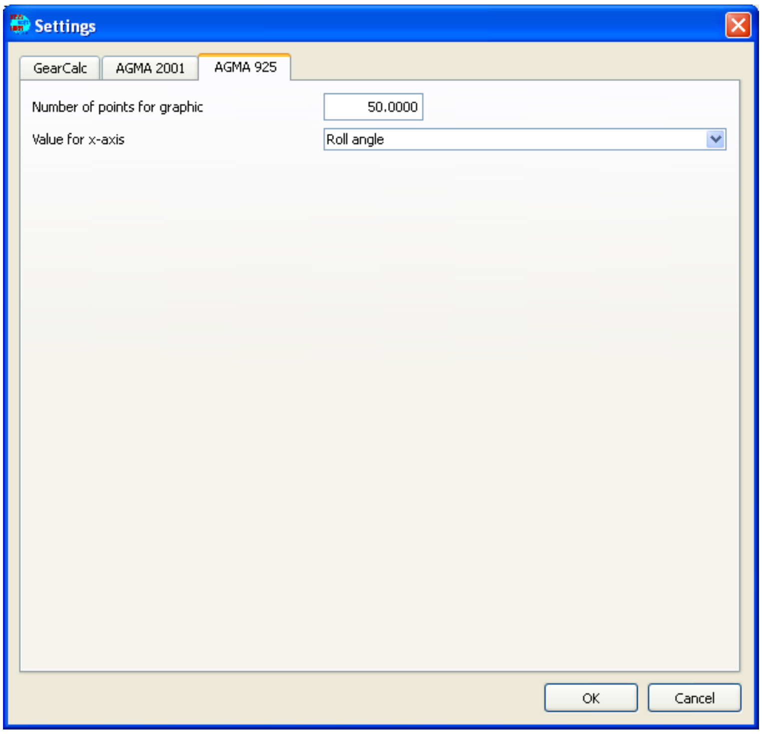


Figure 11.4: GEARCALC settings - page AGMA 925

11.4.1 Number of points for graphics

This cell can be used to determine the total number of points used in the graphics of the AGMA 925 calculation.

11.4.2 X-axis unit

There are three options available in the drop down list to plot X-axis unit values. These are in terms of roll angle, length of path of contact, and diameter.

Chapter 12

AGMA 2001/ 2101

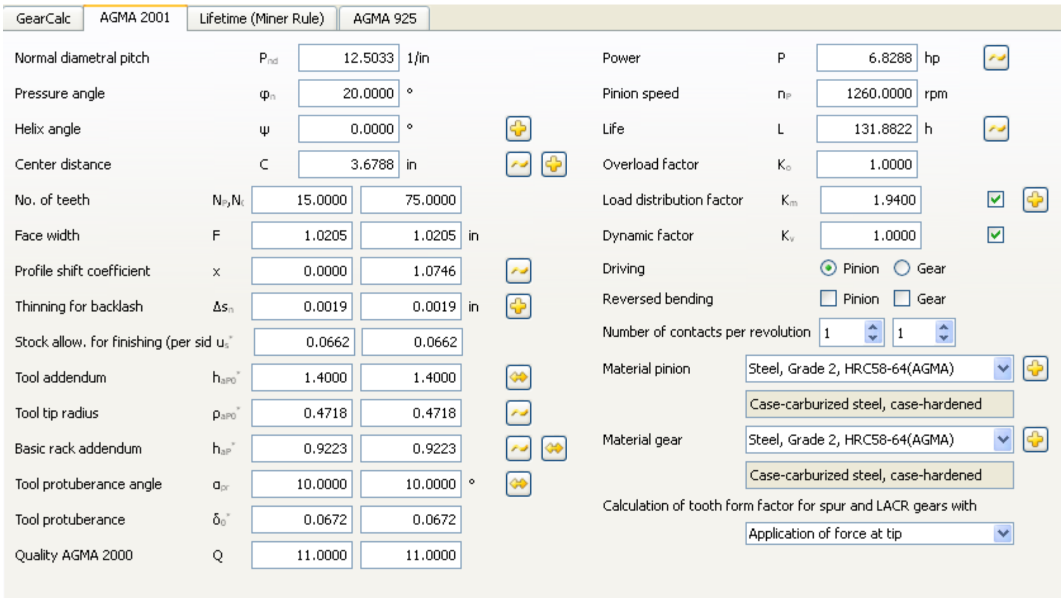


Figure 12.1: GEARCALC AGMA 2001/2101

If metric units (mm, N, kW) are selected AGMA 2101-D04 is used for the calculation, while AGMA 2001-D04 is used for the selection of US Customary units (in, lbf, hp). For US Customary units also diametral pitch Pnd is used instead of the normal module.

12.1 Normal module

The normal module m_n is only shown if metric units are selected (see 1.3.3). It is defined as $m_n = p/\pi$ and standard values are usually given in millimeters and can be found in ISO 54 or DIN 780. The size of the gears is increasing with the module.

The transverse module m_t is the normal module divided by the cosine of the helix angle: $m_t = m_n/\cos\psi$.

12.2 Normal diametral pitch

The normal diametral pitch P_{nd} defines the size of a tooth. It is π divided by the normal pitch $P_{nd} = \pi/p$. So the tooth thickness increases with a decreasing normal diametral pitch.

12.3 Normal pressure angle

$\phi_n\{\alpha_n\}$ is the standard or generating pressure angle. For hobbed or rack-generated gears, it is the pressure angle of the tool. For helical gears, ϕ_n is measured on the generating pitch cylinder in the normal plane. ϕ_n is standardized to minimize tool inventory:

ϕ_n (deg.)	Application
14.5	Low Noise
17.5	
20	General Purpose
22.5	
25	High load Capacity

`Low pressure angle:` Requires more pinion teeth ($N_p\{z_1\}$) to avoid undercut. Gives larger topland for same addendum modification coefficient.

`High pressure angle:` Allows fewer pinion teeth without undercut. Gives smaller topland for same addendum modification coefficient.

12.4 Helix angle

$\psi\{\beta\}$ is the standard or generating helix angle. The helix angle of a gear varies with the diameter at which it is specified. The standard helix angle is measured on the generating pitch cylinder.

For hobbed gears, the helix angle may be freely chosen because the hobbing machine can be adjusted to cut any helix angle. For pinion-shaped gears, the helix angle must correspond to the helical guides that are available for the gear-shaping machine.

ψ (deg.)	Application
0	spur
10-20	single helical
20-40	double helical

`Low helix angle:` provides low thrust loads but results in fewer teeth in contact (smaller face contact ratio, m_F and higher noise generation. For the full benefit of helical action, $m_F\{\epsilon_\beta\}$ should be at least 2.0. If $m_F < 1.0$ the gear is a low contact ratio (LACR) helical gear and is rated as a spur gear. Maximum bending strength is obtained with approximately 15 degree helix angles.

`High helix angle:` provides smooth-running, quiet gearsets but results in higher thrust loads unless double helical gears are used to cancel internally generated thrust loads.

The plus button  at the side of the field can be used to define the 'hand' (left or right) of the helix.

