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Experimental method of the service life estimation of a general kinematic pair of a cam mechanism

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Abstract. The paper deals with the experimental method of the service life estimation of the contact region of a general kinematic pair. In the case of a general cam mechanism, the general kinematic pair is formed by the contact of a cam and a follower working surfaces. We can determine the service life of the cam mechanism both experimentally and theoretically. The theoretical determination has an advantage that it is very quick, but can be inaccurate. Therefore we pay attention to the experimental determination of the service life. The test rig simulates the cam mechanism and allows the testing of the huge number of parameters. We can obtain information this way and use it for refining the mathematical model commonly used for the service life prediction.

Introduction

Part of the cam mechanism design is the determination of suitable material for production. It is possible to choose the theoretical procedures without deeper consideration of the properties of the selected material. This paper describes a test rig for testing materials suitable for cam production.

Rolling contact fatigue of the cam mechanisms

At the contact areas of the cam surface or below these points, contact stress becomes a periodical magnitude related to the angular cam displacement ψ. These transitory stresses are characterised by pulses with a periodicity of 2π (see Fig. 1). The stress limit σ_{hc} is in perfect agreement with the disturbance caused by the transitory stress. [1]

The reduced stresses σ_{red} are limited by the actual strength condition, written in the form (1).

$$\max \sigma_{red} < \sigma_{hc} \tag{1}$$

According to [1] it is a usual relation for steel between the transitory stress limit and the strength limit described by equation (2).

$$\sigma_{hc} = 0{,}66 R_m \tag{2}$$

Also applies (3), where R_e is the sliding limit of the steel.

$$R_e = (0{,}55 \div 0{,}8) R_m \tag{3}$$

Considering equation (2) a (3), we can write the condition (4) stating that no destructive action of plastic deformation is produced in the general pair under operation [1].

$$\max \sigma_{red} < R_e \tag{4}$$

This condition is the starting point for determining the material suitable for the cam mechanism production. Values of the strength limit, sliding limit, and tension fatigue limit for every steel can we found in the material sheets, eventually in the literature, e.g. [2].

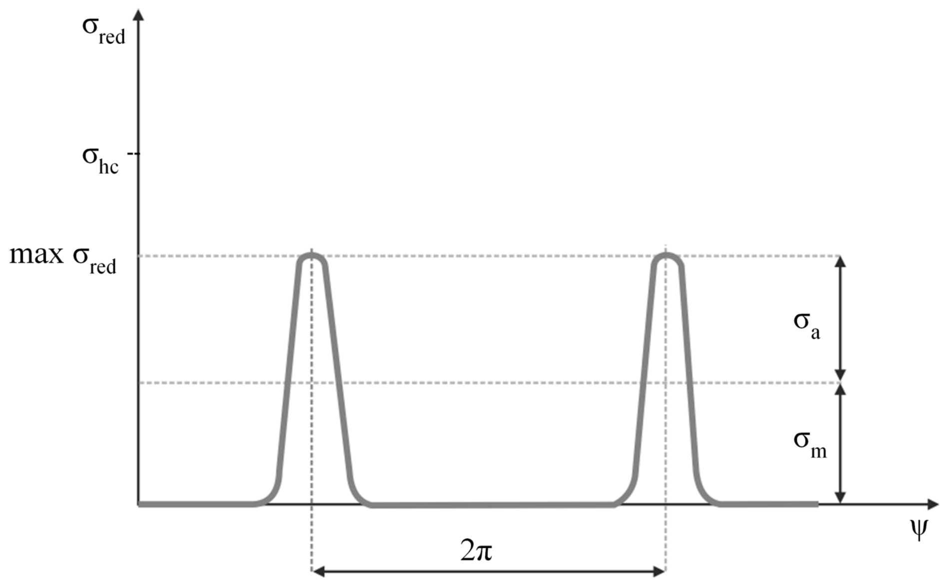

Fig. 1: Load course on the contact areas of the cam surface [1]

The rolling contact fatigue (RCF) occurs in functional surfaces that are exposed to repeated loading (high local pressure) during movement of the functional surfaces of the machine components. We can describe RCF by 3 stages of the material damage [3]:

1. Stage of the material mechanical properties changes
2. Stage of the cracks nucleation
3. Stage of the cracks propagation to fracture

At the loading of the contact surfaces, there are initial changes in properties, cyclic hardening, or softening [4]. As a general rule, the solid materials are reinforced, for example by cold forming, during the fatigue process cyclically softened and, on the contrary, soft materials cyclically hardened. Experiments have shown that cyclic softening occurs for materials with a ratio of strength limit to sliding limit Rm/Re < 1.2 and for cyclic hardening if Rm/Re > 1.4. [5]

