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Experimental investigations and analysis of compressor’s friction losses

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1. Introduction

Friction losses are among the main factors, determining compressor’s efficiency. They not only increase energy consumption. Friction also causes the wear of compressor, decreasing compressors lifetime. This factor now is attributed to ecology in the same manner as energy consumption. The lifetime is related to emission of CO_2 – the worn out compressor must be replaced with new one, causing emission and consumptions of energy and materials at all stages of manufacturing process.

During the last three decades the friction losses were decreased due to increased quality of lubricants, new low viscosity oils, higher manufacturing quality of friction surfaces, new and affordable antifriction materials, commercial production of metal-ceramic.

2. Previous studies

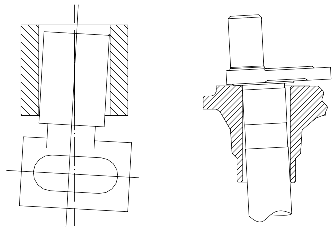
There are same studies [1-3] providing an experimental material related to the evaluation of friction losses in heavily loaded friction pairs. The highest contact stress arises between the connecting rod and the gudgeon pin. The wear occurring at this pair of friction reduces the refrigerant amount pumped per revolution and thereby increases the duration of each operating cycle of the refrigerator. Wear, friction and metal transfer are functions of a number of parameters. The study of these problems is presented in articles [4-6]. Almost all authors emphasize the difference between friction losses estimating in case of hydrodynamic lubrication and in case of boundary or mixed lubrication. N.P.Garland and M.Hadfield [5, 6] P.C. Nautiyal and J.A. Schey [3] and Short G. D. and Rajewsky T. E. [7] highlights the fact that sliding contact occurs even under the presence of a lubricant. Lubricity is an important consideration in determining the life of the compressor and only under the mixed lubrication regime can a life-long investigation be ascertained [8]. However the articles do not give calculation of compressor‘s total friction losses.

The authors of this article in their previous works [9-13] have presented mathematical models of reciprocating compressor with calculation results and analysis. Energy consumption for gas compression was calculated theoretically, taking into account losses in suction and discharge valves. Relationships of various forces, including friction forces were also presented, as well as relationships for friction losses and their analysis. Accuracy of calculations was analyzed in [10, 13], where theoretical and experimental results were compared. The simulated results correlated well with the experiment; however the calculation of the friction losses alone was not validated in the article. The total input power was measured only during calorimetric tests of compressor. The parameters of electric motor (efficiency and torque) were also determined experimentally. So we could estimate power of electric motor without electric losses, which is equal to the load. The load is a sum of three components, which were theoretically calculated in [9-11]. This is energy consumption for vapour compression, energy losses in suction and discharge valves and friction losses. Even if the sum of three components gives good correlation with the experiment, the inaccuracy is still possible if inaccuracy of calculation for each component cross-compensates.

In the articles [14-17] friction losses of compressor are calculated assuming lubrication fully hydrodynamic. This means that the losses are calculated according to Petroff law of friction. According to the law, the friction losses are in direct proportion to viscosity of lubricant. However in this article we will present experimental results which contradict to such linearity.

3 Distinguishing features of compressor’s friction pairs

The household compressor is a complicated machine with almost all friction pairs working at boundary or mixed lubrication conditions. This is not only caused by the high load, but also by the fact that the surfaces in the friction pairs are almost always in the twisted position (Fig. 1). The surfaces are in a contact, which increases the friction coefficient. The coefficient depends less on viscosity and more on magnitude and nature of the load. This will be shown in the next paragraph together with some experimental results.



a b

Fig. 1 Compressors friction pairs under load: *a* - piston – cylinder pair, *b* - crankshaft bearing pair

Another feature of compressor’s friction pairs is related to high concentration of refrigerant in oil. The hydrodynamic lubrication is based on traction of lubricant film into the wedge-shaped clearance between the bearing and journal. However, further the clearance increases, the pressure of the oil film decreases which causes the so-called cavitation - emission of refrigerant vapour from the oil. This decreases load-carrying capacity of the oil film.

Big number of experiments was done in order to decrease friction losses. The comparative experiments were carried out with different diameters of crankshaft, oil viscosity, loads of the bearings etc. The project was focused on the needs of specific company; however the further thorough analysis leads to general results.

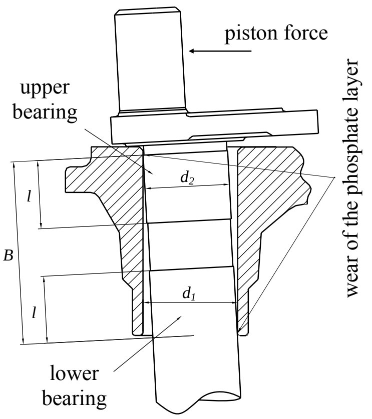


Fig. 2 Crankshaft bearing and its readjusting by wear of the phosphate coating

The experimental results revealed, that friction losses of piston compressors depend on viscosity of the oil, but the relationship is complicated. This is caused by the boundary or mixed lubrication conditions in friction pairs, which are more complicated than hydrodynamic lubrication conditions. At low viscosity of the oil and high loads the dependence between the oil viscosity and friction losses is small.

