Online Calculation of Guide Rings for Hydraulic Cylinders

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Figure 1: Guide rings /7/

Abstract

Hydraulic cylinders are some of the hydraulic components which convert the energy into force and / or movement at the end of the hydraulic circuit. Side loads can occur, which must be absorbed by guide elements. A simplified FEM-based calculation method and test methods of plastic guide rings are described, depending on various parameters (material, deflection, angular misalignment, geometrical conditions, etc.). For the product developers, material-specific prognosis of the guide elements behaviour are possible, regarding stresses, strains and load-bearing capacity of the guide elements, considering the specific dimensions and general conditions, due to a company-internal, parameterized FEM tool. In consequence of an optimized design, the number or the width of the guide elements in a cylinder can possibly be reduced, the resulting advantage could be a shorter length of the cylinder components and corresponding cost reduction.

Keywords: Hydraulic cylinder, bearing ring, guide ring, wear band, FEM Material Modeling
Target audience: Mobile Hydraulics, Stationary Hydraulics, Industrial Hydraulics

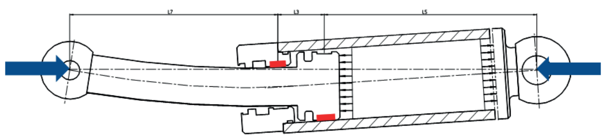


Figure 2:Deflections at a hydraulic cylinder /6/

1 Introduction

When reliability and robustness are needed, hydraulic units and systems are some of the best solutions to go for. The use of guide elements enables a low-friction and low-wear relative movement between the movable components of the hydraulic cylinder. The transverse loads occurring in operation have to be endured in a defined manner and unwanted metal contact between the piston rod or the piston body and the surrounding housing components must be prevented. /1/

1.1 Contact conditions

As a result of the guide clearance, tolerances and the elastic deformation of the components under load and pressure (deflection of the guide element, bending of the shaft, cylinder expansion, Fig.2), an angular deviation develops between the piston rod or the piston body and the counter surface (Fig. 3). Calculation of the transverse forces and collision analysis, both based on idealised contact conditions with parallel axes, cause incorrect results. Excessive tension peaks in the edge area of the guide element (responsible for edge break) are not taken into consideration. Also not regarded are the distance between the metal components (metal contact), which changes with the incorrect position, and the changed force initialization. Depending on the type of guide, the result of an idealised observation in this regard must be evaluated differently. /1/

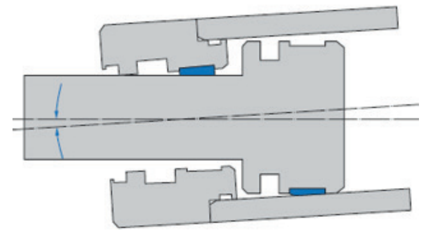


Figure 3:Angular deviation cylinder /6/

1.2 Guide elements

Particularly when a high transverse load is expected many users install traditional cylindrical metal plain bearings. Coaxial parallel metal guides are subjected to considerable tension peaks as a result of the angular misalignment. The permissible value of the surface pressure is reached at a comparatively low transverse load and simultaneous minimal deflection of the guide element. In the pressure zone there is insufficient lubrication. At low sliding speeds stick-slip may occur simultaneously with high load and wear to the counter surface. If the transverse load in the limit range is suddenly applied, a breakage of the edge in the region of the guide is possible. The tension increase in the area of the supporting edge can be reduced by the use of guide sleeves made of synthetic material (thermoplastic or composite fabric materials - fabric-base laminate). The elastic support of the fabric-base laminate guide increases the support length of the guide ring compared to the metal guide. Synthetic guide rings are comparatively economical. But the collision analysis becomes very important due to the elastic behaviour of the synthetic materials. /1/

1.3 Material

Apart from metallic guides, synthetic materials are often used as cost-effective alternatives, e.g. fabric-laminated materials (fabric-base laminate), reinforced thermoplastics and PTFE. The metallic guides offer a high compressive strength and a tight tolerance play, but due to the material and the production they are on a very high price level. When selecting the guide materials, a decision must be made between the requirement for pressure strength (load bearing capacity), possible tolerance play and the cost restrictions for the cylinder. The synthetic materials fabric-base laminate, polyamides and PTFE differ regarding the permissible specific load capacity as a function of the temperature. If temperatures occur in hydraulic cylinders around 100°C, this reduces the possible load of: PTFE guides below 10 N/mm², PA guides around 30 N/mm², whereas fabric-base laminate is able to bear 50 N/mm² and more. In a variety of applications, temperatures of less than 100°C occur, so that the PTFE and polyamide guides are also used. However, it is also always necessary to consider the required load-bearing capacity in relation to the occurred transverse forces. Often additives are added to the basic materials. These are aimed at reinforcing certain properties such as reducing friction or increasing the compressive strength. The influence, however, is limited in comparison with the value level predefined by the basic material. When selecting guide materials, account must be taken of: the specific load capacity, thermal stability, abrasion resistance, impact to the counter surface, elastic behaviour (mounting), manufacturability (e.g. diameter range) and compatibility with the used media. /1/

