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MESYS Rolling Bearing Calculation

Introduction

This rolling bearing calculation (version 1.5) calculates the load distribution, the reference life and the modified reference life according ISO/TS 16281 for the following types of bearings:

- Deep groove radial ball bearings
- Radial angular contact bearings
- Axial angular contact bearings
- Double row 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
- Duplex ball bearings
- Radial cylindrical roller bearings
- Taper roller bearings
- Spherical roller bearings
- Axial cylindrical roller 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 you. 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 on the rolling elements can be considered.

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 Harris: Rolling bearing elements.

The loading can be specified as force/moment or displacement/rotation independently for each of the five degrees of freedom.

The following results can be found in the report:

- Reference life according ISO/TS 16281
- Modified reference life according ISO/TS 16281
- 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

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 and German.

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 and cylindrical roller bearings. Multi row bearings can be considered using bearing configurations.

Installation

When running the installer the installation directory can be selected. The default location is "\Program Files\MesysRBC". 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 and a Demo message is shown on each calculation. 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 MesysRBC.exe). The name of the license file may not be changed since it will not be found by the software.

Requirements

The rolling bearing calculation is available as 32bit windows program running on Windows XP, Vista or Windows 7. In addition to the 32bit version which can be used on 32bit or 64bit operating system also a 64bit version is available. The minimum required processor is Intel Pentium 4 or above.

About 25MB 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.

Input Parameters

The input parameters are shown on five tab pages. To run a calculation first the data on all pages is introduced. Then press or F5 to run the calculation. After all data is defined the calculation can be run from each tab page. So it is easy to make parameter variations.

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

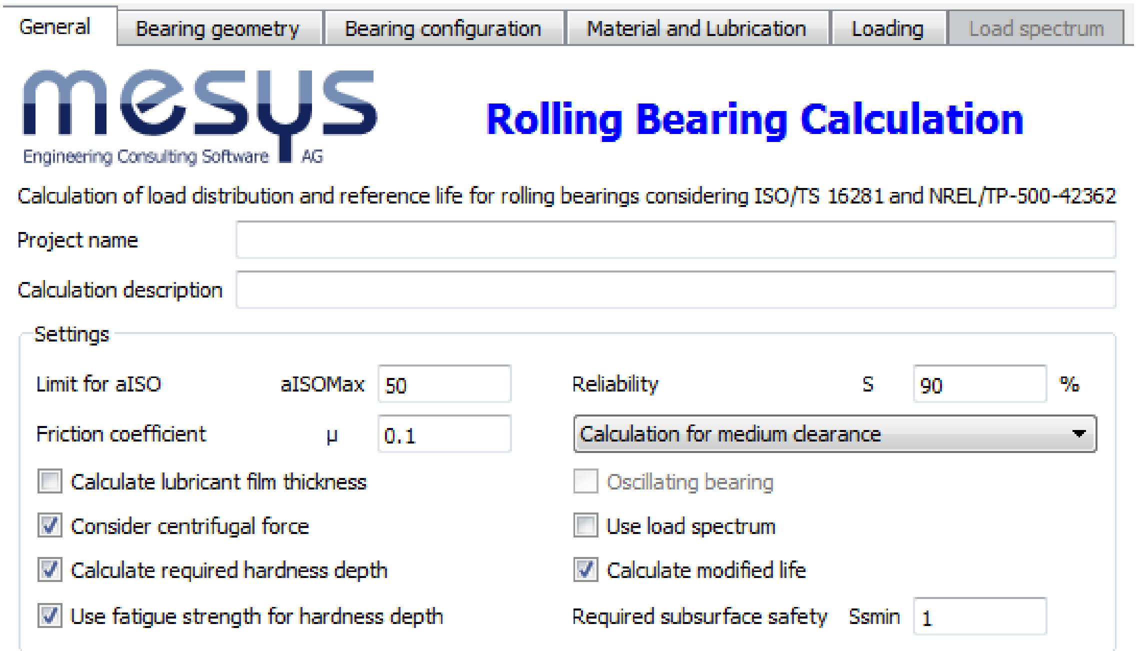
Button	Explanation
	This plus button shows a dialog with additional inputs. Some of these inputs need to be defined, some are just optional.
	This conversion button allows the conversion from other types of input. For example the radial clearance can be converted from an axial clearance
	This proposal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

The software is available in English and German language. The language can also be selected in menu ‘Extras’.

General

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



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.

Limit for a_{ISO}

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 L10r 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.

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.

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.

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, medium or maximum value of the clearance. For other needs the operating clearance can be specified directly as a number.

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.

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.

Consider centrifugal force

The centrifugal force can be considered for ball bearings in this version of the software. It will also be added for 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.

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.

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.

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 $n = \Theta_{osc}/180°*f_{osc}$. 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.

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 shown for the first element of the load spectrum only.

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

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.

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.

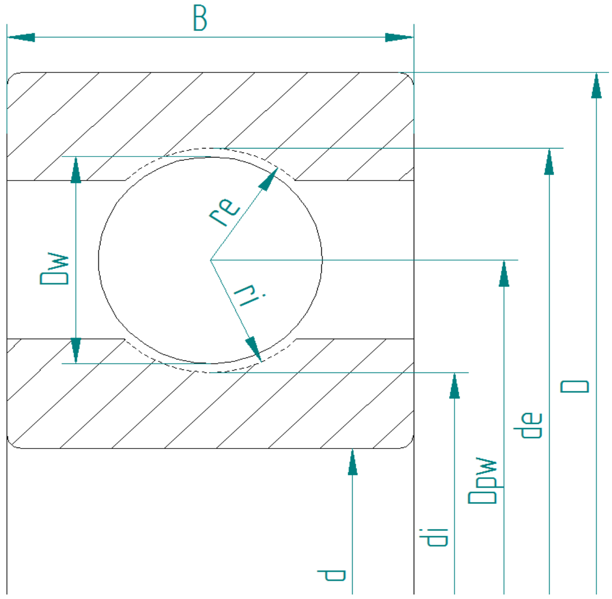
Bearing geometry

Bearing types

Different types of ball and roller bearings can be calculated using the software. In the current version the following types are supported:

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 dependent on the radial clearance in the bearing. The free contact angle $\alpha 0$ is shown in report and result overview.



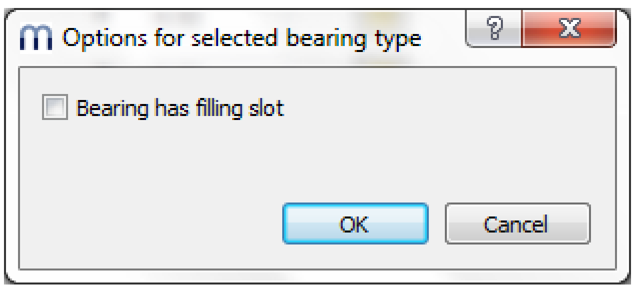
The geometry is described with number and diameter of balls, pitch diameter, conformity of inner and outer race and the diametral clearance. The number of balls in the bearing is restricted, to be able to assemble the bearing.

The conformity is the input parameter defining the radius of the races. ($ri = fi \cdot Dw, re = fe \cdot Dw$)

Usually the conformity is $fi = fe = 0.52$ for deep groove radial ball bearings.

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

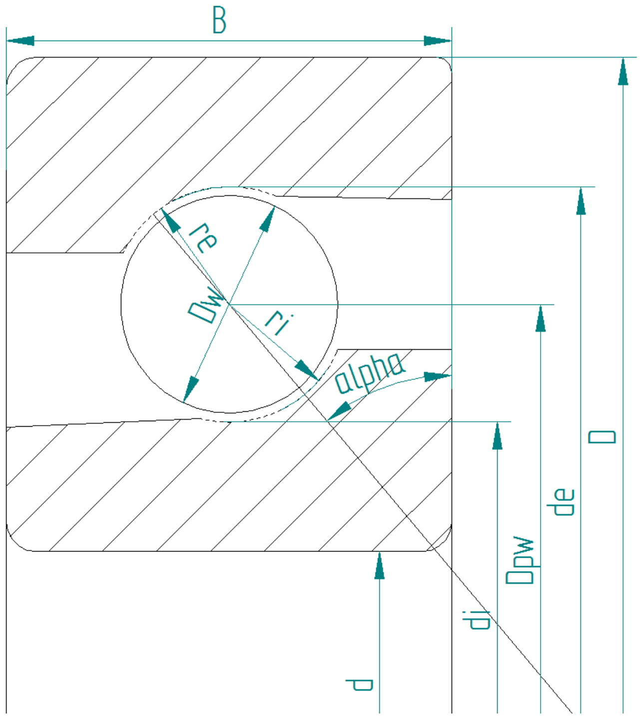
Using the plus button behind the bearing selection, settings depending on the bearing type can be done. For deep groove radial ball bearings it can be selected if a “filling slot” bearings is used, which influences the calculation of the load capacity with factor bm in (ISO 281, 2007).



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 plus button behind the contact angle.

The conversion 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.

Axial angular contact bearing

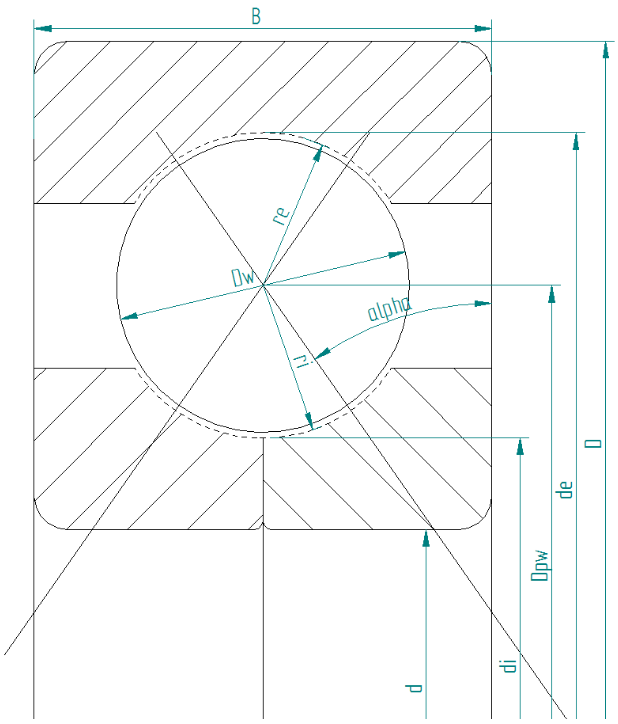
The description of geometry for axial angular contact bearings is the same as for radial angular contact bearings. The only difference in geometry is the default of 0.535 for the conformity is used instead of 0.52 as for radial bearings.

