

MESYS Calculation Software USER MANUAL

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





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General

1 General

This chapter gives some general information about the MESYS Calculation Software. Installation, updates and general usage are covered.

Details about the different calculation programs are provided in the following chapters. Available programs are:

- [Shaft Calculation](#)  42
- [Rolling Bearing Calculation](#)  124
- [Ball Screw Calculation](#)  214
- [Hertzian Stress Calculation](#)  236
- [Axial-Radial-Roller-Bearing Calculation](#)  248
- [Gear Positions](#)  270

1.1 Installation

When running the installer, the installation directory can be selected. The default location is “\Program Files\MESYS 07-2019”. All files are installed into that directory. Also an entry in the start menu is created.

The uninstaller can be called from the start menu. This deletes the installation directory and the entries in the start menu.

Without a license file the software runs as demo version. In the demo version it is not possible to save or load files, a Demo message is shown on each calculation and some functionality is disabled. The demo version may only be used for evaluation of the software.

The license file ‘license.dat’ has to be placed in the installation directory (in the same directory as MesysShaft.exe). The name of the license file may not be changed since it will not be found by the software otherwise.

1.1.1 Configuration with INI-File

Some configuration of the software can be done using ‘mesys.ini’ in the installation folder.

Currently the settings are used for database access and formatting of the report.

Section	Value	Description
General	recentfilenumber	Number of recent files shown in the file menu of the software

Section	Value	Description
	floatinglicense	Path to the licensing file for floating license. It will be written by the software but might be copied to other installations. Format: \\Server\Share\path\license.lic Note: Each '\' has to be doubled.
	floatingtimeout	Time in seconds after a floating license is release if the program is not used. Default is 1800.
	fontsize	Fontsize in points, set 0 for default dependent on operating system.
	style	Either windows, fusion, windowsvista.
	listseparator	Character used as separator for table export. If not set the default setting in operating is used.
	systemlocale	Set to true for decimal point of operating system or false for '.' as decimal separator.
	usecalculatethread	If set to true calculation is done in a separate thread, else set to false.
	ffmpegpath	Path to the program ffmpeg to create videos from animations. It can found at www.ffmpeg.org For path separators either use '/' or '\\' but not '\.
	helpFormat	Use "CHM" to use "mesys.chm" to display the help information. This might not work over the network. Then "WEB" can be used to show the help in a web browser. A third option is "HTML" which would look for the help in subfolder "help", please contact support@mesys.ch in this case as the folder is not provided in the default installation.
	allownesteddock windows	If set to true, docked windows can be set in several columns. The default is false.
database	path	The path to the database file can be defined. The database can be copied onto a server, so all software users share the same database. If the filename is given without path, it is opened from the installation directory. For path separators either use '/' or '\\' but not '\.
	iswritable	Set it to true if the database may be changed. If set to false no changes are made to the database by the software.
	usecache	If set to true the database is read to memory. This speeds up the program in case the database is on a network drive. Default is false.
importdatabases	path_1	A path to additional databases can be defined. These additional databases are read only and optionally encrypted. If the filename is given without path, it is opened from the installation directory.

Section	Value	Description
		For path separators either use '/' or '\\' but not \.
	password_1	The password for the encrypted database
shaft	defaultinputs	Path to a xml-file with default settings For path separators either use '/' or '\\' but not \.
	calculateonfileload	If set to true the calculation is run when a file is loaded. Default is true.
	acceptfiledrop	If set to true a *.xml input file can be loaded by dropping it on the main window
	logo	A logo for use on the system page can be defined here. The format had to be PNG.
	numberofthreads	The number of threads that should be used in FEA calculations. The default is the number of physical cores.
	feamaxmemory	The maximum amount of memory in MB used by the fea solver. If more memory is required it will be swapped to disk.
	feaswappath	Path of a directory for the swapfile for FEA calculation in case not enough memory is available. Default is the temp directory of the operating system. Using this setting a fast drive with enough space can be used instead.
	feamaxtotalmemory	The maximum amount of memory in MB used by the fea solver. If more memory would be required by the solver, the calculation is canceled. Using this setting instead of 'feamaxmemory' avoids swapping to disk and therefore much longer calculation times. On error the mesh sizes would have to be changed.
rbc	defaultinputs	Path to a xml-file with default settings For path separators either use '/' or '\\' but not \.
	calculateonfileload	If set to true the calculation is run when a file is loaded. Default is true.
	acceptfiledrop	If set to true a *.xml input file can be loaded by dropping it on the main window.
	logo	A logo for use on the system page can be defined here. The format had to be PNG.
ballscrew	defaultinputs	Path to a xml-file with default settings For path separators either use '/' or '\\' but not \.
	calculateonfileload	If set to true the calculation is run when a file is loaded. Default is true.
	acceptfiledrop	If set to true a *.xml input file can be loaded by dropping it on the main window.

Section	Value	Description
	logo	A logo for use on the system page can be defined here. The format had to be PNG.
hertz	defaultinputs	Path to a xml-file with default settings For path separators either use '/' or '\\\' but not \'\'.
	calculateonfileload	If set to true the calculation is run when a file is loaded. Default is true.
	acceptfiledrop	If set to true a *.xml input file can be loaded by dropping it on the main window
	logo	A logo for use on the system page can be defined here. The format had to be PNG.
axrabc	numberofthreads	The number of threads that should be used in FEA calculations. The default is 2. More threads have only a small impact on performance, but the calculation time is still reduced.
report	format	The outputted report file can have different formats. By default, this value is equal to "INTERNALPDF", but it can be set to "INTERNALPDF", "DOCX", "DOC", "ODT" or "PDF" (without quotes).
	topmargin	The top margin for the report in mm
	bottommargin	The bottom margin for the report in mm
	leftmargin	The left margin for the report in mm
	rightmargin	The right margin for the report in mm
	papersize	The size of the paper for the report. Available values are A4 and Letter
	template	Path to the created template file used for the report creation. Supported file formats: "DOCX", "DOC" or "ODT"
	logo	A different logo can be defined, which will be used in the report. The format had to be PNG.
	marginbox1\active	The marginbox is used if set to true, else set it to false.
	marginbox1\rect	The size of the marginbox is defined with values in mm. The format is @Rect(x1 y1 width height). The parameters x1 and y1 describe the upper left corner of the box. Positive values are measured from the top/left negative values from the bottom/right. For example @Rect(-35 -20 30 20) is a rectangle at the right bottom.
	marginbox1\text	The text for the margin box. It has to be set in quotations marks (like in "Text"). Either normal text or HTML can be used. Some placeholders are defined: #page, #pageCount, #data, #datetime
	marginbox1\angle	A rotation angle of the margin box can be set in degrees. The orientation of the angle is clockwise

Section	Value	Description
		if positive.
	marginbox1\isHt ml	Either set it to true or false dependent on the type of text.
	marginbox1\draw Box	If set to true a rectangle is drawn indicating the size of the marginbox. Else set it to false.
	marginbox2\...	Like for margin box 1 additional boxes can be defined with increasing numbers.

Please note that the listed options for margin boxes (marginbox1\...) are only valid if format is equal to "INTERNALPDF".

1.1.2 Update

If the software is updated with a new version the database 'mesys.db' should not be overwritten. Either the new installation is done in a new directory or the database file is copied to a different location.

After the new version is started the database can be updated by choosing menu 'Extras'->'Database'->'Import from old database'. All custom entries will be updated. Changes to default data will be lost as only custom data will be copied.

For updates of a floating license in addition to the 'license.dat' the setting 'floatinglicense=' in 'mesys.ini' needs to be copied into the new version.


1.2 Requirements

The software is available as 32bit and 64bit windows program running on Windows Vista, Windows 7, 8 or 10. The 32bit version can be used on 32bit or 64bit operating systems, the 64bit version only runs on 64bit operating systems. For applications with large memory consumption, the 64bit version should be preferred. The 64bit version is also slightly faster. The minimum required processor is Intel Pentium 4 or above.




About 400 MB of hard disk space is required. All dependencies of the software are available in the installation directory. Therefore, it can just be copied to other machines or started from network or removable disks.

For the floating license a directory on a network share with write permissions is required.




1.3 General usage

To run a calculation first the data on all pages is introduced. Then press the button  or F5 to run the calculation. After all data is defined the calculation can be run from each page. So it is easy to make parameter variations.

A symbol is shown the the right of the status bar at the bottom of the program. This shows the current status of the results:

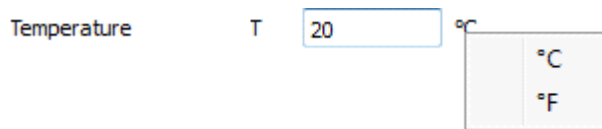
Symbol	Explanation
	The calculation was run and the results are valid.
	Some inputs were changed after the calculation was run. The results might be invalid, dependent on the change of inputs.
	There was an error on running the calculation. The results are invalid.

There are some special buttons used in the user interface, which are explained in the following table:

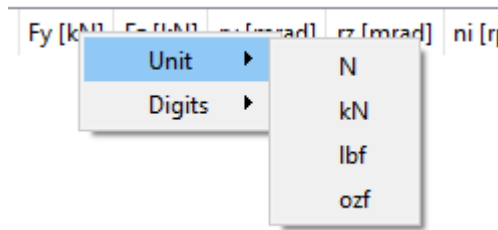
Button	Explanation
	This plus button shows a dialog with additional inputs. Some of these inputs need to be defined, some are just optional.
	This conversion button allows the conversion from other types of input. For example, the radial clearance can be converted from an axial clearance.
	This proposal button provides a suggestion for an input by the software.

The unit system for the input and output can be selected on the menu 'Extras->Unit system' either as metric or US units.

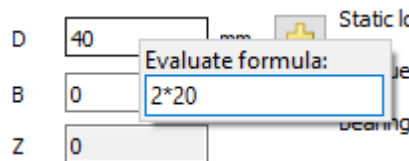
Using the context menu for the units of input fields, the current unit can be changed. This setting is saved for future restarts of the software.



For tables, the unit of columns can be changed using the context menu (right mouse button) on the header row. In addition to the unit, the number of visible digits can be changed.



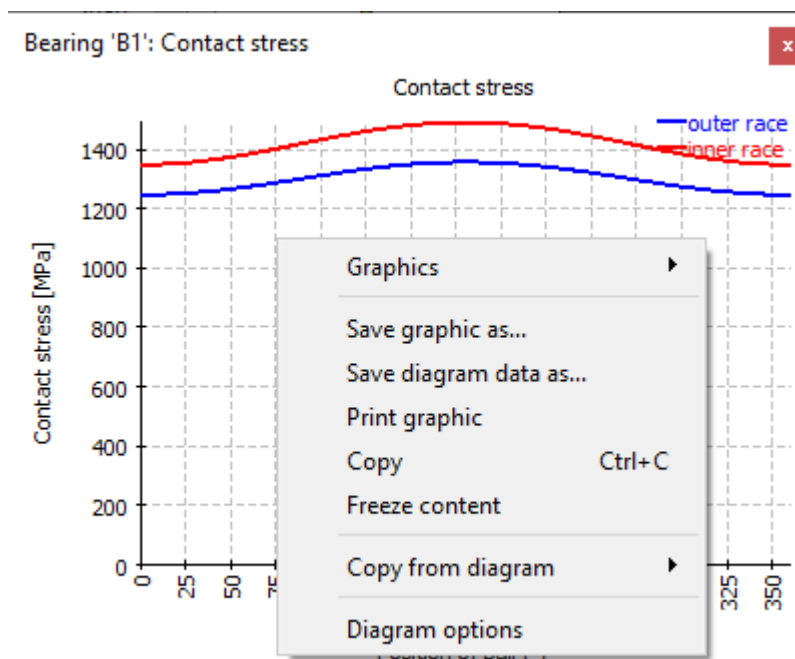
Pressing the right mouse button on an input field, a window for an input of a formula is shown. This can be used for quick calculations.



The software is available in English, German, French, Spanish, Chinese, Korean and Japanese language. The language can also be selected in menu 'Extras'.

1.3.1 Graphics 2D and Diagrams

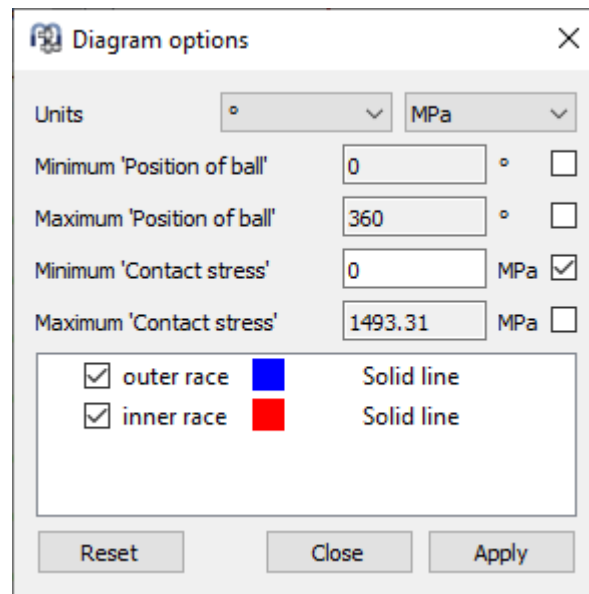
For 2D graphics and diagrams several options are available in the context menu.



All graphics can be printed or exported in different graphic formats like PNG or JPEG using the context menu (right mouse button) in the graphic window. For the export the size of the graphic can be specified. The option "Copy" copies the image to the clipboard, so it can be pasted into other applications. Selecting "Freeze content" suppresses updates of the graphic on further runs of the calculation, this allows a comparison of graphics between calculations with different input data.

For diagrams "Save diagram data as...", "Copy from diagram" and "Diagram options" are available in addition. "Save diagram data as..." allows to save the numeric values for the diagrams in CSV or XLS formats. Using "Copy from diagram" curves from other open diagrams can be copied into the current diagram if they have the same units. This can be used for comparisons.

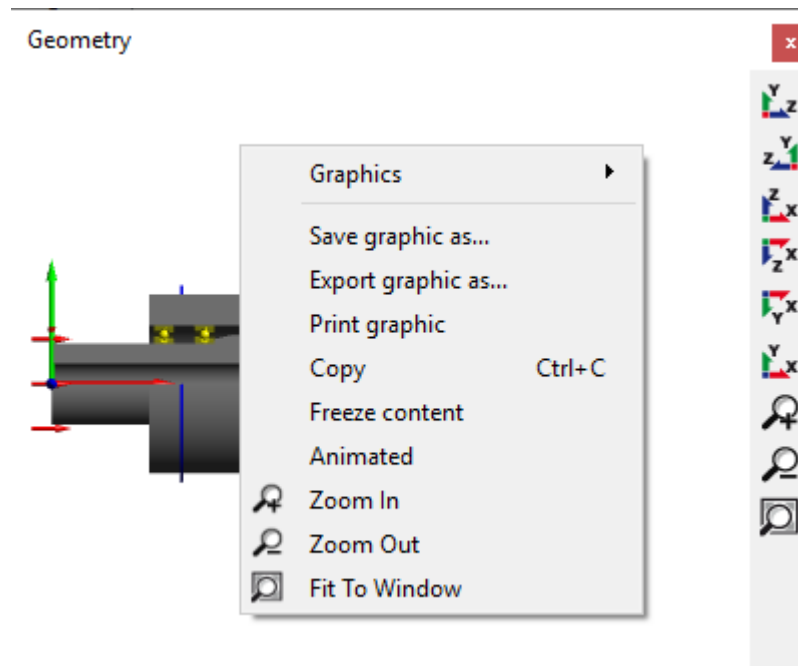
The diagram option allow to change the formatting of the diagram



Both the units and bounds for the axes can be set. Moreover, if any of the graphs is of no interest, it can be disabled with the checkboxes. The name, the color and line style of each curve can be changed too. The changes in diagram options are only used for the current diagram, they are not saved and not used for newly created graphic windows.

1.3.2 Graphics 3D

For 3D graphic windows of the software, there are different buttons for the view manipulation in 3D, such as the zoom-in, zoom-out and fit-to-window functions and also it is possible to select the point of view from different planes. The 3D-model can be dragged with the mouse by holding down Shift-key as well as zoomed in and out by holding down the Ctrl-key. A 3D-mouse is supported in 3D-graphics. Animated 3D-graphics can be saved as video in case a path to ffmpeg is provided in mesys.ini¹⁴.



"Copy" will copy the image to the clipboard, so it can be pasted in other programs.

Activating "Freeze content" will suppress updated of the view on new runs of the calculations. This can be used for comparisons.

The selection "Animated" will rotate the view around the vertical axis.

1.4 Command line parameters

The software supports several command line parameters:

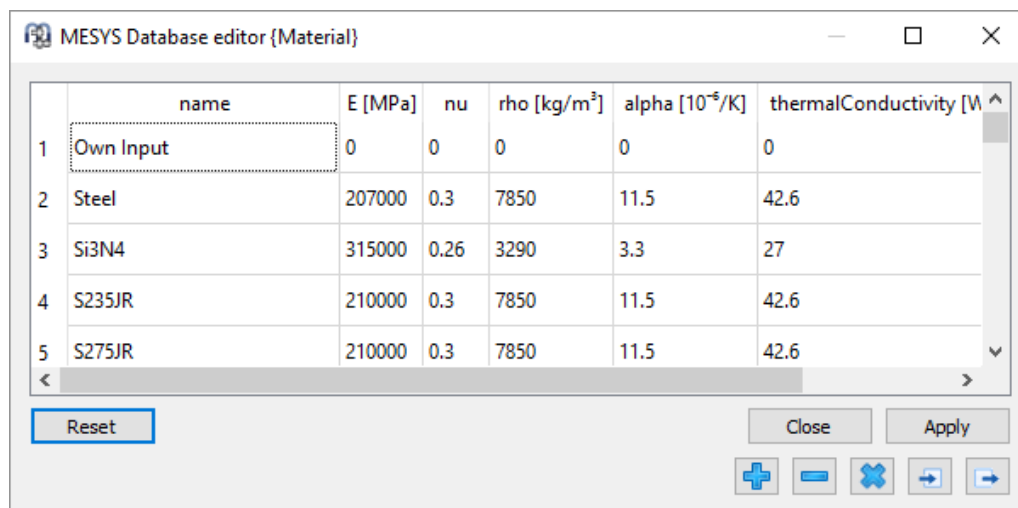
- -disableHighDPI disables scaling and tells the operating system to do the scaling. This is currently the default setting.
- -enableHighDPI enables highDPI scaling by the software. This setting still has some problems but it might be useful on some systems.
- -desktopOpenGL tells the software to use hardware OpenGL, which is the default.
- -OpenGL ES tells the software to translate OpenGL into DirectX. This can be used if the driver for the graphic card does not work correctly and hardware OpenGL does not work.
- -softwareOpenGL tells the software to use a software driver for 3D-graphics. This can be used if the two settings above fail to work.
- -ini=file.ini tells the software to use 'file.ini' for program settings.

- -license=license.dat tells the software to use 'license.dat' as license file. This can be used if different license files are available.
- If a *.xml filename is passed as a parameter, the file is opened by the software. This also allows to drop an input file on a program icon on the desktop.

1.5 Database

As database a SQLite database is used. In addition to the possibilities of the software also standard tools could be used to change the database. Please only add datasets with id > 100000 and don't modify datasets with id < 0 since they are used internally. On future versions datasets with id < 100000 might be changed or added.

In the current version there are database tables for Bearings, Bearing Clearance, Bearing Tolerance, Material, Material Bearings and Lubrication. They can be added using the Menu 'Database' under 'Extras'.



Data is written into the database only when clicking the Apply-button, changes can be removed by selecting 'Reset' before 'Apply' is clicked.

Rows can be added using the + button or a selected row can be deleted using the - button. By using the import button, additional items can be imported from a file. The columns in the file have to be in the same order and unit as shown in the table. Likewise, the table can be exported with the corresponding export button.

1.5.1 Changing names in the database

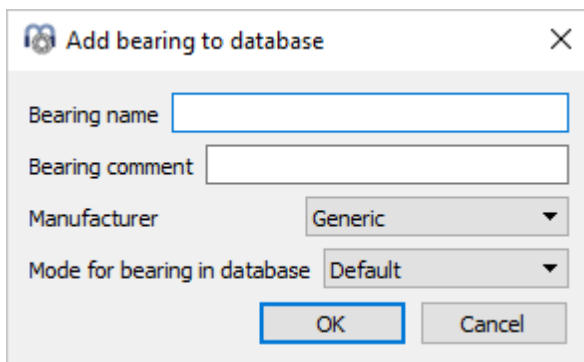
When a calculation file is loaded, the software searches the database for combinations of id and name of a dataset. If the item is not found the inputs are changed to 'own input'.

Therefore, calculations files that are opened by other installations of the software with a different database, will show 'own input' and give the correct results.

If you change the name of entries in the database the same will happen. You will see 'own input' in old files that used this entry and they will use the original data.

1.5.2 Add bearings to database

For adding bearings to the database please enter the geometry data into the calculation windows of the bearing calculation, run the calculation and then select 'Extras->Database->Add bearing to database'. You will be prompted for bearing name, comment, manufacturer and a database mode.



If the combination of name and manufacturer is already found in the database you will be asked if the dataset should be overwritten.

The mode can be

- "Default": Default behavior
- "Approximated": A comment about approximated inner geometry will be shown in the report and damage frequencies will not be shown.
- "Hide inner geometry": Inner geometry will not be available in the report and in input data. This option is for additional encrypted databases.

In case of the shaft calculation the above dialog is shown if 'Extras->Database->Add bearing to database' is selected while the bearing window is active. Else the following dialog is shown:

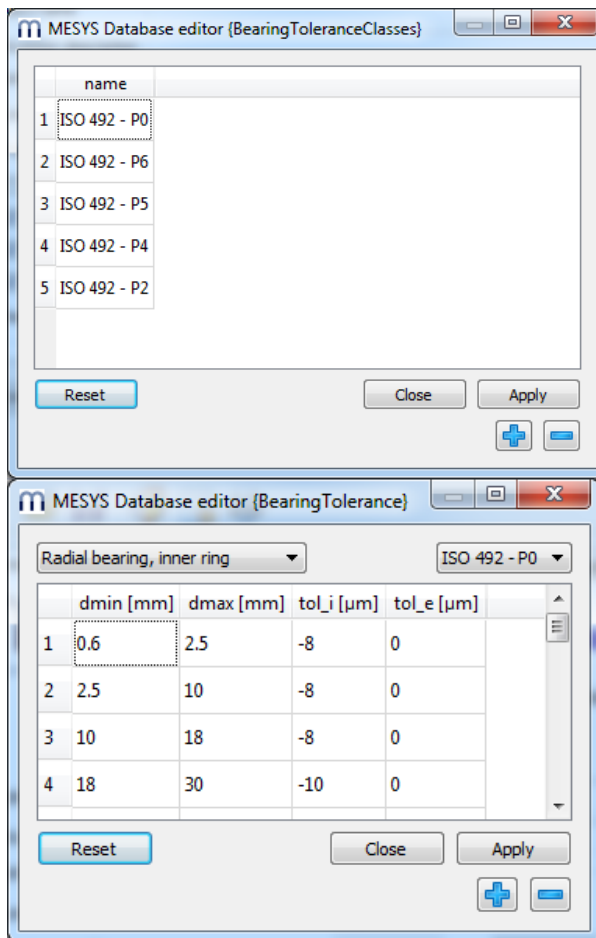
A window with two tab-pages named “Bearing geometry” and “Material and Lubrication” is opened in the shaft calculation. These tab-pages operate in the same way as the ones in the bearing calculation interface. The feature offers the possibility to create and add a new bearing from scratch by entering all the required data or by selecting and modifying an existing bearing of the database. In both cases, the user does not need to know the load capacity data in advance, since the software finds it out from the required geometrical inputs when pressing the button “Calculate” at the bottom. Once all the data is filled, the “Apply” button must be pressed and the user will be prompted to enter a name for the new bearing.

1.5.3 Database Tables

For some database tables more information about usage is given in the following sections.

1.5.3.1 Bearing Clearance and Bearing Tolerance

For clearance and tolerance definitions two tables have to be used. In table ‘BearingClearanceClasses’ or ‘BearingToleranceClasses’ the available names are defined.



In table 'BearingClearance' and 'BearingTolerance' then the actual values are defined dependent on the diameter.

1.5.3.2 Bearing Materials

The table 'Material Bearings' allows to set additional information about permissible stresses for bearing calculations.

1.5.3.3 Bearing Manufacturers

The tables 'Bearing Manufacturers' allows to set additional names for bearing manufacturers.

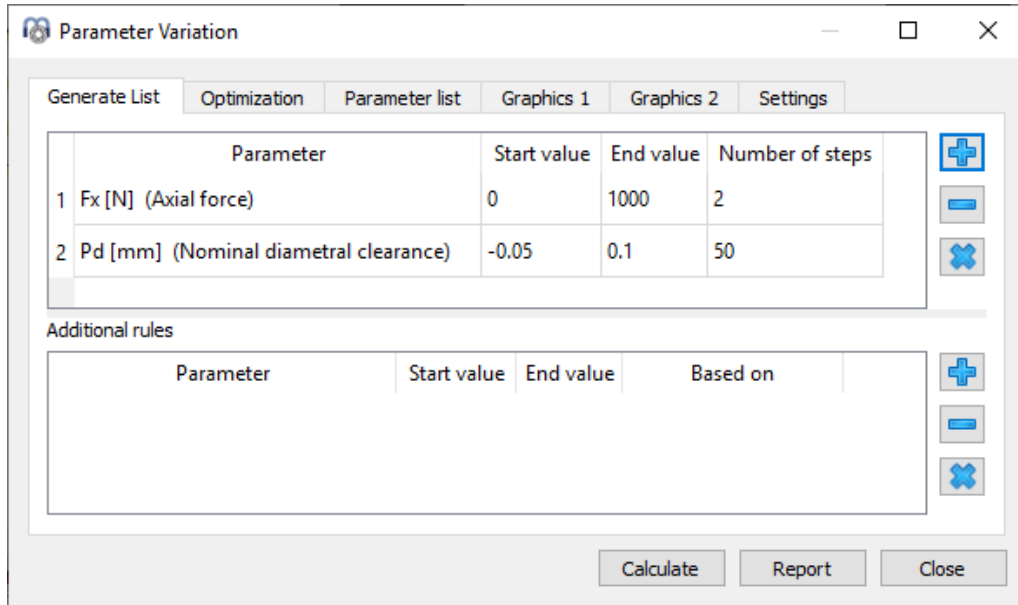
1.6 Parameter variations

Using the menu point "Calculation" -> "Parameter variation" a dialog for parameter variations is shown. It allows the user to do parameter studies with results provided in tables and graphics. Typical applications are, for example, visualizing

life over clearance or displacement over load. An optional optimization for a parameter is available too.

1.6.1 Generate list

Several input parameters can be added to the table “Generate list” and ranges (Start/End values) can be defined for them. In the bearing calculation dependent on the bearing type the calculation can take up to 0.1 sec for a single calculation, using load spectra and defining many steps can lead to long calculation times.



Additional rules can be used to set additional input parameters dependent on parameters on the first list. The additional parameter is interpolated between a start and end value based on another parameter. For example additional parameter y between y1 and y2 based on parameter x in range x1 to x2:

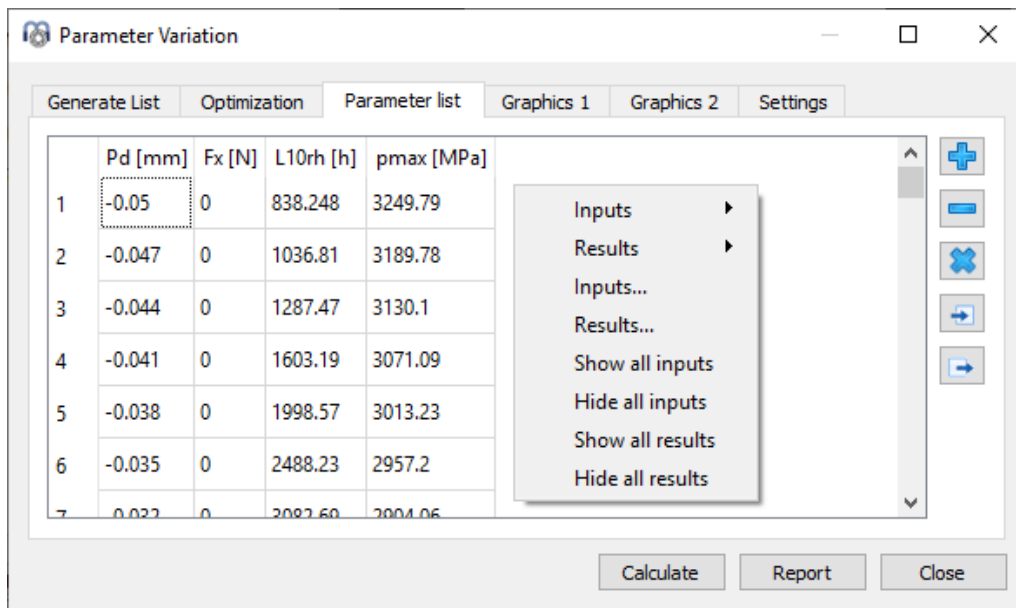
$$y = y_1 + \frac{y_2 - y_1}{x_2 - x_1} (x - x_1)$$

An example for using the rules can be to set the bearing clearances for multiple bearings to the same value within the shaft calculation.

1.6.2 Parameter list

Before we generate a list, it is necessary to define first those outputs or variables we are interested in at the tab window ‘Parameter list’.

As shown in the picture, by doing a right mouse-click on the tab window ‘Parameter list’, a context menu will enable us to add the columns corresponding to the outputs or results we are interested in. The rows will contain the variable values according to the bounds and number of steps.



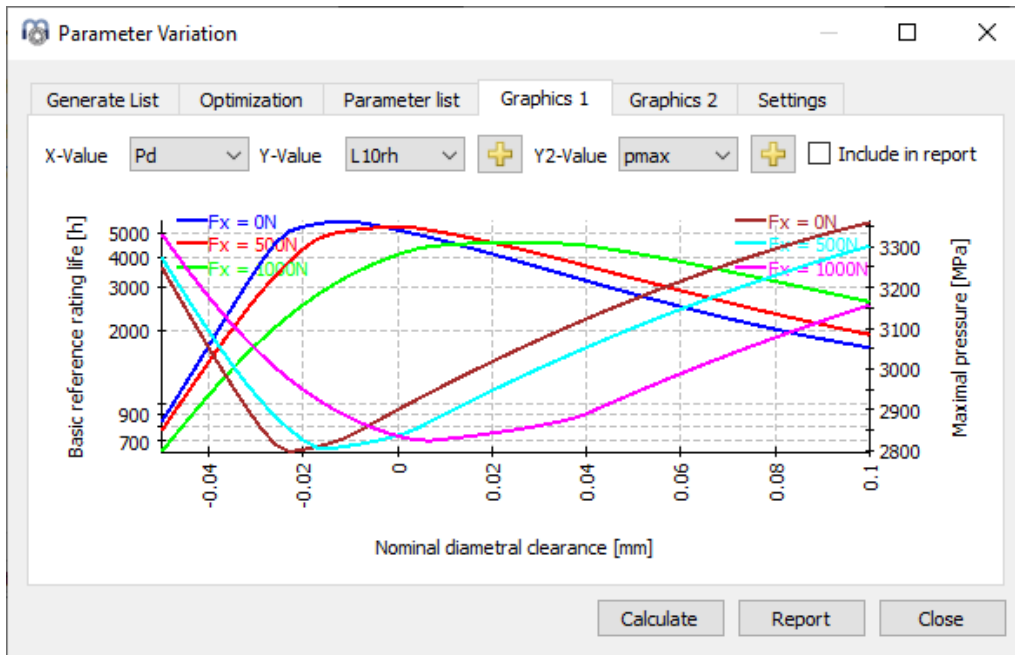
By clicking on 'Calculate' while the "Generate list" tab window is active, a list of all parameter combinations will be generated at the corresponding tab window 'Parameter list' and the analysis will be run. Depending on the chosen number of parameters and steps, the analysis can lead to long calculation times.

A second way to create a table (parameter list) is possible by adding rows and entering values manually with the -button. Moreover, a whole table can be imported from a CSV- or XLS-file using the -button or exported into a file using the -button. Then, in order to run the analysis, we just have to click on "Calculate" while this page is active. Please note that the parameter list is not saved if the dialog is closed.

1.6.3 Graphics

On the pages "Graphics1" and "Graphics2" the results can be shown as graphic.

If the parameter list was generated by "Generate list" and no more than two input parameters are used, lines are used for the diagram. For more parameters or other sources just points are shown.

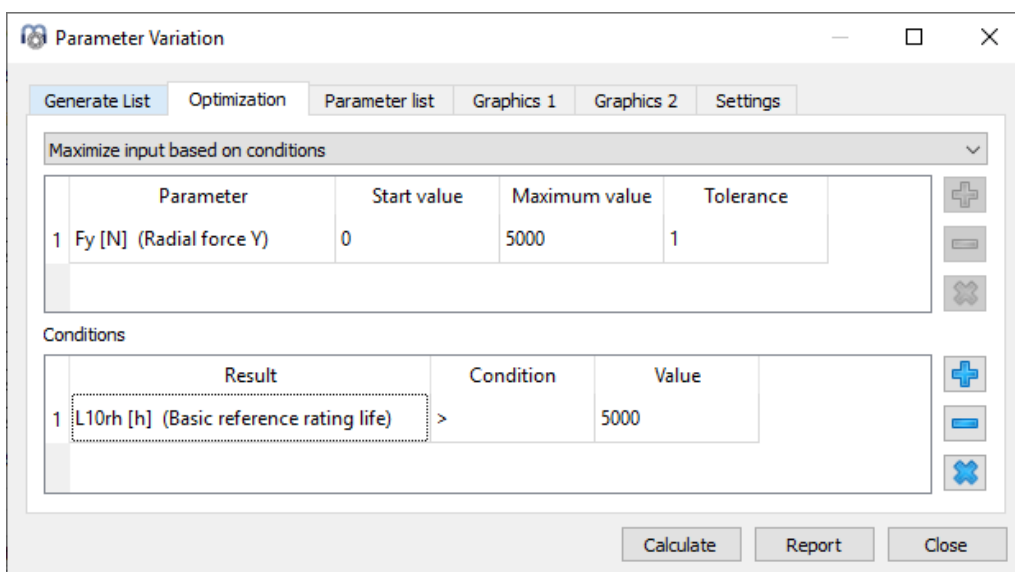


In case two parameters had been varied on "Generate list" multiple curves for the second parameter are shown like in above image showing a variation of nominal clearance and axial force F_x .

The graphic area consists of two different y-axis, thus being possible to compare different types of output results (units) at a time. Furthermore, although the basic life (L10rh) is being primarily analyzed in the graphic, by doing a click on the \oplus -button, it could also be possible to add any other outputs with the same units like Lnmrh of one of the two available y-axis.

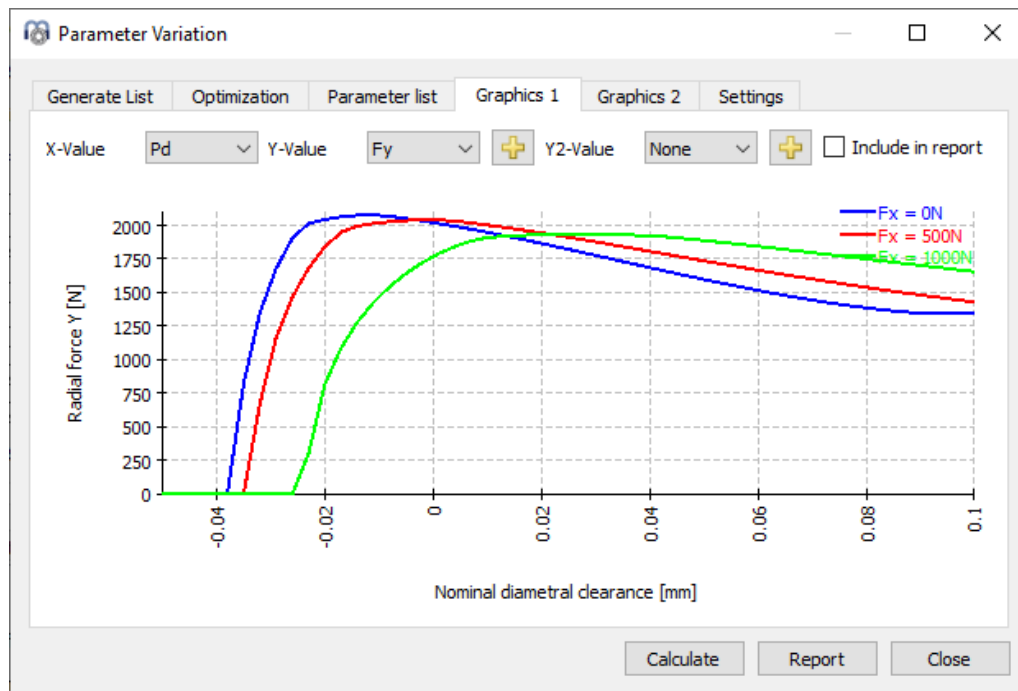
1.6.4 Optimization

Optionally an optimization step can be used. For the example before the radial force F_y should be maximized to that L10rh=5000h is reached.



The conditions have to be fulfilled on the start value of the parameter. If this is not the case, the results are set to zero. If the conditions are still fulfilled at the maximum value, the calculation stops and the results are set to zero too.

The results for F_y in below image show an error for small clearance as the 5000h cannot be reached.



1.6.5 Report

You have to activate the checkbox “Include in report” if the graphic should be added to the report. Moreover, under the tab-page ‘Settings’ another checkbox is available for including the parameter list in the report.

1.7 Statistical parameter variations

Using the menu point “Calculation”->“Parameter variation (statistical)” an analogous tool to the default parameter variation is available. The main difference between them lies in the fact that this tool supports statistical evaluations when doing parameter variations.

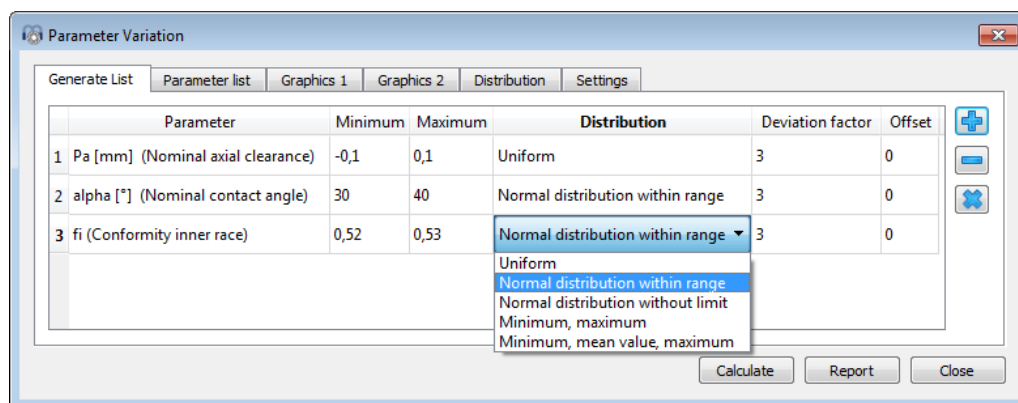
The default version of parameter variation is helpful to visualize the influence of one parameter on one or multiple results. The statistical version can be used if the influence of multiple uncertainties should be evaluated. Instead of manually checking multiple parameters an automatic calculation of the whole ranges can be done. In order to understand the reasons behind the results it will still be

necessary to carry out additional evaluations, but the software shows the ranges of results that have to be expected.

1.7.1 Generate list

As in the normal parameter variation, several input parameters can be added to the table “Generate list” and ranges (Start/End values) can be defined for them. No calculations steps are required for the ranges. However, a limitation on the number of statistical calculations can be set under the tab-page ‘Settings’.

Different types of probability distributions can be chosen from a drop-down list which is shown when double-clicking on a cell under the ‘Distribution’ column for any of the parameters:



- Uniform: The probability is equally distributed for a set of values between the maximum and minimum value of the parameter.
- Normal distribution within range: The probability is calculated for a set of values ranging from the minimum to maximum by using a normal distribution.
- Normal distribution without limit: The probability is calculated for a set of values theoretically in the range of $(-\infty, +\infty)$ by using a normal distribution.
- Minimum, maximum: The probability is equally distributed between the maximum and minimum values of the parameter.
- Minimum, mean value, maximum: The probability is equally distributed between the maximum, mean and minimum values of the parameter.

Additionally, a factor can be set for the deviation and an offset value between -1 and +1 can be used to shift horizontally the ‘bell curve’ in either direction.

1.7.2 Parameter list

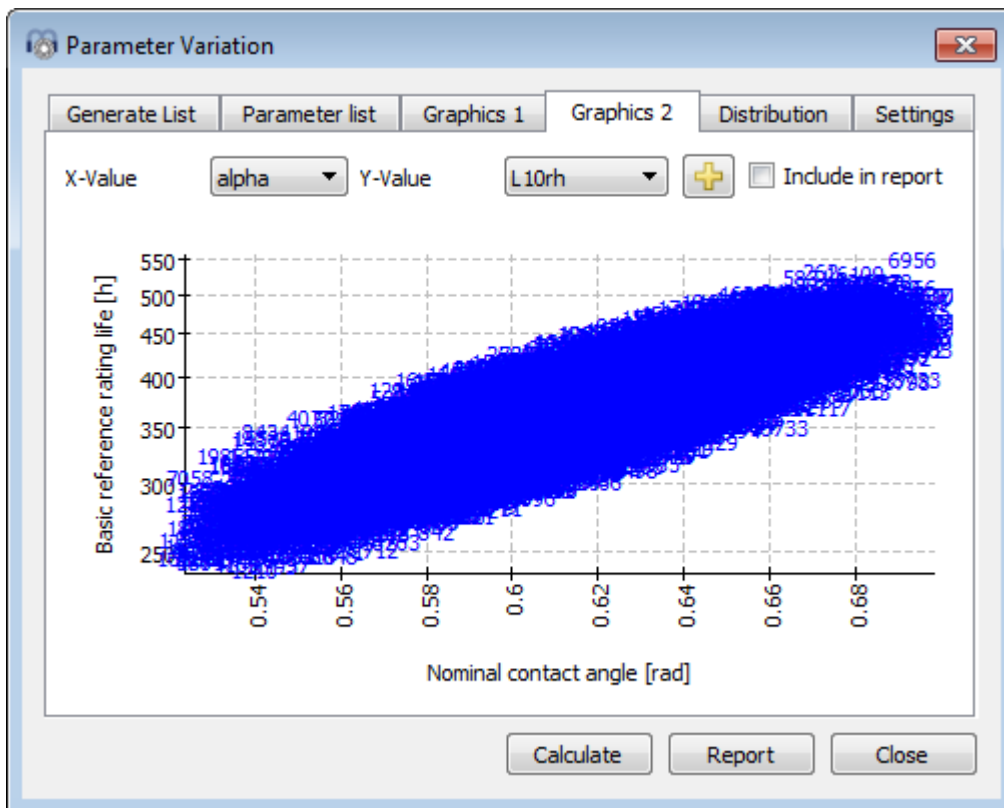
The generation of the parameter list is performed as explained for the non-statistical parameter variation. By clicking on ‘Calculate’ while the “Generate list” tab window is active, a list of all parameter combinations will be generated at the

corresponding tab window 'Parameter list' and the statistical analysis will be run. Depending on the chosen number of parameters and the specified number of calculations, the analysis can lead to long calculation times.


	fi	alpha [rad]	Pa [mm]	L10rh [h]	alpha_eavg [rad]	pmax [MPa]
1	0,524367	0,581674	0,0231761	338,497	0,611126	1798,21
2	0,523759	0,600977	0,00844711	369,786	0,628787	1768,38
3	0,528598	0,617766	-0,00454472	328,565	0,640688	1823,6
4	0,524695	0,64824	0,0417903	422,499	0,669298	1759,09

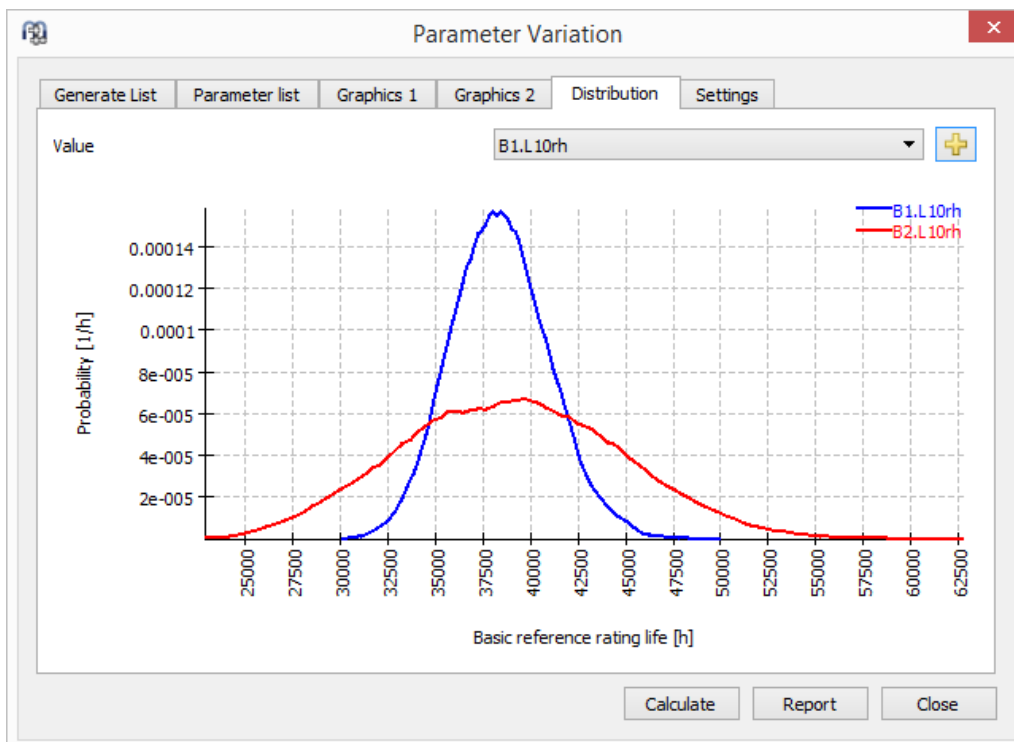
1.7.3 Graphics

On the tab-pages 'Graphics' any variable combination can be visualized in the form of point clouds, where some trends for the relationship between the parameters might be recognized, as shown in the picture below.



1.7.4 Distribution

The probability distribution is graphically represented at the tab-page 'Distribution'. By simply using the -button, it is possible to make comparisons of interest between the statistical results for different mechanical components in a system. As shown in the picture, the probability distribution of basic reference life for the different bearings mounted on the same shaft are compared with each other. The maxima of the curves are close to each other, but the variance for the roller bearing 'B2' is much larger. These analyses are useful to help identify restrictions or weaknesses in the design.



1.7.5 Report

You have to activate the checkbox "Include in report" if the graphic should be added to the report. The report will also show minimum, maximum, mean value and standard deviation for each selected result.

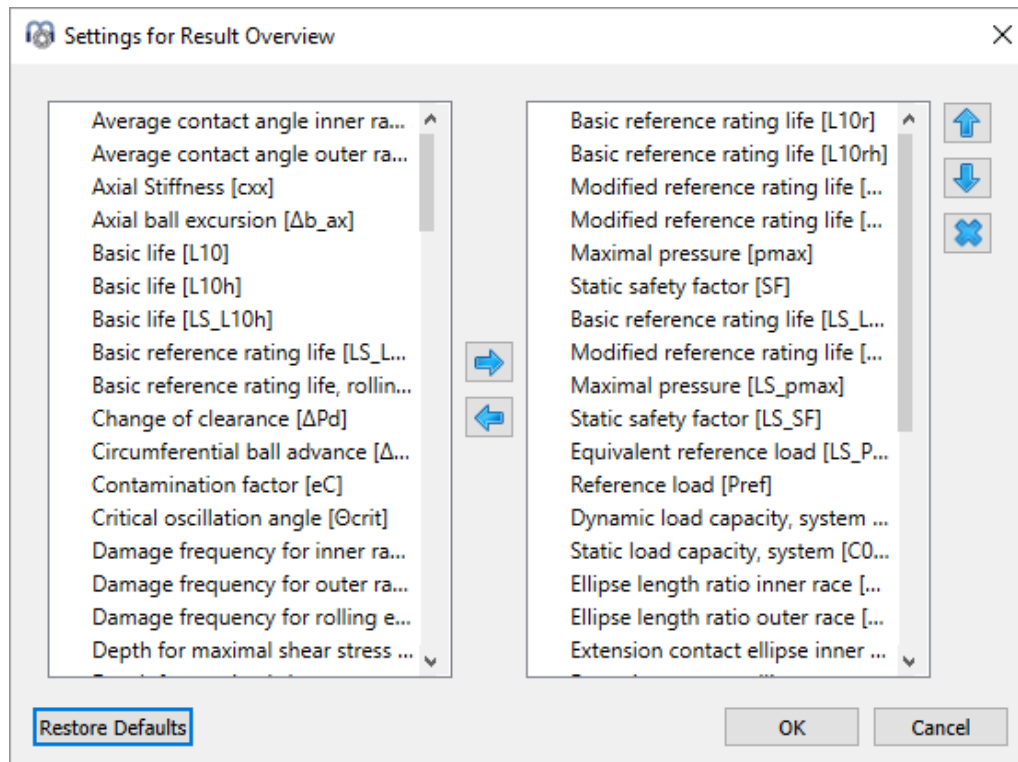
1.8 Results

Results are provided as a small result overview directly in the software, a main text report in PDF format as standard and optional additional reports dependent on the calculation module. Under Report->'Result tables' a spread sheet with result data can be opened in XLSX format as default.

Graphic windows can be docked onto the main program interface and are automatically updated on each calculation.

1.8.1 Results Overview

The results overview on the bottom of the window shows the most important results. The results overview can be configured under Extras->Results overview.



The list on the left shows all available items for the result overview. Using the arrows selected items can be moved to the list on the right which is used for the result overview. The order of the selected results can be changed by the up/down arrows on the right. Note that only valid results values are actually shown in the result overview. So you will see fewer values than selected in the settings.

The settings for the result overview are saved in user settings not in the calculation file. So they will be the same for different calculations.

1.8.2 Main Report

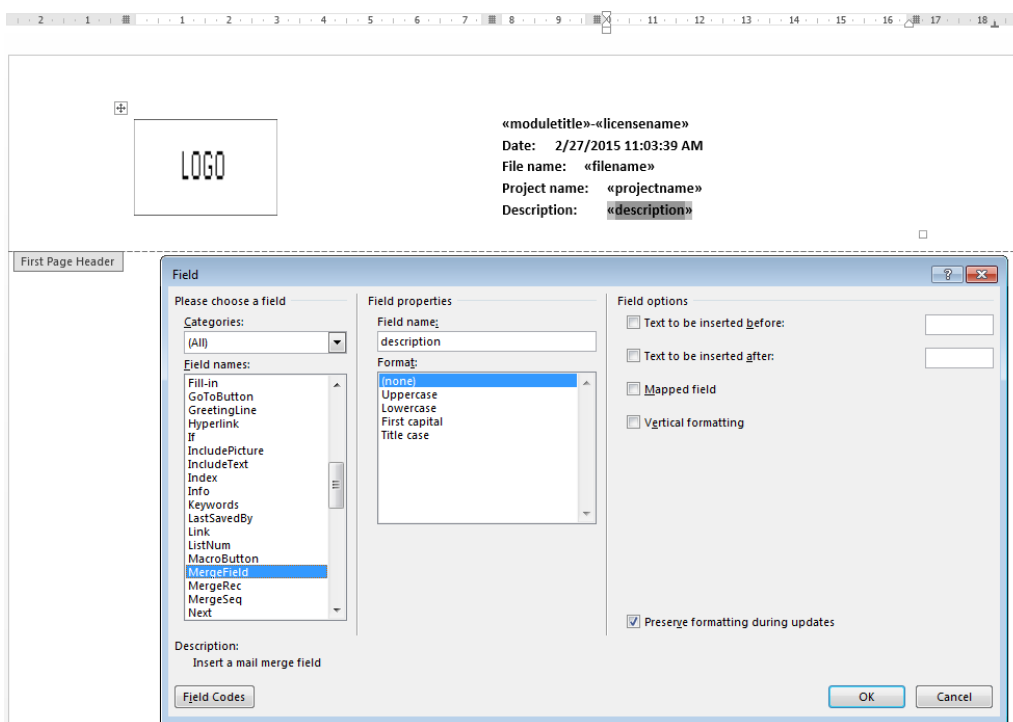
The main report can be generated selecting Report->'Show report'. It can be saved in different formats using Report->'Save report as'.

In the menu Report->Options the contents of the report can be configured. The graphics to be included can be selected and some sections of the report could be discarded if not of interest. The legend for all the table parameters can be shown in the report.

The logo in the report can be configured in 'mesys.ini'. See [Configuration with INI-File](#)

1.8.2.1 Template

A template in (.docx) format can be created and defined in 'mesys.ini' so that it is used when generating the software report. For the current version, only the information contained in the header and footer can be edited, in which it is possible to link information to the software such as 'module title', 'license name', 'date', 'file name', 'project name' or 'description' by means of the text fields option in Microsoft word. Additionally, a company logo can be included, instead of the MESYS logo that it is shown by default at the report:



Available field for the 'Mergefield' option are:

- filename
- projectname
- description
- licensename
- moduletitle

1.8.3 Custom Report Templates

Custom report templates can be added in subdirectory "templates/module-id". Reports based on these templates can be generated by selecting the report template under Report->Report templates.

The report templates are word files with DOCVARIABLE fields. These fields are filled on generation of the report.

Possible fields are:

- DOCVARIABLE VAR name [unit]: the variable with the name 'name' is printed. Optionally a unit can be provided in brackets, for example [mm]. Additionally, a format could be added like %6.3g or %6.3f, where 'f' always generates a fixed point output and 'g' generates an exponential output for large numbers.
- DOCVARIABLE VARU name [unit]: the variable with the name 'name' plus the unit are printed. Optionally a unit can be provided in brackets, for example [mm].
- DOCVARIABLE TABLE tablename: A table is inserted.
- DOCVARIABLE GRAPHIC graphicName width=130 height=70: A graphic is inserted. Optionally height and width can be given in millimeters.

Regarding details for variable names, table names and graphic names please ask support@mesys.ch for additional documentation.

1.8.4 Result tables

In the menu Report->Result tables, it is possible to output the result data by means of tables, which can be generated in different file formats as explained before at [Configuration with INI-File](#)^[14]

1.9 COM Interface

The software can be accessed via a COM-Interface under Windows. The installation of the COM-Interface is optional and it can be selected during installation. Before the first use the COM-server has to be registered.

1.9.1 Registering the COM server

The COM server can be registered by running MesysCOM(64).exe with parameter -regserver as administrator. This has to be done once.

For unregistering the parameter -unregserver can be used. In the installation directory two batch files RegisterMesysCOM and UnRegisterMesysCOM are provided so that the registration can be done without a comment window.

1.9.2 Using the COM server

The COM server can be used to call functions within the MESYS software from other programs.

A simple example in a VisualBasic function:

```
Public Sub test()
Dim mesys As MesysCOM
Set mesys = New MesysCOM
Dim rbc As MesysRBC

Set rbc = mesys.createRBC
Dim d As Long
Dim pmax As Double

Call rbc.setVarInt("inputType", 2)
Call rbc.setVarInt("Z", 12)
Call rbc.setVarDouble("Dw", 5)
Call rbc.setVarDouble("Dpw", 50)
Call rbc.setVarDouble("Fy", 1000)
Call rbc.setVarDouble("ni", 500)

result = rbc.Calculate
Call rbc.showReport("c:/temp/report.pdf")
Call rbc.getVarDouble("pmax", pmax)
Call rbc.getVarInt("Z", d)
Call rbc.calculateBearing(True, 0, True, 1000, True, 0, False, 0, False, 0, 500, 0, 20, 20)
Call rbc.getVarDouble("pmax", pmax2)
Set rbc = Nothing
Set mesys = Nothing

End Sub
```

The server provides the interface MesysCOM which can be used to create calculation modules. Available methods are:

- `void setLanguage(QString lang);`
Pass language as two characters ("de", "en", "fr", "es", "ko", "zh", "ja")
- `int getVersion();`
Returns version in format yyyyymm.
- `MesysRBC* createRBC();`
Create bearing calculation module without user interface.
- `MesysSHAFT* createSHAFT();`
Create shaft calculation module without user interface.
- `MesysHERTZ* createHERTZ();`
Create Hertz stress calculation module without user interface.
- `MesysBALLSCREW* createBALLSCREW();`
Create ball screw calculation module without user interface.
- `MesysGUIRBC* createGUIRBC();`
Create bearing calculation module with user interface.
- `MesysGUIHERTZ* createGUIHERTZ();`
Create hertz stress calculation module with user interface.
- `MesysGUISHAFT* createGUISHAFT();`
Create shaft calculation module with user interface.
- `MesysGUIBALLSCREW* createGUIBALLSCREW();`
Create ball screw calculation module with user interface.

The interfaces MesysRBC, MesysSHAFT and others without GUI create calculation modules without user interface. MesysGUIRBC, MesysGUISHAFT are calculation modules with user interface.

General methods for all calculation modules are:

- `bool calculate();`
Run calculation.
- `bool loadFile(QString filename);`
Load file.
- `bool saveFile(QString filename);`
Save calculation file.
- `bool generateReport(QString filename);`
Generate a report into the filename.
- `bool showReport(QString filename);`
Generate the report into the filename and call a program to view it.
- `bool generateSpecialReport(QString type, QString filename);`
Generate a special report into the filename. Use type as "resultTables" for result tables.
- `bool generateImage(QString imageID, QString filename, int dpi, int widthinMM, int heightinMM);`
Generate an image and save it as file.
- `bool setVarDouble(QString name, double value);`
Set variable to double value.
- `bool setVarBool(QString name, bool value);`
Set variable to boolean value.
- `bool setVarInt(QString name, int value);`
Set variable to integer value.
- `bool setVarString(QString name, QString value);`
Set variable to character value.
- `bool getVarDouble(QString name, double &value);`
Read back double variable.
- `bool getVarBool(QString name, bool &value);`
Read back boolean variable.
- `bool getVarInt(QString name, int &value);`
Read back integer variable.
- `bool getVarString(QString name, QString &value);`
Read back character variable.

Additional methods for bearing calculation:

- `bool calculateBearing(bool enterFx, double Fx_or_Ux, bool enterFy, double Fy_or_Uy, bool enterFz, double Fz_or_Uz, bool enterMy, double My_or_Ry, bool enterMz, double Mz_or_Rz, double ni, double ne, double Ti, double Te);`
Run bearing calculation with given loading.
- `bool setupLoadSpectrum(int count, bool enterFx, bool enterFy, bool enterFz, bool enterMy, bool enterMz);`
Setup load spectrum definition.
- `bool setLoadSpectrumLoad(int loadCase, double frequency, double Fx_or_Ux, double Fy_or_Uy, double Fz_or_Uz, double My_or_Ry, double Mz_or_Rz, double ni, double ne, double Ti, double Te, double T0il);`
Set load spectrum load for each load case
- `bool selectBearingFromDatabase(QString manufacturer, QString bearingName);`
Selects a bearing based on name and manufacturer from the database
- `bool getStiffnessMatrix(double& c11, double& c12, double& c13, double& c14, double& c15, double& c21, double& c22, double& c23, double& c24, double& c25, double& c31, double& c32, double& c33, double& c34, double& c35, double& c41, double& c42, double& c43, double& c44, double& c45, double& c51, double& c52, double& c53, double& c54, double& c55);`

Reads back the bearing stiffness matrix with units N, Nm, mm, rad. The order of rows and columns is like in the report ux, uy, uz, ry, rz.

Additional methods for ball screw calculation:

- `bool calculateBallScrew(bool enterFx, double Fx_or_Ux, bool enterFy, double Fy_or_Uy, bool enterFz, double Fz_or_Uz, double Rx, bool enterMy, double My_or_Ry, bool enterMz, double Mz_or_Rz, double ni, double ne, double Ti, double Te);`
Run ball screw calculation with given loading
- `bool setupLoadSpectrum(int count, bool enterFx, bool enterFy, bool enterFz, bool enterMy, bool enterMz);`
Setup load spectrum definition
- `bool setLoadSpectrumLoad(int loadCase, double frequency, double Fx_or_Ux, double Fy_or_Uy, double Fz_or_Uz, double Rx, double My_or_Ry, double Mz_or_Rz, double ni, double ne, double Ti, double Te, double TOil);`
Sets load spectrum load for each load case
- `bool getStiffnessMatrix(double& c11, double& c12, double& c13, double& c14, double& c15, double& c21, double& c22, double& c23, double& c24, double& c25, double& c31, double& c32, double& c33, double& c34, double& c35, double& c41, double& c42, double& c43, double& c44, double& c45, double& c51, double& c52, double& c53, double& c54, double& c55);`
Reads back the stiffness matrix with units N, Nm, mm, rad. The order of rows and columns is like in the report ux, uy, uz, ry, rz.

Each calculation module with GUI provides following methods:

- `void showWindow();`
Show user interface.
- `void hideWindow();`
Hide user interface.
- `void exec();`
Show user interface and process events until the window is closed.
- `void setLanguage(QString lang);`
Pass language as two characters ("de", "en", "fr", "es", "ko", "zh")
- `MesysModule* getCalcModule();`
Get the calculation module from the user interface.
- `void setCalcModule(MesysModule*);`
Set the calculation module for the user interface.

The names for variables of the calculation modules are the same as provided for the [custom report templates](#)^[33]. Currently the use for the shaft calculation is limited as additional functions for changing loads and supports are missing.

MESYS Shaft Calculation

2 MESYS Shaft Calculation

The shaft calculation (Version 07/2019) calculates the deflections, internal forces and the natural frequencies of several shafts connected by boundary conditions. The following features are supported:

- Definition of multiple coaxial shafts is possible
- Definition of parallel shafts (with extension for shaft systems)
- Definition of shafts in arbitrary direction (with extension for advanced shaft systems)
- Shaft geometry is defined by cylindrical and conical elements
- Inner and outer geometry can be defined independently
- Shear deformation can be considered optionally
- A nonlinear shaft model can be used optionally
- The weight of the shaft can be considered optionally
- An arbitrary number of loads can be defined on each shaft either as point or line loads. Loads (without masses) may be defined outside of the shaft geometry also. Available loading elements are:
 - Force element with three components each for force and moment
 - Eccentric force element with three force components in polar coordinates
 - Helical Gear element using gear data and torque as input
 - Bevel/Hypoid Gear element using gear data and torque as input
 - Worm/Worm Gear element using gear data and torque as input
 - Coupling element for entering a torque only
 - Pulley element for torque and pretension
 - Mass elements for introducing weight and inertia for natural frequencies
 - Imbalance and dynamic force as periodic loads for harmonic response
- An acceleration can be defined for the system and is considered as inertia force
- An arbitrary number of boundary constraints can either be defined between a shaft and a rigid housing or between two coaxial shafts.

- Several housing stiffness matrices can be considered either with three or six degrees of freedom per node
- Import of shaft geometry as 2D DXF or 3D STEP file
- Strength calculation according DIN 743 (with extension for shaft strength)
- Connections by cylindrical gear pairs (with extension for shaft systems)
- Connections by bevel gear pairs (with extension for advanced shaft systems)
- Connections by worm-worm gears (with extension for advanced shaft systems)
- Planetary gear sets (with extension for advanced shaft systems)
- Centrifugal forces are considered on planet shafts
- Calculation of shaft speeds considering given constraints (with extension for shaft systems)
- Data exchange with several programs for gear calculation (with extension for shaft systems)
- Nonlinear rolling bearing stiffness can be considered
- Bearings can be selected from a database
- [MESYS Rolling Bearing Calculation](#)^[124] is fully integrated in the shaft calculation
- [MESYS Ballscrew Calculation](#)^[214] is fully integrated in the shaft calculation
- Calculation using load spectra
- Natural frequencies are calculated considering torsion, bending and axial modes (with extension for modal analysis)
- Campbell diagram (with extension for modal analysis)
- Harmonic response to periodic forces (with extension for modal analysis)
- Thermal expansions, optionally with varying temperature on a shaft
- Parameter variations
- 3D axisymmetric elastic parts considered by 3D FEA (with extension for 3D elastic parts)

- 3D elastic planet carriers defined parametrically or imported as STEP file (with extension for 3D elastic parts)
- Import of housings as STEP file and consideration as stiffness matrix and optionally by modal reduction (with extension for 3D elastic parts)
- Consideration of elastic deformation of bearing rings (with extension for 3D elastic parts and full bearing calculation)
- Consideration of gear body deformation (with extension for 3D elastic parts and shaft systems)

2.1 Input Parameters

The input parameters are shown on several pages which can be selected in a system tree.

2.1.1 System Inputs


If the element 'System' is selected in the system tree on the left, some general data can be defined.

System

▼ System

▼ Shafts

Shaft



Shaft Calculation

Project name
 Calculation description

Settings
Lubrication
Display settings

Consider weight
 Angle for weight β_w °
 Calculate natural frequencies
 Consider gyroscopic effect
 Maximum frequency f_{max} Hz
 Number of frequencies N_{rec}
 Consider gears as stiffness
 Consider gears as point load
 Consider housing stiffness

Housing material
Steel

Housing temperature
 T_h °C

Bearing reliability
 S %

Required life
 H h

Strength calculation
Infinite life according DIN 743

Bearing position
Definition for each bearing

Shear deformations
According Hutchinson

Consider nonlinear shaft model

Consider load spectrum

Calculate modified bearing life

MESYS Calculation Software

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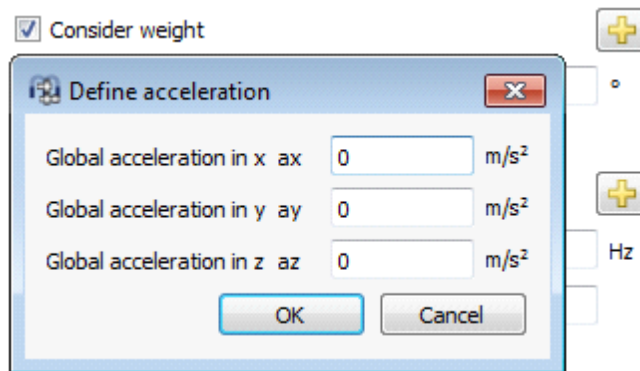
2.1.1.1 Project name and calculation description

The project name and the calculation description are just inputs which are shown in the report header. They can be used to enter information about the purpose of the calculation for documentation.

2.1.1.2 Settings

2.1.1.2.1 Consider weight and accelerations

The weight of shafts and additional masses is considered in the calculation if this option is set.

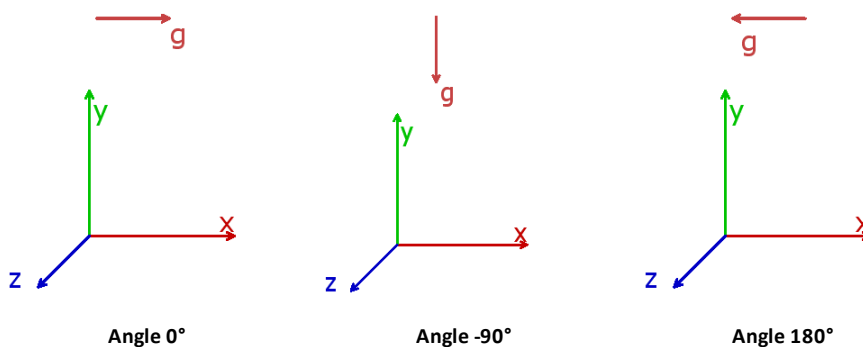


The **+**-button next to the weight checkbox allows the definition of global accelerations for the three axes. The accelerations are also available within the definition of a load spectrum.

The weight is already considered by setting the flag “Consider weight”. It does not have to be entered as acceleration. The input for accelerations is thought for additional accelerations like in gearboxes in vehicles.

2.1.1.2.2 Angle for weight

The direction of the weight can be defined by this setting. The angle is in the x-y-plane, a rotation around z-axis. A value of zero results in a weight in the direction of shaft axis. The weight direction is also shown as an arrow in the shaft graphics.



2.1.1.2.3 Calculate natural frequencies

The calculation of natural frequencies can be activated here. Running a calculation is faster if natural frequencies are not calculated. If a load spectrum is active, the natural frequencies are calculated only for the selected result element.

Calculation of natural frequencies also has to be activated if an harmonic response to dynamic loads should be calculated.

2.1.1.2.4 Consider gyroscopic effect

If this setting is activated, a gyroscopic matrix is considered in the calculation of natural frequencies. The mass inertia around x-axis and the speed of the shaft are required for this gyroscopic matrix.

Campbell diagram options

Start factor for speed: 0

End factor for speed: 5

Number of speeds: 15

Number of harmonics: 1

Maximum number of threads: 0

Accuracy for critical frequency: 0.02 Hz

Increase accuracy for mode changes

Calculate critical frequencies

Consider gear frequencies

Simplified method for harmonic response

Coefficient for global Rayleigh damping α : 0 rad/s

Coefficient for global Rayleigh damping β : 0 s

OK Cancel

The -button allows some additional inputs for the Campbell diagram:

- The diagram is generated for speeds between start and end factor times the current input value for speed.
- The number of speeds is the number of calculation points. Reduce it for faster calculation. For harmonic response over speed a larger value leads to smoother curves.
- The number of harmonics shows additional lines for multiples of shaft speed.
- The Campbell diagram is calculated in parallel as default. The maximum number of threads can be used to limit the number of threads that are

used. This can be applicable if the system memory is not sufficient for multiple parallel calculations of large systems.

- The accuracy for critical frequency is a tolerance for calculation of critical speeds as intersections of shaft speed and natural frequency.
- If 'Increase accuracy for mode changes' is selected, additional points are calculated when an axial mode changed into a radial mode for example.
- If 'Calculate critical frequencies' is selected, the critical frequencies are calculated when the Campbell diagram is shown. Else they are only calculated for the report of critical speeds. The diagram harmonic response over speed gets more accurate in this option is set because additional calculation points at the critical speeds are considered.
- If 'Consider gear frequencies' is selected, additional lines for gear tooth frequencies are shown and the corresponding critical frequencies are calculated.
- If the simplified method for harmonic response is activated, the harmonic response is calculated as superposition of mode shapes. The accuracy of the solutions depends of the calculated number of mode shapes in this case. For large systems the calculation can be faster if the option is activated. The result is more accurate if it is not activated.
- Coefficients for global Rayleigh damping are used to define a global damping matrix as $D=\alpha*M+\beta*K$ using the global mass matrix M and the global stiffness matrix K. Using the sizing button, the values can be calculated so that a user defined damping ratio will result for two given frequencies.

2.1.1.2.5 Maximum frequency

The maximum value for natural frequencies that are of interest can be entered here. If this maximum frequency is set too small, in some cases no results for natural frequencies might be shown.

2.1.1.2.6 Number of frequencies

The number of natural frequencies that should be calculated can be specified here. Normally only the first few modes are interesting. Both limits, the number of frequencies and the maximum are considered. If the Campbell diagram shows not enough lines, these restrictions are often the reason.

2.1.1.2.7 Consider dynamic loads in static equilibrium

If this option is set, dynamic forces and imbalance are considered in the static equilibrium. This approximation is good enough in case the shaft speed is well

below the critical speed. The value of the force is taken for time zero, so the direction of an imbalance is in direction of the input phase.

2.1.1.2.8 Housing material and housing temperature

The material data of the housing is used together with the temperature of the housing to calculate the axial displacement for boundary conditions connected to the housing. For the shaft calculation only the thermal elongation coefficient is important, the other data is only used to transfer it to the bearing calculation.

Property	Variable	Value	Unit
Housing youngs modulus	E_Housing	207000	MPa
Housing poisson number	v_Housing	0.3	
Housing density	ρ_Housing	7850	kg/m ³
Housing thermal elongation coefficient	α_Housing	11.5	10 ⁻⁶ /K
Housing thermal conductivity	λ_Housing	42.6	W/m·K

2.1.1.2.9 Required life

The required life is passed to gear calculations which are connected to the system, and it is used for the shaft strength calculation.

2.1.1.2.10 Bearing reliability

The bearing reliability can be optionally defined on system level and is then passed to all bearing calculations.