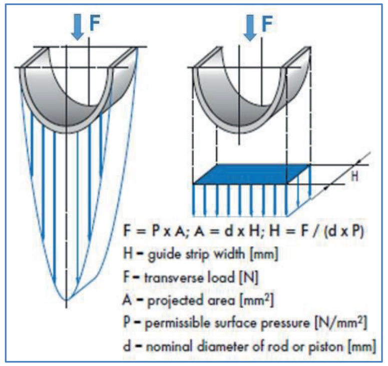


Figure 4:Guide width /1/

4 Guide width

/hen calculating the transverse load, applied in the area of the guide element, in addition to the geometric ›nsideration (external force application, distance between bearings, angular deviation, guide play, ...) we should so take in account the elastic deformation of all components involved (deflection of the guide element, bending ' the piston rod, expansion of the cylinder, etc.). A detailed verification is often neglected because of the efforts ıd complexity of calculations. A realistic estimation of the transverse load must pay particular attention to the nits of the mechanical loading of the metal components. E.g. in the case of long slender cylinders the permissible ansverse load is limited by the bending strength of the piston rod and other factors. The classical assumption, at about 10% of the hydraulic force is applied as a transverse load, would in reality result in bending the piston ›d in many cases. If the magnitude of the normal force applied in the area of the guide element is defined, the inimum required guide width (H) can be determined (Fig. 4). /1/

he permissible specific surface pressure in the specified form is a manageable calculation value with reference the projected area and does not represent the material characteristics. In the definition of the permissible FST ›ecific surface pressure the nonlinear pressure curve over the contact range, the tension increase in the edge area ' the guide rings and an angular misalignment are all considered. When considering the specified FST values of e permissible specific surface pressure (P) of the guide rings it must be noted in comparison that some anufacturers indicate extra safety factors. However, that does not bring any increase in safety into the result of e calculation, because this factor is reset in the associated equation. Metallic contacts between housing ›mponents and the counter surface are unwanted. The maximum permissible deflection (y) of the guide ring is nited by the smallest metal gap inside the sealing system, in general behind the primary seal. Depending on the ıgular misalignment (α) of the piston rod, with reference to all influencing quantities, and the possible deflection ·) the usable guide width is reduced compared to the geometrical total width of the guide elements (H). Only the ıide width actually in contact (H') contributes to holding the load. In the case of large angular misalignments, ıch as those that occur with long-slender cylinders, the guide ring may contact the counter surface on both sides ' the center axis. Here low tolerance levels promote the contact on both sides. The additional contact generates a ıable counteracting force but could also cause stick-slip effects (jamming). In this case the collision check gets ırticular significance. To select the optimum width of the guide the desired service life must also be considered. imit values have to be taken into account in the calculation of the minimum required guide width and also with ference to the permissible surface pressure of the guide elements. Guide elements that are primarily traversed in

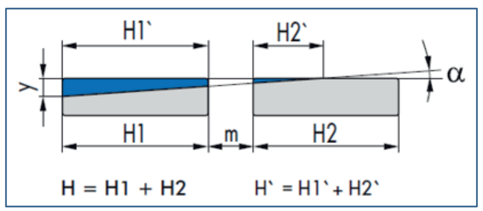


Figure 5:Usable guide width /1/

the range of the maximum possible load have a service life in the lower part of the range. Whether reducing the load by selecting a wider guide is useful in some cases depends on the previously considered safety factors as well as the total loading. /1/

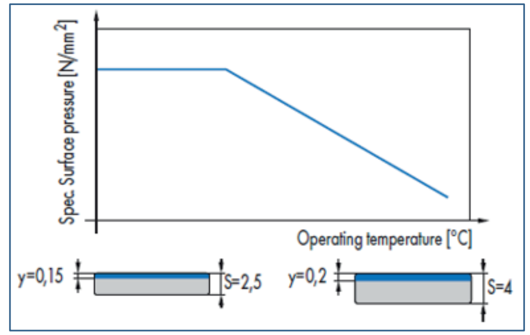


Figure 6:Deflection at maximum surface pressure /1/

1.5 Specific compression per unit area

The permissible surface pressure (dynamic) is specified at a value in the range of 17 to 25 N/mm² for the copper-tin and copper-tin-lead bronze and high load resistant copper-zinc alloys used in the area of the metal plain bearings. High-tensile alloys with values over 25 N/mm² are only used for the edge load of non-critical applications in combination with high-tensile counter surfaces. Guide rings of fabric-laminated materials (fabric-base laminate) have improved function compared to straight metal guides. As a result of the low tension increase in the edge area and the elastic properties of these materials, a higher surface pressure can be accepted. The value of the surface pressure and the characteristics under higher operating temperatures is greatly influenced by the composition of the fabric-based laminated material. Polyester and other plastics and also natural materials such as cotton are used in this application. Polyester, vinyl ester and phenolic resin and also a whole range of plastics with different properties are available for the resin matrix. While some of these compounds show significant thermoplastic characteristics, the factor of the operating temperature on the permissible surface pressure is lower for others. The values for the permissible specific surface pressure depending on the operating temperature can be found in the FST catalog, in the description of the specific guide ring. /1/

Under certain load guide elements show a deformation in the elastic range (reversible). The magnitude of the deformation or deflection (y) is determined directly by the material characteristics, the thickness of the guide sleeve and the magnitude of the load. Assuming similar material characteristics, thicker guide sleeves have softer springing under identical loading. Pressure can only be applied to the guide element at the magnitude of the