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°.

Four point bearing considered as radial bearings

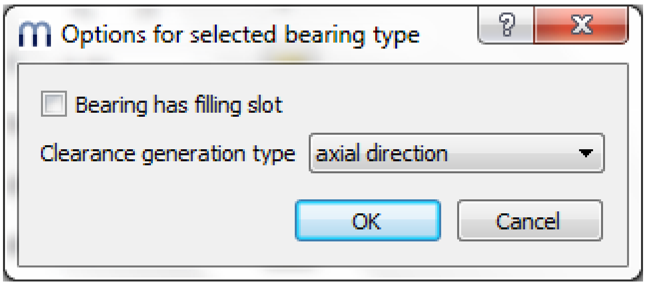
This can be used for standard QJ bearings with 35° contact angle. Load capacity for a four point bearing is calculated as a double row bearing, as usually done in bearing catalogues. The results are reported for both rows. In case of two contact points, there is one contact on the inner ring of one row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



Using the plus 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.



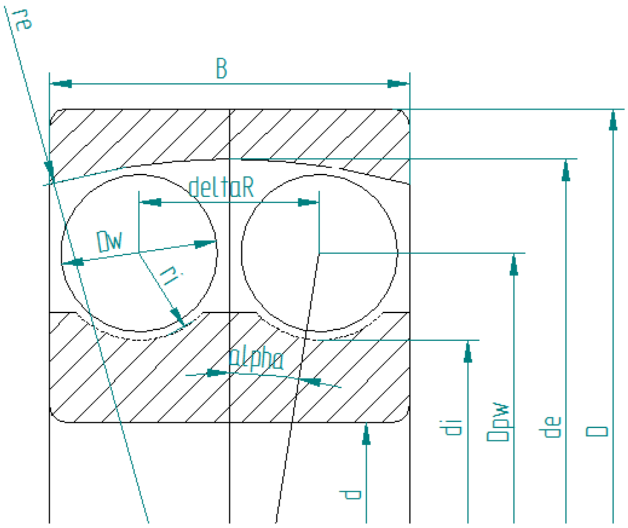
Four point bearings considered as axial bearings

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 capacity is calculated for a single row bearing in this case, but the results are reported for both rows. The load distribution is the same as for the four point bearing considered as radial bearing.

Spherical ball bearings

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

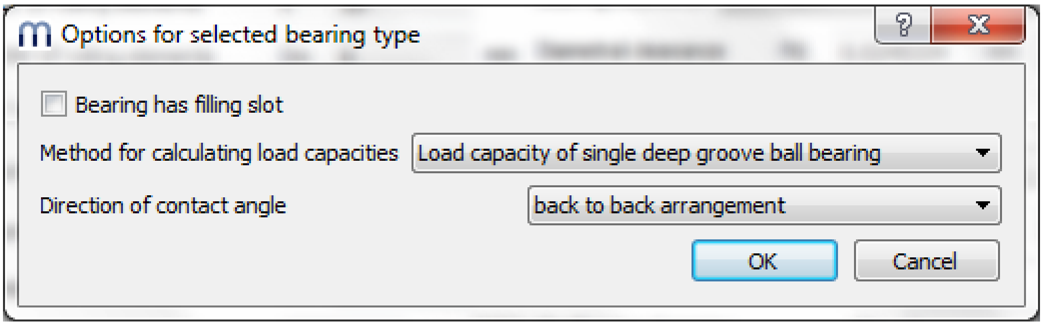
Duplex bearings

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 conversion 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 α_{0eff} 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.

For the calculation of load capacities there are four options:

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.

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. The profile cannot be changed and a profile for the races cannot be defined yet, but the calculation itself is capable of unsymmetrical profiles.

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 outside diameter of the roller.

For cylindrical roller bearings a radial and axial clearance can be specified. It is important to enter a value for axial clearance if tilting occurs. For NU, N, NJ types the axial clearance is large, no axial reactions can occur. For NUP type the axial clearance has influence on the reaction moment.

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 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.

Spherical roller bearings

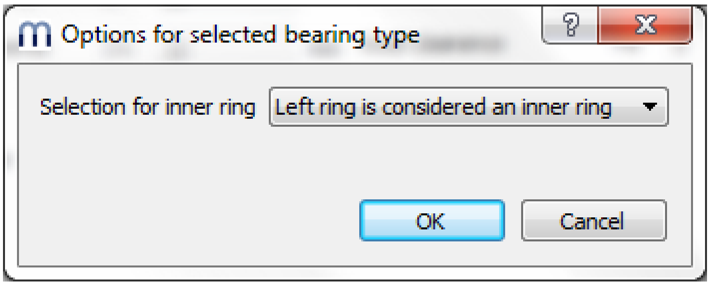
Spherical roller bearings are two row bearings. The distance between both rows is determined the contact angle and the pitch diameter. In contrast to the other bearing types the nominal diametral clearance is only applied to the inner race.

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 any more.

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 uy, uz have to be entered instead of Fy, Fz.

The options dialog (plus 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.

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

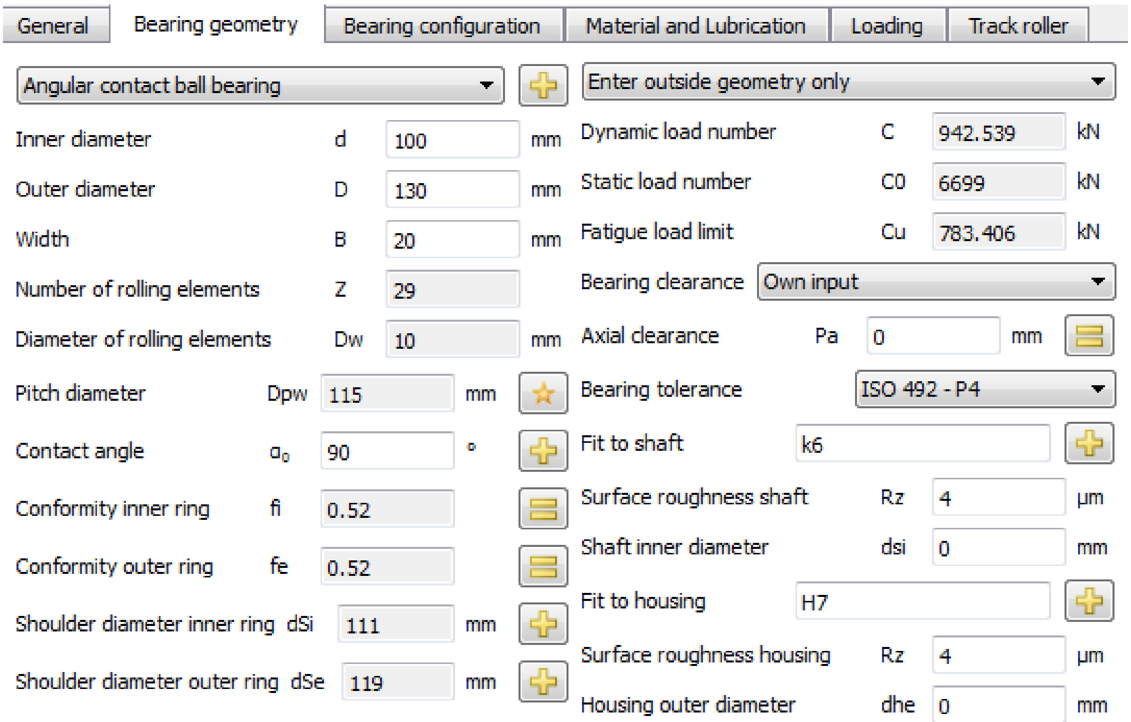
In this case the inner geometry is approximated as before, but the load capacities are provided by the user. The load capacities are usually available in bearing catalogues.

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.



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.

Dynamic load number

The dynamic load number is used for the calculation of bearing life. It is calculated according (ISO 281, 2007) with factor fc according (ISO/TR 1281-1, 2008). The factor bm can be influenced for filling slot bearings using the options dialog for the bearing type (plus button behind bearing type).

Static load number

The static load number is calculated according (ISO 76, 2006) and (ISO/TR 10657, 1991). It is only for documentation and not used in the calculation. It is only used in one case for track roller calculation, see that section for details.

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.3.3.3 for roller bearings.

The fatigue load limit is based on fatigue strength of 1500MPa and it is used for the calculation of modified life.

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.

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.

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.

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.

Contact angle

The contact angle has to be provided for angular contact bearings, four point ball bearings, spherical ball bearings, taper roller bearings ad 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.

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 (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.

For spherical 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.

For spherical ball bearing 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 plus-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.

Conformity of roller

For spherical 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.

Distance between rows

For double row bearings the distance between rows is shown in the user interface. It is the axial distance between the center 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

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

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.