For the conditions of hydrodynamic lubrication the friction losses are calculated according to geometric characteristics of the bearing and properties of the oil. According to the Newton's law of viscous friction the friction force between the moving surfaces is in direct proportion to area of the surface F, velocity of the relative movement υ, viscosity of the oil μ and inversely as the distance between the surfaces, i.e. the thickness of the oil film h.

For the crankshaft of compressor (Fig. 2) the velocity is equal to $\upsilon = \pi n d$ and the area of the surface is $F = \pi d l$, where d is diameter of the crankshaft $d = d_2$, l is length of the bearing $l = l_1 + l_2$ and n is speed of rotation. The friction power is $P_{FR} = T\upsilon$ or after substitution

$$P_{FR} = \pi^3 \mu n^2 d^3 l / h \tag{1}$$

The friction power is in direct proportion to d^3 and decrease of crankshafts diameter from 18 to 16 mm should decrease the friction power by the factor 1.42. This also suggests that the crankshafts diameter should be as small as possible.

The works of known authors are based on the law of viscous friction when the friction coefficient is calculated according to well-known Petroff's equation [18, 19]

$$f_{FR} = \pi \mu \omega / (K \psi) \tag{2}$$

where ω is crankshaft's angular velocity, rad/s; K is load of the bearing, N/m^2; ψ is relative clearance between the bearing and the journal $\psi = 2h/d$.

Here the friction coefficient is in direct proportion to oil's viscosity and friction power should decrease proportionally to decrease of oil's viscosity. On the other hand, lower viscosity of the oil gives lower load-carrying capacity of the bearing which decreases minimal clearance in the bearing and increases its eccentricity. According to [18] the total load-carrying capacity is

$$W = \frac{\upsilon \mu \varepsilon l^3}{c^2 \left(1-\varepsilon^2\right)^2} \frac{\pi}{4} \times \sqrt{\left(\frac{16}{\pi^2} - 1\right)\varepsilon^2 + 1} \tag{3}$$

In the equation $c = h/2 = 0.5(d_1 - d_2)$, where d_1 and d_2 are diameters of the bush and the shaft; ε is the eccentricity ratio, i.e. the ratio of the bearing eccentricity e to clearance c ($\varepsilon = e/c$).

Obviously, W is decreasing when oil's viscosity and velocity of surfaces decreases. The distance between the surfaces decreases and eccentricity ratio ε increases. Increased value of ε extremely increases W; when ε approaches to 1, the value of W approaches infinity. This extreme case is purely theoretical – practically the surfaces come into the contact and in the place of the contact the load carrying capacity W disappears. The contact between the surfaces increases the friction losses due to appearance of additional boundary lubrication friction losses.

The second component of friction losses in equation (3) depends on eccentricity ratio ε of the bearing. When the ratio is small the part of this component in total friction losses is relatively small [18]. However, when ε approaches unity, the friction loses increases extremely. In [18] the following equation is given for the calculation of friction force according the Half-Sommerfeld conditions

$$P_{FR} = \frac{2\pi\mu\upsilon l d}{h} \times \frac{1}{\left(1-\varepsilon^2\right)^{0,5}} \tag{4}$$

The first term of the equation is called the Petroff friction and the second term is called Petroff multiplier. Thus we can assume that for conditions of our research the second term of equation (4) is qualitatively connected to boundary lubrication conditions. The geometry of bearing's surfaces is measured in μm. This means that for boundary lubrication conditions the ε is above 0.90. The friction force increases significantly, but the increase is not equal to the increase which should be according to Petroff multiplier. The Fig. 3 shows relationship of Petroff multiplier subject to ε, which is given in [18].

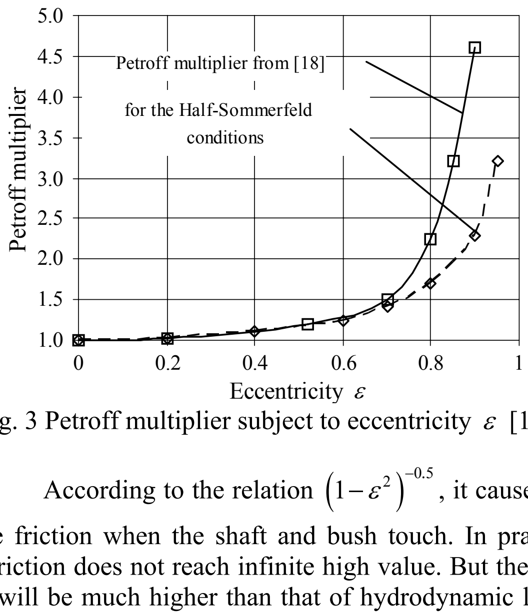


Fig. 3 Petroff multiplier subject to eccentricity ε [18]

According to the relation $\left(1-\varepsilon^2\right)^{-0.5}$, it causes infinite friction when the shaft and bush touch. In practice the friction does not reach infinite high value. But the friction will be much higher than that of hydrodynamic lubrication. So the values of Petroff multiplier are much more higher than those predicted from $(1-\varepsilon^2)^{-0.5}$.