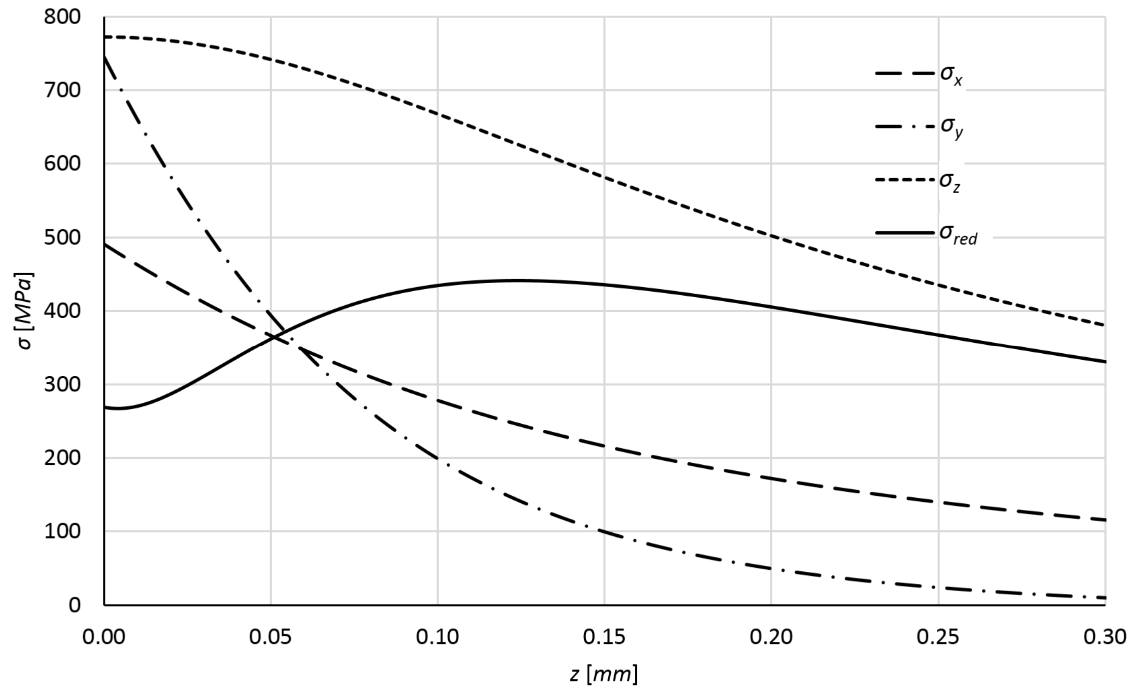

Fig. 2: The state of stress between the crowned roller and the cam in contact [6]

In the next stage, there are such changes in the structure of the material that the first cracks are formed. The crack formation below the surface is connected to the dependence shown in Fig. 2 [6]. There is a maximum of the reduced stress in this area.

If the stress below the surface is high, small plastic deformations can occur in the material and repeated loading may lead to crack formation. These cracks can join together with cracks on the surface and can lead to loosening of the small parts of the surface (see Fig. 3 and Fig. 4). This results in irreversible damage of the contact surfaces and the component must be changed.