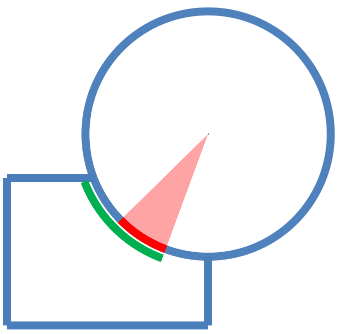
Shoulder diameter of inner and outer ring

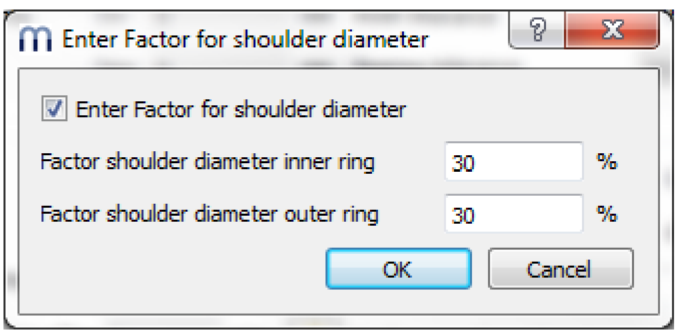
For ball bearings the shoulder diameter of inner and outer ring can be defined. If an input of zero is given it won't be considered.

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 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.

Bearing clearance

The bearing clearance can be set automatically according (ISO 5753, 2009) (C2…C4) for deep groove ball bearings, four point bearings and cylindrical roller bearings.

In addition there are the settings ‘own input as operating clearance’ and ‘own input’.

Own input as operating clearance

Selection ‘own 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.

Own input

Selecting ‘own input’ allows entering the clearance of the bearing before mounting. Influence of temperature or interference fits will be considered additionally.

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. In 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 conversion button allows the input of axial clearance instead of diametral clearance.

Axial clearance

The axial clearance P_a is shown for angular contact bearings, four point bearings and cylindrical roller 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 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.

A pretension can be entered by using a negative value for the axial clearance Pa. A conversion button allows the input of diametral clearance instead of axial clearance. For angular contact bearing axial clearance can also be calculated for a given pretension force.

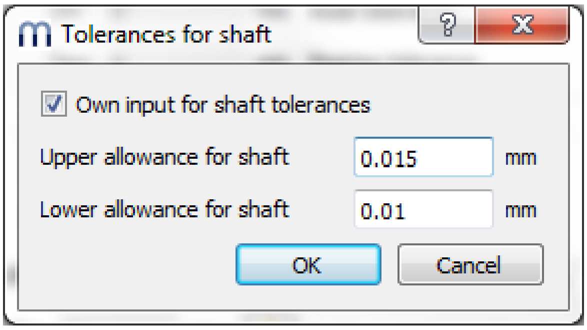
Bearing tolerance

The bearing tolerance can be considered according ISO 492 (P0…P2). If selecting 'Not considered' the interference fit is not considered for operating clearance. 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.

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 plus button next to the input field. Selecting "own input" allows the input of arbitrary tolerances:



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 $2 \cdot D$ will be used in the calculation.

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 $0.8 \cdot Rz$ according (DIN 7190, 2001).

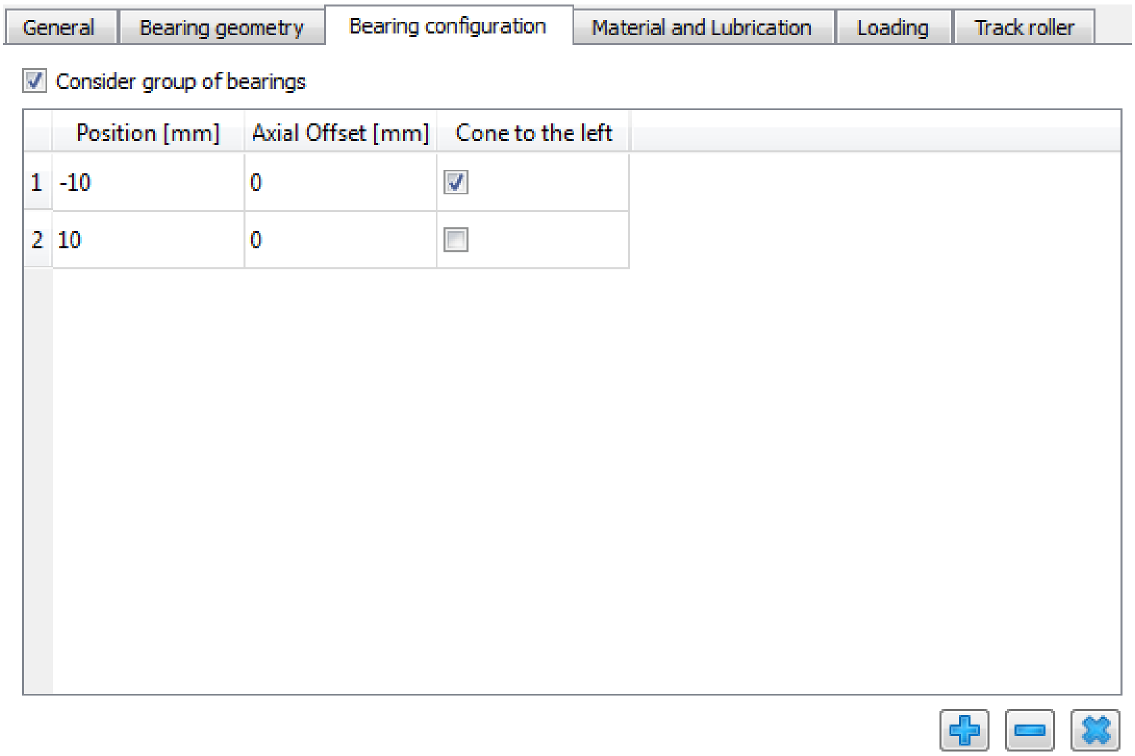
The surface roughness of shaft and housing is only used for the calculation of the interference fit.

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 specifies. In the picture an O-configuration (or back-to-back configuration) is defined.



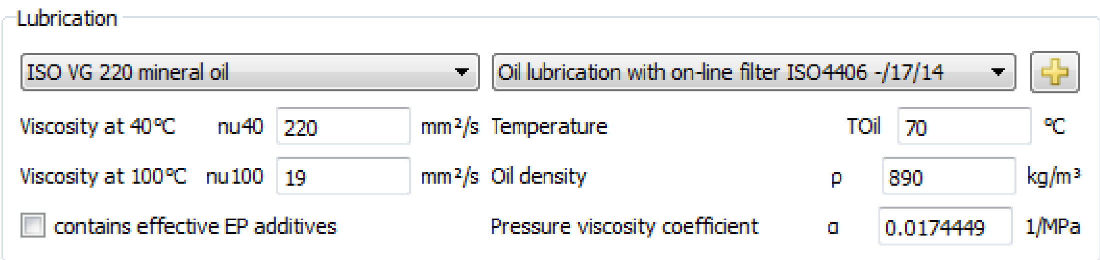
Additional bearings can be added using the '+'-button. A row can be deleted using '-' and the 'X'-button clears all inputs.

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.

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.



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 Plus-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} < 100mm	D_{pw} ≥ 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

Using a value below 0.1, no calculation is possible.

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.

Oil temperature

The oil temperature is only used for the calculation of lubricant properties like operating viscosity. It has no influence on clearance.

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.

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 (ISO/TR 1281-2, 2008) 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 and α in mm^2/s is used according (Baalmann, 1994).

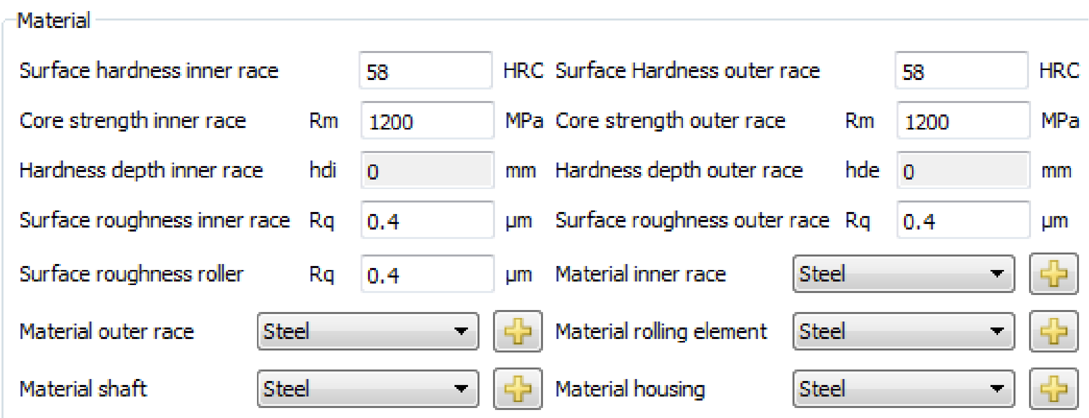
In the software the proposal according (AGMA 925, 2003) is used, because different oil types are considered. The pressure viscosity coefficient is calculated as $\alpha = k \cdot \eta_M^s$, where η_M is the dynamic viscosity at operating temperature. The factors k and s are given for different lubricant 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 only used if a lubricant from the list is selected. If 'own input' is selected, the input value is used for the calculation.

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.



Surface hardness

The surface hardness is given as Rockwell hardness HRC. If the surface hardness is smaller than 58HRC 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.

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

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

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 (Harris, et al., 2009) from the input value of ultimate strength:

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

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.

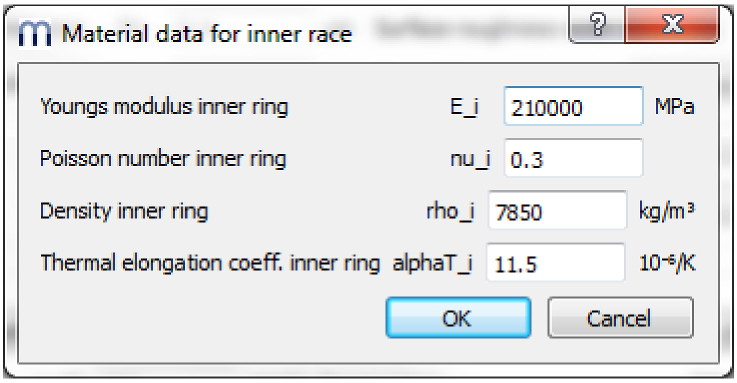
Surface roughness

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 (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 $R_q = 1.2 \cdot 10^{-5} \cdot D_{pw}^{0.55}$

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. Also 'own input' is available and the material data can be defined using the Plus-button.