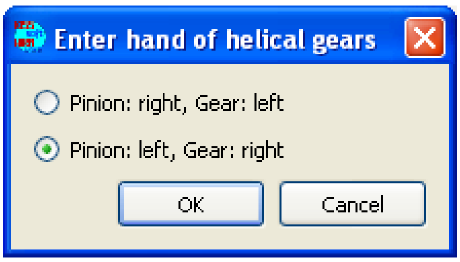


Figure 12.2: AGMA 2001/2101 - Helix angle

12.5 Center distance

The centre distance $C\{a\}$ is the theoretical distance between the origins of the pinion and gear on assembly. The plus button can be used to define an upper and lower tolerance for the centre distance. The sizing button can be used to calculate an appropriate center distance based a given sum for the profile shift coefficients.

Tolerances for the centre distance can be defined using the next to the input field. Normally the tolerances are defined symmetrically so one is positive and the other is negative.

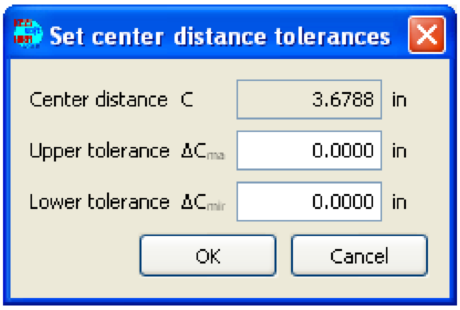


Figure 12.3: AGMA 2001/2101 - Center distance

12.6 Number of teeth

The numbers $N_P\{z_1\}$ and $N_G\{z_2\}$ represent the number of teeth on the 'pinion' and 'gear' respectively. As a default the data of the pinion is input in the left column, the data of the gear in the right column.

For spur gears you need a minimum number of 17 teeth to avoid undercut without any profile shift. You can achive a smaller number of teeth with an appropriate profile shift factor or using helical gears.

12.7 Face width

The face width $F\{b\}$ is the axial length over which the tooth of a gear is formed. This can be entered independently for both gears. The width should be smaller then the pinion diameter as a default, because the load distribution over the width is affeced by a large width of the gear.

12.8 Profile shift coefficient

A profile shift or addendum mofification can be made to have an influence on tooth shape and tooth thickness. According AGMA 908 the factor is called addendum modification coefficient according AGMA 913 and newer ISO standards profile shift coefficient is used.

12.8.1 Gears with standard addenda

For gears with standard addenda, the profile shift coefficients or addendum modification coefficients are zero, i.e.:

$$x_1 = x_2 = 0$$

The standard outside diameters may be calculated from the following equations using the gear ratio $m_G = N_G/N_P$

Standard pitch radii:

$$\begin{aligned} r &= N_P/(2 \cdot P_{nd} \cdot \cos\psi) \\ R &= r \cdot m_G \end{aligned}$$

Standard addenda:

$$\begin{aligned} h_{a1} &= 1/P_{nd} \\ h_{a2} &= 1/P_{nd} \end{aligned}$$

Standard outside diameters:

$$\begin{aligned} d_o &= 2 \cdot (r + h_{a1}) \\ D_o &= 2 \cdot (R + h_{a2}) \end{aligned}$$

Standard inside diameter (internal gears):

$$Di = 2 \cdot (R - h_{a2})$$

NOTE: The inside diameter of an internal gear is frequently made larger than that given by the above equation to avoid interference between the tips of the pinion and gear teeth.

12.8.2 Gears with addendum modification

Gear teeth may have modified addenda in order to avoid undercut, to balance the bending stresses in the pinion and gear, or to vary the relative amounts of approach and recess action. For external gears with increased addendum, there is a corresponding reduction in dedendum; i.e., the teeth are moved outward from the center of the gear. This profile shift, as it is called in newer standards, is expressed in terms of a profile shift coefficient or an addendum modification coefficient x where x is the proportionate distance (in terms of unity normal diametral pitch) by which the datum line of the generating rack (e.g., hob) and the generating pitch circle of the gear are separated.

The profile shift x is positive when the addendum is increased (the tooth thickness is also increased) by shifting the generating rack outward of the material of the generated gear. Existing conventions vary for internal gears; for AGMA2001 we define x_2 as positive when the reference generating rack is shifted out of the material of the gear resulting in an increased tooth thickness of the gear teeth.

The sum of the addendum modification coefficients is given by:

$$\Sigma x = x_1 + x_2$$

Gear pairs with modified addenda may operate on the same standard center distance as unmodified gears if the addendum modification coefficients are chosen as follows:

$$x_2 = -x_1$$

Then Σx = 0 and the gear pair may operate on standard centers.

Alternatively, Σx may be a positive number with the gear pair operating at a center distance larger than standard, or Σx may be a negative number with the gear pair operating at a center diatance smaller than standard.

The sizing button  at the side of this field allows the program to calculate coefficients suitable for a range of operating criteria:

- **General purpose** The profile shift factor is calculated according to a formula by Robert Errichello:

$$x_1 = \frac{\Sigma x}{u+1} + \frac{u-1}{3u} \qquad \text{for speed reducers}$$

$$x_1 = \frac{\Sigma x}{u+1} \qquad \text{for speed increasers}$$

- **Balanced specific sliding** The specific sliding at the beginning and the end of the contact has the same values on the root of the gears.
- **Balanced sliding speed at tip** The sliding velocity at the beginning and the end of the contact has the same values.
- **Best strength against bending** Choose x for the best bending strength
- **Best strength against scoring** Choose x for the best scoring resistance
- **Minimum x_1 without undercut or pointed tip** Choose x_1 so that no undercut occurs at the pinion and the minimal topland of the gear is still large enough.
- **Maximum x_1 without undercut or pointed tip** Choose x_1 so that the minimal topland of the pinion is large enough and no undercut occurs for the gear.

12.9 Thinning for backlash

It is customary to ignore backlash when determining the addendum modification coefficients x_1 and x_2, i.e., x_1 and x_2 are usually nominal values corresponding to zero backlash. The small adjustments (radial shifting) of the generating rack for tooth thinning are indirectly defined by specifying the amount the pinion and gear teeth are thinned for backlash, Δs^*_{n1} and Δs^*_{n2}. With this convention, the outside diameters of the gears are independent of the tooth thinning for backlash, and are based solely on the addendum modification coefficients x_1 and x_2. The root diameters will be changed with the tooth thinning, sinze the tool is moved further into the material.

Tolerances for the maximum and minimum values can be entered using the plus button  at the side of the field.

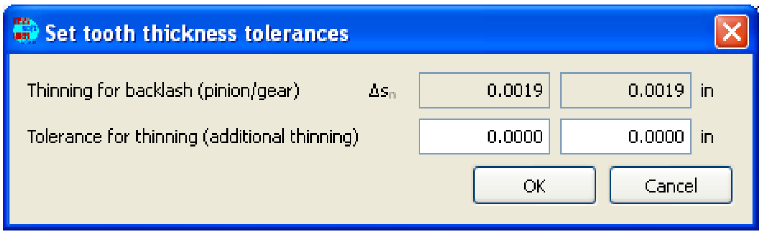


Figure 12.4: AGMA 2001/2101 - Tooth thickness tolerance

12.10 Stock allowance

If the stock allowance u_s^* has been activated under settings (see 11.2.1) then the cells will appear to define the amount of stock to be given on both gear and pinion. The stock allowance is given per side of the gear and in circumferial direction.