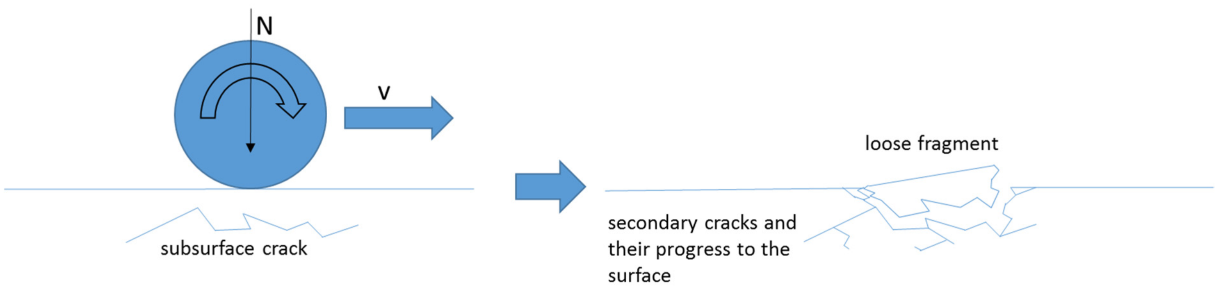

Fig. 3: Pitting formation

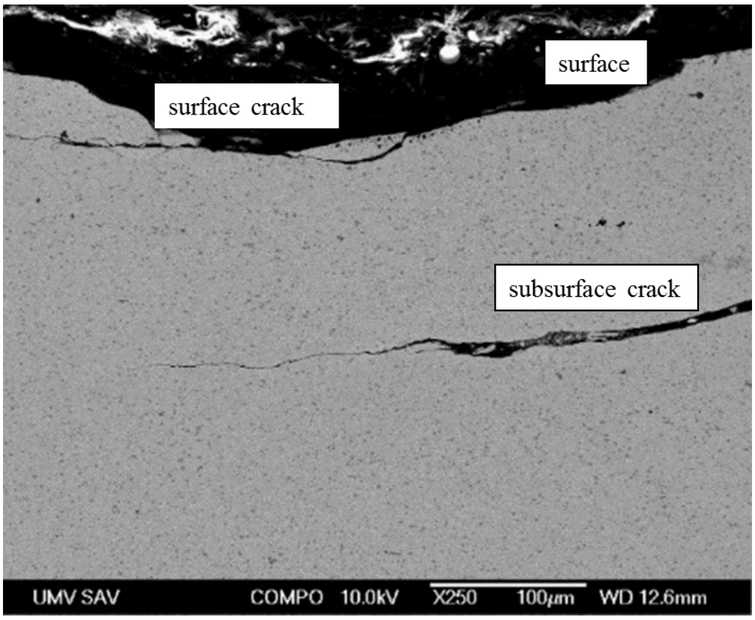

Fig. 4: Detail of the damaged part (acquired in own research)

RCF is very much depend on the mechanical properties of the surface, number and size of impurities and roughness of the surface. The effect of inclusions is showed schematically, acting as a stress concentrator, giving rise to a higher probability of cracking in Fig. 5.

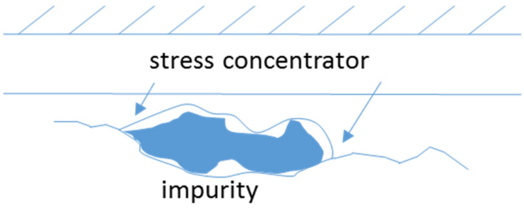

Fig. 5: Material imperfections

Sliding occurs in the operation of the cam mechanism, not just the rolling of the roll on the cam. According to literature [7], a suitable method to protect against fatigue wear is to reduce the coefficient of friction between the two contact surfaces to such a value that the tensile forces do not suffice for delamination during shear or contact fatigue during the shear. Another very important aspect in controlling of the contact fatigue during both sliding and rolling is the purity of the material. Materials with as few inclusions and imperfections as possible should be selected for rolling and sliding. Increasing of the material hardness may also help in some cases, but this method is limited by the embrittlement of the hardened material. Materials for mutual sliding should be selected carefully.

Factors affecting the service life

In general, contact fatigue arises as a result of material imperfection (a material has various constraints and imperfect structure), or working conditions (especially lubrication and load). At the rolling of the body its material must be of high quality, since any imperfection or inclusions manifests as a crack initiation site. The surface of the material should also be of high quality (i.e. as smooth as possible, without bumps) to prevent crack propagation from the surface. Lubrication also has a significant impact on contact fatigue. However, we must not forget other working conditions, such as changes in load (stress) and slippage during sliding, which are related to changes in the relative velocities of the bodies in contact. [7]

When the cam and roller are in contact, they do not only roll together, but also slide. The rolling causes repeated very high stress concentrated on a small area close to the contact site. Pressure stress or frictional stress generation in contact significantly contributes to contact fatigue of both surfaces, with both rolling and sliding. It is generally well known that contact fatigue is very sensitive to sliding. Even a small amount of slips accompanying the rolling process will reduce the service life. [7]

Many factors influence material fatigue. It is therefore necessary to consider the whole set of information not only about the material but also on other influences. There is an overview of this information in the Fig. 6.

Some factors can be described precisely by means of relationships, such as the effect of sample size, the stress gradient, the effect of surface quality, etc. Specific relationships can be found in the literature, e.g. [8].