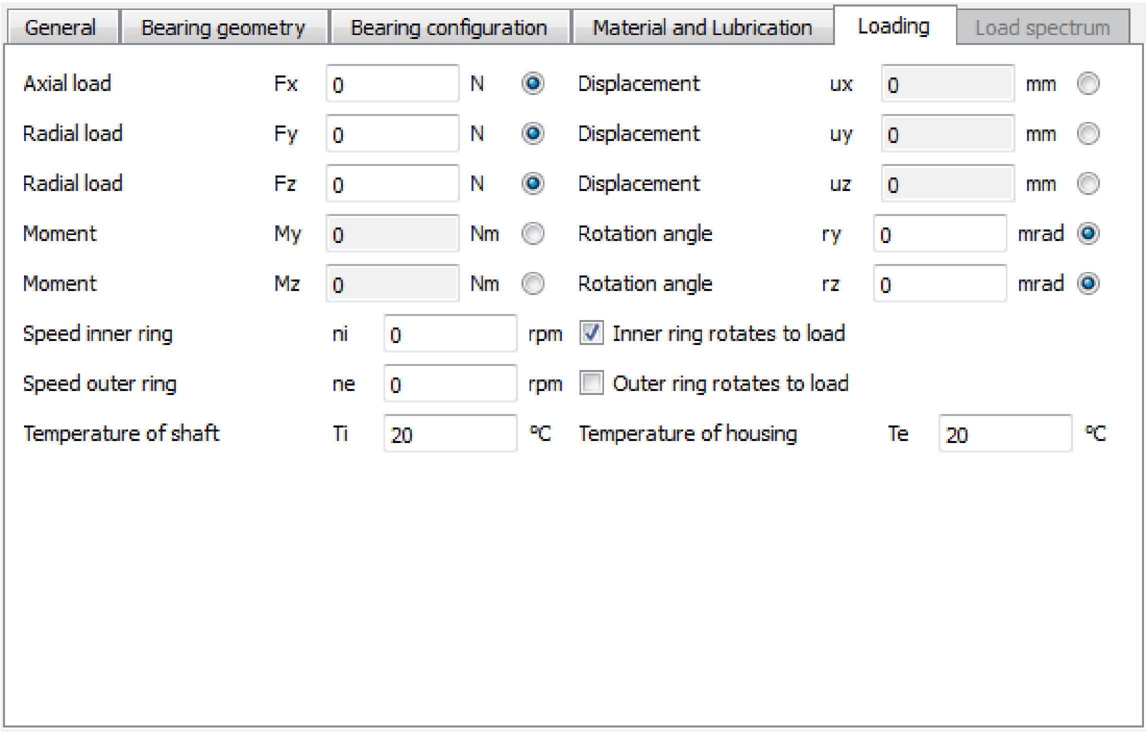
For material data the Youngs modulus, the Poisson number, the density and the thermal elongation coefficient can be defined. This data is used to calculate the operating clearance and the load distribution within the bearing. Please not, that it does not affect the load capacity of the bearing, which is calculated according ISO 281 and ISO 76 in any case.

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.

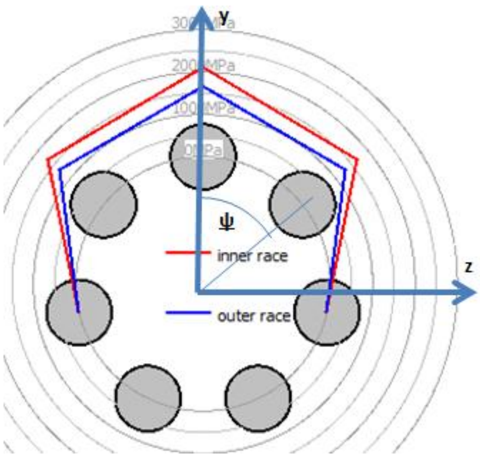
A moment load or a rotation (misalignment) can be entered for two directions only, because the rotation around the bearing axis is not constrained.

Use the radio buttons to select the elements you want to enter.



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.

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.

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.

Temperature of shaft

The temperature of the shaft is used for the calculation of the interference fit between shaft and inner ring only. It affects the operating clearance.

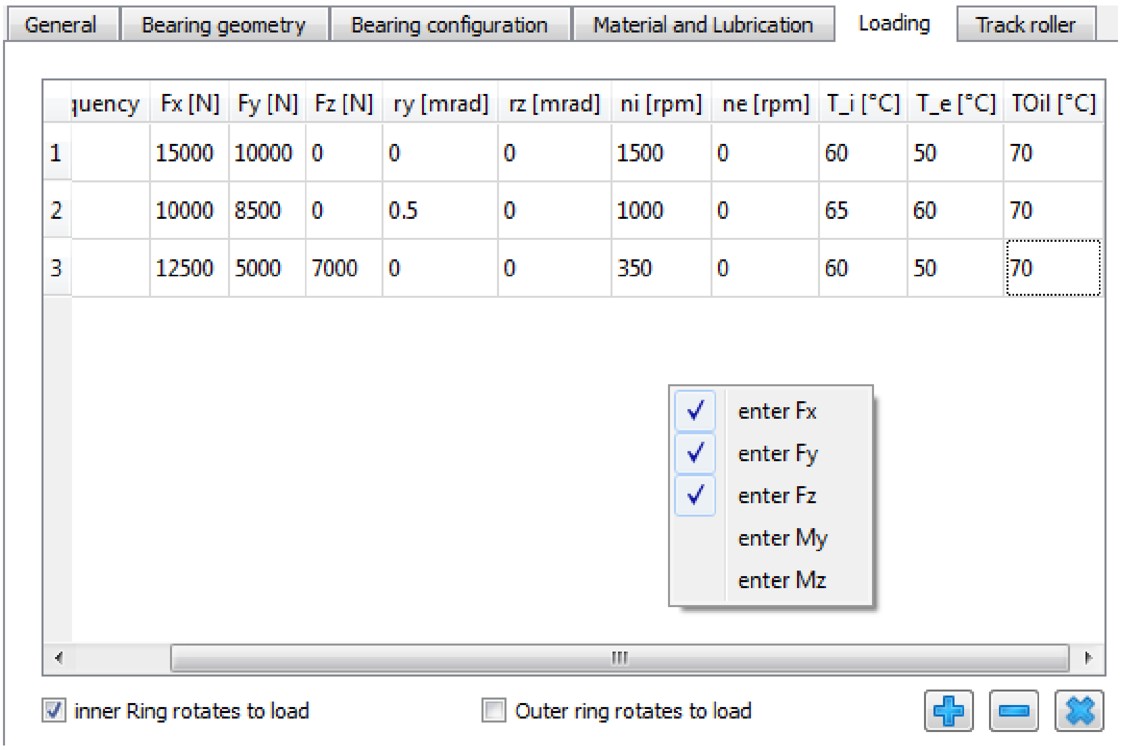
Temperature of housing

The temperature of the housing also affects the operating clearance.

The temperature of the rolling element is assumed to be the mean value of shaft and housing.

Load spectrum

Instead for a single load case the calculation can be done for a load spectrum. The load or displacement, the rotation speed and the temperatures can be specified for each element.



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 'X'-button clears all the inputs.

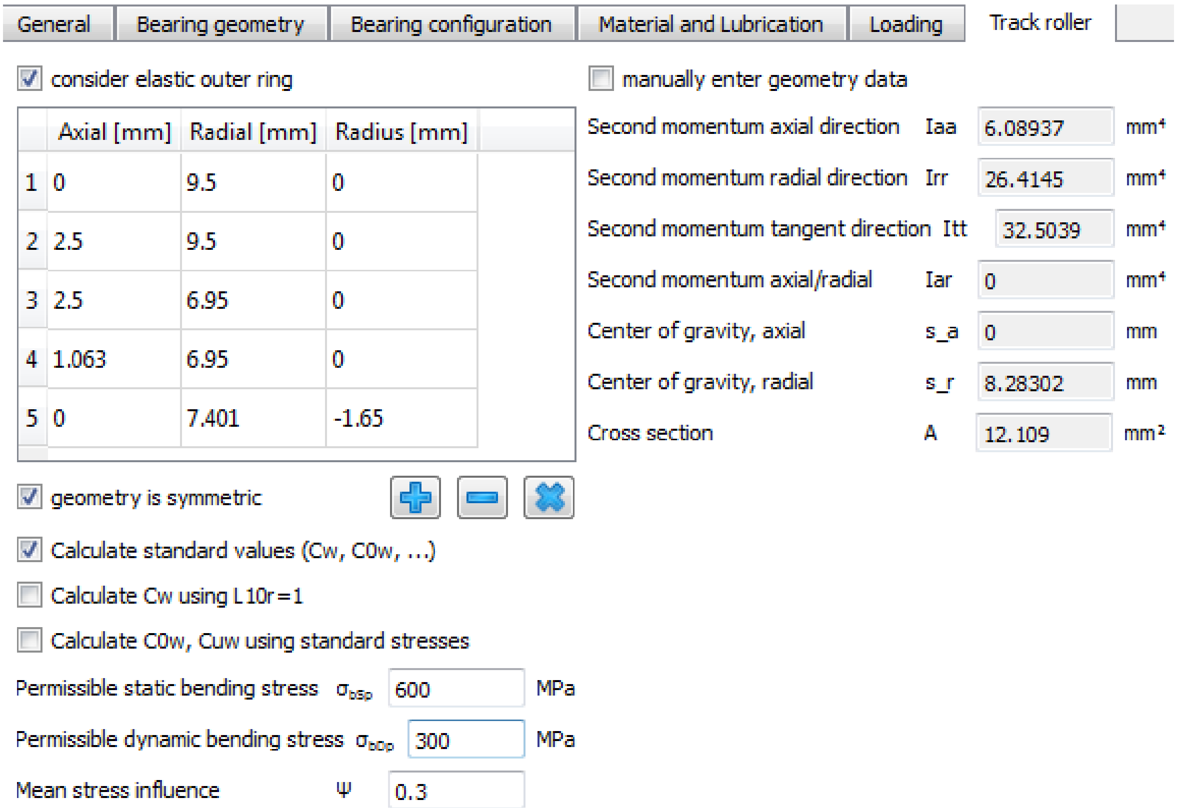
As for the single load case a selection is available if the ring is rotating relative to the load or not. A ring not rotating to the load would be on the safe side.