12.11 Tool addendum

The tool addendum h^*_{aP0} is defined from the datum line with the tooth thickness $\pi/2P_{nd}$ as follows:

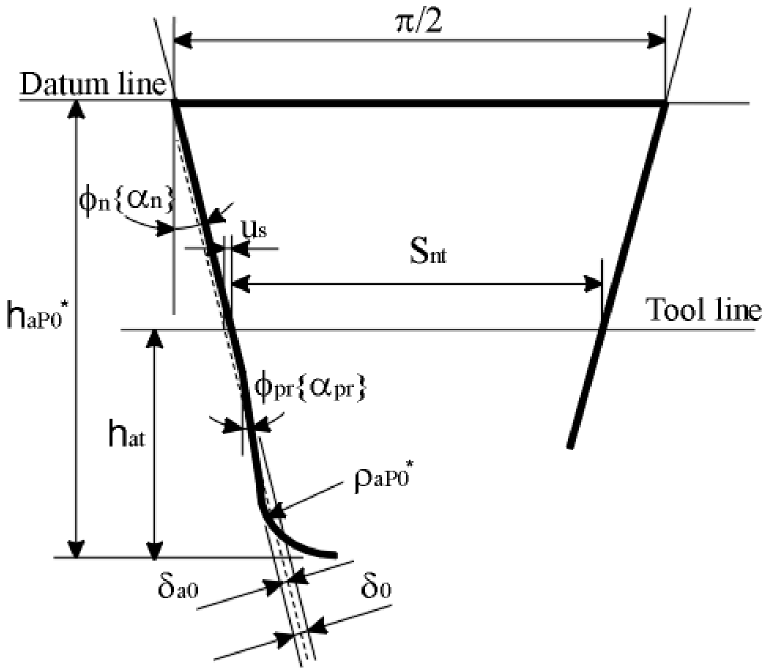


Figure 12.5: This figure shows a normal plane view of a rack-type generating tool (hob, rack cutter or generating grinding wheel).

Using the convert button the tool addendum h^*_{aP0} can also be calculated from an addendum h_{at} measured from a reference line with a different tooth thickness s_nt.

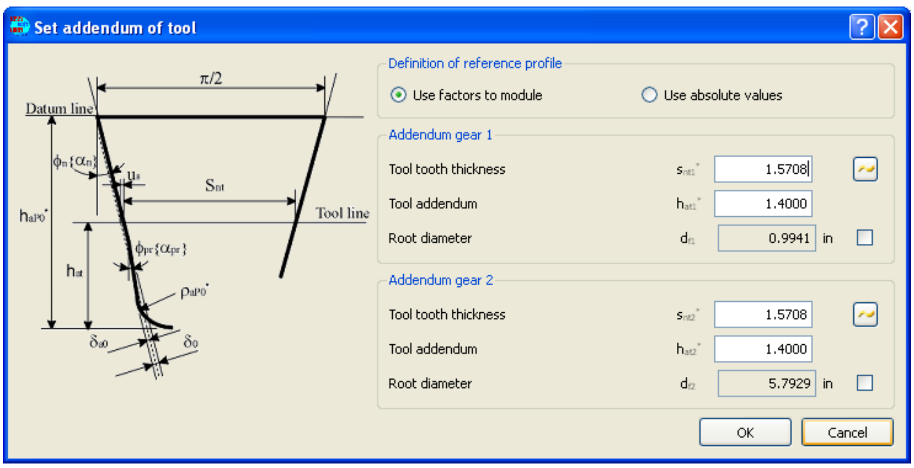


Figure 12.6: AGMA 2001/2101 - Addendum of tool

Where:

s_{nt} = normal tooth thickness of the tool at the tool line. This thickness is usually equal to $\pi/2$ (in terms of Pnd = 1.0) for gears that are not subsequently finished by shaving, grinding, skiving, etc. For gears that are finished by one of the above mentioned finishing methods, the tooth thickness of the rack-type cutting tool is sometimes made thinner than $\pi/2$ to provide stock allowance for finishing i.e.,

$$\begin{aligned} s_{nt} &= \pi/2 - 2 \cdot u_s \\ h_{aP0} &= \text{addendum of tool measured from the tool datum line.} \\ \rho_{aP0} &= \text{tip radius of tool.} \\ \delta_0 &= \text{protuberance of tool.} \end{aligned}$$

The tool addendum can also be calculated by a given root diameter using the convert button.

12.12 Tool tip radius

A tool tip radius $\rho_{aP0}*$ is added to the design is used to remove stress raisers in the finished gear root. A value can be entered for the gear and pinion individually directly into the cells provided.

The sizing button to the side of the cells can be usd to calculate the maximum radius that can be used on top of the tool. It is dependend on the pressure angle and the addendum of the tool.

12.13 Basic rack addendum/Tool dedendum

The gear addendum is created by the tool dedendum for a topping tool. Since topping tools which are also cutting the tip diameter are usual only for very small gears the dedendum of the tool is often bigger than the addendum of the basic rack h^*_{aP}. So the basic rack addendum h^*_{aP} is defining the outside diameter of the gear. The outside diameter of the gear is

$$d_a = (z/\cos\psi + 2 \cdot x + 2 \cdot h^*_{aP})/P_{nd}$$

Alternatively, the input values for the basic rack addendum can be calculated from the outside diameters by pressing the convert button  beside the field.

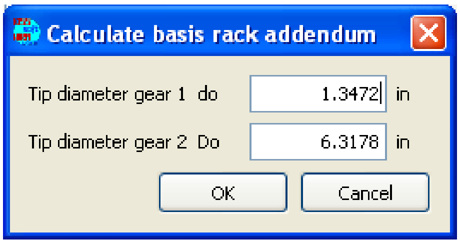


Figure 12.7: AGMA 2001/2101 - Basic rack addendum

The sizing button will set the values of the basic rack addendum to the values needed for constant tip clearance (see 11.1.2).

12.14 Tool protuberance angle

If the stock allowance has been activated under settings (see 11.2.1) then the cells will appear to define the protuberance angle α_{pr}. The convert button at the side of the field can be used to enter a height value, h_k, for the protuberance. The protuberance angle is automatically adjusted for the new values on returning to the main dialog after pressing `OK`.

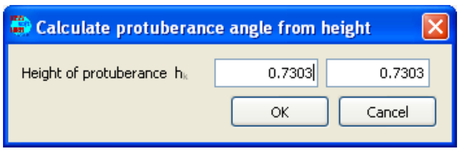


Figure 12.8: AGMA 2001/2101 - Protuberance angle

12.15 Tool protuberance

If the stock allowance has been activated under settings (see 11.2.1) then the cells will appear to define the amount of protuberance δ_0*. A cutting tool is provided with protuberance so that it will generate a relief in the tooth profile of the generated gear in the area of the tooth fillet. See Fig. 12.5. This relief allows the finishing shaving cutter or grinding wheel to run out without notching the root fillet. The protuberance of the cutter is usually made somewhat larger than the amount of finishing stock, i.e.,

$$\delta_0 > u_s \cdot \cos\phi_n\{\alpha_n\}$$

12.16 Quality according to AGMA

The required quality for both the pinion and gear can be defined independently. The scale runs from 15(best) to 3(worst) according to AGMA 2000 or from 2 (best) to 11(worst) according AGMA 2015. In ISO 1328 also the low numbers are for better quality like in AGMA 2015. Under settings (see 11.1.3) the used tolerance standard can be choosen.