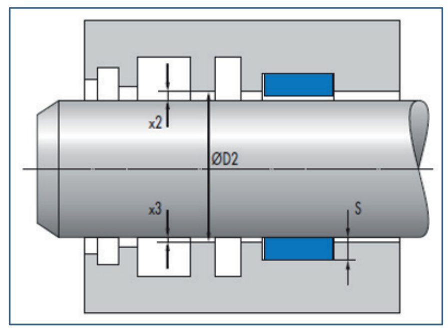


Figure 7:Sealing gap /1/

permissible surface pressure if the associated deflection of the guide element (Fig. 6) can be achieved without metal contact. In a sealing system the collision check is generally executed with reference to the metal gap on the low pressure side behind the primary seal. The minimum permissible magnitude of the sealing gap is determined by the deflection of the guide element. The maximum permissible magnitude of the sealing gap is determined by the form stability of the sealing component. General specifications for the maximum admissible gap width depending on the type of seal, the selected seal profile and the operating pressure can be found in the description of the article in the FST catalog. There is a direct geometrical dependency between the minimum required metal gap (x3) and the maximum permissible extrusion gap (x2) (Fig. 7). The gap dimensions can therefore not be calculated independently of each other. As a result guide elements cannot be subjected to the maximum permissible surface pressure at every pressure stage and with all types of seals, because the minimum required metal gap is not sufficient for complete deflection. /1/

2 Current status of FST-internal tools for limit value analysis

In the "FREUDENBERG-Technical Manual" /1/ you will find the description of Geometrical influences and approach of the limit value analysis. Also there are several analysis tools available at FREUDENBERG but only for internal use: e.g. Extrusion gap calculation programs for limit values (rod seals and piston seals), Angular deflection and gap clearance calculation program, collision check, tolerance check etc. The explanation of all

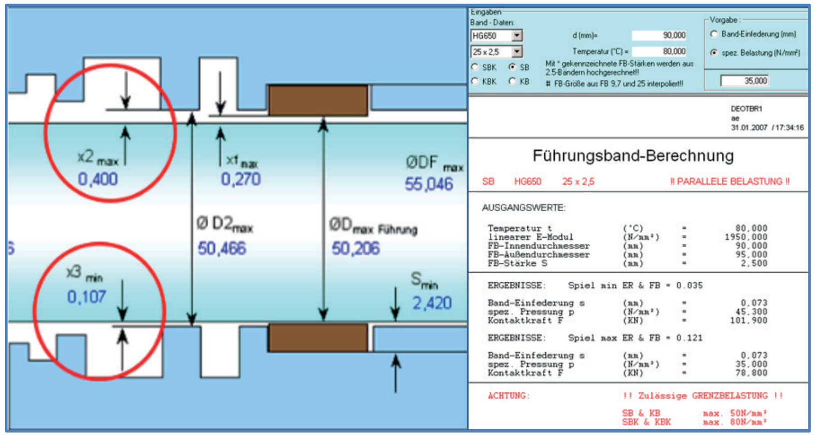


Figure 8:Example internal tool

these calculation tools would exceed the scope of the present document. This article presents a further development: a web based FEM tool for calculations which deal specifically with the topic of guide elements made of phenolic or epoxy resins (fabric-base laminate) and thermoplastics.

3 Freudenberg Web based FEM System

3.1 General Information

Every FREUDENBGERG- product developer worldwide has the possibility to access an Internet-based FEM system. By entering some defined system parameters and input variables, the developer is able to simulate the behavior of specific products without special FEM knowledge. E.g. There are Modules available for :

Figure 9: Example input mask

reciprocating seals, radial seals, diaphragms, Bellows, custom moulded products and the subject of this paper: Guide Elements. FEM System: ABAQUS for calculation with Preprocessor MSC Patran2005r2, Solver aba692 and Postprocessor aba692.

3.2 Current parameterized calculation of rod and piston guide rings

With knowledge of geometrical dimensions, angular displacement, max allowable radial compression/ deflection of the guide rings and the installation layout, the loadability of the guide band can be calculated. The user is guided by parameterized input masks (Fig.10-11).

Figure 10: Example input mask

Earlier versions of the tool were based on a simple linear elastic material model. However, comparison with tests showed that the real force-displacement diagram displayed nonlinearities both at the beginning of the deformation and after the linear elastic curve (Fig.11).

Subject of the presented project was to increase the accuracy of the calculations by the development of a suitable nonlinear model. Because Guide rings get cracks (Fig.12) in areas where the pressure is highest, the material model has to consider plastification, when pressure load is increased above a certain value. The material model has also to take into account, that, when initial pressure load is applied, surface effects yield another nonlinearity.

