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THE ESTIMATION OF OIL LEAKAGE IN HYDRAULIC CYLINDERS USING FEM

WYZNACZANIE PRZECIEKU W SIŁOWNIKACH HYDRAULICZNYCH Z WYKORZYSTANIEM MES

A b s t r a c t

The paper presents FEA of common U-type elastomeric seal used in hydraulic cylinders of heavy duty machines. The achieved results of FE analysis allowed to predict leakage rate for different sets of operating conditions, and can also be useful in studies concerning wearing processes.

Keywords: hydraulic cylinder sealing, leakage estimation

S t r e s z c z e n i e

W artykule zaprezentowano wyniki analizy MES popularnego pierścienia uszczelniającego typu *U*, stosowanego w siłownikach hydraulicznych maszyn roboczych. Uzyskane wyniki pozwalają na oszacowanie przecieku dla różnych warunków pracy oraz mogą być użyteczne w pracach związanych z procesami zużycia uszczelnień.

Słowa kluczowe: uszczelnienia siłowników hydraulicznych, wyznaczanie przecieku

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1. Introduction

Hydraulic cylinders are one of the basic components in the drives of heavy duty machines equipment.

One of the main directions in R&D of hydraulic cylinders is the sealing system of the cylinder, especially the sealing of a piston rod. The development is focused on the elimination of oil leakage into the environment, the reduction of friction and extension of operation time. The proper design and assembly of sealing components enable the extension of durability and operational reliability. These are important factors in cases when a large number of heavy duty machines is used, e.g. in mines, at construction companies and many others. Even minor developments allow savings thanks to the reduction of maintenance time.

The recent years have shown that another important factor which must be taken into consideration at the design stage is the one concerning environmental protection. A hydraulic oil leakage is being constantly reduced due to the application of new materials and new sealing shapes, however the development usually requires many experimental tests which are time-consuming and expensive.

The paper shows the results of a FEM analysis of common U-type elastomeric sealing ring and introduces a seal leakage estimation method utilizing FEM and IHL theory.

The use of CAE tools allow engineers to design sealing components more efficiently and minimizes the risk of undesired prototype features resulting in the necessity of redesigning and repetitive testing.

The traditional manufacturing procedure of sealing components consists of sequenced phases: designing, making prototype (this involves cutting of a mold), testing and, finally, deciding on the possible range of changes. Every change in shape of the seal requires the repetition of the procedure. This approach is time-consuming and relatively expensive.

Modern simulation tools allow the rearrangement of the traditional manufacturing process (Fig. 1).

After the preliminary design is made, the FEM tools can be employed in order to check if the product meets the given requirements. If changes are to be made, they are made in the design phase, and most importantly, before cutting the mold. The next steps which are the same as in the traditional procedure mostly demonstrate compliance with the initial assumptions and requirements.

An analysis of behaviour of hyperelastic rubberlike materials is a complex engineering problem. The material nonlinearities, large deformations, friction and possibility of self to self contact require special tools to solve a particular task.

The process of selection of the proper material for a given application can be also challenging. The selection which was often based on engineers' knowledge and experience, available technology, etc., can be improved using CAE simulation tools .

To conduct FEM analysis shown in this paper ANSYS system has been used. The subsequent step, consisting of an analysis of FEA output data and results, was performed in MAPLE software package.

To describe behaviour of elastomeric material used in the present analysis, the three parameter Mooney-Rivlin hyperelastic material model has been used.

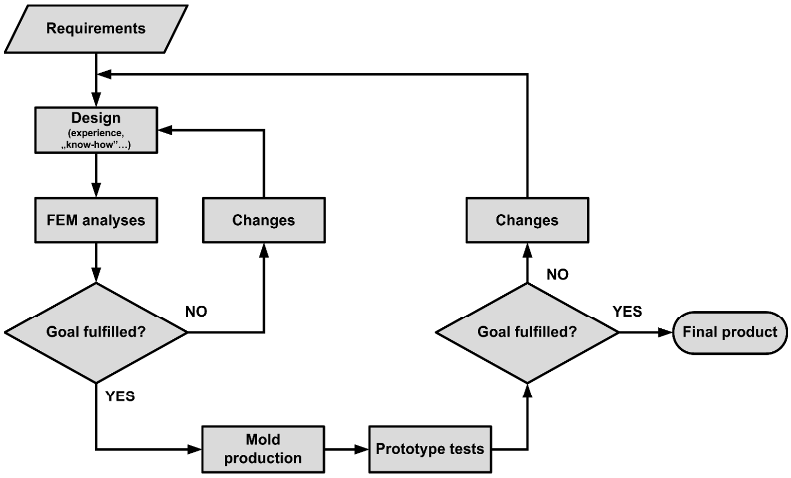

Fig. 1. Block diagram of the modern approach to the sealing manufacturing [12]

Rys. 1. Schemat blokowy nowoczesnego podejścia w produkcji uszczelnień [12]

1.1. Hyperelastic Material Model

Hyperelasticity refers to a specific type of materials that can undergo large elastic strain, without losing initial properties.