The actual quality achieved is dependent upon the manufacturing process used.

12.17 Power

P is the power transmitted per gear mesh. For multiple power paths load-sharing must be considered:

Branched offsets: If the pinion meshes with two or more gears (or the gear meshes with two or more pinions), use the power of the more highly-loaded branch.

Epicyclic Gearboxes: The degree of load sharing depends on the number of planets, accuracy of the gears and mountings, provisions for self-aligning, and compliance of the gears and mountings.

A load increasing because of shocks can be considered using the overload factor K_o (see 12.20). The sizing button  can be used to let the software calculate the maximum power that can be transmitted with the gear set so that the required safeties are reached (see 11.3).

12.18 Pinion speed

Input the rotary speed of the pinion n_p as a positive number. The pinion is the gear with the smaller number of teeth. Here the values for the pinion are taken from the left columns of input data.

12.19 Life

A gearset's design life L is determined by the particular application. Some gears such as hand tools are considered expendable, and a short life is acceptable, while others such as marine gears must be designed for long life. Some applications have variable loads where the maximum loads occur for only a fraction of the total duty cycle. In these cases, the maximum load usually does the most fatigue damage, and the gearset can be designed for the number of hours at which the maximum load occurs.

Typical design lives:		
Application	No. Cycles	Design Life, L(hr)
Vehicle	$10^7 - 10^8$	3000
Aerospace	$10^6 - 10^9$	4000
Industrial	10^{10}	50000
Marine	10^{10}	150000
Petrochemical	$10^{10} - 10^{11}$	200000

The number of load cycles per gear is calculated from the required life (L), the speed (n) and the number of contacts per revolution (q):

$$N = 60 \cdot L \cdot n \cdot q$$

The sizing button  can be used to calculate the lifetime where the required safety factors (see 11.3) are reached.

12.20 Overload factor

The overload factor K_o makes allowance for the externally applied loads which are in excess of the nominal tangential load, W_t. Overload factors can only be established after considerable field experience is gained in a particular application. For an overload factor of unity, this rating mehtod includes the capacity to sustain a limited number of up to 200% momentary overload cycles (typically less than four starts 8 hours, with a peak not exceeding one second duration). Higher or more frequent momentary overloads shall be considered separately. In determining the overload factor, consideration should be given to the fact that many prime movers and driven equipment, individually or in combination, develop momentarypeak torques appreciably greater than those determined by the nominal ratings of either the prime mover or the driven equipment. There are many possible sources of overload which should be considered. Some of these are: system vibrations, acceleration torques, overspeeds, varitions in system operation, split path load sharing among multiple prime movers, and changes in process load conditions.

Examples of operating characteristics of driving machines:

- Uniform – Electric motor, steam turbine, gas turbine.
- Light shock – Multi-cylinder internal combustion engine with many cylinders.
- Medium shock – Multi-cylinder internal combustion engine with few cylinders.
- Heavy shock – Single-cylinder internal combustion engine.

Examples of operating characteristics of driving machines:

- Uniform – Generator, centrifugal compressor, pure liquid mixer.
- Light shock – Lobe-type blower, variable density liquid mixer.
- Medium shock – Machine tool main drive, multi-cylinder compressor or pump, liquid + solid mixer.
- Heavy shock – Ore crusher, rolling mill, power shovel, single-cylinder compressor or pump, punch press.

Operating Characteristics of Driving Machine	Operating Characteristics of Driven Machine			
	uniform	light shock	medium shock	heavy shock
uniform	1.00	1.25	1.50	1.75
light shock	1.10	1.35	1.60	1.85
medium shock	1.25	1.50	1.75	2.00
heavy shock	1.50	1.75	2.00	2.25

12.21 Load distribution factor

This factor allows for the variation in contact brought about by differing manufacturing processes, operating conditions and mounting error on assembly. The load distribution factor K_m can either be defined directly or calculated by the empirical method of AGMA 2001/2101. This empirical method is recommended for normal, relatively stiff gear designs which meet the following requirements:

1. Net face width to pinion pitch diameter ratios less than or equal to 2.0. (For double helical gears the gap is not included in the face width).
2. The gear elements are mounted between bearings, i.e., not overhung.
3. Face widths up to 40 inches.
4. Tooth contact extends across the full face width of the narrowest member when loaded.

The input values used for the empirical method for the load distribution factor calculation can be found by pressing the plus button  beside the field:

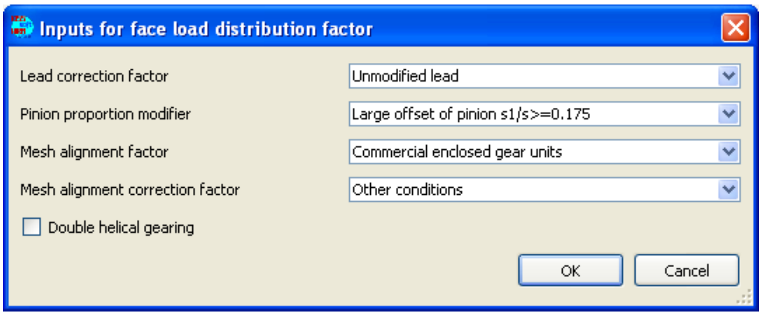


Figure 12.9: AGMA 2001/2101 - Face load distribution factor

12.21.1 Lead correction factor (C_{mc})

The nominal setting 'Unmodified lead' should be used when the machining quality is not known. An option 'Lead properly modified by crowning or lead correction' exists to define a well defined lead modification possible using high quality grinding machines.

Lead modification (helix correction) is the tailoring of the lengthwise shape of the gear teeth to compensate for the deflection of the gear teeth due to load, thermal or other effects. Certain gear grinding machines have the capability to grind the helical lead to almost any specified curve. Many high-speed gears are through-hardened, hobbed and shaved. Usually the gear member is shaved to improve the surface finish, profiles and spacing, but the helix lead is not changed significantly. The pinion and gear are then installed in the housing and a contact pattern is obtained by rolling the gears together under a light load with marking compound applied to the gear teeth. Based on the contact pattern obtained from this test, the pinion is shaved to match the lead of the gear. The process is repeated until the desired no-load contact pattern is obtained.

12.21.2 Pinion proportion modifier (C_{pm})

This setting allows consideration of the degree of alignment change as the pinion is offset under a defelction of the bearings. The C_{pm} value alters the pinion proportion factor, C_{pf}, based on the location of the pinion relative to its bearing center line.

12.21.3 Mesh alignment factor (C_{ma})

The mesh alignment factor C_{ma} accounts for the misalignment of the axes of rotation of the pitch cylinders of the mating gear elements from all causes other than elastic deformation. The factor is dependend on the face width and the follwing options:

- Open – This type of gearing is used in such applications as rotary grinding mills, kilns, dryers, lifting hoists and winches. These gears are frequently of low accuracy because their large size limits the practicable manufacturing methods. The gear shafts are usually supported by separate pedestal bearings with the gears covered by sheet metal shields. The gear mesh alignnent is dependent on the skill and care exercised in the mounting and alignment of the shaft bearings.
- Commercial – This classification pertains to low speed, enclosed gear units, which employ gears that are through-hardened and hobbed or shaped, or hobbed or shaped and surface hardened and which are not subsequently finished by shaving or grinding.
- Precision – This classification pertains to low or high speed, enclosed gear units, which employ gears which are finished by shaving or grinding.