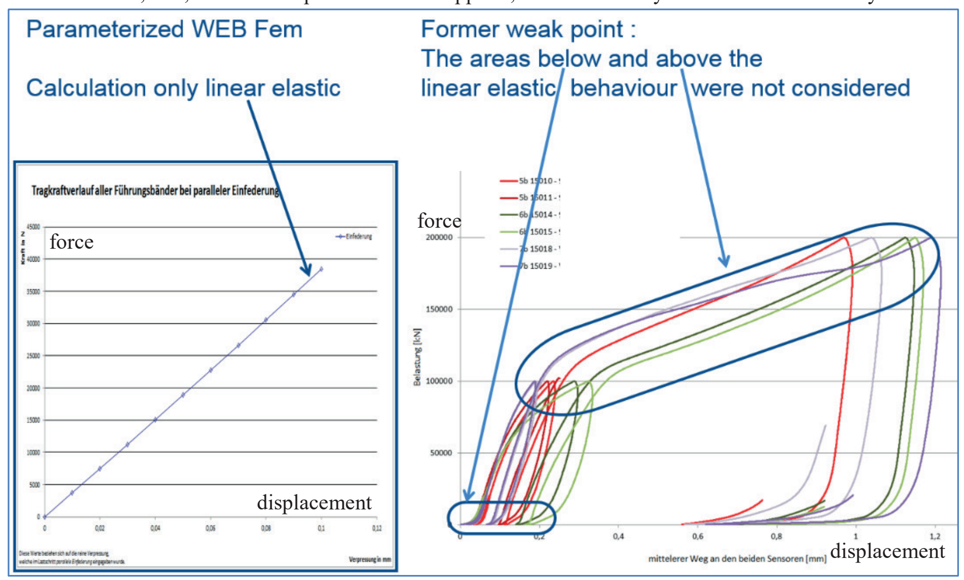


Figure 11:Comparison linear elastic calculation with test

Figure 12:Test specimen fabric-base laminate with cracks

4 Improved parameterized calculation of rod and piston guide rings

4.1 Testing Principles

For the verification of the previous calculation tools and the determination of the material data for a new material model, extensive investigations with comprehensive test series were carried out. We tested a lot of different load intervals, dimensions, materials and surfaces. The main tests were carried out in two versions: static tests (Fig.13) and dynamic tests (Fig.15). The most comprehensive test were made with specimens and housings chosen according to DIN ISO 10766 → 45 x 50 x 9,7 and 45 x 50 x 5

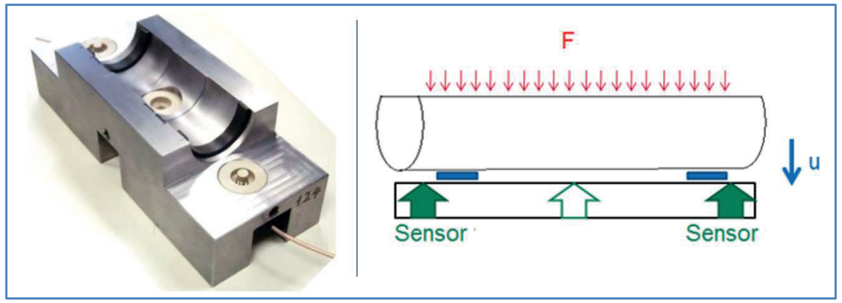


Figure 13:Fixture device and test principle static test

4.1.1 Static Tests

More than 150 static tests were carried out, using 20 different materials and 10 different geometries of the wear band. Different load intervals were executed on compression testing machines. Due to changeable inserts the fixture device (Fig.13) is designed for different guide element dimensions. The ambient temperature of the tests was 23°C and 80°C. We also carried out tests on different surface structures, to see the influence of the surface to the test results. Examples of examined thermoplastics: Freudenberg 4112, 9400, 9800, 9150 and competitor products. [PA 6, PA 6.6 , PPA 35% and 40% glass reinforced, internally lubricated (different Versions), CNC lathe turned, injection moulded]. Examples for phenolic / epoxy resigns: HG 517, HG 650 and competitor products. [phenolic resin or thermoset resin (duroplast), weave cotton yarn or weave synthetic yarn thermoset resin (duroplast), weave synthetic yarn, internally lubricated (different Versions)].

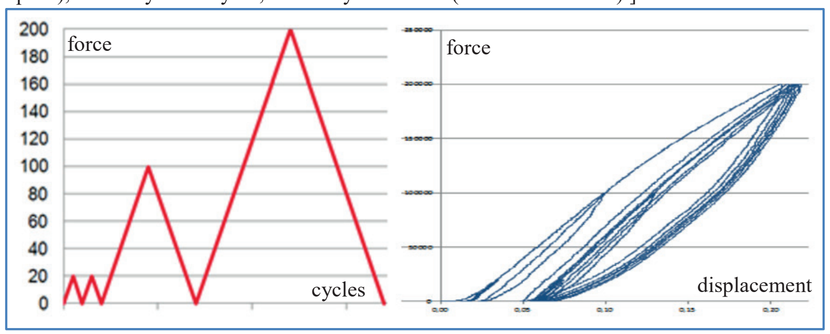


Figure 14: Example load cycles and Typical force-displacement diagram

4.1.2 Dynamic Tests (Side Load)

For the verification of calculation results and for comparative studies of the different materials practice oriented dynamic tests were performed. 50 cycles, sideload: F = 10 kN, housing: 45 x 50 x 10mm, temperature: 23°C, stroke: 350mm, pressure: 100bar

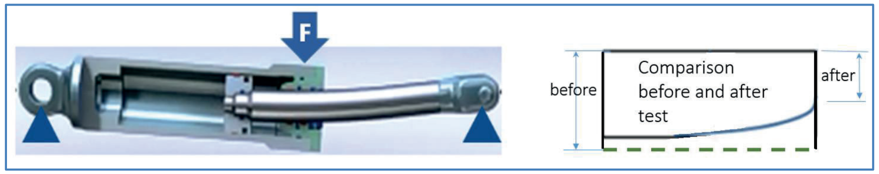


Figure 15: Side load test rig and cross section scheme, side load guide ring deformation

4.2 Improved Material Model

The test results showed a pressure dependent plastification and another nonlinearity at the start of the loading, which was assumed to be due to surface properties. Unfortunately material models which are available in ABAQUS and consider pressure dependent plastification like Cam-Clay-, Drucker-Prager/Cap-, Crushable-Foam- or Gurson-model were not sufficient for the solution of the whole set of problems. First we had to select the plasticity model. In ABAQUS, the crushable foam model showed the best numerical stability and was able to simulate the nonlinearity with high pressure.