The Fig. 3 gives explanation why at higher loads of the bearing the friction power is less dependent on viscosity of the oil or the relationship is nonlinear as will be proved in the next paragraph.

Bigger loads of the bearing increase eccentricity ε to the values which significantly influence Petroff multiplier. On the other hand, oil's viscosity μ also does influence the value of eccentricity. If the viscosity slightly decreases due to increased temperature of the oil, the load-carrying capacity of the bearing W also slightly decreases and eccentricity ε increases accordingly. Even if the increase is not big, it significantly increases the Petroff multiplier (Fig. 3), since the ε values are close to unity. At the same time the first term of Eq. (4), related to losses of friction of hydrodynamic lubrication is decreasing due to decreased oil's viscosity. So we get a case when decreased losses of hydrodynamic friction are partially compensated by the increased losses of boundary friction.

4. Experimental investigations of friction

The compressor, which is used for the analysis in this article, has displacement volume of 8.6 cm^3 with piston's diameter 24 mm and stroke 19 mm. This model was selected because of big amount of available experimental data, including measurements of friction losses. For example, a lot of tests were done to find out how friction losses are influenced by the diameter of crankshaft, oil's viscosity, clearance between the piston and the cylinder, load etc.

The compressor was tested at free-operated conditions as well as under various loads during calorimetric tests. The friction power was estimated from measurements of total input power. The latter is formed power for compression of vapour, power used for suction and discharge through the valves (so-called indicated losses), power losses in electric motor and power losses for friction. Losses for friction involve not only losses in friction pairs, but also losses for friction between moving parts of compressor and vapour, but this part is relatively small. The biggest resistance of approximately 0.1 W is for the movement of rotor, while the resistance of the oil pump is even smaller and does not exceed 0.03 W. Therefore the latter two losses were not taken into account.

Estimation of friction losses for free-operated compressor is less complicated. Input power is formed of friction losses and electric losses of the motor. The losses of the motor can be estimated through measurement of its main characteristics. In this case the motor with the nominal capacity of 120 W was used (see the measured characteristics on Fig. 4).

Fig. 5 gives friction power of free-operated compressor for two different crankshaft diameters at various oil temperatures. For the smaller crankshaft diameter the friction losses are lower, but the difference depends on viscosity (temperature) of oil. At the beginning of the test, when oil's temperature is 30ºC and viscosity is significantly higher, the difference is bigger. The friction power as well as total input power decreases with increase of temperature and decrease of viscosity. The relationship is similar to the relationship of oil's viscosity subject to temperature. One could assume that friction pairs of free-operated compressor are working under the conditions of hydrodynamic lubrication. However the deeper analysis proves that such assumption is not correct.

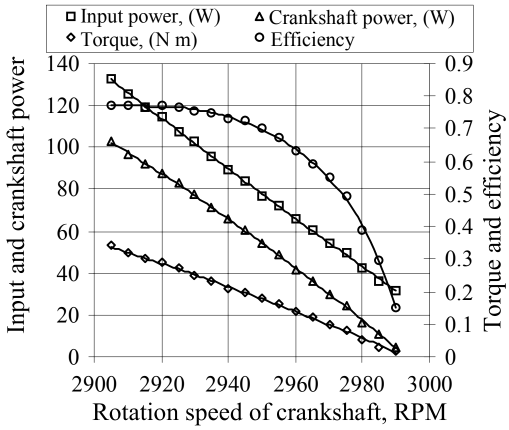


Fig. 4 Characteristics of electrical motor

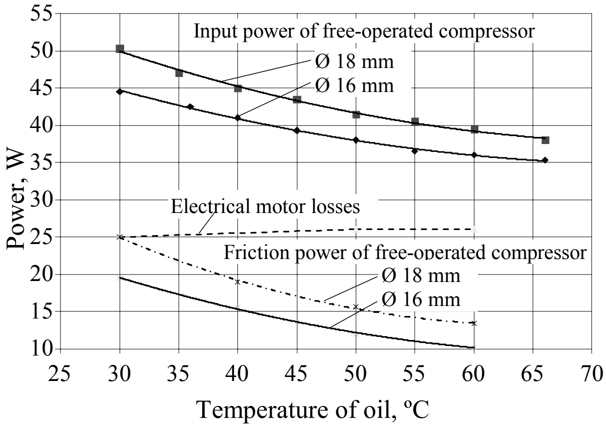


Fig. 5 Friction power of free-operated compressor

From comparison of Fig. 5 and Fig. 6 we can conclude that free-operated compressor is working at the boundary or mixed lubrication conditions. This is also proved by the visual inspection of details after a few hours of operation. Since the details initially were covered with phosphate coating with thickness of 2-3 μm, the traces of wear could be seen.

