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TECHNOLOGIES ASSURING THE SERVICE PROPERTIES OF FRICTION PAIRS WITH CELLULAR MICRORELIEF SURFACES

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A B S T R A C T

Article focuses on the improvement of the technologies used to improve the durability of friction pair components. The authors use the piston compressor to study cellular microrelief surfaces of cylindrical components. The cells are shaped as elliptic paraboloid with uneven positive parameters. The use of cellular microrelief surfaces is highly preferred as they reduce the attrition wear of the friction pairs through assuring the hydrodynamic load capacity of the lubrication layer with the shape of the microrelief. The research goals included the parametric analysis of the lubrication layer behavior in the gap between the microrelief cells. To do this, the authors developed an analytical model based on the theory of hydrodynamic lubrication and constructed a CFD model using the ANSYS Fluent software. To contain the transfer equations, the authors used the turbulence model SST *k–ω*. Both models showed that the maximum hydrodynamic load capacity coincided with the 75%-length of the major axis of the elliptic cell, which also corresponds to 0.128 mm in cell depth. The maximum lifting hydrodynamic pressure on one microrelief cell amounted to 3 kPa. Based on the results of the parametric analysis, the authors claim that the cellular microrelief can be efficiently used to assure the service properties of friction pairs in process units.

K E Y W O R D S

Friction pair;
Cylinder sleeve;
Piston ring;
Cellular microrelief;
Hydrodynamic model;
Mathematical model;
ANSYS Fluent;
Two-dimensional parametric analysis.

Introduction

The data obtained from the operation of process units in the oil and gas industry show that the rubbing surfaces of their components often have insufficient lubrication, which results in the formation of burrs, as well as the chafing and jamming of these rubbing surfaces [1]. Therefore, motors, pumps, and compressors often experience friction pair wear [2] including such pairs as the cylinder sleeve-piston ring; shaft-slider bearing, piston pin-rocker arm, etc. Heavy friction wear has a huge impact on the reliability of the friction pairs and their service properties. The pair cylinder-piston ring has the highest mechanical losses and the most high-wear joints. In particular, the review of publications shows that about half of the mechanical losses in car engines are caused by friction in the cylinder-piston group [3]. The impacts of high-temperature and pressure on those friction pairs result in a complex wear mechanism associated with adhesion, abrasion, etc., which causes frequent failures of these friction pairs and units [4]. Unit failures may result in accidents and significant economic losses [5].

There is practically no more potential design changes that could improve the lubrication feed to the friction surfaces without complicating unit design, or they are economically unfeasible. For instance, there are significant design restrictions due to the well operation, which is crucial for the friction pairs of the rod well pump. One of the few potential solutions is improving the retention of the lubricant on the rubbing surfaces [6]. To implement this solution, we need to optimize the microrelief of the working surfaces of the rubbing components using various relief-forming technologies with essentially unlimited control over all of the

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geometric parameters of the microrelief formed [7-9]. When the operating component, e.g. the piston ring, approaches the microrelief component, its lubricant pressure initially drops and then increases, which creates a pressure differential and forms its load capacity. One of the advantages of this method is that it can be implemented during equipment repairs, e.g. plating the cylindrical surface of the component [10].

Modeling the surface geometry is becoming increasingly popular because it is possible to relatively easily improve the tribological properties of friction pairs via the changing of the surface geometry. The study of various microrelief geometries and their tribological properties has both the theoretical and applied value for design and producing relevant geometries [11]. To study the friction between the piston (piston ring) and the cylinder sleeve using the interchanging surface geometries, several different models have been developed. In particular, the authors of [12,13] claim that the efficiency of lubrication could be higher if the piston ring had a proper microrelief of parabolic pits covering 15% of the surface area.