4.2.1 The crushable foam material model

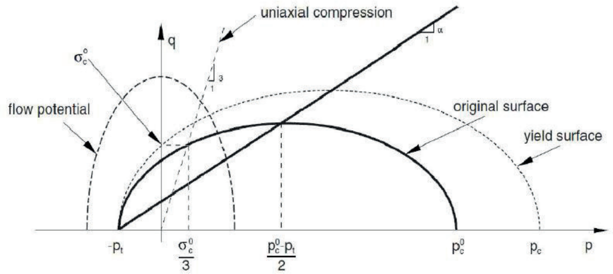


Figure 16: Crushable foam model, volumetric hardening, yield surface and flow potential in the p–q *stress plane*

The model can be used in combination with a simple linear isotropic elastic material model. In ABAQUS the required parameters are the elasticity modulus and the Poisson's ratio. The yield surface for the plastic part of the behavior is a von Mises circle in the deviatoric plane and an ellipse in the meridional plane. Within the crushable foam model there are two different types of hardening possible: volumetric and isotropic hardening. Within the isotropic hardening model the yield ellipse is centered at the origin and as it evolves it retains its original height to width ratio. With the volumetric hardening model the point on the yield ellipse which represents hydrostatic loading is fixed and the evolution of the yield surface is driven by the compressive plastic strain (Fig. 16). The hardening which occurs under tension is negligible. /4/ The crushable foam model with volumetric hardening was chosen, with the following parameters:

$F = \sqrt{q^2 + \propto^2 (p - p_0)^2} - B = 0$ The yield surface for the volumetric hardening model. (1)

$p = -\frac{1}{3}\, trace\ \sigma$ Pressure stress. (2)

$q = \sqrt{\frac{3}{2}}\ S : S$ Mises stress. (3)

$S = \sigma + p\, I$ Deviatoric stress. (4)

$B = \propto A = \propto \frac{p_c + p_t}{2}$ Size of the (vertical) q-axis of the yield ellipse, with size of the (horizontal) p-axis: A (5)

$\propto = B\,/A$ Shape factor of the yield ellipse that defines the relative magnitude of the axes. (6)

$p_0 = \frac{p_c - p_t}{2}$ Center of the yield ellipse on the p-axis. (7)

p_t Strength of the material in hydrostatic tension.
p_c Yield stress in hydrostatic compression (always positive).

The yield surface evolves in a self-similar fashion (constant α); and the shape factor can be computed using the initial yield stress in uniaxial compression, σ_c^0 , the initial yield stress in hydrostatic compression, p_c^0 (the initial value of p_c), and the yield strength in hydrostatic tension, p_t : /4/

$$\propto = \frac{3k}{\sqrt{(3k_t+k)(3-k)}} \quad \text{with} \quad k = \frac{\sigma_c^0}{p_c^0} \quad \text{and} \quad k_t = \frac{p_t}{p_c^0} \qquad (8)\,(9)\,(10)$$

For a valid yield surface the choice of strength ratios must be such that $0 < k < 3$ and $k_t \geq 0$. If this is not the case, Abaqus will issue an error message and terminate execution.
To define the shape of the yield surface, you provide the values of k and k_t . If desired, these variables can be defined as a tabular function of temperature and other predefined field variables./4/
Flow potential**:** The volumetric hardening model uses the following relation to define

$\dot{\varepsilon}^{pl} = \dot{\bar{\varepsilon}}^{pl} \frac{\partial G}{\partial \sigma}$ the plastic strain rate. (11)

$G = \sqrt{q^2 + \frac{9}{2}p^2}$ With G being the flow potential and (12)

$\dot{\bar{\varepsilon}}^{pl} = \frac{\sigma : \dot{\varepsilon}^{pl}}{G}$ the equivalent plastic strain rate. (13)

$\dot{\bar{\varepsilon}}^{pl} = \sqrt{\frac{2}{3}}\,\dot{\varepsilon}^{pl}_{axial}$ The equivalent plastic strain rate is related to the rate of axial plastic strain, $\dot{\varepsilon}^{pl}_{axial}$, (14)

in uniaxial compression.

The shape of this flow potential is depicted in Fig. 16. /4/

4.2.2 Hardening

The model assumes that the yield stress in tension remains the same throughout the whole deformation. The yield ellipse intersects the p-axis at $-p_t$ and pc being the yield stress in tension and compression respectively. Contrary to the yield stress under tension, the yield stress under compression evolves as a result of compaction (or dilation) of the material. The change in the yield surface can be expressed as a function of the size of the yield surface on the hydrostatic stress axis, p_c+p_t , as a function of the value of volumetric compacting plastic strain ε^{pl}_{vol}. With p_t constant, this relation is determined by equation (15) /4/:

$$p_c\left(\varepsilon^{pl}_{vol}\right) = \frac{\sigma_c\left(\varepsilon^{pl}_{axial}\right)\left[\sigma_c\left(\varepsilon^{pl}_{axial}\right)\left(\frac{1}{\alpha^2}+\frac{1}{9}\right)+\frac{p_t}{3}\right]}{p_t + \frac{\sigma_c(\varepsilon^{pl}_{axial})}{3}} \qquad (15)$$

in combination with user provided results of an uniaxial compression test, $\sigma_c\left(\varepsilon^{pl}_{axial}\right)$ along with the fact that $\varepsilon^{pl}_{axial} = \varepsilon^{pl}_{vol}$ uniaxial compression for the volumetric hardening model. /4/

4.2.3 Surface Properties

The next problem was with the nonlinearities at small loads due to the surface properties. In a special test series, surface changes before and after loading were examined in detail for different guide rings. Furthermore, specially manufactured macro-geometries were manufactured and examined before and after the pressure load.