If the details were working at hydrodynamic lubrication conditions they would not be in a contact and the friction power would be decreasing proportionally to decrease of oil's viscosity at higher temperatures. For example, from the Fig. 6 we can see that viscosity of RL10H oil at 60ºC is 2.7 times lower than at 30ºC. At the same time the friction power of compressor decreases only 1.9 times. This could mean that with decrease of oil's viscosity the part of losses due to boundary lubrication is increasing.

Hence, oil's viscosity has dual influence on friction losses. From one side, the losses are decreasing since the moving surfaces are separated by the lower viscosity oil. From other side, oil with lower viscosity creates lower *W*, which leads to contact between moving parts and can increase friction losses several times [19]. The bigger is the share of boundary lubrication, the bigger are friction losses. This can be seen from analysis of Fig. 5. For example, at 30ºC with the crankshaft of Ø16 mm the friction losses are 19.5 W. At 60ºC oil's viscosity is lower by factor 2.7 (Fig. 6). For fully hydrodynamic lubrication the friction losses of compressor should decrease by 12.3 W

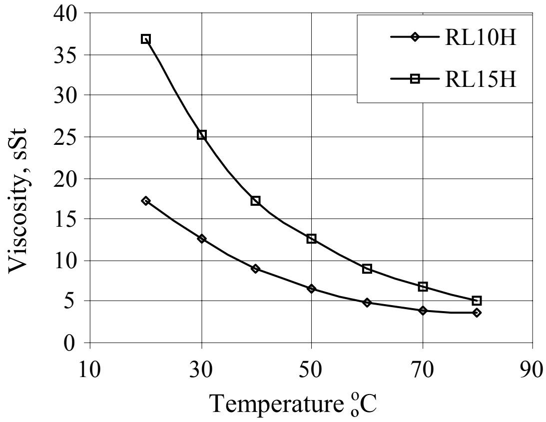


Fig. 6 Oil viscosity subject to temperature

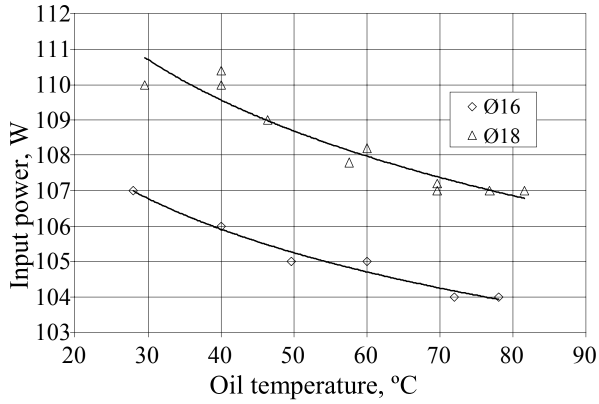


Fig. 7 Input power during calorimetric test

(to 12.3 W), but in reality they decrease only by 9.2 W (to 10.3 W). The same result can be obtained from the analysis of experimental relationship for the crankshaft of Ø18 mm.

For the compressor operating under load the effect is even more pronounced. Corresponding decrease of power consumption is only 3.4 - 4 W (Fig. 7). Similar experimental relationship of capacity and input power subject to oil's temperature is given at Fig. 8. With the RL10H oil the input power decreases from 107.5 W to 104.5 W when oil's temperature increases from 30ºC to 60ºC, whereas with the RL15H oil the corresponding decrease is almost two times bigger – from 112.5 to 107 W.

Figs. 7 and 8 give relationships for fully loaded compressor. The friction losses in such case make about 20% of input power. The fact that oil's viscosity has small effect on the input power proves that friction power of loaded compressor is determined by the boundary lubrication conditions. On the other hand, increase of temperature also changes other components of input power – electric losses of the motor and power for compression of vapour. Let's analyse the influence of these components on compressor's input power during calorimetric test. For isentropic compression the power consumption according to [20] is

$$P_{ISO} = \left(p_S - \Delta p_S\right)\bar{V}_h\theta\frac{k}{k-1}\left[\left(\frac{p_D}{p_S - \Delta p_S}\right)^{(k-1)/k} - 1\right] \quad (5)$$

where $p_S - \Delta p_S$ and p_D are initial and final pressures of compression process; θ is coefficient estimating energy returned by the expanding gas from the dead volume; k is specific heat ratio of refrigerant; p_S is pressure in compressor's shell (at ASHRAE conditions $t_o = -23.3\,^{\circ}\mathrm{C}$ and $p_s = 0.632$ bar); $\bar{V}_h$ is displacement volume per second. According to motor's characteristics at 104 W power consumption the rotation speed is 48.6 1/s and $\bar{V}_h = 0.000418\ \mathrm{m^3/s}$.