- Extra Precision – This classification pertains to high speed, enclosed gear units, which employ gears which are finished by grinding to high levels of accuracy. The lead and profiles of the gear teeth are usually modified to compensate for load deflections and to improve the meshing characteristics.

12.21.4 Mesh alignment correction factor (C_e)

This selection can be used to account for improved corrective action after manufacturing for a better contact condition.

Some gearsets are adjusted to compensate for the no-load shaft alignment error by means of adjustable bearings and/or by re-working the bearings or their housings to improve the alignment of the gear mesh. Lapping is a finishing process used by some gear manufacturers to make small corrections in the gear tooth accuracy and gear mesh alignment. Lapping is done by either running the gear in mesh with a gear-shaped lapping tool or by running the two mating gears together while an abrasive lapping compound is added to the gear mesh to promote removal of the high points of the gear tooth working surface.

12.21.5 Double Helical

For double-helical gears, the mesh alignment factor is calculated based on one helix (one half of the net face width).

12.21.6 Transverse load distribution factor

Since no information about the transverse load distribution factor $C_{mt}\{K_{H\alpha}\}$ is given in AGMA 2001 the load distribution factor is equal to the face load distribution factor. $K_m = C_m f\{K_{H_\beta}\}$

12.21.7 Notes

It usually is not possible to obtain a perfectly uniform distribution of load across the entire face width of an industrial gearset. Misalignment between the mating gear teeth causes the load and stress distribution to be non-uniform along the tooth length. The load distribution factor is used to account for the effects of the non-uniform loading. It is defined as the ratio of the maximum load intensity along the face width to the nominal load intensity, i.e.,

$$K_m = C_m = \text{Maximum Load Intensity}/(W_t/F)$$

Variations in the load distribution can be influenced by:

`Design Factors`
Ratio of face width to pinion diameter
Bearing arrangement and spacing
Internal bearing clearance
Geometry and symmetry of gear blanks
Material hardness of gear teeth

`Manufacturing Accuracy`
Gear housing machining errors (shaft axes not parallel)
Tooth errors (lead, profile, spacing & runout)
Gear blank and shaft errors (runout, unbalance)
Eccentricity between bearing bores and outside diameter

`Elastic Deflection of:`
Gear tooth (bending)
Gear tooth (hertzian)
Pinion shaft (bending and torsional)
Bearings (oil film or rolling elements)
Housing

`Thermal Distortion of:`
Gear teeth, gear blank, shafts, and housing

`Centrifugal Effects`
Centrifugal forces may cause misalignment for high-speed gears

`External Effects`
Misalignment with coupled machines
Gear tipping from external loads on shafts

External thrust from shaft couplings

12.22 Dynamic factor

The dynamic factor K_v accounts for internally generated gear tooth loads which are induced by non-uniform meshing action (transmission error) of gear teeth. If the actual dynamic tooth loads are known from a comprehensive dynamic analysis, or are determined experimentally, the dynamic factor may be calculated from:

$$K_v = (W_d + W_t)/W_t$$

where W_t = Nominal transmitted tangential load and
W_d = Incremental dynamic tooth load due to the dynamic response of the gear pair to the transmission error excitation, not including the transmitted tangential loads.

If the factor is calculated according AGMA, the Transmission Accuracy Grade A_ν is used. A_ν is calculated following formula (21) in AGMA2001, page 15. Therefore A_{nu} is not always identical but close to the gear quality.

CAUTION: This factor has been redefined as the reciprocal of that used in previous AGMA standards. It is now greater than 1.0. In earlier AGMA standards it was less than 1.0.

12.23 Driving

The software needs to know whether pinion or gear is driving when determining the optimum addenda modification for maximum scoring resistance. The driving member influences load-sharing between successive pairs of teeth and load distribution along the path of contact. This in turn influences the flash temperature and scoring resistance.

12.24 Reversed bending

Usually a pair of gears rotate in one direction without torque reversals and the gear teeth are loaded on one side only. For this case, the gear teeth are subjected to one-way bending or uni-directional loading.

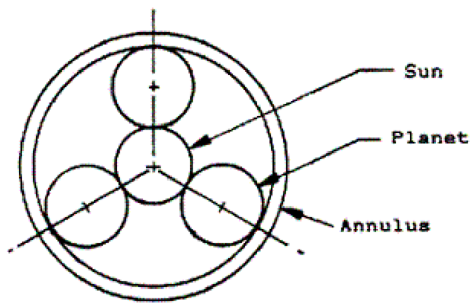
Some gears are loaded on both sides of the teeth and are subjected to reverse bending. Examples are:

- idler gears
- planet gears (planetary or star gear systems)
- gearsets which have fully reversed torque loads

In this case the strength of the gears is reduced.

12.25 Number of contacts per revolution

For a single pinion in mesh with a single gear, each member has one contact per revolution. Some gears have more than one cycle of load contact per revolution. An epicyclic gearset (planetary or star gear) is shown below:

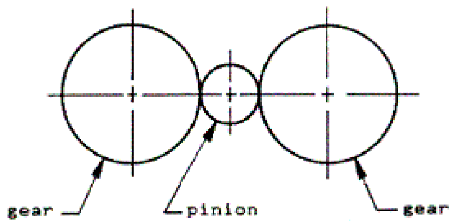


Sun The gear has Q contacts/rev, where Q = number of planets. For the example shown, the sun gear has 3 contacts/rev.

Planet The planet gear has 1 contact/rev because the loads from the sun gear and ring gear occur on opposite sides of the planet gear teeth. The reverse bending that occurs on the planet gear teeth is accounted for with the "Loading-type Code" (See chapter 12.24).

Annulus (planetary gear train) The internal gear has Q contacts per revolution, where Q = number of planets. Although the internal gear in a planetary gearset is fixed, it is analyzed as if it were rotating at the planet carrier speed.

Annulus (star gear train) – the internal gear has Q contacts per revolution of the internal gear where Q = number of planets. An example of a split-power-train (branched) gearset is shown below:



In this example, if the pinion is the driver or is driven, it has 2 contacts/rev. If the pinion is an idler, it has 1 contact per revolution and reversed bending. The mating gears each have 1 contact/rev.