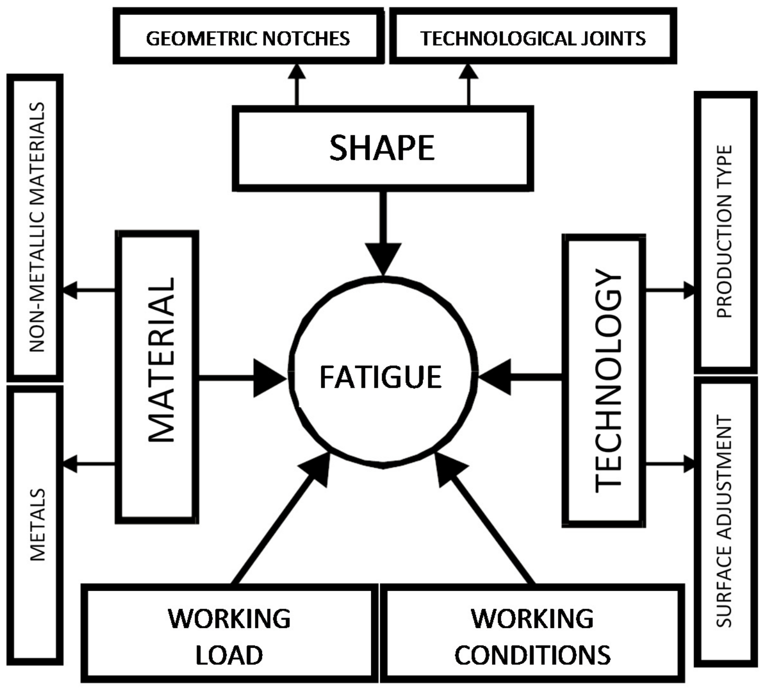

Fig. 6: Main factors affecting the fatigue process [8]

To complete the fatigue curves information, a description of the test conditions is required for the experimental determination. In addition to the type of loading, amplitude and asymmetry of the cycle, information about the sample shape, surface quality, material, and heat treatment is useful. Last but not least, the working environment, sample temperature, type of machine, course of testing, breaks and so on also affect the results of fatigue tests. [8]

Experimental determination of the service life

In addition to the theoretical method of determining the life of the functional surface of the cam [9], it is possible to use the experimental method. This is based on the loading of the test sample surface by the periodic force depending on the angle of rotation of the sample. In this case, the stress on the contact surfaces of the samples has a pulse character. Thus, it is a type of load transient. The loading body represents a cylindrical roller and the loaded body, in analogy with the cam mechanism with radial cam and roller follower corresponds to the cam. The loaded body is stressed by the periodic force N by means of the loading body, the amplitude takes on a size depending on the material of the loaded body and on the method of processing its contact surfaces. A kinematic scheme of the conceptual design of a test rig for testing samples is shown in Fig. 7.

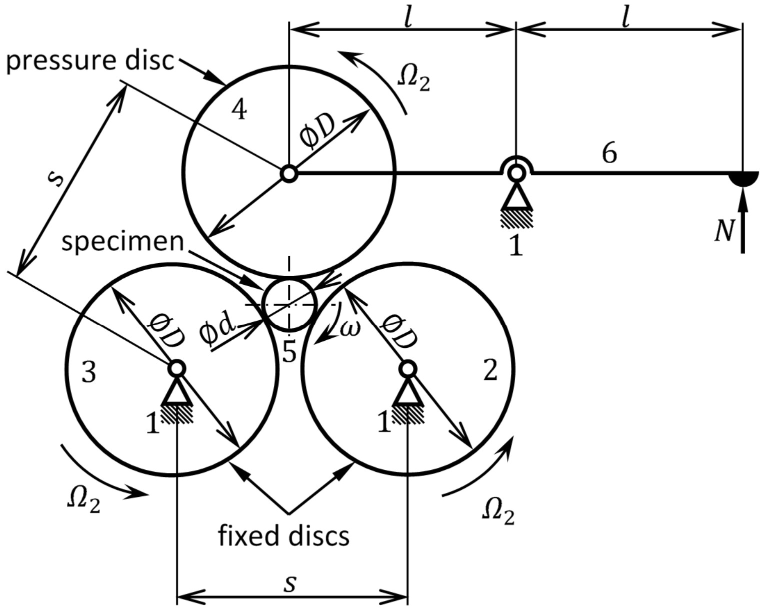

Fig. 7: Kinematic scheme of a test rig

The specimen 5 of the cylindrical shape with an exactly defined width is mounted on the shaft with three rotating discs around it. One of which is pressure (4) and two fixed (2 and 3). The discs are arranged in a common plane at the apices of the equilateral triangle with the sides s, with their rotation axes parallel. Pressure disc 4 is rotatably supported on one of the arms of a pivotally mounted two-arm lever 6 with a length of arms l. The two-arm lever is coupled with its second arm to the loading force N of a defined size. The entire system must be dimensioned due to the maximum load.

In terms of the geometrical arrangement of the load disks and the loaded sample, the sample is loaded with three equally sized N responses from each disk once.

Due to rotation of the test body 5 among three rotating discs rotated with the same angular velocity Ω_2, the loading force N has the character of pulses with a period of $2\pi/3$. Thus, there is a transient type of loading. The three-point contact reduces the testing time to $1/3$, it makes testing much easier as each sample is tested 10^7 to 10^8 cycles. The aim of the experiments is to achieve fatigue damage of the contact area of the specimen after a certain number of cycles.