2.1.1.2.11 Strength calculation

The method for shaft strength calculation can be selected. Currently, only DIN 743 (2012) is available. It can be selected if the calculation should be done considering infinite or finite life. For finite life the number of cycles is calculated using the input for "Required life".

2.1.1.2.12 Bearing position

The position of the first rolling element for rolling element bearings can be defined globally using this settings. See [Position of first rolling element](#)¹³⁰.

In case of bearings with clearance and very small loads it might happen, that only one rolling element is loaded. This can lead to convergence problem as there is zero stiffness normal to the loading. Then setting the option to "Load direction between rolling elements" might be helpful. Else the setting "First rolling element on load direction" leads to the maximum contact stress and the minimum bearing

life in most cases. Usually there is only a noticeable effect on bearing with a low number of rolling elements.

2.1.1.2.13 Shear deformations

Shear deformation should normally be taken into account, since there are these deformations. For comparison with other calculations the shear deformations can be ignored. For nonlinear shaft model the shear deformations should be considered to improve convergence.

There are different options available. Either a fixed value can be entered. Usually 1/1.1 is used for cylinders. The options 'according Cowper' and 'according Hutchinson' are also considering an inner diameter for hollow shafts and the Poisson number of the material. Hutchinson considers higher order terms in his formula than Cowper according to (Hutchinson, January 2001) (Cowper., June 1966):

- Hutchinson (Circular Cross Section):

$$k = \frac{6(1 + \nu)^2}{7 + 12\nu + 4\nu^2}$$

- Hutchinson (Hollow Circular Cross Section):

$$k = \frac{6(a^2 + b^2)^2(1 + \nu)^2}{7a^4 + 34a^2b^2 + 7b^4 + \nu(12a^4 + 48a^2b^2 + 12b^4) + \nu^2(4a^4 + 16a^2b^2 + 4b^4)}$$

where b is the outer radius, a the inner radius and ν the Poisson's ratio.

- Cowper (Circular Cross Section):

$$k = \frac{6(1 + \nu)}{7 + 6\nu}$$

- Cowper (Hollow Circular Cross Section):

$$k = \frac{6(1 + \nu)(1 + m^2)^2}{(7 + 6\nu)(1 + m^2)^2 + (20 + 12\nu)m^2}$$

where $m = b/a$ is the ratio of inner to outer radius and ν the Poisson's ratio.

2.1.1.2.14 Consider nonlinear shaft model

A nonlinear shaft model can be taken into account. The nonlinear model calculates equilibrium of loads in the deformed state.

A beam fixed on one side and radially loaded on the other side will show only radial deflections in the linear model. In the nonlinear model the end point will also move axially and the length of the beam will not increase. A beam fixed on both sides will show an increased stiffness against loading in the center, the

stiffness will increase with the loading because of generated axial loads. The nonlinear model is only needed on large deflections; normal shafts in mechanical engineering do not require a nonlinear model.

2.1.1.2.15 Consider load spectrum

If a load spectrum should be taken into account, select this option. An additional item for the load spectrum will be shown in the tree on the left. See [Calculation with load spectrum](#)^[87] for the definition of the load spectrum.

2.1.1.2.16 Consider gears as stiffness

If this option is selected, the shaft diameter is increased to root diameter plus $0.4 \cdot \text{module}$ automatically. For the root diameter a dedendum of the reference profile of 1.25 is assumed.

2.1.1.2.17 Consider gears as a point load

If this option is selected, the load line from the meshing of gears is converted to a point load. This is useful for calculating bearing loads when the user wants to consider perfectly centered gears loads, that is, no misalignments in tooth mesh.

2.1.1.2.18 Calculate modified bearing life

This setting is passed on to the bearing calculation and activates or suppresses output of modified life in results. It can be disabled if no information about lubrication is available.

2.1.1.2.19 Consider configurations

If multiple groups are used, an additional option is shown to activate configurations. Using the new input page for configurations the status of connections can be changed to allow different options for power flow. See [Configurations](#)^[88] for usage.

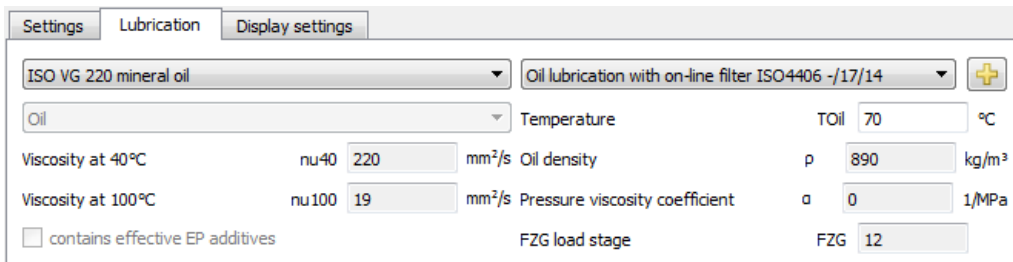
2.1.1.2.20 Consider housing stiffness

When this option is set, an additional page for definition of a housing stiffness matrix is shown in the system tree. See [Housing stiffness matrix](#)^[89] for details of usage.

A housing stiffness can also be considered by using [3D elastic part as housing](#)^[97], this would be easier to use compared to a raw stiffness matrix.

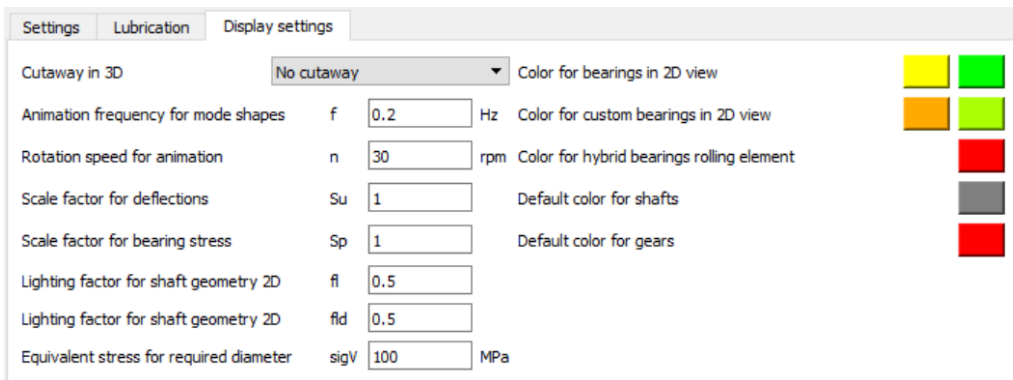
2.1.1.3 Lubrication

The data for lubrication is only to transfer it to bearing or gear calculations. For details regarding the inputs, please consult the documentation of the rolling bearing calculation. The 'FZG load stage' is an additional input which is passed to a gear calculation.

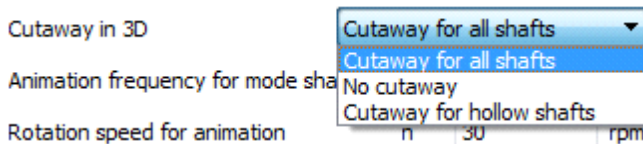


2.1.1.4 Display settings

There are different view options that can be set under the tab-page 'Settings'.



Cutaway in 3D: It is possible to generate a cutaway view for all the existing shafts. If needed, the cutaway can be only applied to the hollow shafts. This setting can be overridden for individual shafts.



With regard to the 3D animation settings, both the frequency for modes shapes and the rotation speed are adjustable. The default frequency of 0.2Hz for the mode shape above, will lead to a period of 5 seconds in the animation. The 3D pressure distribution of the bearings can be easily exaggerated with a scale factor 'Sp'. And with the 'Su' the deformed shape in 3D can be also scaled. Setting 'Su' to zero will show the deformed shape in 3D without scaling, so scale factor of one.

The look of the shaft geometry in 2D can be emphasized by applying a lighting factor 'fl'. The factor 'fld' is used for the shaft background graphic in 2D diagrams.

The input value for 'sigV' allows the user to specify the maximum permissible stress that any shaft shall reach under loading. In this way, the software shows graphically an approximation of required diameter of the shafts subjected to a maximum value of equivalent stress. Please refer to the 'Graphics area' section of this document on how to visualize the required diameter diagram.

Colors for bearings can be specified on the right of the window. The left button is for bearings, which are connected to one shaft and the housing, while the right button is for bearings connected to two shafts.

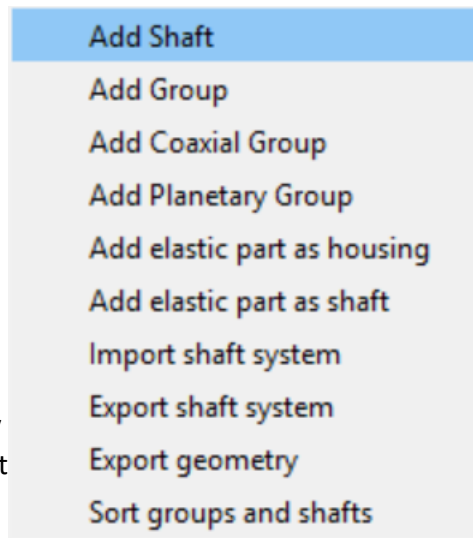
In addition, a default color for shafts and gears can be defined.

2.1.1.5 System options in the system tree

Several options are available in the system tree under "Shafts" and "Bearings" using the context menu.

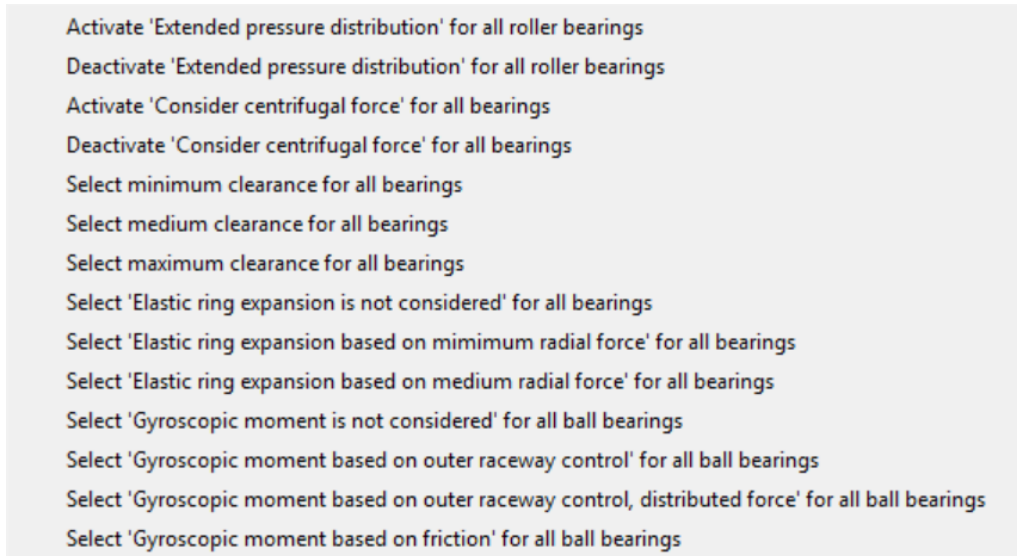
The context menu under "Shafts" provides following options:

- "Add shaft" adds a shaft to the system. If groups are used shafts can only be added to groups.
- "Add groups" adds a group to the systems.
- "Add coaxial groups" adds a coaxial groups. The difference between coaxial groups and groups is, that coaxial groups only have an axial position and support elements can be used between shafts in different coaxial groups.
- "Add planetary groups" adds a planetary group. A planetary group rotates about a carrier shafts and a number of planets can be defined on the group. All elements are generated "number of planets" times.
- "Add elastic part as housing" adds a 3D-elastic part as housing. This part does not rotate. See [3D elastic part as housing](#)^[97] for details.
- "Add elastic part as shaft" adds a 3D-elastic part as shaft. This part can rotate and it can also be a planet carrier. See [3D elastic part as shaft](#)^[92] for details.
- "Import shaft system" allows to import an existing shaft system. Either a file of a MESYS shaft calculation can be selected or a file in REXS format.
- "Export shaft system" allows the export of the system in REXS format. This format allows the data exchange with other CAE software.
- "Export geometry" exports the system as STEP file. Only a simplified geometry is used for gears and bearings. The export could be used to check if it would fit into a housing for example.

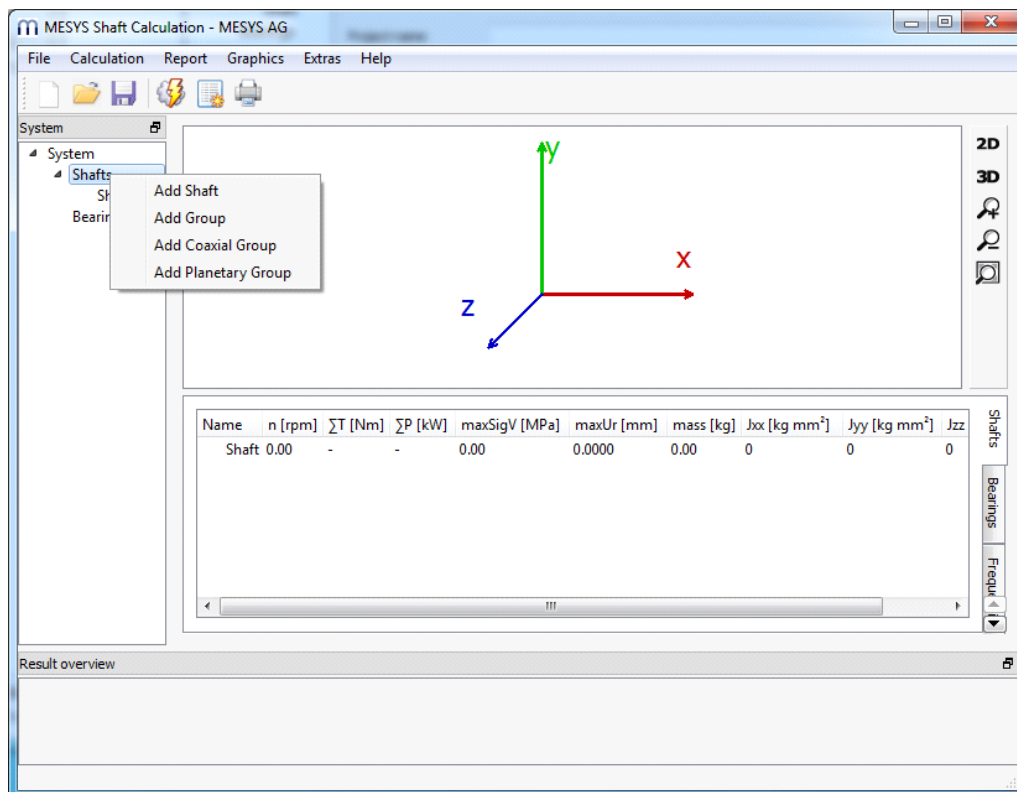


- "Sort groups and shafts" sorts shafts and groups alphabetically. Initially the elements are sorted in creation order.

The context menu under "Bearings" allows to activate or deactivate settings for all bearing calculations at the same time:



2.1.2 Definition of shafts



In the system tree using a right click on “Shafts”, additional shafts can be added to the system. The shaft can be defined by selecting it in the system tree.

Either shafts can be directly added, if only single or coaxial shafts are used. For general shaft systems groups or coaxial groups can be used to build shaft systems. Planetary groups are used for planet shafts to model planetary gear sets.

2.1.2.1 General shaft data

On the first page “General” some general data and data for the strength calculation can be defined for each shaft.

2.1.2.1.1 Name

A name for each shaft can be defined which is used in the system tree, messages and the report to identify the shaft.

2.1.2.1.2 Material

The material can be selected from the data base. The **+**-button allows to see the material data or to change the data if user-input is selected.

2.1.2.1.3 Position

If several shafts are defined, they can have different axial positions. The value for position defines the position of the start point or the left end of a shaft relative to the group. All loads and boundary conditions are then defined relative to this start point.

The shaft position can also be changed by dragging the shaft in the 2D view using left mouse button together with Shift and Control keys.

2.1.2.1.4 Speed

The rotation speed of the shaft has an influence on bearing life and natural frequencies with gyroscopic effect. If the checkbox behind the input is not

selected, the software is trying to calculate it using constraints. It is set to zero if the shaft is not connected to others.

A positive speed is a clockwise rotation looking in the x-direction.

2.1.2.1.5 Temperature

The temperature of the shaft can be defined here. The temperature is transferred to the bearing calculation and will induce axial stresses into the shaft.

In case that the shaft temperature is not uniform, the user must set the flag corresponding to 'Consider temperature per element' at the tab-page 'Settings'. That allows the user to add a temperature difference ' ΔT ' (\pm) for each element created at the outer geometry on the temperature previously set at the tab-page 'General'. The reference temperature with no stress field is 20°C.

2.1.2.2 Data for strength calculation

There are several inputs for the strength calculation according DIN 743 which are defined for each shaft.

2.1.2.2.1 Load factor (static)

For the calculation of static safety factor, the maximal load is multiplied with the static load factor.

2.1.2.2.2 Load factor (fatigue)

For the calculation of safety factor against fatigue the maximal load is multiplied with the load factor (fatigue).

2.1.2.2.3 Overload case

The overload case defines how the permissible stress is calculated. Either "Constant stress ratio" or "Constant mean stress" can be selected. "Constant stress ratio" leads to smaller safety factors and is therefore on the safe side.

2.1.2.2.4 Diameter at heat treatment

The diameter at heat treatment is used to calculate the tensile strength and the yield point for the given material. The software sets it to the maximum shafts diameter automatically, but for hollow shafts the wall thickness can be chosen manually, if heat treatment was done for the hollow shaft.

2.1.2.2.5 Number of load cycles

If a calculation for finite life is done, the number of load cycles is derived from rotation speed and required life as default. But for non-rotating shafts this is not possible. Then the number of load cycles can be entered manually.

2.1.2.2.6 Factor for surface hardening

In DIN 743, a factor for surface hardening KV increases permissible stress. It can be set for hardening or mechanical treatment like shot-peening. As the standard provides a wide range for this value, it has to be defined by the user.

2.1.2.2.7 Stress ratio

The stress ratio R is usually defined as minimum stress value divided by maximum stress value, which leads to values between $-\infty$ and +1.

The software is using a value between -1 and +1 instead and it defines the stress ratio using the following definition:

$$R = \begin{cases} \frac{\sigma_1}{\sigma_2}, & |\sigma_2| < |\sigma_1| \\ \frac{\sigma_2}{\sigma_1}, & |\sigma_1| < |\sigma_2| \end{cases}$$

Using this definition $R = 0$ can be used for pulsating loads between zero and a positive or a negative value. For alternating load $R = -1$ and for constant load $R = +1$.

The stress ratio can be defined for tension, bending and torsion independently. For a rotating shaft the stress ratio for bending should usually be -1. If the stress ratio for torsion is not given, make a calculation with two cases: constant and pulsating for example. It depends on the current loading which case is conservative.

2.1.2.3 Geometry data

The geometry data of a shaft is defined on page “Geometry”. The geometry can be defined using cylindrical and conical elements for outer and inner geometry. The shaft geometry could also be imported from CAD files, see [Import of shaft geometry from CAD files](#) ^[78].

Outer Geometry				Inner Geometry			
L=90mm				L=90mm			
	Length [mm]	Diameter 1 [mm]	Diameter 2 [mm]		Length [mm]	Diameter 1 [mm]	Diameter 2 [mm]
1	20	20		1	40	10	
2	40	30		2	10	10	5
3	20	20		3	40	5	
4	10	20	10				

There is an input table for both inner and outer geometry. The plus and minus buttons allow adding and removing rows, the arrows can be used to move an element up or down in the table.

Length and diameter 1 have to be defined for each element. If diameter 2 is left empty, a cylinder is used else a cone. No negative values are permitted. The total length is shown above the table.

If a hole from the right shall be defined as inner geometry, enter a first element with diameter zero to get to the start position on the hole.

The dimensions of each element can also be changed in the view. Use Shift and left mouse button for it. For changing the length the click needs to be in the right third of an element. For changing the diameter it has to be in the upper third of an element. To change a cylinder into a cone element the mouse needs to be in the left third of an element. The interactive change is restricted to a 1mm grid. Using Shift and Control buttons will move the shaft instead of changing an element.

The table for outer geometry also allows an input for a temperature difference if the corresponding option under settings is activated.

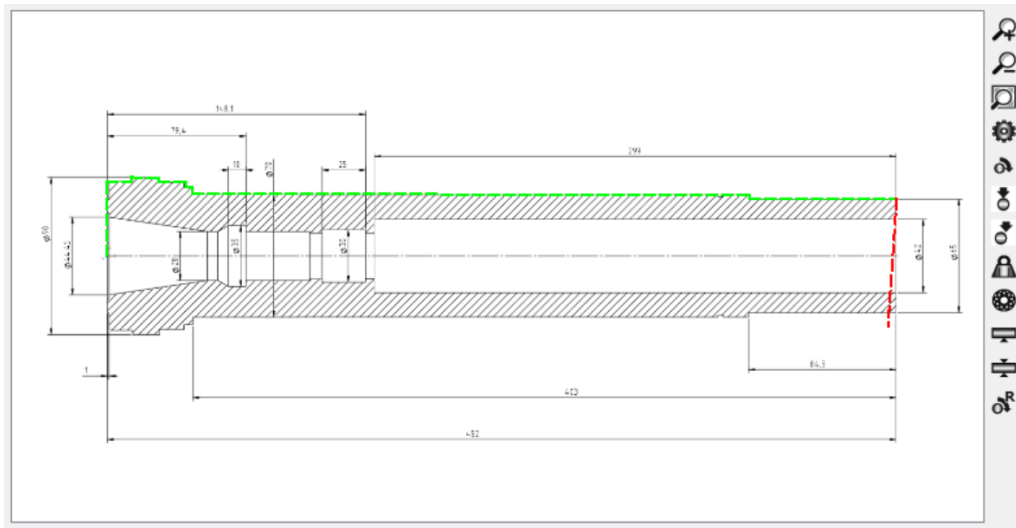
2.1.2.3.1 Background drawing

Using the context menu (right mouse button) a background graphic can be added to the 2D-graphic. Here a CAD-drawing can be set as background image if a PDF drawing is available only.

The background drawing is added using "Set from clipboard" after copying a section in the PDF drawing. Then with "Set origin" the location of the drawing is defined and with "Set horizontal scale" or "Set vertical scale" the scale factor is defined.

The background drawing can also be moved to the foreground using the context menu or it can be rotated.

Then the shaft geometry can be adjusted to the drawing. Either by changing the geometry elements or by using the input as polyline. First define the inner geometry and then the outer geometry.



2.1.2.3.2 Define geometry as polyline

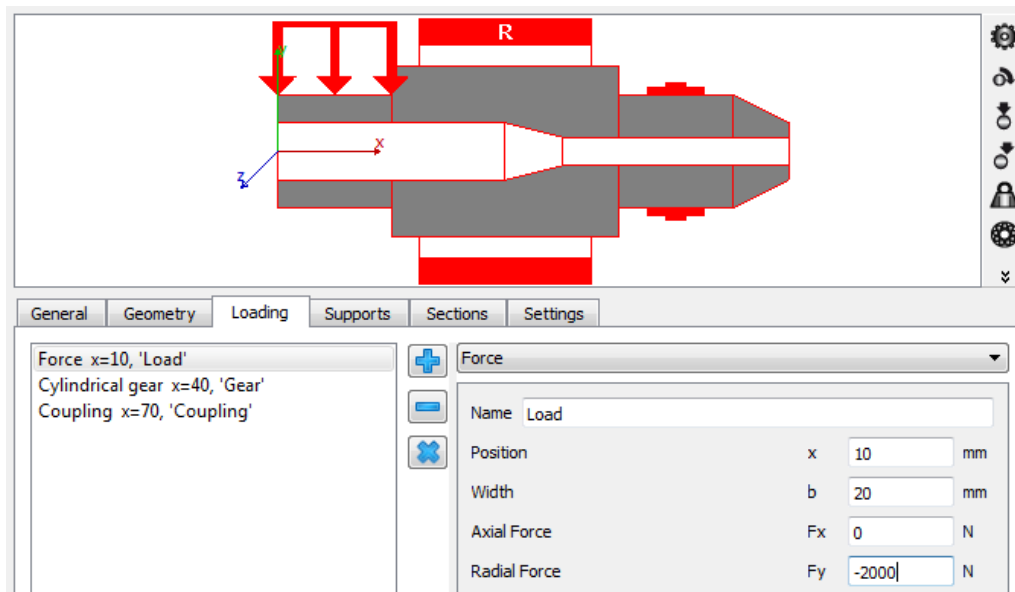
Using the context menu an input of inner or outer geometry as polyline can be activated. This is usually done after setting a background drawing, but is independent of it.

Using the left mouse button, points for the polyline can be added. A red line shows an invalid position for the next point and this will not be accepted. Using Shift plus left button the last selected point is removed again. The right mouse button is used to stop the input and the result is set to the table for inner or outer geometry.

The selected points are rounded to a millimeter.

2.1.2.4 Loading

Loads can be defined on the page "Loading" for each shaft. Loads may be defined outside a shaft; masses may not be outside of a shaft. Loads can also be changed by double-clicking them in the graphics. They can be moved in the graphic by using Shift and the left mouse button.



Loads can be added by the plus button next to the list. The type of the loading can be selected using the list on the right. Each load element has a name to identify it, a position relative to the left end of the shaft and a width.

The graphical representation just shows a symbol for the load, not the actual direction, as it can have six components. The coordinate system is shown on the right. The shaft axis is in x-direction, the y-axis goes up and the z-axis to the front. The weight is in negative y-directions a default (angle $\beta_w = -90^\circ$).

2.1.2.4.1 Force

The load element “Force” allows the definition of a force and a moment with three components each. The moments are moments around an axis. So F_x is the axial force and M_x is the moment around the x-axis so it is a torque.

2.1.2.4.2 Coupling

The coupling only allows a definition of a torque. The direction on the torque can either be defined by its sign or by the selection “Shaft is driven” (torque has the same sign as the rotation speed) or “Shaft is driving” (torque has different sign than the rotation speed).

2.1.2.4.3 Cylindrical gear

The load can be defined by entering geometry data for a cylindrical gear and a

torque.

The direction of torque can either be defined by its sign or by the selection "Shaft is driven"/Shaft is driving". The contact point is given by an angle of contact, which is zero on the y-axis and 90° on the z-axis.

The gear is defined by the usual data on a gear drawing: Number of teeth, normal module, normal pressure angle, helix angle with its direction. Optional inputs are the number of teeth for the mating gear and the center distance. If they are given, the loads are calculated for the operating center distance, which is more accurate than on the reference diameter.

Name	Gear	
Position	x	40 mm
Width	b	30 mm
Torque	T	0 Nm
Direction of torque	Own Input	
Angle to contact	ζ	0 °
Number of teeth	z	25
Normal module	m_n	2 mm
Normal pressure angle	α_n	20 °
Helix angle	β_n	0 °
Helix direction	Spur gear	
Number of teeth of mating gear	z2	25
Center distance	a	50 mm

For internal gears the number of teeth has to be entered as negative value, the center distance is always positive as defined in ISO 21771 for gear geometry.

2.1.2.4.4 Bevel gear

The load can be defined using torque and geometry data for a bevel or hypoid gear.

The direction of torque can either be defined by its sign or by the selection "Shaft is driven"/Shaft is driving". The contact point is given by an angle of contact, which is zero on the y-axis and 90° on the z-axis.

Either the pitch angle can be defined directly or it is calculated using axis angle and number of teeth of the mating gear.

Instead of the mean normal module, the outer pitch diameter can be used as an input.

The width is the width of the gear (the length of the flank), the axial length of the loaded shaft is smaller dependent on the pitch angle.

Name	Bevel gear		
Position	x	40	mm
Width	b	20	mm
Torque	T	0	Nm
Direction of torque	Own Input		
Orientation	Apex on the right		
Angle to contact	ζ	0	°
Number of teeth	z	25	
Normal module	mn	2	mm <input checked="" type="radio"/>
Pitch diameter	de	64.1421	mm <input type="radio"/>
Profile shift coefficient	x	0	
Generated pressure angle	α_{nD}	20	°
Generated pressure angle	α_{nC}	20	°
Helix angle	β_m	0	°
Helix direction	Spur gear		
Pitch angle	δ	45	° <input type="checkbox"/>
Number of teeth of mating gear	z2	25	
Axis angle	Σ	90	°
Center distance	a	0	mm

2.1.2.4.5 Worm

The load can be defined using torque and geometry data for a worm.

The direction of torque can either be defined by its sign or by the selection "Shaft is driven"/Shaft is driving". The contact point is given by an angle of contact, which is zero on the y-axis and 90° on the z-axis. The gear is defined by the usual data on a gear drawing: Number of teeth, axial module, normal pressure angle, lead angle with its direction. The gear friction coefficient, μ_z , can be set for the gear contact.

Name	Worm		
Position	x	1500	mm
Width	b	200	mm
Torque	T	0	Nm
Direction of torque	Own Input		
Angle to contact	ζ	0	°
Number of teeth	z	1	
Axial module	mx	20	mm
Normal pressure angle	α_n	20	°
Lead angle	γ_m	5	°
Helix direction	Helix left hand		
Gear friction coefficient	μ_z	0	

2.1.2.4.6 Worm gear

The load can be defined using torque and geometry data for a worm gear.

The direction of torque can either be defined by its sign or by the selection "Shaft is driven"/"Shaft is driving". The contact point is given by an angle of contact, which is zero on the y-axis and 90° on the z-axis. The gear is defined by the usual data on a gear drawing: Number of teeth, transverse module, normal pressure angle, helix angle with its direction. Optional inputs are the number of teeth for the mating gear and the center distance. The gear friction coefficient, μ_z , can be set for the gear contact.

Name	Worm gear		
Position	x	500	mm
Width	b	200	mm
Torque	T	0	Nm
Direction of torque	Own Input ▼		
Angle to contact	ζ	0	°
Number of teeth	z	40	
Transverse module	mt	20	mm
Profile shift coefficient	x	0	
Normal pressure angle	α_n	20	°
Helix angle	β_m	5	°
Helix direction	Helix left hand ▼		
Gear friction coefficient	μ_z	0	

2.1.2.4.7 Eccentric force

The eccentric force allows entering a force with three components at a point which is not on the shaft axis. The point and the force are defined in polar coordinates. The angle is defined in the same way as the angle of contact in the cylindrical gear (see above).

The radial force is positive if away from the shaft center, the tangential force is positive in the direction of the angle. Moments are calculated by the points and forces given.

2.1.2.4.8 Mass

Additional mass can be defined using the “Mass” element. In addition to its mass, also the mass moment of inertia around three axes can be defined. The weight of the mass is considered and it is considered in the calculation of natural frequencies.

If a width is entered, the mass is equally distributed on this line.

The total mass inertias of this distributed mass are the input values, which results in a minimum value for Jyy and Jzz. A message will be shown if the input values are too small.

If the added mass has an eccentricity, it will cause rotating unbalance and can be considered as a harmonic force according to the formula: $F = me_{cc}\omega^2 \cdot e^{i(\omega t + \varphi)}$ (e_{cc} : eccentricity in mm). A phase shift, φ (°), for the imbalance force can be taken into account too. The imbalance will only affect the results for the harmonic response. Bearing life, contact stress or safety factors for shaft strength will not change. Setting the option [Consider dynamic loads in static equilibrium](#) will lead to a change of bearing life or safety factors, but this is an approximation only.

▼

Name

Position	x	<input type="text" value="0"/>	mm
Width	b	<input type="text" value="0"/>	mm
Mass	m	<input type="text" value="0"/>	kg
Mass moment of inertia	Jxx	<input type="text" value="0"/>	kg m ²
Mass moment of inertia	Jyy	<input type="text" value="0"/>	kg m ²
Mass moment of inertia	Jzz	<input type="text" value="0"/>	kg m ²
Eccentricity	e	<input type="text" value="0"/>	mm
Phase	φ	<input type="text" value="0"/>	°

2.1.2.4.9 Imbalance

If an imbalance should be only considered as a force without adding mass to the shaft, this element has to be used. The imbalance force will be regarded as a harmonic force and calculated in the same way as with an

▼

Name

Position	x	<input type="text" value="0"/>	mm
Mass	m	<input type="text" value="0"/>	kg
Eccentricity	e	<input type="text" value="0"/>	mm
Phase	φ	<input type="text" value="0"/>	°

eccentric mass, that is: $F = me_{cc}\omega^2 \cdot e^{i(\omega t + \varphi)}$, but the inertia and the natural frequencies of the shaft will remain unaffected.

The imbalance will only affect the results for the harmonic response. Bearing life, contact stress or safety factors for shaft strength will not change.

Setting the option [Consider dynamic loads in static equilibrium](#)^[47] will lead to a change of bearing life or safety factors, but this is an approximation only.

2.1.2.4.10 Dynamic force

This element allows the user to consider harmonic loads (moments or forces) on a shaft according to:

$$F = \bar{F} \cdot e^{i(\omega t + \varphi)}$$

A factor h is multiplied to the shaft rotation frequency.

In case that the frequency is not dependent on the shaft speed, then it can be manually specified by checking the box next to the "Frequency" at the software dialog. A phase shift can be defined for the force as well.

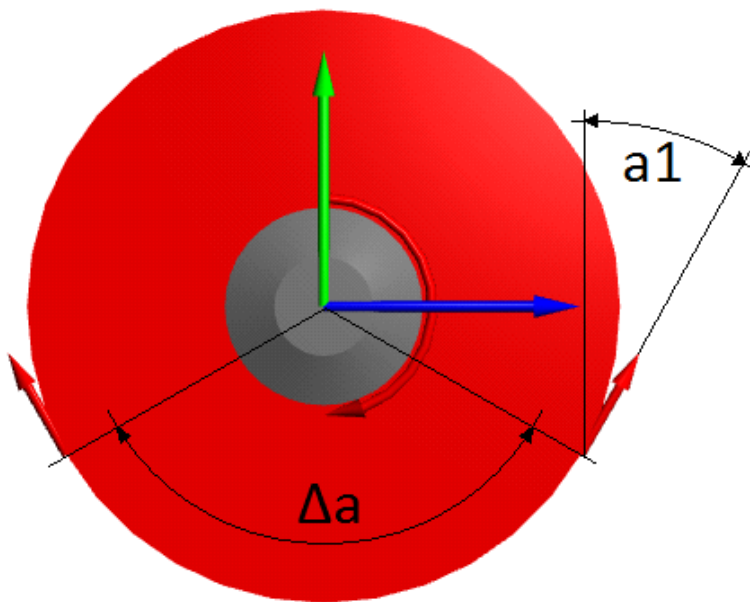
The dynamic force will only affect the results

for the harmonic response. Bearing life, contact stress or safety factors for shaft strength will not change. Setting the option [Consider dynamic loads in static equilibrium](#)^[47] will lead to a change of bearing life or safety factors, but this is an approximation only.

2.1.2.4.11 Pulley

The load can be defined defining a torque on the pulley and its pretension coming from a belt.

Pulley			
Name	Pulley		
Position	x	40	mm
Width	b	10	mm
Pretension force	Fpre	20	N
Torque	T	200	Nm
Direction of torque	Shaft is driving		
Angle of first force	α_1	30	°
Wrap angle	$\Delta\alpha$	120	°
Diameter	d	60	mm <input checked="" type="radio"/>
Number of teeth	z	0	<input type="radio"/>
Pitch	p	0	mm



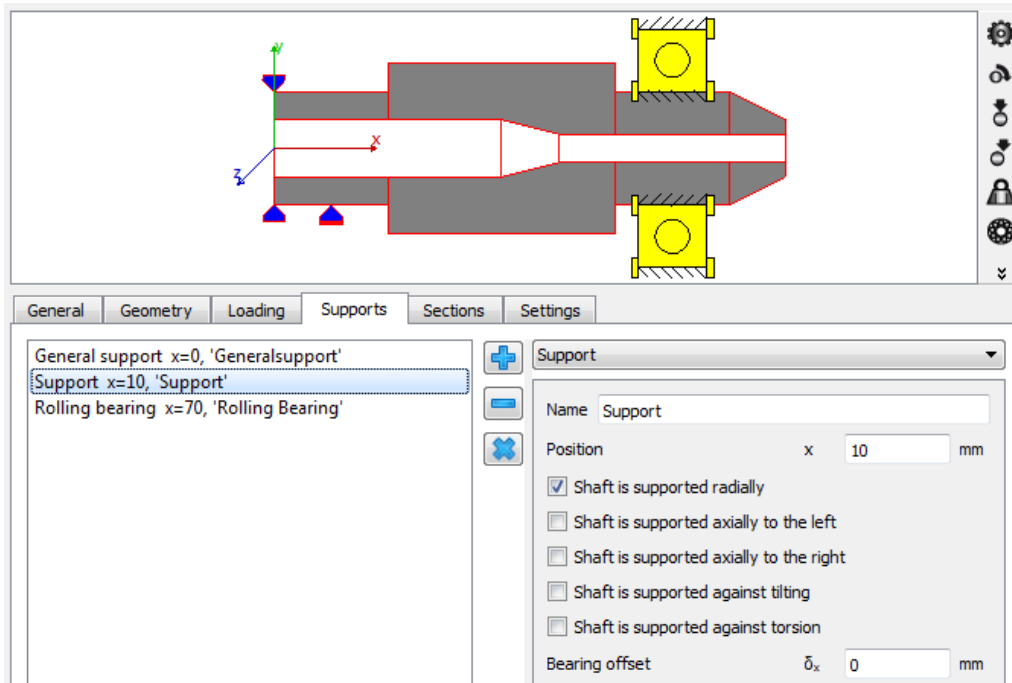
The direction of torque can either be defined by its sign or by the selection “Shaft is driven”/“Shaft is driving”. The positioning of the contact surface is given by an angle of first contact, which starts clockwise from zero on the y-axis, and followed by the wrap angle as shown in the picture below. The pulley diameter can be set straight away or by means of its teeth number and pitch.

A torque will lead to an increase force on one side of the belt and to a decreased force on the other side. This is the correct model in case no tensioning pulley is used.

2.1.2.5 Supports

Boundary conditions can be defined on the page “Supports”. They can also be edited by a double click in the graphics or moved by using Shift and the left mouse

button. Support elements including rolling bearings only define boundary conditions, they have no mass or weight. For most supports an excitation can be defined which is considered in calculation of harmonic response, see [Excitations](#) [74].



2.1.2.5.1 Support

The “Support” element is an easy-to-use element which allows defining constraints between the shaft and the housing. There are five checkboxes to define in which directions the shaft is supported. For axial direction the movement to the right and to the left can be constrained independently.

A bearing offset can be defined for the three directions. An offset means that the point on the housing is moved in this direction. The resulting force on the shaft is in the direction of the offset.

2.1.2.5.2 General support

The “General support” allows the definition of constraints between a shaft and the

housing or between two shafts for all six degrees of freedom independently.

First a selection can be done to which element the shaft is connected to. It can be the housing or a second shaft.

For each degree of freedom, a type of constraint can be defined then:

- No constraint: No constraint in the direction is defined.
- Fixed: The degree of freedom between both elements is coupled in both positive and negative direction. An offset or a clearance can be defined optionally. A positive value for the offset results in a movement of the current shaft in this direction, or a force on the selected shaft in this direction. The clearance is either zero or a positive value, it may not be negative.
- Fixed to the left (or right): The selection “Fixed to the left” means than the selected shaft is constrained only in the negative direction (left on a horizontal axis). “Fixed to the right” is the constraint in positive direction only. For the unidirectional constraint an offset can be defined.
- Stiffness: The “Stiffness” acts like a spring. An offset and a clearance can be defined. Offset and clearance are defined as explained for ‘Fixed’ above.
- Stiffness to the left (or right): Like the rigid constraint also stiffness can be defined in just one direction. The stiffness and an offset can be defined.

2.1.2.5.3 Rolling bearing

The element “Roller bearing” defines a connection to the rolling bearing

calculation.

The current shaft can be connected to the inner or to the outer ring of a roller bearing. The other ring of the bearing can be connected to either the housing or another shaft.

By clicking on the button “Geometry, Material, Temperature, Lubrication’ is connected” the user is prompted, by means of a pop-up window, to set those flags of the data to be transferred to bearing calculation. Shaft diameters, material data, lubrication and temperature data can be transferred to the bearing calculation. Displacements, loads and speeds are always connected.

The dialog box contains the following elements:

- Name: B01
- Position: x 30 mm (with left and right arrow buttons)
- Type: Deep groove ball bearing (6202) (with a plus button)
- Shaft connected to inner ring (dropdown menu)
- Connect outer ring to housing (dropdown menu)
- 'Geometry, Material, Temperature, Lubrication' is connected (button)
- Shaft is supported radially
- Shaft is supported axially to the left
- Shaft is supported axially to the right
- Bearing offset δ_x : 0 mm
- Bearing offset δ_y : 0 mm
- Bearing offset δ_z : 0 mm

The dialog box contains the following elements:

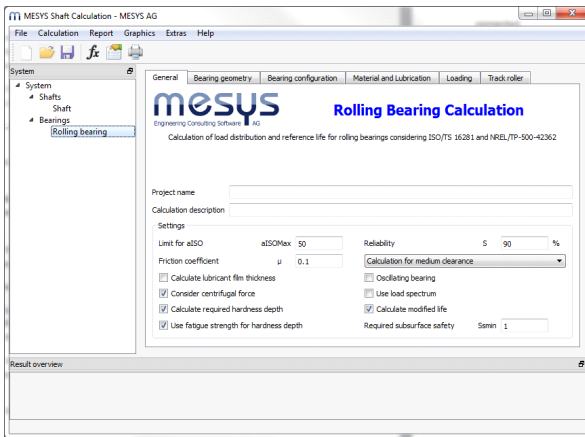
- 'Geometry' is connected
- 'Material' is connected
- 'Temperature' is connected
- 'Lubrication' is connected
- OK button
- Cancel button

The shaft can be supported radially and axially to the left and the right. This setting changes the connection of the bearing outer ring to the shaft/housing. If the bearing cannot constrain a certain movement, there won't be reaction forces in that direction.

The bearing offsets will move the node of the secondary shaft (or housing) in this direction, it will therefore generate forces to the current shaft in the direction of the offset. The offset can be used to generate bearing pretension in axial direction or to take misalignments of the housing into account.

Using the **+**-button next to the bearing type a dialog is shown to select a rolling bearing from the database.

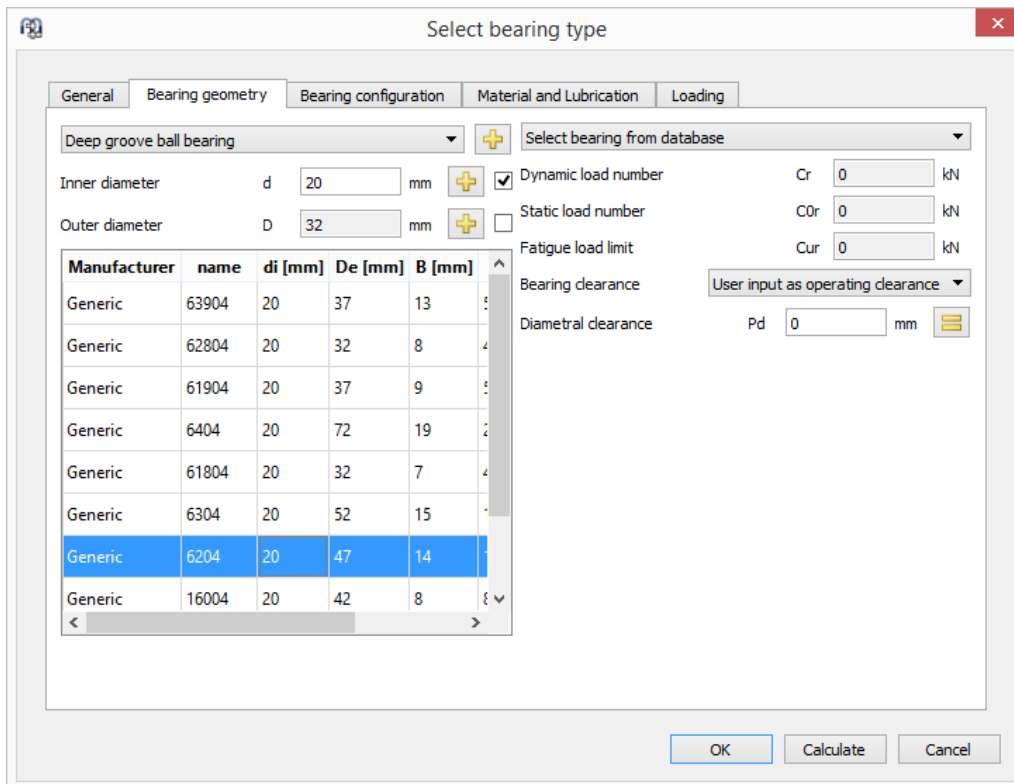
If a rolling bearing element is added, an entry for the bearing in the system tree will be added. If this element is selected, the dialogs of the rolling bearing calculation are active to define the bearing.



2.1.2.5.3.1 Selecting bearing from database

The standard interface of the rolling bearing calculation software is shown for selection of a rolling bearing. Either a bearing is selected from the database or custom input is selected. Restrictions to inner and outer diameter can be applied if the bearing is selected from the database; the shaft diameter at the bearing position is set as default value.

For angular contact bearings or taper roller bearings the direction of contact angle can also be defined at the bottom of the dialog.

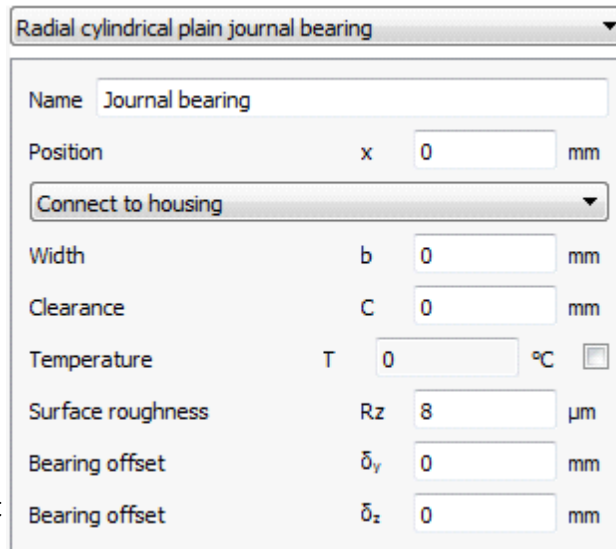


2.1.2.5.4 Radial cylindrical plain journal bearing

The load-displacement behavior of the plain bearing is calculated according to (ISO 7902-2, May 1998). Therefore, the ratio of width to diameter is restricted to $0.25 \leq B/D \leq 1.5$.

The parameters used for the definition of this bearing type are shown in the enclosed figure. Clearance represents the total diametric operating clearance between the journal and the bearing. Any eccentricity is introduced through the corresponding offset inputs in y and x direction.

The optional temperature input does only affect lubricant viscosity. It does not affect clearance.



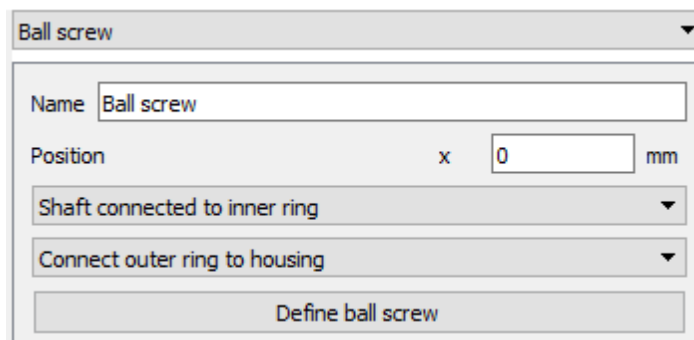
Surface roughness is the average between the shaft and the bearing and it is used to calculate a minimum required film thickness.

The outer ring of the bearing can be connected to either the housing or another shaft.

For modal analysis the unsymmetrical stiffness matrix of the plain bearing is replaced by two orthogonal springs, because of limitations of the FEA library. Therefore, any natural frequencies are an approximation only. No stability analysis can be performed with this software. Damping is not considered either.

2.1.2.5.5 Ball screw

A ball screw can be used as support element. The geometry of the ball screw can be defined at the ball screw element in the system tree. The button "Define ball screw" is not yet active.



2.1.2.5.6 Coupling for reaction torque

A coupling for reaction torque fixes the rotation angle of a shaft. The sum of torques will be taken at this element.

An additional checkbox allows to consider the constraint in modal analysis or not to consider it.

2.1.2.5.7 Stiffness matrix

A general symmetric stiffness matrix can be defined between a shaft and the housing or between two shafts.

Stiffness matrix

Name

Position x mm

Connect to housing

	ux [mm]	uy [mm]	uz [mm]	rx [mrad]	ry [mrad]	rz [mrad]
Fx [N]	0	0	0	0	0	0
Fy [N]		0	0	0	0	0
Fz [N]			0	0	0	0
Mx [Nm]				0	0	0
My [Nm]					0	0
Mz [Nm]						0

2.1.2.5.8 Damping matrix

A general damping matrix can be defined between a shaft and the housing or between two shafts.

Damping matrix

Name Damping matrix

Position x 0 mm

Connect to housing

	vx [m/s]	vy [m/s]	vz [m/s]	ω_x [rad/s]	ω_y [rad/s]	ω_z [mrad/s]
Fx [N]	0	0	0	0	0	0
Fy [N]		500	0	0	0	0
Fz [N]			0	0	0	0
Mx [Nm]				0	0	0
My [Nm]					0	0
Mz [Nm]						0

2.1.2.5.9 Weld point

A weld point is fixing all six degrees of freedom. It is easier to use than a [general constraint](#) with all components set to 'fixed' but has the same functionality. The weld point has the additional option to define a width. In case a width > 0 is defined, multiple weld points are considered automatically.

2.1.2.5.10 Planetary support

A planetary support is available for planetary shafts. The planetary shaft is always supported in radial and tangential direction by the carrier. The constraints in axial direction, against tilting or torsion can be activated independently.

2.1.2.5.11 Axial stiffness

The axial stiffness can be used to define a nonlinear spring in axial direction. An axial clearance can be used to define clearance or pretension, the data in the table has to start with (0, 0). If a diameter is defined, the axial stiffness will act as distributed stiffness on this diameter, therefore the support can also support tilting moments if loaded axially.

The axial stiffness will be set to a very large

Axial stiffness

Name Axial stiffness

Position x 0 mm

Connect to housing

Stiffness to the right

Axial clearance Pa 0 mm

Diameter D 20 mm

	u [mm]	F [N]
1	0	0
2	1	500
3	2	2000

value behind the last point for displacement in the table.

2.1.2.5.12 Cylindrical support

The cylindrical support allows to define a radial clearance over some widths and an axial clearance between two parts. It can be used to model a ring on or in a shaft. The constraint can be rigid or a stiffness can be defined. For the axial constraints a diameter can be defined so it also takes a moment load using a distributed axial stiffness.

Cylindrical constraint

Name

Position x mm

Connect to housing

Width b mm

Diametral clearance Pd mm

Radial stiffness Cr N/mm

Consider temperature for radial clearance

Shaft is supported against torsion

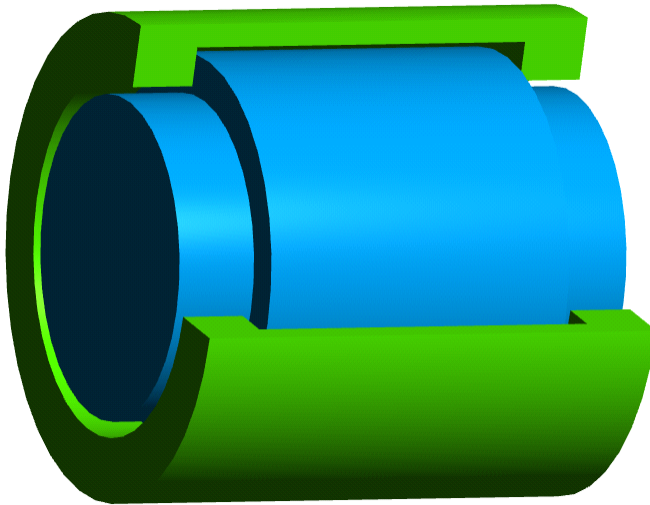
Shaft is supported axially to the left

Axial clearance PaL mm

Diameter DL mm

Axial stiffness CaL N/mm

Shaft is supported axially to the right



2.1.2.5.13 Excitations

Excitations can be defined for supports which are connected to the rigid environment. They are considered in calculation of harmonic response.

Housing:S1

Base excitation

	f [Hz]	phase [°]	ux [mm]	uy [mm]	uz [mm]	rx [rad]	ry [rad]	rz [rad]
1	100	0	0	0.001	0	0	0	0

Excitation based on shaft rotation

	h	phase [°]	ux [mm]	uy [mm]	uz [mm]	rx [rad]	ry [rad]	rz [rad]
1	1	0	0	0.001	0	0	0	0

OK Cancel

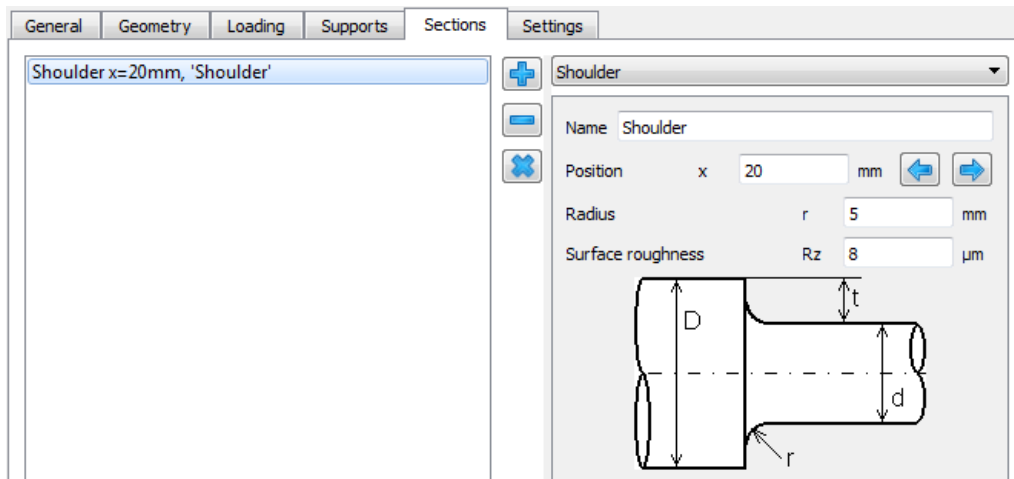
Several types of excitations can be considered:

- A base excitations is a harmonic displacement or rotation of the node in the rigid environment. It has to be defined by giving a frequency. It could be used for vibrations of the grounding.
- An excitation based on shaft rotation will rotate with the shaft. It can be used to model roundness errors. A factor for the harmonic is defined and a factor of one is an excitation with the shaft rotation speed.
- An excitation based on inner ring rotation can be defined for rolling element bearings in case of rigid bearing rings. Here also a harmonic is defined and the excitation rotates with the inner ring.
- An excitation based on outer ring rotation can be defined for rolling element bearings in case of rigid bearing rings. Here also a harmonic is defined and the excitation rotates with the outer ring.

2.1.2.6 Sections

Using "Sections" several cross sections can be defined for the strength calculation. One type of section is "Documentation point" which can be used without the module for strength calculation. In this case just displacements and forces are documented in the report, no strength calculation according DIN 743 is performed.

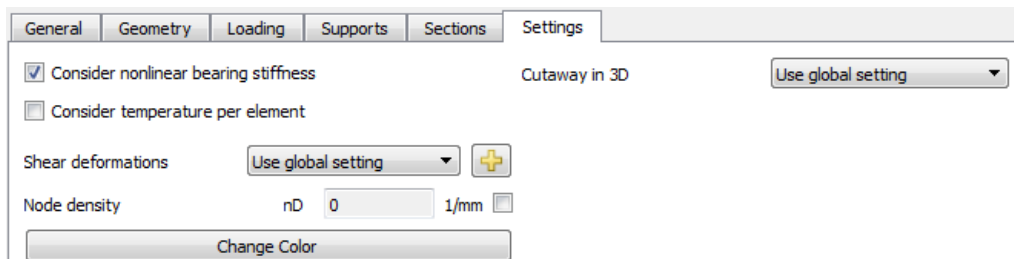
Dependent on the selected type of cross section several inputs are necessary to define details of the notch effect. All types from (DIN 743-2, Dezember 2012) are supported.



The "Documentation point" has the setting "Calculate stiffness". If the option is activated the stiffness in 6 degrees of freedom for this point of the shaft is shown in the results overview and the report. The stiffness is the reciprocal of the main diagonal of the compliance matrix or the system stiffness reduced to this point. So it is for example $c_{xx} = dF_x / d_{ux}$ for constant other forces and moment F_y, F_z, M_x, M_y, M_z .

2.1.2.7 Settings

Some settings can be entered for a single shaft:



2.1.2.7.1 Consider nonlinear bearing stiffness

As default, the nonlinear bearing stiffness is considered for rolling bearings. For large systems this option can be cleared to decrease calculation time. Without this option, bearings are considered as rigid. The option is available for each shaft, so it can be set where bearing stiffness is important and can be cleared on other shafts.

If bearings with contact angle are used, the nonlinear stiffness matrix should be considered to take into account the coupling of axial and radial loads.

2.1.2.7.2 Node density

The user can overwrite a node density for the FEA model. For a static linear calculation, the software adds additional nodes for improved resolution of diagrams. Results are only affected for conical geometry not for cylindrical elements. For modal analysis the number of nodes can make a difference.

Reducing the node density decreases calculation time.


2.1.2.7.3 Change color

The color of the shaft can be changed. The transparency is only considered on 3D view, not in 2D view.

2.1.2.7.4 Consider temperature per element

By setting this flag, the user can add a temperature difference ' ΔT ' (\pm) for each element created at the outer geometry on the temperature previously set at the tab-page 'General'. The temperature difference can also be varied within a load spectrum.

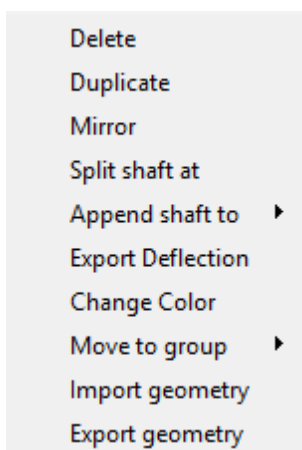
2.1.2.7.5 Shear deformations

Here it is possible to define which shear coefficient value (kappa) is being taken into account for the selected shaft. By default, the one defined at the global settings is used. The other approaches for the calculation of kappa can be selected from the drop-down list. If 'user input for kappa' is selected, then the user has to press the -button in order to enter the desired value.

2.1.2.7.6 Cutaway in 3D

For each shaft the cutaway setting for the 3D view can be set independently of the general option selected under the tab-page 'Display settings'.

2.1.2.8 Options in system tree




The context menu with the right mouse button on the shaft in the system tree provides some additional options:

- "Delete" deletes the shaft.
- "Duplicate" duplicates the shaft. This is helpful if several similar shafts are needed in a gearbox.

- "Mirror" mirrors the shaft geometry and positions of forces and supports. The orientation of forces and supports is not changed.
- "Split shaft at" asks for a position on the shaft and splits the shaft into two shafts.
- "Append shaft to" appends the shaft to another shaft in the same group.
- "Export deflection" exports a *.csv file for the deflection of a shaft. The "Result tables" in menu report also contain this information, but for all shafts instead of one only.
- "Change color" allows to change the color of a shaft.
- "Move to group" allows to move a shaft into another group. This is often used after "Duplicate".
- "Import geometry" allows an input of shaft geometry as 2D DXF or 3D STEP, see below for details.
- "Export geometry" exports the shaft as 3D STEP or IGES. Gears and bearings are included on the export.

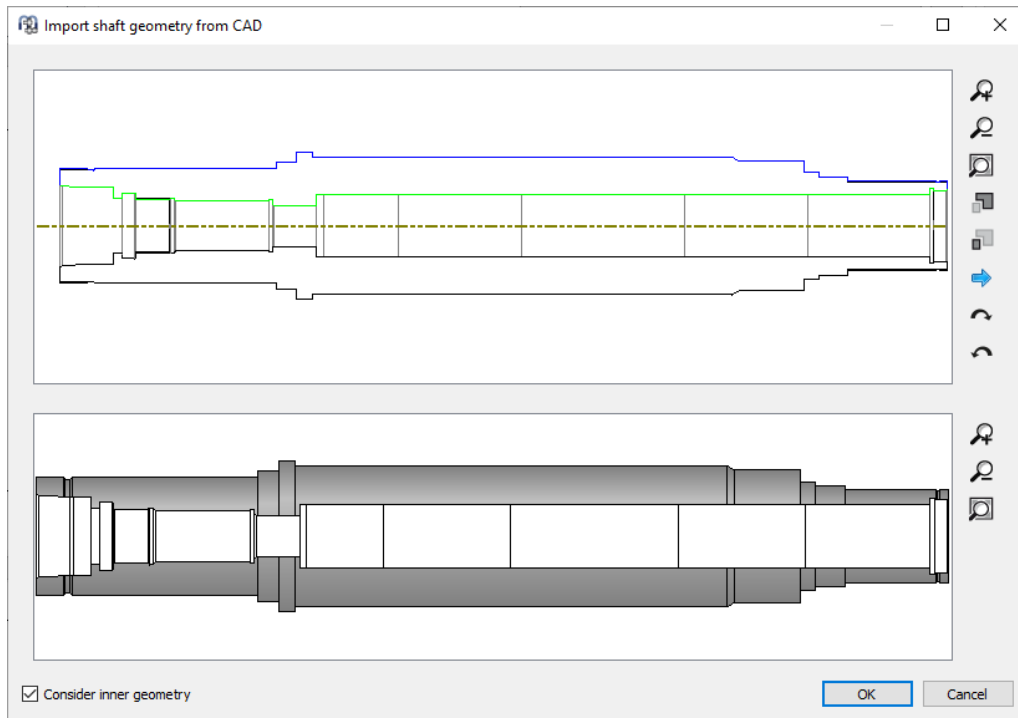
2.1.2.8.1 Import of shaft geometry from CAD files

The shaft geometry can be imported from 2D DXF or 3D STEP files. In case of a 3D-STEP file, the x-axis has to be the axis of the shaft and it has to be only one part in the file. An intersection of the part in the x/y-plane is done and the resulting edges are considered as in the DXF input. Relief grooves in the CAD model can lead to many imported elements.

For the DXF file a normal drawing can be selected. By using the left mouse button, some geometry can be selected. The selected geometry can either be deleted or it can be cut. This allows to select the correct geometry for the import. The -button allows to switch through different layers within the DXF file.

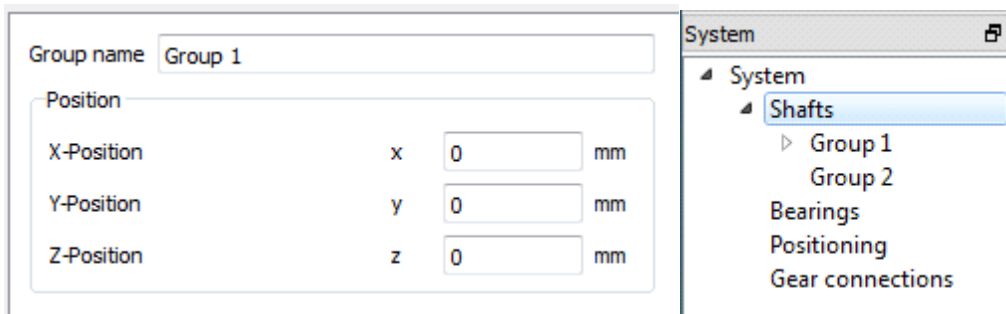
If an inner geometry should be imported too, the flag at the bottom of the window has to be activated.

The DXF file only contains lines. Sometimes the software cannot detect the correct connections between these lines. In this case the wrongly used lines can be selected and deleted. The lower part of the window shows the currently detected geometry input.



2.1.3 Shaft Groups

For systems with shafts that are not coaxial, shaft groups have to be added. For each group a position can be defined.



The positions can also be calculated by the software if constraints are used for positioning. See below.


There are three types of groups:

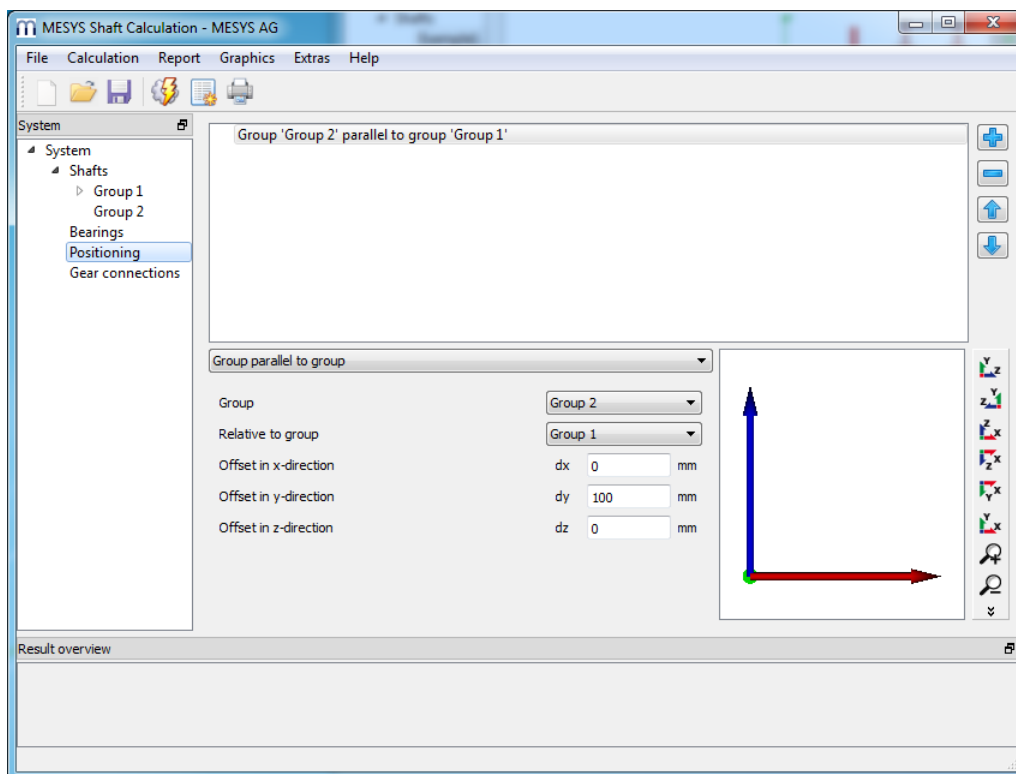
1. Standard group: The standard group can have an arbitrary position and it can hold subgroups. Shafts in different groups can not be connected by supports.
2. Coaxial group: The coaxial group can only be positioned axially relative to its parent group. Shafts in different coaxial groups can be connected by rolling bearings or other supports.

- Planetary group: A planetary group is needed to define planetary gear sets. The number of planets can be defined in addition to position. Shafts in a planetary group rotate around a planet carrier. The centrifugal loads caused by this rotation are considered. For bearings and other supports in planetary groups the result for each planet is shown in the results.

2.1.4 Positioning

If multiple groups are used, an additional entry for positioning is shown in the system tree.

The page “Positioning” can be used to define constraints for group positions. New constraints are added using the -button, they can be sorted using the arrows. The constraints are evaluated once per calculation, so the order can be important.




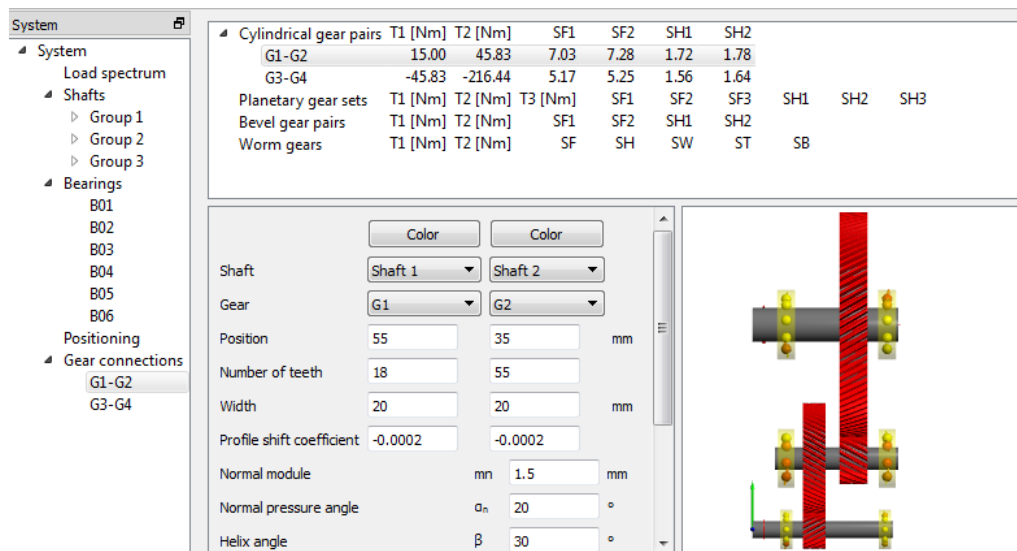
Different constraints are available:

- Group parallel to group: Specify an offset between two groups.
- Group according to gear pair: The selected group is positioned according to the selected gear pair. An additional angle has to be specified, zero is for a center distance on the y-axis. An axial offset can be entered if the centers of the gears should not align axially.
- Group according to two gear pairs: One group is positioned according to the center distances of two gear pairs. The axial position is defined by the first gear pair.

- Group according bevel gear pair: One group is positioned according to geometry data of a bevel gear. The angle of contact for the reference group can be defined.
- Shaft according to gear pair: This constraint is moving a shaft axially, so that the centers of the gears are aligned. An axial offset can be entered optionally.
- Gear according to gear pair: This constraint is moving a gear axially so that it is aligned with its mating gear. Here an axial offset can be entered optionally, too.
- Group according worm gear pair: The selected group is positioned according to the selected worm gear pair. An additional angle has to be specified, zero is for a center distance on the y-axis. An offset can be entered axially to the worm.

2.1.5 Gear connections

If multiple groups are defined, a page for gear connections is shown. Gear connections can be added using the -button. The shafts and gears in contact can be defined and the basic data for the gear pair is shown. The data for the gears can be modified in this window in addition to the inputs at the single shaft but using the connection allows to change data for both gears at the same time.



Cylindrical gear pairs	T1 [Nm]	T2 [Nm]	SF1	SF2	SH1	SH2
G1-G2	15.00	45.83	7.03	7.28	1.72	1.78
G3-G4	-45.83	-216.44	5.17	5.25	1.56	1.64

Planetary gear sets	T1 [Nm]	T2 [Nm]	T3 [Nm]	SF1	SF2	SF3	SH1	SH2	SH3
Bevel gear pairs	T1 [Nm]	T2 [Nm]		SF1	SF2	SH1	SH2		
Worm gears	T1 [Nm]	T2 [Nm]		SF	SH	SW	ST	SB	

Configuration for G1-G2:

- Shaft: Shaft 1, Shaft 2
- Gear: G1, G2
- Position: 55, 35 mm
- Number of teeth: 18, 55
- Width: 20, 20 mm
- Profile shift coefficient: -0.0002, -0.0002
- Normal module: 1.5 mm
- Normal pressure angle: α_n 20 °
- Helix angle: β 30 °

The torque on each gear pair is shown in the list at the top; the safety factors are shown if a connection to a gear calculation program is set up.