12.26 Material

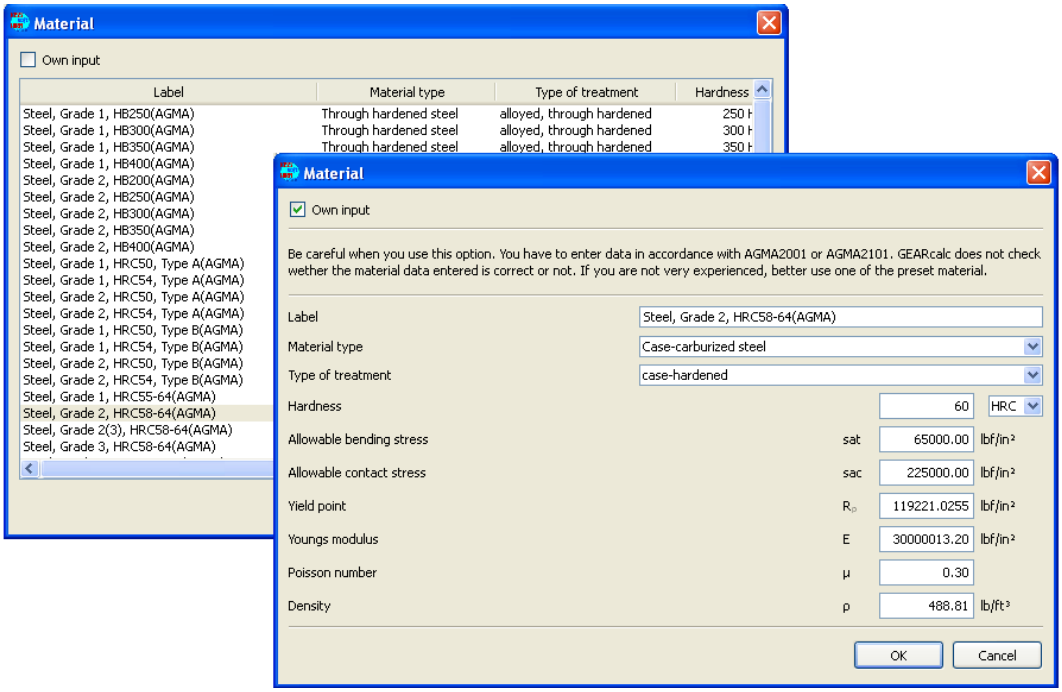


Figure 12.10: AGMA 2001/2101 - Material

The material of the gears can be selected from the material database. The strength is dependend of material type, treatment and quality.

12.26.1 Material treatment

There are different possibilities for heat treatment: through hardened, nitrided, induction hardened and case hardened materials:

- **Through hardened:** annealed, normalized or quenched and tempered. Carbon content ranges from 0.30 to 0.50%. Alloy content ranges from plain carbon steels (e.g. MSI 1040) for tiny gears, to Cr-Ni-Mo alloys (e.g.AISI 4340) for large gears. The best metallurgical properties are obtained with quenched and tempered steels. Hardness ranges from HB = 180 for lightly-loaded gearsets, to the limit of machinability (approximateby HB = 360) for highly-loaded gears.

Good tooth accuracy (typically $Q = 10$ acc. AGMA2000) can be obtained by hobbing the teeth after heat treatment, eliminating heat treatment distortion from the generated tooth forms. Hardenability must be adequate to obtain the required hardness at the root diameter.

- **Nitrided gears** are quenched and tempered to obtain the desired core properties, then the teeth are cut and finished, followed by the nitriding process. fle gears are placed in an ammonia gas atmosphere where nitrogen is absorbed into the surface bayers of the gear teeth and forms hard fron nitrides. Because nitriding is performed at the relatively low, temperature of 950-1050 °F, and there is no quench, the distortion due to heat treatment is small. Surface hardness ranges from HB = 432 for alloys such as AISI 4340 to HB = 654 for Nitralloy 135M and 2.5% chrome alloys. The practical limit on case depth is about 0.025 in, which limits the application of nitriding to pitches finer than approximately $P_{nd} = 8$.

- **Induction hardened** gear teeth are heated by electromagnetic induction from a coil or inductor and are immediately quenched. Because only the surface layers of the gear teeth are hardened, heat treat distortion is minimized. Very tight controls of every step of the process are necessary for satisfactory results, and it is best for high-volume production where the process can be optimized. Several gears from each production run must be destructively inspected for case depth to ensure that the induction hardening is properly controlled. Carbon content of induction hardened gears is usually 0.40 or 0.50%. Plain carbon steels (e.g. AISI 1050) may be used for small gears, while alloys such as AISI 4350 may be used for large gears.

- **Carburized gears** are first cut, then heated in a carbon atmosphere (usually gas carburizing) which causes carbon to diffuse into the surface layers of the gear teeth. The gears are either quenched from the carburizing temperature or cooled, reheated and quenched later. Most gears are tempered at 300-400 °F after carburizing and quenching. Carbon content of carburizing steels range from 0.15 to 0.25%. Low alloy steels (e.g. AISI 8620) are used for small gears and moderate loads while high alloy steels (e.g. AISI 4820) are used for large gears and high loads. Minimum surface hardness ranges from HB = 615 to HB = 654. Because carburized gears are subjected to a drastic quench from a high temperature the distortion is large, and grinding is usually required to obtain acceptable accuracy.

12.26.2 Material quality

Material quality strongly influences pitting resistance and bending strength. For high quality material, the following metallurgical variables must be carefully controlled:

- Chemical coposition
- Hardenability
- Toughness
- Surface and core hardness
- Surface and core microstructure
- Cleanliness/inclusions
- Surface defects (flanks and root flllets)
- Grain size and structure
- Residual stress pattern
- Internal defects, seams or voids
- Microcracks
- Carbide network
- Retained austenite
- Intergranular oxidation
- Decarburization

There are three basic grades of material:

Grade 1: Commercial quality typical of that obtained from experienced gear manufacturers doing good work. Modest level of control of the metallurgical variables.

Grade 2: High quality typical of aircraft quality steel with cleanliness certified per AMS 2301 or ASTM A534. Close control of critical metallurgical variables.

Grade 3: Premium quality typical of premium aircraft quality with cleanliness certified per AMS 2300 or .ASTM A535. Absolute control of all metallurgical variables.

12.26.3 Own input of material data

Using the plus button  next to the material list the material values can be entered directly by the user. You have to be careful choosing the values since they are not checked by the software. Important for the calculation are the allowable stress numbers $s_{ac}\{\sigma_{Hlim}\}$ and $s_{at}\{\sigma_{Flim}\}$. The youngs module is needed for the hertzian stress and the yield point for the static strength. The hardness value is only used for documentation.

12.27 Calculation of tooth form factor

The point of force to be assumed by the calculation of tooth form factor for spur and LACR gears is defined here. The drop down list allows the definition of force applied at tip or at the high point of single tooth contact (HPSTC). For low quality gears loading at the tip should be choosen because of the influence of pitch errors. For high quality gears the single contact point can be choosen to consider load sharing between several pairs of teeth. See AGMA 908-B89 Table 5-1 for limits of the load sharing.

For helical gears with an axial contact ratio $m_F \epsilon_\beta > 1$ this input is not used.

Chapter 13

Lifetime (Miner Rule)

13.1 Calculating Lifetime according Miners rule

The Palmgren-Miner Linear-cumulative-fatigue-damage-theory (Miner's Rule) is used to calculate the resultant pitting or bending fatigue lives for gears that are subjected to loads which are not of constant magnitude but vary over a wide range. According to Miner's Rule, failure occurs when:

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + ... + \frac{n_i}{N_i} = 1$$

	n_i	$=$	number of cycles at the i^{th} stress level.
where:	N_i	$=$	number of cycles to failure correspontiing to the i^{th} stress level.
	n_i/N_i	$=$	damage ratio at the i^{th} stress level.

Instead of load cycles we can alo use lifetimes:

$$\frac{l_1}{L_1} + \frac{l_2}{L_2} + ... + \frac{l_i}{L_i} = 1$$

	l_i	$=$	time at a the i^{th} stress level.
where:	L_i	$=$	permissible lifetime at the i^{th} stress level.
	l_i/L_i	$=$	damage ratio at the i^{th} stress level.