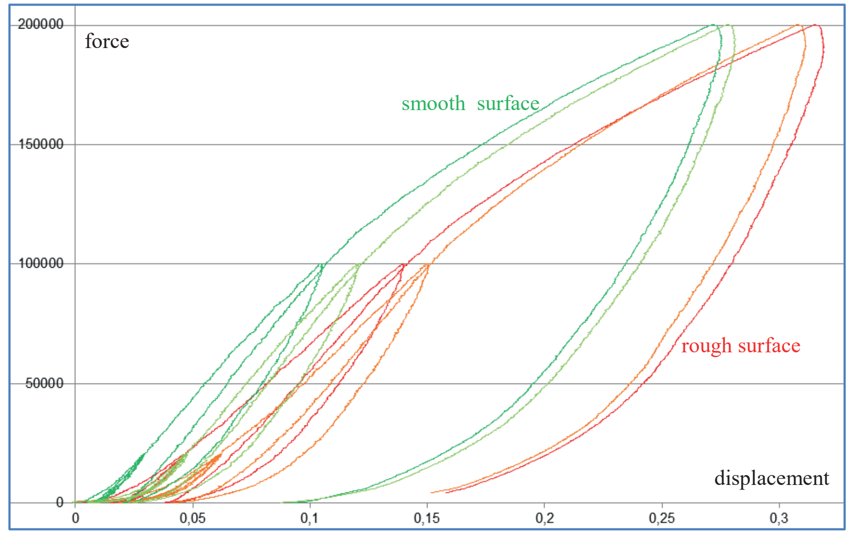
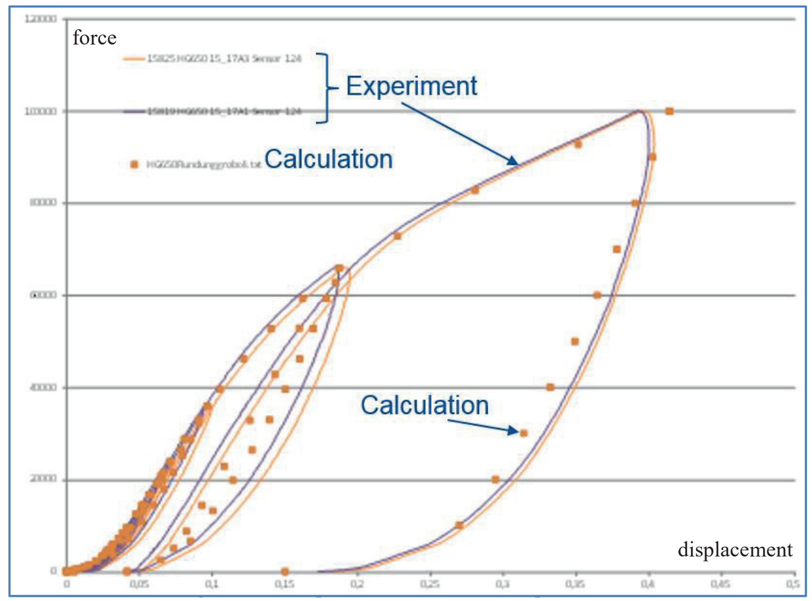


Figure 17: Example surface roughness

A theoretical approach was discussed with the aim to allow conclusions from surface measurement, roughness parameters and bearing ratio. At the end it was possible to find a special approach based on an extended crushable foam model with volumetric hardening and parameters determined in the described different tests. This material model simulates the practical pressure test sufficiently accurately.



4.3 Improved parameterized calculating – Input Examples

First of all, the necessary input parameter have to be determined. For this purpose, tools and programs described in paragraph **2** are useable (at FST) to determine the relevant factors, e.g.: calculating of the angular displacement (Fig. 19), predefinition of the allowable guide compression, sketch with dimensioning of the installation situation and the guide rings.

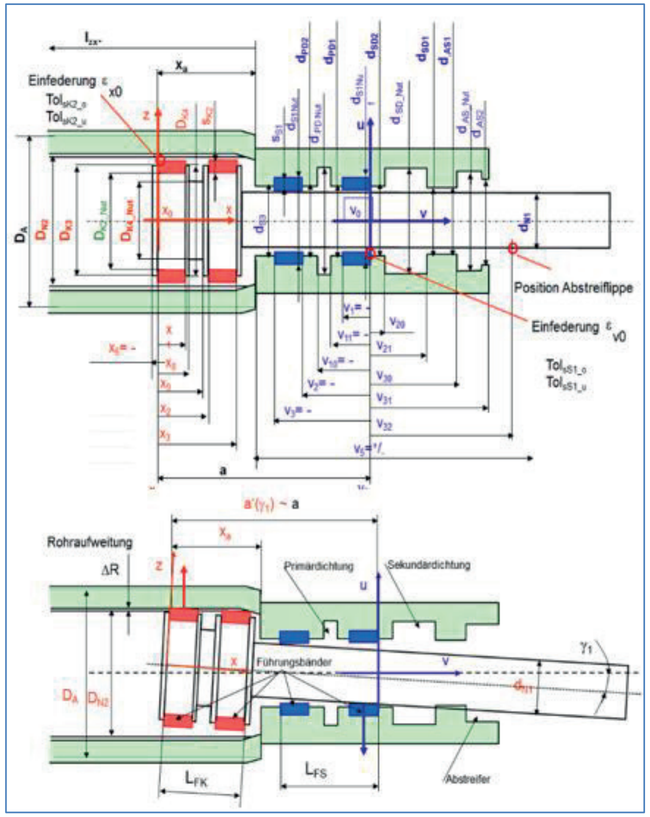


Figure 19: Parameter for angular displacement calculation (preparatory work)

Program input information : Dimensions, material data, load definition, contact definition, coefficient of friction and temperature.