The cyclic movement of the operating component of the machine results in a gradual wear of any given microrelief type at the periphery along the movement axis of the component due to the formation of increased contact pressure areas. This results in the deterioration of the initial microrelief into the cellular form, i.e. the deterioration of the rotational paraboloid into the elliptic paraboloid with uneven positive parameters. This was discovered in the operation experience and experimental research [6, 14]. Thus, a significant proportion of material is worn-out during the machine attrition and operation. This results in the deterioration of intended service properties of the microrelief and the cylindrical surface, the secondary flaking, and other adverse effects. Experiments showed that the initially-set elliptical microrelief reduces the attrition wear [14]. That being said, there is no data on the service properties of cylindrical surfaces with the cellular microrelief in current publications. The service properties of the microrelief surface are primarily assured by the hydrodynamics of the lubrication layer, the hydrodynamic load capacity that occurs due to the microrelief form.

Based on the above, the goal of this research is to analyze the service properties of the cylindrical surfaces with the cellular microrelief. Thus, we determined the following research objectives: creating the analytical and the computer hydrodynamic models (CFD) for the behavior of the lubrication layer on the cylindrical surface with the elliptic cell microrelief, their analysis, and result comparison.

Analytical model

The theory of hydrodynamic lubrication is based on the Reynolds differential equation [15]:

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(h^3\frac{\partial p}{\partial y}\right)=6\mu\nu\frac{dh}{dx}+12\mu U \tag{1}$$

where x, y are the coordinates for the tangential to the internal surface of the bearing recess and for the thickness of the lubrication layer; $p=p(x, y)$ is the pressure in the lubrication layer; ν is the rubbing surface speed projection to the axis x; U is the closing speed of the rubbing surfaces; μ is the dynamic viscosity of the liquid; $h(x)$ is the lubrication layer thickness function.

Assuming there is no movement along the y-coordinate, and the respective pressure value does not change, equation (1) shall look as [16]:

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)=6\mu\nu\frac{dh}{dx} \tag{2}$$

Its integral:

$$\frac{dp}{dx}=6\mu\nu\frac{h-h_0}{h^3} \tag{3}$$

where h is the gap height; h_0 is the gap height at which the pressure gradient along the gap length $dp/dx=0$ and $p=\max$.

Following the procedures described in [17] and taking into account (2), along with the calculation model in figure 1, we obtained expression (4) describing the distribution of the hydrodynamic oil pressure for the arbitrary (integrated) function of the lubrication layer thickness $h(x)$ over the length L:

$$p(x)=6\mu\nu\left[\frac{\int_0^L\frac{dx}{h^2(x)}}{\int_0^L\frac{dx}{h^3(x)}}\int_0^x\frac{dx}{h^3(x)}-\int_0^x\frac{dx}{h^2(x)}\right] \tag{4}$$

Using figure 1, we constructed equations describing the gap profile in a non-dimensional form. To simplify the task, as the sizes are quite small, assume that the lubrication layer thickness function is linear. Then, $h(x)$ shall be described with a system of equations:

$$h(x)=\begin{cases}\frac{xh_p}{l}+h_0, & x\in[0;l];\\ -\frac{xh_p}{l}+h_0, & x\in[l;L].\end{cases} \tag{5}$$

Inserting the obtained integral expressions in expression (4), we get the system of equations describing the distribution of the hydrodynamic oil pressure in the gap for two profile sections:

$$p(x)=6\mu\nu\begin{cases}\frac{h_plx(l-x)}{(2h_0+h_p)(h_0l+h_px)^2}, & x\in[0;l];\\ \frac{h_pl(2l^2-3lx+x^2)}{(2h_0+h_p)(h_0l+2h_pl-h_px)^2}, & x\in[l;L].\end{cases} \tag{6}$$

Finding the linear (unit-width) hydrodynamic load capacity P_n of the elliptic cell. As the profile gap is symmetrical, it will be sufficient to find the load capacity for half of the profile. Integrate expression (6) using variable x within the limits of 0 and l, and after elementary transformation, we get the load capacity expression for half an elliptic cell profile:

$$P_n=6\mu\nu\int_0^l\frac{h_plx(l-x)}{(2h_0+h_p)(h_0l+h_px)^2}dx= \tag{7}$$

$$=6\mu\nu\frac{l^2\left(2h_p-2h_0\ln\left(\frac{l(h_0+h_p)}{h_p}\right)-h_p\ln\left(\frac{l(h_0+h_p)}{h_p}\right)+2h_0\ln\left(\frac{lh_0}{h_p}\right)+h_p\ln\left(\frac{lh_0}{h_p}\right)\right)}{h_p{}^2(2h_0+h_p)}$$