Assuming the contact of two cylindrical bodies with parallel axes touching the length of l and the length of which is the uniformly distributed loading force N, the condition (5) can be derived for its size.

$$N \leq \frac{l\pi R_{eq}}{E^*}\left(\frac{K_R \cdot R_m}{0.6}\right)^2, \qquad K_R \in \langle 0.55; 0.8\rangle \tag{5}$$

In condition (5) the equivalent radius of curvature of a cylindrical roller of radius R_{eq} in contact with a cylindrical cam of radius r can be expressed as (6),

$$R_{eq} = \frac{Dd}{2(D+d)} \tag{6}$$

where d is the diameter of the sample and D is the diameter of the disks. The constant E^* defines the material elasticity of the bodies in the contact being defined by the formula (7),

$$\frac{1}{E^*} = \sum_i \frac{1-\nu_i^2}{E_i}, \qquad i = 1{,}2 \tag{7}$$

where constants μ_i are Poisson´s ratio and E_i Young's modulus of elasticity of the given body i. In our case it is the specimen and discs. Parameter R_m represents the ultimate tensile strength of the material from which the specimen is made and the constant K_R expresses the ratio of the yield tensile strength $R_{p0.2}$ just to the ultimate strength. [9]

The influence of the shape of the discs profile on the stress of the specimens

The geometrical shape of the disc profile itself has a significant effect on the stress distribution, due to the load in the contact areas of the specimen. The contact of the disc and specimen forms the general kinematic pair. In this case, the contact stress on the surface of the contact surfaces of the bodies in contact with and under the respective surface is periodic, which results from the method of loading. The applied stresses are transient, with pulses having a period of $2\pi/3$ in the case of specimens and 2π in the case of discs.

It is possible to use both the conclusions of the contact mechanics [10] for the respective case of the contact of two elastic bodies as well as the possibility of the finite element method for the calculation of contact stresses. Assuming that the shapes of the specimens and discs would be cylindrical, it can be used to determine the distribution of the contact stress of the results of Hertzian contact theory for contacting cylindrical bodies with parallel axes [10], but with some limitations, as will be explained below. The finite element method is particularly useful when it comes to contact of the sample and the disc with a profile shape other than cylindrical or convex.

In the case of conventional cylindrical discs, there are discontinuities in the intersections of the cylindrical profile with the specimen profile, i.e. if one contact part is axially shorter than the other, as well as in the chamfer of the disc edges, see Fig. 8. In the vicinity of those profile discontinuities, the contact between the disc and the specimen cannot be considered to be straightforward to which Hertzian theory of contact can be applied, but to be a more complex three-dimensional type of contact. These discontinuities cause a very rapid increase in the pressure distribution in the respective contact area of the bodies. In fact, these local increases in pressure distribution can exceed the strength limit of the given material, thereby causing plastic deformations, residual stresses in the material or steel hardening. Furthermore, the area of question will be more susceptible to fatigue damage to the contact surfaces, i.e. to pitting or spalling of the material.

In order to reduce the excess edge stress in the case of the cylindrical discs, it is possible to achieve a crown axial cross section with such a shape that includes a straight line and a single circular arc or a combination of several circular arcs, see Fig. 8. However, such shape of a disc crown whose one segment is cylindrical and the subsequent segment thereof is convex, leads to a certain concentration of stress in the transition from the cylindrical section to the convex one.

According to [11], the logarithmic profile of the disc crown, it can achieve uniform stress distributions for different load levels of the general kinematic pair, Fig. 8. A characteristic feature of this profile is that it descends monotonously from its centre to the edge according to the logarithmic function. It should be noted here that the production of the general profile of the disc crown is a technologically demanding issue in the required precision and quality.