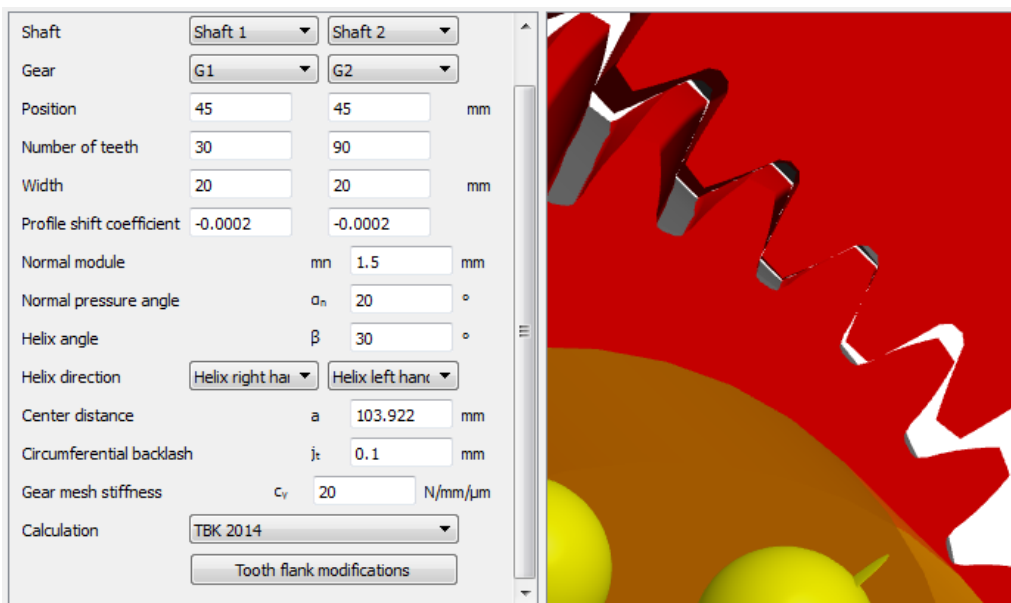
With a click on the items for the gear pairs in the system tree on the left the gear calculation program can be started to define details for the gear pair. The shaft calculation is passing basic gear geometry (z , b , x , m_n , α_n , β , x , a) to the gear calculation and loading; it is reading back the same geometry data and the safety

factors. The colors of the gears can be changed by clicking on the “Color” button shown in the picture.

If a gear is considered in a connection, its torque input is hidden at the corresponding input window under the page ‘Loading’.

2.1.5.1 Tooth flank modification

With the goal of ensuring optimum tooth flank contact, the teeth of gear pairs can undergo flank corrections based on three different approaches according to the standard (ISO 21771:2007, September 2007), i.e. flank line crowning, flank line slope modification and flank line end relief. By pressing on ‘Tooth flank modifications’ at the page ‘Gear connections’, a pop-up window allows the user to enter the corresponding geometrical data in relation with these approaches. For each gear it is possible to have either symmetric or unsymmetrical modifications by setting or clearing the available flags respectively at the bottom. By activation of an unsymmetrical flank modification, the corresponding data column is added in the window.



Tooth flank modifications

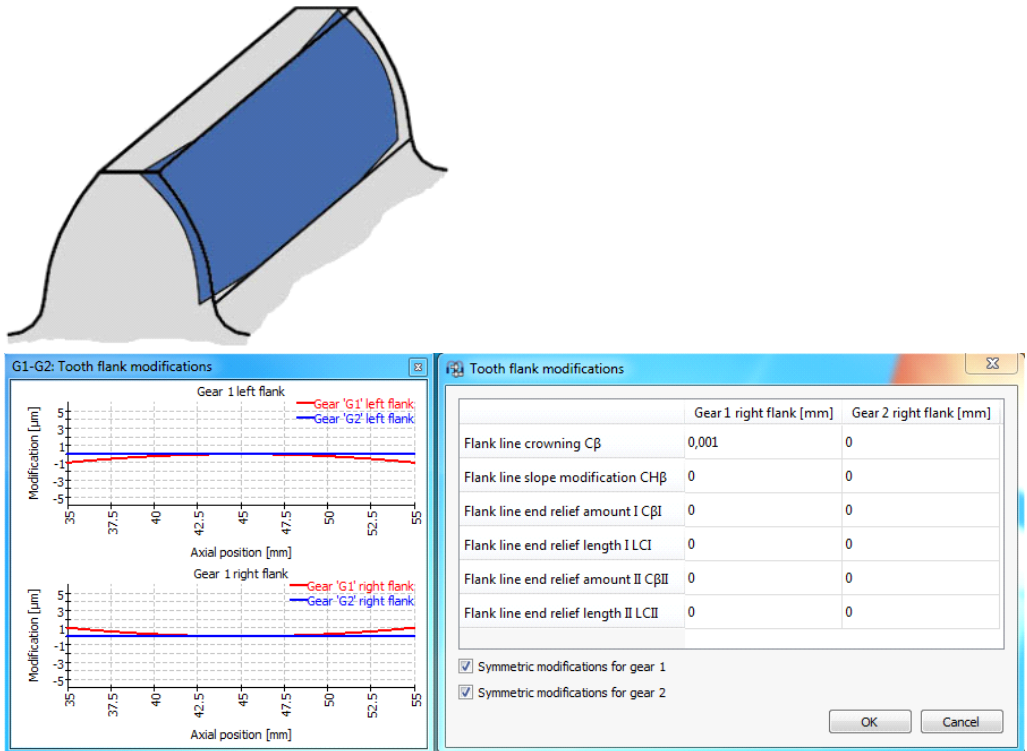
	Gear 1 right flank [mm]	Gear 1 left flank [mm]	Gear 2 right flank [mm]	Gear 2 left flank [mm]
Flank line crowning $C\beta$	0	0	0	0
Flank line slope modification $CH\beta$	0	0	0	0
Flank line end relief amount I $C\beta I$	0	0	0	0
Flank line end relief length I LCI	0	0	0	0
Flank line end relief amount II $C\beta II$	0	0	0	0
Flank line end relief length II $LCII$	0	0	0	0

Symmetric modifications for gear 1
 Symmetric modifications for gear 2

OK Cancel

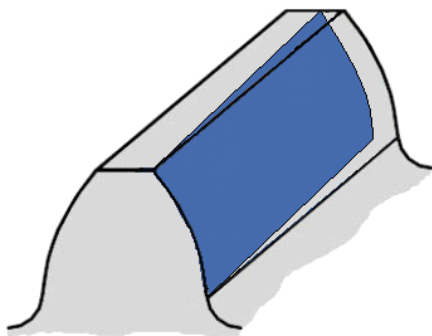
2.1.5.1.1 Flank line crowning

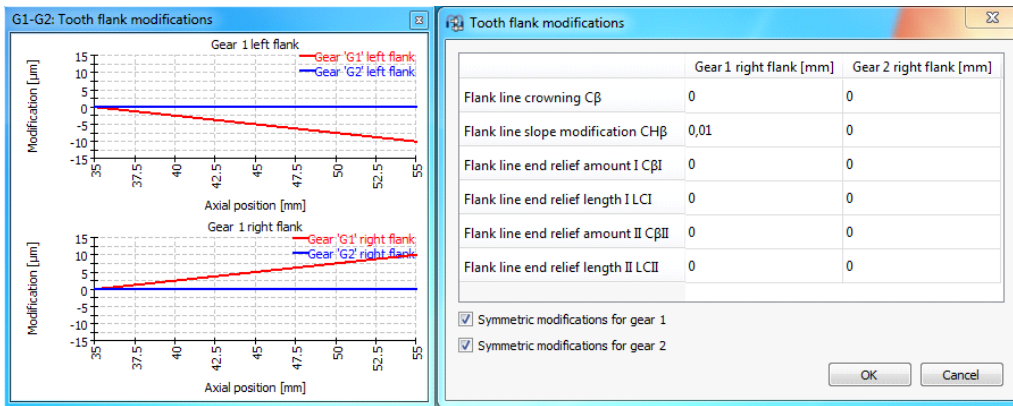
It consists of a relief based on a circular arc which is tangent to the mid-point of the tooth width. The relief amount, which is measured from the end of the tooth, must be specified.



2.1.5.1.2 Flank line slope modification

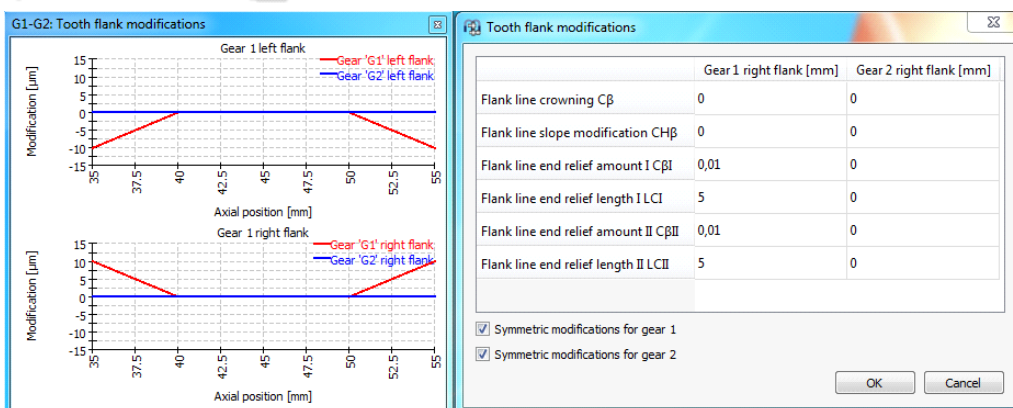
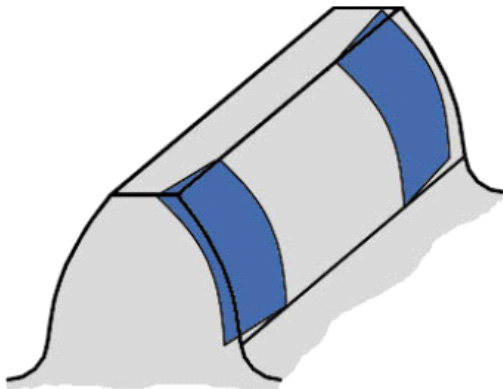
It consists of an increasing linear relief over the whole tooth width. So the slope of the relief line starting from one end to the other must be specified.



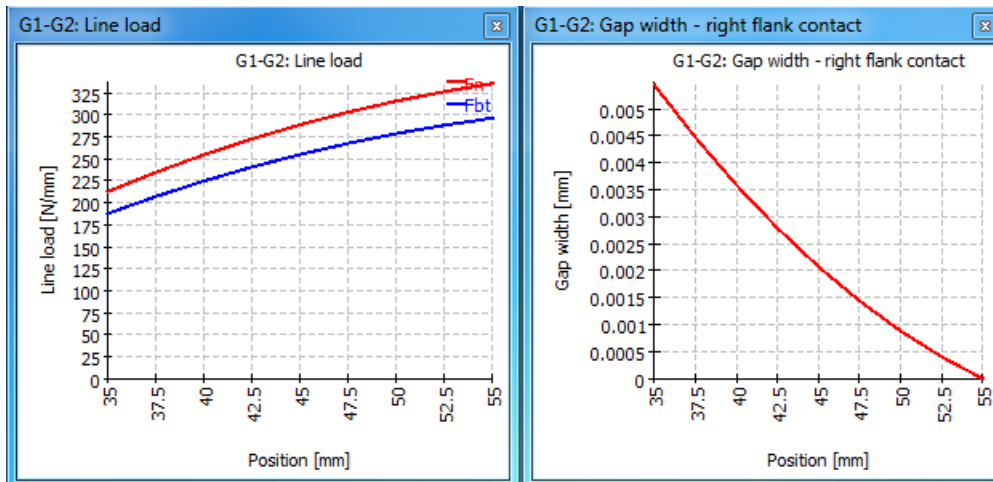


2.1.5.1.3 Flank line end relief

It consists of an increasing linear relief starting from a certain point until the end of the flank line. The relief amount and length, which are measured from both ends of the tooth, must be specified.



All in all, the user can make use of the line load and gap width graphics as guidance to establish a flank correction proposal. If, for instance, we start from a gear tooth mesh condition like this:

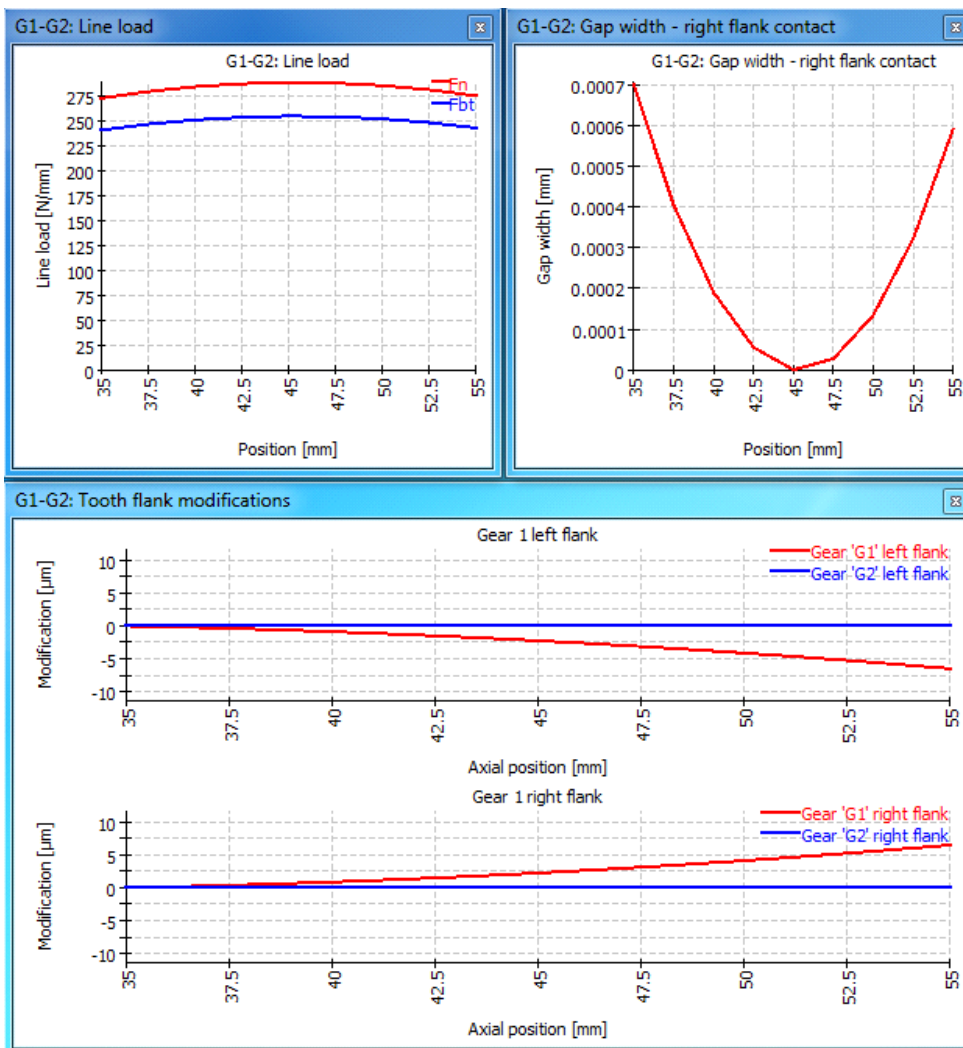


	Gear 1 right flank [mm]	Gear 2 right flank [mm]
Flank line crowning $C\beta$	0,001	0
Flank line slope modification $CH\beta$	0,0065	0
Flank line end relief amount I $C\beta I$	0	0
Flank line end relief length I LCI	0	0
Flank line end relief amount II $C\beta II$	0	0
Flank line end relief length II $LCII$	0	0

Symmetric modifications for gear 1
 Symmetric modifications for gear 2

OK Cancel

And we apply a combination of flank line crowning and flank line slope modification, it significantly helps to have an homeogenous line load as shown in the following pictures:

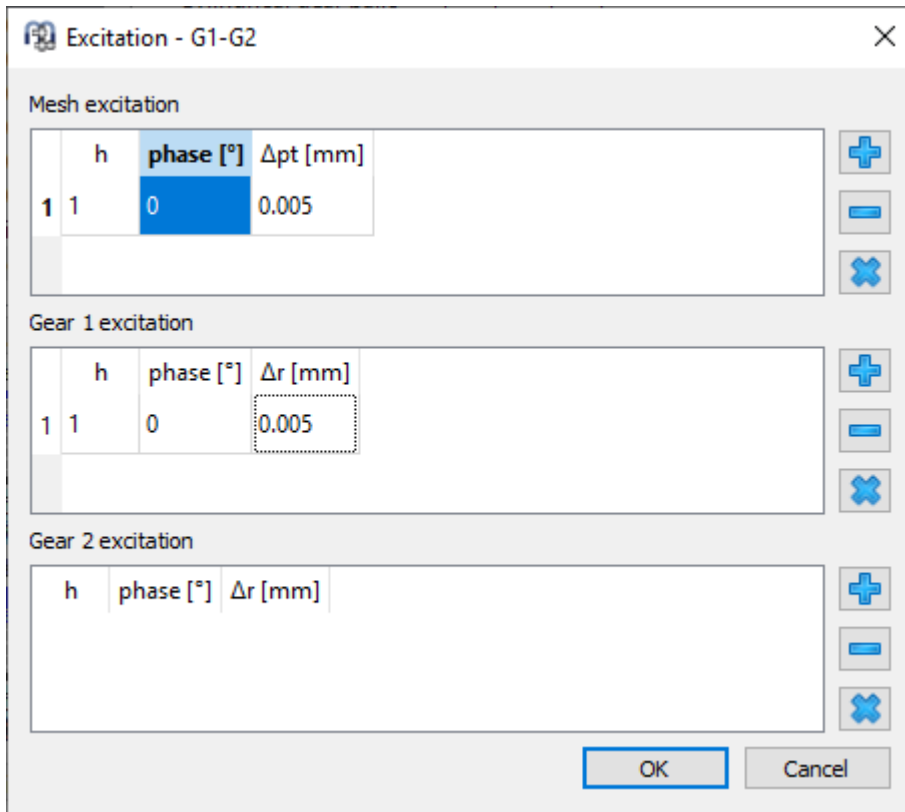


2.1.5.2 Excitations

For cylindrical gear pairs three types of excitations can be defined:

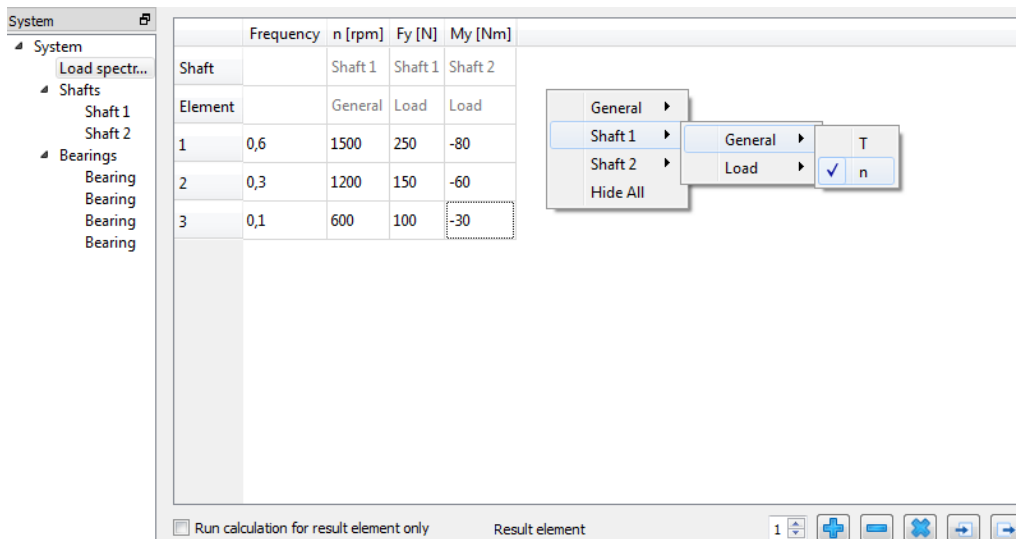
- A mesh excitation is defined on the path of contact. Pitch errors will lead to excitations based on gear mesh frequencies. Higher orders can be defined using the input of harmonic.
- An excitation based on roundness errors of the gear 1 can be defined. Here a variation of the radius is used. The base frequency is the rotation speed of the gear.
- An excitation based on roundness errors of the gear 2 can be defined like for gear 1.

The excitations are considered in the calculation of harmonic response. They will not affect the results of quasi-static calculation and will not affect safety factors for gears.




2.1.6 Calculation with load spectrum



If the flag for the calculation with load spectrum is set on the 'System' page, an additional item titled load spectrum is shown in the system tree.



By default, the table is empty. Only the column header for the load frequency is shown. By right-clicking somewhere on the window, different calculation parameters for the load cases can be added optionally as column headers. These parameters are classified as "General" and "Load". Concerning the rows in the table, they can be added using the -button and also deleted by selecting it and using the -button. These rows are called "Element" and are taken into account as


a load case where the values (inputs) corresponding to the calculation parameters can be entered. The -button clears all the inputs of the table, also the elements (rows).

You can hide any header by right-clicking on it and selecting "Hide". For all load components that are not shown in the table the default value is used.

By using the -button, the load spectrum is read from a file. The columns in the file have to be in the same order and unit as shown in the table. Likewise, a created load spectrum table can be exported with the corresponding -button.

The default graphics for deflection are shown for the selected "Result element". Natural frequencies are also calculated for this "Result element".

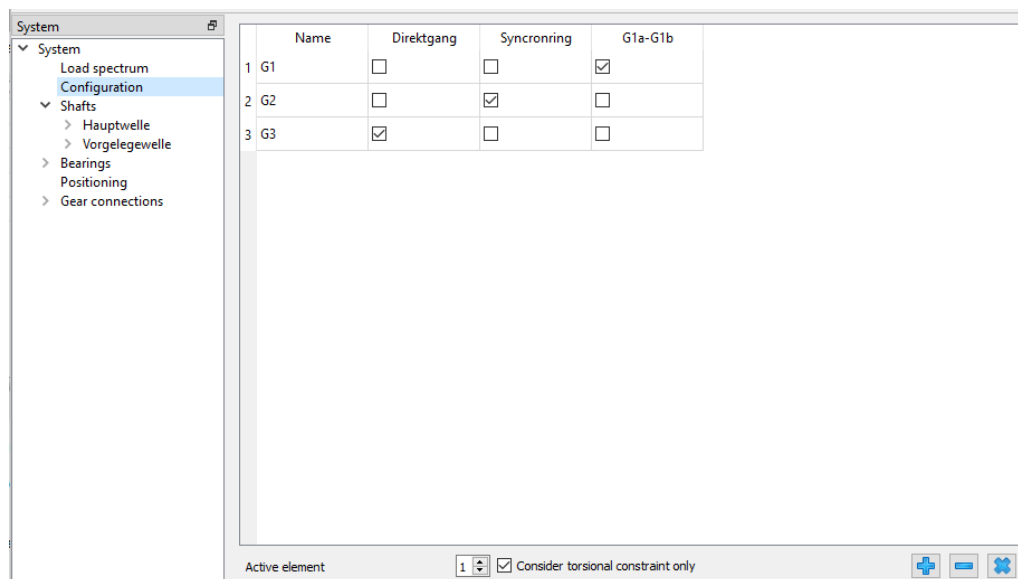
2.1.6.1 Running calculation for one load spectrum element only

It is possible to run the calculation for only one element (load case) if the user sets the flag 'Run calculation for result element only'. The desired load case is chosen by either using the arrows of the small box next to the -button or typing the element number into it directly. The graphics will be only shown for the selected element.

This allows using the load spectrum calculation as input interface for independent load cases, which are calculated independently.


2.1.7 Configurations

Configurations can be activated on the first input page for settings if multiple groups are defined.



Using the right mouse button additional columns can be added to the view. Available are gear connections, reaction couplings, general constraints, supports

and cylindrical constraints. The constraints are only available if the torsion is fixed in the constraint.

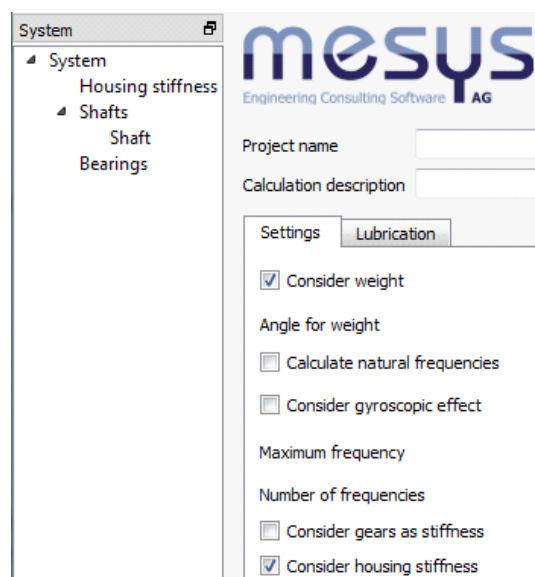
Rows can be added using the -button and with the check box the active elements for each configuration can be selected.

If the option "Consider torsional constraint only" is active, clearing the flags for the constraints only disables the torsional constraint of general constraints, constraints in all other directions are not affected. If the option is not active the constraints is deactivated for all directions.


The active configuration can be selected at the bottom of the window or within a load spectrum.


2.1.8 Housing stiffness matrix

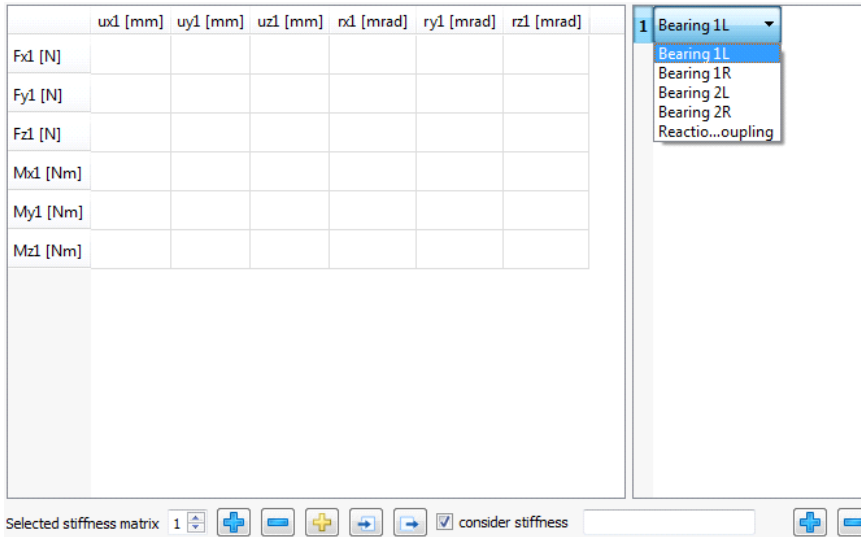
A symmetric stiffness matrix can be defined between the supports (only those which are connected to the housing) and the housing, thus acting as boundary condition at its connecting point. This enables the user to simulate different ways of stiffness interaction, also between the bearings or supports by means of the housing.




In order to make use of this option, the corresponding flag of 'Consider housing stiffness' must be activated at the tab-page 'Settings' which it is shown when selecting 'System' from the software tree.

The -button, located under the matrix, can be used to create any additional type of stiffness matrix, which is referenced with a number shown in a box titled 'Selected stiffness matrix'. Any one or all of these created matrices can be excluded for the calculation at any time by using the flag titled 'Consider stiffness'. In this way, any matrix data can be kept, even though it is not being used for the

analysis. In case we do want to erase any of the matrices, the -button can be employed for it. The entry field can be used to name them and their names will be shown in the report.






By clicking once the -button on the right lower corner, a new numbered element is listed on the window above it and a matrix is shown on a bigger window. With a double click on this element a drop-down list is released letting us choose the support to consider. By repeating this procedure, we could add other existing bearings or supports on the list, and therefore their corresponding lines and columns of the stiffness matrix would be shown and taken into account.

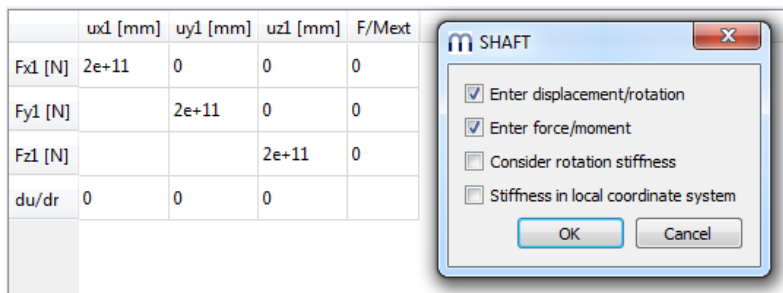
	ux1 [mm]	uy1 [mm]	uz1 [mm]	ux2 [mm]	uy2 [mm]	uz2 [mm]
Fx1 [N]	2e+11	0	0	0	0	0
Fy1 [N]		2e+11	0	0	0	0
Fz1 [N]			2e+11	0	0	0
Fx2 [N]				1.5e+11	0	0
Fy2 [N]					1.5e+11	0
Fz2 [N]						1.5e+11

1 Bearing 1L

2 Bearing 1R

All the values for the stiffness can be entered manually with double click on the cells or also be loaded as a matrix, which is sometimes obtainable from a finite element analysis, by using the import button . Equally, a created matrix can be exported with the corresponding export button .

If we use the -button, a pop-up window allows activating four additional options. As shown in the following picture, we can for instance omit the consideration of rotation stiffness, and activate the possibility to define for the connecting point, in any available direction required, a force or a moment with 'Enter force/moment', and also a certain displacement or rotation with 'Enter displacement'.



The usage of the last option lets us to consider the stiffness in the local coordinate system of the connecting point instead of the global coordinate system.

2.1.9 3D elastic parts

The context menu under “Shafts” or under groups allows to insert “3D elastic parts as shaft” or “3D elastic parts as housing”. A new part will then be shown in the shaft/group tree.

The 3D elastic parts combine the FEA beam model used for the usual shafts with 3D solid meshes. The stiffness of the 3D solids is reduced to a stiffness matrix, which is then considered in the solver for the beam model. On the first use of the 3D elastic part the reduced stiffness matrix is generated, which can take a while dependent on the size of the mesh. Afterwards the reduced stiffness is saved with the file and only needs to be regenerated if material, mesh or selections for faces are changed.

The 3D elastic parts are considered as stiffness and optionally by modal reduction. No mass is taken into account without modal reduction. Thermal expansion and weight are considered. A “3D elastic parts as shaft” must have the x-axis as rotation axis and a rotation speed can be defined. A “3D elastic parts as housing” can not rotate and should be fixed in space.

If the option “calculate displacements” for the 3D elastic parts is selected, the displacement field of the surface is calculated for each connected degree of freedom on initial calculation of reduced stiffness. This increases the initial calculation time and the file size, but allows to visualize the deformed shape afterwards. The same is valid for the option “calculate mode shapes”; this is used for visualization only, too.

In case modal analysis is supported, the displacement field and the mode shapes need to be calculated. Therefore options “calculate displacements” and “calculate mode shapes” won't affect the required time for reduction. They only affect the required file size.

If the option "use second order mesh" is activated quadratic shape functions are used instead of linear shape functions. An eight node hexahedral will get a 27-node hexahedral instead. As the number of nodes increase, reduction time and

memory usage will increase too. Generally the second order mesh will be more accurate. It is better to use a larger mesh size and a second order mesh instead of a fine linear mesh.

Selected faces are reduced to one node (or elastic bearing or gear) and can be connected to a support element. Following options are available:

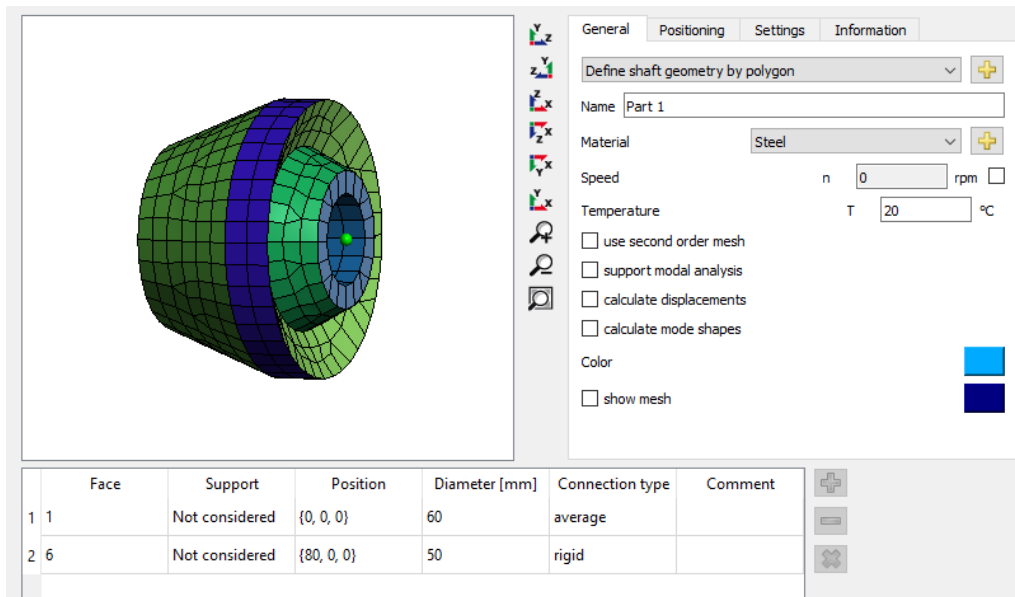
- An existing support can be selected. This support is then connected to the 3D-elastic part instead of the rigid environment
- A new support can be defined as possible for shafts. Only a subset of the available types in the shaft calculation is supported so far. It can be used to support an imported housing by a stiffness.
- A force can be added on the reduced node
- A shaft can be selected. This will lead to a rigid connection to the shaft in all six degrees of freedom. It is the same as adding a weld-point to the shaft and connection this to the 3D-elastic part.
- "Set as fixed" is available for "3D-elastic parts as housing". This node will be connected to the rigid environment.

For the reduction of a face four options are available as connection type:

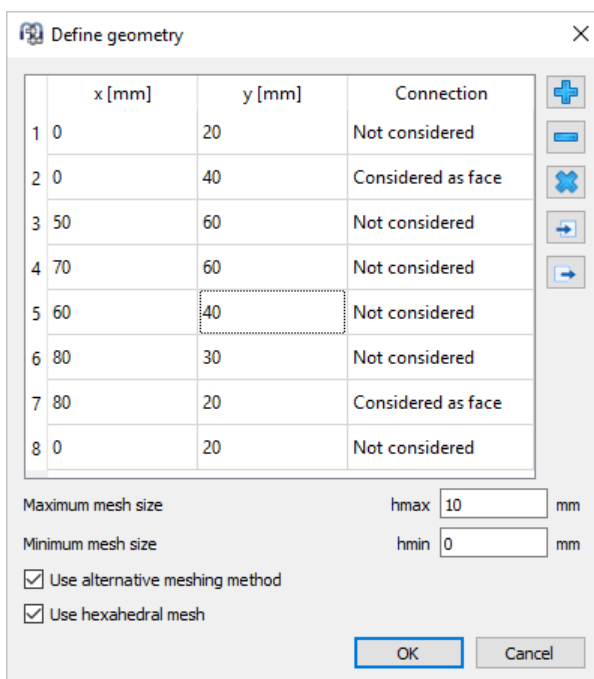
- average: The reaction force will be distributed on all nodes of the face and an average value of the displacement of the face will be used. This option will not stiffen the part.
- rigid: A rigid connection of the reduced node to all nodes of the face will be added. This will stiffen the part.
- elastic bearing: This option may only be connected to a roller bearing and will consider the elastic deformations of the part in the load distribution of the bearing. A license for the bearing calculation software including ring deformations is required. Currently an elastic outer ring is only permitted for fixed bearings.
- elastic gear: This option may only be connected to a cylindrical gear so far and will consider elastic deformations of a gear body. The shaft system option is required for this

2.1.9.1 3D elastic part as shaft

A 3D elastic part as shaft is a rotating part with one axis which can be connected to other shafts. Currently, axisymmetric parts defined by a polygon or a STEP import can be considered.



Using the **+**-button behind “Define shaft geometry by polygon”, a dialog opens to define the geometry using a polygon. It should be defined clockwise. “x” is the axial coordinate, “y” the radius. Optional connections to normal shafts can be defined as “Considered as face” or “Considered as edge”. Either all points on a face or only points on an edge will be connected to a shaft support element.



In addition, the mesh size can be defined in this dialog.

Additional settings are the material, the rotation speed, the option to select a second order (quadratic) mesh and the option if the displacements should be calculated. Using a quadratic mesh is recommended for better accuracy. The calculation of displacements increases the calculation time and would only improve the deformed shape in the 3D view.

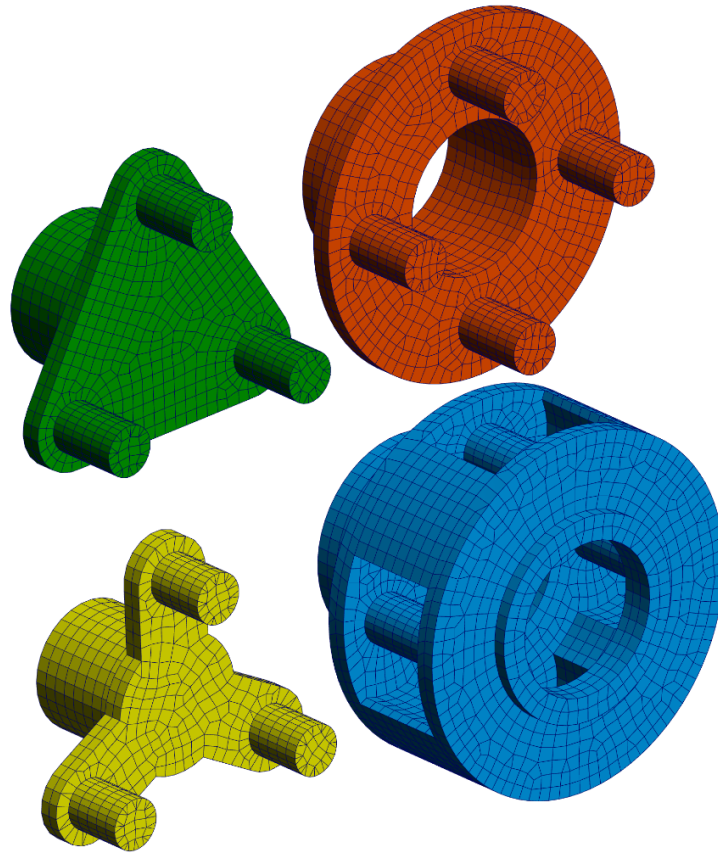
Under "Positioning" the axial position can be defined.

In the table of the main view the support elements of the shafts have to be selected under "Support". The connection type "rigid" enforces a rigid connection of the selected face/edge to the beam node of the shaft. The connection type "average" enforces an averaged connection to the node, which leads to less stiffening of the 3D elastic part.

In case of a STEP import faces can be selected by double click in the 3D-view. Faces can be merged by selecting two faces, marking them in the table and selecting "Merge faces" in the context menu (right mouse button). Only neighbouring faces can be merged. For an imported shaft the x-axis must be the rotation axis. Under Positioning the orientation can be changed selecting "Permanent coordinate transformation". If faces are inside the part a clipping of front faces can be activated using the context menu in the 3D-view. Using Alt-Button and left mouse button the clip-plane can be moved.

2.1.9.2 3D elastic planet carriers

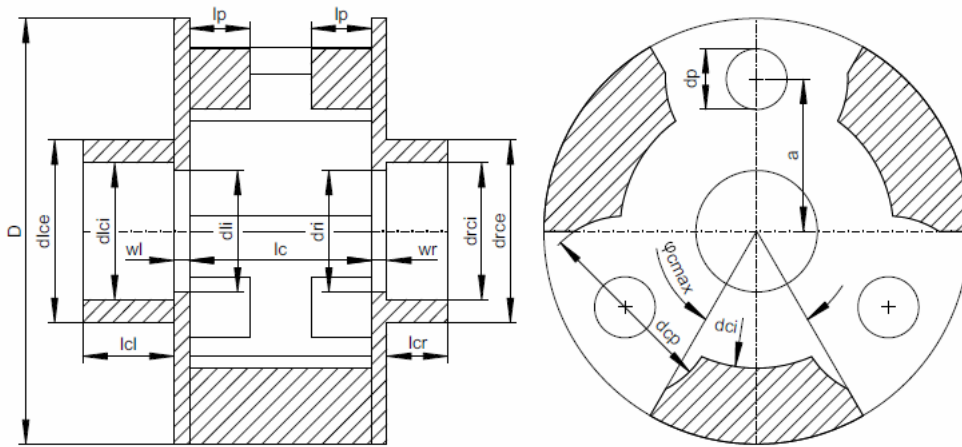
In case of a 3D elastic shaft, also planet carriers can be defined. Either imported as STEP file or as parametric planet carrier. For a parametric planet carriers, the following basic shapes are available:



For an imported planet carrier, the rotation axis has to be the x-axis. On page "Positioning" a permanent coordinate transform can be used to change the imported orientation to the x-axis as rotation axis.

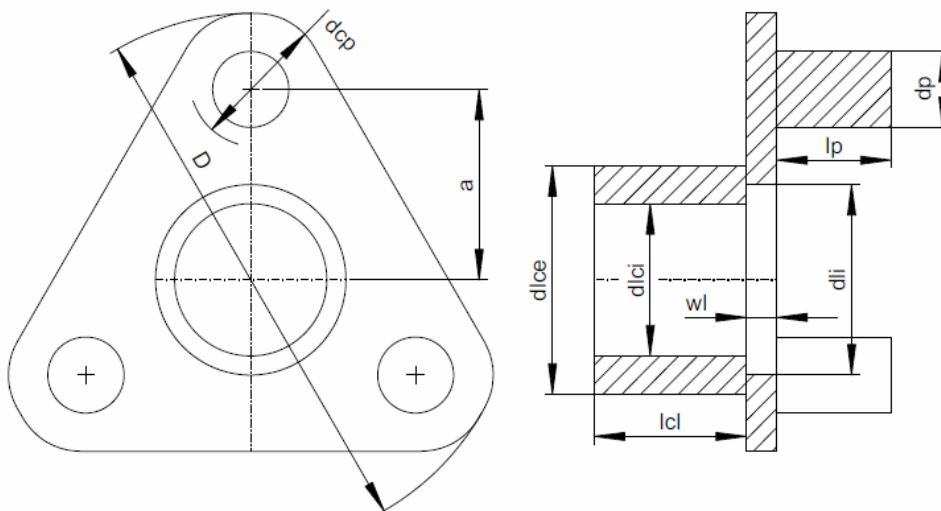
2.1.9.2.1 Parametric circular planet carrier

A parametric circular planet carrier can be defined either single or double sided. The geometry is shown in following image:



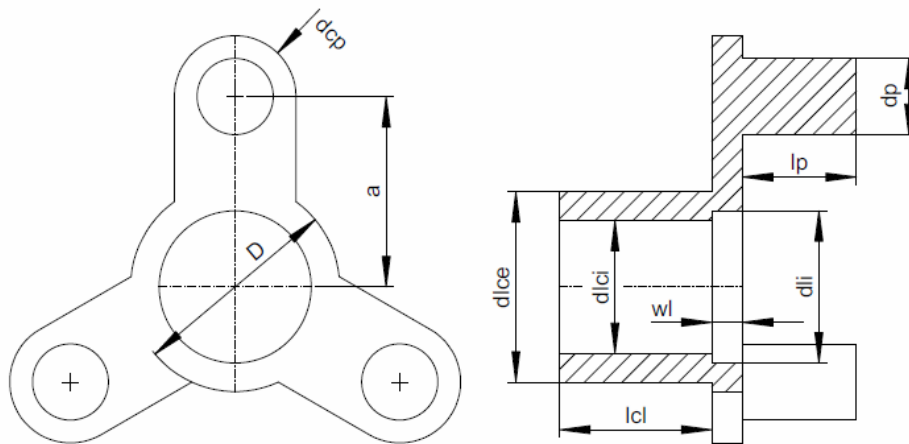
2.1.9.2.2 Parametric polygonal planet carrier

The geometry of the polygonal parametric carrier is shown in following image:



2.1.9.2.3 Parametric star-type planet carrier

The geometry of the star-type parametric carrier is shown in following image:



2.1.9.3 3D elastic part as housing

A 3D elastic part as housing can be connected to shafts in different groups, it does not rotate and at least one face has to be set as fixed. Currently, it can only be imported from CAD geometry as STEP file or as Nastran mesh.

The screenshot displays a 3D meshed model of a housing part with two circular cutouts. To the right, the 'Settings' panel is open, showing the following configuration:

- Settings: Positioning
- Import CAD geometry: [dropdown menu]
- Name: Part 1
- Material: Steel
- Temperature: T 20 °C
- use second order mesh:
- support modal analysis:
- calculate displacements:
- calculate mode shapes:
- consider stiffness:
- Color: [blue and dark blue swatches]
- show mesh:

Below the settings is a table defining the support conditions for the meshed part:

Face	Support	Position	Connection type	Comment
1 6	Set as fixed	{0.00141155, 0.000495987, -2.5}	rigid	Set bottom face as fixed
2 38	Not considered	{-39.9998, -33.5, 59.9998}	rigid	
3 39	Not considered	{-40.0005, 33.5, 60.0003}	rigid	
4 40	Not considered	{20.0015, -33.5, 60}	rigid	
5 41	Not considered	{20.0021, 33.5, 59.9993}	rigid	

Using the **+**-button behind “Import CAD geometry” a STEP file can be selected. An assembly with connected parts is glued together automatically. Afterwards the required mesh size can be entered. Alternatively, a Nastran mesh can be imported; only the surface mesh is kept in this case.

In the 3D view for the parts now multiple faces can be selected by mouse double click on a face. The faces can then be connected to a shaft support element or "Set as fixed". As above a rigid connection or an averaged connection can be selected. To merge two connected faces, select both faces in the table, press the right mouse button and select "Merge faces". If faces are inside the part a clipping of front faces can be activated using the context menu in the 3D-view. Using Alt-Button and left mouse button the clip-plane can be moved.

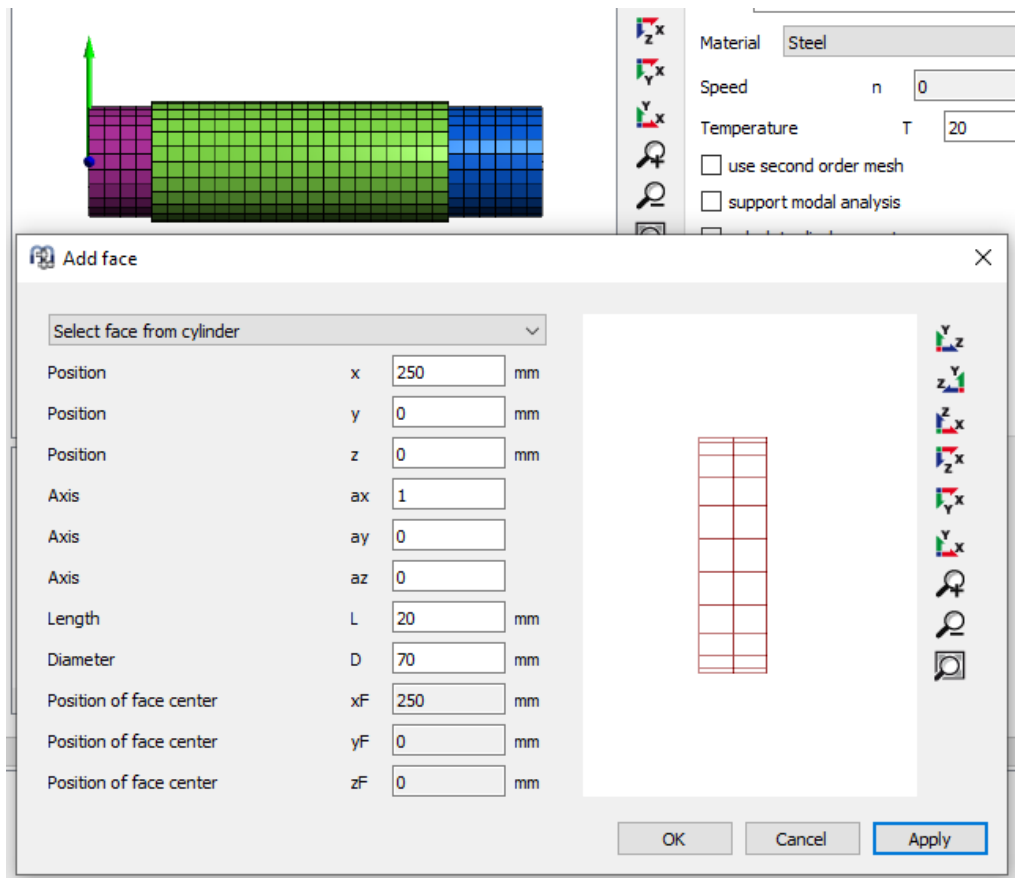
Under settings as before the material and options for a second order (quadratic) mesh and for calculation of the displacements can be selected. In addition, a 3D elastic housing can be supported in modal analysis; if "calculate mode shapes" is selected, the mode shapes can be shown, too. Under "Positioning" the position and rotation of the 3D part can be defined.

The selected support element will be connected to the nodes of the selected faces. A rigid or averaged connection will be considered, no contact and no clearance will be considered.

2.1.9.4 Selecting faces on imported geometry

After importing a STEP file or a mesh multiple faces can be selected by mouse double click on a face. To merge two faces, select both faces in the table, press the right mouse button and select "Merge faces". If faces are inside the part a clipping of front faces can be activated using the context menu in the 3D-view. Using Alt-Button and left mouse button the clip-plane can be moved.

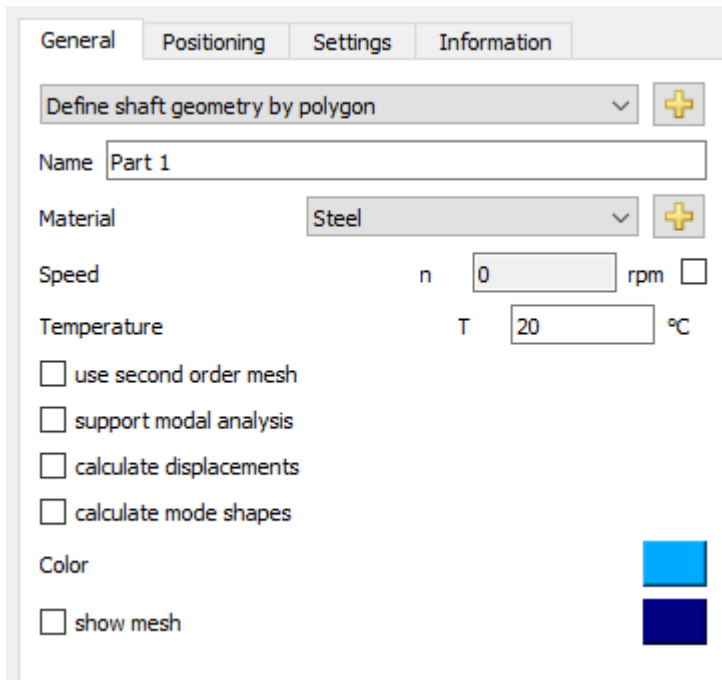
New faces can be defined using the context menu for the table and selecting "Add face". A new face can be selected based on a cylinder a bearing or a box:



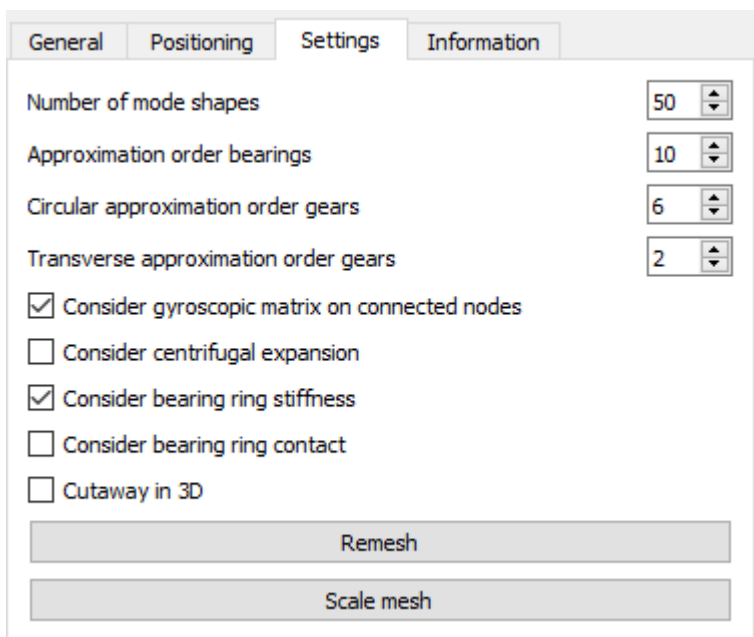
Define the position of the center of the cylinder and its diameter and length. Clicking "Apply" will select all elements in the range and will display the new elements in red. Existing selected faces are shown as mesh in black. For a box distances dx , dy , dz in 3 directions need to be defined instead of diameter and length. For a bearing, position, diameter and lengths are set automatically. This allows to add faces to an imported part without doing a new import.

2.1.9.5 Settings

There are some settings available for the 3D-elastic parts.



On page "General" the material, speed, temperature can be defined. The color for the part and the mesh can be defined which is used in the standard 3D-geometry views. A change of material required a new reduction for the part.



On page "Settings" the number of mode shapes used for modal reduction can be defined. A larger number increases accuracy but required more memory and reduction time. Usually a value larger 20 should be used. The maximum frequency is shown on page "Information". This frequency should be larger than the maximum frequency that is of interest for the system.

An approximation order for elastic bearing rings and elastic gears can be defined here. The transverse approximation order for gears of 2 means a parabola is used

for transverse deformations. A factor of 1 would only use a straight line. For gears with a large face width larger orders than 2 might be appropriate.

The reduction of the 3D-elastic parts lead only to mass and stiffness matrices. No gyroscopic matrix is considered. As approximation the polar inertia can be considered a gyroscopic matrix on the connected reduced nodes on the x-axis. The polar inertia will be distributed equally on all available nodes if the option "Consider gyroscopic matrix on connected nodes" is set.

The option "Consider centrifugal expansion" will add the effect of centrifugal expansion to the part. This is only active for 3D-elastic shafts.

The option "Consider bearing ring stiffness" will add the stiffness of the bearing rings to elastic bearings. If the ring is already included in the 3D-elastic part, this option should be disabled. Using "Consider bearing ring contact" will add a contact model between bearing ring and part. Then the fitting calculation will not directly set an operating clearance of the bearing, but the fitting will be influenced by expansion or compression of the elastic parts. If the stiffness of the part is not uniform, there will be already a deformation of the bearing ring after mounting.

Activating the option "Cutaway in 3D" will lead to a cut view in the 3D-view for the system like the same setting available for shafts.

A change of the numeric values on the settings page will lead to an update of the reduction, while changing the check boxes won't require an update of the reduction.

The button "Remesh" allows to remesh the part. Either to change the mesh size or the type hexahedral/tetrahedral. Note that this is a remesh of the previous mesh, generally it would be better to do a new import of the CAD file in case the mesh size is changed. "Scale mesh" allows to apply a scale factor on the mesh. This could help, in case the unit system of the import was not correct. The use of either button requires a new reduction of the part.

2.2 Calculation of natural frequencies and harmonic response

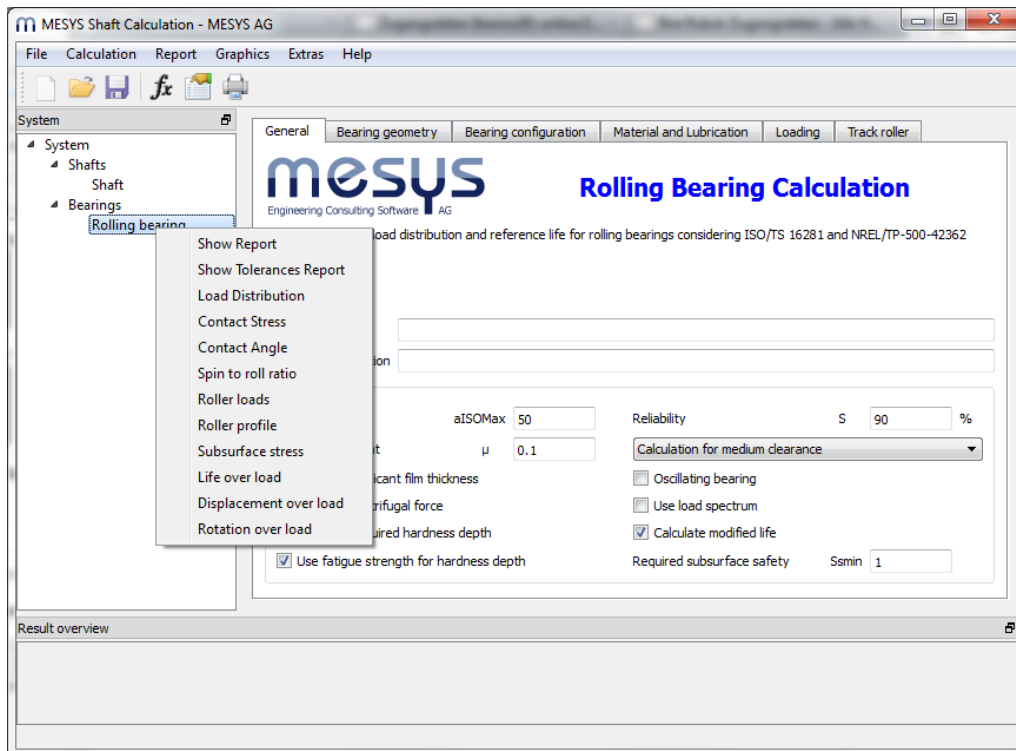
Optionally natural frequencies and harmonic response can be calculated using the software. The setting [Calculate natural frequencies](#)^[46] has to be activated for this.

First the nonlinear static equilibrium is calculated. Then the natural frequencies and the harmonic response are calculated using the linear equations of motion with the tangent stiffness matrices from the static equilibrium. Therefore the results are only valid for small amplitudes around the static solution.

The natural frequencies are calculated using the given rotation speed. The critical frequencies are the intersections between shaft speed and natural frequency at this shaft speed. The critical frequencies are shown in a special report or they can be found in the Campbell diagram.

2.3 Interface to rolling bearing calculation

The dialogs of the [MESYS Rolling Bearing Calculation](#)^[124] are fully integrated into the shaft calculation.

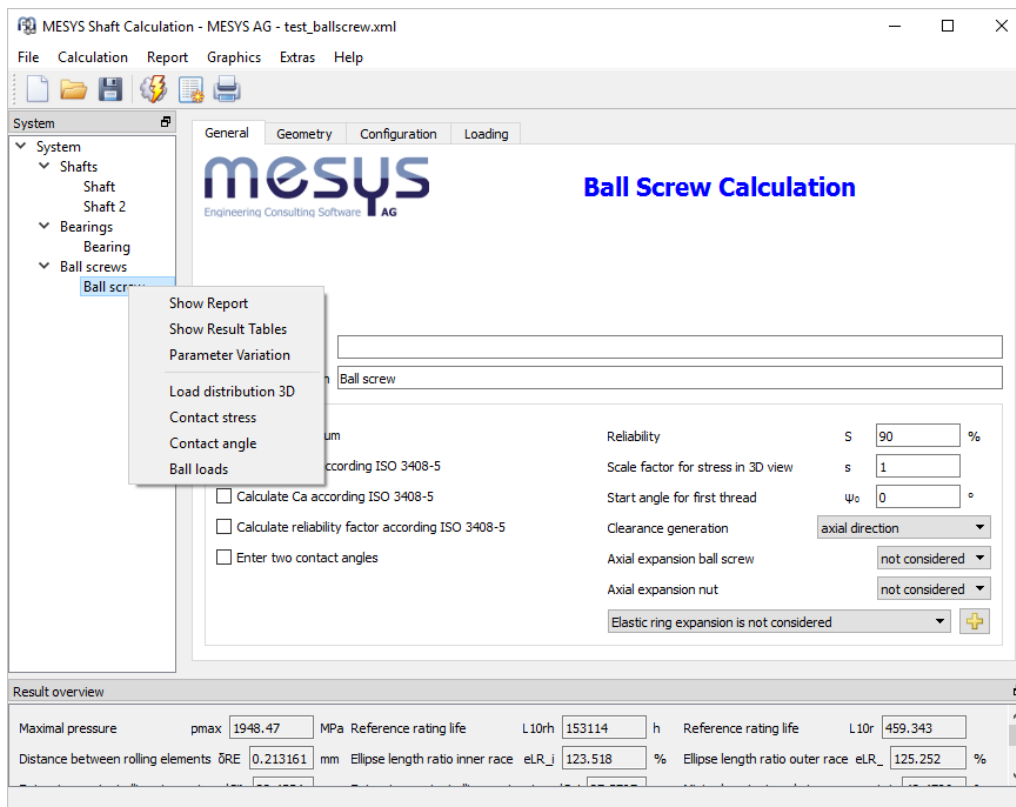


While the bearing calculation is open, the functions for file operations, calculation and report generation are used for the bearing calculation only. Therefore, only the bearing will be calculated and you will get results for the bearing in the results overview. However, it exists the possibility to run the whole shaft calculation, while the bearing calculation is in use, by pressing the key combination CTRL+F5.

Graphics and the tolerance report can be opened using the right mouse button in the system tree. Graphics are also available in the menu for graphics. Graphics from different bearings can be shown at the same time.

2.4 Interface to ball screw calculation

All dialogs of the [ball screw calculation](#)^[214] are fully integrated into the shaft calculation like for the rolling bearing calculation.



While the ball screw calculation is open, the functions for file operations, calculation and report generation are used for the ball screw calculation only. Therefore, only the ball screw will be calculated and you will get results for the ball screw in the results overview. However, it exists the possibility to run the whole shaft calculation, while the ball screw calculation is in use, by pressing the key combination CTRL+F5.

Graphics and can be opened using the right mouse button in the system tree. Graphics are also available in the menu for graphics. Graphics from different ball screws can be shown at the same time.

2.5 Results

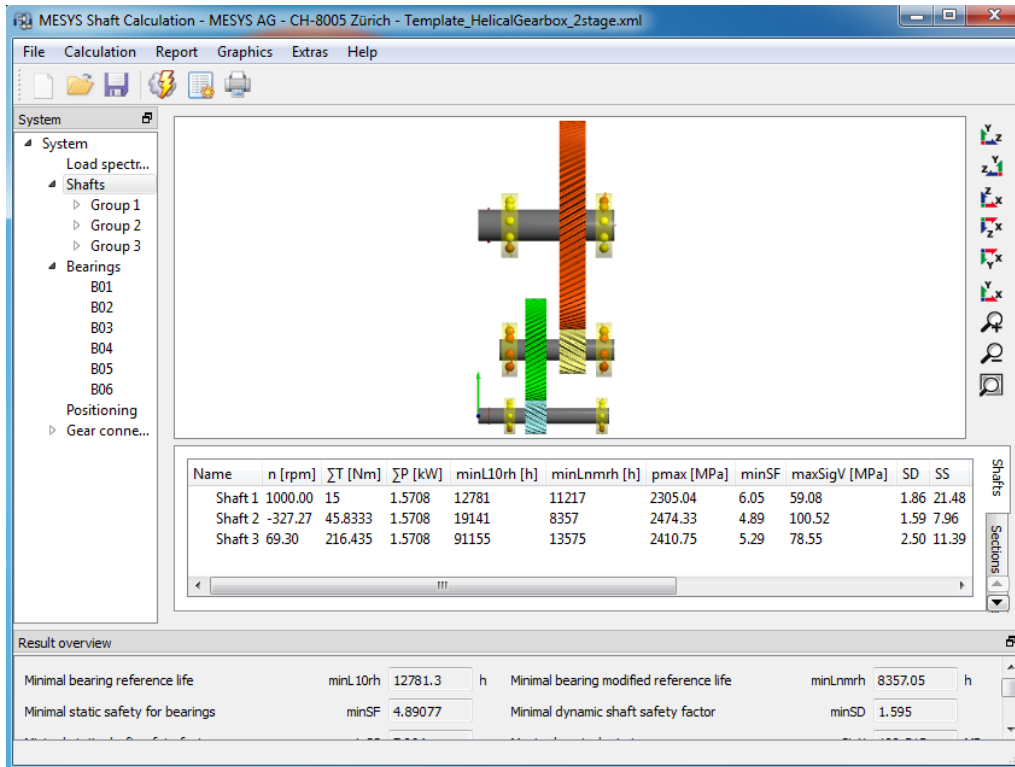
Results are available in different outputs. There is the default result overview on the bottom of the user interface, an overview of bearing forces and natural frequencies, several graphics and the report.

2.5.1 Results Overview

The results overview on the bottom of the window shows minimal bearing life, minimum static bearing safety and maximum shaft deflection. The results overview can be configured under Extras->Results overview.

2.5.2 Result Tables

Several result tables are available if the pages “Shafts”, “Bearings” or a group is selected.



2.5.2.1 Table for shafts

A table for shafts is showing results for each shaft:

Name	n [rpm]	ΣT [Nm]	ΣP [kW]	minL10rh [h]	minLnmrh [h]	pmax [MPa]	minSF	maxSigV [MPa]	SD	SS	maxUr [mm]	mass [kg]	Jxx [kg mm ²]	Jyy [kg mm ²]	Jzz [kg mm ²]
S1	1000.00	100	10.472	3330	5121	2678.03	3.86	41.06	3.94	8.28	0.0380	1.38	196.17	4204.48	4204.48
S2	-395.35	252.941	10.472	2757	1931	2920.93	2.97	23.86	3.96	11.92	0.0506	3.09	1025.99	8636.34	8636.34
S3	149.35	669.55	10.472	5764	1685	2849.53	3.20	52.04	3.88	5.53	0.0334	4.06	1707.99	12817.2	12817.2

Available results are:

- 'n': shaft rotation speed
- ' ΣT ': sum of all torques with the same sign
- ' ΣP ': sum of all power inputs with the same sign
- 'minL10rh': minimum reference life L10rh for all bearings on this shaft. Alternatively L10h according to ISO 281 is shown.
- 'minLnmrh': minimum modified reference life Lnmrh for all bearings on this shaft. Alternatively Lnmh according to ISO 281 is shown.
- 'pmax': maximum contact stress for bearings

- 'minSF': minimum static safety factor for all bearings on this shaft
- 'maxSigV': maximum equivalent stress with in this shaft
- 'SD': safety for fatigue according to DIN 743 for this shaft
- 'SS': static safety factor according to DIN 743 for this shaft
- 'maxUr': maximum radial displacement of the shaft
- 'mass': mass of this shaft
- 'Jxx, Jyy, Jzz': moments of inertia of the shaft

2.5.2.2 Table for sections

The table for sections shows details for each section for every shaft. The type of notch effect is shown, safety factors for strength, forces, moments and displacements:

Name	Type	x [mm]	SD	SS	Fx [kN]	Fy [kN]	Fz [kN]	Mx [Nm]	My [Nm]	Mz [Nm]	ux [mm]	uy [mm]	uz [mm]	rx [°]	ry [°]	rz [°]
4 S1			3.94	8.28												
A	Shoulder	38	5.35	8.28	0.000	-1.034	-2.826	-100.00	-22.57	8.22	0.0048	-0.0107	-0.0282	1.20	0.01	-0.00
B	Shoulder	172	27.83	87.42	0.000	0.403	1.095	0.00	-8.74	3.20	0.0048	-0.0063	-0.0165	1.18	-0.02	0.01
C	Interference fit	55	4.47	10.46	0.000	-1.033	-2.826	-100.00	-70.61	25.78	0.0048	-0.0122	-0.0324	1.19	0.01	-0.00
D	Interference fit	95	3.94	12.44	0.000	0.361	0.995	-2.57	-93.09	34.06	0.0048	-0.0132	-0.0351	1.18	-0.00	0.00

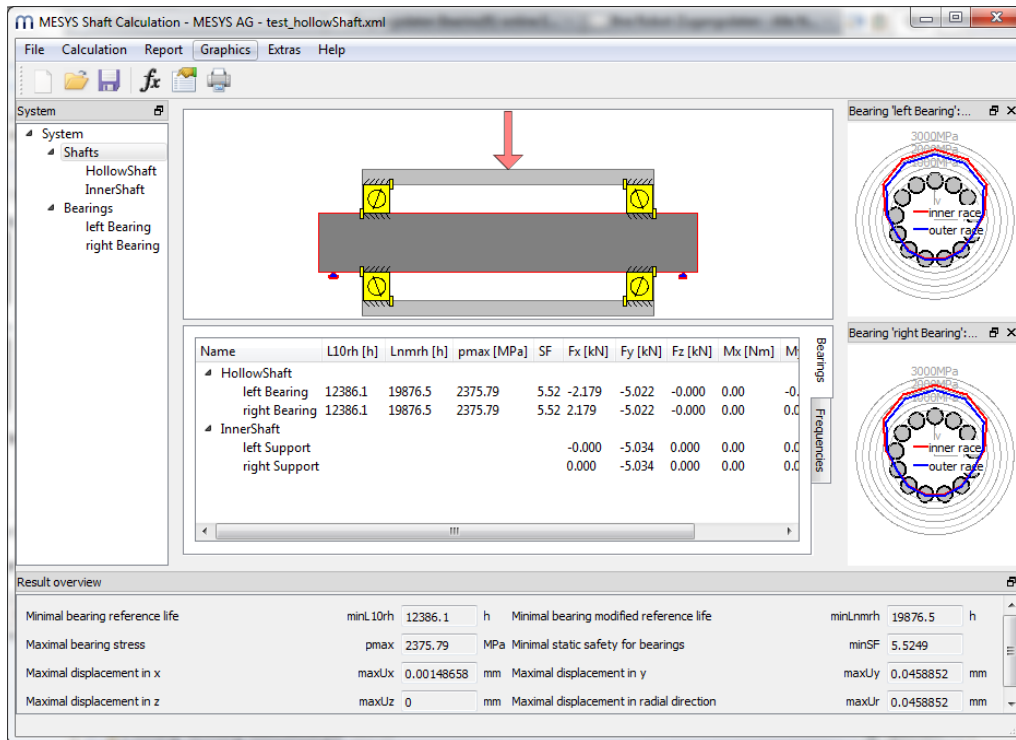
Available results are:

- 'SD': safety for fatigue according to DIN 743 for this section
- 'SS': static safety factor according to DIN 743 for this section
- 'Fx, Fy, Fz': forces in the section of the shaft
- 'Mx, My, Mz': moments in the section of the shaft
- 'ux, uy, uz': axial and radial displacement for this position
- 'rx, ry, rz': torsion and tilting angles for this position
- 'cxx, cyy, czz': stiffness of the system reduced to this point. The reciprocal of the main diagonal of the compliance matrix is shown
- 'crx, cry, crz': rotational stiffness of the system reduced to this point. The reciprocal of the main diagonal of the compliance matrix is shown

2.5.2.3 Table for bearings

A table for bearings shows life and loading of bearings and supports. The sign convention for the loads is that the force from the shaft to the bearing is shown.

For rolling bearings the life and pressure is shown in addition to forces and moments. Using the right mouse button additional results like operating clearance or spin-to-roll ratio can be shown.



Available results are:

- 'L10h': nominal life according to ISO 281
- 'Lnmh': modified life according to ISO 281
- 'L10rh': nominal reference life according to ISO/TS 16281
- 'Lnmrh': modified reference life according to ISO/TS 16281
- 'pmax': maximum contact stress of the bearing
- 'SF': static safety factor
- 'pmin': minimum contact stress of the bearing. If larger zero all rolling elements are loaded.
- ' $\omega_{spin}/\omega_{roll}$ ': spin to roll ratio for ball bearings
- ' Δb_{circ} ': circumferential ball advance for ball bearings
- ' ΔT ': temperature increase according to DIN 732 for this bearing
- 'nt': maximum permissible speed according to DIN 732
- ' κ ': viscosity ratio for this bearing. It should usually be between 1 and 4
- 'Pdeff': effective radial clearance of the bearing at operating conditions
- 'Paeff': effective axial clearance of the bearing at operating conditions

- ' α_{0eff} ': effective free contact angle for ball bearings
- ' $\Delta\alpha$ ': variation of contact angle between inner and outer ring for ball bearings
- ' $p_{FitShaft}$ ': pressure in interference fit between shaft and inner ring. This should be larger than zero usually.
- ' e_{LR_i}, e_{LR_e} ': extension of contact ellipse for ball bearings. Values below 100% indicate truncation.
- ' F_x, F_y, F_z, F_r ': bearing forces, F_r is radial force
- ' M_x, M_y, M_z, M_r ': moments, M_r is tilting moment
- ' u_x, u_y, u_z, u_r ': relative displacements of shaft node to second node (or housing), u_r is radial displacement
- ' r_x, r_y, r_z, r_r ': relative rotation of shaft node relative to second node (or housing), r_r is resulting tilting angle

2.5.2.4 Table for frequencies

In the overview also the natural frequencies can be shown. If a frequency is selected, the corresponding mode is shown in the graphics. Three displacements and the torsion angle are shown in 2D view. In 3D view the mode shapes are shown animated.