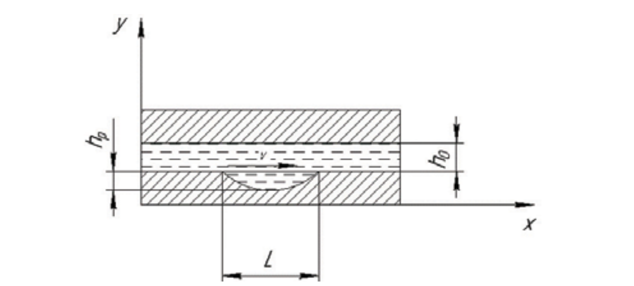

Fig. 1. The calculation model for the elliptic cell

According to GOST 25347-2013 and [18], the use of the running fit H7/f7 is preferred for compressor pistons and cylinders. For a cylindrical joint with a diameter of 280 mm, we have the upper cylinder displacement of +0.052 mm, the lower displacement of 0, the upper displacement of the piston ring of -0.056, and the lower displacement of -0.108 mm. The tolerance range for the cylinder-piston system is shown in figure 2.

For the gap h_0, take average gap S_c, then $h_0 = 0.108$ mm.

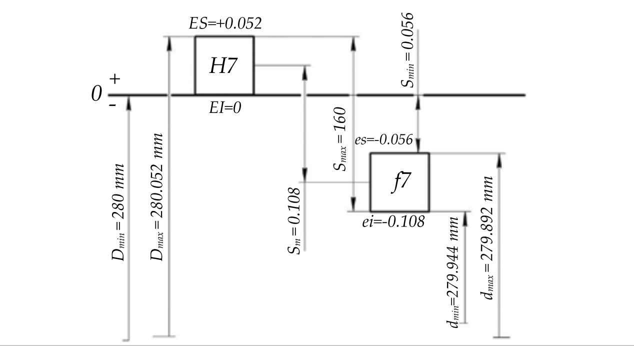

Fig. 2. Tolerance range location

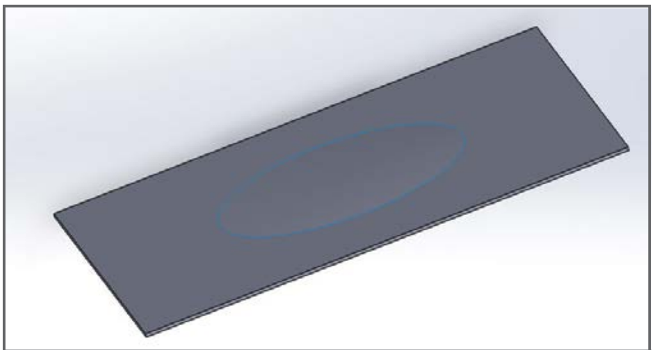

Fig. 3. The 2D geometry of the computational region

Table 1

Geometry of cells

Parameter	Value
Maximum cell depth, mm	0.260
Minor axis, mm	3.78
Major axis, mm	8.0
Gap height, mm	0.108

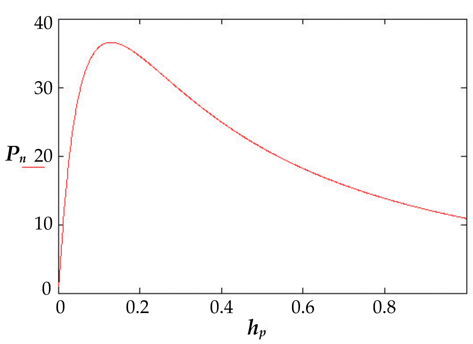

Fig. 4. The dependency of the linear hydrodynamic load capacity of the lubricant and the current depth h_p of the elliptic cell

CFD analysis methodology

The geometrical parameters of an elliptical cell (table 1) required for the computerized hydrodynamic modeling were taken from the mathematical model shown above. The visualization of the computational region is shown in figure 3. It was created using the SolidWorks software.