There are a few models (e.g.: Arruda-Boyce, Mooney-Rivlin, Neo-Hookean) describing hyperelastic materials which are employed in FEM systems. The literature survey has shown that the Mooney-Rivlin model gives satisfactory results for applications like the one shown in the article - elastomeric seals or other selected rubber elements (e.g. car tyres).

It has to be clearly stated here, that the presented FE model, does not take the phenomena related to cyclic loading and unloading into account.

According to [4] and [7], the constitutive description of hyperelastic materials is based on strain energy density function. The stresses can be calculated from the derivatives of the strain energy density function W to the strains:

$$S = \frac{\partial W}{\partial E} = 2\frac{\partial W}{\partial C} \tag{1}$$

The E is the Green-Lagrange strain tensor and the C is the Right Cauchy-Green stretch tensor.

Under the assumption that material is isotropic, it is convenient to express the strain energy density function in terms of strain invariants I or principal stretches λ:

$$W = W(I_1, I_2, I_3) = W(\lambda_1, \lambda_2, \lambda_3) \tag{2}$$

The hyperelastic materials are usually considered as purely or nearly incompressible. Thus, the function W is split into a deviatoric part and hydrostatic part. The deviatoric part describes a constant volume deformation and the hydrostatic part describes a uniform compression of expansion. To fulfill the separation of the deviatoric and hydrostatic part, a modified set J of deformation measures is used.

Finally, the strain energy density function can be defined as:

$$W = W(J_1, J_2, J) = W(\bar{\lambda}_1, \bar{\lambda}_2, \bar{\lambda}_3, J) \tag{3}$$

It is worth noting here, that for purely incompressible materials, the third principal invariant (volume change ratio) $I_3 = 1$ and the new measures do not differ from the originals. For an incompressible, initially isotropic material, the Rivlin formulation can be used to construct any type of strain energy density function. This formulation can also be used for nearly incompressible materials if the principal invariants I_1 and I_2 are substituted by the modified invariants J_1 and J_2. The three parameter Mooney-Rivlin model used in FEM analyses can be derived from Rivlin formulation and it is given as follows:

$$W = \mathrm{c}_{10}(J_1 - 3) + \mathrm{c}_{01}(J_2 - 3) + \mathrm{c}_{11}(J_1 - 3)(J_2 - 3) + \tfrac{1}{d}(J - 1)^2 \tag{4}$$

The c_{ij} are material constants, determined form physical tests and d is material incompressibility parameter.

1.2. The Object of Research

A standard U-type sealing ring for rod diameter of 50 mm has been examined. The seal is made of TPU (Thermpolastic Polyurethane). The coefficients of the material model and other properties were taken from literary sources [8].

Standard gland with sealing points indicated is shown in Fig. 2.

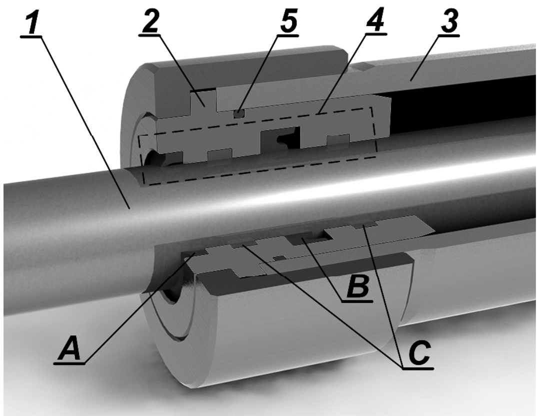

Fig. 2. A fragment of hydraulic cylinder cross-section: 1 – piston rod, 2 – gland, 3 – cylinder, 4 – piston rod dynamic sealing, 5 – piston rod gland static sealing, A – wiper, B – sealing ring, C –bearings

Rys. 2. Fragment przekroju siłownika hydraulicznego: 1 – tłoczysko, 2 – dławnica, 3 – cylinder, 4 – uszczelnienie dynamiczne, 5 – uszczelnienie statyczne, A – pierścień zgarniający, B – pierścień uszczelniający, C – pierścienie prowadzące

2. FEA Investigation of the Sealing Ring

The model was created as an APDL (ANSYS Parametric Design Language) batch file for the ANSYS system. An axisymmetric 2D model of a cylinder sealing was used in analysis. The geometry of the piston rod, seal and housing was parametrized, in order to be easily modified.

The mesh of the seal was created using PLANE182 (2-D 4-node structural solid quadrilateral) element type with mixed u-p formulation activated. Contact pairs consist of CONTA171 and TARGE169 elements with both initial geometrical penetration and offset included. Each iteration of contact analysis is based on current mean stress of underlying elements.

The Augmented Lagrange method was used in the contact analysis. This contact algorithm minimizes penetration and is conditioned better than the Pure Penalty method. It is also less sensitive to contact stiffness value, but may require more iterations than the Pure Penalty method. In typical applications this method has proven to produce good quality results [7, 9].

To avoid problems with solution convergence and to obtain accurate results, the mesh was refined in critical areas. A fine mesh can be found near to the initial contact and closure areas shown in Fig. 3.