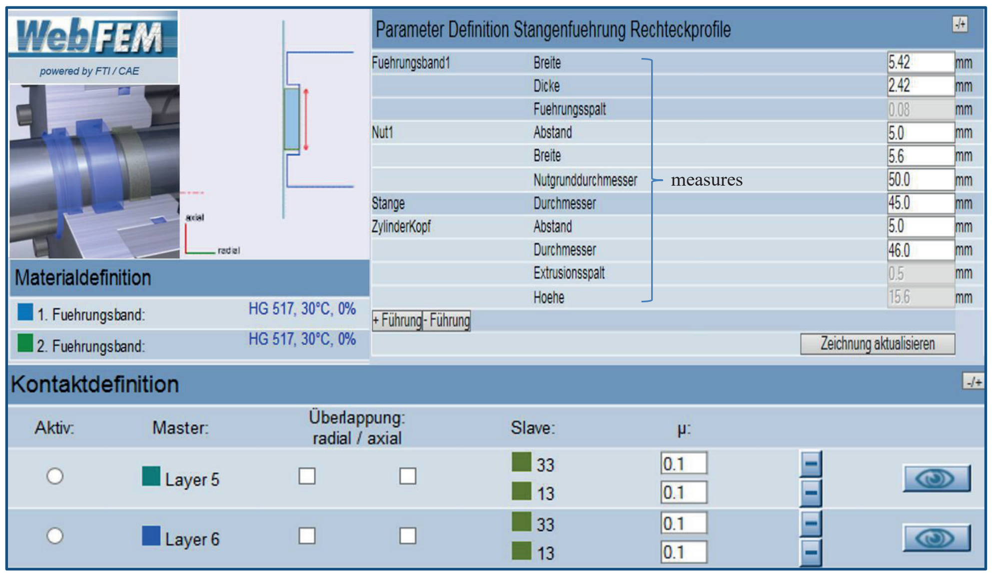


Figure 20: Input mask diameter and distances, contact definition, coefficient of friction, material etc.

Figure 21: Input mask load definition

4.4 Improved parameterized calculating – Output Examples

After preprocessing, solving and postprocessing, results like expansion, strain and loadability are available.