All of the calculations were carried out based on the finite-volume method using the ANSYS Fluent software. This method is based on the second order of accuracy to solve the transfer equations for both the laminar and turbulent models. To contain the equations, we used the turbulence model of Shear-Stress Transport (SST) *k–ω*. This model assures higher accuracy and reliability of forecasts for currents near the walls because it accounts for the transfer of turbulent tangential strains and assures accurate flow forecasts for cases with adverse pressure gradients attributed to flow breakaway. The detailed description of the SST k–ω model and the transfer equations we use is shown in [19]

The computational grid was created. It was calibrated based on the sensitivity analysis based on the grid size independence. The refinement was carried out until the results began to correlate with the values produced in the mathematical model. We used quadrilateral elements with compacted cell sizes from 0.03 mm in the cell to 0.04 mm in the remaining part of the grid. The total number of elements was 36190.

The set properties of the service liquid (oil) were as follows: viscosity – 10 mm^2/s, density – 800 kg/m^3.

The boundary conditions were set on the faces prearranged during the construction of the grid. Since we took zero boundary conditions, the input and output pressure was assumed to be equal to zero. The upper boundary imitates the cylinder sleeve wall, therefore we set it as stationary. The lower boundary imitates the piston ring, therefore we set its speed at 1 m/s.

Results and discussion

As shown in previous research works [17], the maximum load capacity was achieved with the microrelief with a pitch comparable to the set axial height of the piston ring (ceteris paribus). For instance, assume $l = 8$ mm.

Insert values in expression (7), we get the dependency of the linear hydrodynamic load capacity of the lubricant and the current depth of the elliptic cell. The dependency graph is shown in figure 4.

Figure 4 shows that the maximum hydrodynamic load capacity occurs at $h_p = 0.128$ mm.

To check whether the microrelief with a pitch comparable to the set axial height of the piston ring assures the maximum hydrodynamic load capacity, assume the load capacity dependency of two variables, h_p and $l/2$ (fig. 5).

As we can see from figure 5, the maximum linear hydrodynamic load capacity of the lubricant is achieved at $l/2 = 4$ mm. This value cannot be increased as this will result in the reduction of the compression ratio of the piston compressor.

The visualization of the lubricant hydrodynamic load capacity distribution in the gap with the elliptic cell is shown in figure 6.

As we can see from figure 6, the hydrodynamic load capacity $P(x)$ decreases nonlinearly and then increases to 0 over the first half of the section. This happens because the gap height increases along with the movement. After that,

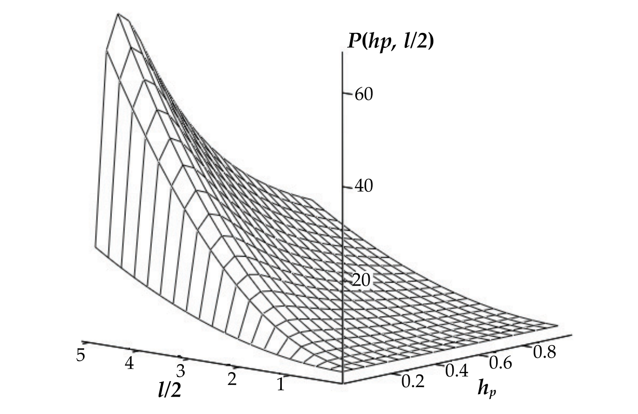

Fig. 5. The dependency of the linear hydrodynamic load capacity and the current depth h_p of the elliptic cell length $l/2$