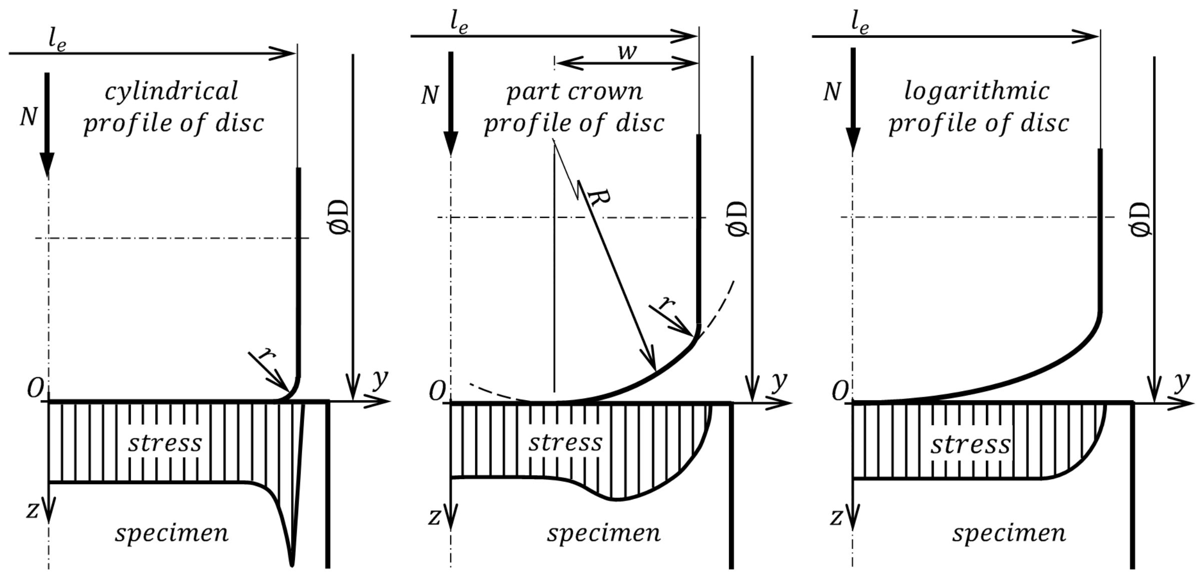

Fig. 8: Schematic drawing of the contact of the various discs profiles with the specimen

From the nature of the course and the results of the tests, it is necessary that the distribution of contact stresses in the contact area and its vicinity of the sample is as even as possible. This requirement is also placed on the stress in the vicinity of the shape discontinuities, which are apparent in the cylindrical profile of the disc, see Fig. 9. For this reason, attention was paid to the influence of the shape of the loading discs profile on the stress in the test specimen. The effect of the discs was analyzed by FEM. In the foregoing we have mentioned that the logarithmic profile of the disc crown is advantageous both in terms of uniform distribution of contact stress in the contact area but also in defining of the disc and specimen misalignment. However, this type of profile is extremely difficult to manufacture. A certain compromise solution seems to be the design of the disc with convex crown segments, with the middle crown section being cylindrical, see Fig. 9 - $Profile\ A$. The convex segments are characterized by their radius of curvature ρ_y in the plane of Oyz, whose size is among other things the object of optimizing the shape of the discs. The characteristic dimensions of such a disc are its diameter D, the radius of the convex part of the crown $\rho_y \equiv R$, the width of the convex part w and the radius of the rounding edge r. Another optimization parameter of such a crown profile is, besides the radius of the convex section $\rho_y \equiv R$, its width w. The optimization of the crown shape itself is carried out in order to distribute the contact stress as evenly as possible with the greatest possible load exerted by the effect of the force N.

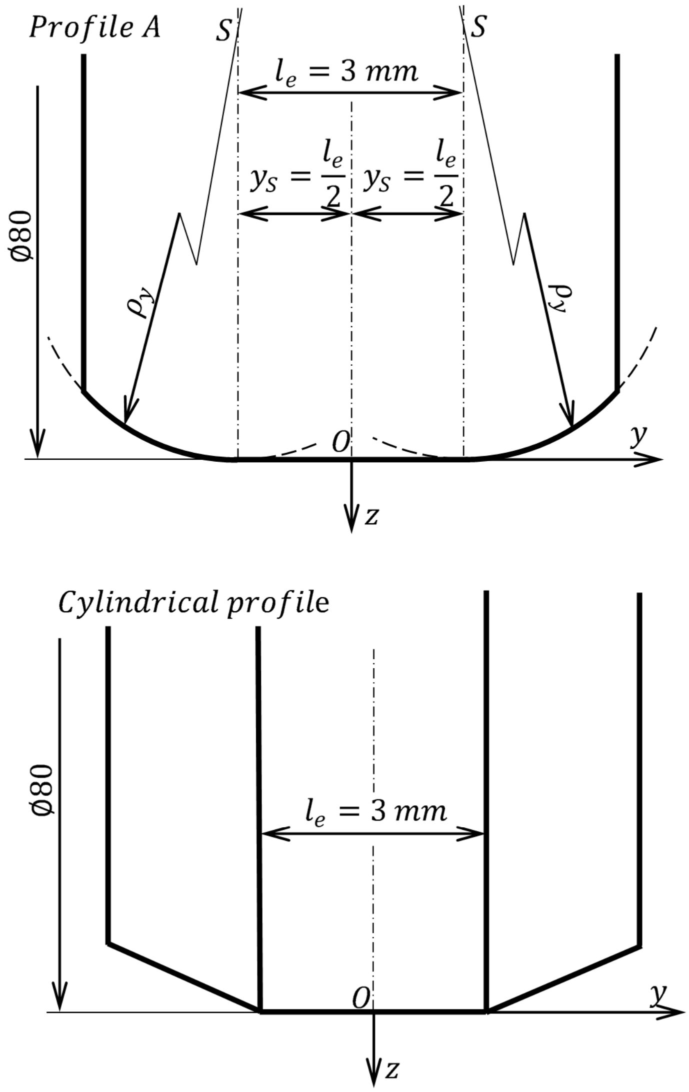

Fig. 9: Schematic drawing of the various discs profiles

The types of disc profiles according to the schematic representation shown in Fig. 9 were analyzed in detail. Each of the said disc types comprises of a cylindrical part, which is connected to either convex segments or conical segments. In summary, the typical disc dimensions are included in Table 1. The material parameters of the load discs and the test specimen are included in Table 2.