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 first element of the load spectrum.

Track rollers

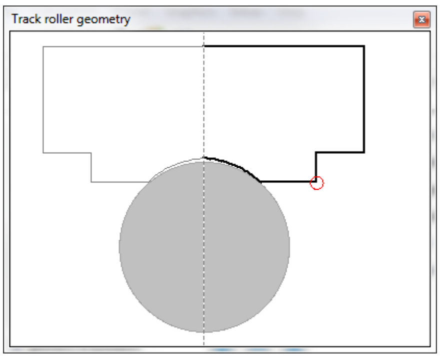
Elastic deformations of the outer ring can be optionally enabled. This is an extension to the base bearing software.

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.



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.

Symmetric geometry

A symmetric geometry starts at axial position zero on the outside diameter and ands on axial position zero on the inner diameter.

An unsymmetrical geometry should end with its start point. Note that the axial position zero is used as position for loading for the calculation of effective load capacity, so the axial position of the unsymmetrical geometry makes a difference there. Choose the axial position zero for the point where the force is usually applied.

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 be 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 option 'Calculate C_{0w}, C_{uw} using standard stresses' 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 were 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.

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, Cw will lead to the same life for the elastic bearing as C for the rigid bearing.

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.

For roller bearings edge stresses can occur on high loading leading to different results than usually found in bearing catalogues. Therefor a second option exists for the calculation of effective static load capacity C_{0w}. If the option is not set C0w 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.

For ball bearings this option does not affect the results, it only makes a difference for roller bearings.

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.

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.

$$\sigma_{max} = \sigma_{b,max} + F_t/A$$

$$\sigma_{min} = \sigma_{b,min} + F_t/A$$

$$\sigma_m = (\sigma_{max} + \sigma_{\text{min}})/2$$

$$\sigma_a = (\sigma_{max} - \sigma_{\text{min}})/2$$

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.

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.

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.

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.

Center of gravity

The center of gravity is defined using its axial and radial coordinate. For symmetric cross sections the axial position is zero.

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.

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.

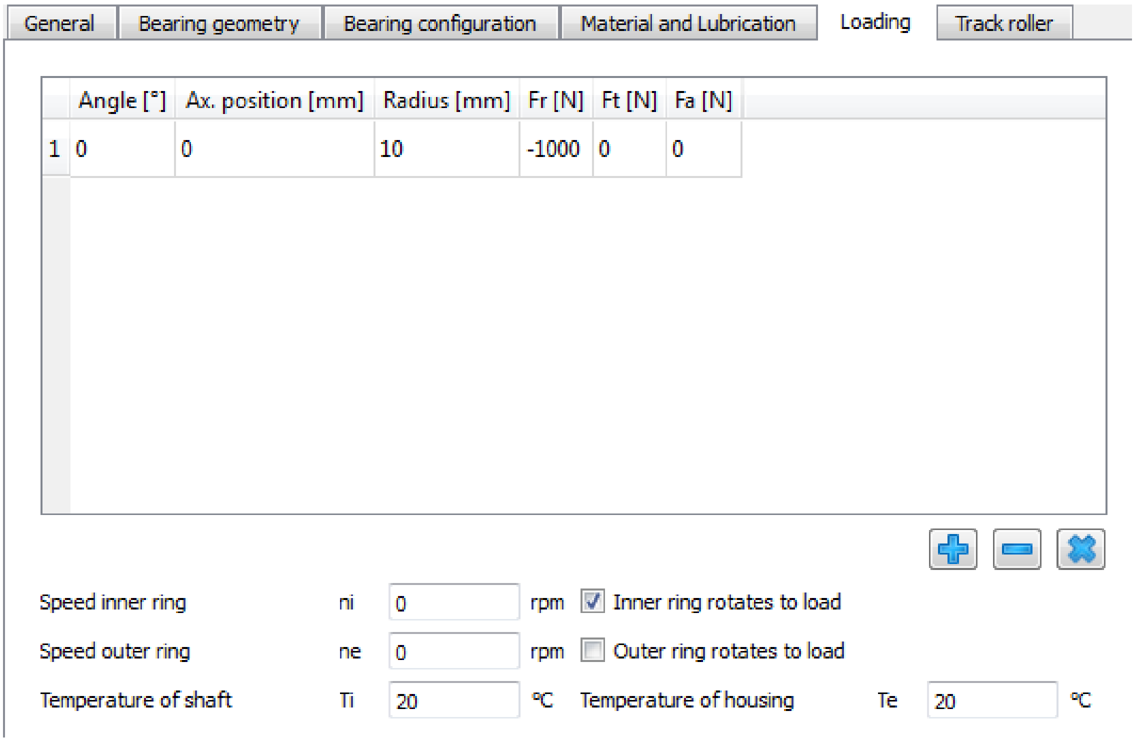
Track roller loading

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 axial position can be used if you assume that the radial load is not centric. An excentrical 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.

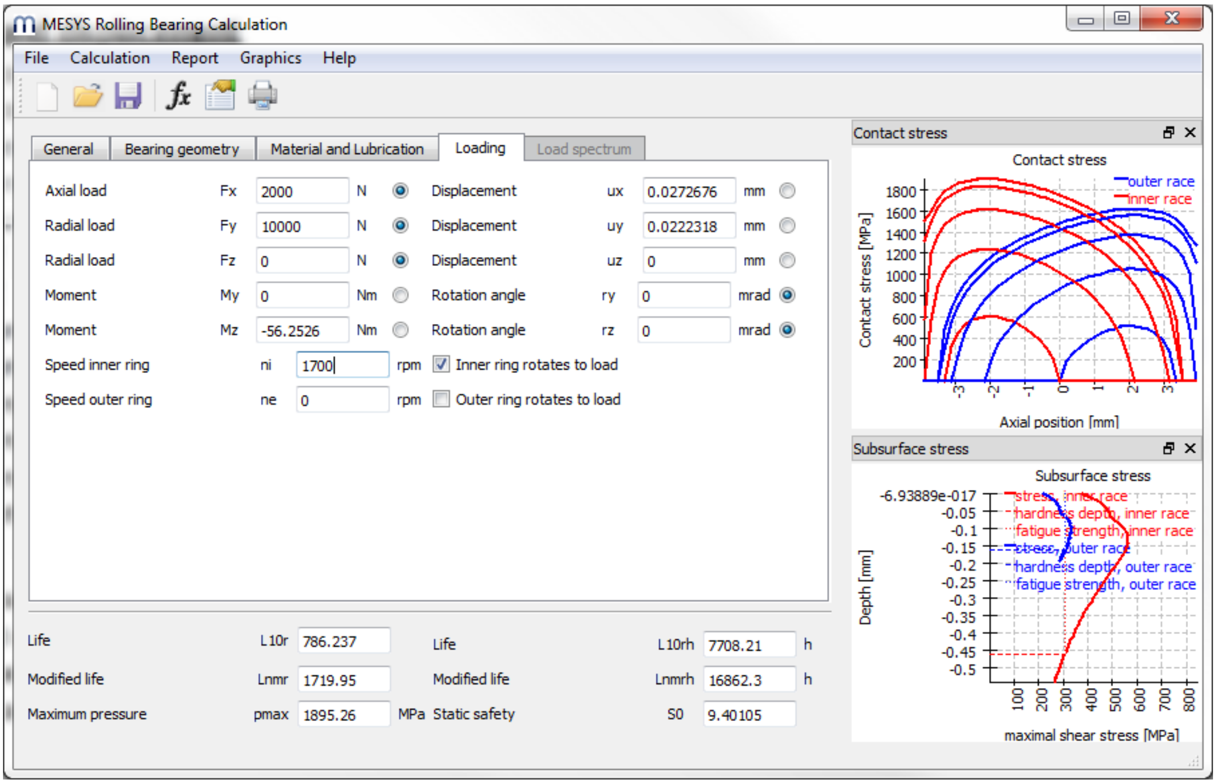


Load spectra for track rollers

Using load spectra only one load is acting on the outer ring. An angle of zero is used and the radius is half the outside diameter of the bearing. Please contact support@mesys.ch if you require more possibilities.

Results

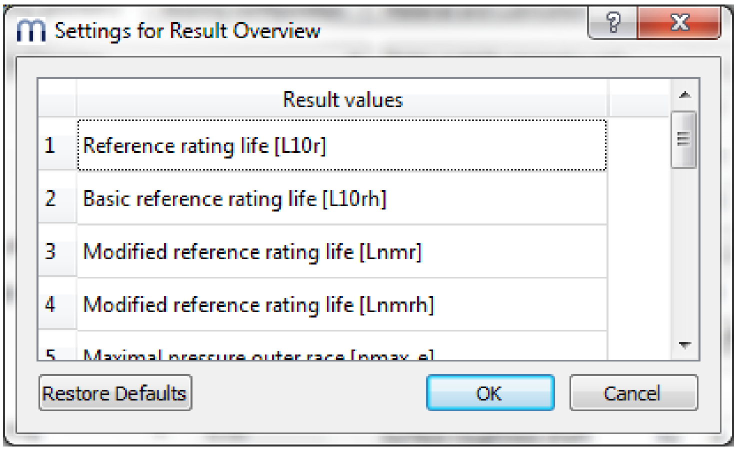
Results are provided as a small result overview directly in the software, a main PDF text report, a tolerances report and separate graphics windows. Graphic windows can be docked onto the main program interface and are automatically updated on each calculation.