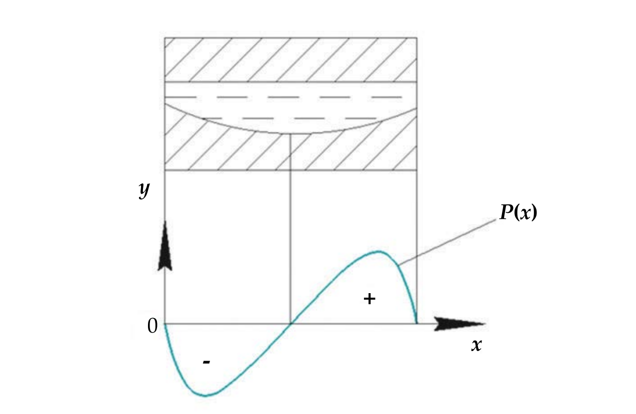

Fig. 6. The distribution of the lubricant hydrodynamic load capacity in the gap with the elliptic cell

the gap decreases, resulting in the increased load capacity. The maximum load capacity is observed on part 21/25 of the major axis of the cell ellipse, after which it decreases to 0.

Using the circuit function in the ANSYS Fluent software, we obtained the distribution of the pressure in the lubrication layer shown in figure 7. For illustrative purposes, we used 100 circuits. The overlaid turquoise curve in figure 7 illustrates the results of the load capacity distribution from the analytical model.

Comparing the results of the CFD-model and the analytical solution, we can assess the adequacy of the result obtained. Since we used zero boundary conditions, the input and output pressure, as well as the pressure in the center of the cell, takes zero as its value. The location of the lubrication layer pressure maximum and minimum also aligns with the similar data on the load capacity obtained from the analytical model as shown in figure 7. The numerical results show that the elliptical cell microrelief applied on the sleeve can potentially reduce the wear of the piston ring because a single cell can provide additional lifting hydrodynamic pressure of 3 kPa. The issue of elliptic cell geometry assuring the highest load capacity shall be tackled in the following papers, as well as the problem of the technology used for the formation of cellular microrelief surfaces.

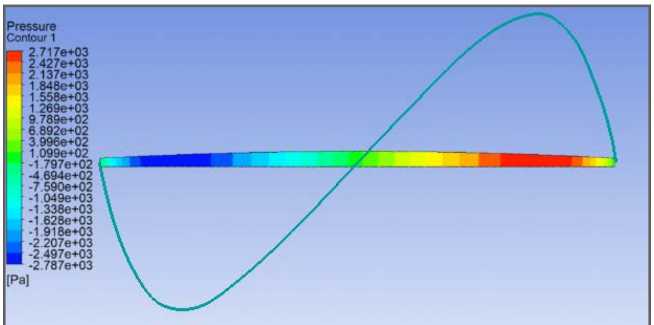

Fig. 7. The distribution epure for the hydrodynamic pressure of the oil in the gap with the elliptical cell

Conclusions

- We made a hypothesis about the high potential efficiency of using the elliptical cell microrelief to assure the service properties of cylindrical surfaces and reduce the attrition wear of the process units.
- The analytical model of lubrication layer behavior on the cylindrical surface with elliptical cell microrelief produced based on the theory of hydrodynamic lubrication.
- Using the ANSYS Fluent software, we produced the computerized hydrodynamic model (CFD) for the behavior of the lubrication layer on the cylindrical surface featuring elliptical cell microrelief.
- The analytical model showed that the maximum hydrodynamic load capacity occurs at the cell depth of 0.128 mm, which corresponds to part 21/25 of the major axis of the elliptic cell.
- Using the hydrodynamic CFD model, we calculated the lifting hydrodynamic pressure in one cell, and it equaled ~3 kPa.
- The location of the lubrication layer pressure maximum and minimum in the cell aligns with the similar data on the load capacity obtained from the analytical model, which means that the models are sufficiently accurate.
- Based on the results of the research, we can confirm the significant efficiency of using an elliptic cell microrelief to assure the service properties of cylindrical surfaces. This requires additional research of cellular microrelief in terms of both the optimal cell geometry and their formation technology.