The screenshot displays the MESYS Shaft Calculation software interface. The main window shows a 3D model of a shaft with two bearings. A red arrow indicates a load applied to the shaft. The software interface includes a menu bar (File, Calculation, Report, Graphics, Extras, Help), a toolbar, and a system tree on the left. The system tree shows the following structure:

- System
 - Shafts
 - HollowShaft
 - InnerShaft
 - Bearings
 - left Bearing
 - right Bearing

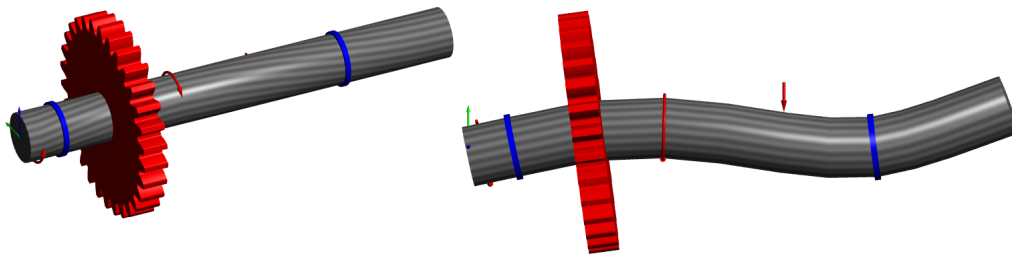
The central area displays a 3D model of the shaft and bearings. A table of natural frequencies is shown below the model:

Number	f [Hz]	f [rpm]	Type
1	442.299	26537.9	
2	1052.89	63173.7	
3	1110.75	66645.2	
4	1713.37	102802	
5	1894.91	113695	
6	1989.42	119365	
7	2366.65	141999	
8	3084.29	185058	
9	4009.59	240575	
10	5101.2	311478	

On the right side, there are two plots showing the stress distribution for the left and right bearings. The plots show the inner and outer races of the bearings, with a maximum stress of 3000 MPa indicated. The plots also show the inner and outer races of the bearings.

At the bottom, there is a 'Result overview' section with the following data:

Minimal bearing reference life	minL10rh	12386.1	h	Minimal bearing modified reference life	minLnmrh	19876.5	h
Maximal bearing stress	pmax	2375.79	MPa	Minimal static safety for bearings	minSF	5.5249	
Maximal displacement in x	maxUx	0.00148658	mm	Maximal displacement in y	maxUy	0.0458852	mm
Maximal displacement in z	maxUz	0	mm	Maximal displacement in radial direction	maxUr	0.0458852	mm



Available results are:

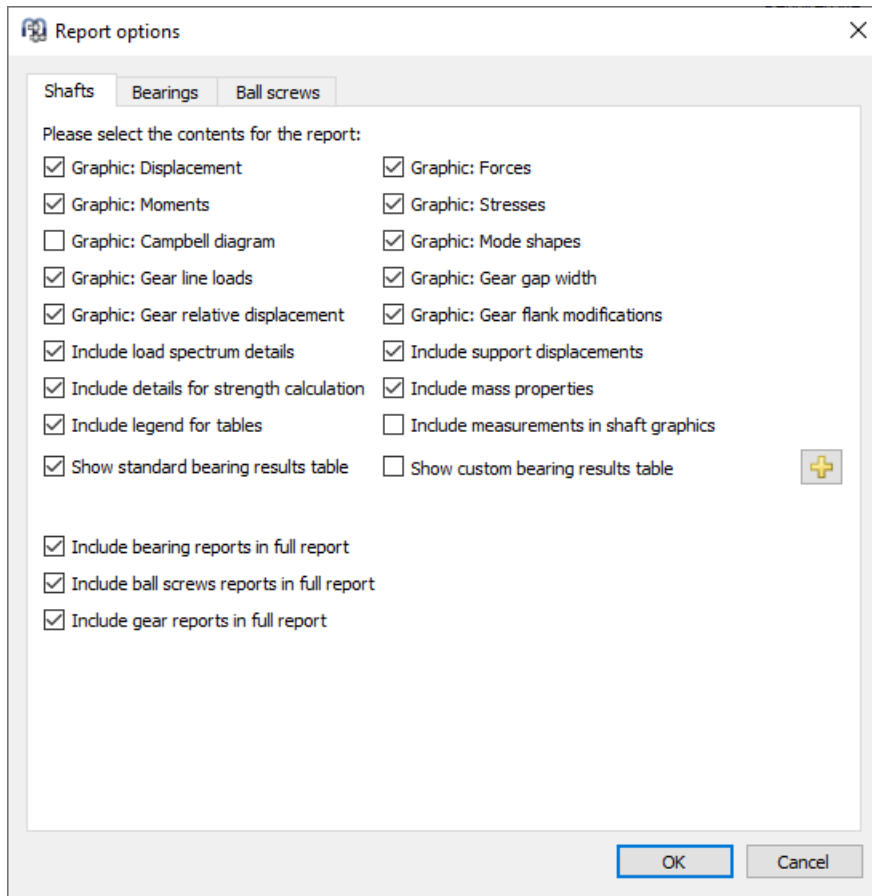
- 'f': frequency in Hz and per minute
- 'D': Damping ratio of the mode in case a damping is defined
- 'Type': Try's to classify the type of mode based on the kinetic energy of the modes

2.5.3 Report

Using the toolbar button or "Report"->"Show Report", a report for the shaft calculation is generated which only gives an overview for the bearing results. There is also "Report"->"Full report" which is generating a full report with results of the shaft calculation and the full reports of the bearing calculations.

2.5.3.1 Report options

Using "Report"->"Report options" the contents of the report can be configured.



One page is available for outputs of the shaft calculation, one page for the bearing calculation and one page for the ball screw calculation. The settings for bearings and ball screws are passed to the corresponding calculations. The legend for all the table parameters can be shown in the report. Custom tables for bearing results can be configured if the license for the bearing calculation is available.

2.5.4 Result tables

In the menu "Report" -> "Result tables", it is possible to output the result data by means of tables, which can be generated in different file formats as explained before at "Configuration with INI-File".

Results tables with detail results for bearings and ball screws can be opened with a right click on the bearing or ball screw in the system tree.

2.5.5 Custom Report Templates

Custom report templates can be added in subdirectory "templates/shaft". Reports based on these templates can be generated by selecting the report template under "Report" -> "Report templates".

The report templates are word files with DOCVARIABLE fields. These fields are filled on generation of the report.

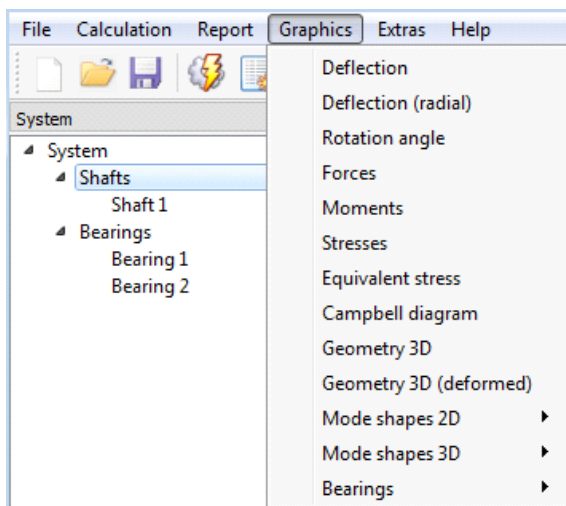
Possible fields are:

- DOCVARIABLE VAR name [unit]: the variable with the name 'name' is printed. Optionally a unit can be provided in brackets, for example [mm]. Additionally a format could be added like %6.3g or %6.3f, where 'f' always generates a fixed point output and 'g' generates an exponential output for large numbers.
- DOCVARIABLE VARU name [unit]: the variable with the name 'name' plus the unit are printed. Optionally a unit can be provided in brackets, for example [mm].
- DOCVARIABLE TABLE tablename: A table is inserted.
- DOCVARIABLE GRAPHIC graphicName width=130 height=70: A graphic is inserted. Optionally height and width can be given in millimeters.

Regarding details for variable names, table names and graphic names, please ask support@mesys.ch for additional documentation.

2.5.6 Graphics

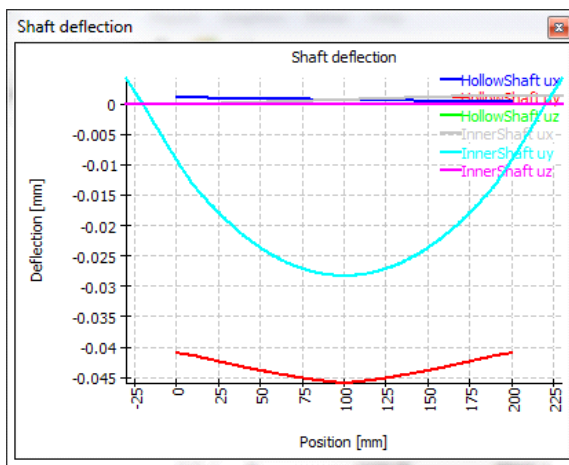
Several results can be shown as graphics. Most graphics are also appended to the report, so that a full documentation is provided in one document.



For all diagrams diagram options are available using the right mouse button. It is possible to change coordinate system and units. Also export and printing is available in the context menu.

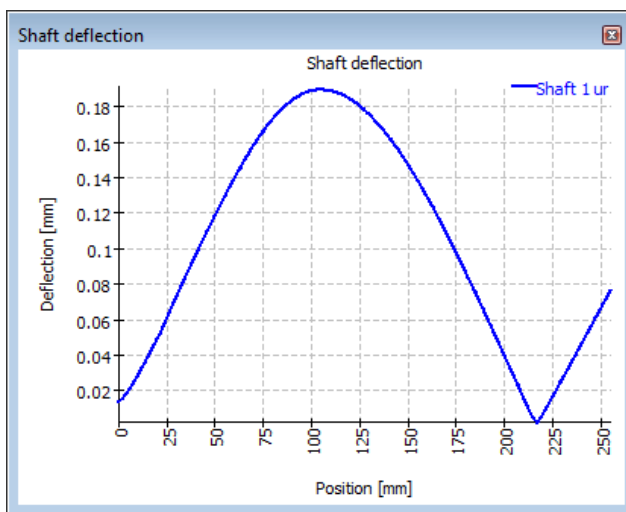
2.5.6.1 Shaft deflection

The deflection of the shafts is shown for the three displacement coordinates for each shaft.



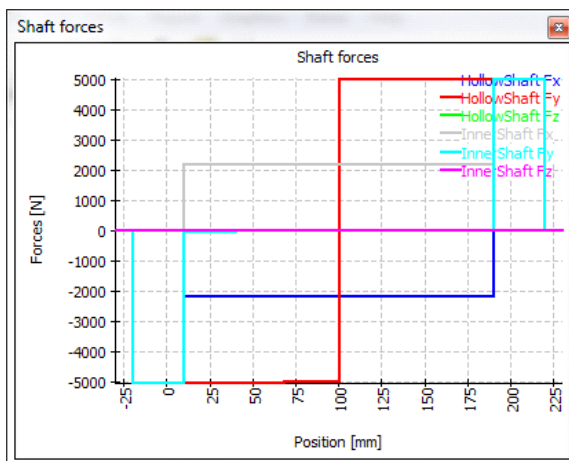
2.5.6.2 Shaft deflection (radial)

The radial deflection chart shows the absolute value of total radial deflection for the shafts.



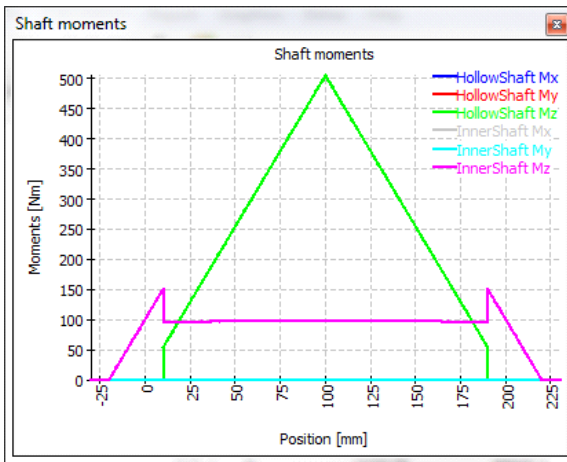
2.5.6.3 Shaft forces

The three force components are shown for each shaft.



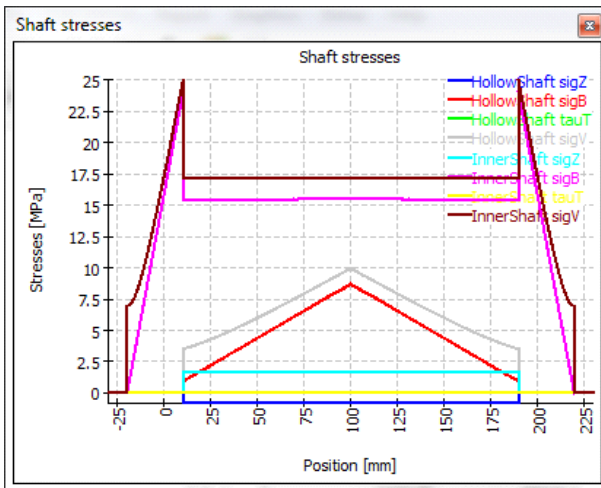
2.5.6.4 Shaft moments

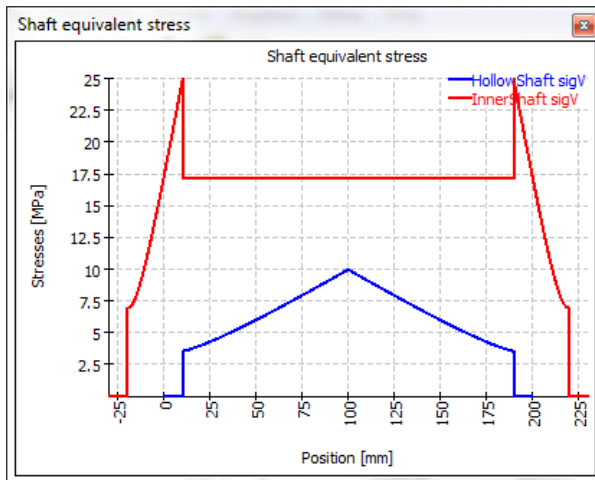
The two bending moments and the torque are shown for each shaft.



2.5.6.5 Shaft stresses

The stress components are shown for all shafts. Since there are too many curves in that diagram if several shafts are used, there is also a diagram which only shows the equivalent stress.

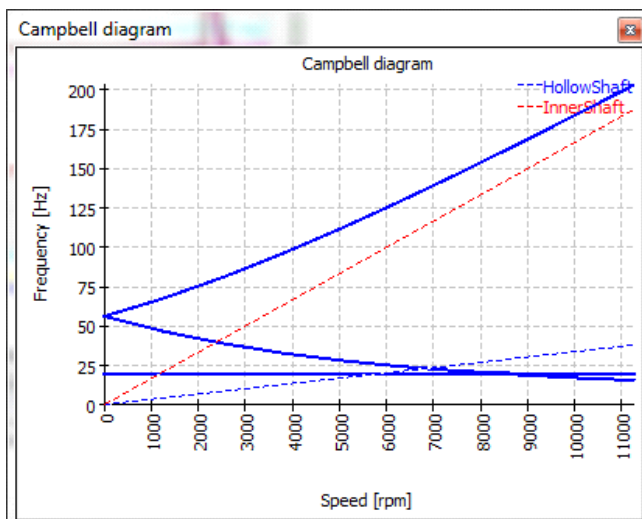




2.5.6.6 Campbell diagram

The Campbell diagram shows the change of natural frequencies over the shaft speed. The speed of all shafts is multiplied by the same factor in this calculation. The speed of the shafts is shown using dashed lines.

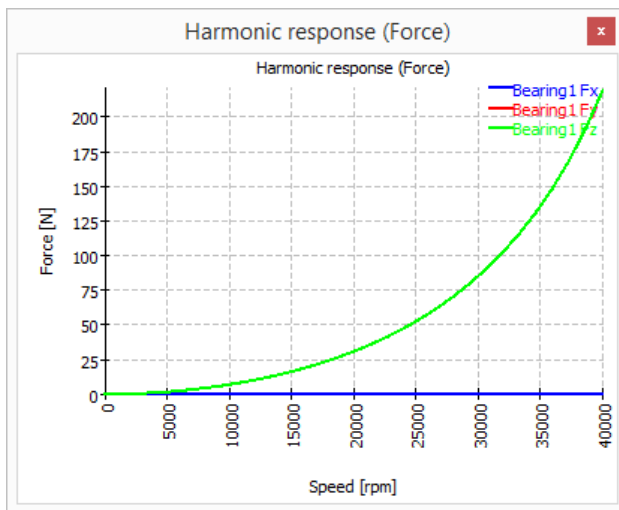
This calculation is always done considering the gyroscopic effect.



2.5.6.7 Harmonic response over speed

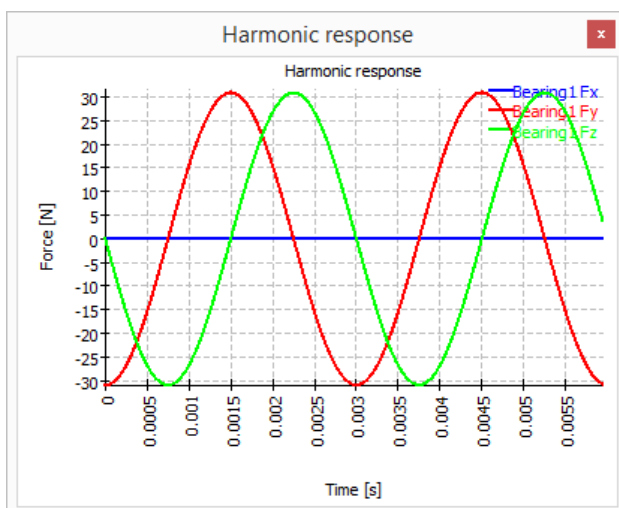
Harmonic response can be shown over speed. The calculation is done together with the Campbell diagram.

There are diagrams for forces available at all supports and diagrams for displacements available at positions of loads and sections of type "Documentation point"



2.5.6.8 Harmonic response over time

Harmonic response over time is showing the variation of reaction forces and displacements over time. This calculation is done for the current rotation speed.



2.5.6.9 Harmonic response 3D

Harmonic response 3D is showing the dynamic displacements over time as 3D view.

2.5.6.10 Diagrams for bearing analysis

In addition, there are several diagrams for bearing analysis. Please check the documentation of the bearing analysis for details. See [result graphics for bearings](#)¹⁹⁶.

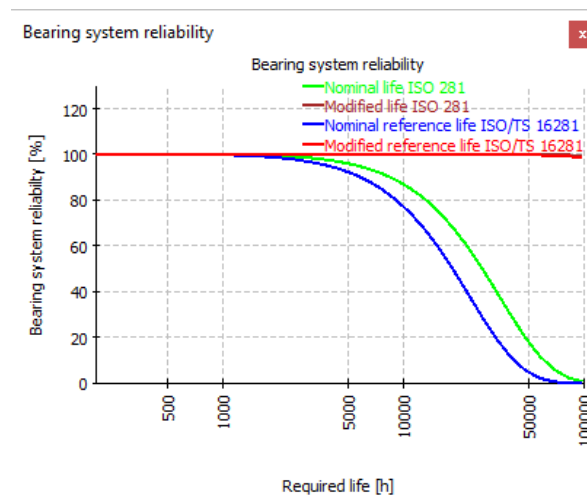
2.5.6.11 Bearing overviews

Several graphics for bearings are available as overview graphics, showing the same diagram for each bearing. This allows to get a quick overview over the behavior of all bearings.



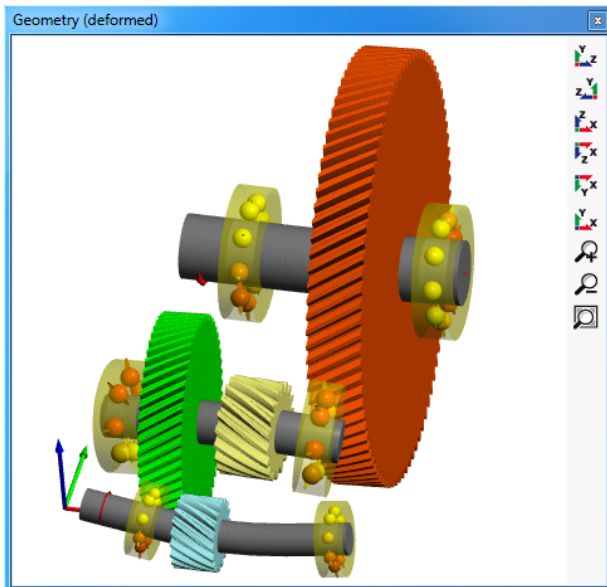
2.5.6.12 Bearing system reliability

The system reliability considering all bearings is shown in a diagram. The reliability is shown for the four different results for bearing life.



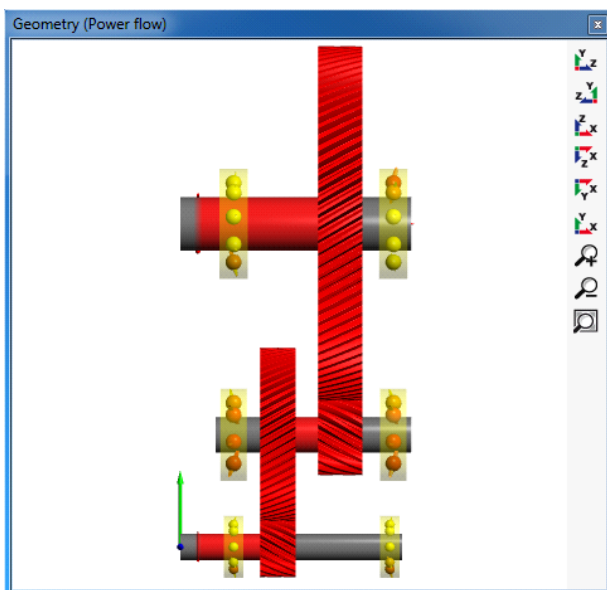
2.5.6.13 Geometry 3D

The 3D geometry can be shown without deformations or with exaggerated deformations. The diagram is available for the whole systems and for each group.



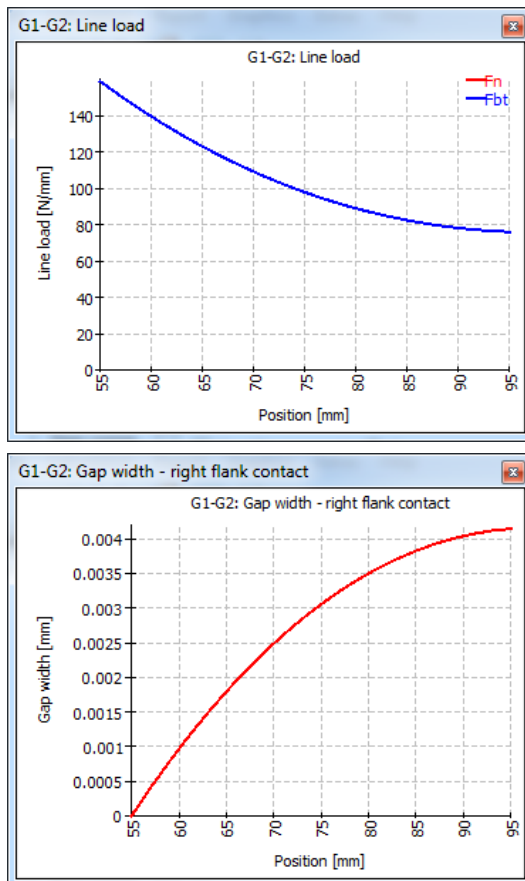
2.5.6.14 Geometry 3D (power flow)

The power flow of transmission systems can be shown in the 3D graphics.



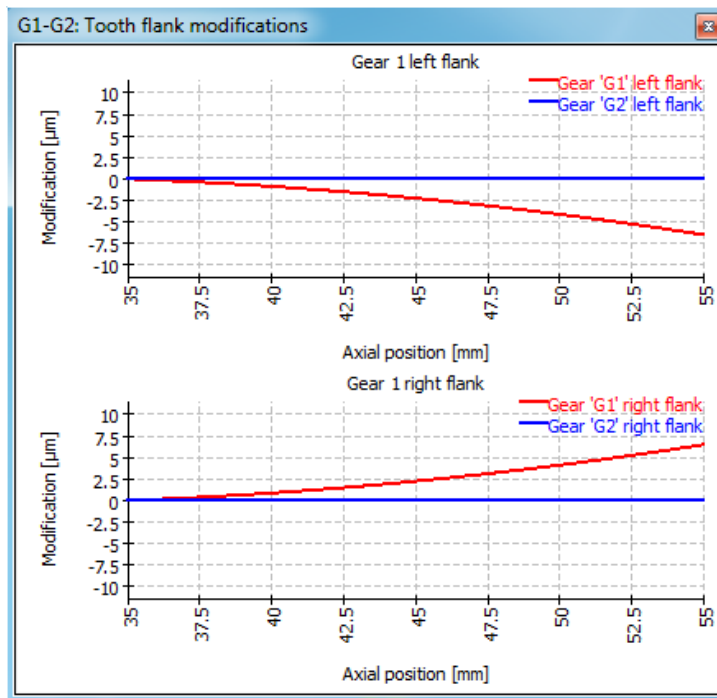
2.5.6.15 Gear line load and gap width

The line load on a gear pair can be shown and also the gap width. For these diagrams either the gears should be considered as stiffness under settings or the shaft diameter has to be large enough to take into account the stiffness of the gear body. The gap width can be used for a proposal of flank line corrections. Both diagrams are also available for load spectra showing results for all elements.



2.5.6.16 Tooth flank modifications

It shows the resulting profile of the tooth flank from the corrections applied in order to obtain an optimum tooth contact. In this way, the x-axis represents the tooth width in mm and the y-axis the geometrical variation in μm which has been undergone by the flank.



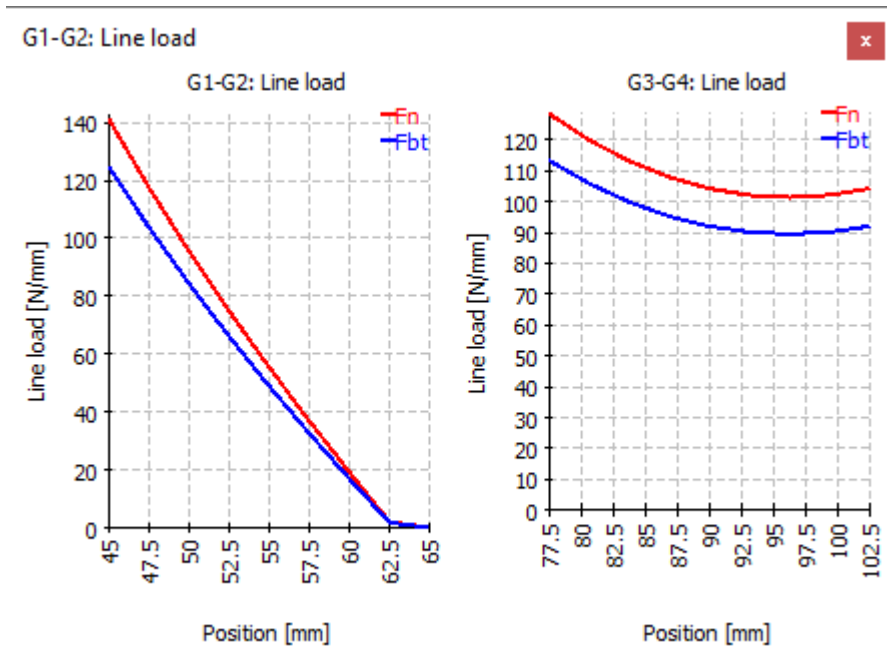
2.5.6.17 Relative displacements

A diagram for relative displacement of shafts is shown for each cylindrical gear pair. It shows the relative displacement of both shafts over the width of the gears. The axial displacement is a change of the center distance, the normal displacement a translation normal to the plane of the two shafts.

The diagram helps to understand required modifications or changes or clearance.

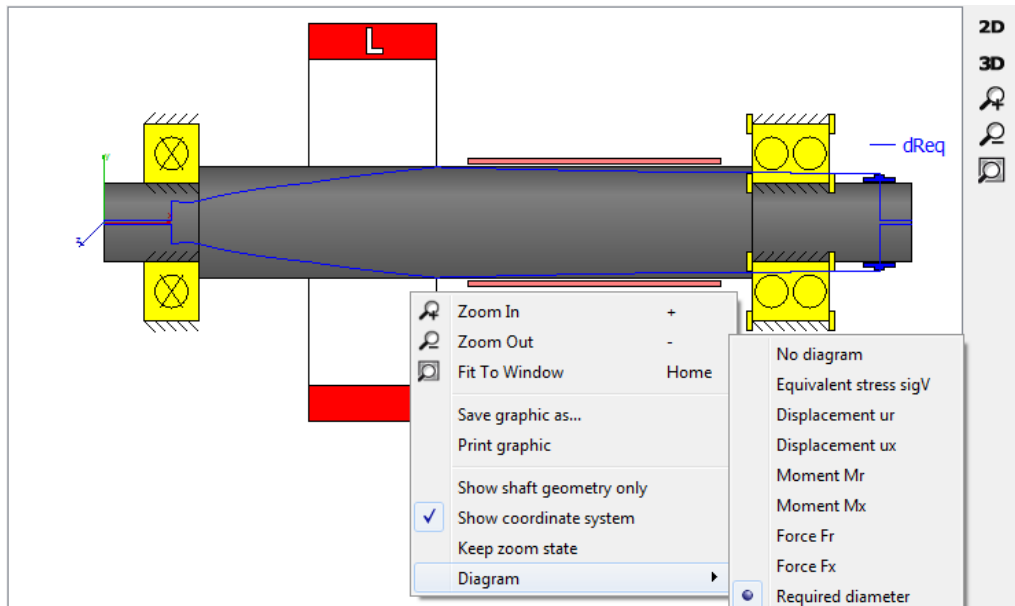
2.5.6.18 Connection overviews

Several graphics for connections are available as overview graphics, showing the same diagram for each connection. This allows to get a quick overview over the behavior of all connections.



2.5.7 Graphics area

By right-clicking the main graphics area in 2D mode, a context menu called 'Diagram' offers the possibility of overlaying some shaft-related diagrams on the actual geometry of the shaft.



An additional diagram called 'Required diameter' can be overlaid to the 2D geometry of the shaft. It provides a clue as to how the size of the diameter should be in order not to exceed a maximum equivalent stress value (see the image above), which is defined under the tab-page 'Display settings'.

2.6 Bibliography

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MESYS Rolling Bearing Calculation

3 MESYS Rolling Bearing Calculation

This rolling bearing calculation (Version 07/2019) calculates the load distribution, the reference life and the modified reference life according ISO/TS 16281 (DIN 26281) for the following types of bearings:

- Deep groove radial ball bearings
- Double row deep groove ball bearings
- Axial deep groove ball bearings
- Radial angular contact bearings
- Axial angular contact bearings
- Double row radial angular contact bearings
- Double row axial angular contact bearings
- Single row spherical ball bearings
- Double row spherical ball bearings
- Four-point ball bearings considered as radial bearings
- Four-point ball bearings considered as axial bearings
- Three-point bearings with split inner ring
- Three-point bearings with split outer ring
- Duplex ball bearings
- Radial cylindrical roller bearings
- Double row radial cylindrical roller bearings
- Axial cylindrical roller bearings
- Barrel roller bearings
- Needle bearings
- Tapered roller bearings
- Double row tapered roller bearings
- Axial tapered roller bearings
- Radial spherical roller bearings
- Axial spherical roller bearings

- Cross roller bearings considered as radial bearings
- Cross roller bearings considered as axial bearings
- Angular roller bearings considered as radial bearings
- Angular roller bearings considered as axial bearings

Additional bearing types will be added in the future.

The inner geometry of the bearings can be approximated by the software or provided by the user. The operating clearance can be specified. The influence of interference fits, temperature and centrifugal load on the clearance can be taken into account. Centrifugal loads and gyroscopic moments on the rolling elements can be considered for ball bearings only.

In extension of ISO/TS 16281 the influence of reduced material hardness can be considered according NREL guideline DG03. This includes a check of the case core interface.

The lubricant film thickness can be calculated by the software. This is done according to Harris: Rolling bearing elements.

The loading can be specified as force/moment or displacement/rotation independently for each of the five degrees of freedom. Bearing sets can be defined by using a configuration with multiple rows. This can be used for spindles or multiple row cylindrical roller bearings.

The following results can be found in the report:

- Reference life according ISO/TS 16281
- Modified reference life according ISO/TS 16281
- Basis life and modified life according ISO 281 for comparison
- Load distribution between rolling elements
- Reaction forces/moments and displacements/rotations
- Contact pressure for each contact
- Required shoulder height for contact ellipsis in ball bearings
- Static safety factor
- Maximum subsurface shear stress and stress at case core interface
- Load dependent friction torque for ball bearings based on coulomb friction

- Change to clearance because of interference fit and temperature
- Wear parameters like PV and QV for ball bearings

In addition to the report, the results are shown in several graphics. Report and user interface are both available in metric or US customary units. Supported languages for user interface and report are English, German, French, Spanish, Chinese and Korean.

Parameter variations can be made using ranges for several parameters. The results of parameter variations are given as tables or graphics. Proposals are provided for several inputs and conversions like axial clearance in radial clearance are possible.

Elastic deformations of the outer ring can be considered with an extension of the base software. The loading can be specified on several points in radial, axial or tangential direction and the life and load distribution are calculated with a deformed outer ring. The main use of this feature is for track rollers but also deformations in a planetary gear as outer ring can be considered. The elastic outer ring can be considered with following bearing types: Deep groove ball bearing, radial angular contact bearing, four-point ball bearing, radial cylindrical roller bearings and double row tapered roller bearings. Multi row bearings can be considered using bearing configurations.

3.1 Input Parameters

The input parameters are shown on five tab pages.

3.1.1 General

On the first input page in addition to a project name, several settings can be done.

3.1.1.1 Project name and calculation description

The project name and the calculation description are just inputs which are shown in the report header. They can be used to enter information about the purpose of the calculation for documentation.

3.1.1.2 Reliability

As default, the bearing life is calculated for a reliability of 90%. The required reliability can be changed here with limits 90% and <100%. The life modification factor for reliability a_1 is calculated according to this input using the three parameter Weibull relationship as given as formula in (ISO/TR 1281-2, 2008). The table in ISO 281 (2007) is also calculated according to this formula and has changed to previous versions of the standard.

3.1.1.3 Limit for aISO

The life modification factor for systems approach a_{ISO} takes into account lubrication properties and fatigue limit of the bearing and it is multiplied to the L_{10r} life to get the modified reference life. According to ISO 281, this factor is limited to a maximum value of 50. In some cases, for example in wind turbines, a smaller maximum limit of 3.8 is required.

The maximum limit for the a_{ISO} factor can be specified here. The input is only an upper limit for the a_{ISO} factor which is calculated according to (ISO 281, 2007).

3.1.1.4 Friction coefficient

The friction coefficient is used to calculate the friction moment of the bearing assuming coulomb friction in the contacts. In the actual version, this is only available for ball bearings. The friction moment is only considering the load dependent part of the friction; the no load part is not considered.

A proposal by (Harris, et al., 2007) is a value of 0.1 for the friction coefficient.

3.1.1.5 Calculate lubricant film thickness

The calculation of modified reference life requires a viscosity ratio κ . This ratio is calculated by the lubricant viscosity and a reference viscosity ν_1 or it can be calculated from the lubricant film parameter Λ as $\kappa = \Lambda^{1.3}$ according (ISO 281, 2007) and (ISO/TR 1281-2, 2008).

Since the viscosity ratio is calculated using standard settings for surface roughness, loading, pressure viscosity coefficient of the oil and geometric properties, the usage of lubricant film thickness takes into account more parameters of the actual bearing. For the definition of the reference viscosity ν_1 see the derivation in (Baalmann, 1994) or (Heemskerk, 1980).

If this setting is activated the lubricant film thickness is calculated according (Harris, et al., 2007). For ball bearings the minimal film thickness is calculated using equation 4.60 (Harris, et al., 2007) by Hamrock and Dowson. For roller bearings equation 4.57 (Harris, et al., 2007) according Dowson and Higginson is used. The same formulas are used for the calculation of Λ in (ISO/TR 1281-2, 2008). The software uses the minimal film thickness for the calculation of Λ , in contrast to (Heemskerk, 1980) who is using the central film thickness.

In addition, a thermal correction is applied from version 08-2016 on. The thermal correction factor is calculated according to (Koch, 1996) using the formula 4.62 from Wilson and Murch. Different literature provides different correction factors, see (Baly, 2005) or (Gohar, 2001). As temperature the input for the lubricant temperature is used. For high speed the thermal correction factor reduces the film thickness.

This calculation requires the input of surface roughness, oil density and oil pressure viscosity coefficient. The minimal film thickness for all contacts is then used for ball bearings. For roller bearings the minimal film thickness for each section is calculated and used for the calculation of life modification factor for systems approach a_{ISO} for this section.

3.1.1.6 Consider centrifugal force

The centrifugal force can be considered for ball bearings and radial cylindrical roller bearings in this version of the software. It will also be added for other roller bearings in the future.

Centrifugal forces will increase the pressure at the outer race, but decrease the pressure at the inner race. It will lead to different contact angles on inner and outer race and an increased spin to roll ratio.

3.1.1.7 Consider temperature gradient in fits

If this option is activated, the shaft and housing temperatures are available as inputs in addition to inner and outer ring temperatures. This option should only be activated if a temperature gradient between shaft and inner ring or between outer ring and housing should be considered.

If this option is activated be sure to define all four temperatures especially it used from the shaft calculation. The shaft temperature in the shaft calculation will set the shaft temperature for the bearing, but the inner ring temperature still needs to be defined by the user.

3.1.1.8 Oscillating bearing

Some bearings are not fully rotating but they are just oscillating. In this case, the effective number of load cycles is smaller than for a rotating bearing which can be considered by the software. The calculation is done according (Harris, et al., 2009) which is based on (Harris, et al., 2007).

An oscillating angle and an oscillating speed (oscillations per minute) have to be provided in this case. The oscillation angle is defined as the angle between the two and points of oscillation, so it is twice the amplitude.

The rotation speeds n_i and n_e are used for the calculation of centrifugal forces and lubrication film thickness, oscillation speed f_{osc} is used for the calculation of life in h. For the speed used to calculate the lubrication factor (Houpert, 1999) proposes $n = \Theta_{osc}/180^\circ * f_{osc}$. The speed has to be entered by the user.

In case of $\Theta_{osc} < \Theta_{crit}/2$ fretting corrosion can occur; the bearing should be rotated by a larger angle from time to time to redistribute the lubricant. A warning is shown in this case.

3.1.1.9 Calculate required hardness depth

The hardness depth is an input for material parameters. If it should be calculated by the software or if it is of no interest for through hardened bearings for example, then set this flag.

If the flag is not set, a check is made if the hardness depth is large enough.

3.1.1.10 Use fatigue strength for hardness depth

If this flag is set, the hardness depth is calculated using the fatigue strength of the core. If it is not set, the yield point is used.

If the calculation is done using a maximal load, the yield point can be used. If it is done using the equivalent load the fatigue strength is recommended.

3.1.1.11 Required subsurface safety

A required safety factor for the subsurface stresses against the yield point can be defined. It will be used for the calculation of the required hardness depth.

3.1.1.12 Selection for clearance

You can select which clearance is taken into account for the calculation. For interference fit and clearance a range of tolerances is given. The calculation can either use the minimum, minimum probable, medium, maximum probable or maximum value of the clearance. For other needs, the operating clearance can be specified directly as a number.


The probable values are calculated assuming a normal distribution within the tolerance field and an independence of tolerances for inner ring fit, outer ring fit and bearing clearance. In addition linearity is assumed which is not true in case a fit changes from interference to clearance within the tolerance range. For comparison probable values could be calculated using the [statistical parameter variation](#)^[30].

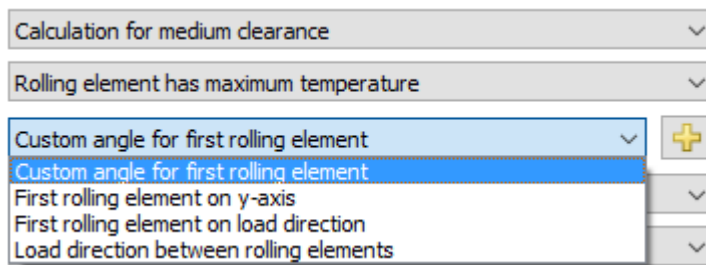
3.1.1.13 Rolling element temperature

As default, the temperature of the rolling element for calculation of operating clearance is set to the maximum temperature of inner and outer ring. This default setting can be changed to either the inner or outer ring temperature or a user input can be selected, which will be shown on page "Loading" together with the inner and outer ring temperatures.

If the heat is generated by the bearing, the maximum temperature is a reasonable selection. If the heat is generated by other heat sources, the mean temperature could be a better choice.

3.1.1.14 Position of first rolling element

As the circumferential position of the rolling elements within the bearing may affect the results, an option to define the position of the first rolling element by means of the angle is available. If a custom angle is desired, it can be entered by using the -button.



The selection “First rolling element on load direction” uses the radial displacement for definition of the angle. For pure moment load the angle is set to zero. In most cases this settings leads to the maximum contact stress and the lowest life.

The selection “Load direction between rolling elements” uses the radial displacement for definition of the angle and adds half the pitch angle. For pure moment load the angle is set to zero.

The option is not yet available for track roller calculation with elastic outer ring.

3.1.1.15 Options for gyroscopic moment

In case of different directions of forces on inner and outer ring contact, the gyroscopic moment on the ball will affect the load distribution. For high speed ball bearings often “outer race control” is assumed which means that spinning speed at outer race contact is zero. The available options are:

- “Gyroscopic moment is not considered”: This is the same behavior than older versions of the software. No gyroscopic moment is considered and spinning speeds are calculated by Coulomb friction.
- “Gyroscopic moment based on outer race control”: The ball rotation vector is calculated assuming no spinning at outer race contact. The gyroscopic moment is only causing a friction force at outer race contact.
- “Gyroscopic moment based on outer race control, distributed force”: The ball rotation vector is calculated assuming no spinning at outer race contact. The gyroscopic moment is causing a friction force at inner and outer race contact. The friction forces at each contact are proportional to normal force.
- “Gyroscopic moment based on inner race control”: The ball rotation vector is calculated assuming no spinning at inner race contact. The gyroscopic moment is only causing a friction force at inner race contact.
- “Gyroscopic moment based on inner race control, distributed force”: The ball rotation vector is calculated assuming no spinning at inner race contact. The gyroscopic moment is causing a friction force at inner and

outer race contact. The friction forces at each contact are proportional to normal force.

- "Gyroscopic moment based on mixed control": Dependent on the loading for each ball the rotation vector of the ball is calculated based on inner or outer race control assumption. The gyroscopic moment is causing a friction force at inner and outer race contact. The friction forces at each contact are proportional to normal force.
- "Gyroscopic moment based on friction": On each iteration for ball equilibrium the frictional forces based on Coulomb friction are calculated in the contact ellipses. The ball rotation vector is based on this calculation. The resulting gyroscopic moments is distributed on both contacts proportionally to the normal loads. This options leads to much longer calculation time and it can lead to convergence problems if not all balls are loaded.

For high speed ball bearing usually "outer race control" is assumed, this limits the selection to two options only. For low speed the gyroscopic has usually only very little influence and can be ignored.

For three-point bearings and four-point bearings the options based on outer ring, inner ring or mixed control can only be used on two point contact. In case of more contact points the gyroscopic moment is set to zero and a warning is shown.

3.1.1.16 Life for rolling elements

The bearing life calculation usually only takes into account fatigue life of the bearing races.

Life for rolling elements can optionally be calculated. The rolling element life is calculated analogous to (ISO/TS 16281, 2008) like described by (Correns, 2015). For the combination of life for single rolling elements to the life of the rolling element set, two options for the Weibull exponent can be considered, either $e=10/9$ for ball bearings and $e=9/8$ for roller bearings as in (ISO/TS 16281, 2008) and (ISO/TR 1281-1, 2008), or $e=1.5$ as in (ISO/TR 1281-2, 2008) section 4.2.1 for the reliability factor a_1 . This is different to (Correns, 2015) where an exponent of $e=10/3$ for ball bearings and $e=9/2$ for roller bearings is used without giving a reason for these exponents.

If this option is activated, the usual bearing life is not changed just the life of the rolling element set `L10r_RESet` is calculated in addition.


3.1.1.17 Elastic ring expansion


As default, only the elasticity of the contact between rolling element and race is considered. If, for example, an angular contact bearing with clearance between

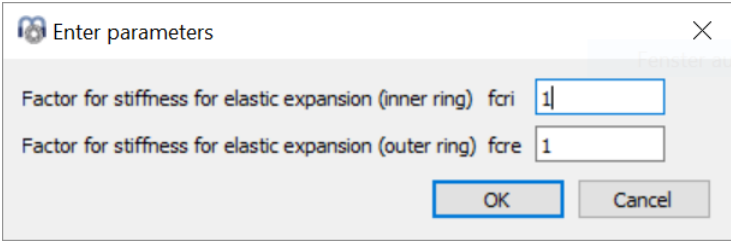
outer ring and housing is under axial pretension, the outer ring can expand radially. This will reduce the pretension.

In the current version, two options for consideration of elastic expansion of bearing rings are available. Both options are based on the assumption of a thick-walled ring as in the calculation of interference fits.

Either the minimal or the mean radial force in the load distribution is converted into a constant radial pressure which is then taken into account as in a calculation for interference fits. Both rings inner/outer ring and shaft/housing are considered in the calculation of the fit. This calculation approach is only valid if the variation of radial forces is small. So the axial load should normally be larger than the radial load.

The ring diameter is taken as $D_{pw} \pm D_w$ as default, so a stiffening effect of shoulders is not taken into account. Using the -button behind the [bearing type](#)¹³⁵ instead the use of an equivalent cross section can be activated. Then the ring diameter is defined so that the cross section is equivalent to the cross section of the bearing ring. Both options for the ring diameter can be compared with FEA results using the graphic "[Radial expansion of races](#)"²⁰⁹.

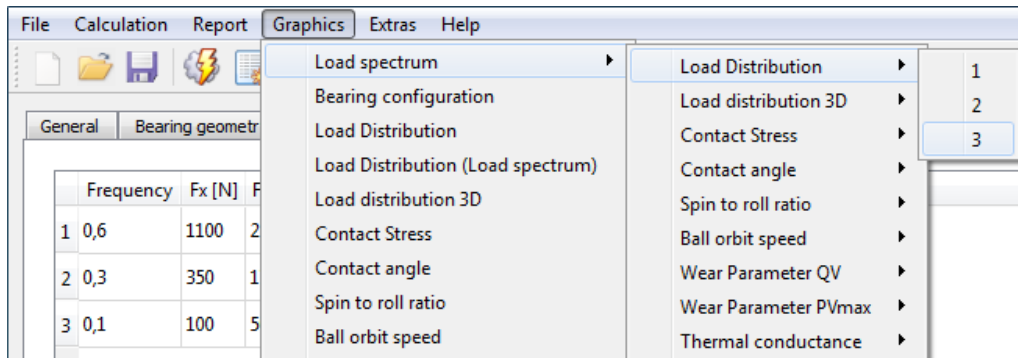
An additional factor for the ring stiffness of the inner and outer ring can be defined using the -button behind the selection box. This factor is applied to the surface used for conversion of the rolling element force to a pressure: $p = F / (f_{cr} * A)$.



The elastic ring expansion is set to zero in case deformations from the shaft calculation are considered as the ring expansion should not be considered twice.

3.1.1.18 Use load spectrum

A load spectrum can be used instead of a single load case. This option can be activated here. For each load case a full calculation using all factors is made. The resulting life is calculated using the life of each element. Result graphics are only shown in the report for the selected result element of the load spectrum. However, the result graphics corresponding to any element number (load case) of the load spectrum can be shown under the menu 'Graphics' -> 'Load spectrum' (only until 25 elements), as we can see in this picture:



All other graphics available under the menu 'Graphics are only valid for the selected element.

Additionally, under 'Graphics'->'Load distribution (Load spectrum)', all the load distribution cases are superimposed on the chart at the same time. This is also possible for the roller graphics.

Please note that more intermediate results are printed in the report if a calculation is done for a single load case.

3.1.1.19 Calculate modified life

If this flag is set, the modified reference life L_{nmrh} will be calculated. You can clear the flag if no information about lubrication is known or lubrication should not be considered in the calculation.

3.1.1.20 Use extended method for pressure distribution

In (ISO/TS 16281, 2008) a simplified approach to calculate edge stresses in roller bearings is used and extended method according other literature are recommended. If this option is set the pressure distribution is calculated according to (de Mul, et al., 1986), which is the newest literature in the proposals of the standard.

This extended method is available for radial and axial cylindrical roller bearings, taper roller bearings and spherical roller bearings.

3.1.1.21 Calculate static safety factor based on stress

If this options is set the static safety is calculated based on maximal contact stress.

For ball bearings $SF = \left(\frac{p_{perm}}{p_{max}}\right)^3$ is used, for roller bearings $SF = \left(\frac{p_{perm}}{p_{max}}\right)^2$ is used, so that the safety factor is proportional to the load. The permissible stress is based on (ISO 76, 2006).

If the option is not set, the static safety factor is calculated based on maximum rolling element load and Q_{max} in (ISO/TR 10657, 1991).

For ball bearings both options usually show the same results, but for roller bearings edge stresses are only considered if the static safety factor is based on stresses.

3.1.2 Bearing geometry

3.1.2.1 Bearing types

Different types of ball and roller bearings can be calculated using the software. In addition to the calculation of a single bearing, a configuration of several rows can be defined using "Bearing configuration".

The screenshot shows a dialog box titled "Options for selected bearing type" with a close button (X) in the top right corner. The dialog contains the following options and input fields:

- Bearing has filling slot
- Bearing inner ring is shaft
- Bearing outer ring is housing
- Use ring diameter for equivalent cross section for calculation of fits +
- Calculate load capacity for hybrid bearings
- Permissible ellipse length ratio: %
- Lower stress limit for truncation $p_{min}(eLR)$: MPa
- Limit for conformity for dynamic load capacity f_{limCr} :
- Limit for conformity for static load capacity f_{limC0r} :
- Tolerance for conformity inner race Δf_i :
- Tolerance for conformity outer race Δf_e :
- Friction coefficient for fitting μ_{fit} :
- Reduction of load capacity because of hardness according to Harris (dropdown menu)

At the bottom of the dialog are "OK" and "Cancel" buttons.

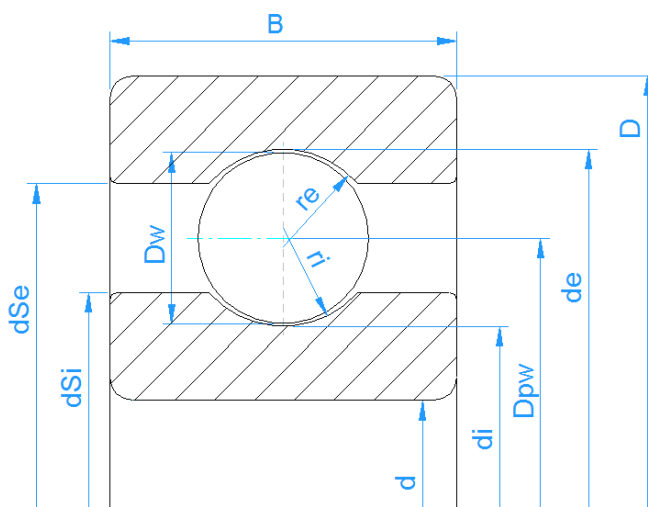
By using the +-button behind the bearing type, some options for the selected bearing type can be defined.

- For all bearings, it can be selected that inner ring and shaft or outer ring and housing are identical. This affects the material input and the input of tolerances.
- The option "Use ring diameter for equivalent cross section for calculation of fits" will lead to a larger ring thickness of inner and outer ring for the calculation of fits. Both variants can be compared using the graphic ["Radial expansion of races"](#)^[209]. By using the +-button behind the option, the race diameters can be overridden.

- The option "Calculate load capacity for hybrid bearings" will use (ISO 20056-1, 2017) and (ISO 20056-2, 2017) for the calculation of load capacities which leads to an increased static load capacity because of higher permissible stresses. The Youngs modulus of the rolling elements is required to be larger than 300GPa for the increased static permissible stress.
- For ball bearings the "Permissible ellipse length ratio" affects a warning about truncation and an output for a permissible axial force in the report.
- For ball bearings the "Lower stress limit for truncation" ignores truncation for all contacts with a contact stress less than this limit.
- For ball bearings, a limit for the conformity used in calculation of load capacity can be defined. The load capacity is calculated by the maximum value of the given limit and the actual geometry input.
- For deep groove ball bearings a tolerance for the conformity can be defined. This will lead to additional outputs in the [Tolerances Report](#)¹⁹⁶.
- If a friction coefficient for fitting is defined, an axial mounting force is shown in the tolerance report.
- The reduction of the load capacities because of hardness can be calculated according to Harris or Schaeffler. The option according to Schaeffler leads to larger load capacities for very low hardness.

In the current version, the following bearing types are supported:

3.1.2.1.1 Deep groove radial ball bearing



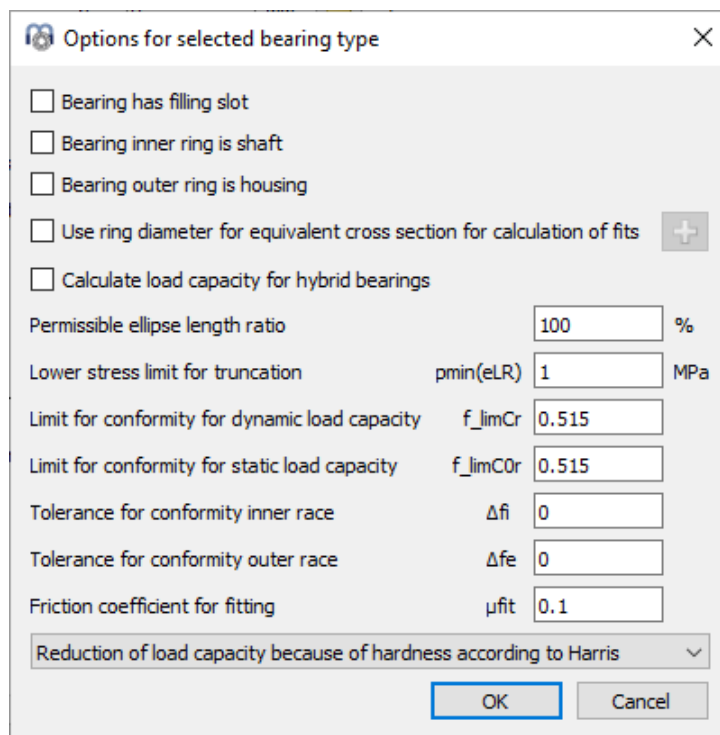
Deep groove radial ball bearings are cheap standard bearings. Both radial and axial loads can be transmitted. The nominal contact angle is 0° which will increase under axial load depending on the radial clearance in the bearing. The free contact angle α_0 is shown in report and result overview.

The geometry is described with number and diameter of balls, pitch diameter, conformity of inner and outer race and the diametral clearance. The number of balls in the bearing is restricted, to be able to assemble the bearing.

The conformity is the input parameter defining the radius of the races. ($r_i = f_i \cdot D_w, r_e = f_e \cdot D_w$)

Usually the conformity is $f_i = f_e = 0.52$ for deep groove radial ball bearings.

The diametral clearance is defined as $Pd = de - di - 2Dw$.

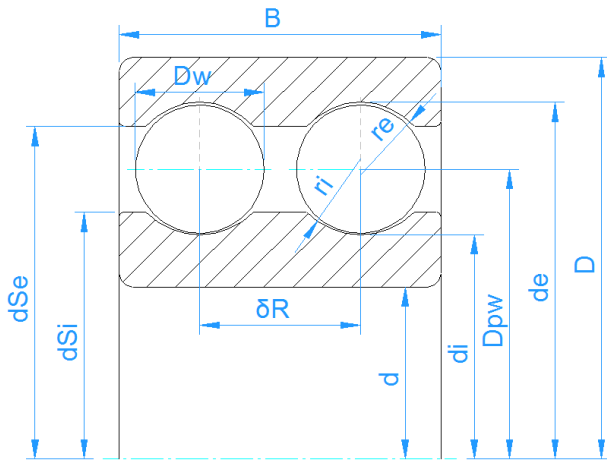


The **+**-button behind the bearing selection also offers some other settings depending on the bearing type. For deep groove radial ball bearings, it can be selected if a “filling slot” bearings is used which influences the calculation of the load capacity with factor b_m in (ISO 281, 2007).

The permissible ellipsis length ratio controls the warning about cut off of contact ellipsis. A value below 100% would allow some amount of cut off, a value larger 100% adds an additional distance. This permissible value is used for the calculation of permissible axial load limit given in the report too.

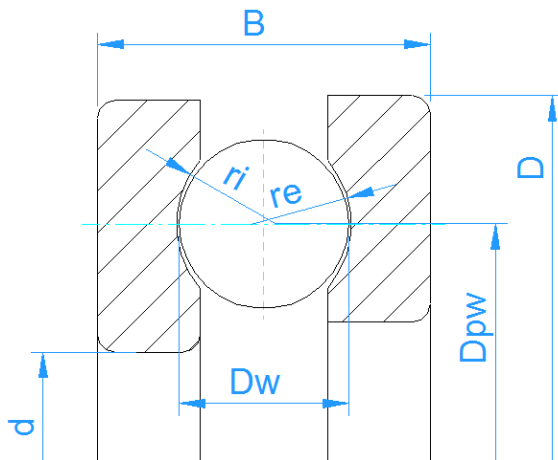
3.1.2.1.2 Double row deep groove ball bearing

For double row deep groove ball bearings, a distance between rows ‘ δR ’ has to be defined in addition to the parameters of the single row deep groove ball bearing.

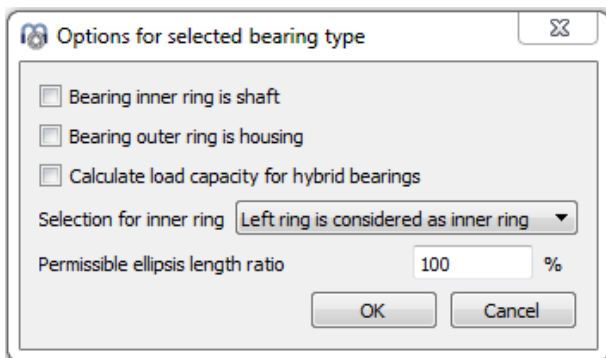


3.1.2.1.3 Axial deep groove ball bearings

For axial deep groove ball bearings, the nominal contact angle is 90° . The default conformity is $f_i = f_e = 0.535$.

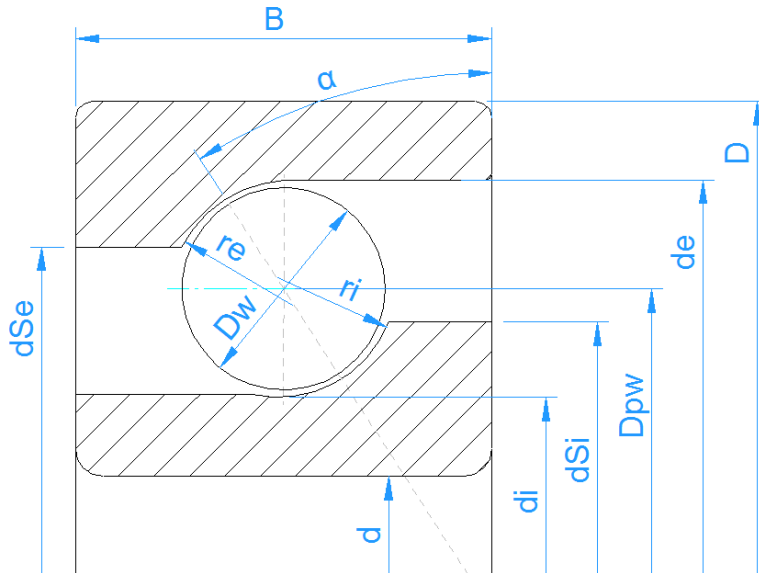


In the bearing type options dialog, it can be defined if the left or the right ring shall be considered as inner ring. This also changes the sign of the axial force which is given for the inner ring.




No tolerances are considered for axial deep groove ball bearings.


3.1.2.1.4 Radial angular contact bearing

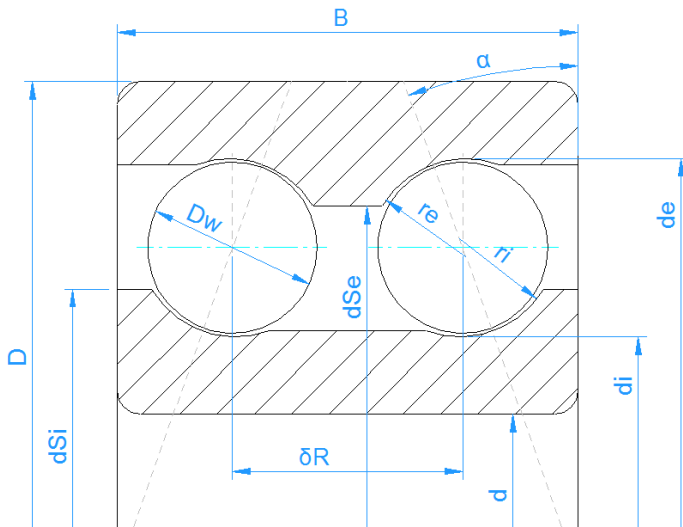


The angular contact bearing is similar to the deep groove ball bearing, but the contact angle is larger. Standard bearings have contact angle of 15°, 25° or 40°.

Double row angular contact bearings can be either defined using a single row bearing and a configuration of two bearings or a double row angular contact bearings can be defined directly. This allows the input of radial clearance of the two row bearing. In both cases life modification factor a_{ISO} is calculated for both rows separately.

The direction of contact angle can be defined using the -button behind the contact angle. Additionally, the input of direction of contact is available in the bearing type options.

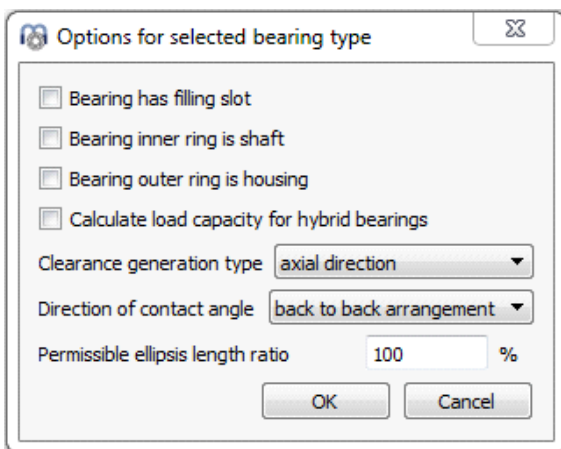
The -button for axial clearance allows to calculate the axial clearance for given pretension or for given radial clearance in case of double row angular contact bearing.



3.1.2.1.5 Double row radial angular contact bearing

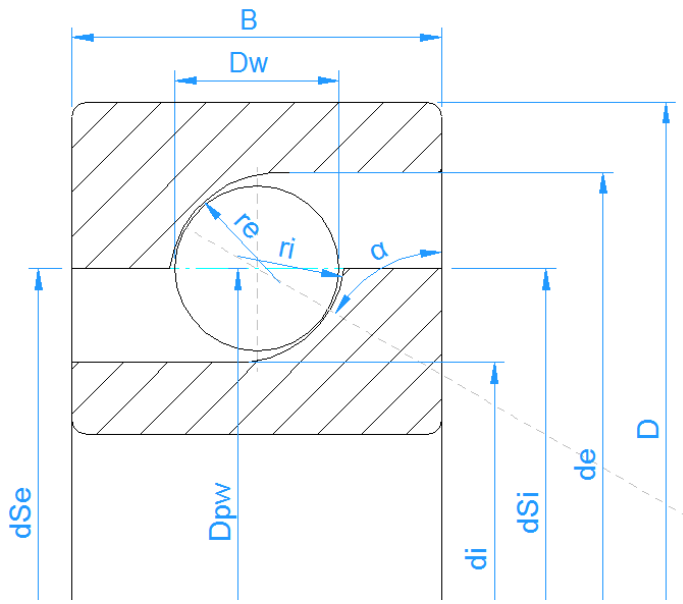
For double row angular contact bearings, the configuration of contact angles can be defined in the options dialog.

The clearance can be generated in three different ways. See at four-point bearing below.



In addition to the inputs for the single row angular contact bearing, the distance between rows has to be entered.

3.1.2.1.6 Axial angular contact bearing

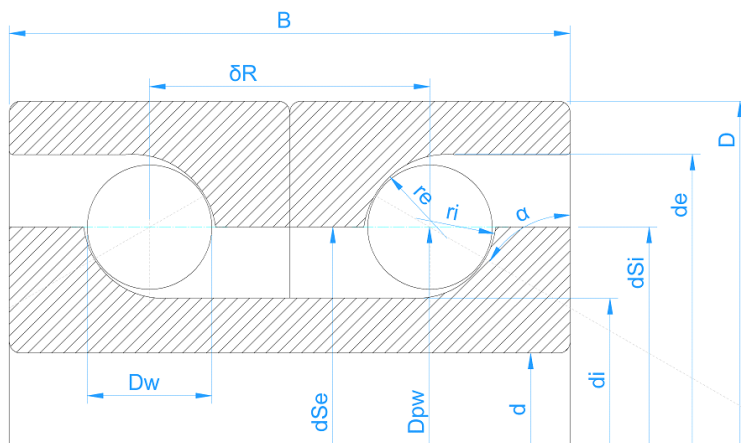


The description of geometry for axial angular contact bearings is the same as for radial angular contact bearings. The only difference in geometry is a value of 0.535 for the conformity, instead of the 0.52 used for radial bearings.

The results for load distribution are the same for the selection of axial or radial angular contact bearings, but for axial bearings additional reduction factors are considered in the calculation of load capacity. Therefore, the resulting life is smaller if the bearing is calculated as axial bearing.

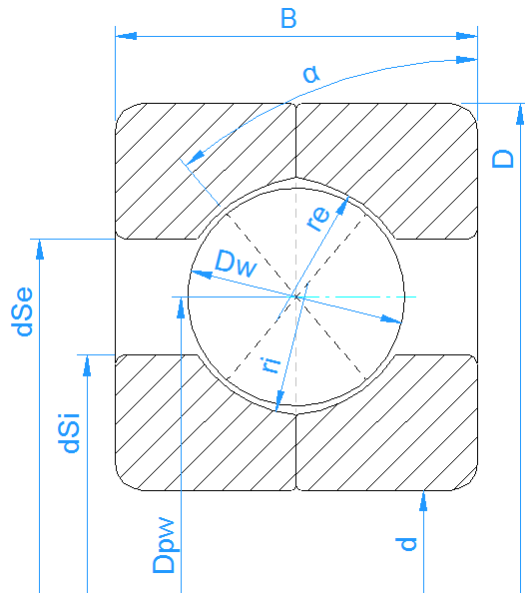
Standard axial angular contact bearings have a contact angle of 60°.

3.1.2.1.7 Double row axial angular contact bearing




Just as for the radial case, double row axial angular contact bearings can be either defined using a single row bearing and a configuration of two bearings or a double row angular contact bearings can be defined directly.

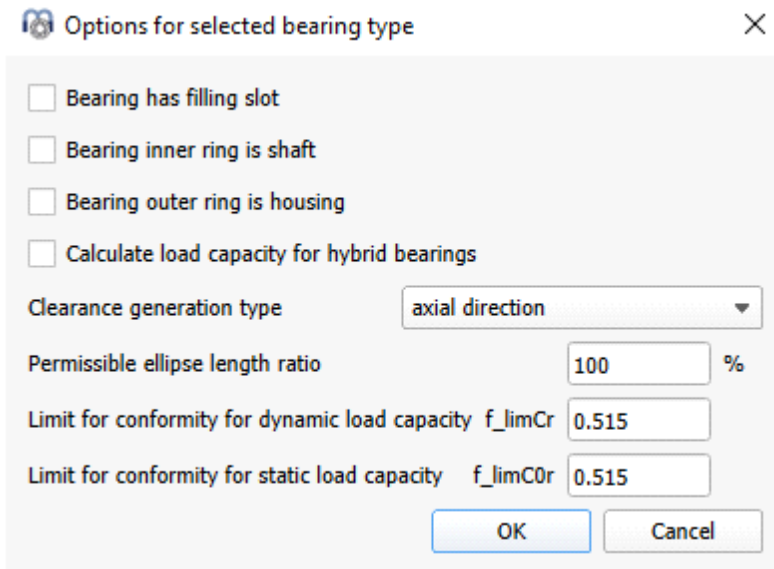
3.1.2.1.8 Four-point bearing considered as radial bearing



This can be used for standard QJ bearings with 35° contact angle. Load capacity for a four-point bearing is calculated as a double row bearing, as usually done in bearing catalogues. The results are reported for both rows. In case of two contact points, there is one contact on the inner ring of one row and on the outer ring of the other row. All possible four contacts are considered in life calculation.

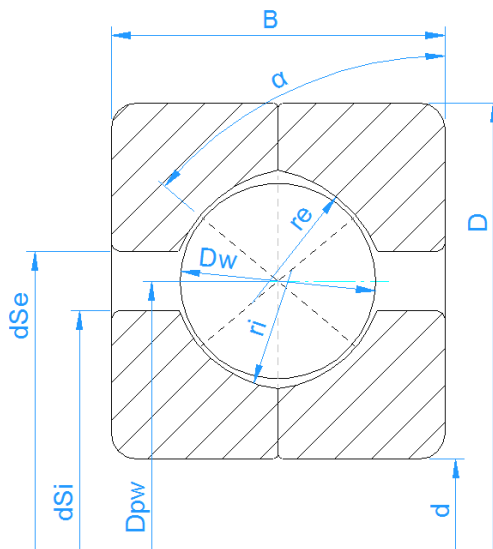
Using the -button behind the bearing type a “filling slot” can be defined and the method clearance is generated for the bearing. There are three ways to generate clearance in the software:

1. Centers of curvature are moved in axial direction. This leads to a decreased contact angle for radial load. It corresponds to a four-point considered as angular contact bearing.
2. Centers of curvature are moved in radial direction. This leads to an increased contact angle for axial load.
3. Centers of curvature are moved along the nominal contact angle.



3.1.2.1.9 Four point bearings considered as axial bearing

This selection can be used for slewing rings as done in (Harris, et al., 2009). Because of additional reduction factors for axial ball bearings, the calculated life will be smaller than considered as a radial bearing. The load distribution is the same as for the four-point bearing considered as radial bearing.



3.1.2.1.10 Three point bearings

Three point bearings with split inner ring or split outer ring can be considered. The split ring has two radii like in a four-point bearing and the second ring has a full radius. Therefore there can be two or three contact points.

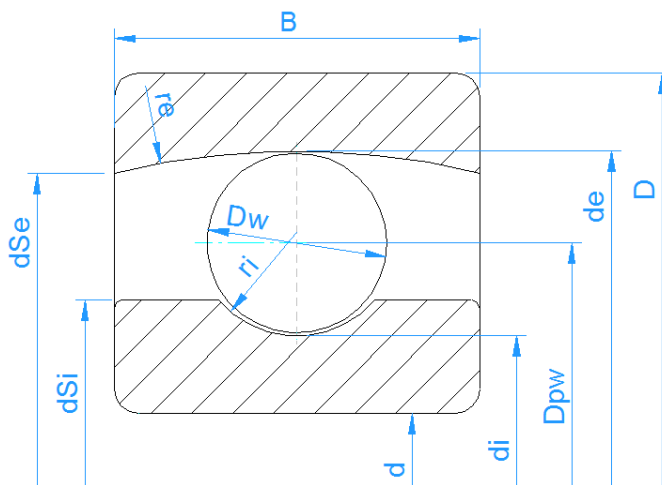
In contrast to the other bearing types, the input for the contact angle is the contact angle on the split ring under radial contact. The input field is therefore shown as shim angle. The operating contact angle will depend on radial clearance. The radial clearance can be calculated from the required free contact angle.

The free contact angle is calculated based on two point contact. If the resulting free contact angle is less than the shim angle, there will be three point contact under low axial load.

The load capacity is calculated based on a single row radial angular contact ball bearing using the free contact angle. Because the free contact angle is used, the load capacity is dependent on the nominal clearance.