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.

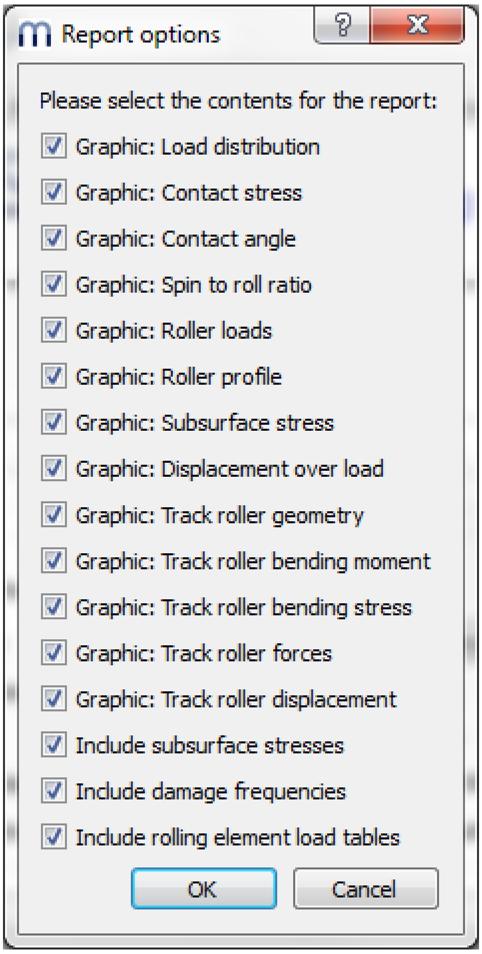


The result overview can be customized using “Extras->Result overview”. The items shown in the overview can be selected and additional items can be added using the context menu with the right mouse button. Note that only values are actually shown in the result overview that are valid. So you will see 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.

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.

Saving main report

The main report can be saved using the menu “Report->Save Report As” in different formats. It can be saves as PDF or ODT (Open Office) format which is including Text and Images or in HTML, which does not include Images. The HTML export is still useful, for accessing the result data for other programs for example.

Customizing report logo

As default a MESYS logo is shown on top of the report. This can be customized by placing a ‘logo.png’ into the installation directory. 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.

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.

Result Graphics

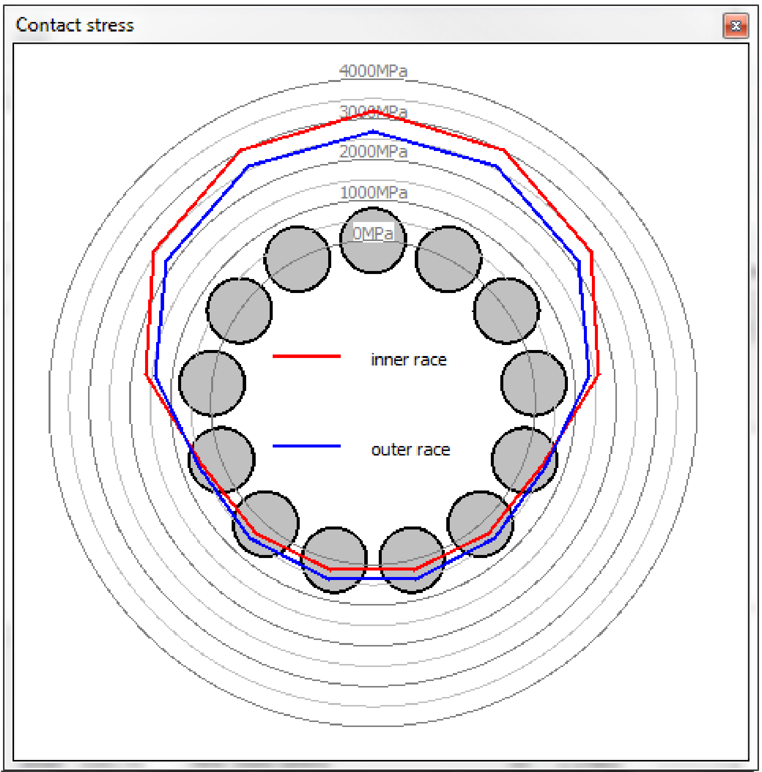
Several results can be shown as graphics.

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.

Load distribution

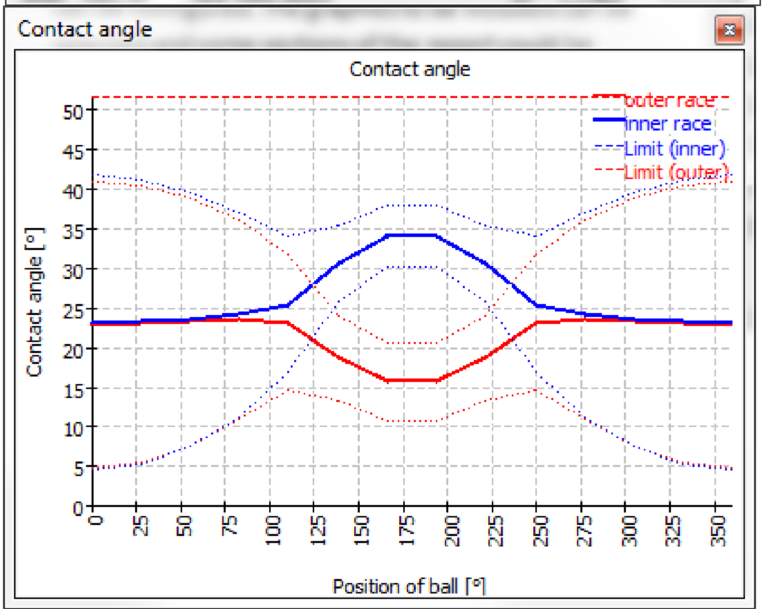
Here the contact stress of an angular contact bearing with radial and axial load and relatively high speed is shown. The contact stress is larger on the inner race (red) in the load zone but slightly larger on the outer race on the opposite side because of centrifugal loads.



Contact angle

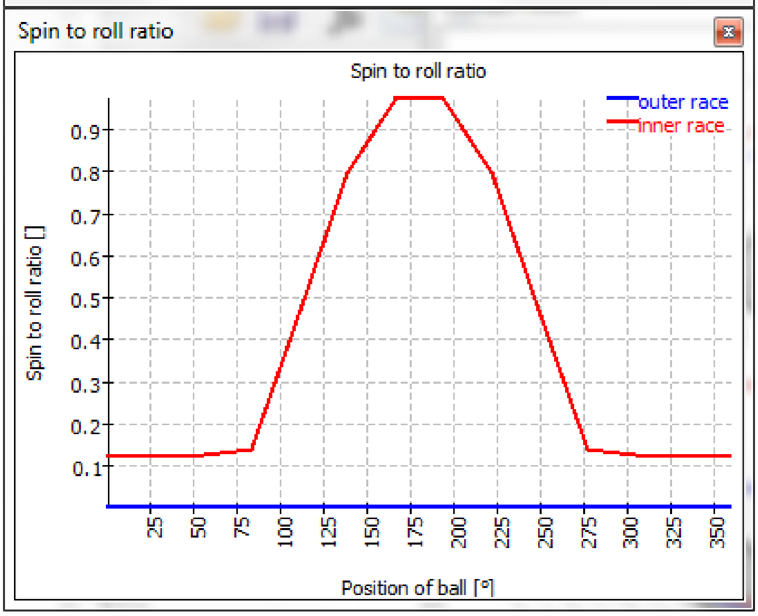
The contact angle is shown for an angular contact bearing with nominal contact angle of 20°. In the load zone the contact angles of inner and outer race are almost equal, but there is a big difference opposite to the load zone because of the centrifugal load.

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.



Spin to roll ratio

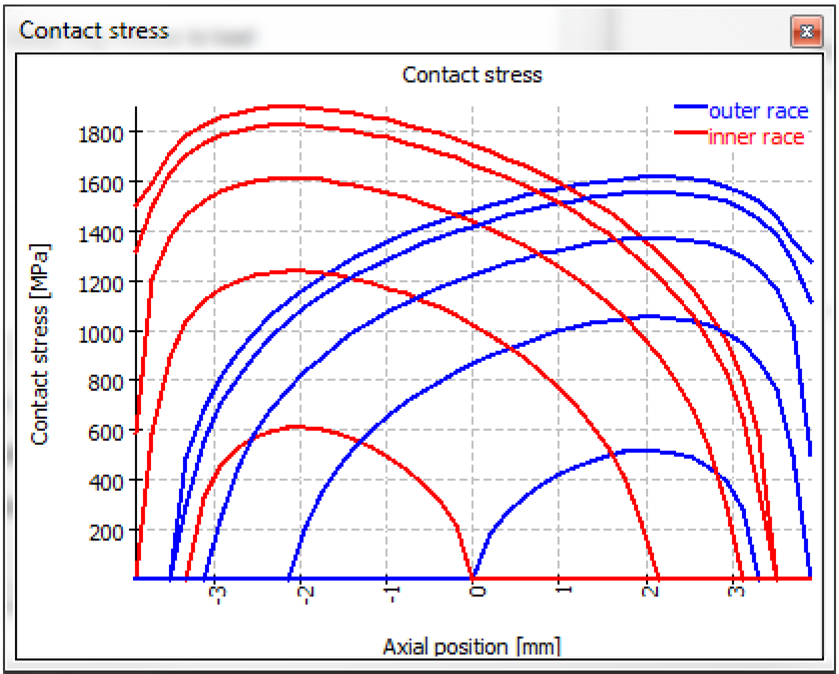
The spin to roll ratio can be shown also. The calculation assumes 'outer race control' so there is no spinning at the outer race and all the spin occurs at inner race.