Geometry of housing was simplified and in general consists of non deformable rigid walls. The initial interference fit of seal was performed by applying radial displacement to the piston rod and housing walls. Due to the large deformation for both the interference fit and loading process, the total displacement and total pressure were applied in series of load steps. Direct application of displacement or pressure may lead to convergence problems and solution instability.

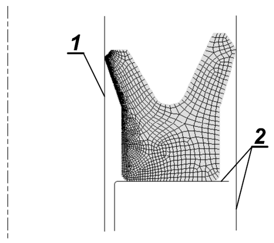

Fig. 3. FE mesh applied on an axisymmetric seal model: 1 – piston rod, 2 – groove

Rys. 3. Siatka elementów skończonych nałożona na osiowosymetryczny model uszczelnienia: 1 – tłoczysko, 2 – rowek

As one of the results obtained from the conducted FEM analysis, Von Mises stress distribution over sealing ring pressurized to 16 MPa is shown in Fig. 4.

Except for fluid leakage estimation, the presented model also allows to study the influence of the closure size on the process of extrusion of the seal material.

An analysis of the sealing under applied pressure shows the deformation processes and stress concentrations that can influence seal wearing. Large deformation and extensive extrusion of seal material into closure can lead to damage in this area. Due to higher than average contact pressure, also intensified abrasive wear occurs near the closure area.

Depending on working conditions, material properties, and groove geometry, the abrasive wear and cutting of outer layer may appear with various intensity. Generally, seals made of less harder material are less endangered for abrasive wear but more for cutting by groove edge.

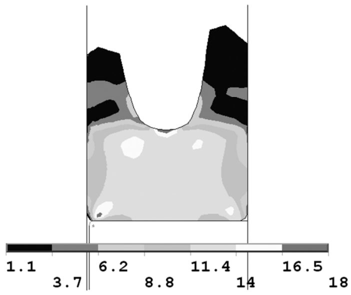

Fig. 4. Von Mises stress distribution over U-type sealing ring cross-section [MPa]

Rys. 4. Rozkład naprężeń na przekroju pierścienia typu U [MPa]

3. Estimation of Fluid Leakage Rate

The fluid leakage rate estimation shown in this work is based on the Inverse Hydrodynamic Lubrication theory (IHL). It is assumed that fluid film does not affect contact pressure (lubricating film thickness is much smaller than elastic deformations of the seal). This approach is considered to be useful for high pressure applications as the one shown in the article [1–3], [6].

Experiments have shown that for modern high pressure seals, static and dynamic pressure gradients do not show major differences in both directions of motion [1].

The leakage rate can be derived using instroke and outstroke sealing gap height at the point where the pressure value reaches maximum. The gap heights can be determined from:

$$h_{\mathrm{id0/od0}} = 0,94\left(\frac{\eta u_{\mathrm{i/o}}}{g_{\mathrm{i/o}}}\right)^{0.5} \tag{5}$$

The i/o indices in Eq. 5 indicate concerned direction of motion (instroke and outstroke of piston rod), g is maximum gradient of sealing pressure, η – dynamic oil viscosity and u – rod velocity.

The fluid velocity distribution in the gap at the point of maximum pressure is linear, and has constant value of $u_{\mathrm{i/o}}$ outside the gap. Thus, assuming flow continuity, the following relation for determination of oil film thickness on outgoing and retracting rod is valid:

$$h_{\mathrm{i/o}} = \frac{1}{2} h_{\mathrm{id0/od0}} \tag{6}$$

Assuming $h_{\mathrm{i}} < h_{\mathrm{o}}$, the leakage rate for N cycles can be determined from:

$$V = N\pi dL\left(h_{\mathrm{o}} - h_{\mathrm{i}}\right) = N\pi dL\sqrt{\frac{2}{9}\eta}\left(\sqrt{\frac{u_{\mathrm{o}}}{g_{\mathrm{o}}}} - \sqrt{\frac{u_{\mathrm{i}}}{g_{\mathrm{i}}}}\right) \tag{7}$$

where: d is rod diameter and L stroke length.
The lubricating film thickness can be derived from the Reynolds equation:

$$\frac{dp}{dx} = 6\eta u \frac{h(x) - h_{\mathrm{id0/od0}}}{h^3(x)} \tag{8}$$

The $h(x)$ in the equation above stands for the current height of lubrication gap.

4. Conclusions

The paper presents a FEM analysis of common U-type elastomeric seal used in hydraulic cylinders of heavy duty machines.

The achieved results allow to predict and improve operational sealing capabilities and can also be useful in works regarding wearing processes.

The fluid leakage rate has been estimated, using IHL theory. Assumed simplifications caused higher than initially expected value of leakage, nevertheless the final results are considered quite satisfactory.

The majority of researched scientific papers regarding this method show that the pressure profiles were obtained in time-consuming experiments rather than from FEA. It seems that improving FEA towards more realistic models could possibly result in a step forward in the development of sealing solutions.

The performed models and achieved results may be used in further, more developed analyses covering e.g. different thermal conditions.

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