The change of temperature does influence the only variable of equation 5 – the specific heat ratio k. When temperature increases, the ratio decreases. For example, suction temperature to the shell is 32ºC, pressure is 0.632 bar and $k = 0.098$. Whereas, the temperature at theend of suction (before the compression) is about 80ºC and $k = 0.083$. This decreases power consumption for compression about 1% which is 0.6 W. Thus with increase of temperature the power consumption for compression slightly decreases. Contrary to that, losses of electric motor with increase of temperature do increase. These losses are made of three parts – losses in stator's pack, losses in stator's winding and losses in rotor. The first part makes about half of all electric losses [21] and does not depend on temperature. Losses in the rotor's pack also do not depend on temperature and these losses are not big. About one third of electric losses (approximately 8 W) make losses in stator's winding. These losses are in direct proportion to resistance of copper wire and increase by approximately one fifth, i.e. by 1.6 W.

Hence, the temperature has a small influence on total input power. The capacity consumption for vapour compression decreases a bit, and electric losses increase. When the temperature changes from 30 to 60ºC, the input power (for compression and electric losses) increases only about 1 W, which does not change the meaning of Figs. 7 and 8. The main meaning of the relationships is that friction losses almost do not depend on oil's viscosity. This best can be seen from the Fig. 9, where the relative values of oil viscosity and friction losses are given. At 40ºC the said relative values are equal to 1. As can be seen, the oil's viscosity continues to decrease even at 80ºC, but the friction losses during the calorimetric test becomes almost constant. For the free-operated compressor the friction losses more depend on oil's temperature. However, at oil's temperature above 70ºC the losses again become almost constant, while decrease of viscosity is still significant.

The fact that the friction losses are almost constant while oil's viscosity is decreasing can hardly be explained by the theory of hydrodynamic lubrication. Even the Eq. (4), which takes into account increase of friction force due to eccentricity, ε does not explain that, since at ε approaching unity, friction forces approach infinity. However, experimental research reveals that compressor's friction pairs are working partially in contact. The places of contact can be identified by the different colour which they have due to wear of phosphate coating.

The authors in their article [11] presented calculation of compressor's bearings relative eccentricity ε subject to rotation angle of crankshaft (Fig. 10). As already mentioned, the upper bearing (Fig. 2) is more heavily loaded than lower bearing, which can be seen from the relationships on Fig. 11.

According to the theory of hydrodynamic lubrication, with increase of load the eccentricity ε will also in-

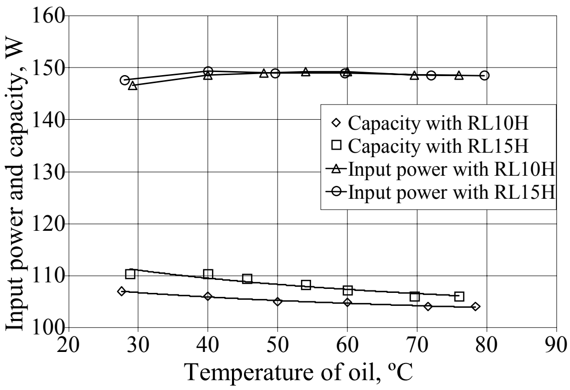


Fig. 8 Capacity and input power during calorimetric test

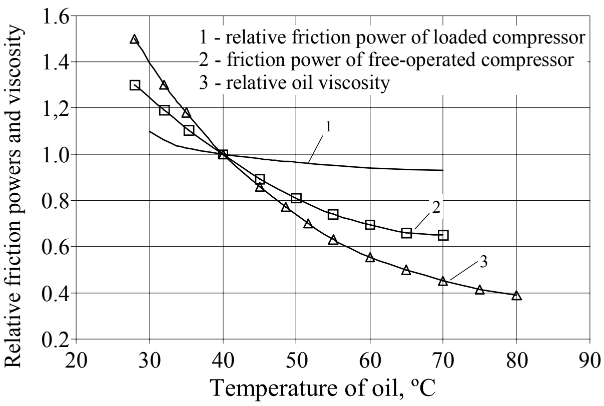


Fig. 9 Relational friction power and oil viscosity during calorimetric test

crease. However, when the eccentricity is high enough, and especially when it approaches unity, the load carrying capacity of the bearing increases extremely, limiting further increase of eccentricity. This can be seen on Fig. 10. At the end of compression the load of upper bearing is two times bigger than the load of lower bearing, but the relative eccentricity is almost the same – 0.81 and 0.75 respectively. When the loads of the bearings are lower, they have the bigger influence on the eccentricity. For example at the beginning of compression process ($\tau = 4\pi/8$) the loads are not big although the difference between the loads of upper and lower bearings is almost the same as at the end of compression process – about 150 N. But the eccentricities are very different ($\varepsilon = 0.23$ for the lower bearing and $\varepsilon = 0.69$ for the upper bearing).