3.1.2.1.11 Self-aligning ball bearing

Spherical ball bearings can be selected as single or double row bearings. A single row bearing has a contact angle of zero.



3.1.2.1.12 Self-aligning ball bearing (double row)

For double row bearings the distance of rows is defined by the contact angle. The distance between rows is then

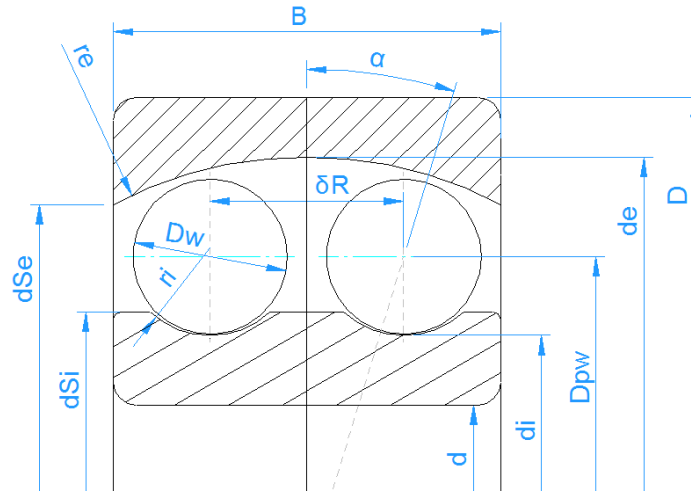
$$\delta_R = D_{pw} \cdot \tan \alpha$$

For spherical ball bearings, the conformity of the outer race is defined as

$$f_e = \frac{r_e}{D_{pw} / \cos \alpha + D_w}$$

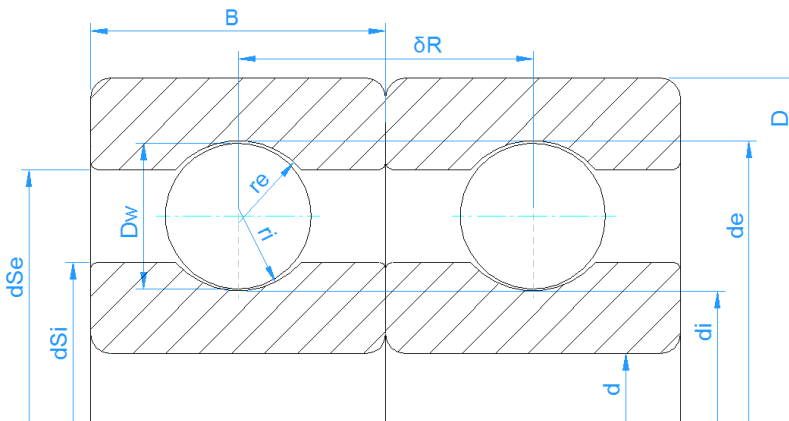
So it is the ratio of radius and outer race diameter which is 0.5 as standard.

The shoulder diameter of the outer ring is automatically limited by the width and the radius of the outer race. Still a larger value as the limit could be entered by the user. The life modification factor a_{ISO} is calculated for both rows separately.




3.1.2.1.13 Duplex bearings

Two deep groove ball bearings can be calculated as a set by selecting “Duplex bearings”. The same could be done by using a single deep groove ball bearing and a configuration of two bearings but the input is more flexible using “Duplex bearings”.



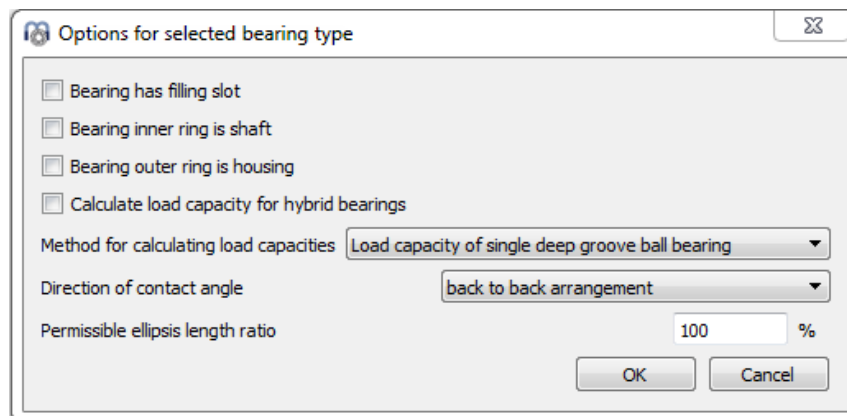
The geometry data is defined for a single deep groove ball bearing, additionally the distance between the two rows is an input value. The input of the diametral clearance is for the single bearing only.

The clearance of the configuration can then be changed by an axial offset δ_{CC} between inner and outer ring of each bearing. Using the -button it can be

calculated from a given axial clearance, radial clearance or pretension force for the bearing configuration.

The free contact angle of the bearing α_0 and $\alpha_{0\text{eff}}$ are shown in the report and in the results overview.

Options for the bearing allow selecting of face-to-face or back-to-back configuration. This has an influence of the load distribution on moment loads or a tilted bearing.

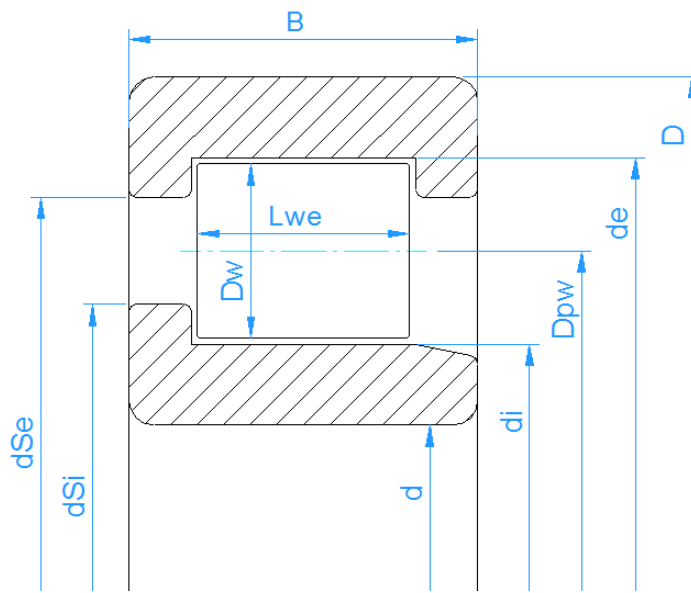


There are four options for the calculation of load capacities:

1. *Load capacity of single deep groove ball bearing:* The input values are for a single bearing only. These are the values which are given in the documentation of the single bearing. In this case, the load capacities of the pair are shown as C_{sys} in the report.
2. *Load capacity of paired deep groove ball bearings:* Here the calculation is done using load capacities for two paired deep groove ball bearings using factors of $2^{0.7}$ for dynamic and 2 for static load capacity.
3. *Load capacity of double row deep groove ball bearing:* Here the load capacities are calculated using the factors for a double row bearing. The dynamic load capacity is smaller than for the second case.
4. *Load capacity of double row angular contact ball bearing:* Here the load capacity is calculated using the free contact angle of the bearings.

If thermal effects are considered, the axial offset δ_{CC} will be modified during the calculation to take the different axial elongation of inner and outer ring into account. The results are the same as if using a single radial deep groove ball bearing with a configuration of two bearings.

3.1.2.1.14 Radial cylindrical roller bearings

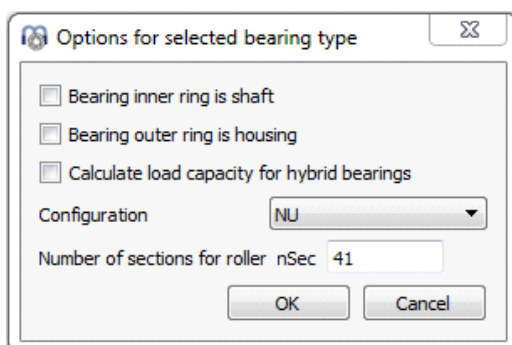


The radial cylindrical roller bearing is a bearing providing a high load capacity for radial loads but it does not support high axial loads or misalignment of inner and outer ring.

In addition to the parameters used for ball bearings, the effective length of the roller L_{we} is a required input parameter. The effective length is a little smaller than the length of the roller because of radii at the end of the roller. The contact angle is always zero for cylindrical roller bearings.

The roller profile is considered as defined by ISO/TS 16281 as default. If the extended calculation for pressure distribution is selected, an input of profile for races and roller is possible using the \oplus -button behind the roller effective length. Several options are available including reading the profile from a file.

The axial load is considered as shown by (Harris, et al., 2007). The axial load brings a tilting moment to the roller and an unsymmetrical load distribution on the races occurs. The axial forces are considered at the half height of the shoulders. If 20% is entered for the height of the shoulder, the axial force is acting at 10% of roller diameter.



The type of cylindrical roller bearing NU, NJ, NUP ... can be selected in the options dialog of the bearing.

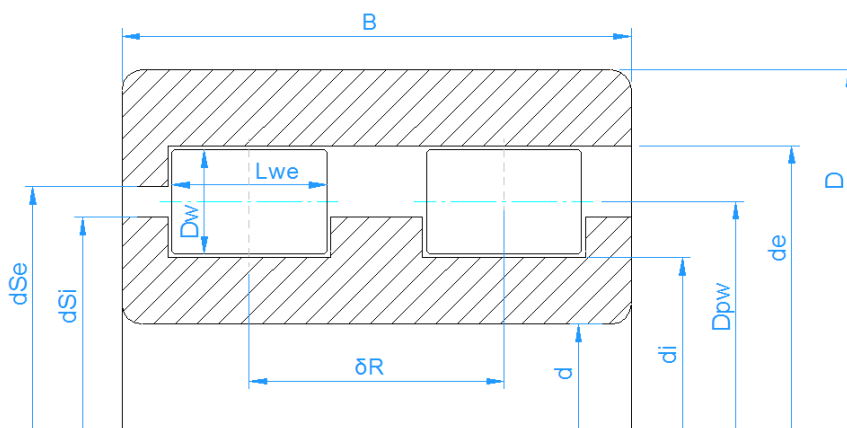
For cylindrical roller bearings which support axial loads, a radial and axial clearance can be specified. It is important to enter a value for axial clearance if tilting occurs. For NUP type, the axial clearance has influence on the reaction moment.

For NUP type, the axial clearance is measured between left and right positions of the rings like for deep groove ball bearings. For directional bearings like NJ, the axial clearance is between reference position and one side like for angular contact ball bearings or taper roller bearings.

For types which do not support axial loads like NU, the axial displacement 'ux' has to be selected as input instead of 'Fx'.

The number of sections for the calculation of load distribution can be changed in bearing options, too. The minimum is 31 sections. A larger value reduces the edge stresses by the approximation function in ISO/TS 16281 if the extended method for pressure distribution is not active.

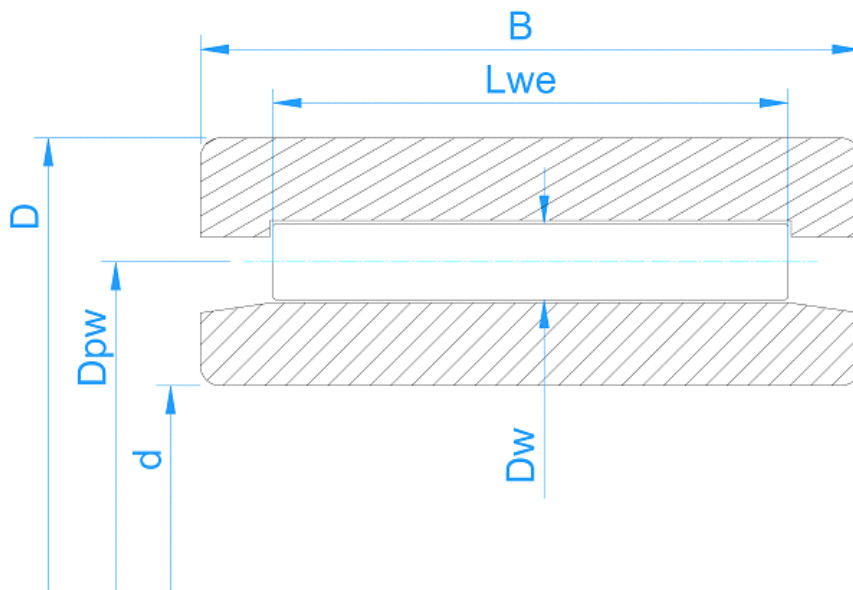
3.1.2.1.15 Radial cylindrical roller bearings (double row)



In addition to the inputs for a single row bearing, the distance between the row centers has to be defined. Generally, the same results are obtained as if using a single row bearing with a bearing configuration of two rows; however, the results will differ if the configuration shows different positions of shoulders under axial load. An additional difference is the load capacity which is shown for the double row bearing.

The different bearing types can be selected in the options dialog as well.

3.1.2.1.16 Needle bearings



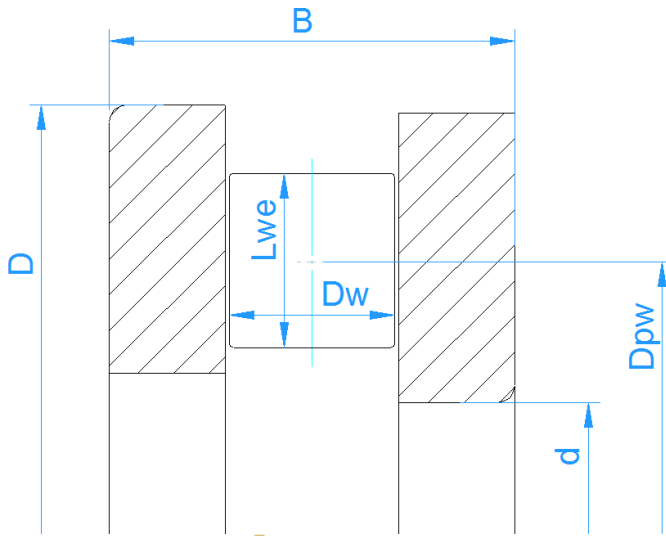
Needle bearings can be calculated using the selection 'Cylindrical roller bearing', too. The type 'needle bearing' was only added so the types can be separated in the database.

Needle bearings do not support axial forces. Therefore 'ux' has to be selected as input instead of 'Fx'. The axial clearance is not available as an input.

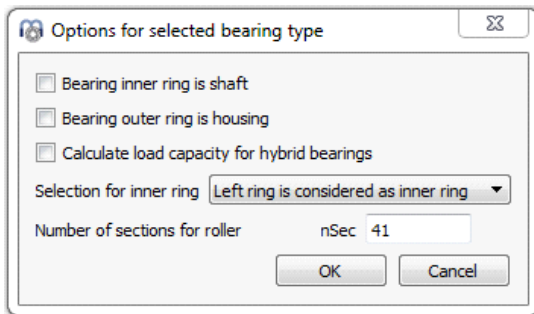
For drawn cup needle roller bearings the b_m -factor for the dynamic load capacity should be set to 1.0 instead of 1.1 in case of machined bearing rings. The type can be selected under options for the bearing type using the \oplus -button behind the bearing type.

3.1.2.1.17 Axial cylindrical roller bearings

Axial cylindrical roller bearings have a contact angle of 90° . They only allow axial forces and bending moments. No radial forces can be used. Therefore, u_y , u_z have to be entered instead of F_y , F_z .



The options dialog (+-button behind bearing type) allows to specify if the left or the right ring shall be considered the “inner ring” on which loading applies. The default setting “Left ring is considered as inner ring” results in a positive axial load to be entered.

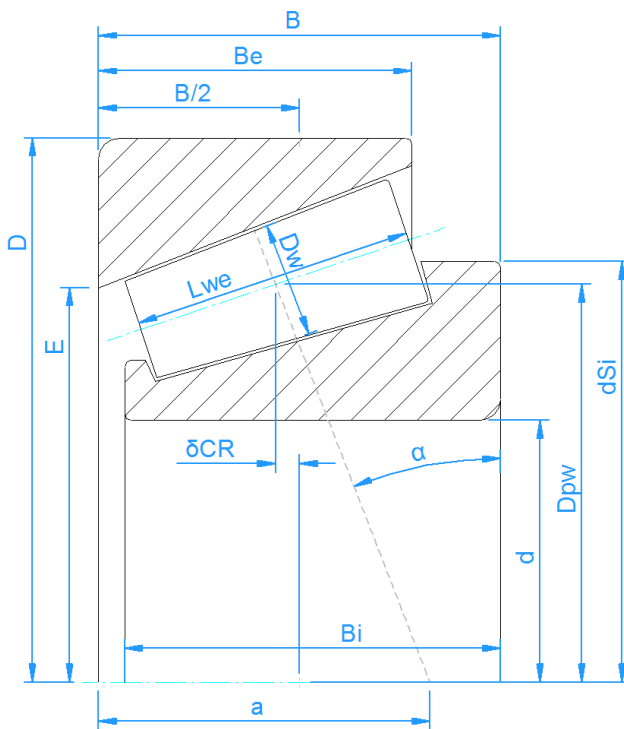


An axial clearance can be entered; its only influence is an offset to the axial displacement.

No tolerances are considered for axial cylindrical roller bearings.

3.1.2.1.18 Radial tapered roller bearings

Tapered roller bearings use a conical roller instead of a cylindrical roller. The input roller diameter is given for the middle of the roller and also the pitch diameter D_{pw} is defined for the middle of the rollers. The clearance is defined as axial clearance.

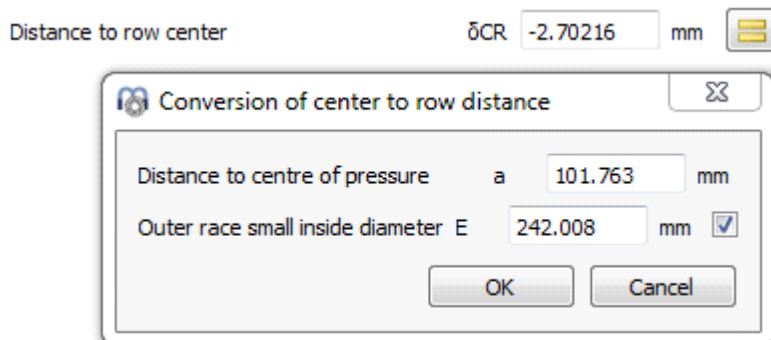


The contact angle should be the direction of the load. Therefore, the angle of the outer ring, the cup, has to be specified for the contact angle. The angles for the roller and the cone are then calculated so that all cones intersect on no load condition.

If the axial force is too small, a calculation error can occur since the bearing will fall apart. You have to enter an axial force large enough or to specify an axial displacement instead.

The direction of contact angle can be defined using the -button behind contact angle or bearing type.

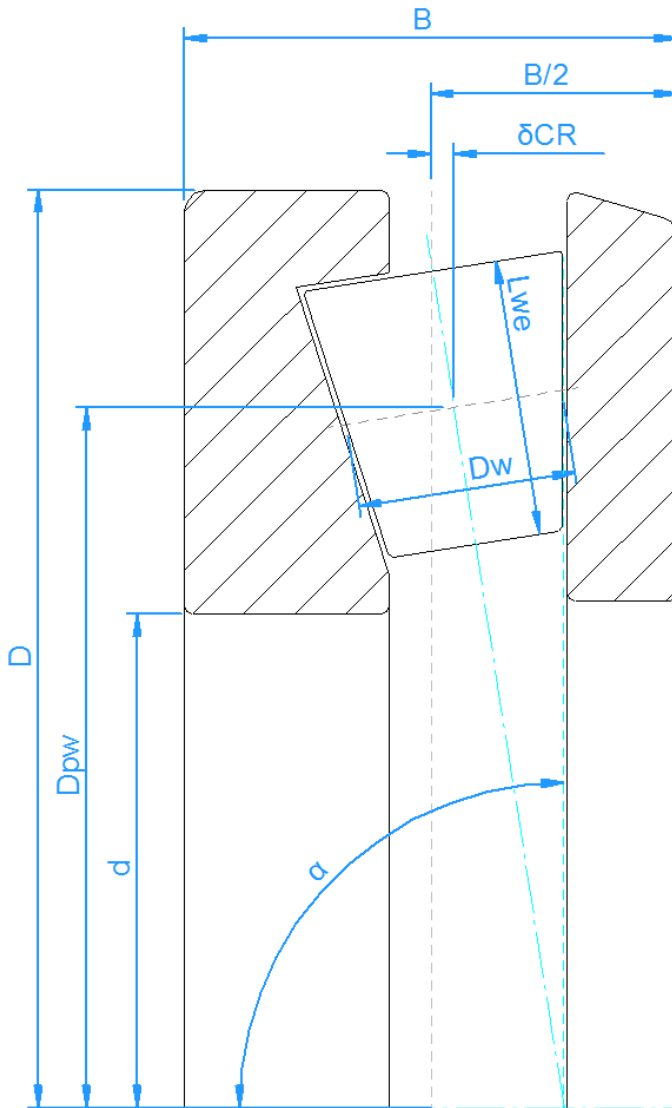
The diameter of the shoulder of the inner ring can be defined. The force is assumed to be at the medium height of the shoulder.



The distance between center of bearing and center of roller 'δCR' can be defined in two different ways by using the -button. It can be converted either from the

distance to the center of pressure 'a', or from the outer race small inside diameter 'E' by setting the corresponding flag. Note that changing 'E' will in turn modify 'Dpw' according to their geometric relations.


3.1.2.1.19 Axial tapered roller bearing



The inputs which define a tapered roller thrust bearing are a bit simpler than for the radial case of tapered roller bearings, since the distance between center of bearing and center of roller ' δCR ' is directly entered by the user. The clearance is also defined as axial clearance.

As shown in the picture, a contact angle ranging from 90° to 0° can be set, but it should always be between 90° and 45° in order to suit its axial load-carrying design.

No radial forces can be accommodated, so the radial displacements ' uy ' and ' uz ' under loading have to be set to zero.

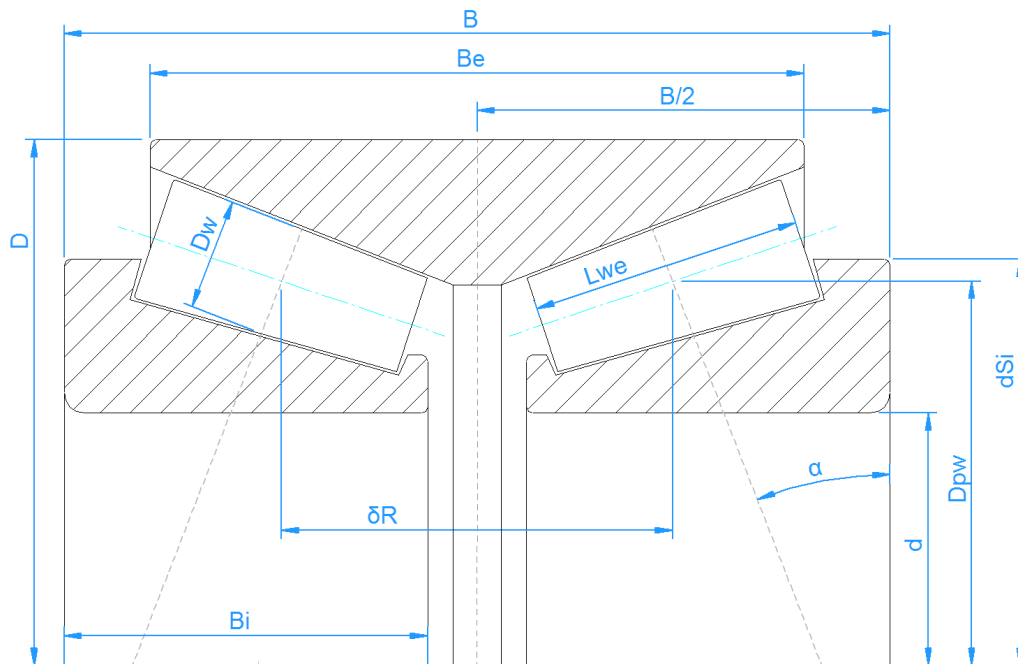
The direction of contact angle can be defined using the -button behind contact angle or bearing type.

The height of the shoulder of the inner diameter can be defined through the factor 'fSi' (%) in terms of percentage of the roller diameter. The force is assumed to be located at the medium height of the shoulder.

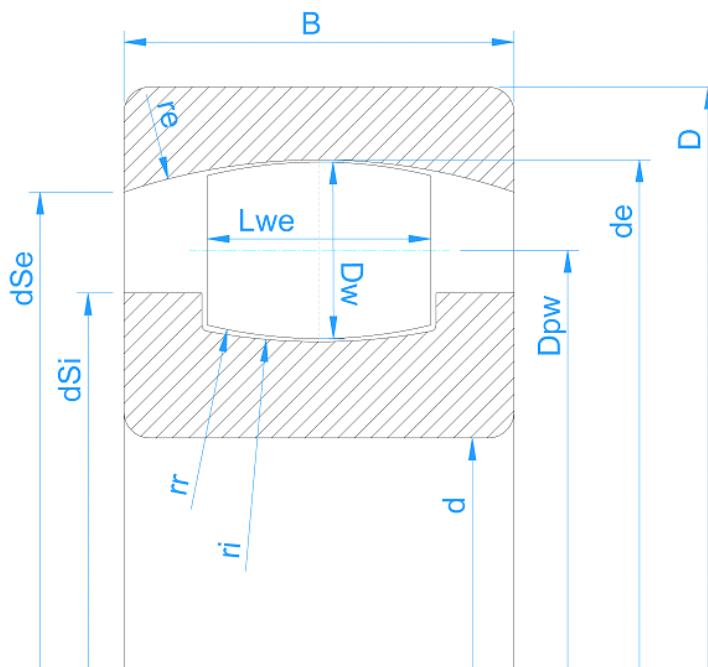
No tolerances are considered for axial tapered roller bearings.

3.1.2.1.20 Radial tapered roller bearing (double row)

As for other double row bearings types, the arrangement of contact angles can be defined in the options dialog. Unlike the single row radial tapered roller bearing, the distance between rows measured from the center of the corresponding rollers has to be entered. The same results are calculated as if using a single row bearing with a bearing configuration of two rows. The only difference is the load capacity which is shown for the double row bearing. As in the other bearing types, the clearance can be also defined as radial clearance.



3.1.2.1.21 Barrel roller bearings



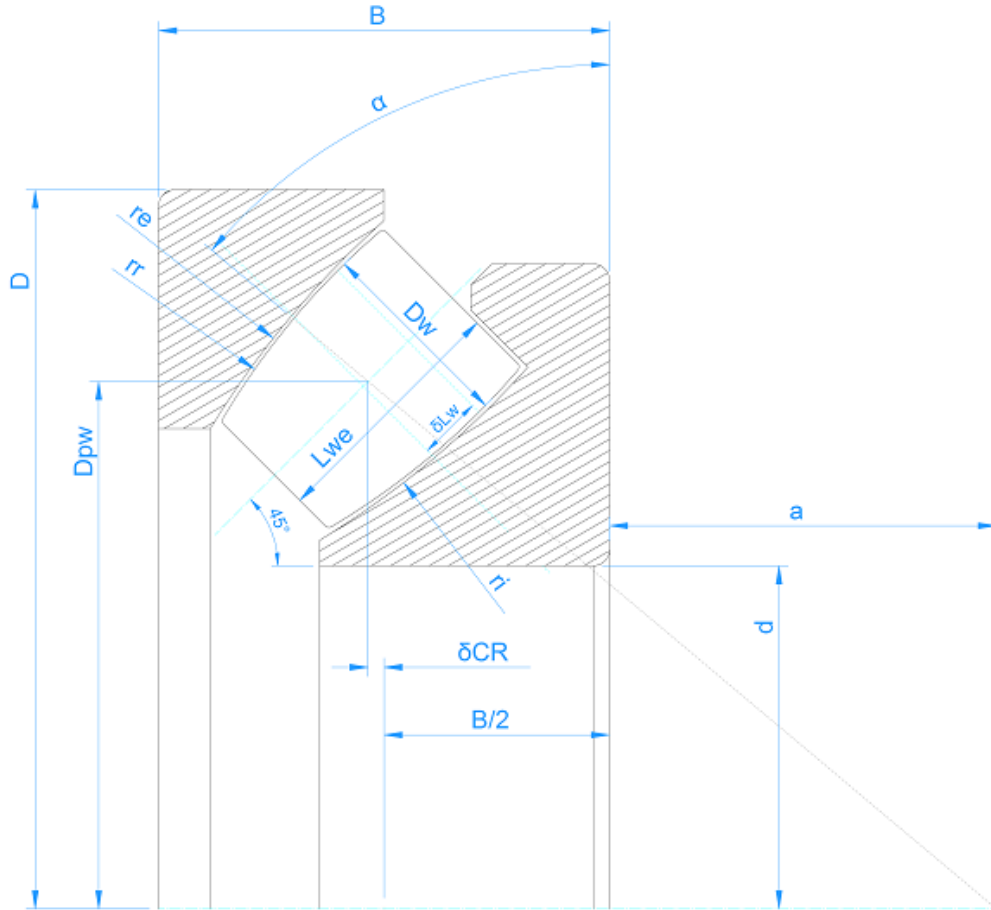
Barrel roller bearings are single row spherical roller bearings. As for double row spherical roller bearings, the outer race is a sphere and the bearing does not support moment loads. Therefore, the tilting angle has to be defined instead of moment load.

The radius of inner race, outer race and roller can be specified as ratio to the nominal diameter of the outer race: $d_e = D_{pw} + D_w$. Default parameters are: $f_e = 0.5$; $f_i = 0.5$; $f_r = 0.485$. If the radius for the outer race is chosen differently, the bearing cannot rotate freely any more.

The load capacity is calculated using $b_m=1$ as this seems to be the case in catalog data of major manufacturers.

3.1.2.1.22 Axial spherical roller bearing

The pitch diameter D_{pw} is defined as intersection of contact angle α and axis of roller. Usual values for contact angle are 45° to 50° .

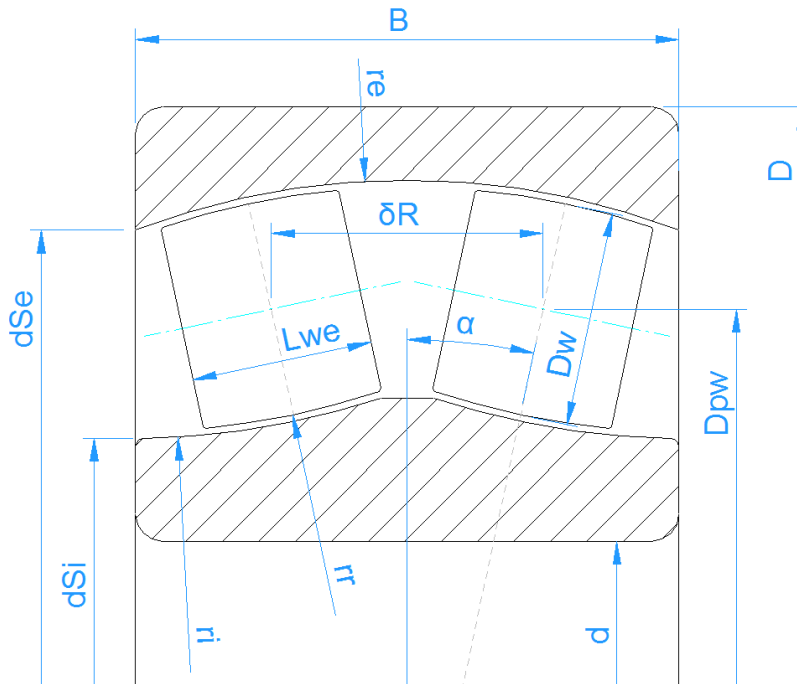


Radii of inner race and roller are defined by factors f_i and f_r like for the radial spherical roller bearings. They are defined as $r_i = f_i \cdot (2 \cdot r_e)$ and $r_r = f_r \cdot (2 \cdot r_e)$. Default values are $f_i = 0.5$ and $f_r = 0.485$. If the three radii r_i , r_e , r_r are entered using the button behind f_i , f_r , the factors f_i , f_r and the pitch diameter D_{pw} are calculated.

The unsymmetry of the roller is defined by δL as offset between center of roller and position of maximum roller diameter D_w . A sizing button is available to calculate δL so that the contact point is at the center of the roller.

The load capacities are calculated using the roller diameter D_{we} at the contact point. The contact stress on the roller is always shows from inner roller end to outer roller end, instead of left to right.

3.1.2.1.23 Spherical roller bearing

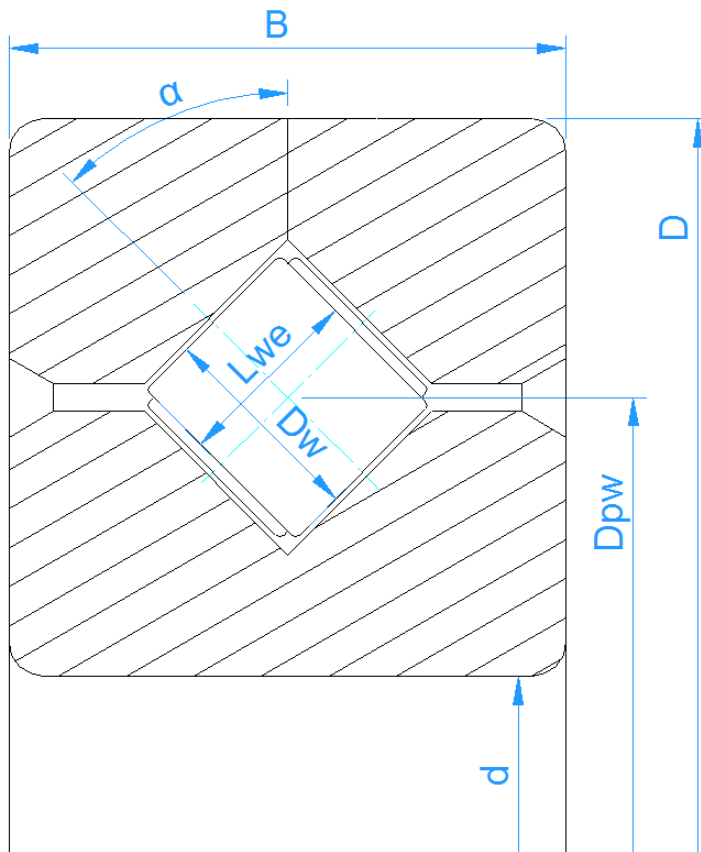


The distance between both rows is determined the contact angle and the pitch diameter. In contrast to the other bearing types, the nominal diametral clearance is only applied to the inner race.

The radius of inner race, outer race and roller can be specified as ratio to the

nominal diameter of the outer race: $d_e = \frac{D_{pw}}{\cos(\alpha)} + D_w$. Default parameters are: $f_e = 0.5; f_i = 0.5; f_r = 0.485$. If the radius for the outer race is chosen differently, the bearing cannot rotate freely anymore.

3.1.2.1.24 Cross roller bearings



Cross roller bearings have a contact angle of 45° . They can be selected as radial or axial bearings. Differences are the calculation of load capacity and the input of clearance as radial or axial clearance.

The number of rollers has to be entered for one row; so it is half of the total number of rollers. The length of the roller has to be smaller than its diameter.

3.1.2.1.25 Angular roller bearings

Angular roller bearings are similar to cross roller bearings, but all rollers are mounted in one direction. Therefore, the restriction on roller length does not apply anymore and the contact angle is available as input.

3.1.2.2 Approximation of bearing geometry

If the inner bearing geometry is not available, it can be approximated by the software. Four possibilities are available:

Enter outside geometry only

In this case only the outside geometry of the bearing is defined with inner diameter d , outer diameter D and width B . Additionally, the contact angle and the clearance have to be defined.

The number and size of rolling elements are approximated by the software. The load capacities are then calculated using this inner geometry. This does not lead to

accurate results, because the real bearing geometry is not used. But still influences of moment loads and other parameters could be seen.

Enter outside geometry and load capacities

In this case, the inner geometry is approximated as before, but the load capacities are provided by the user. The load capacities are usually available in bearing catalogues.

Enter inner geometry

Using this selection, you have to enter all the dimensions for inner geometry. The load capacities are calculated according to the standards.

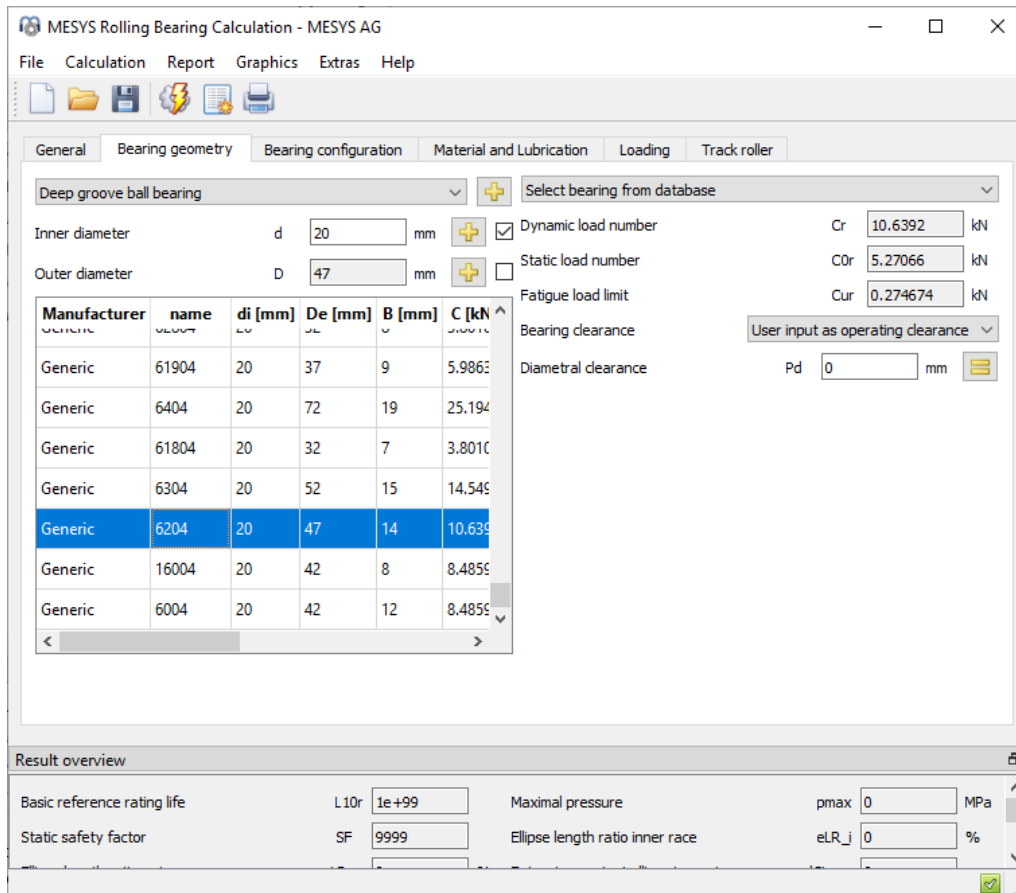
Enter inner geometry and load capacities

Since bearing manufacturers often use load capacities larger than calculated according to standards, it is possible to enter both: the inner geometry and the load capacities. The load capacities are then used for the calculation of life.

Select bearing from database

Instead of entering bearing geometry by the user, it can be selected from a database.

Inner and outer diameter can be defined optionally. This restricts the number of bearings shown in the list. By clicking on the titles of the columns, the data can be selected according to this column.



Double-clicking of a row reads in the bearing data and updates the values which are read from the database.

The data of inner geometry of the bearings provided with the database are approximated from outer dimensions. Additional databases from bearing manufacturers are available. Catalog data with approximated inner geometry is available from Schaeffler and SKF and is included in the installation. Encrypted bearing databases including inner geometry are available from GMN, HQW and IBC, please contact the bearing manufacturer for these databases.

Bearings with approximated bearing inner geometry and bearings with manufacturer setting "Generic" are shown with default font, custom bearings and databases with inner geometry information are shown in bold font.

General	Bearing geometry	Bearing configuration	Material and Lubrication	Loading	Track roller		
Angular contact ball bearing		Enter outside geometry only					
Inner diameter	d	95	mm	Dynamic load number	Cr	61.88	kN
Outer diameter	D	145	mm	Static load number	CO _r	63.5152	kN
Width	B	24	mm	Fatigue load limit	Cur	3.02162	kN
Number of rolling elements	Z	22		Bearing clearance	User input		
Diameter of rolling elements	D _w	14	mm	Axial clearance	Pa	0	mm
Pitch diameter	D _{pw}	120	mm	Bearing tolerance	ISO 492 - P4		
Contact angle	α ₀	25	°	Fit to shaft	k6		
Conformity inner ring	f _i	0.52		Surface roughness shaft	Rz	4	μm
Conformity outer ring	f _e	0.52		Shaft inner diameter	d _{si}	0	mm
Shoulder diameter inner ring	d _{Si}	114.4	mm	Fit to housing	H7		
Shoulder diameter outer ring	d _{Se}	125.6	mm	Surface roughness housing	Rz	4	μm
				Housing outer diameter	d _{he}	0	mm

3.1.2.3 Load capacities

Dependent on the setting for the approximation of bearing geometry, the load capacities can be an input or an output. If they are given by the user, they will not be changed because of surface hardness of the material. The surface hardness is only considered if the values are calculated by the software.

3.1.2.3.1 Dynamic load number

The dynamic load number is used for the calculation of bearing life. It is calculated according (ISO 281, 2007) with factor f_c according (ISO/TR 1281-1, 2008). The factor b_m can be influenced for filling slot bearings using the options dialog for the bearing type (+-button behind bearing type). If the option for hybrid bearing is selected in the options dialog, the dynamic load capacity is calculated according to (ISO 20056-1, 2017), leading to larger b_m and smaller f_c , but unchanged load capacity compared to steel bearings.

The dynamic load number can be influenced by a modification factor available at material properties.

3.1.2.3.2 Static load number

The static load number is calculated according (ISO 76, 2006) and (ISO/TR 10657, 1991). It is only for documentation and only used in the calculation of static safety factor if the static safety is not calculated based on stresses (see settings "Calculate static safety factor based on stresses"). It is also used in one case for track roller calculation, see that section for details.

If the option for hybrid bearing is selected in the bearing options dialog, the static load capacity is calculated according to (ISO 20056-2, 2017), leading to larger static

load capacities in case of silicon nitride rolling elements with Youngs Modulus larger than 300GPa . Note that (ISO 20056-2, 2017) also uses higher values for recommendations of static safety factors.

The static load number is based on a permissible stress which can be changed at inputs for material data.

For bearings with low speed higher loads are permitted sometimes. For example, (ISO 1002, 1983) allows a radial force of over five times the static load capacity for nonrotating ball bearings.

3.1.2.3.3 Fatigue load limit

The fatigue load limit is calculated according (ISO 281, 2007) section B.3.2.1.2 for ball bearings and according section B.3.2.1.3 for roller bearings. For roller bearings, the standard calculation according ISO/TS 16281 is used, not the extended method for pressure distribution.


The fatigue load limit is based on fatigue strength of 1500MPa and it is used for the calculation of modified life. The fatigue strength can be modified using inputs for material data.



3.1.2.4 Inner, outer diameter and width

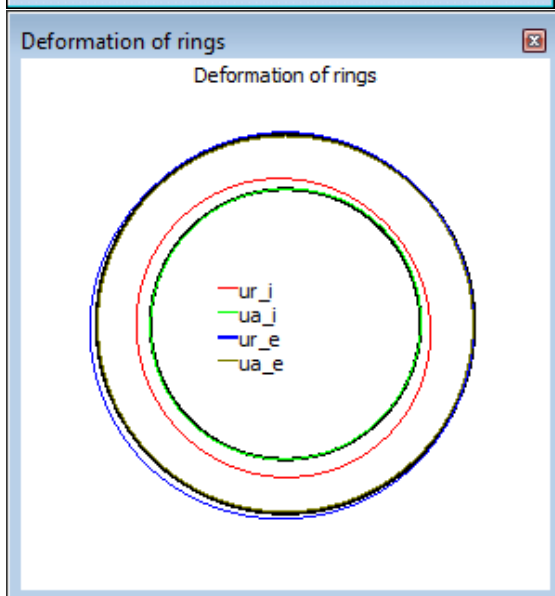
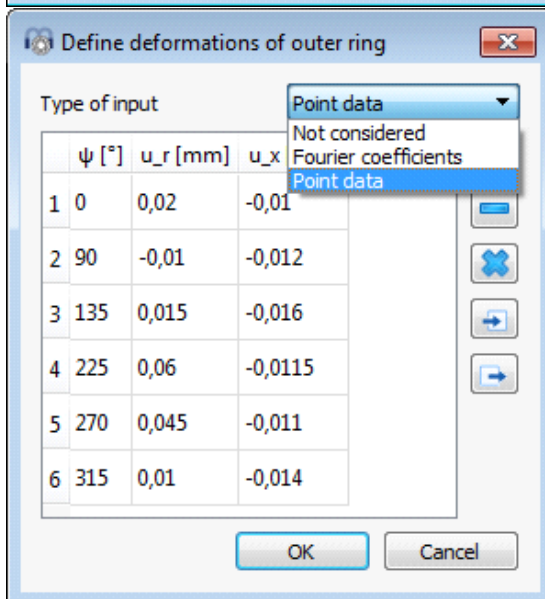
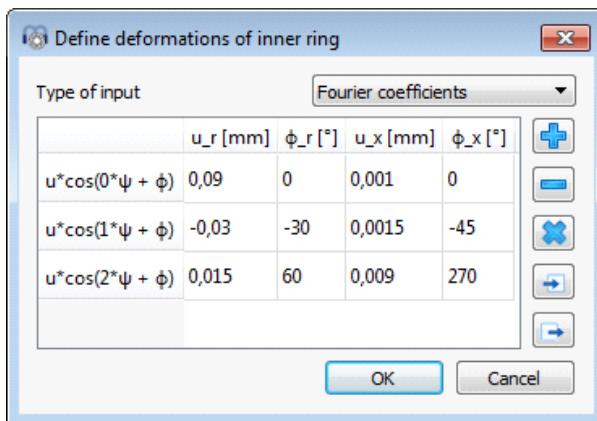
The inner diameter, outer diameter and width are only needed for documentation and for the approximation of inner geometry. They are not used in the calculation itself.

If the inner geometry is provided, these values could be set to zero.



3.1.2.5 Deformations of rings

If deformations of outer and inner ring are, for instance, known by means of FE analysis, they can be entered into the software to evaluate their influence on bearing life and contact stress by clicking on the -button behind both diameter input fields.

Inner diameter	d	100	mm	
Outer diameter	D	140	mm	



They are two possibilities. On one hand, with the option 'Point data', as many points of deformation as needed can be added to the data table. Any point is positioned circumferentially by its angle and both the axial and radial deformation

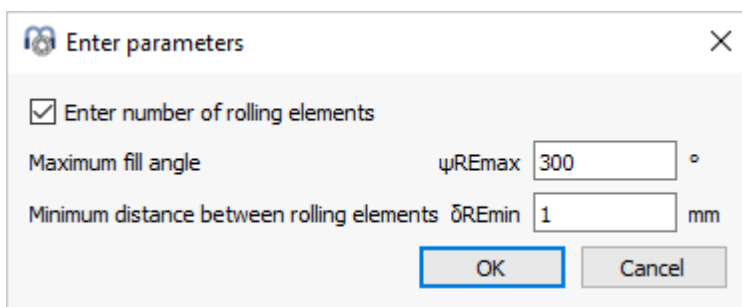
can be defined. Note that the deformations along the rings between any defined points will be linearly interpolated. On the other hand, there is an option to define an approximate deformation curve by 'Fourier coefficients' as shown in the picture below. Additionally, a whole table can be imported from a csv-file using the -button and any created table can be exported into a file using the -button.

In case of shaft calculations with 3D-elastic parts a shaft deformation can be transferred to the bearing calculation. In this case a check-box "Consider shaft deformations" is shown. If this check-box is cleared the shaft deformations are removed and not considered any more.

These deformations and deformations transferred from the shaft calculation are not considered in outputs for bearing clearance or the tolerance report. They only affect the calculation of load distribution and life.


In the current version, the definition of deformations cannot be used together with track roller calculation with elastic outer ring.

3.1.2.6 Number of rolling elements



The number of rolling elements has to be specified. A minimum number is three; the maximum number depends on the bearing pitch diameter. A warning is shown if the rolling elements overlap, the distance between rolling elements is available in the results overview and report.


For cross-roller bearings, this is the number of rollers for one row.

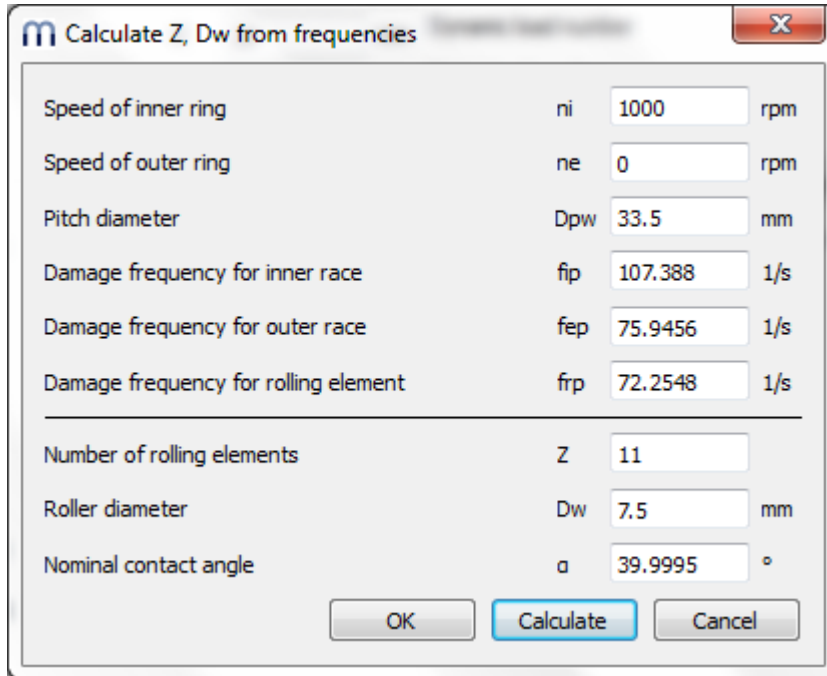
Using the -button the number of rolling elements can be calculated automatically based on a maximum fill angle and a minimum distance between the rolling elements. For deep groove ball bearings, the fill angle would be 180-200°, the minimum distance can be based on requirements for the cage.

This option is mainly thought to allow the variation of the rolling element diameter in the parameter variation with an automatic setting of the number of rolling elements.

3.1.2.7 Diameter of rolling elements

The diameter of the rolling elements is specified here. For taper roller bearings, the diameter in the middle of the roller is used.


Using the -button, the number of rolling elements, roller diameter and contact angle can be calculated from given damage frequencies. This can be used if damage frequencies for a bearing are given and geometry data is missing.




Parameter	Symbol	Value	Unit
Speed of inner ring	n_i	1000	rpm
Speed of outer ring	n_e	0	rpm
Pitch diameter	D_{pw}	33.5	mm
Damage frequency for inner race	f_{ip}	107.388	1/s
Damage frequency for outer race	f_{ep}	75.9456	1/s
Damage frequency for rolling element	f_{rp}	72.2548	1/s
Number of rolling elements	Z	11	
Roller diameter	D_w	7.5	mm
Nominal contact angle	α	39.9995	°

3.1.2.8 Pitch diameter


The pitch diameter is the diameter of the centers of rolling elements when they have the same amount of clearance to both races. The diameters of inner and outer race are calculated by this value, the diameter of rolling element, the contact angle and the clearance.

Usually, the mean value of inner and outer diameter of the bearing is taken, but it can be different. For example, it is not valid for a needle bearing without inner ring. The mean value can be selected using the -button behind the input.

3.1.2.9 Contact angle

The contact angle has to be provided for angular contact bearings, four-point ball bearings, spherical ball bearings, taper roller bearings and spherical roller bearings. For taper roller bearings, the cup angle is used since this is the direction of the force. Using the -button, the direction of contact angle can be selected.

For double row angular contact bearings, the back-to-back arrangement has the larger width between the centers of pressure and it is also called O-configuration, while the face-to-face arrangement is the same as the X-configuration.


For three point bearings this input is used for the shim angle of the split ring. The -button allows to calculate the shim angle by given shim width. The resulting contact angle for axial contact is available as free contact angle in the results.

3.1.2.10 Conformity of inner and outer ring


The conformity is the ratio between radius of curvature of a race and the ball diameter. For geometric reasons, the value has to be larger than 0.5. The values used for the calculation of tables in ISO 281 are $f_i=f_e=0.52$ for radial ball bearings and $f_i=f_e=0.535$ for thrust ball bearings as stated in (ISO/TR 1281-1, 2008).

According to (ISO/TS 16281, 2008) usual values are $f_i=0.52$ and $f_e=0.53$ for radial ball bearings and $f_i=f_e=0.54$ for thrust ball bearings. (ISO 76, 2006) also uses $f_i=f_e=0.54$ for thrust ball bearings.

For spherical or barrel roller bearings values of $f_i=f_e=0.5$ should be used. Here the diameter of the outer race d_e is used as reference.

According to (ISO 281, 2007) and (ISO 76, 2006) a larger conformity reduces the load capacity and therefore a reduction of f_c is required, but a smaller conformity does "not necessarily increase the load-carrying ability". Therefore, the software allows to enter a limit for the conformity used for the calculation of load capacity. See the -button behind the bearing type for ball bearings.

For spherical ball bearings, the diameter of outer race d_e is used as reference for f_e . So $f_e=0.5$ should be used as default.

Using the -button behind the input field, the conformity can be calculated from a given radius. Ensure that you input the correct roller and pitch diameter before.

3.1.2.11 Conformity of roller

For spherical or barrel roller bearings, the conformity of the roller has a usual value of $f_r=0.485$. The diameter of the outer race d_e is used as reference, so $r = f_r * d_e$.


3.1.2.12 Distance between rows

For double row bearings, the distance between rows is shown in the user interface. It is the axial distance between the centers of balls. For double row angular contact bearings, it is an input, for the other bearing types it is an output since the distance is calculated by pitch diameter and contact angle. The distance between rows is

$$\delta_R = D_{pw} \cdot \tan \alpha$$

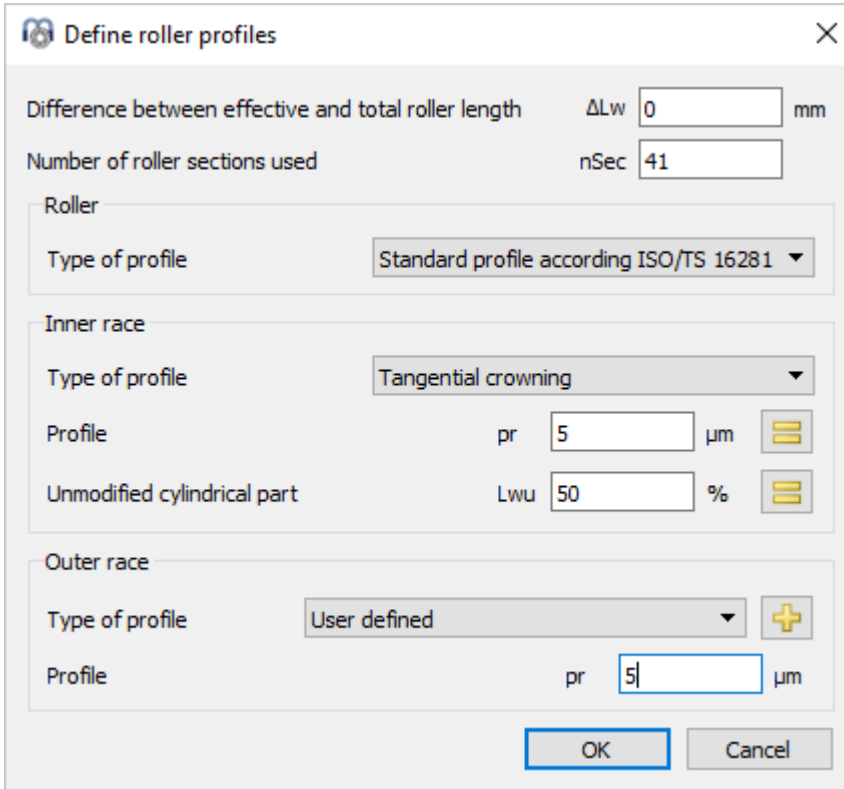
3.1.2.13 Effective length of roller

The effective length of the roller is the length that can be loaded. A radius at the end of the roller has to be subtracted to get the effective length.

If the extended method pressure calculation for pressure distribution is selected on page 'General', the roller profile can be defined using the -button.

3.1.2.13.1 Defining roller profiles

Profiles can be defined for inner race, outer race and roller.



Define roller profiles

Difference between effective and total roller length ΔL_w mm


Number of roller sections used nSec


Roller

Type of profile


Inner race

Type of profile

Profile pr μm 

Unmodified cylindrical part Lwu % 

Outer race

Type of profile 

Profile pr μm

As general inputs, the difference between effective and total roller length can be defined. As it increases the stiffness of the roller, it has an influence on the calculation.

The number of sections for the calculation has to be larger 30 and has to be an odd number. The calculation accuracy and the calculation time increase with a larger number of sections.

Four options are available for the profiles:

1. Standard profile according ISO/TS 16281:
This is using the defaults of the standard, which is logarithmic profile of the roller and no profile on the races.
2. Logarithmic profile:
The amount of relief can be defined and a cylindrical part that should not be modified. The following formula is used:

$$P(x) = Pr \cdot \ln \left(\frac{1}{1 - \left\{ \frac{2|x| - L_{wu}}{L_{we} - L_{wu}} \right\}^2} \right)$$

For the roller, a sizing button provides the default relief according ISO/TS 16281. The default relief of $0.00035 \cdot D_w$ for short cylindrical roller bearings corresponds to a contact stress of 3000MPa.

3. Modified logarithmic profile:

The amount of relief is calculated according following formula derived from (Fujiwara, et al., 2010):

$$P(x) = A \cdot \ln \left(\frac{1}{1 - \left\{ 1 - e^{-\frac{Pr}{A}} \right\} \left(\frac{2|x| - L_{wu}}{L_{we} - L_{wu}} \right)^2} \right)$$

using

$$A = 2D_w \cdot \left(\frac{p \cdot (1 - \nu^2)}{E} \right)^2$$

The maximum relief is not infinity at the edge of the roller like in the logarithmic profile according ISO/TS 16281 but limited to the maximum value of pr .

Using $p = 3000\text{MPa}$ and $pr = \text{Infinity}$, the results are the same as for the logarithmic profile according ISO/TS 16281 above.

The material data for this profile is always taken from the contact roller/inner race.

4. Tangential crowning:

A circular arc tangential to the cylinder is used to generate the profile. The amount of relief and a percentage of unmodified cylinder length can be defined.

5. Crowning:

A circular arc with center positioned axially on the center of the roller is used to generate the profile. The amount of relief and a percentage of unmodified cylinder length and a transition radius can be defined.

6. Two tangential radii:

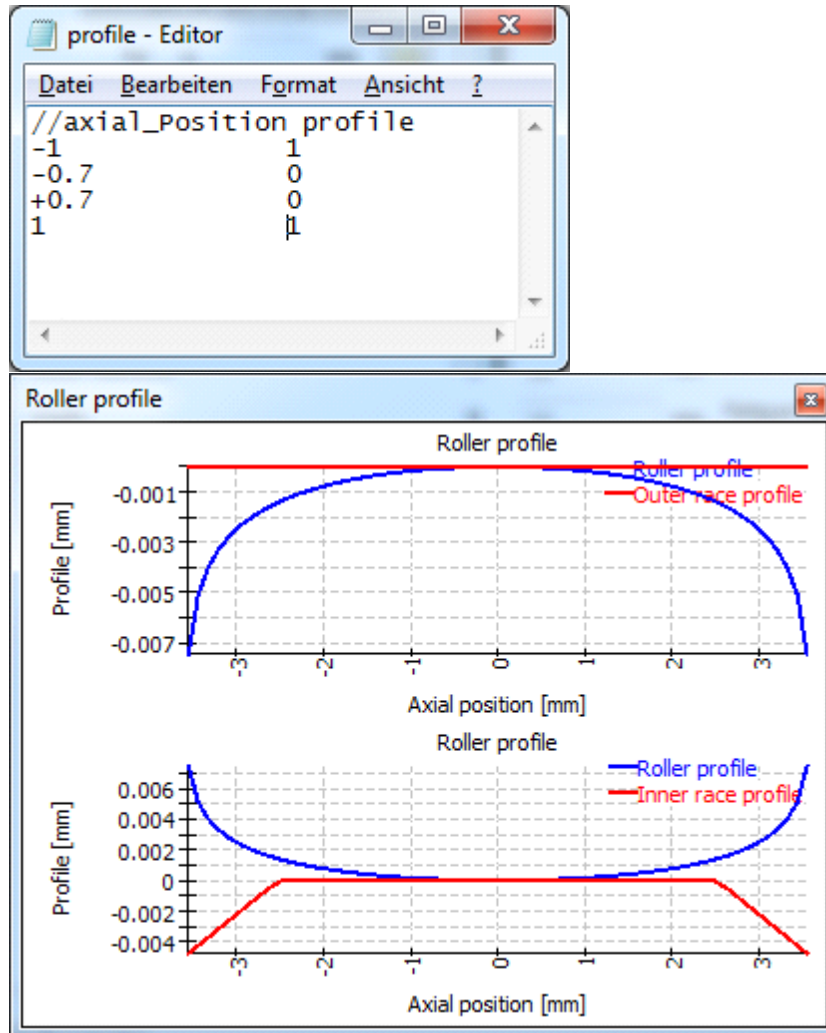
A first cylindrical part with width " L_{wu} " is extended by two tangential radii. The width of the first radius is defined as percentage " L_{wr1} ". A requirement is " $L_{wr1} > L_{wu}$ ". " L_{wu} " can be set to zero to have two radii only.

7. Read file:

The profile can be read from a file. The data has to be specified with two values on each line, an axial position and the profile relief. The axial position is scaled with the roller length and the relief is scaled with the

input for the profile relief. So a range from -1 to +1 can be used for the axial position and 0 to 1 for the relief.

A very simple file is shown here. It was selected for the inner race profile on the right:

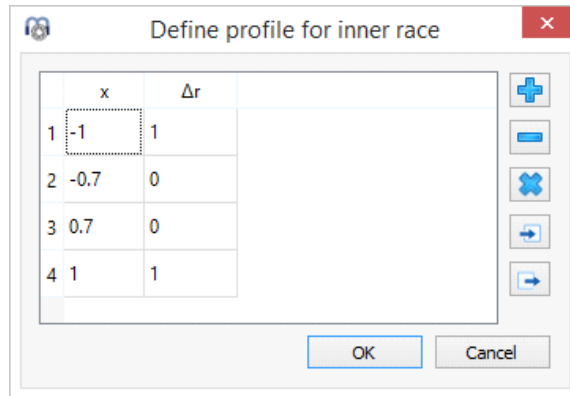


The unsymmetrical profile is used from left to right for most bearings. For taper roller bearings and axial spherical roller bearings, the profiles is used from left to right if direction of contact angle is left. For contact angle to the right, the profile is mirrored. So mounting conditions do not influence the profile.

8. Read file without scaling:
Like option "Read file" the data is read from a file. Using this option no scaling will be done. The axial extension has to be defined from $-L_{we}/2$ to $+L_{we}/2$ or from 0 to L_{we} . The profile is defined in "mm" and won't be scaled too.
9. User Input:
Instead of defining a file name, the points for the profile can be defined in

a table. Again as above, use a range of -1 to +1 for the axial position and a value between 0 and 1 for the profile relief.

The table in the diagram on the right shows the same data as the file above.



The profiles can be shown in the graphics for roller profile.

3.1.2.14 Offset for roller center

For axial spherical roller bearings, an offset for the roller center δL can be defined. It is the distance between the center of the roller and the position of the maximum diameter D_w .

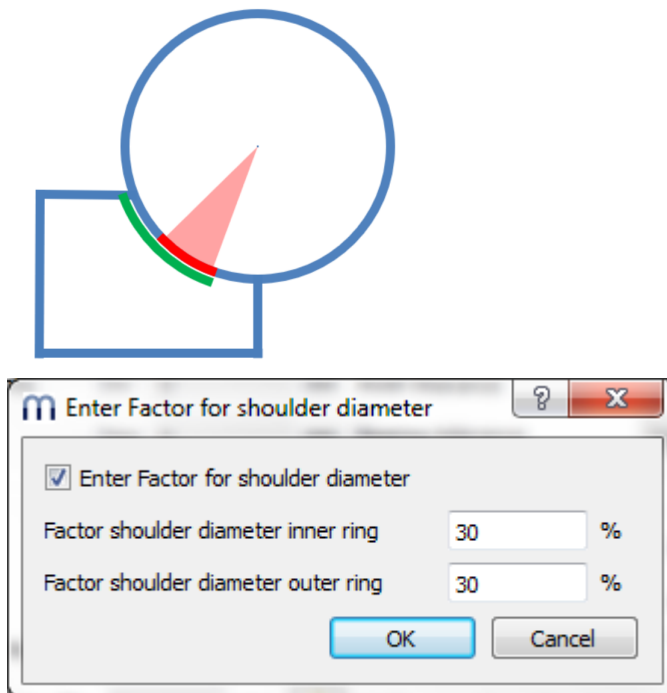
3.1.2.15 Shoulder diameter of inner and outer ring

For ball bearings and radial cylindrical roller bearings, the shoulder diameter of inner and outer ring can be defined. For tapered roller bearings, the shoulder for the inner race can be defined. If an input of zero is given, it will not be considered.

The shoulder diameter is compared to the maximum extension of the contact ellipsis. A warning is shown if the ellipsis would extend above the shoulder. The Hertzian stress is not valid in that case.

The required shoulder diameter is shown in the report together with a length ratio eLR_i , eLR_e which is showing a safety of shoulder length. The length ratio is defined as the length from lower end of contact ellipsis to shoulder (the green line in the graphics) divided by the length of the contact ellipsis $2a$ (red line in the graphics). The value should therefore be larger than 1 or 100%.

For spherical ball bearings, the shoulder diameter of the outer race is limited by the radius of outer race and bearing width automatically if the bearing width is entered.



Instead of using an absolute value, the shoulder height can be defined using a percentage of ball diameters. A factor of 50% would be a shoulder up to the pitch diameter, so the factor should be between 10% and 40%. Using this factor allows to have default geometry on changing ball diameter or pitch diameter.

For radial cylindrical roller bearings and tapered roller bearings, it is assumed that the axial force is acting at half the height of the shoulder.

3.1.2.16 Distance to row center


This input is only available for taper roller bearings and axial spherical roller bearings. It is the distance between center of roller and center of bearing. It can be calculated from the distance to center of pressure 'a' provided in bearing catalogs.

3.1.2.17 Bearing clearance

The bearing clearance can be set automatically according (ISO 5753, 2009) (C2...C5) for deep groove ball bearings, four-point bearings, self-aligning ball bearings, spherical and cylindrical roller bearings.

In addition, there are the settings 'from database', 'own input as operating clearance' and 'own input':

- If 'From database' is selected, the minimal and maximal values for the bearing clearance are taken from the bearing database. An error message is shown if the bearing data is not selected from database. It is not necessary to use the clearance from database, also the other options for clearance can be selected.

- Selection 'user input as operating clearance' means that no modification to the clearance will be done by the software. No interference fit or temperature is taken into account. This is the recommended setting if you know the clearance you want to consider.
- Selecting 'user input' allows entering the clearance of the bearing before mounting. Influence of temperature or interference fits will be considered additionally.
- Selecting 'user input as range' allows entering the clearance of the bearing before mounting as a range using the -button. Influence of temperature or interference fits will be considered additionally. The load distribution will be calculated using the clearance according to the setting as minimum, medium or maximum value of the range.


3.1.2.18 Diametral clearance

The diametral clearance P_d is shown for deep groove radial ball bearings, spherical ball bearings, cylindrical roller bearings and spherical roller bearings. It is the distance between the upper and the lower position the inner ring can have without load in a fixed outer ring. It can be calculated as

$$P_d = d_e - d_i - 2 \cdot D_w$$

For pretension, a negative clearance can be entered.

The same value is sometimes described as radial clearance (in contrast to axial clearance), but the measurement is still a difference of diameters.

A -button allows the input of axial clearance instead of diametral clearance. For deep groove ball bearings, the clearance can also be calculated for given free contact angle.

3.1.2.19 Axial clearance

The axial clearance P_a is shown for angular contact bearings, four-point bearings, cylindrical roller bearings and all axial bearings. The axial clearance is the difference between the possible axial displacement in positive and negative direction for four-point bearings and cylindrical roller bearings.


For single row angular contact bearings or tapered roller bearings, it is the axial distance between inner and outer ring when the bearing rings are moved together with a very small axial load. So in this case, the clearance is between center and one direction and for double row angular contact bearings, four-point bearings and cylindrical roller bearings, it is the maximum distance between left and right movement.

For cylindrical roller bearings, a value larger than zero should be entered for the axial clearance, since the bearing usually has axial clearance. If the calculation is done with clearance zero, you will get some axial loads on the rollers if there is a tilting angle of the bearing. This should be the case for a locating cylindrical roller bearing, but not for a non-locating bearing. So the correct clearance should be entered in both cases.

Calculation of axial clearance Pa

Pretension force	Fp	82.7936	N	<input checked="" type="radio"/>
Unmounted pretension force	Fpu	76.2862	N	<input type="radio"/>
Effective pretension force	FpEff	341.757	N	<input type="radio"/>

OK Cancel

A pretension can be entered by using a negative value for the axial clearance Pa. A -button allows the input of diametral clearance instead of axial clearance. For angular contact bearings or tapered roller bearings, the axial clearance can also be calculated for a given pretension force.

For roller bearings, the calculation of axial clearance from pretension force does not yet take into account the extended pressure calculation or changes of profiles.

Dependent on the bearing type, several options for the pretension force are available. The calculation with “Fp” is using nominal dimensions for the bearings, the calculation using “Fpu” is using nominal dimensions together with a radial elastic expansion without limit; this option can be used for the case of loading an unmounted bearing axially during manufacturing. The effective pretension force “FpEff” is calculated using the dimensions after fitting.

3.1.2.20 Bearing tolerance

The bearing tolerance can be considered according ISO 492 (P0...P2). If selecting ‘Not considered’, the interference fit is not considered for operating clearance, in this case only temperature will affect the operating clearance, but centrifugal forces are not considered.


If selecting ‘Nominal dimensions’, the tolerances of the bearing are considered zero. Then the interference fit can be defined with the inputs for the shaft only.

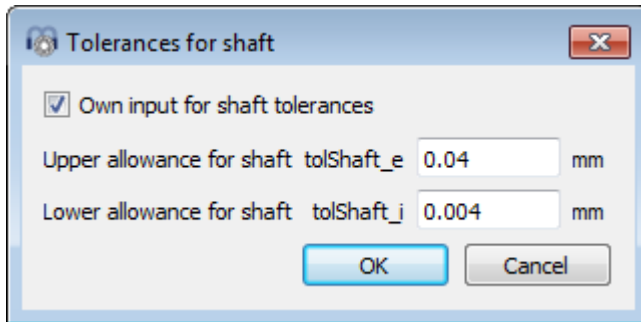
Selecting ‘User input’ allows the definition of custom tolerances.

If no tolerance range should be considered, the selection 'Define interference' is useful. Then the interference between shaft/inner ring and between housing/outer ring can be defined directly. No reduction because of surface roughness is considered in this case.

3.1.2.21 Fit to shaft/housing

The fit between inner ring and shaft can be defined as a tolerance according (ISO 286-1, 2010). For example, it can be specified as 'k6'. The same is possible for the fit between outer ring and housing.

A numerical value can be specified directly using the -button next to the input field. Selecting "own input" allows the input of arbitrary tolerances:



3.1.2.22 Shaft inner diameter/Housing outer diameter

The inner diameter of the shaft can be specified for a hollow shaft. This has an effect on the interference fit.

For the housing, the outer diameter can be specified. If zero is entered, a value of 2·D will be used in the calculation.

3.1.2.23 Surface roughness shaft and housing

The surface roughness of the shaft and the housing is only used for the calculation of the interference fit. The effective interference of the parts is reduced by $0.4 \cdot R_z$ according (DIN 7190-1, 2017). In previous versions, a reduction of $0.8 \cdot R_z$ was used as in the previous version of the standard.