Contact stress for roller bearings

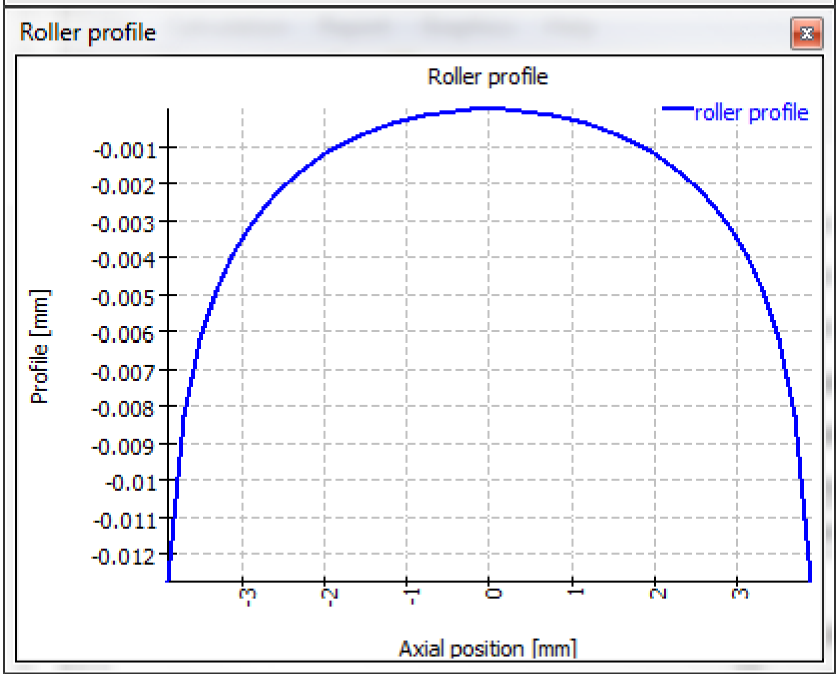
The contact stress for roller bearings is shown over the axial position or the length of the roller. This example is for a cylindrical roller bearing under radial and axial load and without misalignment between inner and outer race.

Because of the tilting of rollers under axial load the radial load is not distributed equally over the length of the roller.



Roller profile

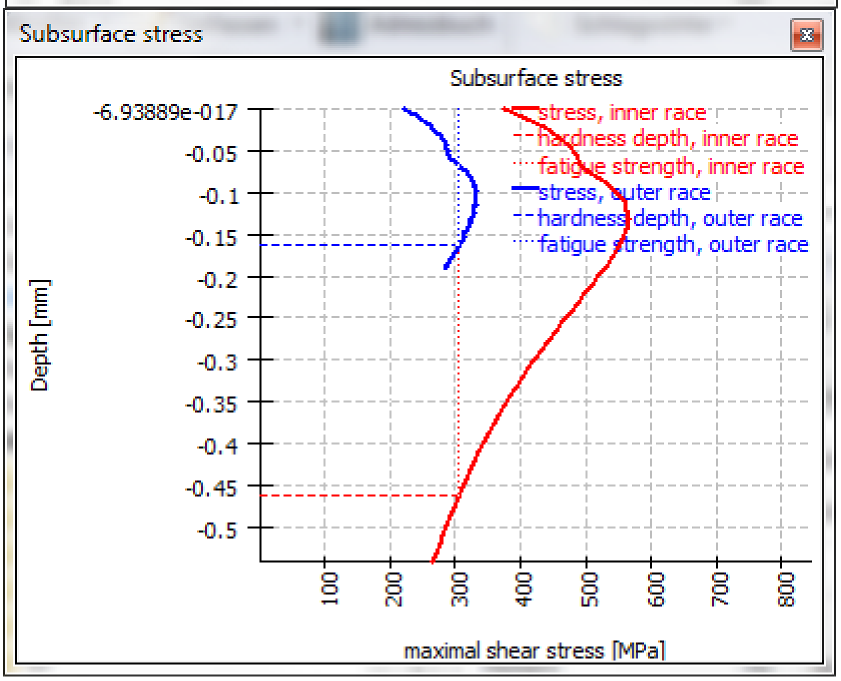
The roller profile can be shown graphically. In the current version a logarithmic profile according ISO/TS 16281 is always used.



Subsurface stress

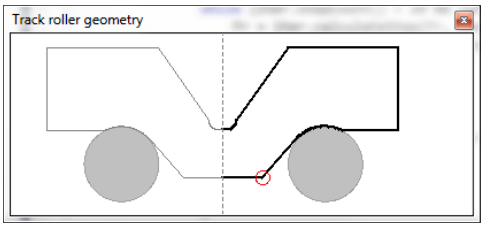
The subsurface shear stress is shown for the contact with the highest surface pressure. The required hardness depth is shown in the diagram dependent on the fatigue strength of inner and outer race.

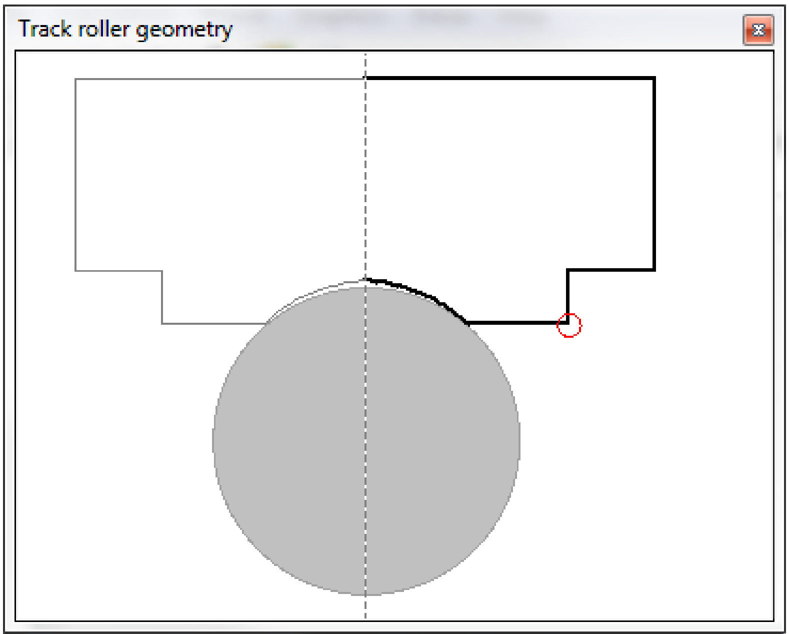
The red curve is for the shear stress of the inner race, the blue curve for the outer race.



Track roller geometry

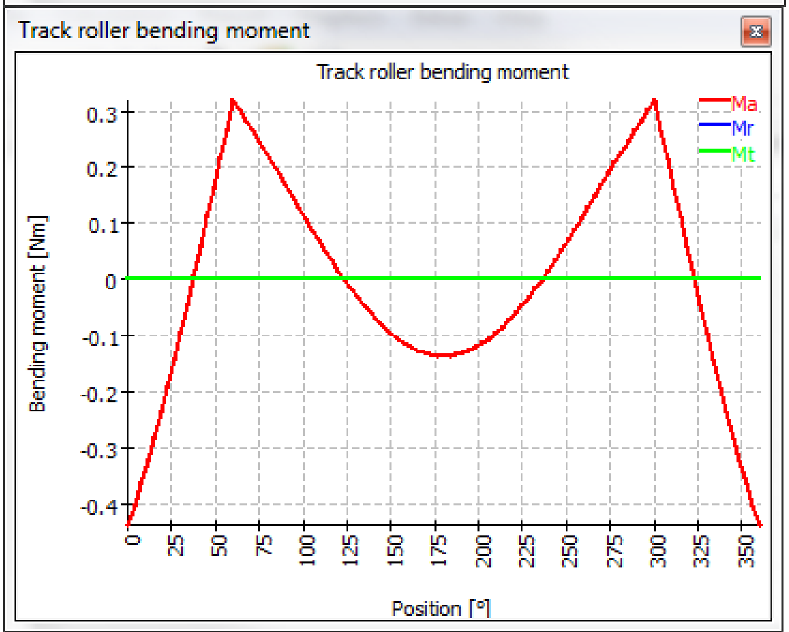
The track roller geometry is shown if defined using point data. For symmetric cross sections the mirrored is shown in grey and the critical point for bending is marked by a red circle.





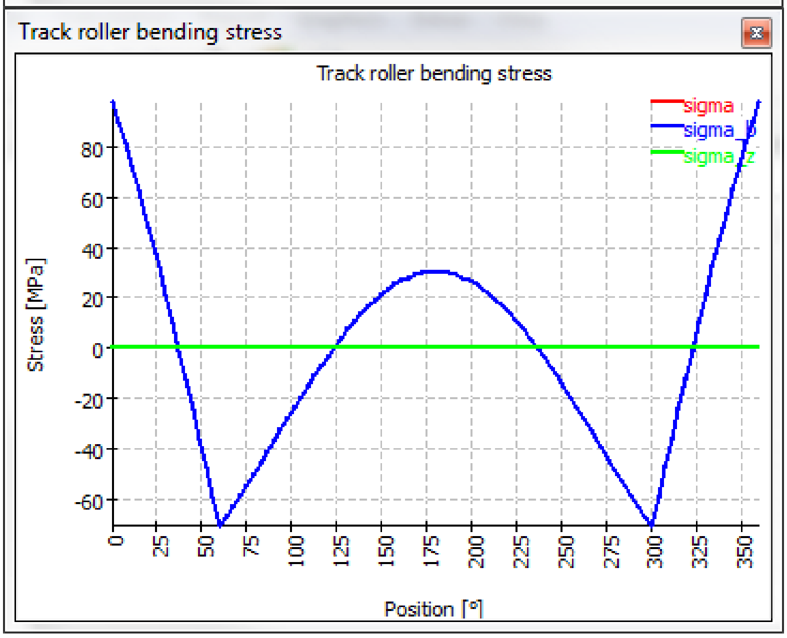
Track roller bending moment

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.