From the Fig. 10 we can see that even at the highest load the bearing's eccentricity ε should not reach 0.9. Even for the highest load of upper bearing the eccentricity reaches only 0.8, But for appearance of contact the eccentricity should be significantly higher, more than 0.9. However, the research proves that both upper and lower bearings do wear, which means that the eccentricity ratio reaches unity at significantly lower values of the load.

5. Analysis and some practical remarks

Obviously that in reality the load-carrying capacity W is not increasing so extremely, as it should according to Eq. (3) and it does not increase to infinity at minimal clearances in the bearings, when $\varepsilon \to 1$.

Throughout the literature there are three sorts of boundary conditions to solve Reynolds equation for hydrodynamic lubrication. The boundary conditions are known as the Full-Sommerfield, Half-Sommerfeld and Reynolds.

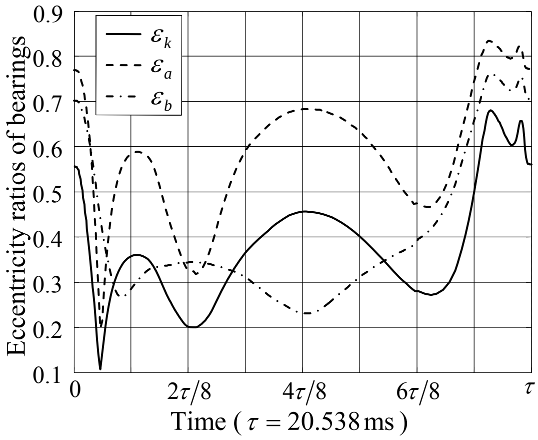


Fig. 10 Eccentricity ratio of the crankshaft [14]: ε_a - upper bearing, ε_b - lower bearing, ε_k - short end bearing

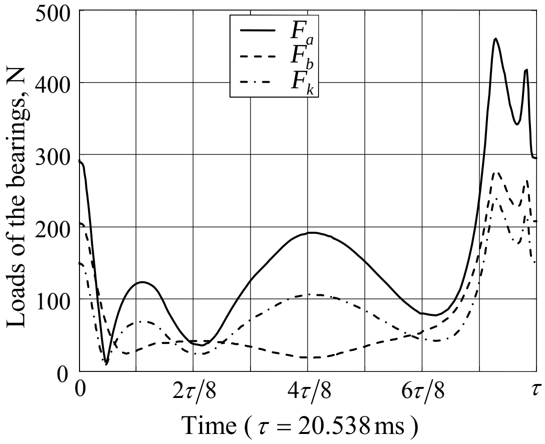


Fig. 11 Loads of crankshaft bearings [14]; F_a - upper bearing, F_b - lower bearing, F_k - short end bearing

The Full-Sommerfeld condition is the simplest of the boundary conditions: the pressure is equal to zero at the edges of the wedge. But this condition is unlikely to apply to real fluids because of large negative pressure. Furthermore, because of opposing negative and positive pressures the predicted load capacity W is zero.

Half-Sommerfeld boundary conditions assume that the negative pressure does not exist and in fact it is equal to zero at this region. From engineering viewpoint Half-Sommerfeld condition is simple and easy to apply (Eq. (4). Errors are small; however it is erroneous since discontinuity of flow at diverging region between the zero and nonzero pressures.

Reynolds assumed that in the diverging region the pressure gradient and pressure are equal to zero $p = dp/dx = 0$. In this region the lubrication film starts to divide into streamers of lubricant and gas spaces. As the film thickness continues to increase, the proportion of space occupied by lubricant streamers is correspondingly reduced. Volume of gas space increases and we have cavitations phenomena [18].

Discontinuity of lubrication film flow means that the total load W that hydrodynamic lubrication bearing will support, decreases. In a clearance between the bearing and journal appear gas cavities. Their number and size depend on how easy does the gas gets into the clearance. Some gas is dissolved in oil. When pressure decreases, the emission of gas from oil begins. Moreover, gas flows into the clearance from outside more easily. When the clearance between bearing and journal decreases, the pressure of oil film and gas cavities increases. The gas is partially absorbed with the oil, but remaining part significantly decreases load carrying capacity *W*. Gasses are compressible and have much lower viscosity. They easily flow through the sides of the bearing, dragging together the oil. The relative eccentricity ε increases faster at lower loads, contact between the bearing and the journal occurs and hydrodynamic lubrication conditions change to boundary lubrication conditions.