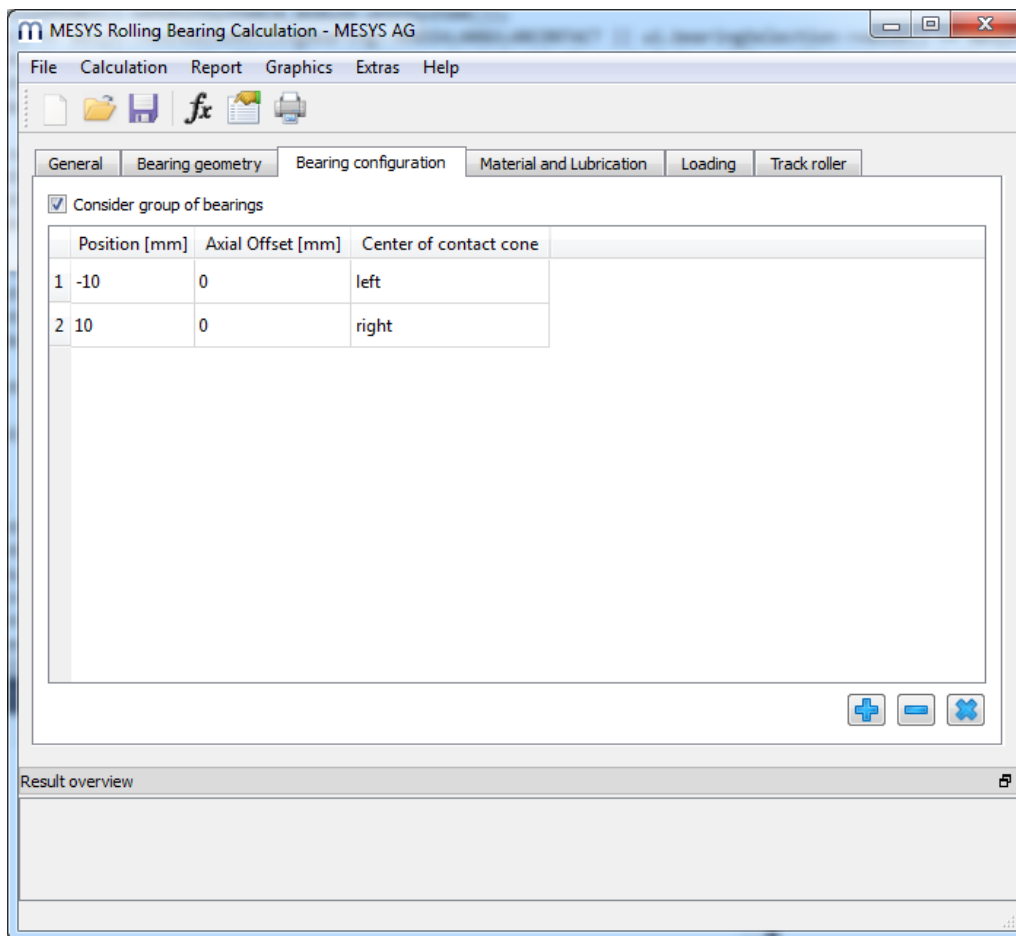
The surface roughness of shaft and housing is only used for the calculation of the interference fit.

3.1.3 Bearing configuration

A group of bearings can be considered in the calculation. The bearings are assumed to be connected to a rigid shaft and housing. The loads or displacements are given for the origin of the bearing group instead for each single bearing.

A relative position can be entered for each bearing. An axial offset allows the definition of pretension in the system. The offset is an additional movement of the outer ring of a bearing. The displacements, forces and moments under loading are applied at the reference position 0. So in most cases, the positions should be chosen symmetrically around zero.

For bearings with a contact angle, the position of the cone center can be specified. In the picture, an O-configuration (or back-to-back configuration) is defined.



Additional bearings can be added using the **+**-button. A row can be deleted using the **-** - and the **X**-button clears all inputs.

The possibility of bearings groups allows the calculation of 8-point ball bearings as two rows of four-point bearings; it allows sets of angular contact bearings, paired taper rollers or sets of cylindrical roller bearings which could be used in a planetary gear.

With the context menu (right mouse button) in the table, several results values can be shown in the table. This allows seeing quickly how load or life is distributed among the bearing rows.

3.1.4 Lubrication

The lubrication has an effect on the modified life rating. A large oil film thickness prevents direct contact between rolling element and races and therefore can increase the life.

Some lubricants can directly be selected from the list. Mineral oils and PAG-based synthetic oils are provided. For other types of oil, 'own input' can be selected.

Lubrication

ISO VG 220 mineral oil Oil lubrication with on-line filter ISO4406 -/17/14 +

Viscosity at 40°C nu40 mm²/s Temperature TOil °C

Viscosity at 100°C nu100 mm²/s Oil density ρ kg/m³

contains effective EP additives Pressure viscosity coefficient α 1/MPa

3.1.4.1 Contamination factor

The contamination factor is used for the calculation of life modification factor a_{ISO} . It is considering the cleanness of the lubricant and can be calculated with the viscosity ratio, the bearing size and a selection of oil cleanness according ISO 4406. The selection according ISO 4406 defines how many particles of certain size may exist.

Also an own input of the contamination factor can be selected and the factor can be entered using the +-button. The contamination factor is in the range from 0.1 to 1 and guide values can be selected according (ISO 281, 2007):

Contamination level	e_C	
	$D_{pw} < 100\text{mm}$	$D_{pw} \geq 100\text{mm}$
Extreme cleanliness Particle size of order of lubricant film thickness, laboratory conditions	1	1
High cleanliness Oil filtered through extremely fine filter; conditions typical for bearings greased for life and sealed	0.8 to 0.6	0.9 to 0.8
Normal cleanliness Oil filtered through fine filter; conditions typical for bearings greased for life and shielded	0.6 to 0.5	0.8 to 0.6
Slight contamination	0.5 to 0.3	0.6 to 0.4
Typical contamination Conditions typical of bearings without seals; course filtering; wear particles from surroundings	0.3 to 0.1	0.4 to 0.2
Severe contamination Bearing environment heavily contaminated and bearing arrangement with inadequate sealing	0.1 to 0	0.1 to 0
Very severe contamination	0	0

Note that for small bearings, the values for e_C calculated from oil cleanliness can be much lower than in the table.

3.1.4.2 Kinematic viscosity

The kinematic viscosity has to be given for two temperatures: ν_{40} at 40°C and ν_{100} at 100°C. The actual viscosity at the given temperature will then be interpolated using these values.

If a single kinematic viscosity at operating temperature is given, it can just be entered in both inputs and therefore will be used in the calculation.

A high viscosity results in a larger lubricant film, but efficiency will be reduced if it is too large.

3.1.4.3 Oil temperature

The oil temperature is only used for the calculation of lubricant properties like operating viscosity. It has no influence on clearance. If the oil temperature is not selected as input, it will be set to the temperature of the rolling elements as calculated from the temperatures of the rings dependent on the settings or as given on page "Loading".

3.1.4.4 Oil density

The oil density is used for the calculation of dynamic viscosity. If a lubricant from the list is selected, the density is modified according to the oil temperature according (Niemann, et al., 2005). If 'own input' is selected, the input value is directly used for the calculation.

3.1.4.5 Pressure viscosity coefficient

The pressure viscosity coefficient α is used for the calculation of lubricant film thickness. If the film thickness should not be calculated, you can just enter zero.


According to (ISO/TR 1281-2, 2008) the pressure viscosity coefficient α can be calculated by the kinematic viscosity ν_0 in cm^2/s as

$$\alpha = 0.1122 \cdot \left(\frac{\nu_0}{10^4} \right)^{0.163}$$

To calculate the reference viscosity ν_1 , a value of $\alpha = 0.0077 \cdot \nu^{0.204}$ using ν in mm^2/s and α in mm^2/N is used according (Baalmann, 1994).

In the software, the proposal according (AGMA 925, 2003) is used, because different oil types are considered. The pressure viscosity coefficient is calculated as $\alpha = k \cdot \eta_M^s$, where η_M is the dynamic viscosity at operating temperature. The factors k and s are given for different lubricant types as shown in the following table:










Oil type	k	s
Mineral oil	0.010471	0.1348
PAO based synthetic non-VI improved oil	0.010326	0.0507
PAG based synthetic	0.0047	0.1572

This formula is used if a lubricant from the list is selected. If 'own input' is selected, either the formula according to (ISO/TR 1281-2, 2008) 11.8 or according to (AGMA 925, 2003) or a direct input can be selected using the -button next to the input field.

3.1.5 Material

Material properties for rolling element, inner and outer ring, shaft and housing can be defined. This data is used for the calculation of load distribution and interference fit between bearing and shaft/housing. Also hardness and surface roughness can be specified.

Material

Surface hardness inner race	<input type="text" value="58"/>	HRC	Surface Hardness outer race	<input type="text" value="58"/>	HRC
Core strength inner race	Rm <input type="text" value="1200"/>	MPa	Core strength outer race	Rm <input type="text" value="1200"/>	MPa
Hardness depth inner race	hdi <input type="text" value="0"/>	mm	Hardness depth outer race	hde <input type="text" value="0"/>	mm
Surface roughness inner race	Rq <input type="text" value="0.167006"/>	μm		Surface roughness outer race	Rq <input type="text" value="0.167006"/>
					
Surface roughness roller	Rq <input type="text" value="0.167006"/>	μm		Material inner race	<input type="text" value="Steel"/>
					
Material outer race	<input type="text" value="Steel"/>		Material rolling element	<input type="text" value="Steel"/>	
Material shaft	<input type="text" value="Steel"/>		Material housing	<input type="text" value="Steel"/>	

3.1.5.1 Surface hardness

The surface hardness is given as Rockwell hardness HRC. If the surface hardness is smaller than 58 HRC, a reduction in dynamic and static load capacity is taken into account according (Harris, et al., 2009). This reduction is only considered as long as the load capacity is calculated by the software. The reduction factor for dynamic load capacity is:

$$C' = C \left(\frac{HRC}{58} \right)^{3.6}$$

An input value for load capacity is not modified because of the hardness value. If a modification factor for dynamic load capacity or the permissible stress for static capacity is defined in material data, a reduction because of hardness is not considered.

The static safety factor is also affected by a reduced hardness. This is also done according (Harris, et al., 2009) using the factor f_s :

$$f_s = 1.5 \left(\frac{HV}{800} \right)^2 \text{ (Ball bearings) or } f_s = 2 \left(\frac{HV}{800} \right)^2 \text{ (roller bearings), with } f_s \leq 1$$

For this formula, HV is calculated as $HV = (223 \cdot HRC + 14500) / (100 - HRC)$ according (Schlecht, 2010).

3.1.5.2 Core strength of inner and outer race

The ultimate strength of the inner and outer ring core and the corresponding hardness depth are used for a check of case core interface. The actual shear stress at hardness depth is compared with the shear yield stress τ_{yield} and the shear fatigue strength τ_a which are calculated according to (Harris, et al., 2009) from the input value of ultimate strength:

$$\tau_{yield} = 0.425 \cdot R_m; \tau_a = 0.6 \cdot \tau_{yield}$$


If the material is though hardened, the software can be set to calculate the hardness depth automatically. Then no warnings will occur, just a larger hardness depth is calculated on load increase.

3.1.5.3 Surface roughness

The surface roughness R_q is used for the calculation of the specific lubricant film thickness Λ . It is a root mean square roughness R_q or R_{RMS} . According to (Niemann, et al., 2005), an approximation is given as $R_q \approx 1.25 \cdot R_a$.

According to (Baalmann, 1994), an estimate for a usual composite roughness is given by $R_q = 1.2 \cdot 10^{-5} \cdot D_{pw}^{0.55}$. This formula according Baalmann is used when the sizing button next to the input field is clicked.

3.1.5.4 Material for races, rolling element, shaft and housing

The material of the races, the rolling element, shaft and housing can be selected to be steel or Si3N4. The list is taken from the database therefore it can be extended by the user. Also 'own input' is available and the material data can be defined using the -button.

Property	Symbol	Value	Unit	Checkbox
Young's modulus inner ring	E_j	207000	MPa	<input type="checkbox"/>
Poisson number inner ring	ν_j	0.3		<input type="checkbox"/>
Density inner ring	ρ_j	7850	kg/m ³	<input type="checkbox"/>
Thermal elongation coeff. inner ring	αT_j	11.5	10 ⁻⁶ /K	<input type="checkbox"/>
Thermal conductivity inner ring	λ_j	42.6	W/m·K	<input type="checkbox"/>
Static permissible stress inner ring (ball bearing)	pCO_{Bi}	0	MPa	<input type="checkbox"/>
Static permissible stress inner ring (roller bearing)	pCO_{Ri}	0	MPa	<input type="checkbox"/>
Fatigue strength inner ring (ball bearing)	pCu_{Bi}	0	MPa	<input type="checkbox"/>
Fatigue strength inner ring (roller bearing)	pCu_{Ri}	0	MPa	<input type="checkbox"/>
Dynamic load capacity modification factor inner ring	bm^*_j	0		<input type="checkbox"/>

For material data, the Young's modulus, the Poisson number, the density, the thermal elongation coefficient and the thermal conductivity can be defined. This data is used to calculate the operating clearance and the load distribution within the bearing. The thermal conductivity is used to calculate the thermal resistance for ball bearings according (Nakajima, 1995).

Optionally, a static permissible stress can be defined which is used for the calculation of static load capacity and static safety factor. The default value is 4000MPa for roller bearings and 4200MPa for ball bearings. For spherical ball bearings, the permissible stress is multiplied by 4600/4200 for the outer ring internally by the software.

The fatigue strength is used for the calculation of the fatigue load limit C_u . The default value is 1500MPa, it can be overwritten by the user.

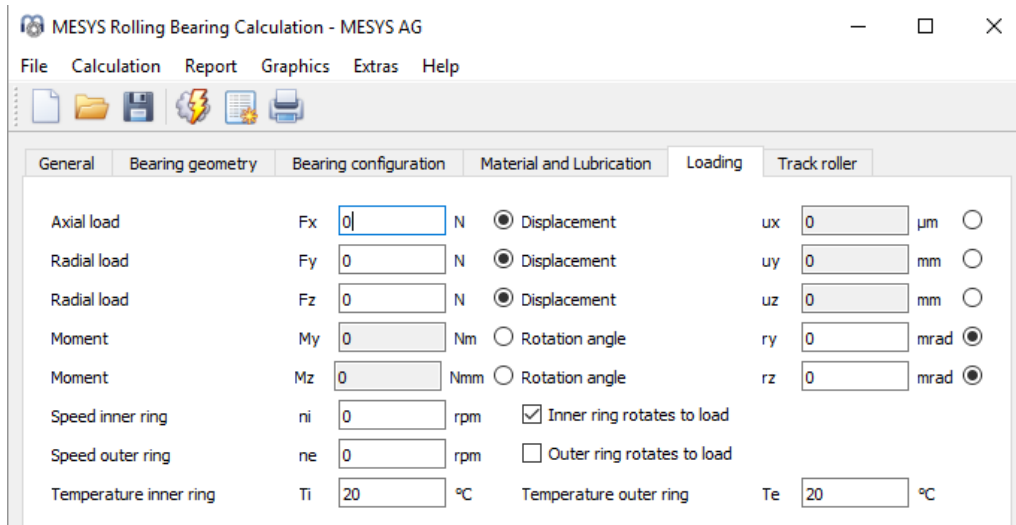
The dynamic load capacity can be modified dependent on the material selection, too. The bm^* -factor is multiplied to the dynamic load capacity according ISO 281. This allows to consider a material influence on dynamic load capacity. A different value can be used for inner and outer race. The value of bm^* -factor for the rolling element is only used for the calculation of rolling element set life, it does not affect the dynamic load capacity of the bearing.

3.1.6 Loading

For each coordinate direction, you can select if a force or a displacement should be entered. For example, if a calculation of a tapered roller bearing should be done considering only radial load, the displacement in axial direction can be fixed to zero and the reaction force in axial direction will be calculated.

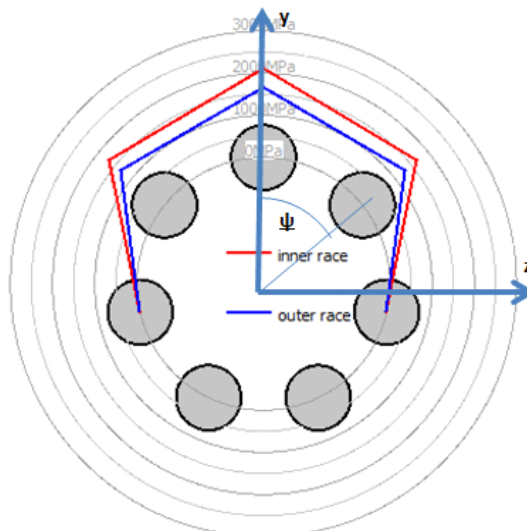
A moment load or a rotation (misalignment) can be entered for two directions only, because the rotation around the bearing axis is not constrained.

Use the radio buttons to select the elements you want to enter.



3.1.6.1 Coordinate system

The coordinate system is defined with x as the axial direction. The y-axis points up to the first rolling element and the angle ψ is positive around the x-axis or clockwise in the diagram on the right (looking in the direction of the x-axis). The angle starts with zero at the first rolling element on the y-axis. Moments are positive if acting around the corresponding axis.



The load is acting on the inner ring (different for track rollers with elastic outer ring), so a positive load in y-direction leads to a loading of the rolling elements on the top as shown in the graphics on the right.

3.1.6.2 Inner ring rotates to load

The resulting life is slightly different if the load distribution on a race is stationary or rotating. For many bearings, the inner ring rotates relative to the load and the

outer ring is stationary to the load. This is the case if the inner ring is connected to a rotating shaft and the load is stationary in space.

What is the correct input if only an axial load occurs? In this case, the input doesn't matter since the results are the same for both settings if all the rolling elements have the same loading.

3.1.6.3 Temperature of inner ring

The temperatures of the inner and outer ring are used for the calculation of the interference fit between shaft and inner ring or between housing and outer ring only. They affect the operating clearance. Optionally, the temperatures of the shaft and housing can be entered, too, to consider a temperature gradient, see [Consider temperature gradient in fits](#)¹²⁹.

The temperature of the rolling element is assumed to be the maximum value of the ring temperatures as default. This can be changed under settings and a manual input of the rolling element temperature is available optionally, too.




3.1.7 Load spectrum



If at the tab-page 'General' the flag for the calculation with load spectrum is set, instead of for a single load case, the calculation will be done for a load spectrum. The load or displacement, the rotation speed and the temperatures can be specified for each element.

The screenshot shows the 'Loading' tab in the MESYS software. It features a table with columns for Frequency, Fx [N], Fy [N], Fz [N], ry [°], rz [°], ni [rpm], ne [rpm], T_i [°C], T_e [°C], and TOil [°C]. The table contains three rows of data. A context menu is open over the table, listing options: enter Fx, enter Fy, enter Fz, enter My, and enter Mz. At the bottom of the interface, there are checkboxes for 'inner Ring rotates to load' (checked) and 'Outer ring rotates to load' (unchecked), along with a 'Results for No' dropdown set to '1' and several navigation icons.

	Frequency	Fx [N]	Fy [N]	Fz [N]	ry [°]	rz [°]	ni [rpm]	ne [rpm]	T_i [°C]	T_e [°C]	TOil [°C]
1	0,5	15000	10000	0	0	0	1500	0	60	50	70
2	0,3	10000	8500	0	0,5	0	1000	0	65	60	70
3	0,2	12500	500	7000	0	0	350	0	60	50	70

Using the context menu that opens clicking the right mouse button, you can select which forces or moments should be used as input values. If a value is not selected, the corresponding displacement/rotation can be entered.

Rows can be added using the -button or a selected row can be deleted using the -button. The -button clears all the inputs.

Using the -button, the load spectrum is read from a file. The columns that are currently shown with its units have to be used for the input file. As the data from the file is added to the inputs, you must clear the inputs first to define a full load spectrum. Likewise, a created load spectrum table can be exported with the corresponding -button.

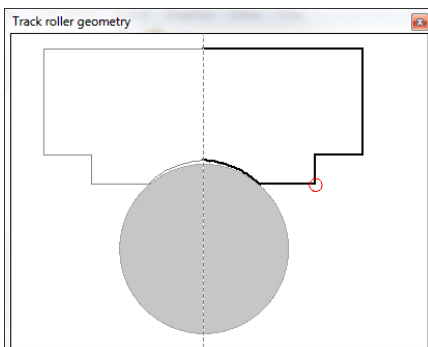
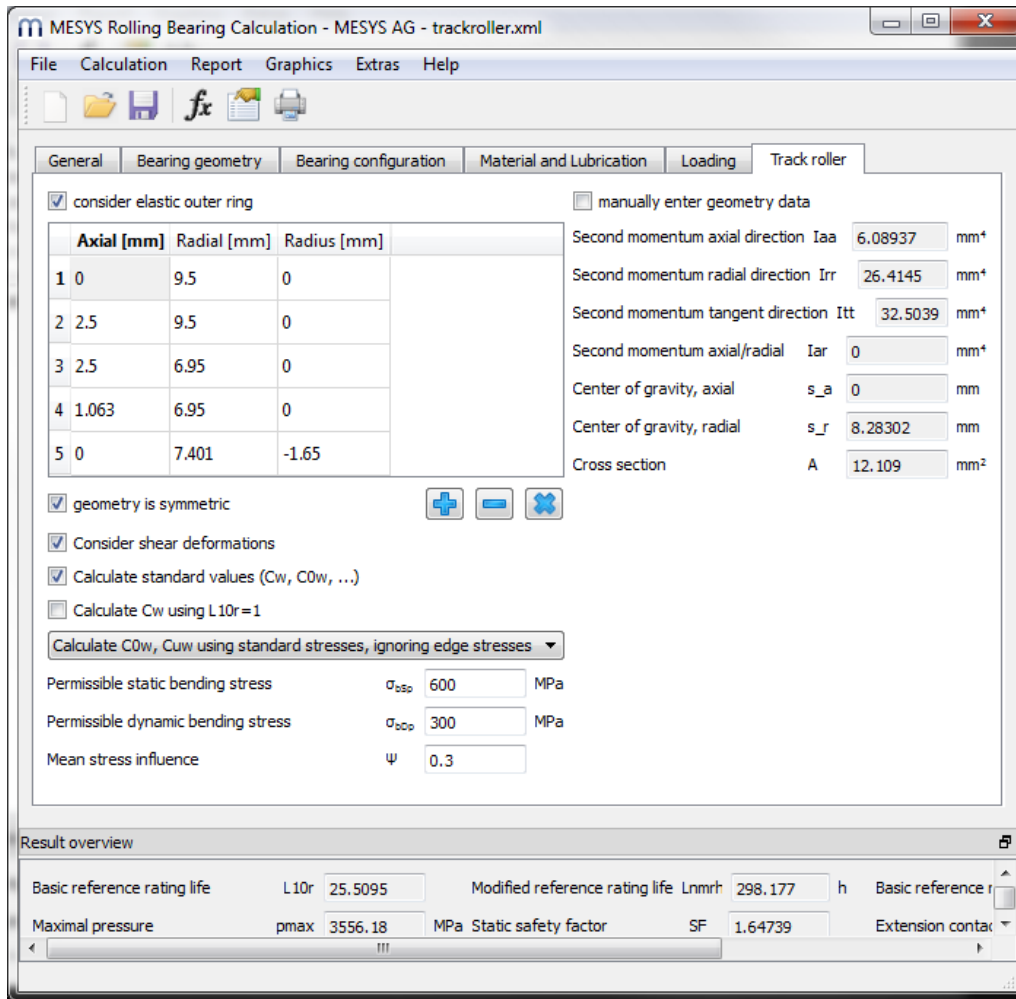
As for the single load case, a selection is available if the ring is rotating relative to the load or not. A ring not rotating to the load would be on the safe side.

The load distribution is calculated for each load case. The resulting life for the whole load spectrum is shown as result and the minimum static safety of all elements. The result graphics are only shown for the element of the load spectrum that is chosen as results on this page.

3.1.8 Track rollers

Elastic deformations of the outer ring can be optionally enabled. This is an extension to the base bearing software.

The outer ring of the bearing is modeled by a Timoshenko beam element. So bending, axial deformations and shear deformations are considered. The shear correction factor is fixed to 1. The calculation should only be used if the approximation as beam seems to be appropriate. The beam does not consider deformations within the cross section for example in axial direction for multi row bearings.



3.1.8.1 Supported bearing types

The calculation with elastic outer ring is only available for several bearing types. The calculation is possible for:

- Deep groove ball bearing
- Radial angular contact bearing
- Double row radial angular contact bearing
- Radial four-point ball bearing

- Radial cylindrical roller bearing
- Double row cylindrical roller bearings
- Double row tapered roller bearings

Multi-row bearings can be considered using the “bearing configuration”.

3.1.8.2 Track roller geometry

The track roller geometry can be defined using several points. The contour starts at axial position zero on the outside of the bearing and is then defined in clockwise direction.

A radius can be defined in the third column. An arc from the last point to the new point will be generated. For a convex contour, use a positive value, for a concave contour, use a negative value. A clockwise arc has a positive radius, a counterclockwise arc a negative radius.

Either a full geometry is defined back to the start point or only half the geometry is defined for a symmetric outer ring. In the graphic for the symmetric track roller geometry, the mirrored part is shown in grey.

3.1.8.3 Symmetric geometry

A symmetric geometry starts at axial position zero on the outside diameter and ends on axial position zero on the inner diameter.

An unsymmetrical geometry should end with its start point.

3.1.8.4 Consider shear deformations

As default shear deformations in the elastic outer ring should be considered. Since other software and old literature, see (Paland, 1968), is only considering deformations because of bending, the influence of shear deformations can be ignored by clearing this setting. The effective load capacities will be higher because the deformation of the outer ring is reduced.

3.1.8.5 Calculate standard values

Some standard values for track rollers can be calculated automatically using a pure radial load at axial position zero.

Because of elastic deformations of the outer ring, track rollers show a different load distribution than rigid bearings. This leads to a decrease of life which is taken into account by giving reduced effective load capacities in bearing catalogs.

The effective dynamic load capacity C_w is given in bearing catalogs for track rollers. There are two options to calculate this value. See option 'Calculate C_w using $L_{10r} = 1$ ' below. Also for effective static load capacity, there are two options. See section 'Calculation options for C_{0w} and C_{uw} ' below.

The load capacities are calculated using a radial load on the angle of zero, so it is acting directly on a rolling element.

The static and dynamic permissible radial loads are the loads where the permissible bending stress is reached. See below. The permissible static and dynamic loads are calculated using a radial load which is acting between two rolling elements. This is the critical case.

If a bearing configuration is used, effective load capacities and permissible radial loads are calculated for the whole bearing.

3.1.8.6 Calculate C_w using $L_{10r} = 1$

As the dynamic load capacity is defined in (ISO 281, 2007), a radial bearing loaded with $F_r = C$ should have a life $L_{10} = 1$ (One million load cycles). This definition is used for the definition of the effective dynamic load capacity if the option 'Calculate C_w using $L_{10r} = 1$ ' is set. The software iterates for a radial load which leads to $L_{10r} = 1$ and this is shown as effective dynamic load capacity C_w .

For bearings with a small number of rolling elements the life $L_{10r} = 1$ is not reached for the rigid bearing loaded with $F_r = C$. The reason is that (ISO 281, 2007) assumes a load zone of 180° which cannot be reached if only six rolling elements are used for example. Therefore, a different definition for C_w is available if the option is not set. In this case, C_w is defined to the radial load which leads to the same life as a rigid bearing with $F_r = C$.

This option will only have an influence for a small number of rolling elements. If the option is not set, C_w will lead to the same life for the elastic bearing as C for the rigid bearing.

3.1.8.7 Calculation options for C_{0w} and C_{uw}

Several options for the calculation of effective static load capacity are available:

Calculate C_{0w} , C_{uw} using standard stresses

If the option 'Calculate C_{0w} , C_{uw} using standard stresses' is set, the effective static load capacity C_{0w} is defined as the radial load which leads to $p_{max} = 4200$

MPa (or 4000 MPa for roller bearings). The effective fatigue limit is defined similar, but with a stress of 1500 MPa in this case.

Calculate C_{0w} , C_{uw} using standard stresses, ignoring edge stress

For roller bearings edge stresses can occur on high loading leading to different results than usually found in bearing catalogues. Therefore, a second option exists for the calculation of effective static load capacity C_{0w} .

If this option is selected, the stress in the center of the roller is compared with the permissible stress as above. So edge stresses are not taken into account.

For ball bearings, this option does not affect the results, it only makes a difference for roller bearings.

Calculate C_{0w} , C_{uw} using roller load

If the option is set, C_{0w} is defined as the radial load which leads to the same maximum roller load than a rigid bearing loaded with $F_r = C_0$. Also for the calculation of C_{uw} , the comparison of roller loading instead of stress is made.

If this option is selected, the resulting C_{0w} has the same meaning than C_0 for the rigid bearing. It can be useful if own input is used for the static load capacity.

3.1.8.8 Permissible static bending stress

The permissible static bending stress is used to calculate the permissible static radial load and a safety factor for the bending stresses in the outer ring. The stress is calculated as maximal value of bending stress plus tension in the outer ring.

$$\sigma_{max} = \sigma_{b,max} + F_t/A$$

If not defined manually, the software automatically selects the point of input geometry which leads to the highest stress. Only points in the geometry list are considered, no intermediate values. This could make a difference for convex radii.

3.1.8.9 Permissible dynamic bending stress

The permissible dynamic bending stress is used to calculate the permissible dynamic radial load and a safety factor for the bending stress amplitude. The stress is considered at the point where the maximal stress occurs.

$$\sigma_{max} = \sigma_{b,max} + F_t/A$$

$$\sigma_{min} = \sigma_{b,min} + F_t/A$$

$$\sigma_m = (\sigma_{max} + \sigma_{min})/2$$

$$\sigma_a = (\sigma_{max} - \sigma_{min})/2$$

The input value is a permissible alternating stress with is reduced by the mean stress influence coefficient ψ . The safety factor is calculated as

$$S_{dyn} = \frac{\sigma_{bDperm} - \Psi \cdot \sigma_m}{\sigma_a}$$

The permissible dynamic radial load is calculated so that a safety factor of 1 is reached.

3.1.8.10 Mean stress influence

The mean stress influence is considering the mean stress for calculating permissible amplitude of bending stress.

The mean stress influence coefficient can be calculated as

$$\Psi = \sigma_{bDperm} / (2 \cdot R_m - \sigma_{bDperm})$$
 using the definition according DIN 743.

3.1.8.11 Axial position for the load

If an asymmetric geometry is defined, an axial position for the load can be defined. It is used for the calculation of standard values like C_w , C_{0w} ...

For symmetric geometry, the load is applied at position zero.

3.1.8.12 Manually enter geometry data

It is not mandatory to enter the cross section using point data. Also cross section data can be introduced directly after calculating them by a CAD system for example.

If the cross section data is entered manually, also the critical point for calculating bending stresses has to be defined manually. Unfortunately, this point is load-dependent. If you just have a radial load on a symmetric cross section, this is not a problem, but you have to be careful in other cases.

3.1.8.13 Second momentums

The second momentums are given on polar coordinates for the cross section. They are given relative for the center of gravity, not for the origin of the coordinate system.

The second momentum in axial direction I_{aa} and in radial direction I_{rr} are usually available. For the tangent direction, the torsional moment should be given, but since this is not usually known the polar moment $I_{tt} = I_{aa} + I_{rr}$ can be used as

replacement. This is only exact for a circular cross section, but the torsion load is not the main loading to the outer ring, so it could be acceptable.

The deviatory momentum is defined as $I_{ar} = + \iint a \cdot r \, dA$. Depending on the literature, also definitions with a negative sign are usual. The deviatory momentum is zero for symmetric cross sections.

3.1.8.14 Center of gravity

The center of gravity is defined using its axial and radial coordinate. For symmetric cross sections, the axial position is zero.

3.1.8.15 Cross section

The cross section of the outer ring has to be entered here. Since the calculation is considering tension and shear deformations, it has an influence on the load distribution.

3.1.8.16 Critical point

The critical point for the calculation of bending stress has to be entered if the geometry data is entered manually. The bending stress is calculated using this point.

Be sure to enter the correct value for the given load conditions. The critical point is marked with a red circle in the graphics of track roller geometry.

3.1.8.17 Track roller loading

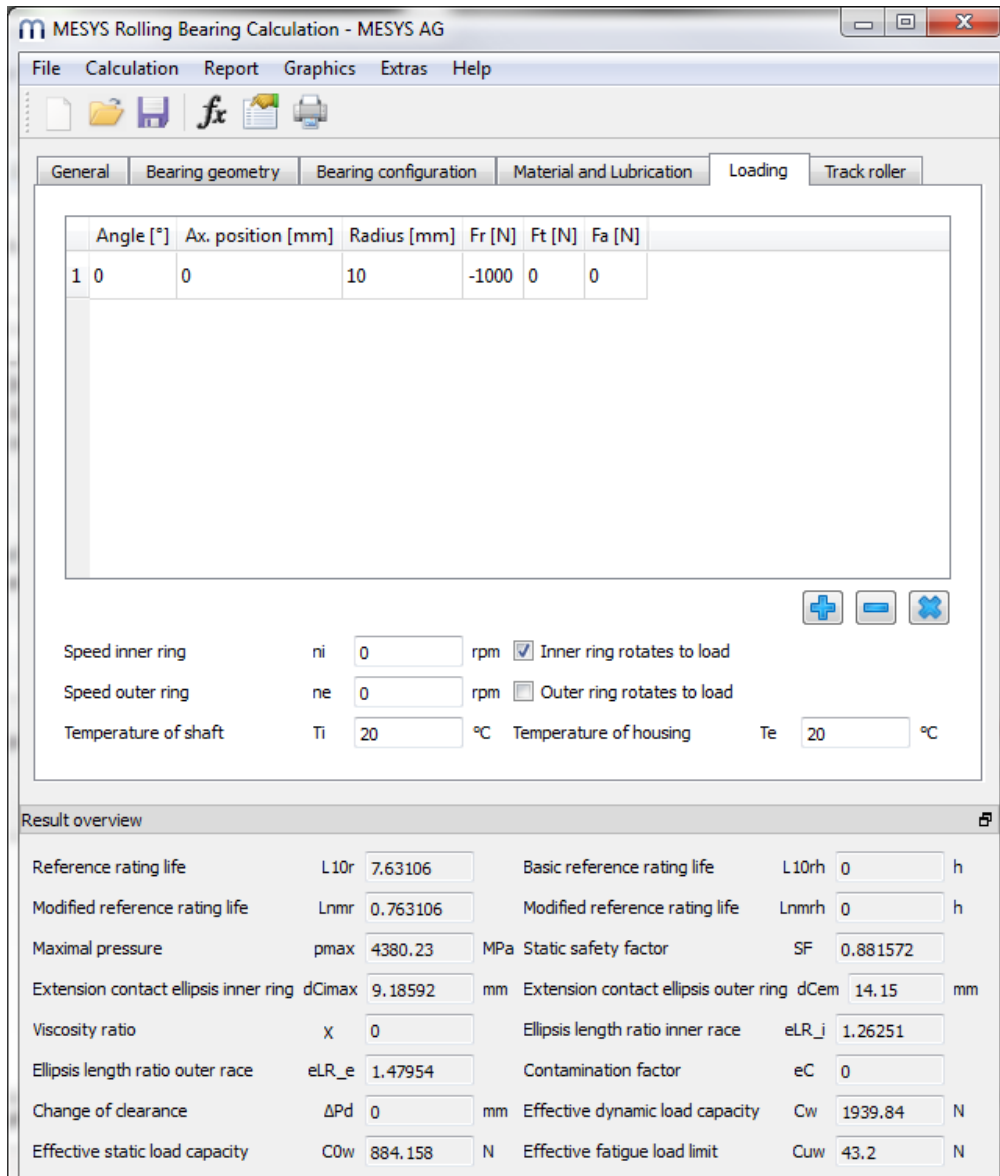
In the case of elastic outer ring, the input of loading is changed. Instead of the load onto the inner ring, the load on the outer ring can be defined on several points. It is not possible to define displacements or rotations directly.

An angle, axial and radial positions define the point where the load is acting. The load can be defined in axial, radial and tangential direction. Since the overall torque has to be zero, an input of tangential load can only be used if several loads are defined.

The coordinate system is a standard cylindrical coordinate system with r-direction pointing to the outside. Therefore, Fr should usually be entered as negative value for pressure.

The axial position can be used if you assume that the radial load is not centric. An eccentric load leads to torsion of the outer ring.

The first rolling element on the y-axis is at an angle of zero. The results will be different if the load is applied on the position of a rolling element or between them. Both cases should be considered in doubt.



3.1.8.17.1 Load spectra for track rollers

If load spectra are used for track rollers, as a first step, the location of forces has to be defined in the loading page without activating load spectra. If the load spectra are activated afterwards, a column for each force is available as input.

	Frequency	ni [rpm]	ne [rpm]	T_i [°C]	T_e [°C]	T_r [°C]	TOil [°C]	Fr_1 [N]	Ft_1 [N]	Fa_1 [N]
1	0.6	0	500	20	20	20	70	-1000	0	0
2	0.4	0	200	20	20	20	70	-500	0	0

inner Ring rotates to load
 Outer ring rotates to load
 Results for No: 1

3.2 Thermal permissible speed

The thermal reference speed according (ISO 15312, 2003) and the thermal permissible speed according (DIN 732, 2010) are calculated by the software. The calculation can be run separately under menu 'Calculation' -> 'Thermal permissible speed'.

Thermal permissible speed

Thermal reference speed

Factor for load independent losses f_{0r}

Factor for load dependent losses f_{1r}

Surface for heat transfer A_r mm²

Thermal transmission coefficient k_q W/m²·K

Thermal permissible speed

Method

Factor for load independent losses f_0

Factor for load dependent losses f_1

Assume fixed temperature for lubricant

Temperature difference between bearing and surroundings $\Delta\theta$ °C

Temperature difference of oil flow $\Delta\theta_{Oil}$ °C

Oil volume flow V_{Oil} l/min

Thermal reference speed n_{tr} rpm

Thermal permissible speed n_t rpm

Most of the parameters can be overwritten by the user. Note that the calculation is just according to the standard. The equivalent load is calculated by axial and radial load, moment load is not considered. Also clearance is not considered.

For the thermal permissible speed in addition to DIN 732, the friction forces can be derived from ISO 14179-2. The ISO standard provides different factors for grease and oil lubrication. As default the lubricant temperature as defined in the main window is used. Clearing the option "Assume fixed temperature for lubricant" uses the bearing temperature for the oil viscosity. The bearing temperature is the temperature of bearing surroundings plus temperature increase for the bearing.

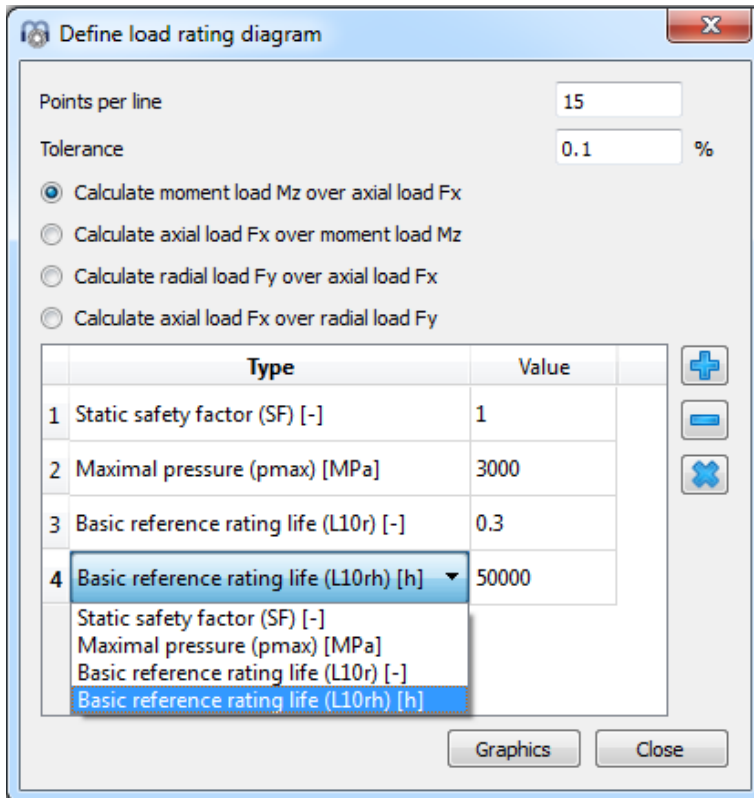
If a bearing geometry is not considered in the standard, the input values for f_{0r} , f_{1r} and f_1 can be entered by the user and the calculation can be run.

The results for thermal reference speed and for thermal permissible speed are also shown in the main report.

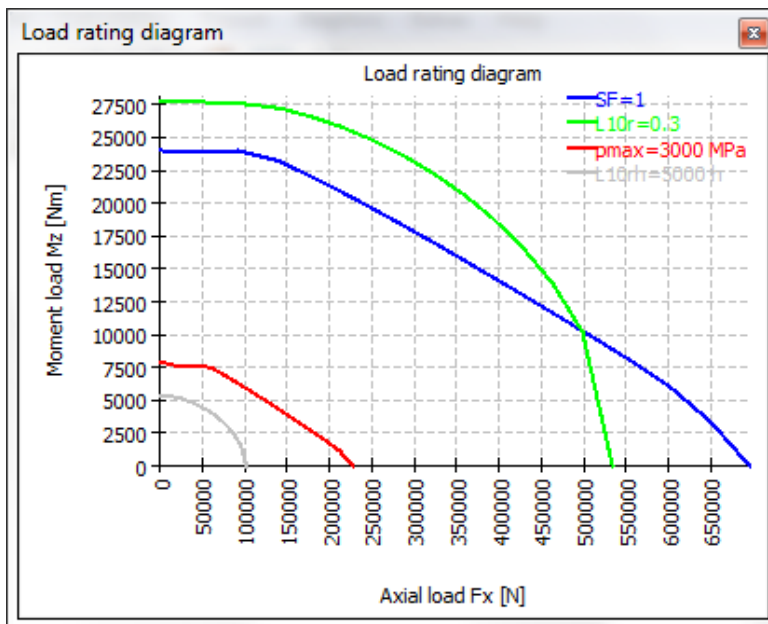
3.3 Load rating diagram

A load rating diagram can be configured at menu 'Calculation' -> 'Load rating diagram'. It allow us to show four possible line charts, where the variables of the

XY axes are compared against each other (M_z - F_x , F_x - M_z , F_y - F_x , F_x - F_y) while maintaining constant four types of values predefined by the user (see the picture). These diagrams are often used for slewing rings.



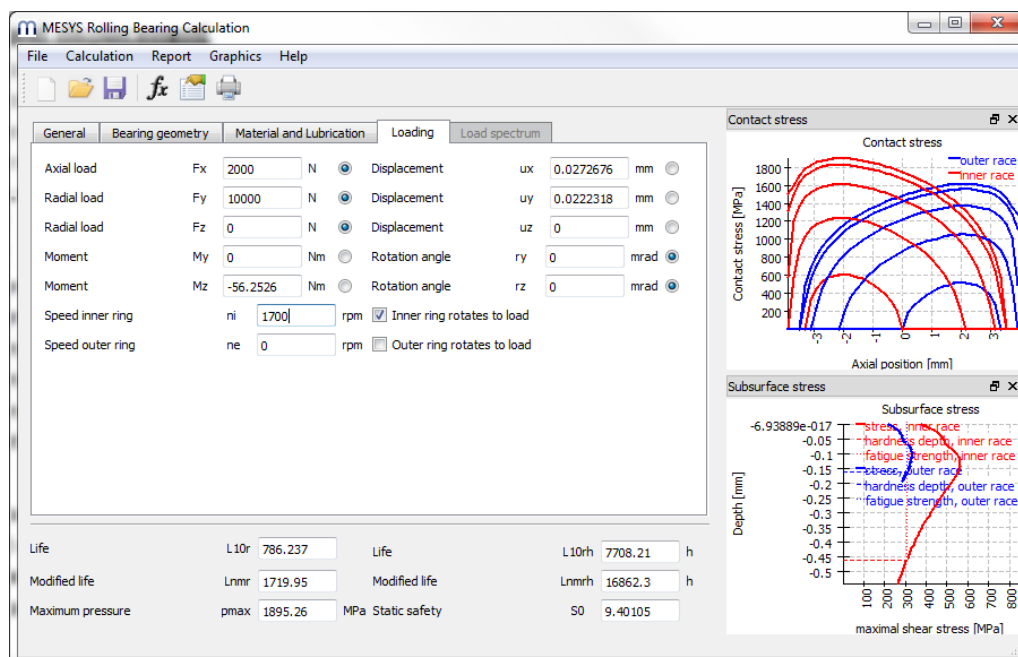
In the following picture it is shown a graphic example of the load rating diagram by which tilting moment (M_z) is compared against axial load (F_x) for a safety factor of 1, maximal pressure of 3000 MPa, 300.000 cycles and 50000 h.



For each axial load, the tilting moment is increased until the selected criteria are reached. The number of points used for one line and a tolerance for the calculation can be defined.

3.4 Results

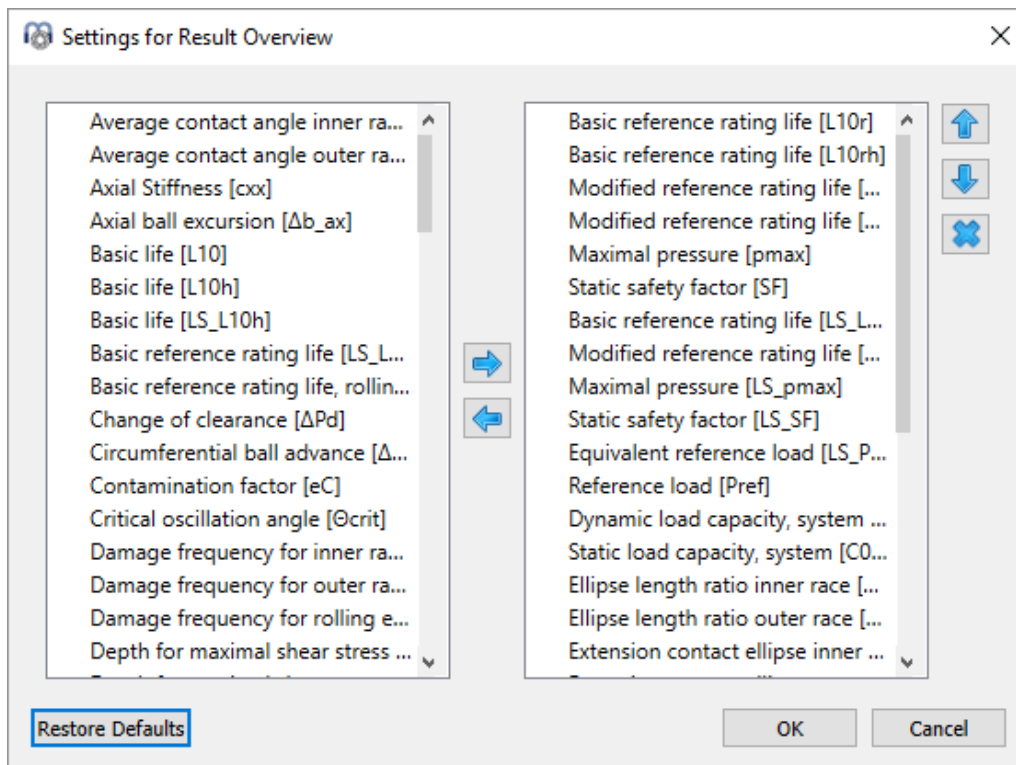
Results are provided as a small result overview directly in the software, a main PDF text report, a tolerance report and separate graphic windows. Graphic windows can be docked onto the main program interface and are automatically updated on each calculation.




3.4.1 Result Overview

The result overview shows the reference life L_{10r} in 10^6 rotations and the reference life L_{10rh} in hours. The modified reference life L_{nmr} considering reliability and lubrication is also shown in 10^6 rotations and in hours.

The maximum pressure and the static safety provide information about the maximum loading.

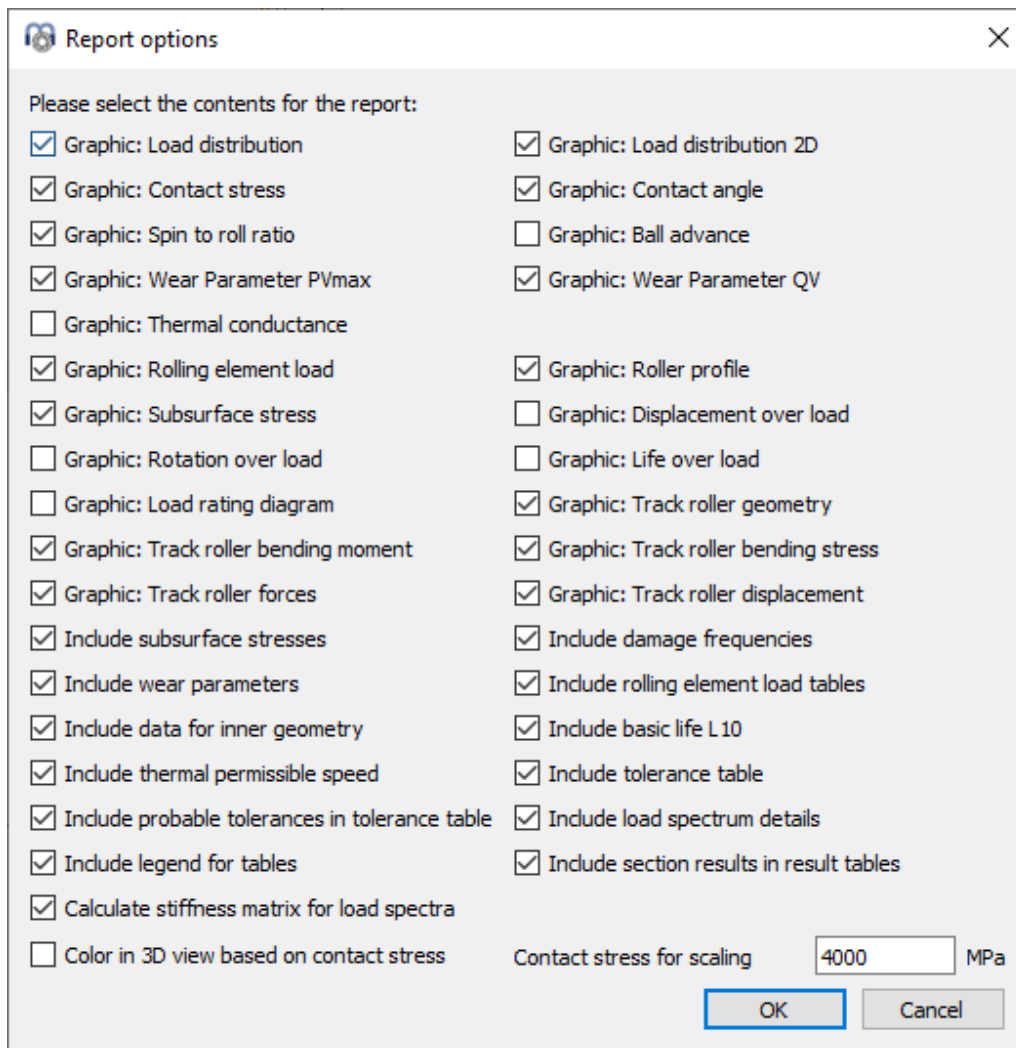


The result overview can be customized using “Extras->Result overview”. The items shown in the overview can be selected and additional items can be added using the -button or the context menu with the right mouse button. Note that only values are actually shown in the result overview that are valid. So you will see fewer values than selected in the settings.

The settings for the result overview are saved in the windows user profile, not in the calculation file. So they will be the same for different calculations.

3.4.2 Main Result Report

The text report is provided in PDF format. An example report is available as separate document. The results are provided in sections ‘Bearing inner geometry’, ‘Forces and displacement’, ‘Life’, ‘Subsurface stresses’ and ‘Damage frequencies’.



In the menu "Report->Options", the contents of the report can be configured. The graphics to be included can be selected and some sections of the report could be discarded if not of interest. The legend for all the table parameters can be shown in the report.

3.4.2.1 Saving main report

The main report can be saved using the menu "Report->Save Report As" in different formats. It can be saved as PDF, DOC, DOCX or ODT (Open Office) format.

3.4.2.2 Customizing report logo

As default, a MESYS logo is shown on top of the report. This can be customized by defining a logo in the "mesys.ini" file. The logo should be created using a size, so that it fits into the header table. The DPI setting in the logo is considered, so that a high-resolution image could also be used. Additionally, it also exists the possibility to create an own template of the supported file formats (DOC, DOCX and ODT) and generate the report from it, thus having the chance to customize the logo.

3.4.3 Tolerances Report

In the menu "Report->Tolerance" report a special report for tolerances can be generated. While the tolerances for the selected calculation case are included in the main report, the additional report shows tolerances and pressure for maximum, medium and minimum clearance. If selected under options, the minimum probable and maximum probable clearances are included too.

3.4.4 Custom Report Templates

Custom report templates can be added in subdirectory "templates/rbc". Reports based on these templates can be generated by selecting the report template under "Report->Report templates".

The report templates are word files with DOCVARIABLE fields. These fields are filled on generation of the report.

Possible fields are:

- DOCVARIABLE VAR name [unit]: the variable with the name 'name' is printed. Optionally, a unit can be provided in brackets, for example [mm]. Additionally, a format could be added like %6.3g or %6.3f, where 'f' always generates a fixed point output and 'g' generates an exponential output for large numbers.
- DOCVARIABLE VARU name [unit]: the variable with the name 'name' plus the unit are printed. Optionally a unit can be provided in brackets, for example [mm].
- DOCVARIABLE TABLE tablename: A table is inserted.
- DOCVARIABLE GRAPHIC graphicName width=130 height=70: A graphic is inserted. Optionally, height and width can be given in millimeters.

Regarding details for variable names, table names and graphic names, please ask support@mesys.ch for additional documentation.

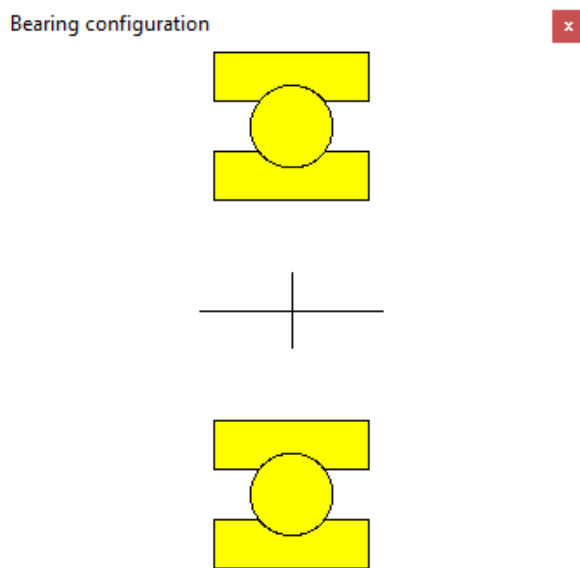
3.4.5 Result Graphics

Several results can be shown as graphics. All graphics are also appended to the report, so that a full documentation is provided in one document.

Using the context menu (right mouse button) in a graphics window, the graphic can be exported as file or be printed. For diagrams, "Diagram options" allow to adjust units and range of axes.

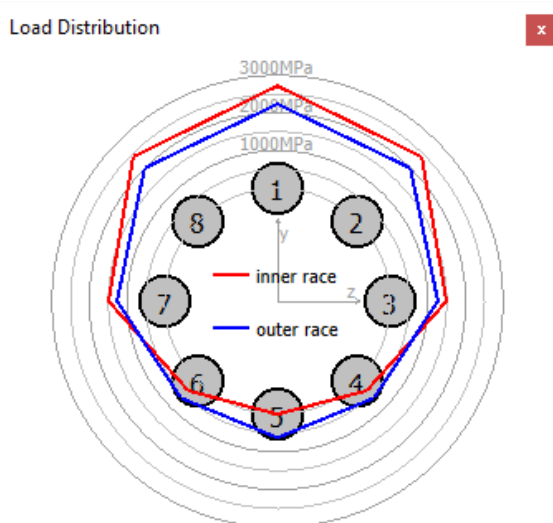
3.4.5.1 Bearing configuration

The bearing configuration shows the bearing rows and their position. It is mainly used together with the bearing configuration. The bearing inner and outer diameter have to be defined for generating this image.



3.4.5.2 Load distribution

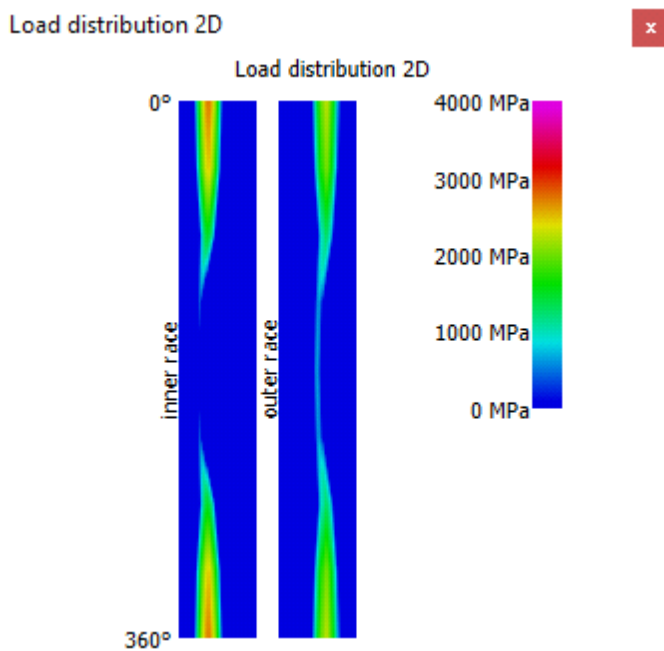
Here the contact stress of an angular contact bearing with radial and axial load and relatively high speed is shown. The contact stress is larger on the inner race (red) in the load zone but slightly larger on the outer race on the opposite side because of centrifugal loads.



3.4.5.3 Load distribution 2D

The 2D load distribution shows the pressure distribution on inner and outer race as color strip. The maximum stress for the scale can be defined under report options. The graphic is available for a single load case and for the whole load spectrum.

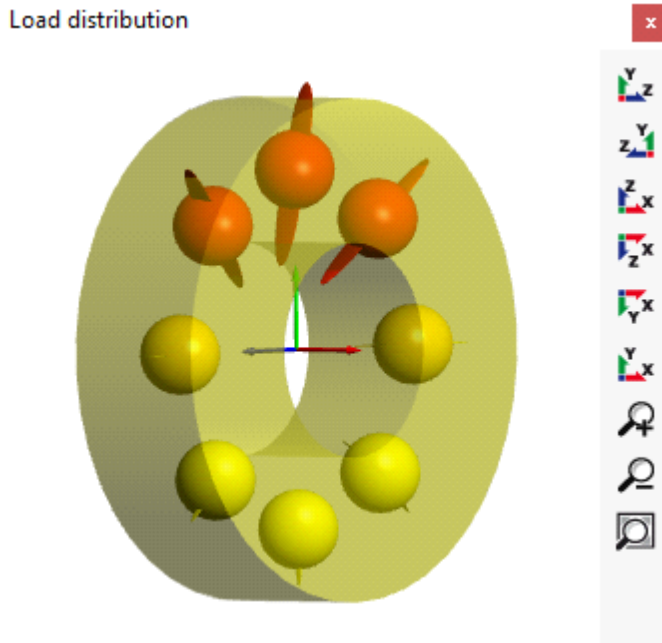
This diagram also gives an impression of contact angle variation.



3.4.5.4 Load distribution 3D

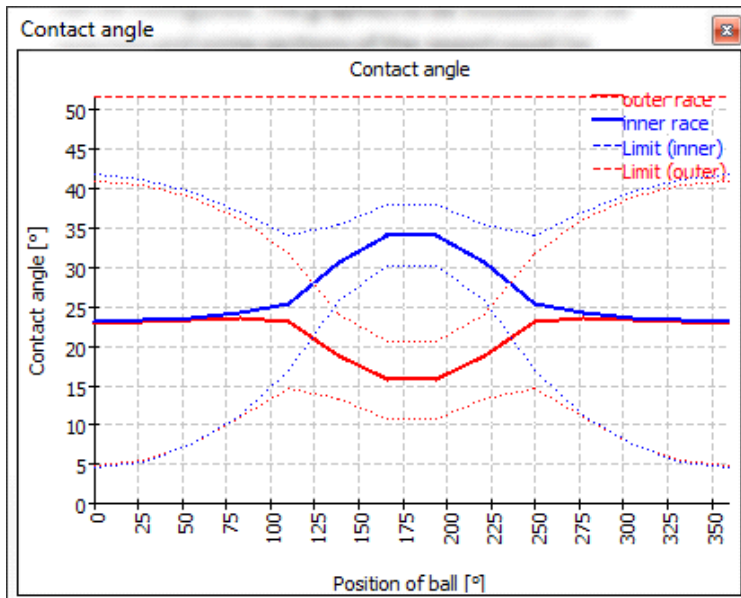
This diagram shows the load distribution on a 3D visualization of the bearing. The stress level is indicated by the color of the rolling element and the contact ellipse. The visualization can be accurately rendered in any perspective.

The 3D views can be rotated with the left mouse button. Panning is activated by pressing "Shift", zooming by pressing "Ctrl".



3.4.5.5 Contact angle

The contact angle is shown for an angular contact bearing with nominal contact angle of 20°. In the load zone the contact angles of inner and outer race are almost equal, but there is a big difference opposite to the load zone because of the centrifugal load.

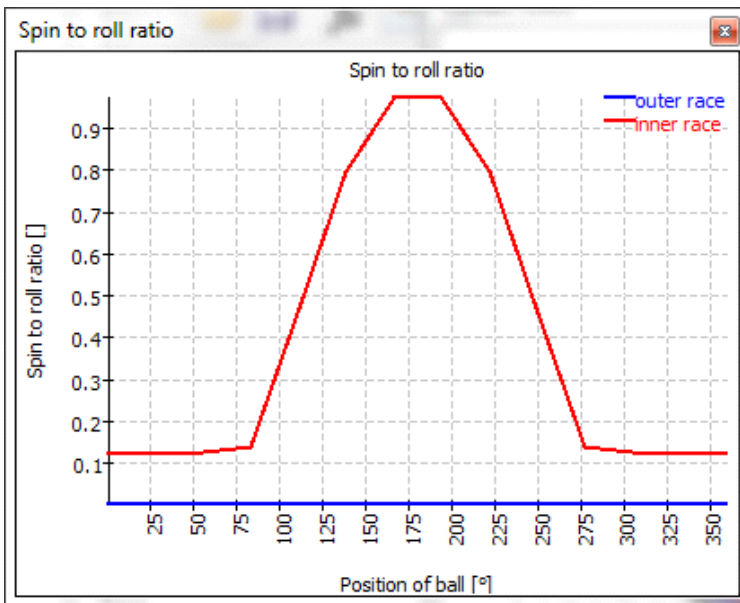


The extension of the contact ellipsis is shown with the dotted lines, so we have a wide contact ellipsis at 0° and a smaller one at 180°. The contact angle corresponding to the shoulder is shown as dashed straight line at the top, so the shoulder is high enough in this case.

3.4.5.6 Spin to roll ratio

The spin to roll ratio can be shown also. The calculation assumes 'outer race control' if either relative speed or friction coefficient is zero so there is no spinning at the outer race and all the spin occurs at inner race.

If speed and friction coefficient are not zero, spinning and rolling speed are calculated considering Coulomb friction.



3.4.5.7 Circumferential ball advance

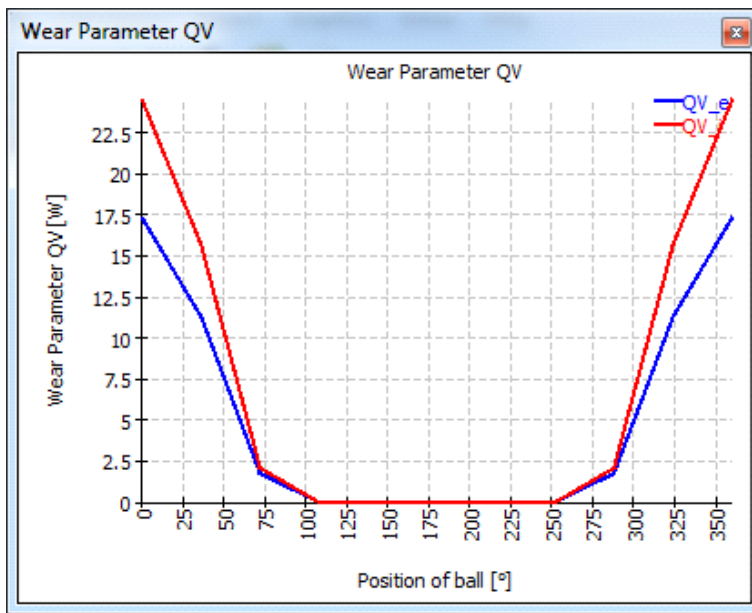
The circumferential ball advance is shown as graphics. It is calculated based on the assumption that the orbit speed of each ball is constant in one section. Because of variation of contact angles the ball orbit speed varies between the rolling element. This leads to variations in the distance between the rolling elements. The ball advance should be smaller than the clearance in the cage pocket in case of high speed.

3.4.5.8 Wear parameter QV

For ball bearings spinning and rolling speed are calculated in the contact ellipsis.

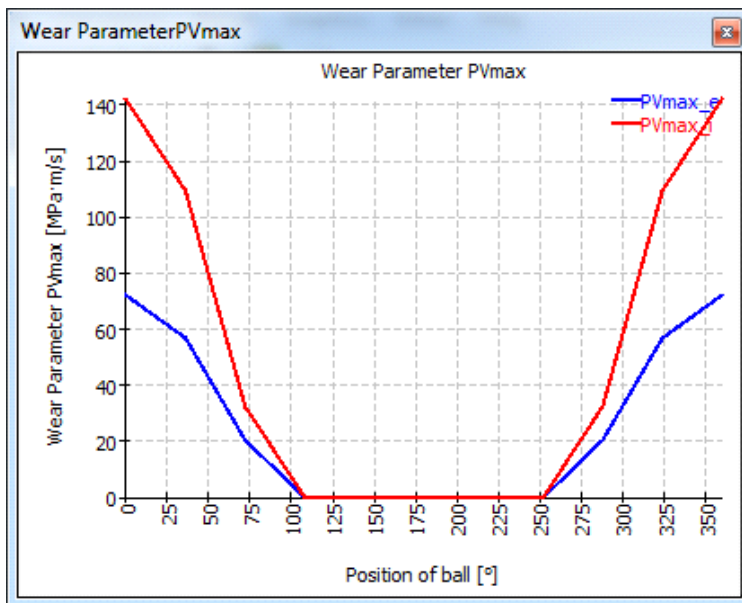
The wear parameter QV is calculated for each contact zone as

$$QV = \int p \cdot v \, dA$$



3.4.5.9 Wear Parameter PVmax

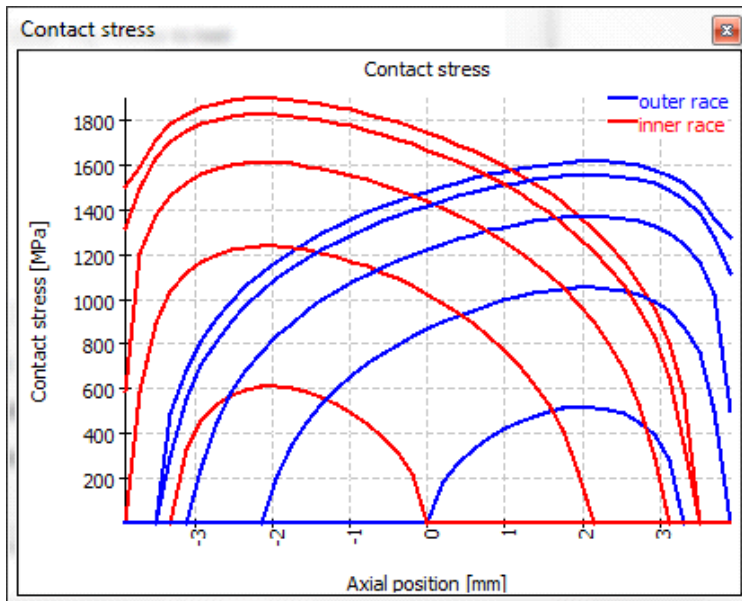
The wear parameter PVmax is the maximum value of the product pressure times sliding velocity in the contact ellipsis.



3.4.5.10 Contact stress for roller bearings

The contact stress for roller bearings is shown over the axial position or the length of the roller. This example is for a cylindrical roller bearing under radial and axial load and without misalignment between inner and outer race.

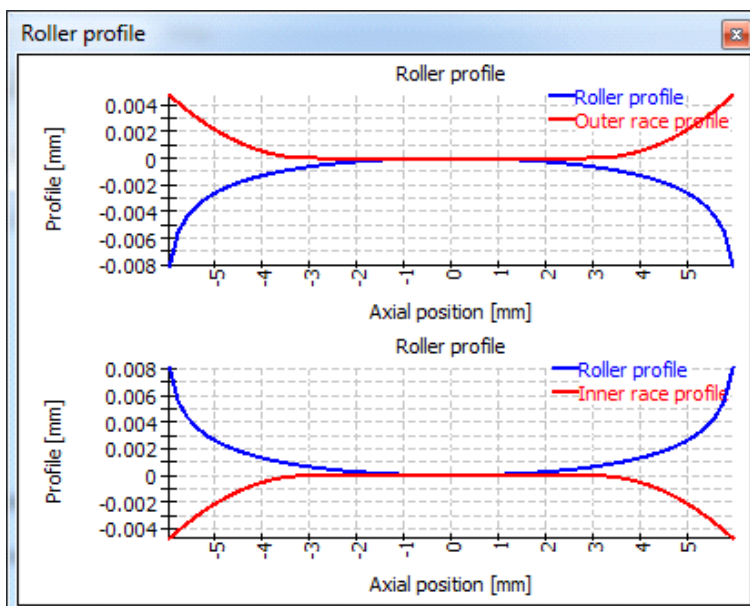
Because of the tilting of rollers under axial load the radial load is not distributed equally over the length of the roller.



For axial cylindrical roller bearings and axial spherical roller bearings the roller orientations is inside to outside, while it is left to right for the other bearing types.

3.4.5.11 Roller profile

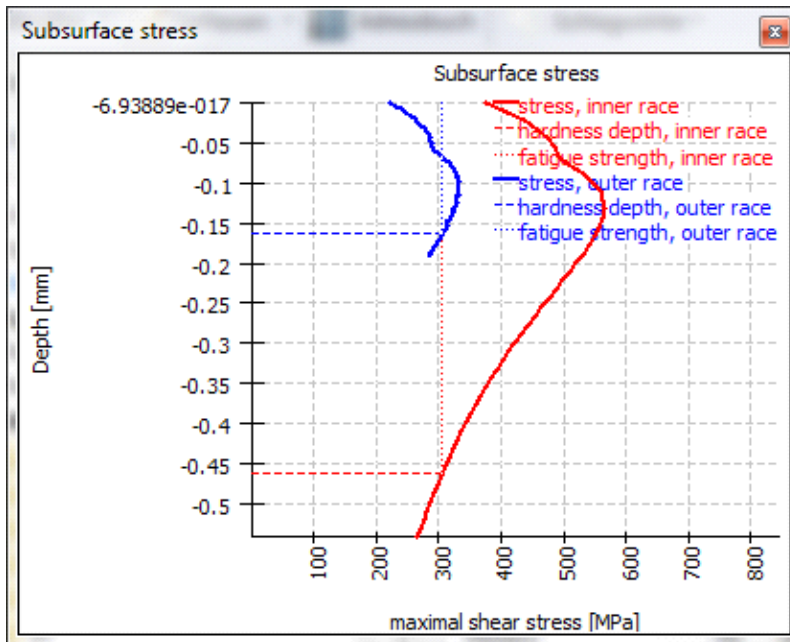
The roller profile and the profile of the races can be shown graphically. The profile can only be user-defined if the extended method for pressure distribution is activated.