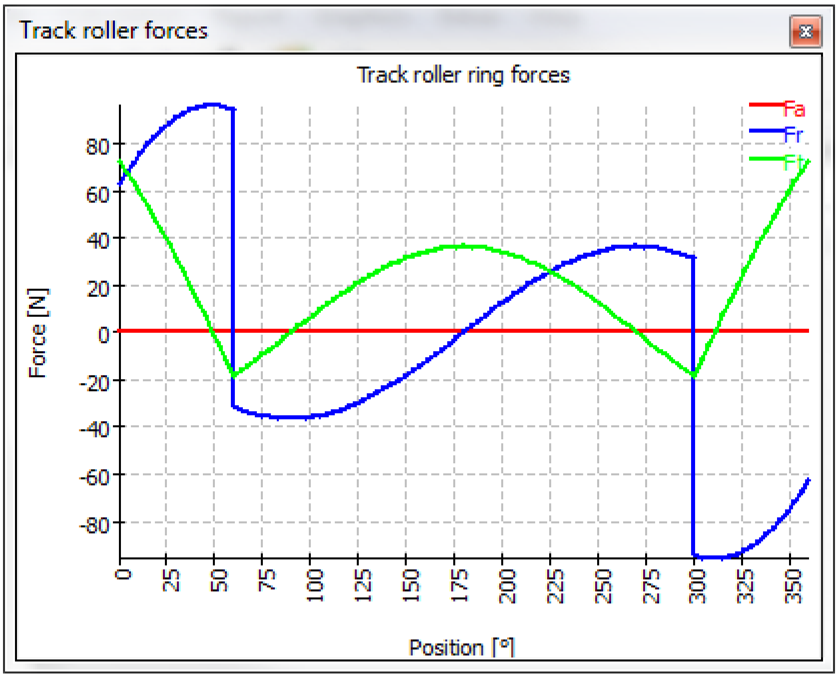
Track roller bending stress

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.



Track roller forces

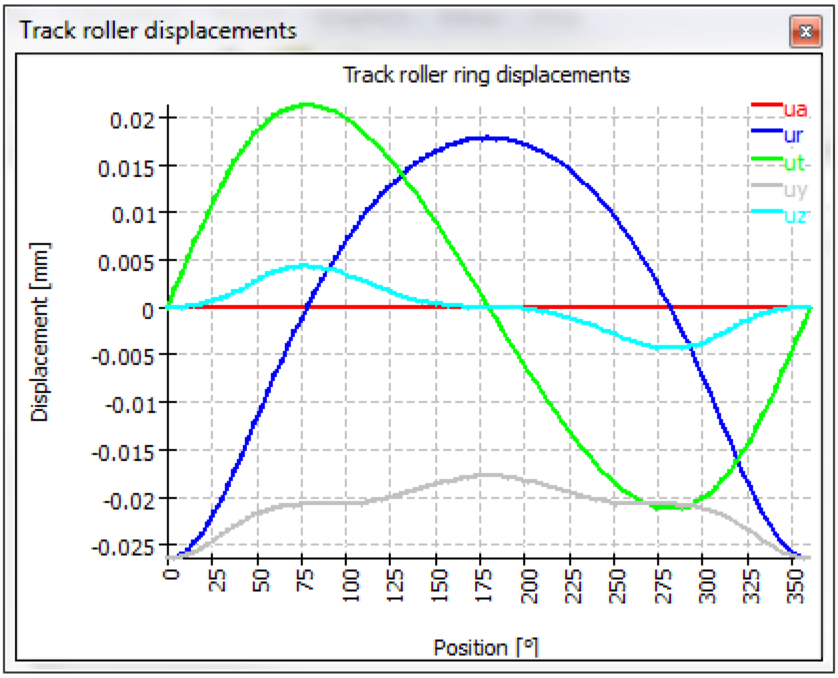
The forces in the beam modeling the outer ring are shown. The force F_a is in the direction of be bearing axis, therefore it is a shear force in the beam. The force F_r is a shear force in radial direction and F_t is the tangential force so it is tension (positive) or pressure (negative) in the beam.



We can also see that loads are only introduced at 0°, 60° and 300°, so only three rolling elements are under load.

Track roller displacements

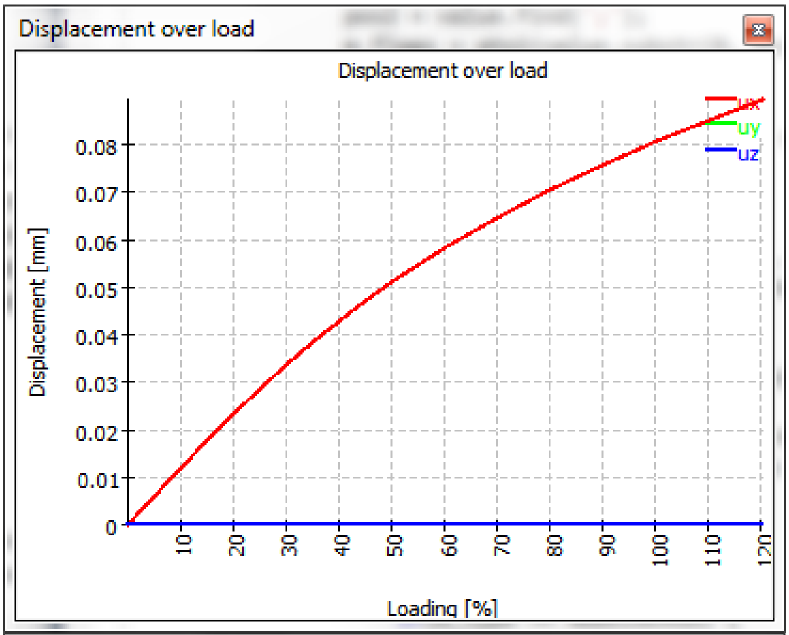
The displacements are shown in local coordinates and in global coordinated. The global coordinates allow seeing quickly that this example makes mainly a rigid body movement of the outer ring in negative y-direction. In other examples the deformations of the ring might be different, so the values in local coordinates are more important.



Displacement over load

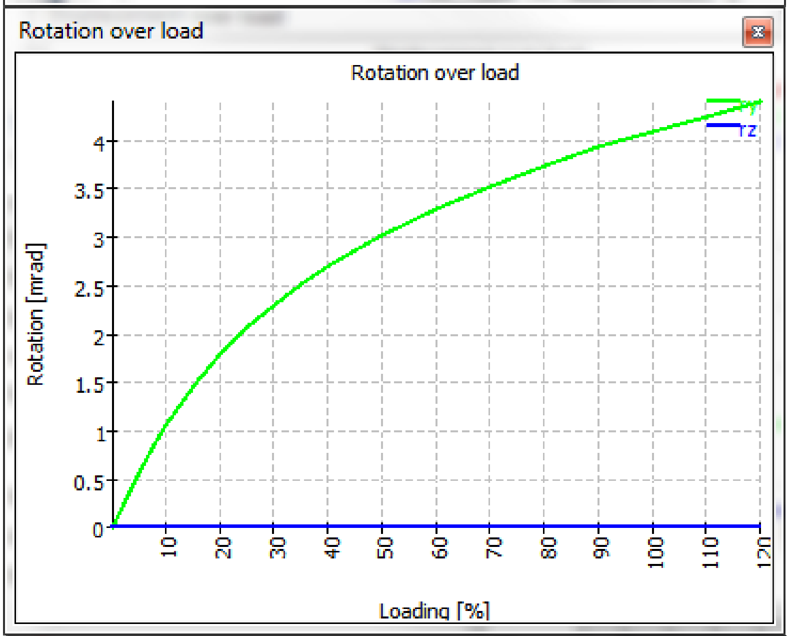
The displacement over load can be shown as graphics. All load components are multiplied with a load factor and the displacements are shown. Only loads are multiplied with the load factor, displacements or rotations entered are not modified.

This diagram can be used as information about stiffness. Dependent on the type of loading (axial or radial) different stiffness can be calculated.



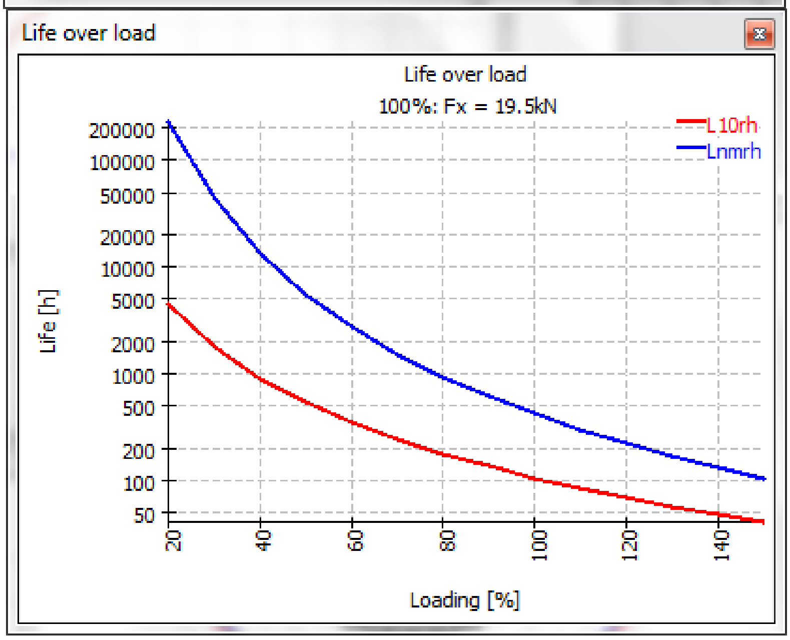
Rotation over load

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.



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.



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