The refrigerating compressor is working in hermetic system, filled with refrigerant, which has very good solubility in oil. The amount of refrigerant in oil is much higher than the amount of air (when the bearing is working in open system). During cavitation the emission of refrigerant from oil is also more intense. So, in the bearings of refrigerating compressor the boundary lubrication occurs at lower loads. This is one of distinguishing features of refrigerating compressor's bearings.

In compressor the crankshaft usually is working under cantilever load which causes misalignment of axes of the bearing and the journal. Because of misalignment the smallest clearance appears at the top of upper bearing and at the bottom of lower bearing (Fig. 2), and that are the places where the most intense wear occurs. On the other hand, the wear improves performance of the bearings since the surfaces do readjust.

The experiments show that friction losses in just-assembled compressor are by a few W higher than the losses after the short break-in – usually a few hours are enough for it. The short break-in period is ensured by thin and fast-wearing antifriction coating, e.g. phosphate.

The phosphate coating does decrease friction losses also due to its structure – it results a regular crystalline structure on the metal surface. The crystals have a concave structure and lubricant (especially lubricant of low viscosity) fills the concaves, forming microreservoirs. This leads to good retention of lubricant on the metal surface. We think that phosphate coatings become even more important tor compressor operating with very low viscosity oils.

For operation with current oils of average viscosity the higher value of clearance between the bush and the shaft is required (about 20-25 μm). But the thickness of phosphate coating is about 2-3 μm. When it wears off, the (length) of irregular clearance between the bush and the shaft decreases only by one third. The area of contact for boundary lubrication conditions increases respectively. With the oils of lower viscosity it is possible to decrease the clearance between the bush and the shaft, which also decreases misalignment. Thus the wear of phosphate coating does decrease the irregular clearance more significantly, increasing the area of heavy-loaded surfaces. If the low viscosity oil is combined with lower diameter of crankshaft, there is the decrease of bearings length l and increase of length B (Fig. 2), the bearing of crankshaft may be designed in such a way, that after the first few hours of operation it will reach the maximal load-carrying capacity W. Through the whole bearings height l the clearance between the bush and the shaft will become regular, which may decrease the friction losses by 2-3 W.

One more remark considers the relationship between the oil's viscosity and clearance between the piston and the cylinder. This clearance is very important to efficiency – capacity of the compressor is decreased because of blow-by losses. On the other hand, the small clearance increases friction losses. For slider-link driven compressors the clearance is bigger than for connecting rod driven compressors and reaches 9-11 μm. This optimal value was determined experimentally after thorough research, which was aimed to ensure maximal efficiency.

The Fig. 12 gives the compressor's calorimetric tests results. Since changing of the piston required reassembling the compressor (with setting up linear dead volume etc.) the tests with each piston were repeated 2 or 3 times. As it can be seen from the chart, from the efficiency point of view the optimal clearance between the piston and the cylinder is 10 ± 1 μm. Here we also want to draw attention to the fact, that at the beginning of calorimetric test the blow-by losses are significantly higher.

At the very beginning of calorimetric test, when the oil's temperature is low and its viscosity is high, the compressor's capacity is exceptionally low (Figs. 7 and 8). This phenomenon could be called 'crisis of capacity'. However, when the temperature reaches ~ 40ºC, compressor's capacity increases. Input power increases a bit or remain constant. With the further increase of temperature, both values do change logically. For the connecting rod driven compressors such phenomena was also observed, though not so pronounced. It is also less pronounced for the slider-link driven compressors when the clearance between the piston and the cylinder is smaller (8-9 μm).

In our opinion the capacity crisis phenomenon is related to adhesive properties of oil. When oil's temperature is higher, adhesion of oil's molecules to the surface of

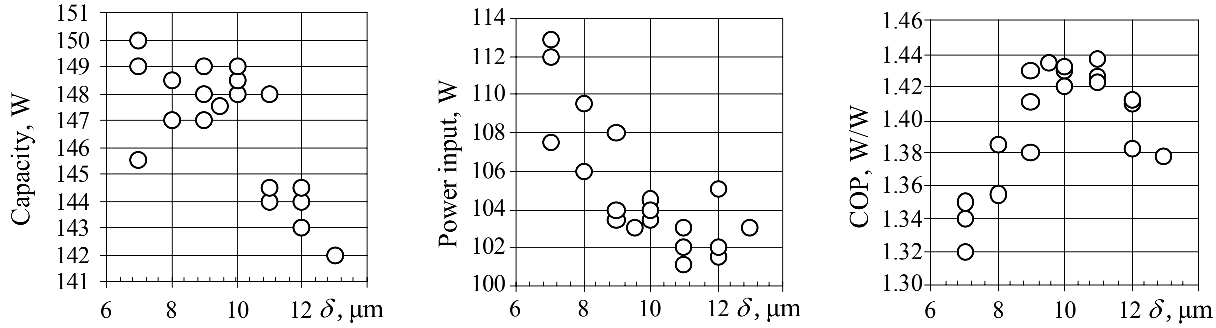


Fig. 12 Calorimetric data of the isobutane compressor (displacement 8.6 cm^3)