3.4.5.12 Subsurface stress

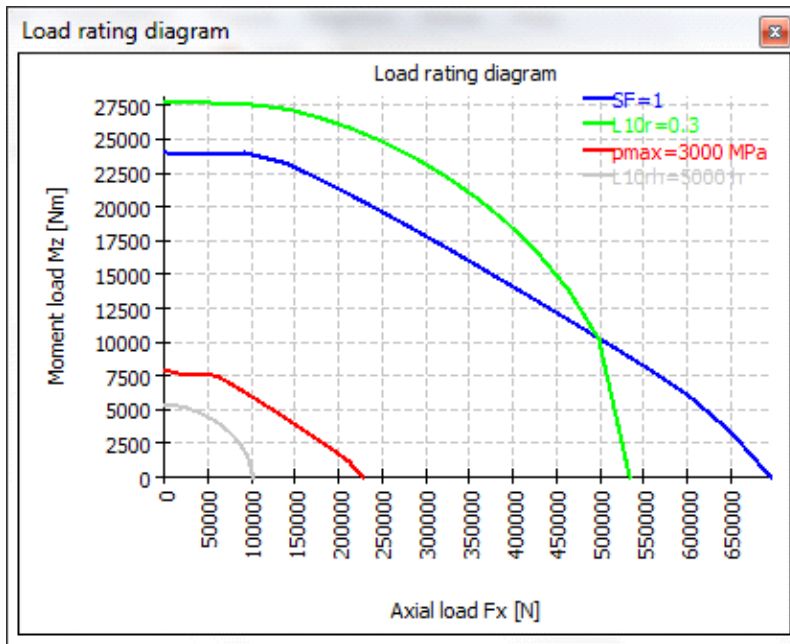
The subsurface shear stress is shown for the contact with the highest surface pressure. The required hardness depth is shown in the diagram dependent on the fatigue strength of inner and outer race.

The red curve is for the shear stress of the inner race, the blue curve for the outer race.



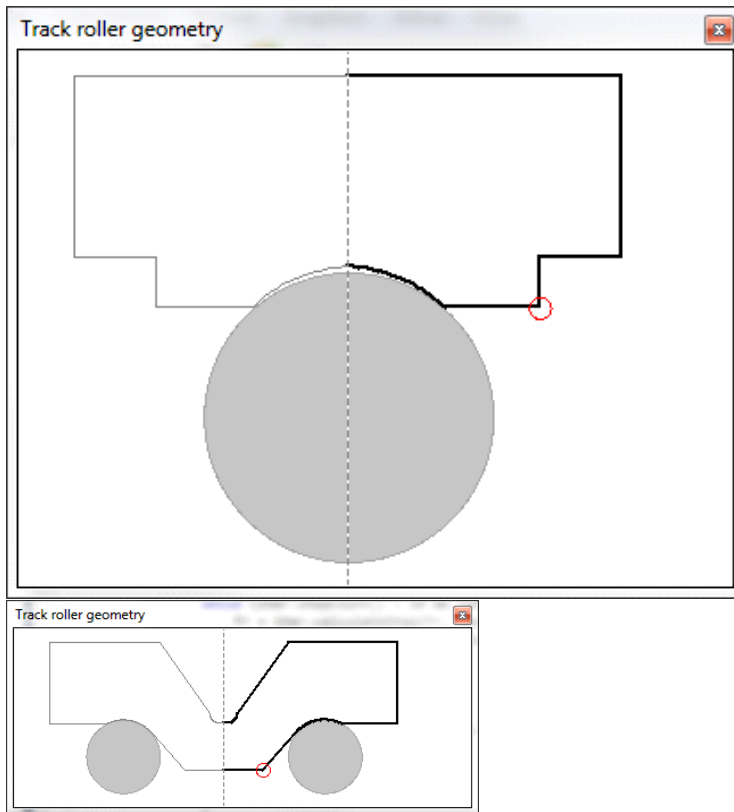
3.4.5.13 Load rating diagram

The load rating diagram shows the permissible range for tilting moment and axial load for different criteria which can be configured under 'Calculation' -> 'Load rating diagram'.



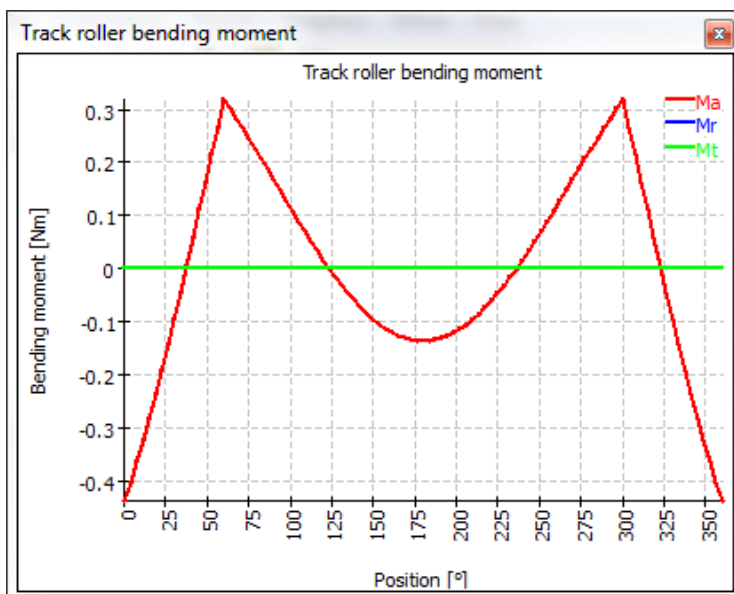
3.4.5.14 Track roller geometry

The track roller geometry is shown if defined using point data. For symmetric cross sections, the mirrored is shown in grey and the critical point for bending is marked by a red circle.



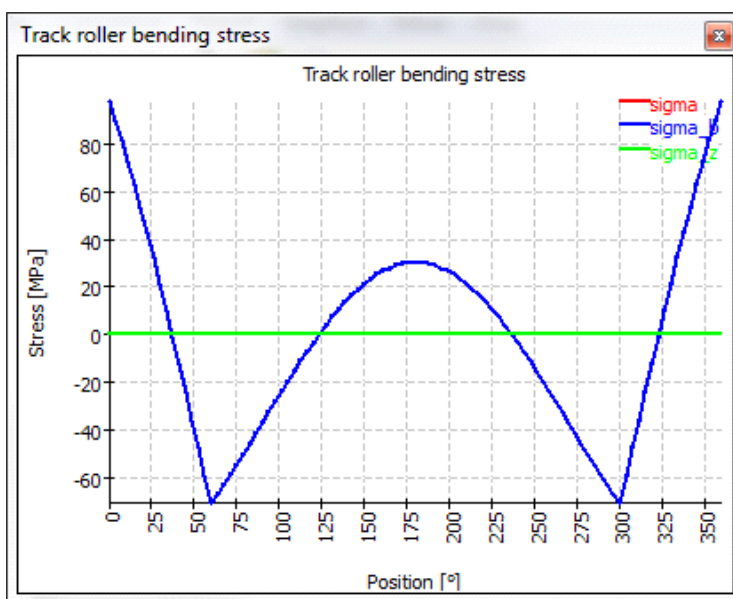
3.4.5.15 Track roller bending moment

The track roller bending moment is shown using the local coordinate system. For a pure radial load on a symmetric cross section, only the Moment around the bearing axis M_a occurs. The moment M_r is a tilting around a radial axis and the moment M_t is a torsional moment in the beam describing the outer ring.



3.4.5.16 Track roller bending stress

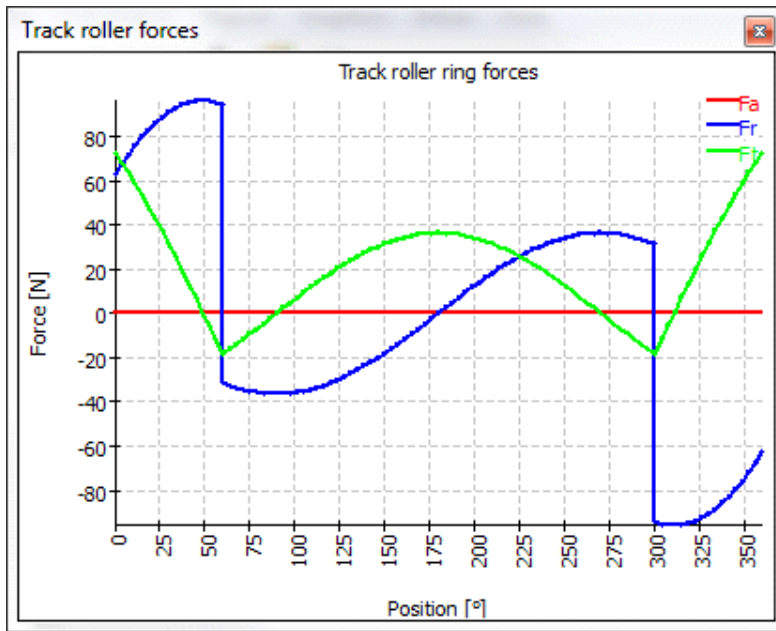
The bending stress and the stress due to tension are shown in the diagram. The bending is calculated for a critical point either defined manually or automatically by the software. These stresses are used to calculate a static and dynamic safety factor for the outer ring.



3.4.5.17 Track roller forces

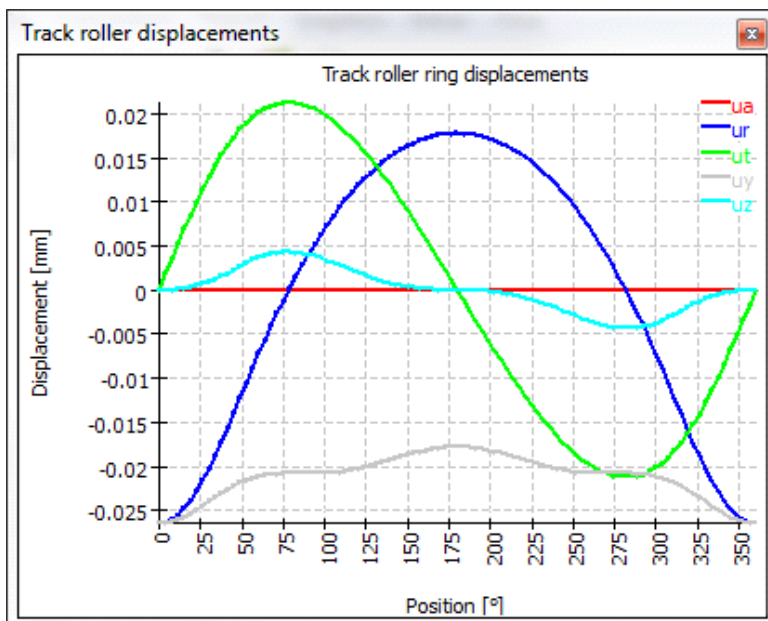
The forces in the beam modeling the outer ring are shown. The force F_a is in the direction of the bearing axis, therefore it is a shear force in the beam. The force F_r is a shear force in radial direction and F_t is the tangential force, so it is tension (positive) or pressure (negative) in the beam.

We can also see that loads are only introduced at 0° , 60° and 300° , so only three rolling elements are under load.



3.4.5.18 Track roller displacements

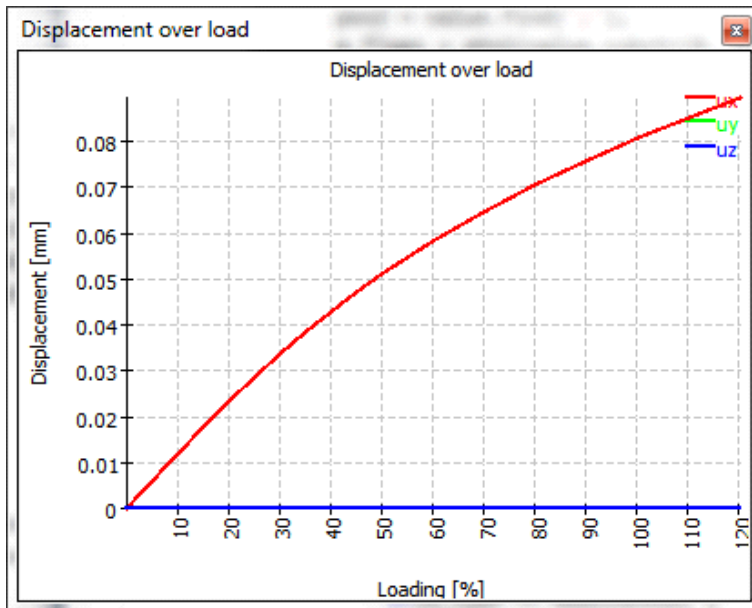
The displacements are shown in local coordinates and in global coordinated. The global coordinates allow seeing quickly that this example makes mainly a rigid body movement of the outer ring in negative y-direction. In other example,s the deformations of the ring might be different, so the values in local coordinates are more important.



3.4.5.19 Displacement over load

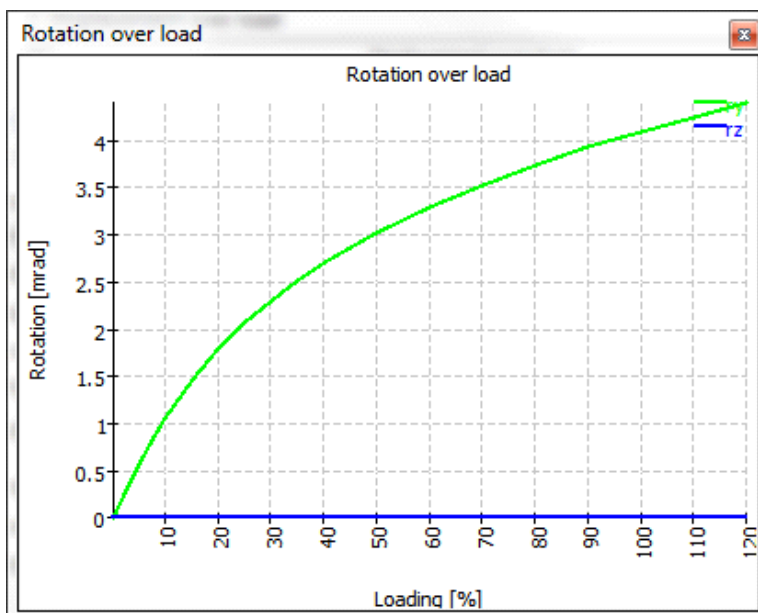
The displacement over load can be shown as graphics. All load components are multiplied with a load factor and the displacements are shown. Only loads are multiplied with the load factor, displacements or rotations entered are not modified.

This diagram can be used as information about stiffness. Dependent on the type of loading (axial or radial), different stiffness can be calculated.



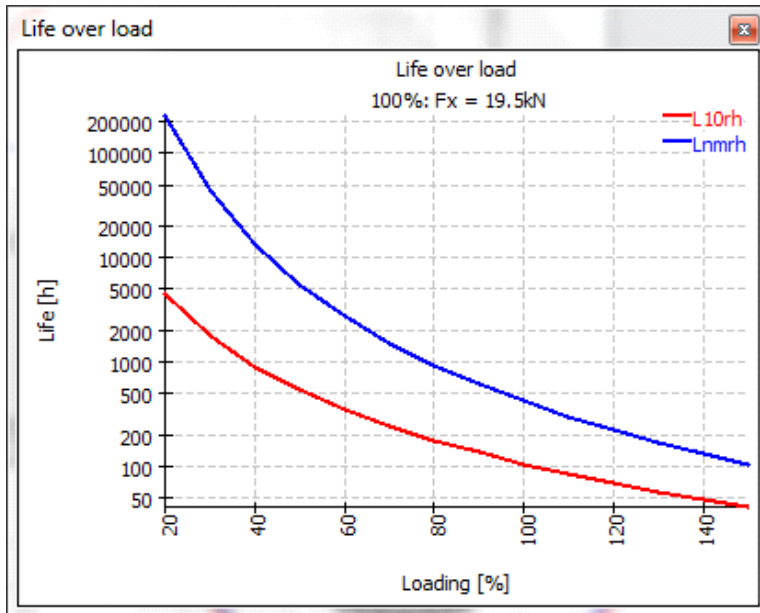
3.4.5.20 Rotation over load

Like in the graphics before, the rotations are shown dependent on a load factor. This graphic can only be shown if the moment is an input not for the tilting angle as an input. All loads are multiplied with the load factor.



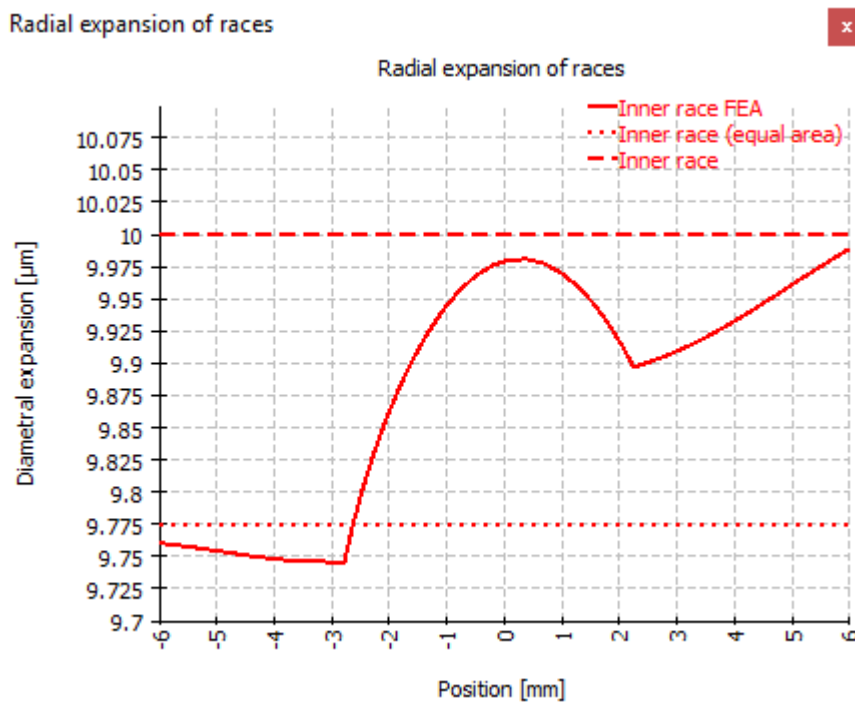
3.4.5.21 Life over load

The life over a load factor can be shown as a diagram. All loads and moments are multiplied by a load factor; all inputs for displacements/rotations stay constant. As reference the loading is shown on top of the diagram.



3.4.5.22 Radial expansion of races

The radial expansion of races is calculated using axisymmetric FEA considering fitting, temperatures and centrifugal forces. As comparison the radial expansion according to thick ring geometry as used for the calculation of fitting is shown. The dashed line shows the default calculation of fitting, the dotted line the calculation using equivalent cross section. This graphic can be used to check which approach is suited better for the actual conditions.



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MESYS Ball Screws

MESYS Ball Screw Calculation

4 MESYS Ball Screw Calculation

This Ball Screw calculation (Version 07/2019) calculates the load distribution and the reference rating life according to ISO/TS 16281 (DIN 26281).

The inner geometry of the ball screws must be provided by the user. The operating clearance can be specified. The loading can be specified as force/moment or displacement/rotation independently for each of the five degrees of freedom. Bearing sets can be defined by using a configuration with multiple rows.

The following results can be found in the report:

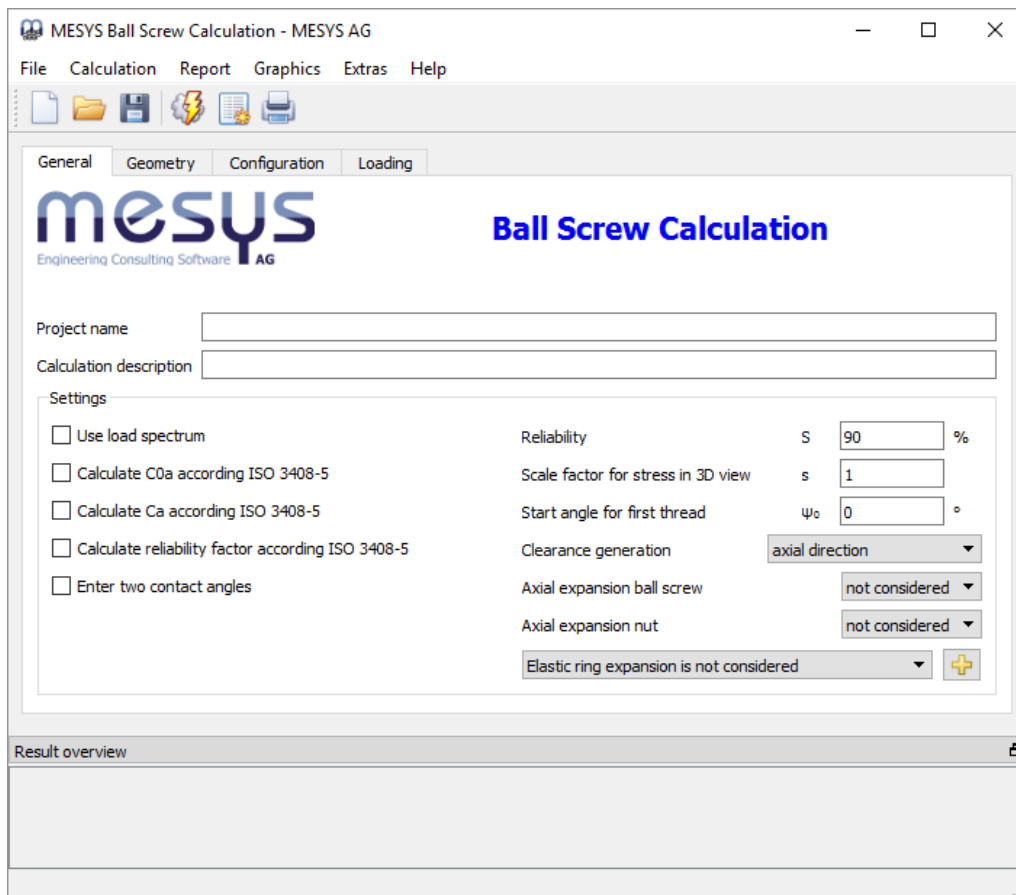
- Reference rating life according ISO/TS 16281
- Load distribution between rolling elements
- Reaction forces/moments and displacements/rotations
- Contact pressure for each contact
- Static safety factor

In addition to the report, the results are shown in several graphics. Report and user interface are both available in metric or US customary units. Supported languages for user interface and report are English, German, French, Spanish, Korean and Chinese.

4.1 Input Parameters

4.1.1 General

On the first input page in addition to a project name, several settings can be done.



4.1.1.1 Project name and calculation description

The project name and the calculation description are just inputs which are shown in the report header. They can be used to enter information about the purpose of the calculation for documentation.

4.1.1.2 Use load spectrum

A load spectrum can be used instead of a single load case. This option can be activated here. For each load case, a full calculation using all factors is made. The resulting life is calculated using the life of each element (load case). Result graphics are only shown in the report for the selected result element of the load spectrum. However, the result graphics corresponding to any element number (case number) of the load spectrum can be shown under the menu 'Graphics' -> 'Load spectrum' (only until 25 elements), as we can see in this picture.

All other graphics available under the menu 'Graphics' are only valid for the selected element.

Additionally, under 'Graphics' -> 'Load distribution (Load spectrum)', all the load distribution cases are superimposed on the chart at the same time. This is also possible for the roller graphics.

Please note that more intermediate results are printed in the report if a calculation is done for a single load case.

4.1.1.3 Calculate COa according to ISO 3408-5

If this option is selected, the static load capacity is calculated according to (ISO 3408-5, 2006). In this case, material properties like Young's modulus have no effect on the static load capacity.

If the option is not selected the static load capacity is calculated according (ISO 3408-5, 2006), but using f_0 according (ISO/TR 10657, 1991) instead of k_0 . In this case, the static load capacity is based on a permissible contact stress of 4200 MPa as for axial ball bearings according (ISO 76, 2006).

4.1.1.4 Calculate Ca according to ISO 3408-5

If this option is set, the dynamic load capacity is calculated using the exponent 0.86 in equation 8 of (ISO 3408-5, 2006). If the option is not set, the exponent 0.7 is used as in (ISO 281, 2007), (ISO/TR 1281-1, 2008) or (Lundberg, et al., 1947).

The load capacity is calculated according

$$C_a = f_c \cdot (i \cdot \cos \alpha)^{0.86} \cdot \tan \alpha \cdot z_l^{\frac{2}{3}} \cdot D_w^{1.8} \cdot (\cos \varphi)^{1.3}$$

with either an exponent 0.86 or an exponent of 0.7.

4.1.1.5 Calculate reliability factor according to ISO 3408-5

The modification factor for reliability is calculated according to the two parameter Weibull distribution according (ISO 3408-5, 2006) if this option is set. If the option is not selected, the three parameter Weibull distribution as in (ISO 281, 2007), (ISO/TR 1281-1, 2008) is used.

4.1.1.6 Enter two contact angles

Either one or two contact angles can be defined. Different contact angles can be used for screw and nut. The contact angle is defined by radial contact of the ball to either screw or nut.

4.1.1.7 Reliability S

As default, the bearing life is calculated for a reliability of 90%. The required reliability can be changed here with limits 90% and <100%. The life modification factor for reliability 'fac' is calculated according to this input using either the three parameter Weibull relationship as given as formula in (ISO/TR 1281-2, 2008) or the two parameter Weibull relationship used in (ISO 3408-5, 2006).

4.1.1.8 Scale factor for stress in 3D view s

The stress distribution is scaled to $s \cdot D_w$ for a contact stress of 4000MPa in the 3D view.

4.1.1.9 Start angle for first thread Ψ_0

The start angle defines the angular position of the left end of the first thread. With an angle of 0° , the start is on the y -axis.

4.1.1.10 Clearance generation

The nominal contact angle, ball diameter and radii of races are used to define the positions for centers of curvature in the case of zero clearance.

There are four possibilities to define clearance:

1. Centers of curvature are moved axially. This is only possible for gothic profiles with contact angle larger zero. The maximum clearance that can be achieved is limited.
2. Centers of curvature are moved radially.
3. Centers of curvature are moved in direction of contact angle. This cannot be used in case of contact angle zero.
4. Clearance is generated by reduction of ball diameter. This slightly reduces load capacities as the reduced ball diameter is used for calculation of load capacity.

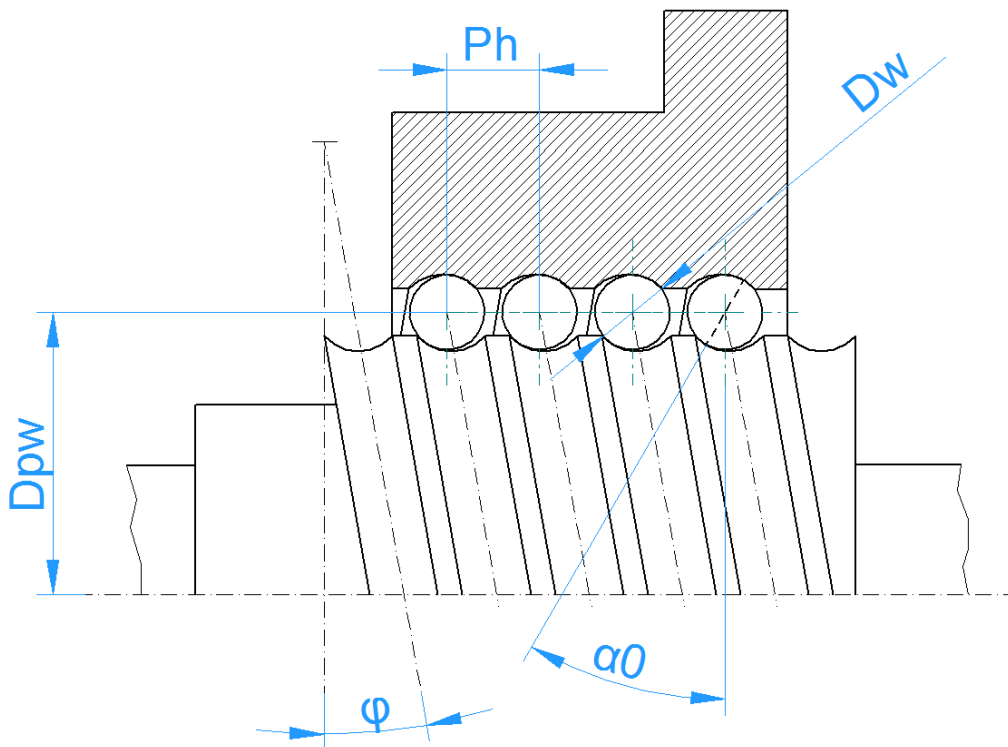
4.1.1.11 Axial expansion of ball screw and nut

The axial expansion of the ball screw and the nut three options are available: "Not considered", "fixed left", "fixed right". In case the load is introduced on one side, the screw will axially expand or compress because of the axial stresses. "Fixed left" means that the load is introduced on the left of the element and the axial stress will reduce to the right.

4.1.1.12 Elastic ring expansion

An elastic expansion/compression because of radial forces can be considered. The calculation is based on thick ring theory with constant pressure at the outside or inside. Either the minimum or the medium contact force of the balls is used to calculate this elastic deformation. The deformation is constant over the whole length of the nut. Therefore, the method is not suited for a pure moment load.

4.1.2 Ball screw geometry



General	Geometry	Configuration	Loading						
Number of starts	ns	1	Dynamic load number	Ca	0	kN	<input type="checkbox"/>		
Number of turns	nt	1	Static load number	C0a	0	kN	<input type="checkbox"/>		
Lead	Ph	0	mm	<input checked="" type="radio"/>	Axial clearance	Pa	0	mm	<input type="checkbox"/>
Lead angle	phi	0	°	<input type="radio"/>	Tolerance class	5			
Direction	right		<input type="checkbox"/>	Material ball screw	Steel			<input type="button" value="+"/>	
Number of loaded balls per turn	zl	0		Material ball nut	Steel			<input type="button" value="+"/>	
Ball diameter	Dw	0	mm	Material rolling element	Steel			<input type="button" value="+"/>	
Pitch diameter	Dpw	0	mm	Ball screw inner diameter	dsi	0	mm		
Contact angle	α	0	°	Nut outer diameter	dhe	0	mm		
Conformity ball screw	frs	0.52	<input type="button" value="+"/>	Ball screw lead error	Δ_{Ph}	0	mm		
Conformity ball nut	frn	0.52	<input type="button" value="+"/>						
Shoulder diameter ball screw	dSi	0	mm	<input type="button" value="+"/>					
Shoulder diameter ball nut	dSe	0	mm	<input type="button" value="+"/>					

4.1.2.1 Load capacities

The load capacities can be an input or an output. If they are given by the user they will not be changed because of surface hardness of the material. The surface hardness is only considered if the values are calculated by the software.

The load capacity will be calculated using the free contact angle α_0 , not the nominal contact angle α . This mainly makes a difference for nominal contact angle of zero in case of a full radius in races.

4.1.2.1.1 Dynamic load number

The dynamic load number is used for the calculation of bearing life. It is calculated according (ISO 3408-5, 2006) with factor f_c according (ISO/TR 1281-1, 2008). Either 0.86 or 0.7 according (ISO 281, 2007) is used as an exponent.

The dynamic load number can be influenced by a modification factor available at material properties.

4.1.2.1.2 Static load number

The static load number is calculated according (ISO 76, 2006) and (ISO/TR 10657, 1991) or according (ISO 3408-5, 2006).

4.1.2.2 Number of starts

The number of starts defines the number the ratio between lead and pitch or the number of parallel ball tracks.

4.1.2.3 Number of turns

Each ball track has this number or turns. It does not have to be an integer value.

4.1.2.4 Lead

The lead is the axial travel of the nut for a 360° rotation.

4.1.2.5 Lead angle

Instead of the lead, the lead angle can be used as input. The lead is $Ph = D_{pw} * \pi * \tan(\alpha)$

4.1.2.5.1 Direction

The lead can either be right or left handed. The sense of rotation is defined by this input.

4.1.2.6 Number of loaded balls per turn

The number of loaded ball per turn us usually calculated according (ISO 3408-5, 2006). It can be overwritten in case spacers are used.

4.1.2.7 Ball diameter

The diameter of the balls is specified here.

4.1.2.8 Pitch diameter

The pitch diameter is the diameter of the centers of rolling elements when they have the same amount of clearance to both races. The diameters of inner and

outer race are calculated by this value, the diameter of rolling element, the contact angle and the clearance.


4.1.2.9 Contact angle

Either one or two contact angles can be defined. For contact angle larger zero a gothic profile is used, for contact angle of zero a full radius is used.

If a full radius should be used, the clearance has to be generated either radially or by reduction of ball diameter.

4.1.2.10 Conformity of ball screw and ball nut

The conformity is the ratio between radius of curvature of a race and the ball diameter. For geometric reasons, the value has to be larger than 0.5. The values used for the calculation of tables in ISO 281 are $f_i=f_e=0.52$ for radial ball bearings and $f_i=f_e=0.535$ for thrust ball bearings as stated in (ISO/TR 1281-1, 2008).

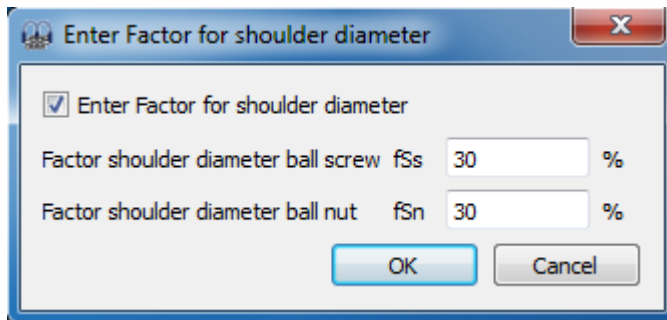
Using the -button behind the input field, the conformity can be calculated from a given radius. Ensure that you input the correct roller and pitch diameter before.

4.1.2.11 Shoulder diameter of inner and outer ring

The shoulder diameter of inner and outer ring can be defined.

The shoulder diameter is compared to the maximum extension of the contact ellipsis. A warning is shown if the ellipsis would extend above the shoulder. The Hertzian stress is not valid in that case.


The required shoulder diameter is shown in the report together with a length ratio eLR_i , eLR_e which is showing a safety of shoulder length. The length ratio is defined as the length from lower end of contact ellipsis to shoulder (the green line the graphics) divided by the length of the contact ellipsis $2a$ (red line in the graphics). The value should therefore be larger than 1 or 100%.



Instead of using an absolute value, the shoulder height can be defined using a percentage of ball diameters. A factor of 50% would be a shoulder up to the pitch diameter, so the factor should be between 10% and 40%. Using this factor allows to have default geometry on changing ball diameter or pitch diameter.

4.1.2.12 Axial clearance


The axial clearance is the difference between the possible axial displacement in positive and negative direction. It can be negative for gothic profiles, but it has to be positive for full radius profiles.

A pretension can be entered by using a negative value for the axial clearance P_a . A -button allows the input of radial clearance instead of axial clearance or a free contact angle can be used for full radius profiles.

4.1.2.13 Tolerance class

The tolerance class will be considered in calculation of static and dynamic load capacity.

4.1.2.14 Material

The material of the ball screw the nut and the balls can be defined here. Using the -button, the hardness and material treatment can be defined, too.

The material of the ball does only affect the contact stiffness in calculation of load distribution, it is not considered for the calculation of life.

4.1.2.15 Ball screw inner diameter

The inner diameter in case of a hollow screw. This input is only used in case axial or radial expansion is considered.

4.1.2.16 Nut outer diameter

The outer diameter of the nut. This input is only used in case axial or radial expansion is considered. In case the input value is zero, a diameter of $D_{pw} + 20 \cdot D_w$ is used internally.

4.1.2.17 Ball screw lead error

An optional lead error for the ball screw. This will affect the load distribution.

4.1.3 Bearing configuration

A group of nuts can be considered in the calculation. The nuts are assumed to be connected to a rigid shaft and housing. The loads or displacements are given for the origin of the bearing group instead for each single bearing.

A relative position can be entered for each bearing. An axial offset allows the definition of pretension in the system. The offset is an additional movement of the ball nut. The displacements, forces and moments under loading are applied at the reference position 0. So in most cases the positions should be chosen symmetrically around zero.

	Position [mm]	Offset [mm]
1	-30	0
2	30	0

Additional bearings can be added using the **+**-button. A row can be deleted using the **-**-button and the **X**-button clears all inputs.

With the context menu (right mouse button) in the table, several results values can be shown in the table. This allows seeing quickly how load or life is distributed among the nuts.

4.1.4 Loading

For each coordinate direction, you can select if a force or a displacement should be entered.

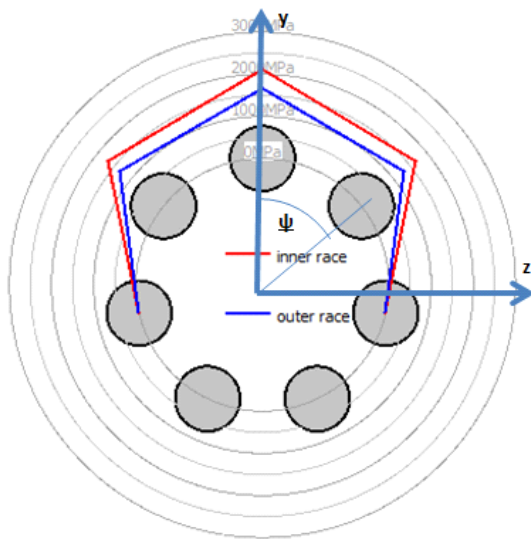
A moment load can be entered for two directions only, because the torque around the axis is connected to the axial force. The rotation angle rx will affect the axial displacement only.

Use the radio buttons to select the elements you want to enter.

Parameter	Value	Unit	Option	Value	Unit	Option
Axial load	0	N	<input checked="" type="radio"/> Displacement	0	μm	<input type="radio"/>
Radial load	0	N	<input checked="" type="radio"/> Displacement	0	mm	<input type="radio"/>
Radial load	0	N	<input checked="" type="radio"/> Displacement	0	mm	<input type="radio"/>
Moment	0	Nm	<input type="radio"/> Rotation angle	0	mrad	<input checked="" type="radio"/>
Moment	0	Nm	<input type="radio"/> Rotation angle	0	mrad	<input checked="" type="radio"/>
Moment	0	Nm	<input type="radio"/> Rotation angle	0	mrad	<input checked="" type="radio"/>
Speed inner ring	0	rpm	<input checked="" type="checkbox"/> Inner ring rotates to load			
Speed outer ring	0	rpm	<input type="checkbox"/> Outer ring rotates to load			
Temperature of shaft	20	°C		20	°C	

4.1.4.1 Coordinate system

The coordinate system is defined with x as the axial direction. The y-axis points up to the first rolling element and the angle ψ is positive around the x-axis or clockwise in the diagram on the right (looking in the direction of the x-axis). The angle starts with zero at the first rolling element on the y-axis as default; it can be changed under settings. Moments are positive if acting around the corresponding axis.



The load is acting on the inner ring, so a positive load in y-direction leads to a loading of the rolling elements on the top as shown in the graphics on the right.

4.1.4.2 Inner ring rotates to load

The resulting life is slightly different if the load distribution on a race is stationary or rotating. For many bearings, the inner ring rotates relative to the load. This is the case if the inner ring is connected to a rotating shaft and the load is stationary in space.

What is the correct input if only an axial load occurs? In this case, the input doesn't matter since the results are the same for both settings if all the rollers have the same loading.

4.1.4.3 Outer ring rotates to load

For the outer ring, the same is valid as for the inner ring. In many applications the outer ring is stationary to the load.

4.1.4.4 Temperature of screw and nut

The temperatures of the shaft and the nut are considered in axial elongation of the screw and the nut and they will affect the radial clearance due to radial thermal expansion.




4.1.5 Load spectrum



If at the tab-page 'General' the flag for the calculation with load spectrum is set, instead of for a single load case, the calculation will be done for a load spectrum. The load or displacement, the rotation speed and the temperatures can be specified for each element.

	Frequency	Fx [N]	Fy [N]	Fz [N]	rx [mrad]	ry [mrad]	rz [mrad]	ni [rpm]	ne [rpm]	Ti [°C]	Te [°C]
1	0.45	8000	0	0	0	0	0	1000	0	20	20
2	0.35	5000	0	0	0	0	0	50	0	20	20
3	0.1	2000	0	0	0	0	0	100	0	20	20

Inner Ring rotates to load
 Outer ring rotates to load
 Results for No 1

Using the context menu that opens clicking the right mouse button, you can select which forces or moments should be used as input values. If a value is not selected, the corresponding displacement/rotation can be entered.

Rows can be added using the -button or a selected row can be deleted using the -button. The -button clears all the inputs.

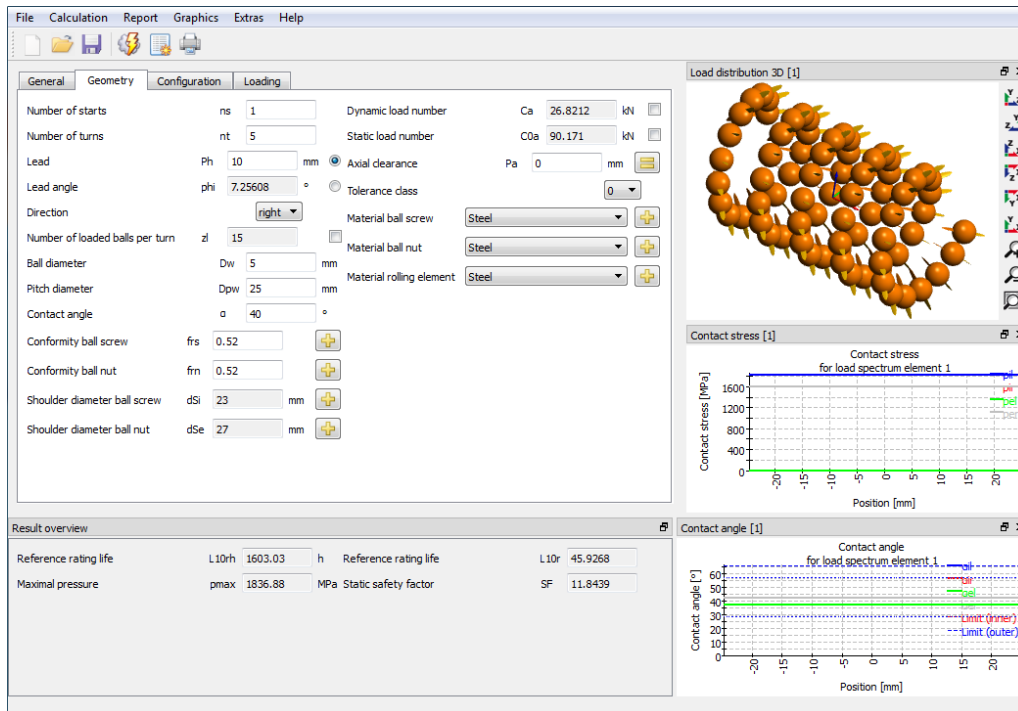
Using the -button, the load spectrum is read from a file. The columns that are currently shown with its units have to be used for the input file. As the data from the file is added to the inputs, you must clear the inputs first to define a full load spectrum. Likewise, a created load spectrum table can be exported with the corresponding -button.

As for the single load case, a selection is available if the ring is rotating relative to the load or not. A ring not rotating to the load would be on the safe side.

The load distribution is calculated for each load case. The resulting life for the whole load spectrum is shown as result and the minimum static safety of all elements. The result graphics are only shown for the element of the load spectrum that is chosen as results on this page.

4.2 Results

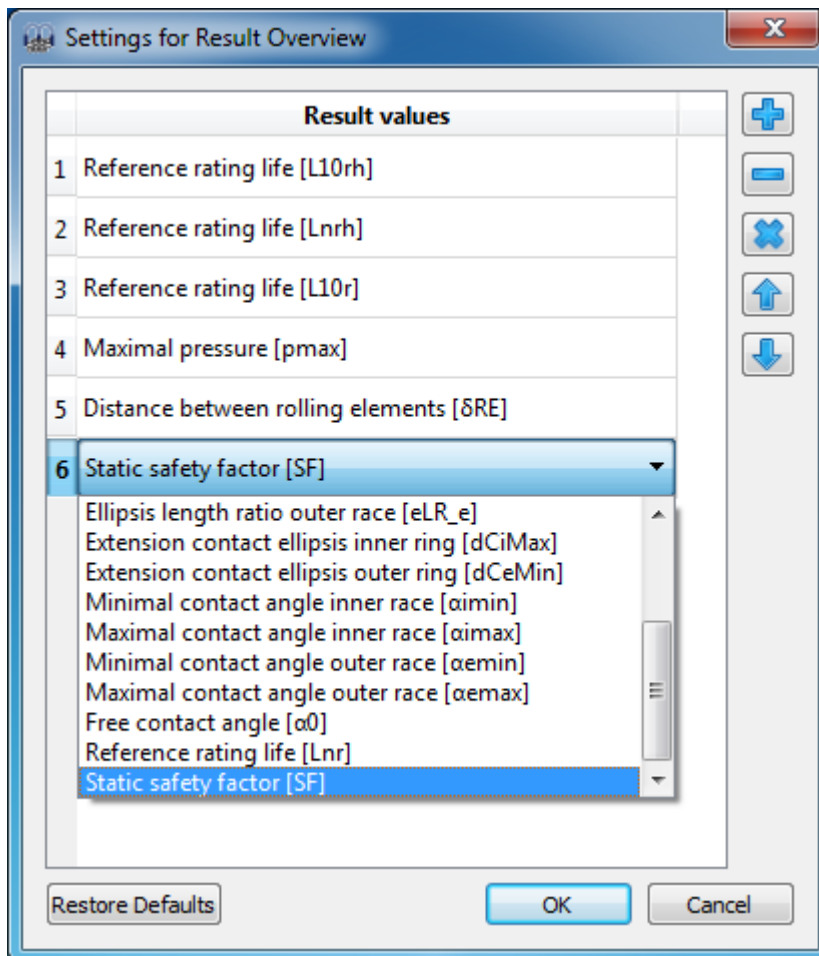
Results are provided as a small result overview directly in the software, a main PDF text report, a tolerances report and separate graphic windows. Graphic windows can be docked onto the main program interface and are automatically updated on each calculation.



4.2.1 Result Overview

The result overview shows the reference life L_{10r} in 10^6 rotations and the reference life L_{10rh} in hours. The modified reference life L_{nmr} considering reliability and lubrication is also shown in 10^6 rotations and in hours.

The maximum pressure and the static safety provide information about the maximum loading.



The result overview can be customized using “Extras->Result overview”. The items shown in the overview can be selected and additional items can be added using the **+**-button or the context menu with the right mouse button. Note that only values are actually shown in the result overview that are valid. So you will see fewer values than selected in the settings.

The settings for the result overview are saved in the windows user profile not in the calculation file. So they will be the same for different calculations.

4.2.2 Main Result Report

The text report is provided in PDF format.

In the menu "Report->Options" the contents of the report can be configured. The graphics to be included can be selected and some sections of the report could be discarded if not of interest. The legend for all the table parameters can be shown in the report.

4.2.2.1 Saving main report

The main report can be saved using the menu "Report->Save Report As" in different formats. It can be saved as PDF, DOC, DOCX or ODT (Open Office) format.

4.2.2.2 Customizing report logo

As default, a MESYS logo is shown on top of the report. This can be customized by defining a logo in the "mesys.ini" file. The logo should be created using a size, so that it fits into the header table. The DPI setting in the logo is considered, so that a high-resolution image could also be used. Additionally, it also exists the possibility to create an own template of the supported file formats (DOC, DOCX and ODT) and generate the report from it, thus having the chance to customize the logo.

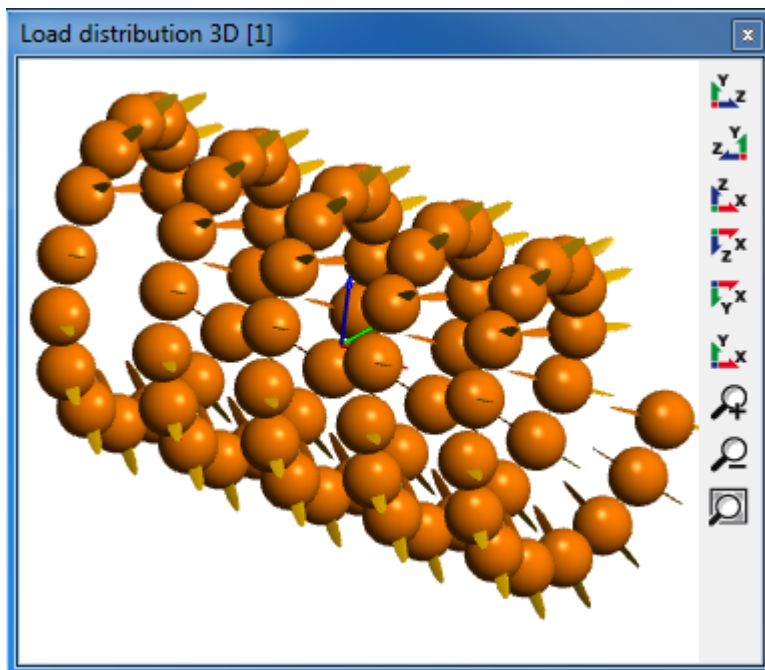
4.2.3 Result Graphics

Several results can be shown as graphics. All graphics are also appended to the report, so that a full documentation is provided in one document.

Using the context menu (right mouse button), in a graphics window the graphic can be exported as file or be printed.

4.2.3.1 Load distribution 3D

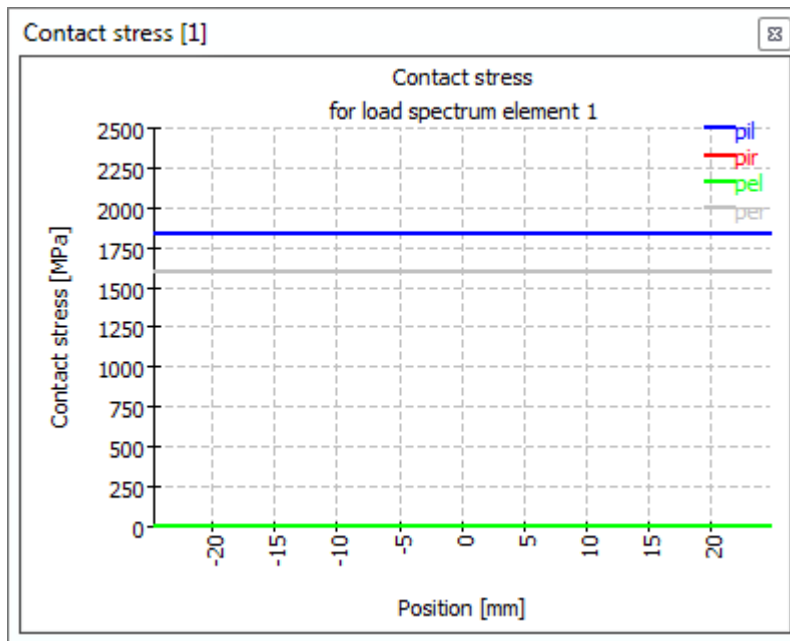
Here the contact stress of an angular contact bearing with radial and axial load and relatively high speed is shown. The contact stress is larger on the inner race (red) in the load zone but slightly larger on the outer race on the opposite side because of centrifugal loads.



With this chart, it can be noticed the contact stress by means of color-based representation of the contact stress levels on each rolling element and the corresponding contact ellipse on them. The visualization can be accurately rendered in any perspective.

4.2.3.2 Contact stress

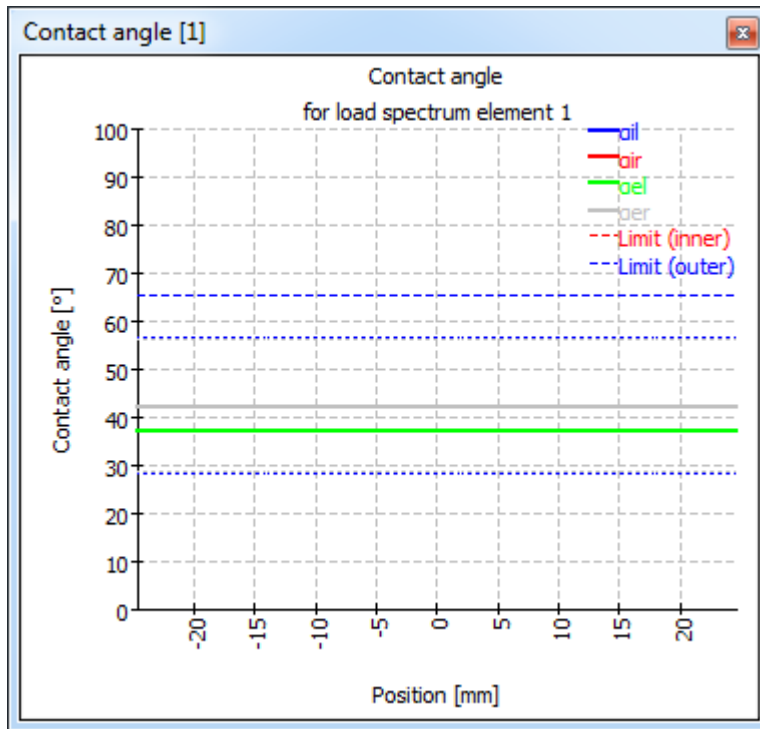
The contact stress is shown for each ball and each of the four possible contacts.



4.2.3.3 Contact angle

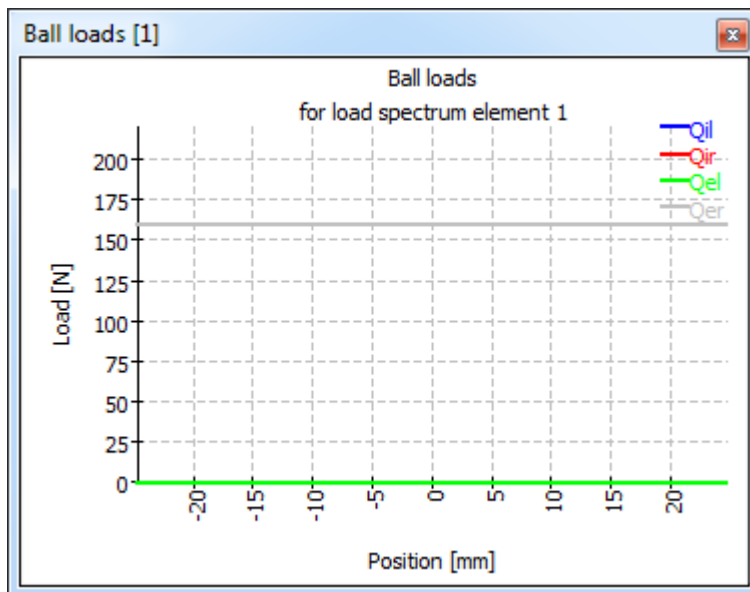
The contact angle is shown for a nominal contact angle of 40°. As for typical axially loaded ball screws, both the load level for all balls and the contact angles of inner and outer race are almost equal along the axial position.

The extension of the contact ellipsis is shown with the dotted lines. The contact angle corresponding to the shoulder is shown as dashed straight line at the top, so the shoulder is high enough in this case.



4.2.3.4 Ball loads

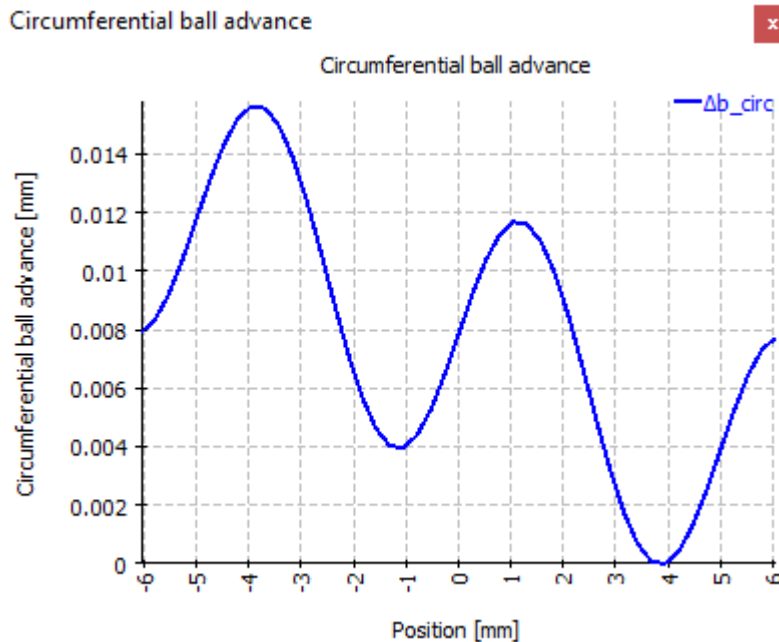
This diagram shows the forces on each ball for the up to four contacts.



4.2.3.5 Circumferential ball advance

The circumferential ball advance is shown as graphics. It is calculated based on the assumption that the orbit speed of each ball is constant in one section. Because of variation of contact angles the ball orbit speed varies between the rolling element. For calculation is done in the same way as for ball bearings, the lead is

not taken into account. A decrease in ball advance means that the next ball is slower and this will lead to forces between the balls. The calculation of orbit speed does not take into account a contact between the balls.



4.3 Bibliography

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Niemann, G., Winter, H. und Höhn, B.-R. 2005. *Maschinenelemente Band 1: Konstruktion und Berechnung von Verbindungen, Lagern, Wellen*. Heidelberg : Springer, 2005.

Paland, E.-G. 1968. Die Lastverteilung in einem auf der Ebene abgestützten Zylinderrollenlager. *Maschinenbautechnik (Schmierungstechnik)*. 17, 1968, Bd. 8.

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MESYS Hertz Calculation

MESYS Hertz Calculation

5 MESYS Hertz Calculation

The MESYS Hertz Calculation (Version 07/2019) calculates the Hertzian stresses and subsurface stresses for point or line contact.

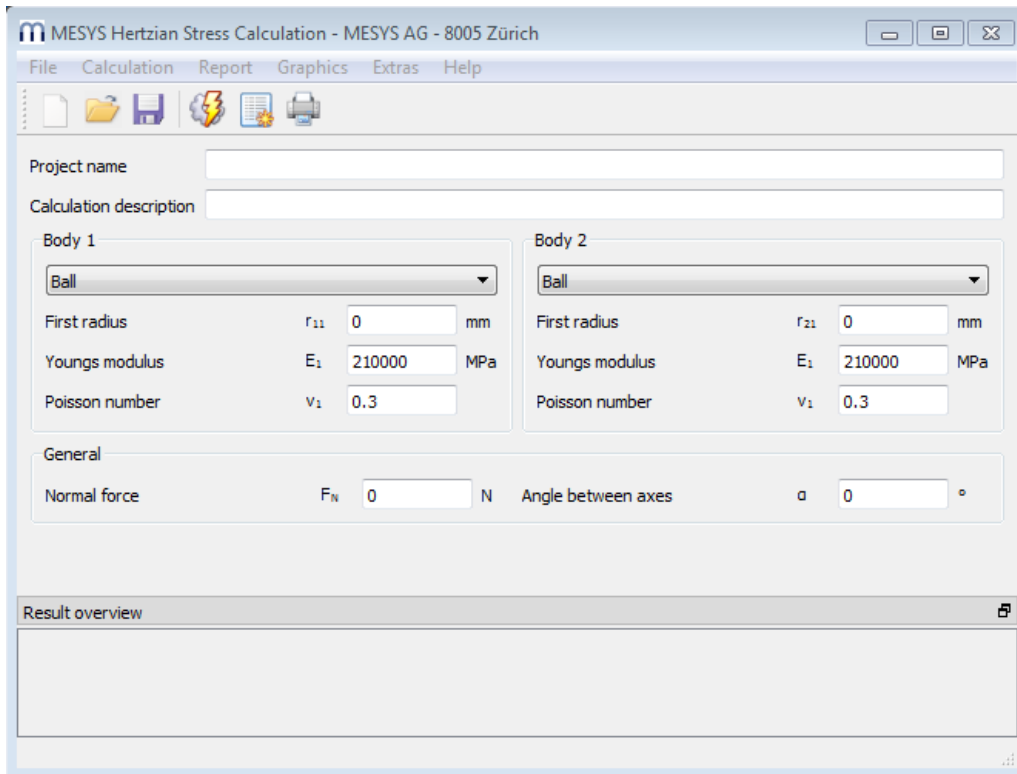
The following results can be found in the report:

- Major half axis of contact ellipsis a
- Minor half axis of contact ellipsis b
- Approach of both bodies
- Contact stiffness R
- Hertzian stress
- Maximal shear stress
- Maximal octahedral shear stress
- Depth for max. shear stress
- Depth for max. octahedral shear stress
- Equivalent stress (Tresca)
- Equivalent stress (Mises)

In addition to the report, the results are shown in several graphics. Report and user interface are both available in metric or US customary units. Supported languages for user interface and report are English, German, French, Spanish, Chinese and Korean.

5.1 Input Parameters

The input parameters are all shown at a unique page:

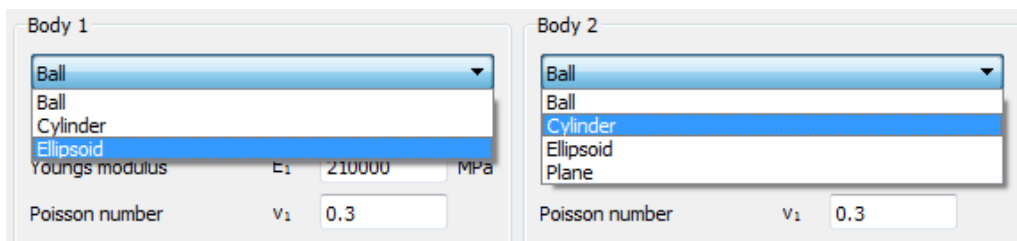


5.1.1 Project name and calculation description

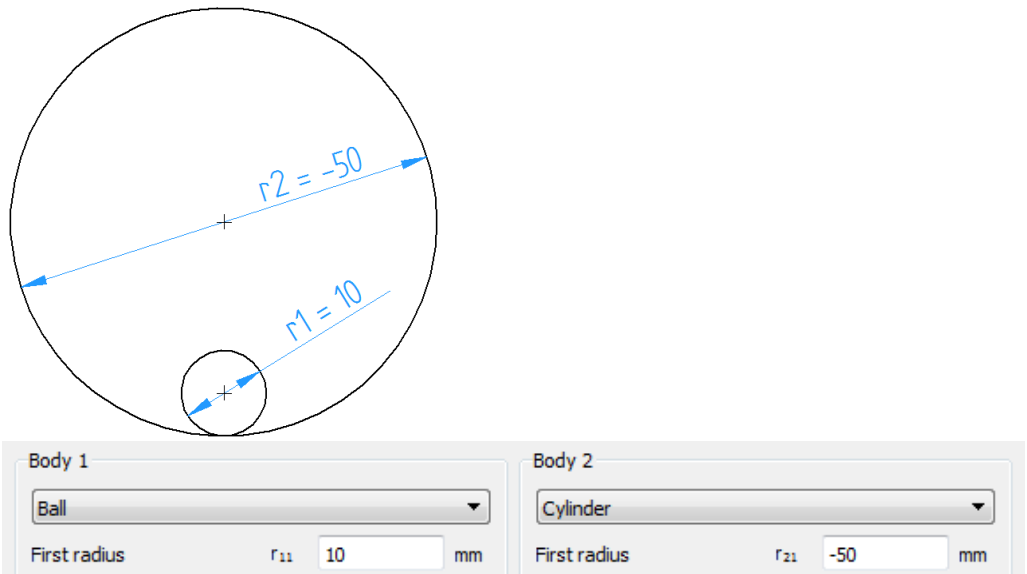
The project name and the calculation description are just inputs which are shown in the report header. They can be used to enter information about the purpose of the calculation for documentation.

5.1.2 Bodies in contact

Body types such as ball, cylinder, ellipsoid and plane are available on the drop-down lists corresponding to the 'Body 1' and 'Body 2' sections. All combinations are possible, apart from the plane-plane contact for obvious reasons.

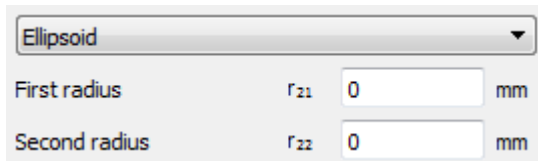


5.1.2.1 Body radii



Only one radius is needed for balls and cylinders. Note that concave surfaces have to be defined by means of negative values.

It will be necessary to specify the two radii of an ellipsoid, so the input is activated when selected:



5.1.2.2 Material properties

Only young modulus and poisson's ratio are needed for the bodies in contact. The stresses for both properties are the same for equal materials, for different materials they can differ.

5.1.3 General



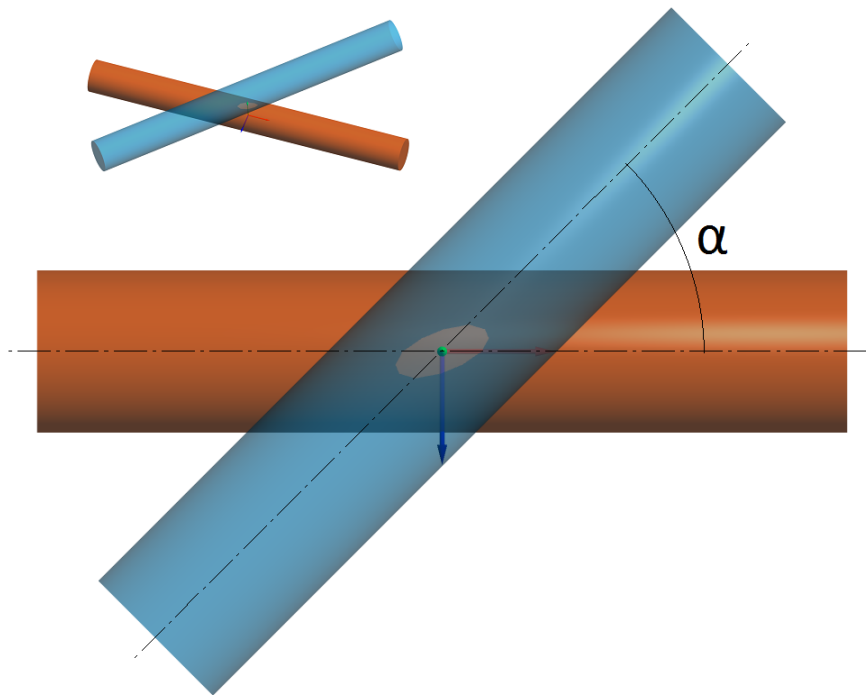
5.1.3.1 Normal force

Here the contact load is specified as normal force.

5.1.3.2 Angle between axes

The radii of curvature for two bodies can be given in two planes and an angle between these planes can be specified. The angle is defined between the lines

which are normal to the planes where the radii r_{11} and r_{21} lie, as shown in the picture:

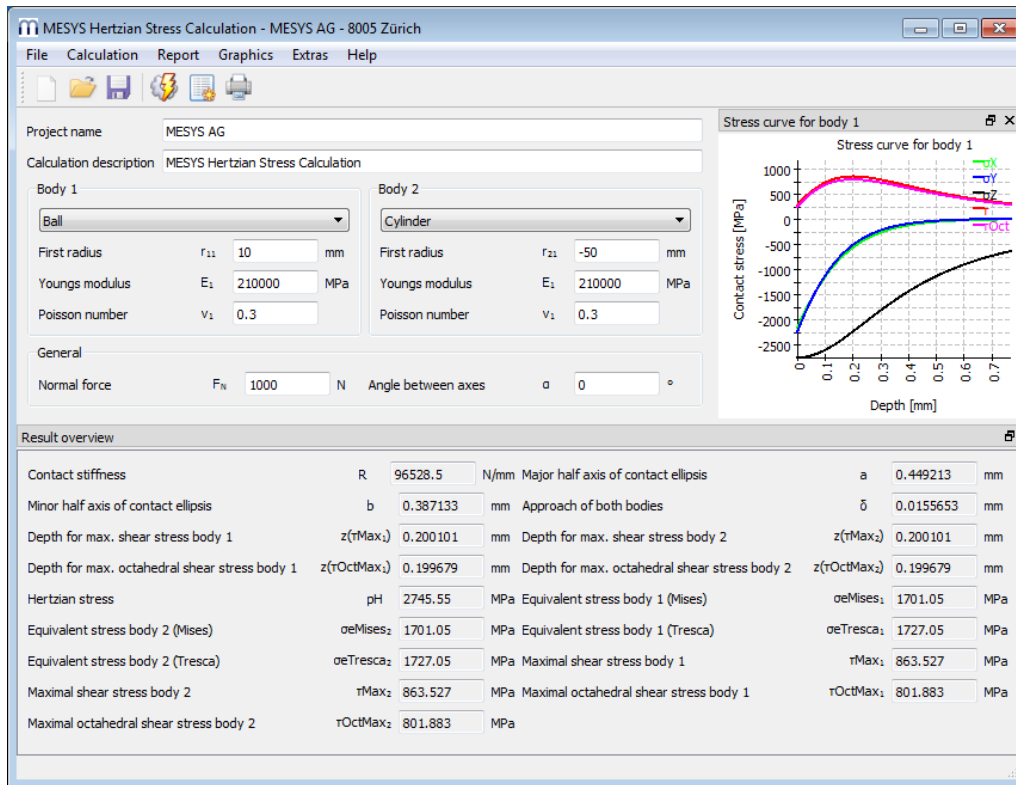


5.1.3.3 Length for line contact

For the case of two cylinders in contact, it is required to define the length for line contact, so the corresponding input field will be activated under 'General' as shown in the picture above.

5.2 Results

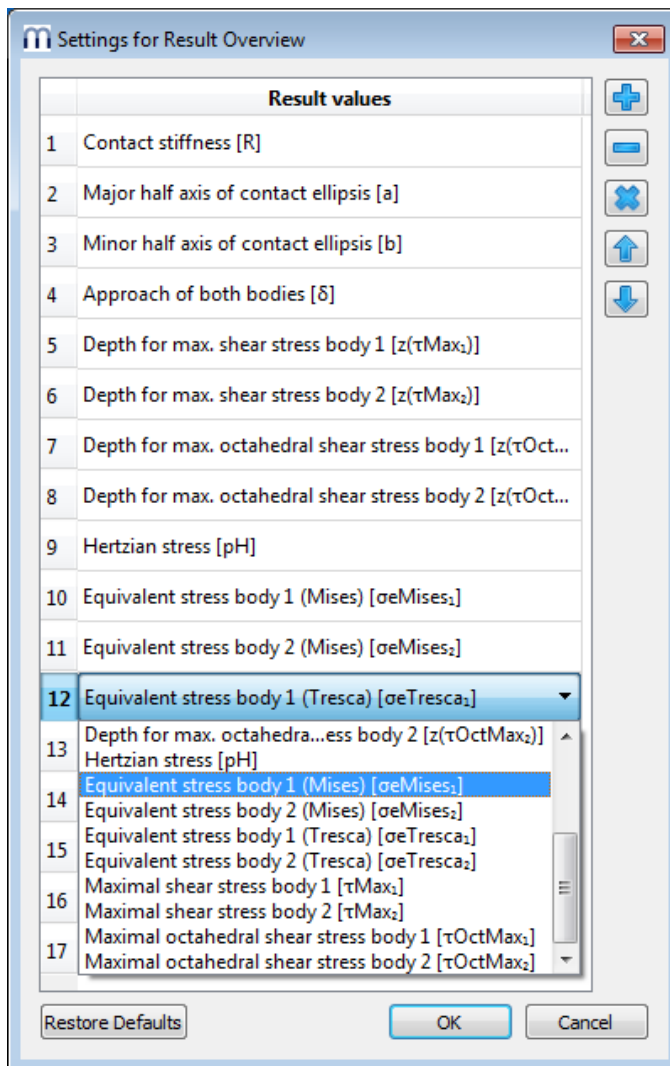
Results are provided as a small result overview directly in the software, a main PDF text report, a tolerances report and separate graphic windows. Graphic windows can be docked onto the main program interface and are automatically updated on each calculation.



5.2.1 Result Overview

Each and every one of the results can be chosen to be displayed at the results overview window of the software. Under the menu “Extras->Result overview” all the items are listed in a table called ‘Result values’. By doing a double-click on the table items, a drop-down list allows the user to select the result to be shown. By using the and buttons, any item of the table can be added or removed, as well as it is possible to clear the whole table with -button or rearrange it with the up and down arrows.

The settings for the result overview are saved in the windows user profile not in the calculation file. So they will be the same for different calculations.



5.2.2 Main Result Report

The text report is provided in PDF format.