Assuming the fraction of time at each stress level is known rather than the actual number of cycles or times, then:

$$
\begin{aligned}
l_1 &= \alpha_1 \cdot L \\
l_2 &= \alpha_2 \cdot L \\
l_i &= \alpha_i \cdot L
\end{aligned}
$$

where:

α_i	=	fraction of time at the i^{th} stress level.
L	=	Resultant number of cycles to failure under the applied load spectrum.

Defining the time ratio as:

$$\alpha_i = l_i/L = n_i/N$$

Miner's Rule may be rewritten as:

$$\alpha_1 \frac{L}{L_1} + \alpha_2 \frac{L}{L_2} + \ldots + \alpha_i \frac{L}{L_i} = 1$$

Which may be solved for the resultant life:

$$L = \frac{1}{\frac{\alpha_1}{L_1} + \frac{\alpha_2}{L_2} + \ldots + \frac{\alpha_i}{L_i}}$$

The load spectrum is defined by the time ratio, α_i, and the load ratio, β_i and additionally a speed ratio ω_i is needed for the calculation of the permissible lifetimes L_i.

where:

β_i	=	instantaneous load/baseline load
ω_i	=	instantaneous speed/nominal load

The baseline load is entered with the Load Data input screen by specifying the transmitted horsepower and speed of the pinion. The load spectrum is entered on the page `Lifetime`:

13.2 Define a lifetime calculation

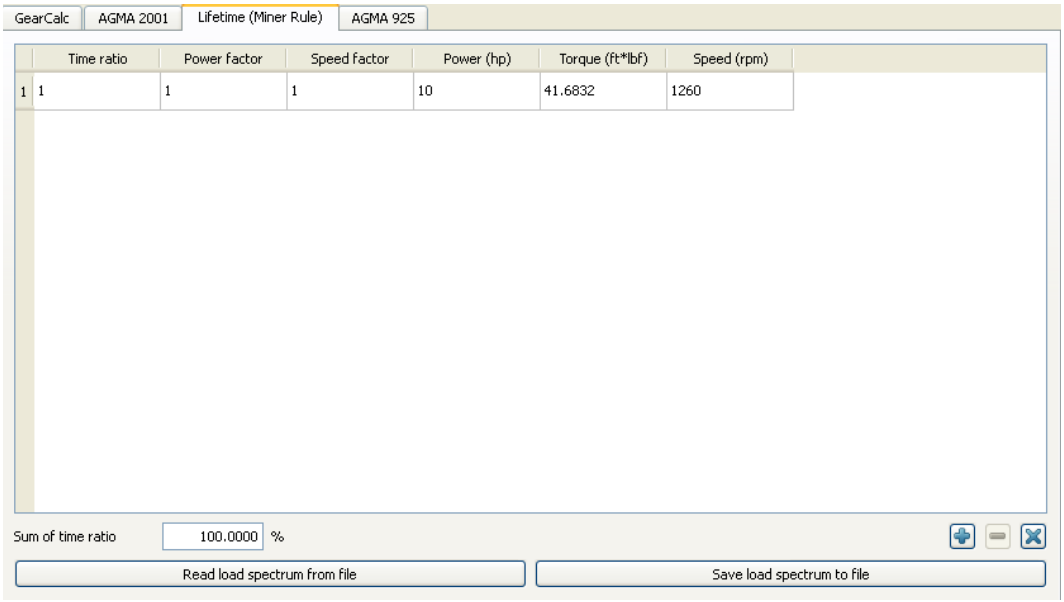


Figure 13.1: Gearcalc - Lifetime calculation

13.2.1 Create a load spectrum element

On this screen is a table containing at least one row. Each row element is used to define the individual characteristics for a proportion of running time at a specified load. A collection of more than one elements for multiple operating levels represents a load spectrum. Each element entry contains six characteristics;

```
Time Ratio
Power Factor
Speed Factor
Power
Torque
Speed
```

Three buttons at the bottom right of the table control the construction of the elements in the load spectrum. The [+] button adds another row element to the table. The [-] button will delete the any row currently selected in the table. The [x] button will clear the table of all but one row entry.

13.2.2 Sum of time ratio

This represents the total operating time (as a percentage) defined by the sum of the ratios in the first column of the table. The time ratio column is summed and multiplied by 100.

13.2.3 Save spectrum

An table which has been defined can be stored for future use or in association with other designs. On pressing the button indicated under the table a directory window opens to allow the user to specify the file name and directory required for storage.

13.2.4 Reload spectrum

An existing table containing a saved load spectrum can be reloaded using the button indicated. A directory window opens to allow the user to select the file required.

Chapter 14

AGMA 925 - Scoring

The AGMA925-A03 **Effect of Lubrication on Gear Surface Distress** is currently the only standard that calculates the conditions in the lubrication gap over the tooth contact. AGMA925 describes the calculation of the height of the lubrication gap taking into account the curvature of the flanks, properties of the lubricant, sliding speed and the local stress load. On this basis, the standard calculates the probability of wear (by means of metallic contact by the surfaces if the lubrication gap is too small).The standard itself does not provide any notes on protection against micropitting. It is known, however, from literature and research results that there is a direct correlation between the minimum lubrication gap size and the occurrence of micropitting. The calculation method can therefore be used when gearing is to be optimized to resist micropitting.

The probability of the occurrence of scuffing is also determined in accordance with AGMA925. This calculation has the same basis (Blok's equation) as the calculation of scuffing in accordance with the flash temperature criteria under DIN3990 part 4. The determination of the permitted scuffing temperature under AGMA925 is somewhat problematic because comprehensive or generally applicable notes are missing in this area. In particular there is no reference to the scuffing load load capacity specification according to the FZG test. Oils with active EP additives therefore have a tendency to be undervalued.

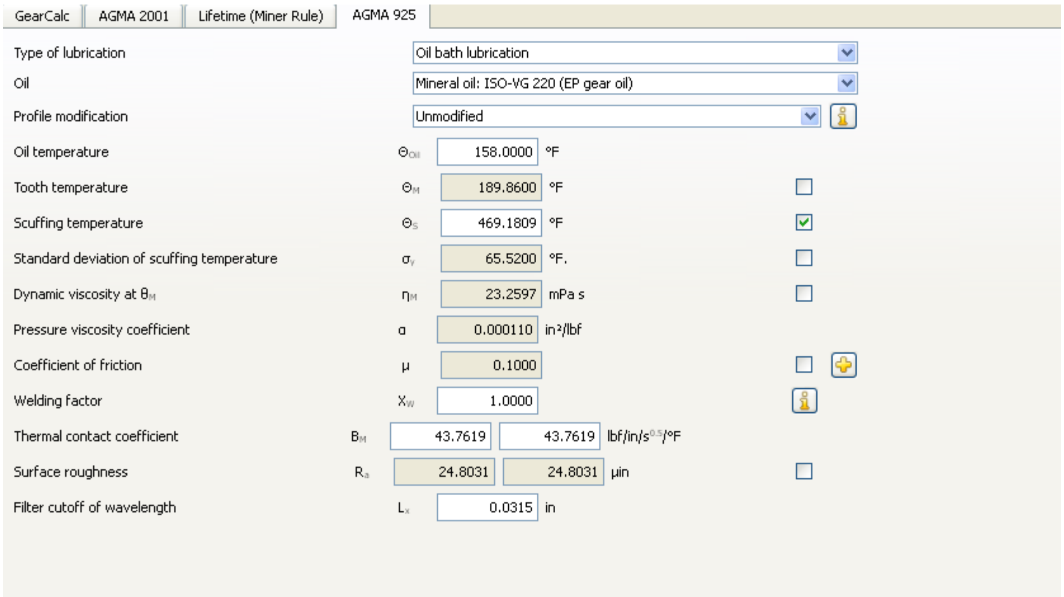


Figure 14.1: GEARCALC - AGMA 925

14.1 Type of lubrication

Grease or oil lubrication (oil bath, oil mist, or oil injection process) are the options in the list.