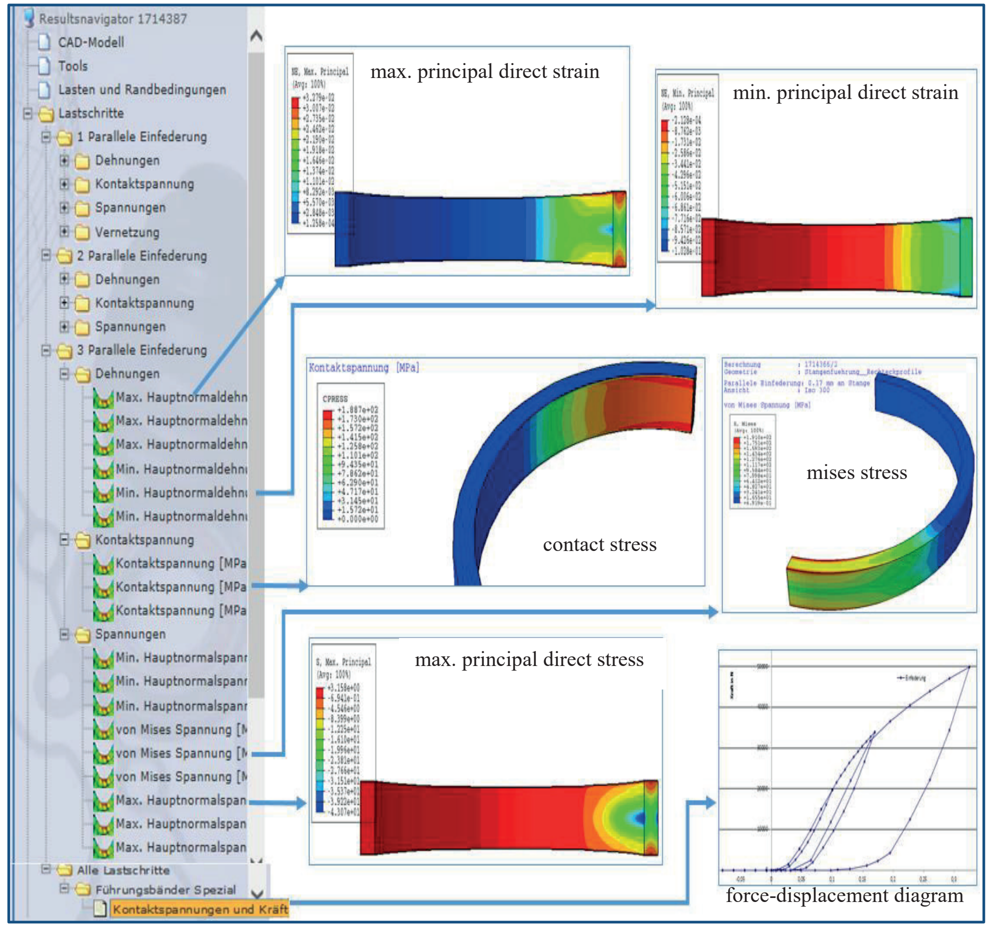


Figure 22: Example of result information

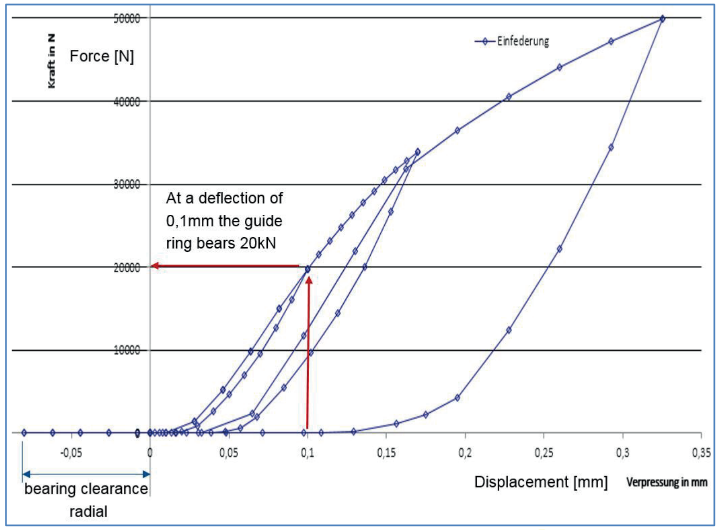


Figure 23: Output example of load-bearing capacity

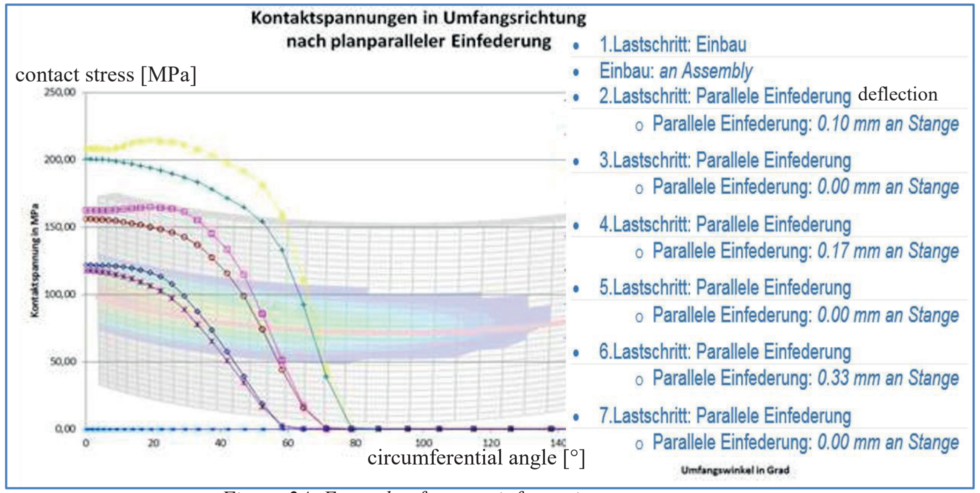


Figure 24: Example of output information, contact pressure

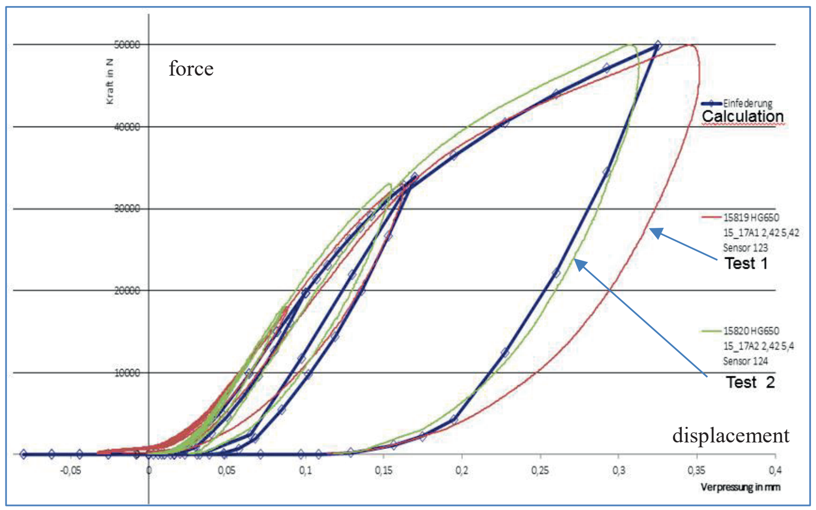


Figure 25: Comparison calculation with test

5 Summary and Conclusion

Within the Freudenberg Group, every product developer worldwide has the opportunity to access an online FEM system. Among other products, plastic guide rings can be calculated parametrised. The relevant material model has been further developed and validated so that the non-linear behavior of various guide elements and materials can be simulated realistically. Consequently it is possible in future to make accurate predictions regarding stresses, strains and load capacity. With an optimal design, selection and utilization of the products, e.g. it could be possible to reduce the number of guide elements in a cylinder and / or to save installation space or cylinder length. These improved design options ultimately can lead to cost advantages for the cylinder manufacturer.

Nomenclature

Variable	*Description*	*Unit*
σ	Stress	[MPa]
p	Pressure stress	[MPa]
q	Mises stress	[MPa]
S	Deviatoric stress	[MPa]
ε	Strain	[-]

Abbreviations

FST Freudenberg Sealing Technologies GmbH

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