In the menu 'Report->Options' the contents of the report can be configured. The graphics to be included can be selected and some sections of the report could be discarded if not of interest. The legend for all the table parameters can be shown in the report.

5.2.2.1 Saving main report

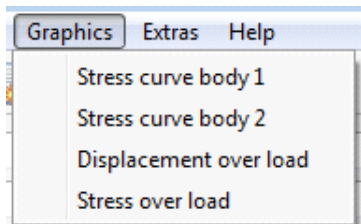
The main report can be saved using the menu "Report->Save Report As" in different formats. It can be saved as PDF, DOC, DOCX or ODT (Open Office) format.

5.2.2.2 Customizing report logo

As default, a MESYS logo is shown on top of the report. This can be customized by defining a logo in the "mesys.ini" file. The logo should be created using a size, so that it fits into the header table. The DPI setting in the logo is considered, so that a high resolution image could also be used. Additionally, it also exists the possibility

to create an own template of the supported file formats (DOC, DOCX and ODT) and generate the report from it, thus having the chance to customize the logo.

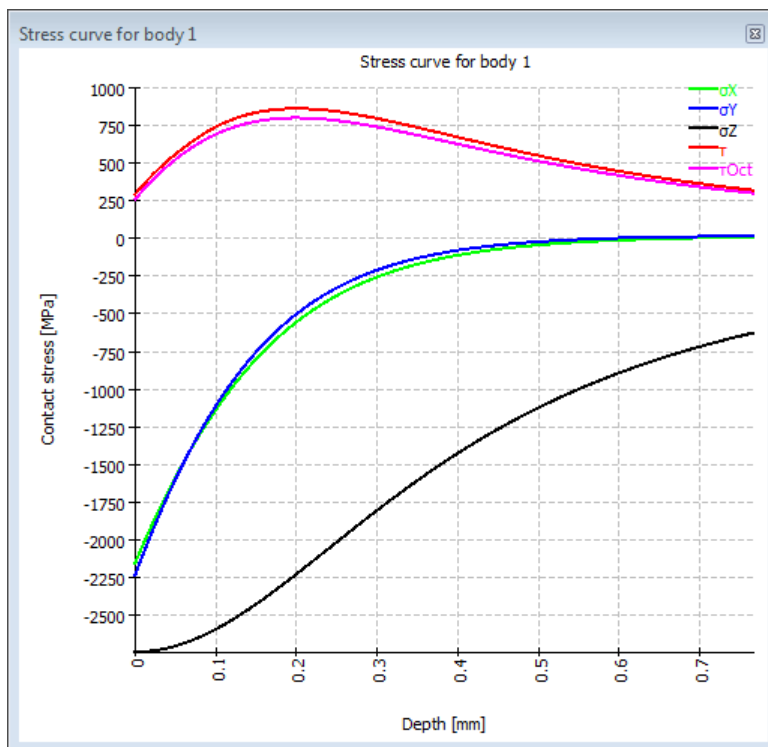
5.2.3 Result Graphics



All the available charts are under the menu 'Graphics'. For the current version, the stress curves for both bodies as well as displacement over load or stress over load can be analyzed.

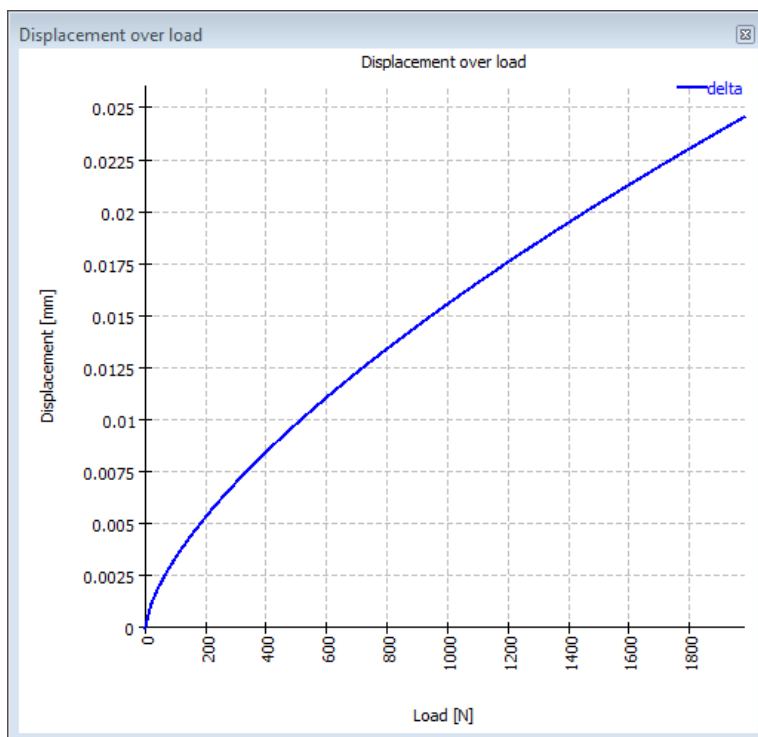
5.2.3.1 Stress curves for body 1 and body 2

Maximum principal stress (σ_{xx} σ_{yy} σ_{zz}), maximum shear stress (τ) and maximum octahedral shear stress (τ_{Oct}) are represented for at the line of contact under the surfaces.



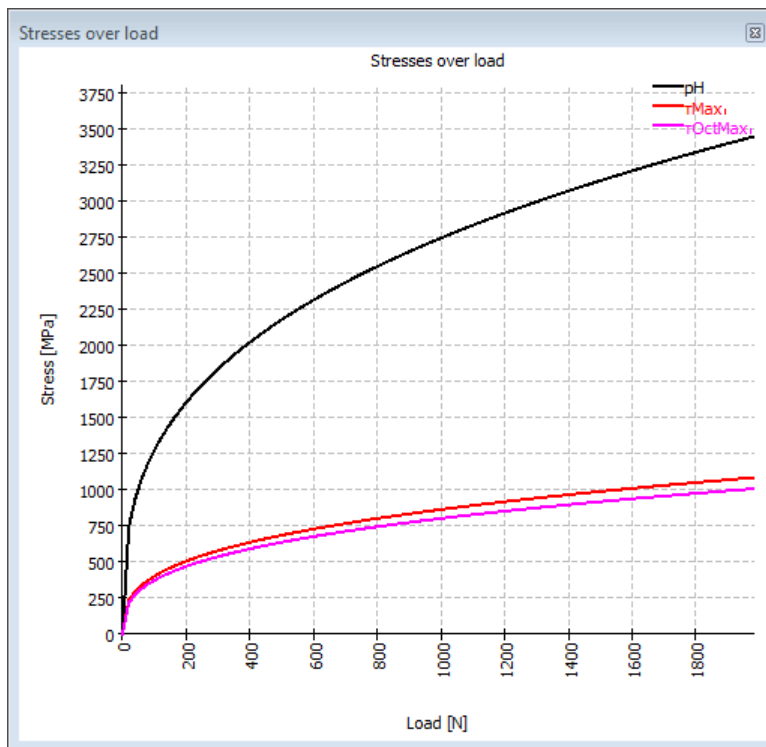
5.2.3.2 Displacement over load

This chart represents how the bodies (1 and 2) move towards each other by amounts of δ_1 and δ_2 as the load increases. Therefore, delta is the total decrease of distance by an amount $\delta = \delta_1 + \delta_2$ between points on the bodies not affected by the local deformation near 0.



5.2.3.3 Stress over load

Here the Hertzian stress, maximum shear stress and maximum octahedral stress are represented against the load.



5.2.4 Bibliography

Boresi, Arthur P. and Schmidt, Richard J. 2005. *Advanced Mechanics of materials.* s.l. : John Wiley&Sons, 2005.

Calculation for axial-radial-roller bearings

The software calculates the stiffness, the load distribution and the life of axial-radial-roller bearings. The geometry of parts is defined by horizontal and vertical lines; several parts can be defined. The parts can be connected axially or radially by bearings. A bolted joint and loads on surfaces or point loads can be considered.

The deformation of parts is calculated by a nonlinear FEA, while the coupling to the rollers is done analytically according ISO/TS 16281. Bolts are modeled using beam elements.

A first calculation step is done to evaluate the bolt pretension without further loads. Further calculation steps are then done with loading.

MESYS Axial-Radial-Rollerbearings

6 MESYS Axial-Radial-Rollerbearings

The calculation software for axial-/radial-roller-bearings or three row slewing rings is considering deformations of bearing rings using finite element analysis. The bearing geometry of axial symmetric rings can be defined by the user by a polygon in the cross section. An arbitrary number of axial or radial cylindrical roller bearings or cross roller bearings can be defined and the pretension of bolts is considered. Loads can be applied on a surface or on single points. For the definition of stiffness a calculation of a series with variation of load is possible.

As results the following data is available in several graphics and a PDF report:

- The pressure distribution on the rollers
- The bearing life for each row and the system according ISO/TS 16281
- The deformation and the stiffness defined by two points
- The maximal and minimal load on the bolts

6.1 General usage

The software can be called with several file names on the command line. All calculations are run sequentially and the report is generated without showing a user interface. Example: „MesysAxRaRBC.exe file1.xml file2.xml“

For larger models, the calculation time can be up to an hour. Therefore the following approach is recommended: Choose ‘Only preprocessing’ and build the model interactively. Save the model and run the calculation using the command line. Afterwards, the results can be loaded into the program to evaluate the results.

If several calculations have to be run, the software can be called with several file names. These calculations are done sequentially. The FEA is using up to two processor cores. If more cores are available the program can be started in parallel to run multiple calculations if the amount of memory is sufficient.

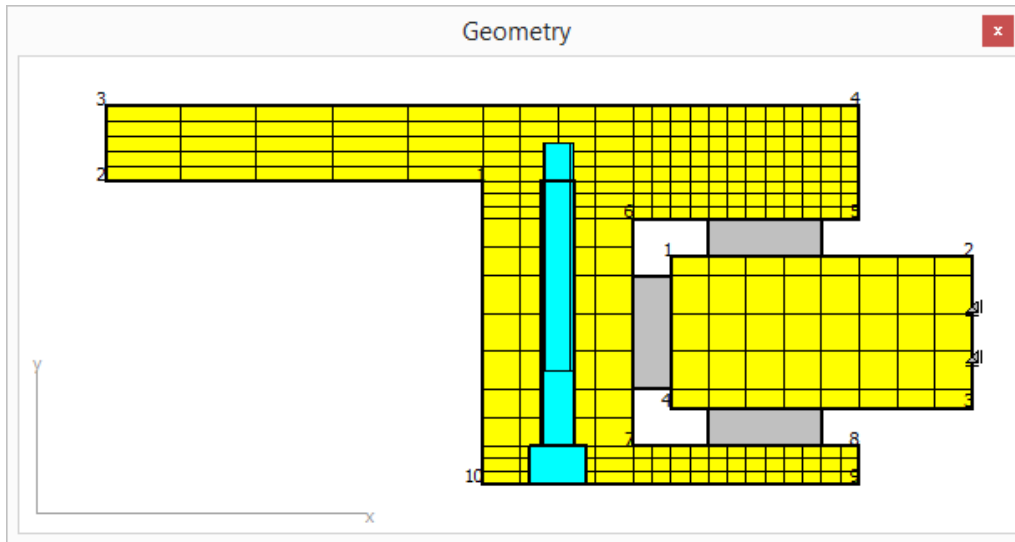
The number of processor cores used in FEA can be changed by changing following entry in mesys.ini:

```
[axrarbc]
numberofthreads=2
```

Using more than two threads will lead only to small improvements.

6.2 Inputs

The input data is provided on several input pages which are described on the following pages. The input data is for the following geometry:



6.2.1 General

General	Geometry of parts	Bolts	Bearings	Cross roller bearings	Loading	Load spectrum	Results
General							
Project name <input type="text"/>							
Calculation description <input type="text"/>							
Number of parts	<input type="text" value="1"/>	Number of radial bearings	<input type="text" value="0"/>				
Maximum angle for mesh	Ψ_{max} <input type="text" value="0"/> °	Number of axial bearings	<input type="text" value="0"/>				
Number of bolted joints	<input type="text" value="0"/>	Number of cross roller bearings	<input type="text" value="0"/>				
<input type="checkbox"/> Consider load spectrum		Number of surface loads	<input type="text" value="0"/>				
<input checked="" type="checkbox"/> Preprocessing only		Reliability for bearings	Z <input type="text" value="90"/> %				
<input checked="" type="checkbox"/> Allow general geometry input							
Calculation of series							
<input type="checkbox"/> Activate calculation for series		Number of steps	<input type="text" value="3"/>				
Start factor	<input type="text" value="0"/>	End factor	<input type="text" value="1"/>				
Lubrication							
Viscosity at 40°C	nu40 <input type="text" value="220"/> mm ² /s	Oil lubrication with on-line filter ISO4406 -/17/14 <input type="button" value="+"/>					
Viscosity at 100°C	nu100 <input type="text" value="19"/> mm ² /s	Temperature	TOil <input type="text" value="50"/> °C				
<input type="checkbox"/> contains effective EP additives		<input checked="" type="checkbox"/> Calculate modified life					

6.2.1.1 Number of parts

The bearing can be build using several parts. In usual cases two parts are used. There is no restriction on the number of parts.

6.2.1.2 Number of bearings

The number of radial, axial and cross roller bearings can be defined here. In usual cases one radial and two axial bearings are needed.

6.2.1.3 Maximum angle for mesh

For the meshing in circumferential direction, a maximum angle can be defined. Using a value of zero, the angle results from number of bolts and rollers. The angle can only be reduced but not be increased.

A large angle can lead to increased stiffness of the parts. To estimate the influence of the angle, some calculations for comparison can be made.

6.2.1.4 Number of bolted joints

One or more rings of bolts can optionally be considered. Only one bolted joint can be defined for one part.

6.2.1.5 Consider load spectrum

A load spectrum can be considered. If a calculation of series is active, only multiple speeds can be defined. Without calculation of series, also different loads can be defined per load spectrum element.

6.2.1.6 Only preprocessing

If this option is activated only the model is created but no FEA is performed on running the calculation.

6.2.1.7 Allow general geometry input

If this option is not set, the geometry is defined by horizontal and vertical lines only. This allows to enter one number only per geometry line.


If the option is activated, all values for the geometry coordinated have to be defined, but a general polygon is permitted as geometry. A meshing algorithm is used then, that will lead to less elements than with horizontal and vertical lines only.

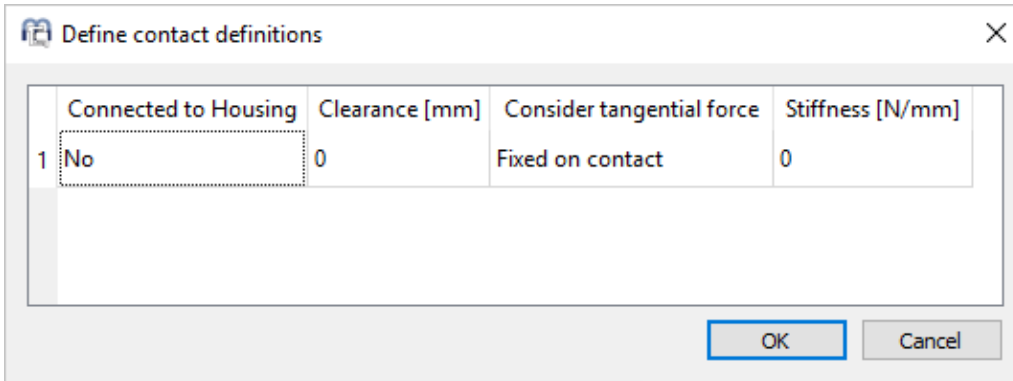
6.2.1.8 Number of surface loads

The number of loads acting on surfaces can be defined using this input value.

6.2.1.9 Number of contacts

A contact between parts or between a part and the rigid environment can be defined if 'Allow general geometry input' is activated. The number of contacts is the number of contact definitions. Each definition can be used multiple times.

The -button allows to define the properties for each contact definition:



	Connected to Housing	Clearance [mm]	Consider tangential force	Stiffness [N/mm]
1	No	0	Fixed on contact	0

The first column defines if the contact is between a part and the rigid environment (the 'housing') or between two parts.

The second column allows to define the clearance in the contact.

All contacts lead to a normal force in case of contact, but there are different options for the tangential force, which can be defined in the third column:

- "Fixed on contact" will constrain the tangential movement in case of contact.
- "No constraint" will lead to no constraint in tangential direction.
- "Welded" will lead to a tangential constraint always. In addition, also the normal forces can act in both directions.

The stiffness input in the last column should be zero in most cases. In case a value larger zero is used, it will be used for the normal stiffness. This can be useful for fixing a part in space with low stiffness. As the stiffness is used per node, its effect depends on the mesh size.

6.2.1.10 Reliability of rolling bearings

The bearing reliability is 90% as default. The input is used for the calculation of life modification factor a_1 .

The reliability is set for all bearings to the same value.

6.2.1.11 Series calculation

A series of load cases can be calculated by multiplying the load with a factor.

The load factor for the first calculation is the 'start factor' for the last calculation the 'end factor' is used. A given number of steps is calculated in between. For the results, a selection is available for the load step shown in graphics.

6.2.1.12 Lubrication

For the calculation of modified life, the definition of the lubrication is needed. The lubricant viscosity is calculated for the lubricant temperature using the reference values at 40°C and 100°C.

The selection 'Consider lubrication' defines if the modified life L_{nmrh} and the data for lubrication is shown in the report. For a small rotation speed, there will be no effective lubrication and the aISO factor approaches 0.1.

The lubricant cleanliness is considered by the factor e_C . It can be selected if the lubricant contains effective EP additives which can have an influence on aISO factor on small speed.

The e_C Factor can either be defined by lubricant cleanliness or it can be input by the user. An overview of recommended e_C factors is shown in the following table:


Contamination level	e_C	
	$D_{pw} < 100mm$	$D_{pw} \geq 100mm$
Extreme cleanliness Particle size of order of lubricant film thickness, laboratory conditions	1	1
High cleanliness Oil filtered through extremely fine filter; conditions typical for bearings greased for life and sealed	0.8 to 0.6	0.9 to 0.8
Normal cleanliness Oil filtered through fine filter; conditions typical for bearings greased for life and shielded	0.6 to 0.5	0.8 to 0.6
Slight contamination	0.5 to 0.3	0.6 to 0.4
Typical contamination Conditions typical of bearings without seals; course filtering; wear particles from surroundings	0.3 to 0.1	0.4 to 0.2
Severe contamination Bearing environment heavily contaminated and bearing arrangement with inadequate sealing	0.1 to 0	0.1 to 0
Very severe contamination	0	0




6.2.2 Geometry of parts

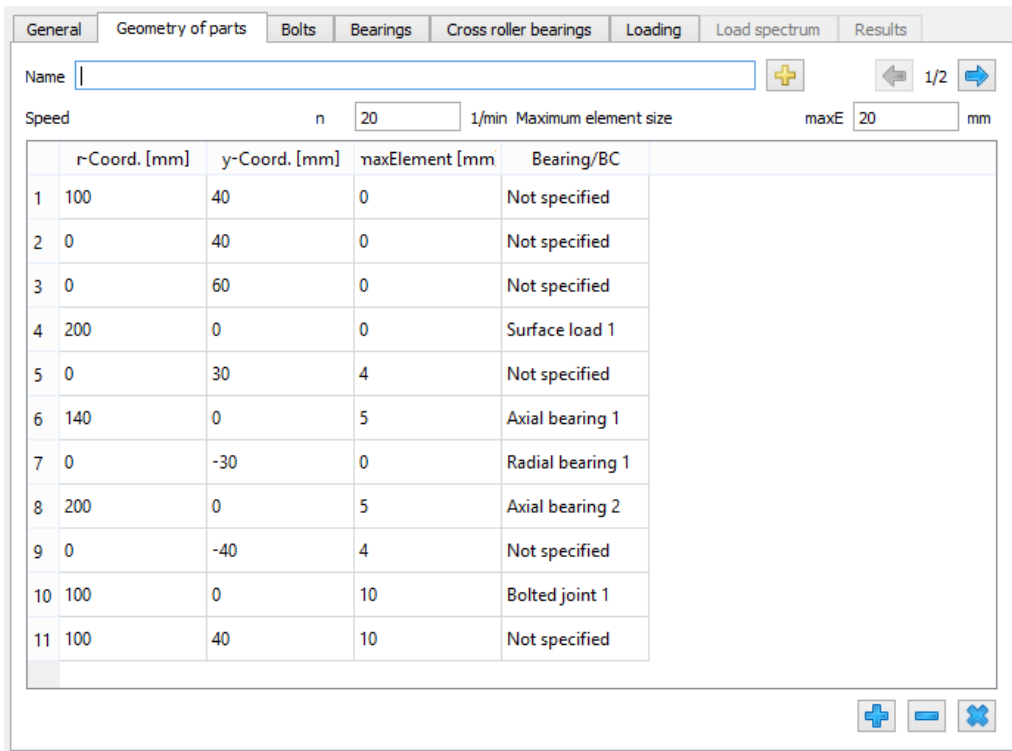
The geometry of parts is defined in a table. The points (radius and y-coordinate) have to be entered clockwise.

If the general geometry input is not activated, the geometry can be defined using horizontal and vertical lines. A coordinate which is not changed to the previous point can be set to zero. So the number of input values can be reduced by factor two.




If the general geometry input is activated, the geometry can be defined by lines in arbitrary direction. All coordinate values have to be defined in this case.

The -button behind the name of the part allows to define the material properties for this part. The material data is only used for the FEA calculation of this part, not for the stiffness of the contact between roller and races.

Using the button on the lower right additional lines can be added or deleted. The -button copies the selected line or appends a new line if nothing is selected. The -button deletes the selected line and the -button clears the whole table.






General Geometry of parts Bolts Bearings Cross roller bearings Loading Load spectrum Results

Name   1/2 

Speed n 1/min Maximum element size maxE mm

	r-Coord. [mm]	y-Coord. [mm]	maxElement [mm]	Bearing/BC
1	100	40	0	Not specified
2	0	40	0	Not specified
3	0	60	0	Not specified
4	200	0	0	Surface load 1
5	0	30	4	Not specified
6	140	0	5	Axial bearing 1
7	0	-30	0	Radial bearing 1
8	200	0	5	Axial bearing 2
9	0	-40	4	Not specified
10	100	0	10	Bolted joint 1
11	100	40	10	Not specified

6.2.2.1 Element size

The size of elements can be defined as a maximal size per part or in the third column of the table for a single line. An input of zero leads to the standard value.

If bending is expected in a part, more elements are required. One element can model a constant stress state correctly. For a linear stress distribution, several elements are necessary.

6.2.2.2 Speed

The rotation speed for the bearing life calculation can be defined for each part. The sign is considered, so one part can have a positive and one part can have a negative speed.

6.2.2.3 Boundary conditions

Boundary conditions can be defined line by line. The following selections are available:

- Radial fixed: All nodes for this surface are fixed radially.
- Axial fixed: All nodes for this surface are fixed axially.
- Radial and axial fixed: All nodes for this surface are fixed radially and axially.
- Bolted joint: The head of the bolts is on this plane.
- Radial bearing: A radial bearing is connected to this plane .
- Axial bearing: An axial bearing is connected to this plane. This input is used for axial positioning.
- Cross roller bearing: A cross roller bearing is connected to this surface.
- Surface load: A load is applied to this surface.
- Contact: A contact to a second part or to the rigid environment can be considered.

6.2.3 Bolted joint

The data for the bolted joint can be defined on this page.

General	Geometry of parts	Bolts	Bearings	Cross roller bearings	Loading	Load spectrum	Results
Bolts							
← 1/1 →							
Number of bolts	Z	<input type="text" value="36"/>	Pitch diameter	D	<input type="text" value="240"/>	mm	
Nominal diameter	d	<input type="text" value="8"/>	mm Bolt length	L	<input type="text" value="80"/>	mm	
Bore diameter	dh	<input type="text" value="9"/>	mm Depth of bore	H	<input type="text" value="80"/>	mm	
Diameter of counterbore	ds	<input type="text" value="15"/>	mm Depth of counterbore	HS	<input type="text" value="10"/>	mm	
Assembly preload	F	<input type="text" value="20"/>	kN <input checked="" type="checkbox"/> Shank length	Ls	<input type="text" value="30"/>	mm	
Tightening torque	MA	<input type="text" value="0"/>	Nm Amount of embedding	fz	<input type="text" value="8"/>	µm	
Effective diameter of bolt head area	Dkm	<input type="text" value="0"/>	mm Pitch of the thread	P	<input type="text" value="1.25"/>	mm	
Friction coefficient head	µk	<input type="text" value="0.1"/>	Friction coefficient thread	µe	<input type="text" value="0.1"/>		

The bore for the bolts and the thread are in the same part. The depth of the bore is measured from the surface; the thread in the part is starting below.

If no counterbore exists, the diameter and the depth of the counterbore can be set to zero. The cylinder with the diameter of the counterbore may not intersect and edges of the part. If necessary, the diameter has to be reduced for the calculation.

The pretension is defined per bolt. All bolts have the same pretension. Either the effective pretension can be defined or a mounting torque and a subsidence value. According to VDI 2230, the subsidence value for surface roughness between $10 < Rz < 40$ is $3\mu\text{m}$ for the head and the thread and $2\mu\text{m}$ for each layer. For most cases, it will be $f_z = 3+3+2 = 8\mu\text{m}$.

The pitch of the thread is needed for the calculation of the core diameter of the bolt needed for its stiffness. Also, it is needed for the calculation of pretension force from mounting torque.

If the mounting torque is entered, the mean diameter of head support and the friction values for thread and head support are required.

6.2.4 Bearings

The geometry of the bearings can be defined on this input page.

The screenshot shows a software interface with several tabs: General, Geometry of parts, Bolts, Bearings, Cross roller bearings, Loading, Load spectrum, and Results. The 'Bearings' tab is active and contains two sections: 'Radial bearings' and 'Axial bearings'.

Radial bearings section:

- Number of rolling elements: Z = 70
- Diameter of rolling elements: Dw = 10 mm
- Pitch diameter: Dpw = 290 mm
- Effective length of roller: Lwe = 30 mm
- Radial clearance: Pd = 0 mm
- Axial position: pa = 0 mm
- Type of profile: Profile according ISO/TS 16281
- Surface hardness: HRC = 58
- Dynamic load number: Cr = 304.491 kN
- Static load number: C0 = 892.138 kN
- Fatigue load limit: Cu = 79.0494 kN

Axial bearings section:

- Number of rolling elements: Z = 70
- Diameter of rolling elements: Dw = 10 mm
- Pitch diameter: Dpw = 350 mm
- Effective length of roller: Lwe = 30 mm
- Axial clearance: Pa = -0.01 mm
- Axial position: pa = 25 mm
- Type of profile: Profile according ISO/TS 16281
- Surface hardness: HRC = 58
- Dynamic load number: Ca = 537.953 kN
- Static load number: C0 = 4620 kN
- Fatigue load limit: Cu = 386.908 kN

6.2.4.1 Number of rolling elements

The number of rollers has to be defined here.

6.2.4.2 Diameter of rollers

The diameter of the rollers is defined here.

6.2.4.3 Pitch diameter

The pitch diameter through the centers of the rollers has to be defined for both bearing types. For radial bearings a warning is shown if the pitch diameter is not between the selected planes.

6.2.4.4 Effective length of rollers

The effective length of rollers L_{we} is the length which is the basis of the calculation of the load capacity. It is the total length of the roller reduced by its radii.

6.2.4.5 Radial-/axial clearance

The clearance is not defined by the dimensions of the parts, so that there is one input for clearance and it is not needed to define the geometry with many digits. The clearance is the nominal clearance before tightening the bolts.

A pretension is defined as negative clearance.

6.2.4.6 Axial position

The axial position of the center of the roller is entered here. For axial bearings the axial position is determined by the position of the selected planes of the parts.

6.2.4.7 Crowning

There are three selections for the crowning of the rollers:

Profile according to ISO/TS 16281

The default profile according ISO/TS 16281 can be selected. This is a logarithmic profile. No additional inputs are necessary.

Logarithmic profile

For the logarithmic profile, the length of the unmodified cylinder and the amount of crowning has to be defined.

The profile is calculated as:

$$P(x) = Pr \cdot \ln \left(\frac{1}{1 - \left\{ \frac{2|x| - L_{wu}}{L_{we} - L_{wu}} \right\}^2} \right)$$

Using $Pr = 0.0005 \cdot Dw$ and $Lwu = Lwe - 2.5 \cdot Dw$ leads to the default profile according ISO/TS 16281 for long rollers.

Crowning

The crowning is an arc-shaped profile. Here the length of the unmodified part of the cylinder and the amount of profile relief has to be defined, too.

6.2.4.8 Surface hardness

The surface hardness is used to reduce the load capacity of the bearings according to NREL guideline DG03. Instead a user input of load capacities can be used to consider the hardness. Only a hardness less than 58 HRC leads to a change of load capacity.

6.2.4.9 Dynamic load capacity

The dynamic load capacity is calculated according ISO 281. Its value can be overwritten by the user. The hardness is considered as follows for values less than 58HRC:

$$C' = C \left(\frac{HRC}{58} \right)^{3.6}$$

If the load capacity is entered by the user, no further reduction by hardness is considered.

6.2.4.10 Static load capacity

The static load capacity is calculated according ISO 76. Its value can be overwritten by the user. The load capacity is multiplied by the following value f_s for reduced hardness:

$$f_s = 2 \left(\frac{HV}{800} \right)^2$$

If the load capacity is entered by the user, no further reduction by hardness is considered.

6.2.4.11 Fatigue load

The fatigue load is calculated according ISO 281 appendix B.3.3.3. The value can be overwritten by the user.

6.2.5 Cross roller bearings

The screenshot shows the 'Cross roller bearings' configuration window with the following parameters and values:

Parameter	Value	Unit	Additional Info
Number of rolling elements (Z)	0		Navigation: 1/1
Diameter of rolling elements (Dw)	0	mm	Surface hardness: HRC 58
Pitch diameter (Dpw)	0	mm	Dynamic load number: Cr 0 kN
Effective length of roller (Lwe)	0	mm	Static load number: C0r 0 kN
Radial clearance (Pd)	0	mm	Fatigue load limit: Cur 0 kN
Axial position (pa)	0	mm	Type of profile: Profile according ISO/TS 16281
Configuration	Cross roller bearing		
Calculated as	Radial bearing		
Contact angle (α)	0	°	

For cross roller bearings, three additional inputs compared to the other bearing types are necessary:

Configuration

In addition to cross roller bearings also a calculation using angular roller bearings is available. All rollers have the same orientation in this case.

The number of rollers is the total number in each case. For cross roller bearings, it has to be an even number.

Calculate as

The bearing can be calculated as axial or radial bearing. Dependent on the selection, either C_a , C_{0a} , P_a or C_r , C_{0r} , P_d are shown as inputs.

Contact angle

The contact angle can be chosen between 40° - 50° . Standard is 45° . For angular roller bearings, the permissible range is 20° - 70° .

6.2.6 Loading

The load can be defined as point or surface load. Surface loads should be preferred because of smoother load introduction. Point loads can only be set inside of a part.

The screenshot shows the 'Loading' tab in the MESYS software. It is divided into two main sections: 'Surface loads' and 'Point loads'.

Surface loads:

- Force in x-direction: Fx = 0 kN
- Force in y-direction: Fy = -200 kN
- Force in z-direction: Fz = 0 kN
- Moment around x-axis: Mx = 0 Nm
- Moment around z-axis: Mz = 0 Nm
- A checkbox labeled 'is varied in series' is checked.

Point loads:

A table with the following columns: x-Pos. [mm], y-Pos. [mm], z-Pos. [mm], Fx [kN], Fy [kN], Fz [kN]. The table is currently empty.

6.2.6.1 Surface loads

Surface loads can be defined by an input of three force components and two moment components. A torque around the y-axis cannot be taken by the bearing, therefore it is not available as input.

For each load, a selection can be made if the load should be multiplied by a load factor in a calculation for series. If not selected, the load is considered constant for all load steps.

6.2.6.2 Point loads

Point loads are defined by a force vector with three components and a position. The position has to be inside of a part.

For point loads it can be defined, too if the load should be multiplied with a load factor in calculation for series or taken as constant.

The point load is introduced using several nodes in the neighborhood of the given position, so the load center is on the selected position.

6.2.7 Load spectrum

If the load spectrum calculation was activated, it can be defined on page "Load spectrum". Only loads are shown where the flag "is varied in series" is set.

	Frequency	P1-n [rpm]	P2-n [rpm]	SL1-Fx [kN]	SL1-Fy [kN]	SL1-Fz [kN]	SL1-Mx [Nm]	SL1-Mz [Nm]
1	0.5	20	0	0	-200	0	0	0
2	0.5	30	0	100	-300	0	0	0

6.3 Results

General	Geometry of parts	Bolts	Bearings	Cross roller bearings	Loading	Load spectrum	Results
<p>Results</p> <p>Clear results</p> <p>Current result 3/3</p> <p>Factor for displacements: 200</p> <p>Angle for graphic deformation: α 0 °</p> <p>Minimal stress for graphics: σ_{min} -200 MPa</p> <p>Maximal stress for graphics: σ_{max} 200 MPa</p> <p>Calculation of stiffness</p> <p>Radius point 1: r1 20 mm</p> <p>Radius point 2: r2 230 mm</p> <p>Axial position point 1: y1 60 mm</p> <p>Axial position point 2: y2 -20 mm</p> <p><input type="checkbox"/> generate short report</p>							

If the results page is active, evaluation of results is possible without a new FEA calculation. Pressing the calculate button just evaluates the results.

6.3.1 Factor for displacements

The factor for displacements defines the exaggeration of deformations used in graphics. It is used in 3D view and 2D view for deformation of a section.

6.3.1.1 Angle for deformation graphics

The angle for the section used in 2d view for deformations can be selected here. The same angle is used in the diagrams for stresses within the parts.

6.3.1.2 Minimal and maximal stress for graphics

For the stress diagrams, the range for the color scale can be set here.

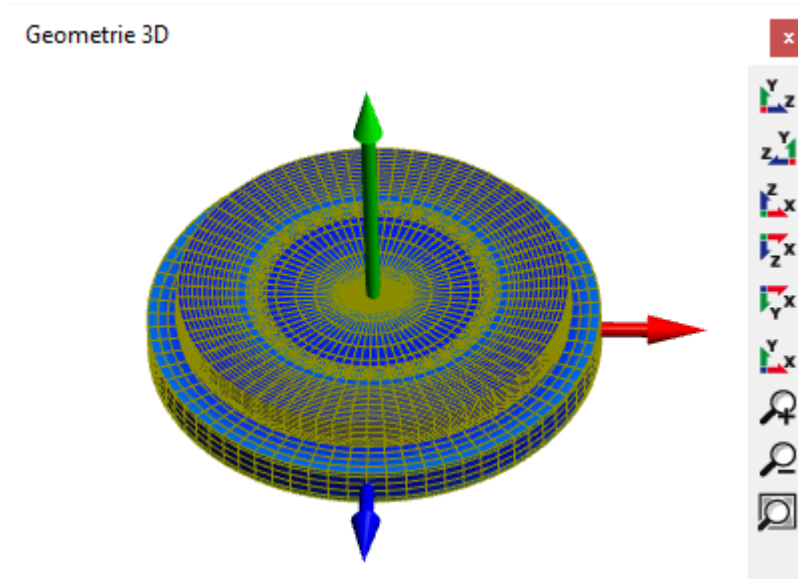
6.3.1.3 Calculation of stiffness

For the calculation of stiffness and the diagrams for deflection and tilting angle, two points have to be defined. The software averages the deflections around the circumference for these points and the result is used to calculate a stiffness value.

6.3.2 Graphics

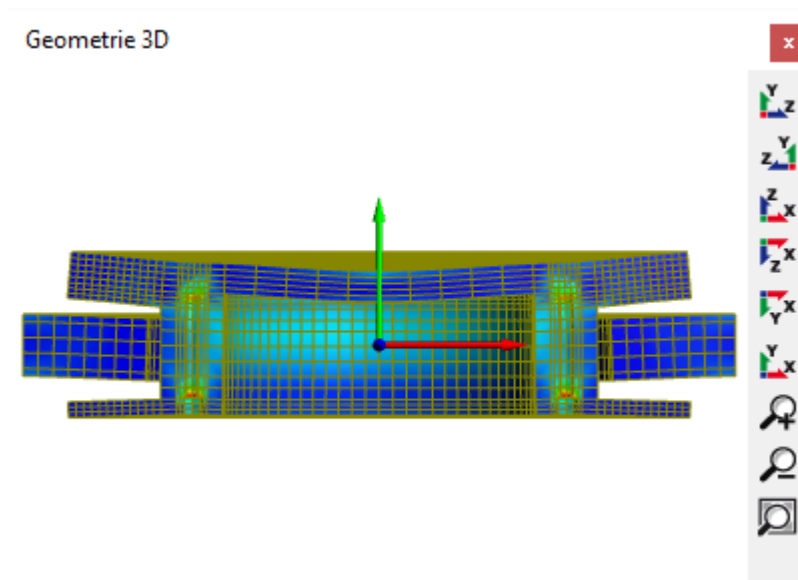
6.3.2.1 3D-View

The 3D-view can be rotated using the left mouse button. Using the Ctrl-button, the size can be changed.



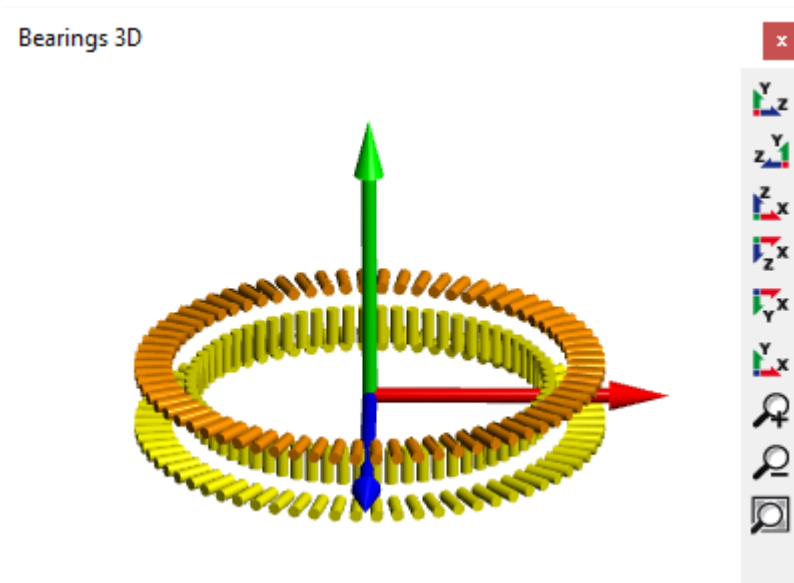
6.3.2.2 Geometry 3D Cut view

The cut view is a cut of the 3D geometry using the angle defined on the result page.



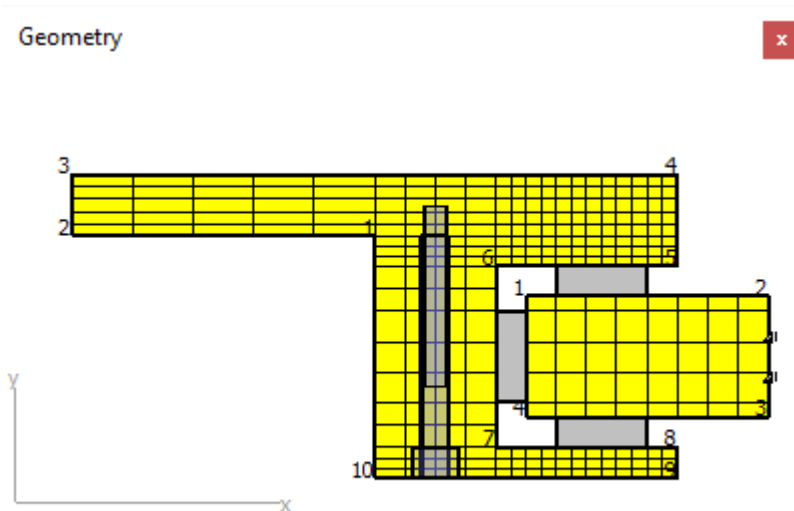
6.3.2.3 Bearings 3D

The load distribution can be shown on the 3D bearing set.



6.3.2.4 Geometry

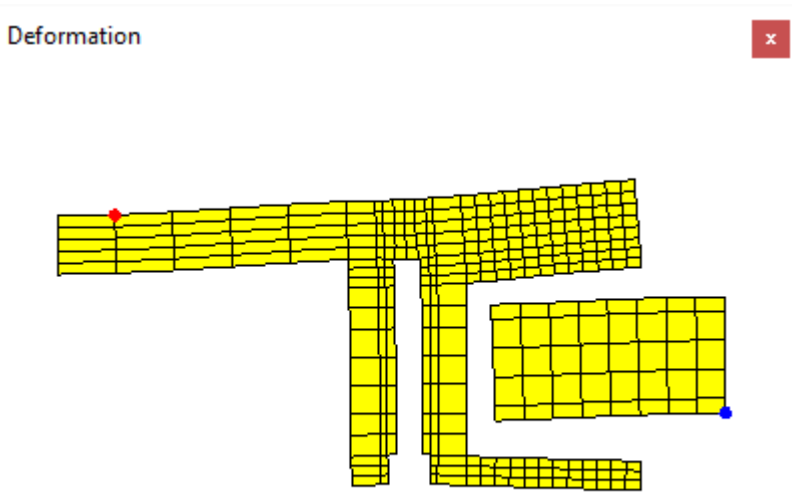
The graphics 'Geometry' is available after preprocessing. It can be used to check meshing and the position of bolts and bearing rows.



6.3.2.5 Deformation

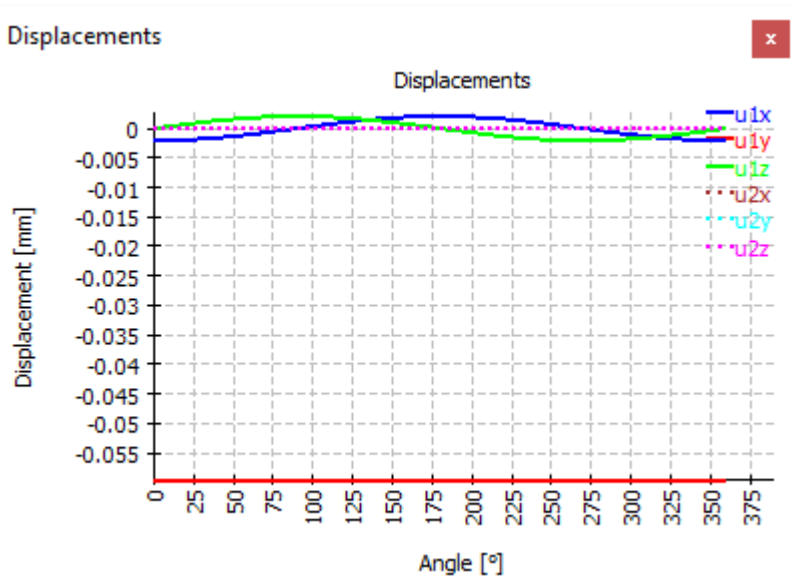
The graphics 'Deformation' shows the deformed mesh for a given angle. The two points for evaluating the displacements are marked with color.

The first point is shown in red, the second in blue.



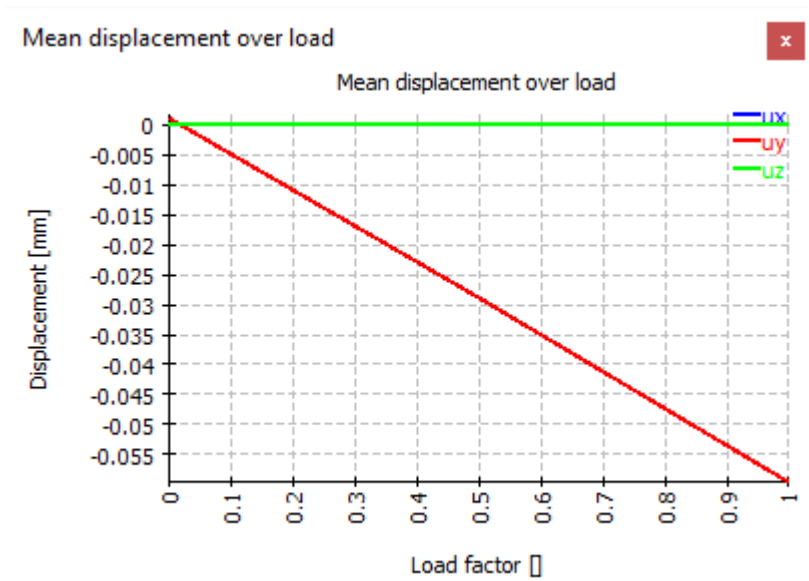
6.3.2.6 Displacements

The displacements for the selected points are shown over the circumference.



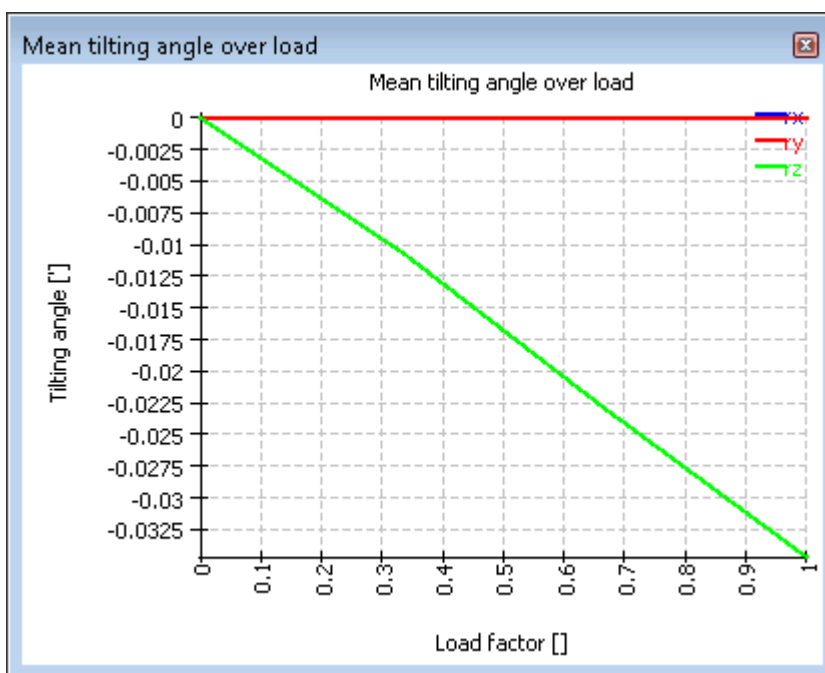
6.3.2.7 Displacements over load factor

In case of a series calculation, the mean displacements are shown over load factor.



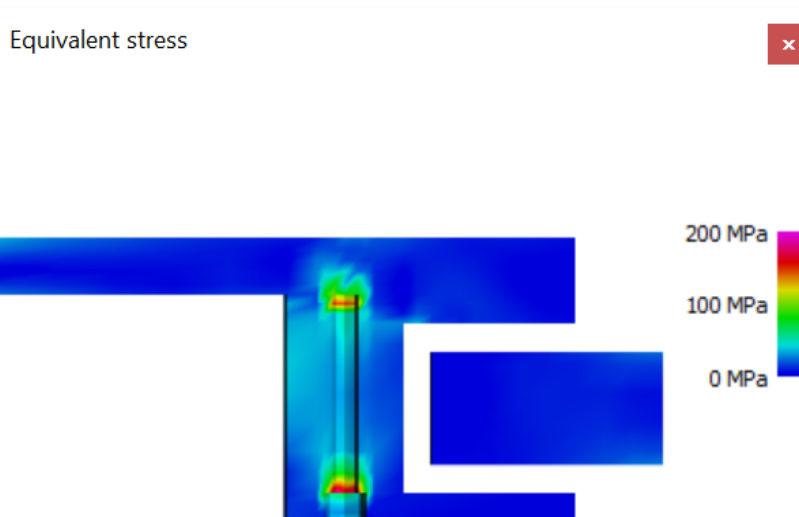
6.3.2.8 Tilting angle over load factor

Like the mean displacement, also the mean tilting angle can be shown dependent on the load factor in a diagram.



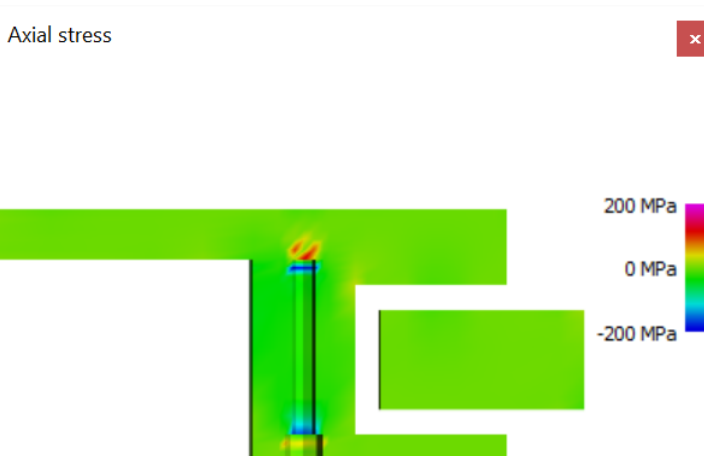
6.3.2.9 Equivalent stress in parts

The equivalent stress in part is shown in this diagram.



6.3.2.10 Axial, radial and tangential stress in parts

Further diagrams show the axial, radial and tangential stress in the parts. The range for the color scale can be defined on the results page.

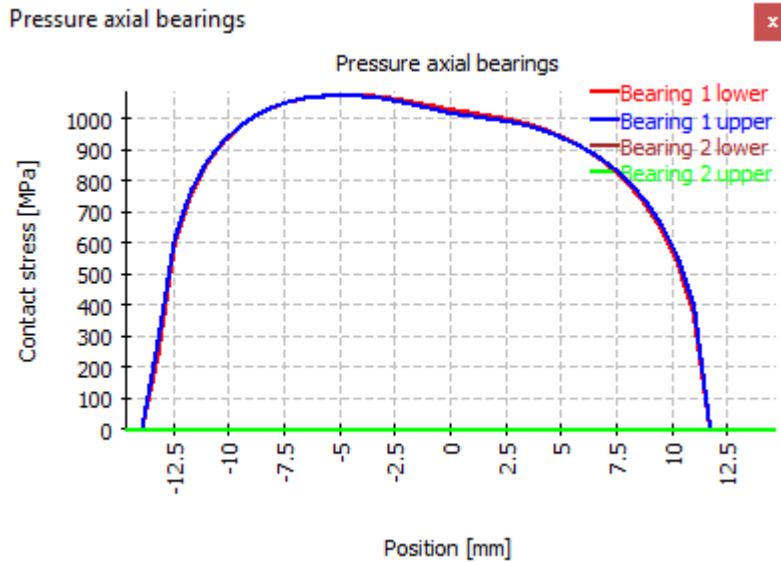


6.3.2.11 Contact stress for bearing

The contact stress is shown over the length of the rollers. In the user interface, there is one graphic for all bearings of one type, in the report, one diagram is shown for each bearing.

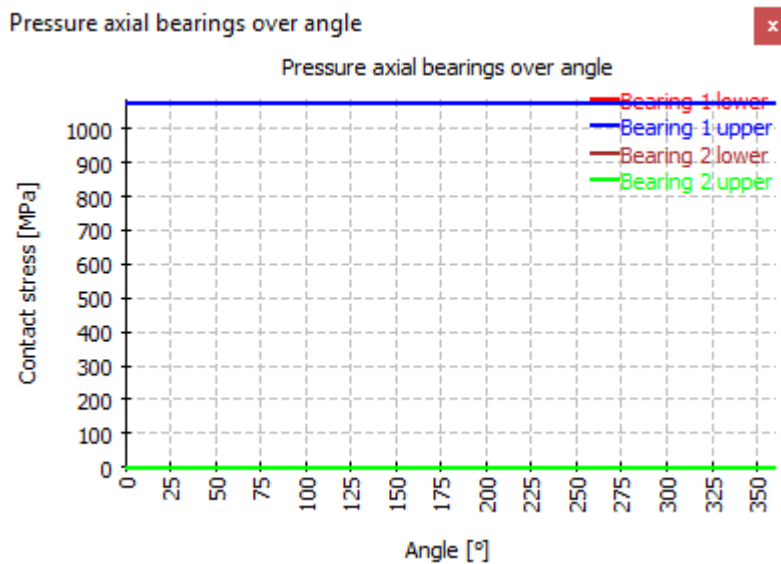
The position in the diagram is either the axial or radial coordinate with the origin at the center of the roller. For radial bearings, the orientation is like the

orientation of the y-axis. The position is running from bottom to the top of the bearing. For axial bearings, the orientation is the radius; the position is running from inside to outside.



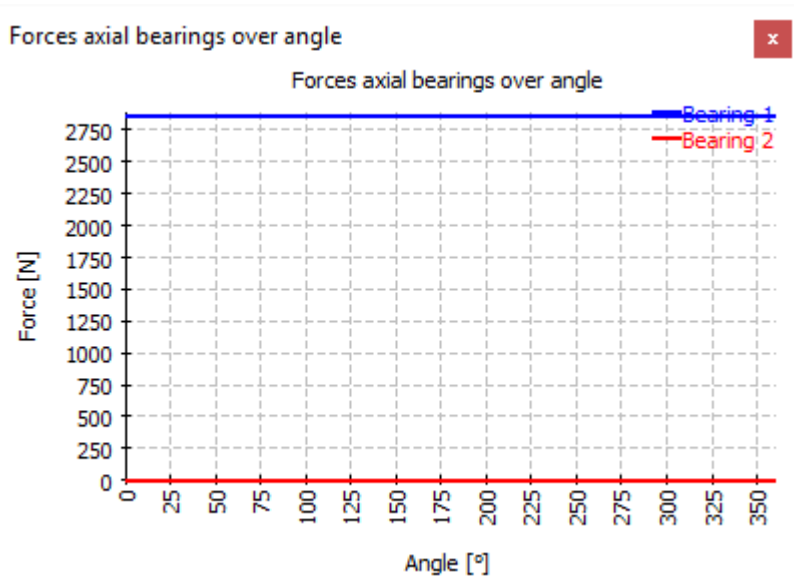
6.3.2.12 Contact stress over angle

The contact stress can also be shown over the angular position. The maximal contact stress per roller is shown. Angle zero is in the x-axis, 270° on the z-axis.



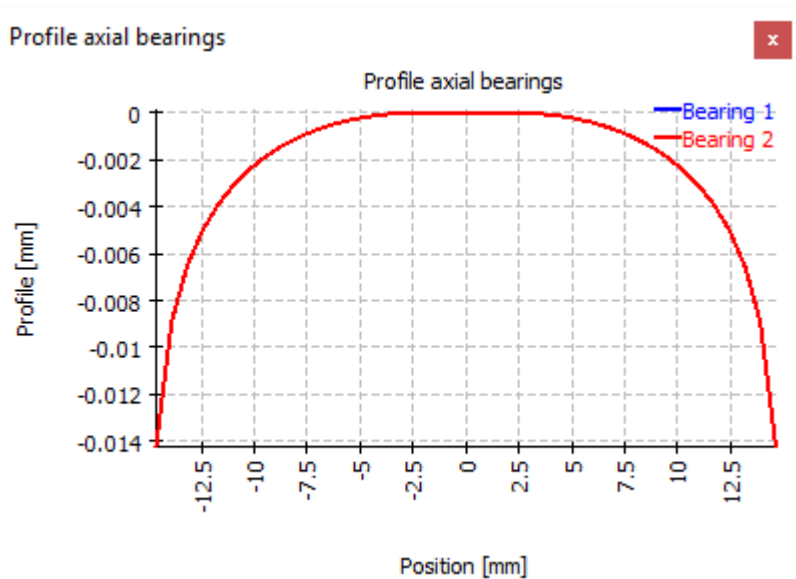
6.3.2.13 Forces over angle

The distribution of roller forces over angle is shown in this diagram. For force is the total force between roller and race.



6.3.2.14 Roller profile

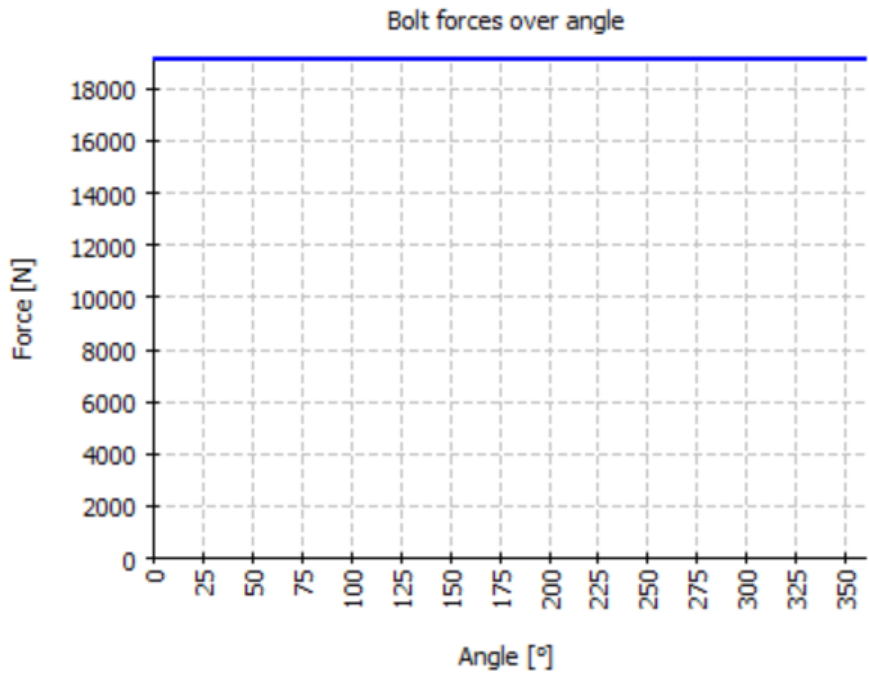
The profiling of the rollers is shown in this diagram. It is mainly used for check of inputs.



6.3.2.15 Bolt forces over angle

This diagram shows the bolt forces over the angle.

Bolt forces over angle



MESYS Gear Positions

7 MESYS Gear Positions

MESYS Gear Positions (Version 07/2019) calculates the positioning of the gears in a system that must meet specific criteria related to gear design data and geometric constraints between the gears.

The following results can be found in the report:

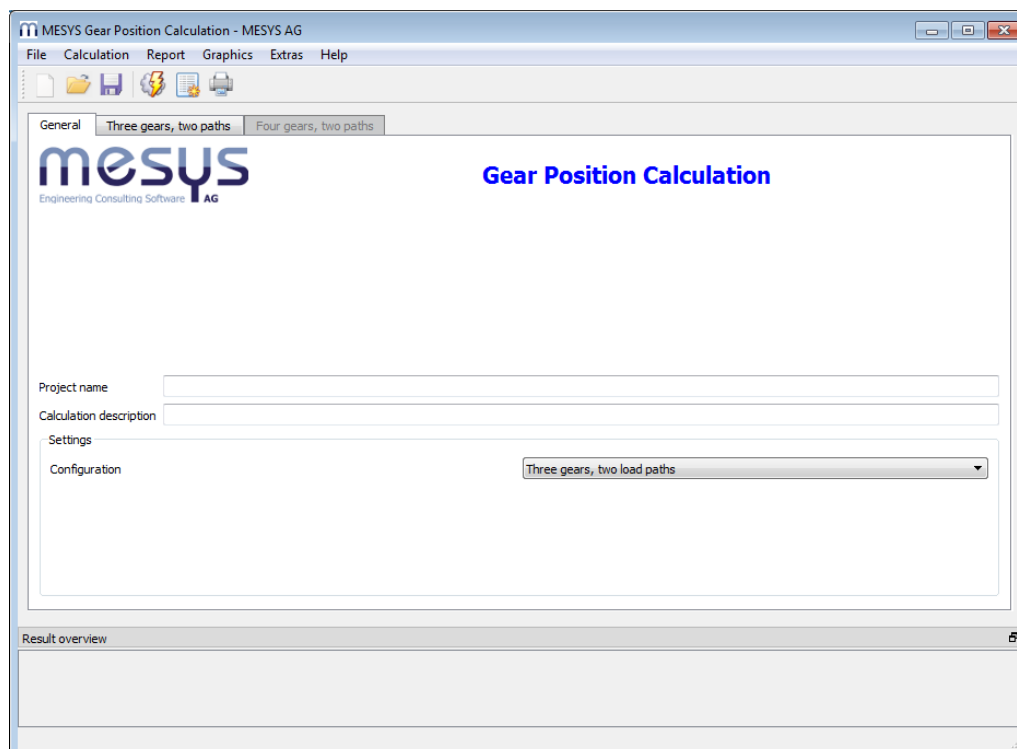
- Center coordinates of all gears of the system
- Graphical representations of the feasible gear configurations

In addition to the report, the results are shown in several graphics. Report and user interface are both available in metric or US customary units. Supported languages for user interface and report are English, German, French, Spanish and Korean.

7.1 Input Parameters

The input parameters are all shown at three different tab pages.

7.1.1 General



Project name and calculation description

The project name and the calculation description are just inputs which are shown in the report header. They can be used to enter information about the purpose of the calculation for documentation.

Settings

Two configurations are available:

- Three gears, two load paths
- Four gears, two load paths

7.1.2 Three gears, two paths

The screenshot displays the 'MESYS Gear Position Calculation - MESYS AG' software window. The interface includes a menu bar (File, Calculation, Report, Graphics, Extras, Help) and a toolbar with icons for file operations. The main area is divided into tabs: 'General', 'Three gears, two paths' (selected), and 'Four gears, two paths'. The 'Three gears, two paths' tab contains the following input fields:

System data			
Normal module	m_n	1	mm
Normal pressure angle	α_n	20	°
Helix angle	β	0	°
Basic rack addendum	haP	1	
Required clearance	c	0	mm

Gear 1			
Number of teeth	z	0	
Profile shift	x	0	

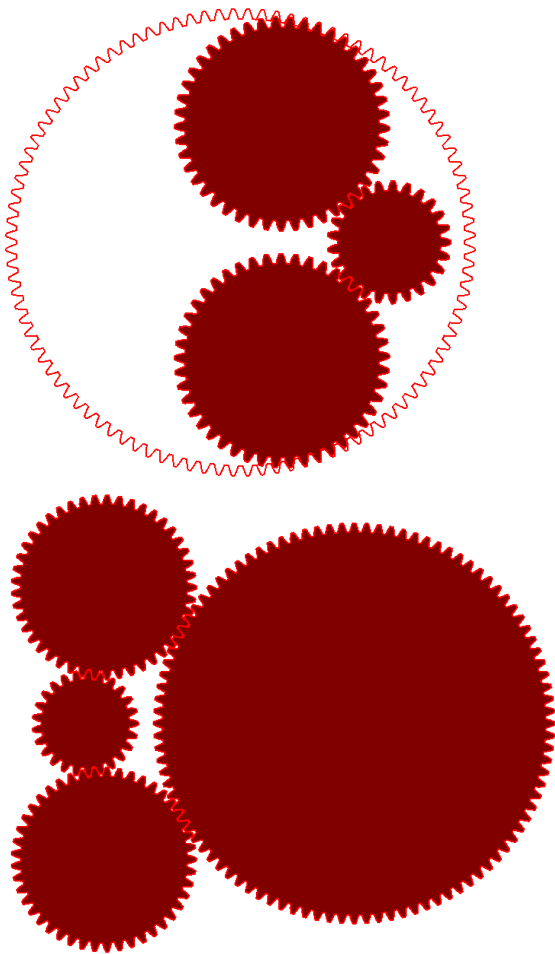
Gear 2			
Number of teeth	z	0	
Profile shift	x	0	

Gear 3			
Number of teeth	z	0	
Profile shift	x	0	

$x1$ [mm]	$y1$ [mm]	$x2$ [mm]	$y2$ [mm]

At the bottom of the window, there is a 'Result overview' section which is currently empty.

This module of the program allows a configuration of a symmetric system with two different paths for transmitting the power composed of three gears each.



7.1.2.1 System data

At this section, common general data and settings for the whole gear system can be introduced, such as:

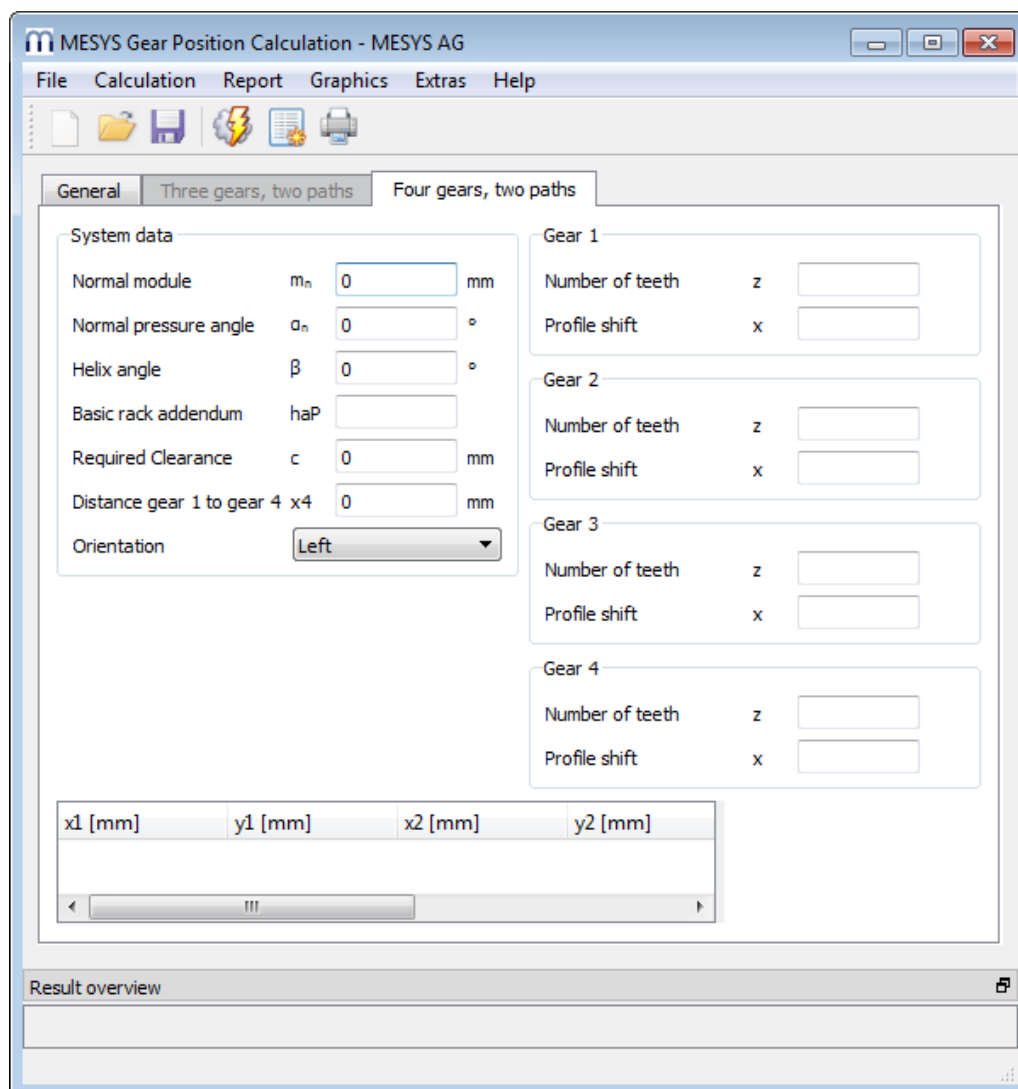
- Normal module
- Normal pressure angle
- Helix angle
- Basic rack addendum: h_{aP} is the coefficient to define the addendum of the standard basic rack tooth profile defined as addendum = $h_{aP} * mn$. This value is normally equal to 1, thus being the addendum = mn .

- Required clearance: The required clearance is a restriction that sets a minimum permissible distance between the tooth tips of unconnected gears to prevent them from colliding with each other.

7.1.2.2 Gears

For each of the three gears, the number of teeth and profile shift can be specified as design parameters. A positive value for the teeth will be understood as an external gear by default, thus being negative values used for modeling internal gear teeth. Due to assembly reasons, only the third gear can accept negative values for the teeth.

7.1.3 Four gears, two paths

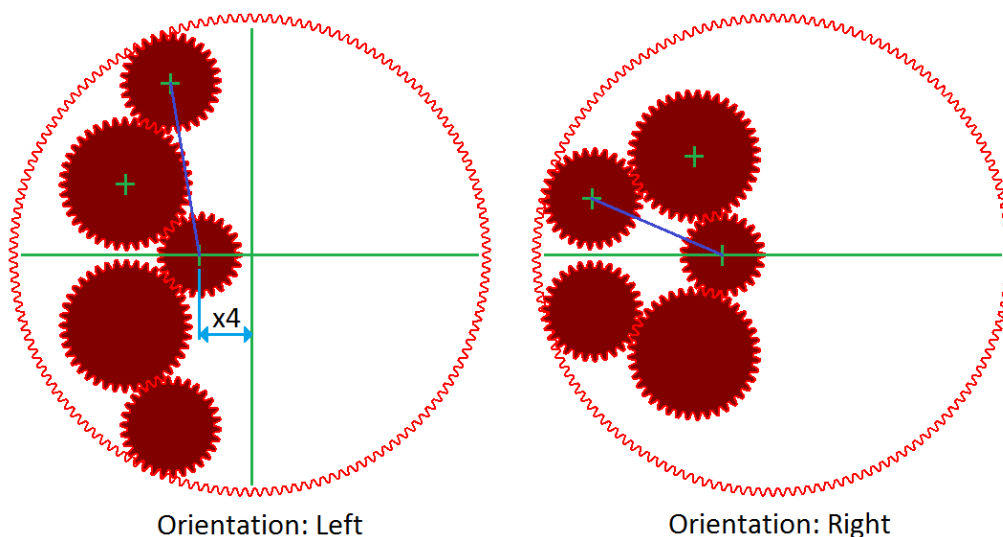


This second module of the program allows a configuration of a symmetric system with two different paths for transmitting the power composed of four gears each.

7.1.3.1 System data

As for before at this section, common general data and settings for the whole gear system can be introduced, such as:

- Normal module
- Normal pressure angle
- Helix angle
- Basic rack addendum: h_{aP} is the coefficient to define the addendum of the standard basic rack tooth profile defined as $\text{addendum} = h_{aP} * mn$. This value is normally equal to 1, thus being the addendum = mn .
- Required clearance: The required clearance is a restriction that sets a minimum permissible distance between the tooth tips of unconnected gears to prevent them from colliding with each other.
- Distance gear 1 to gear 4: This distance is defined as the distance between the gear centers of the gear 1 and gear 4. Positive values will drag the center of gear 4 rightwards with respect to the center of gear 1.
- Orientation: The orientation can be selected as 'Right' or 'Left', which will constrain the possible results to those configurations where the 2nd gear is restricted to maintaining its position either on the left or the right side of the center line between the 1st and 3rd gear, as shown below:



7.2 Results

Results are provided as a small result overview directly in the software, a main PDF text report, a tolerances report and separate graphic windows. Graphic windows

can be docked onto the main program interface and are automatically updated on each calculation.

x1 [mm]	y1 [mm]	x2 [mm]	y2 [mm]	x3 [mm]	y3 [mm]	x4 [mm]	y4 [mm]	cMin [mm]
0	0	-17,1074	14,6146	3,96374	39,3585	20	0	2,22924
0	0	-15,3923	16,4111	6,60614	40,3343	20	0	5,82229
0	0	-13,4067	18,0696	9,45352	41,1706	20	0	9,13916
0	0	-11,1271	19,556	12,5316	41,8387	20	0	11,2395
0	0	-8,51245	20,8276	15,8838	42,3002	20	0	12,6907
0	0	-5,4801	21,8224	19,5951	42,4981	20	0	14,4522
0	0	-1,82268	22,4261	23,8748	42,323	20	0	16,5926
0	0	3,40671	22,2406	29,6693	41,3854	20	0	17,4812
0	0	10,32	19,9937	36,5901	39,1282	20	0	12,9874
0	0	13,1398	18,2646	38,9175	38,0576	20	0	9,5292

Result 1

Result overview

Gear ratio i: 6 Number of results n: 16

Result 5 Result 10 Result 15

- 3 -

3D elastic part as housing 97
 3D elastic parts 91
 3D elastic planet carriers 94

- A -

Accelerations 45
 Add shafts 52
 aISO 127
 Angular roller bearings 157
 Axial angular contact bearing 140
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