Table 1: Characteristic dimensions of the discs

		Profile A	Cylindrical profile
Diameter	$D\ [mm]$	82.4	82.4
Length of cylindrical part	$l_e\ [mm]$	3.0	3.0
Fillet radius	$\rho_y\ [mm]$	200	$-$

Table 2: Characteristic material parameters

		Disc: 90MnCrV8	Specimen: 16MnCr2
Density	$\rho\ [kgm^{-3}]$	7850	7850
Young's modulus of elasticity	$E\ [GPa]$	210	206
Shear modulus	$G\ [GPa]$	80	79
Poisson's ratio	$\nu\ [-]$	0.3125	0.3038

In the model, the respective surface of the disk was subjected to a surface load induced by the force effects of (8).

$$N = 4000\ N \tag{8}$$

Based on the analyses, the distribution of the contact stress was determined in the contact area and its vicinity induced effect of the load force N.

Fig. 10 shows the course of the maximum reduced stress in a depth of z_e depending on the half width of the specimen for all types of discs in contact with the cam. Depth z_e expresses the depth under the surface of the cam where just the maximum value of the reduced stress depending on the distance from the contact surface is reached.

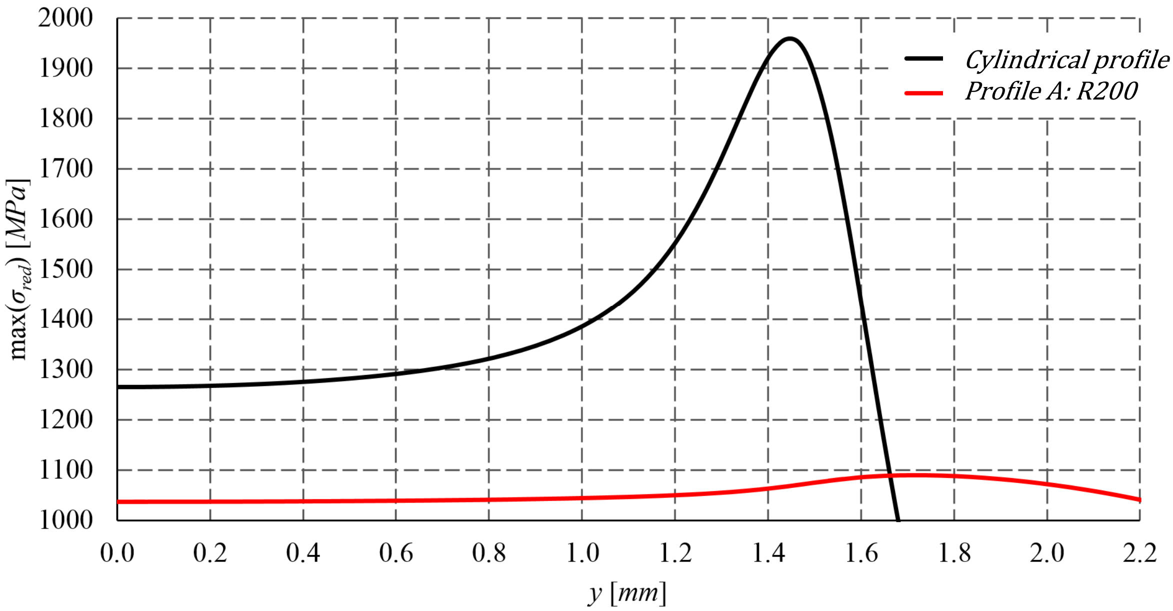

Fig. 10: Course of the maximum reduced stress depending on the width of the specimen

Fig. 11 shows the distribution of the reduced stress induced by the contact of the specimen and the disc with a radius of curvature $\rho_y = 200\ mm$ in the case of $profile\ A$ and the cylindrical disc. It can be seen from the above figures that in the vicinity of the profile discontinuities, the magnitude of reduced stresses increases. This is particularly evident in the case of a cylindrical disc, where the contact between the disc and the specimen cannot be considered simply straight, but rather a more complex three-dimensional contact type. It is also clear that a significantly more uniform stress distribution is achieved by the shape of the disc according to Fig. 9 - $Profile\ A$. We have also chosen this type in terms of acceptable technological demands of its production in the required quality and precision.