14.2 Oil

There are numerous oils and greases from which an appropriate option can be selected. The data for this oil type will be used by the calculation.

14.3 Profile modification

You can make corrections to the theoretical involute (profile modification). The type of profile modification has an impact on the calculation of the scoring safety. The Distribution factor (or Force Distribution factor) XGam is calculated differently depending on the type of profile modification. There is a significant difference between cases with and without profile correction.

The difference between profile correction 'for high load capacity' gears and thise 'for smooth meshing' however is not so important. The calculation procedure requires that the C_a (of the profile correction) is sized according to the applied forces, but does not indicate an exact value.

14.4 Oil temperature

The Oil Temperature Θ_{oil} is the input required for the calculation of the effective oil viscosity.

14.5 Tooth temperature

The tooth temperature (bulk temperature) Θ_M that is relevant to the analysis of flash temperature and film thickness is the bulk temperature of the surfaces of the gear teeth just before they engage. The gear tooth bulk temperature is an important component of the total temperature that occurs during engagement of the gear teeth, which consists of the bulk temperature plus the instantaneous flash teruperature rise, i.e.:

$\Theta_B = \Theta_M + \Theta_{fl}$

It is the total contact temperature,Θ_B , which controls the scoring (scuffing) mode of gear tooth failure. Besides being an important contributor to the gear tooth total temperature, the bulk temperature controls the operating viscosity of the lubricant which is entrained into the gear tooth contact. The entrained lubricant is in thermal equilibrium with the surfaces ot the gear teeth and its viscosity determines the thickness of the EHD oil film. lt is therefore imperative that an accurate value of gear bulk temperature be used as input to Scoring.

In some cases, the equilibrium gear bulk temperature may be significantly higher than the temperature of the oil supplied to the gear mesh. For example, reference tested high-speed, single-helical gears typical of gears used in the turbo-machinery of the petro-chemical industry. With oil nozzles supplying lubricant to the outgoing side of the gear mesh, the temperature of the pinion teeth was 180 deg. F (76 deg. F rise over the inlet oil temperature) at a pitch line velocity of v_{tr} = 20,000 fpm, and 275 deg. F (171 deg. F rise) at v_{tr} = 40,000 fpm. For the mating gear the temperature was 138 deg. F (34 deg. F rise) at v_{tr} = 20,000 fpm, and 208 deg. F (104 deg. F rise) at v_{tr} 40,000 fpm.

This example indicates that the bulk temperature of ultra-high-speed gears may be significantly higher than the temperature of the oil supply (171 deg. F rise at $v_{tr} = 40{,}000$ fpm) and that the pinion can be very much hotter than the gear (67 deg. F difference at $v_{tr} = 40{,}000$ fpm).

14.6 Scuffing temperature

In the list can the user to select from three options for determining the scuffing temperature Θ_S:

1. Own input.
2. Calculation according to AGMA925 (equations 94/95).
3. Calculation according to ISO/ TR 13989-1 (2000).

14.7 Standard deviation of scuffing temperature

This is a statistical measure defining the variation in scuffing temperature σ_V.

14.8 Dynamic viscocity at Θ_M

This is the viscocity η_M of the oil expected at the bulk temperature achieved during operation.

14.9 Coefficient for pressure viscocity)

The coefficients k and s are used to determine the pressure viscocity coefficient, α. The 'k' value is a linear multiple, while the 's' value is an exponential power for the dynamic viscocity, η_M. These coefficients are found under `Lubricant Data` in the report.

14.10 Coefficient of friction

There are three options for determining the coefficient of friction μ:

- Own input of constant value.
- Constant value calculated according to AGMA925 equation 85.
- Constant value calculated according to AGMA925 equation 88.

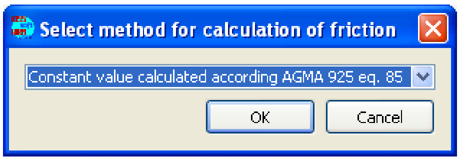


Figure 14.2: AGMA 925 - Calculation of friction

The value for the coefficient of friction can be entered directly by checking the box at the side of the field or accept the program default for a constant value. Alternatively, the user may request a variable coefficient of friction in which case Scoring calculates according to AGMA925.

14.11 Thermal contact coefficient

The thermal contact coefficient B_M accounts for the influence of the material properties of pinion and gear:

$$B_{M1} = \sqrt{\lambda_{M1} \cdot \rho_{M1} \cdot c_{M1}}$$
$$B_{M2} = \sqrt{\lambda_{M2} \cdot \rho_{M2} \cdot c_{M2}}$$

For martensitic steels the range of heat conductivity, λ_M, is 41 to 52 N/sK and the product of density times the specific heat per unit mass, $\rho_M \cdot c_M$ is about $3.8N/[mm^2K]$, so that the use of the average value $B_M = 13.6N/mms^{0.5}K$ for such steels will not introduce a large error when the thermal contact coefficient is unknown.

14.12 Surface roughness

The initial (as manufactured) surface roughness R_a of the working profiles of gear teeth depends primarily on the manufacturing method. The surface roughness to be used as input data for Scoring should be the surface roughness (micro-in rms) of the gear tooth profiles after they are run-in. The degree of improvement in surface roughness depends on the surface hardness of the gear teeth, the initial as-manufactured surface roughness and the operating conditions of load, speed and lubrication regime. The surtace roughness of slow speed, low hardness gears with an initial surface roughness of 80 micro-in rms might have up to a 4:1 improvement by running-in to 20 micro-in rms. Medium-hard, medium-speed gears commonly have 2:1 improvements by running-in from say 60 micro-in rms to 30 micro-in rms, while the surfaces of high-speed carburized gears may improve from 25 micro-in rms to 17 micro-in rms by running-in.

Users should obtain data for the surface roughness after run-in from tests on their particular gears. In lieu of this data, the following table gives typical values of surface roughness before and after run-in:

Surface Roughing (micro-in rms)		
Gear Tooth Manufacturing Method	As Manufactured	After run-in
Milling	64 - 125	32 - 64
Shaping	32 - 125	25 - 50
Hobbing	30 - 80	20 - 45
Lapping	20 - 100	20 - 40
Shaving	10 - 40	10 - 25
Grinding	10 - 40	10 - 25
Honing	6 - 20	5 - 15

14.13 Filter cut-off of wavelength

This setting can be used to define the wave length limit L_x for the surface roughness calculation. No wavelength with an amplitude above this value will be considered.

Standard values are shown in the following table:

mm	in
0.08	0.003149606
0.25	0.009842520
0.80	0.031496063
2.50	0.098425197
8.00	0.314960630

Part III

Appendix: Bibliography and Index

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