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Технологическое обеспечение эксплуатационных свойств пар трения поверхностями с микрорельефом ячеистого типа

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Реферат

Работа посвящена совершенствованию технологических методов повышения износостойкости деталей пар трения. На примере поршневого компрессора представлено исследование цилиндрических поверхностей деталей с микрорельефом ячеистого типа. Форма ячейки представляет собой эллиптический параболоид с неравными положительными параметрами. Высокий потенциал использования ячеистого микрорельефа связан со снижением приработочного износа пар трения при обеспечении формой микрорельефа гидродинамической несущей способности смазочного слоя. Задачи исследования свелись к параметрическому анализу поведения смазочного слоя в зазоре с ячейкой микрорельефа. Для этого построена аналитическая модель, основанная на теории гидродинамической смазки, а также построена CFD-модель в программном обеспечении ANSYS Fluent. Для замыкания уравнений переноса была принята модель турбулентности SST *k–ω*. Обе модели показали, что максимальная гидродинамическая несущая способность приходится на 75% длины большой оси эллипса ячейки часть, что приходится на 0.128 мм по глубине ячейки. Максимальное подъемное гидродинамическое давление на одной ячейке микрорельефа составило 3 кПа. По результатам параметрического анализа можно констатировать существенный потенциал использования ячеистого микрорельефа для обеспечения эксплуатационных свойств пар трения технологических агрегатов.

Ключевые слова: пара трения; гильза цилиндра; поршневое кольцо; ячеистый микрорельеф; гидродинамическая модель; математическая модель; ANSYS Fluent; двухмерный параметрический анализ.

Gözcük tipli mikrorelyefli səthlər ilə sürtünmə cütlərinin istismar xassələrinin texnoloji təminatı

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Xülasə

Məqalə sürtünmə cütlərinin hissəciklərinin aşınmaya davamlılığının artırılması texnoloji üsullarının təkmilləşdirilməsinə həsr edilmişdir. Pistonlu kompressor nümunəsində gözcük tipli mikrorelyefi olan hissəciklərin silindrik səthlərinin tədqiqi təqdim olunmuşdur. Gözcüyün forması qeyri-bərabər müsbət parametrlərə malik elliptik paraboloiddir. Gözcüklü mikrorelyefdən istifadənin yüksək potensialı yağlayıcı təbəqənin hidrodinamiki daşıyıcılıq qabiliyyətinin mikrorelyef forma ilə təmin edilməsi zamanı sürtünmə cütlərinin aşınmasının azalması ilə əlaqədardir. Tədqiqatın məqsədlərinə mikrorelyef gözcükləri arasındakı boşluqda yağlayıcı təbəqənin özünü necə aparmasının parametrik analizi daxildir. Bunun üçün hidrodinamik yağlama nəzəriyyəsinə əsaslanan analitik model və ANSYS Fluent proqram təminatı vasitəsilə CFD modeli qurulmuşdur. Daşınma tənliklərinin ehtiva edilməsi üçün SST *k–ω* turbulentlik modeli qəbul edilmişdir. Hər iki model göstərmişdir ki, maksimum hidrodinamiki daşıyıcılıq qabiliyyəti gözcük ellipsinin əsas oxunun uzunluğunun 75%-də baş verir ki, bu da gözcüyün dərinliyinin 0.128 mm-nə təsadüf edir. Mikrorelyefin bir gözcüyündəki maksimum qaldırıcı hidrodinamiki təzyiq 3 kPa təşkil etmişdir. Parametrik analizin nəticələrinə əsasən təsdiq etmək olar ki, texnoloji aqreqatların sürtünmə cütlərinin istismarının təmin edilməsi üçün gözcüklü mikrorelyefdən səmərəli istifadə etmək olar.

Açar sözlər: sürtünmə cütlüyü; silindr qolu; piston halqası; gözcüklü mikrorelyef; hidrodinamiki model; riyazi model; ANSYS Fluent; ikiölçülü parametrik analiz.