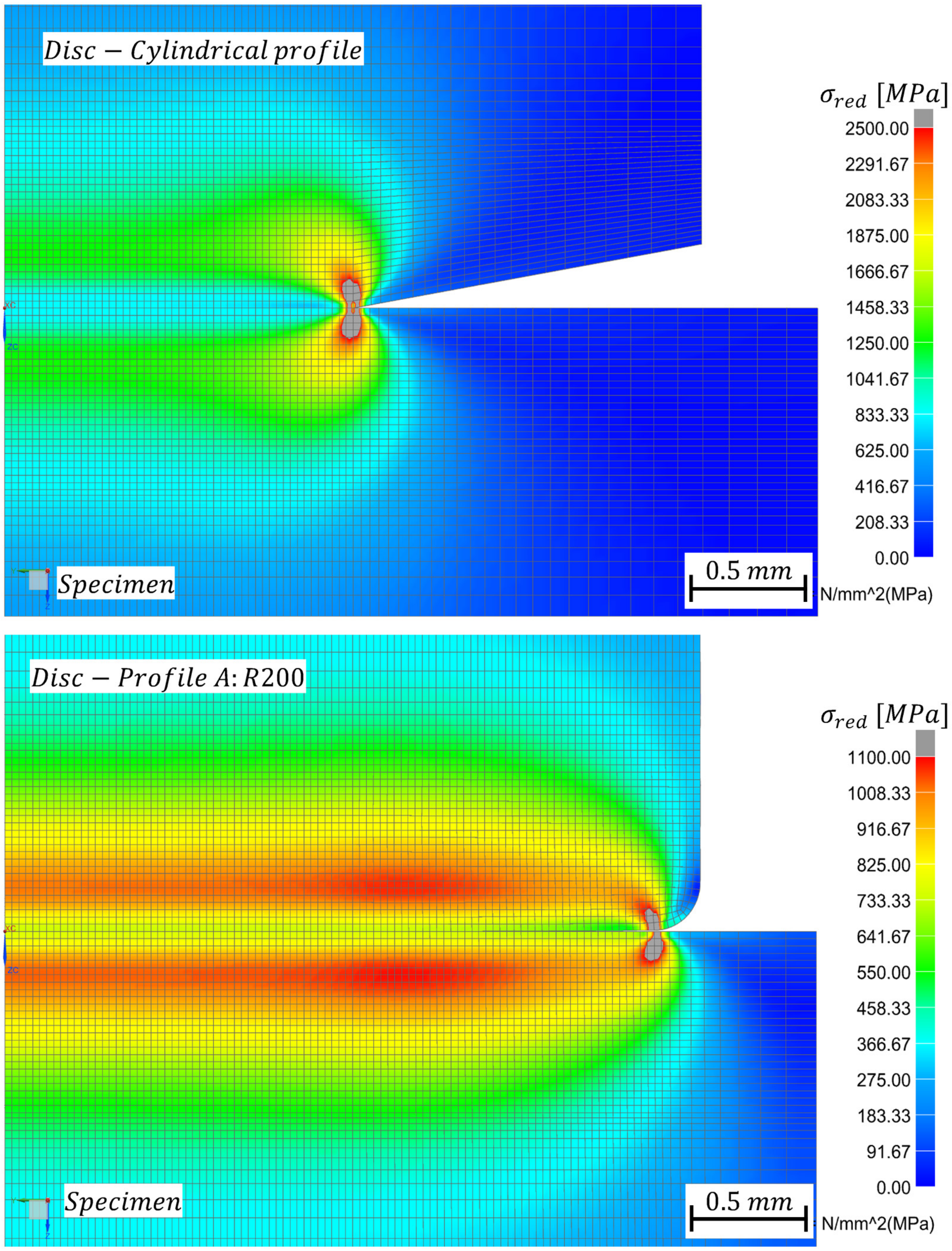

Fig. 11: Distribution of the reduced stress in the contact area of disc and specimen

Conclusions

This paper discusses a design of a test rig for testing materials intended for the manufacture of cam mechanisms. It allows us to respect and test a whole range of parameters. It simulates cam and roller contact. However the test runs three times faster than the real cam mechanism.

Suggested test rig allows testing under the following conditions:

- working conditions
    - Slide Roll Ratio (0% (pure rolling) to +/- 200% (pure sliding))
    - Lubrication (no lubrication or influence of various types of lubricants)
    - Temperature (at normal operating temperature or reduced / increased)

- steels (types, production technology, with / without (chemically-)heat treated layer)
- other materials (non-metallic materials, composites, etc.)
- coatings, finishes

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