the moving parts is more effective. The oil is dragged into the clearance without breaking the continuity of film flow. The oil completely fills the clearance and compressed vapour can not leak through it. When the clearance is smaller, the capacity crisis is less pronounced. The explanation for this could be lower blow-by losses even without the sealing function of oil (due to smaller clearance). Considering connecting rod driven compressors we should also take into account that their pistons are made of metal-ceramic which has porous structure, potentially improving oils adhesion. On the other hand, the clearance in such compressors is also significantly smaller.

There is one more remark, which we would like to discuss. It considers the compressor's test standards. The tests are done at the conditions, which are far away from the conditions at which the modern refrigerating systems are operating.

The test conditions of compressors and household refrigerators are defined in different standards. The compressors are tested at harder conditions with higher compression ratios and higher condensing temperatures. Moreover, during the last 15-20 years the operation cycle of refrigerator changed significantly. Due to strict restrictions the refrigerator manufacturers optimized the cycle to minimize compressor's energy consumption. The compressor's operation conditions become easier, nevertheless the test conditions remain unchanged.

The compressor manufacturers have to optimize for example valves, dead volume and clearance between the piston and the cylinder for the test conditions. Let us look again to the problem of clearance between the piston and the cylinder. At 55ºC condensing temperature (55ºC for CECOMAF and 54.4ºC for ASHRAE conditions) the refrigerant R600a must be compressed to 7.81 bar and R134a – to 14.92 bar. In modern refrigerator average condensing temperature is about 35ºC, and the corresponding pressures are significantly lower – 4.69 and 8.89 bar. The blow-by losses are proportional to the difference of second powers of condensing and evaporation pressures. Thus for the compressor test conditions the blow-by losses will be almost three times higher. The compressor manufacturers are forced to decrease the clearance in order to decrease the losses. This not only increases the friction losses in the pair, but also increases manufacturing cost of corresponding details.

The similar problem is with the application of lower viscosity oils. At the high temperature which is reached at the end of calorimetric test (approximately 80ºC) the positive effect of application of low viscosity oil is comparatively low (Fig. 8). However, during operation in modern household refrigerator the temperature is significantly lower – about 45ºC. At such temperature RL10H oil gives more than 4 W effect (Fig. 7). Since average energy consumption of modern refrigerator is about 80 W, energy consumption would be decreased by 5%.

According to the standards the manufactures of compressors run life tests at even higher temperatures, estimating possible risks of low viscosity oils. The positive effect of low viscosity oils is estimated through the calorimetric tests. In case of RL10H oil the effect hardly reaches 2 W. But the risk for compressor's durability increases because of more intense wear. Even if total friction losses decreased, the part of friction losses related to boundary lubrication conditions increased. However, when the compressor is working in refrigerating system this part of friction losses are significantly lower.

6. Conclusions

The friction losses in the sense of ecology are more important than other losses influencing compressors efficiency. These losses are related to compressors wear and decrease its durability. The analysis of experimental data presented in the article suggests that due to big amount of refrigerant solved in oil the wear of refrigerating compressors is more intense. This factor decreases load carrying capacity of oil film due to more intense cavitation. Big number of experiments indicates that friction losses of refrigerating compressors almost do not depend on oils viscosity. This should mean that in refrigerating compressors we observe a case when decreased losses of hydrodynamic friction are compensated by increased losses of boundary lubrication friction.

The wear of compressors parts as well as its life time depends on boundary lubrication friction. The duration of compressors life time is evaluated experimentally through the life test and the life test conditions are very different from real operation conditions in modern refrigerators. During the last decades the operation conditions in refrigerators become significantly easier. However, the life tests done by compressors manufacturers does not show the increased life time, since the test conditions remain unchanged. This supports more conservative approach, limiting positive effect which could be gained from easier operation conditions. Such design approach could promote wider use of lower viscosity oils, less expensive materials and simpler manufacturing process of some details, for example pistons and cylinders pair.

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KOMPRESORIAUS TRINTIES NUOSTOLIŲ EKSPERIMENTINIS TYRIMAS IR ANALIZĖ

R e z i u m ė

Straipsnyje apibendrinami ir analizuojami buitinių šaldymo kompresorių trinties nuostolių eksperimentinio tyrimo rezultatai. Pagrindinis tyrimo tikslas buvo pasiekti aukštesnį kompresoriaus efektyvumą. Nustatyta, kad šiuolaikinių kompresorių trinties nuostolius tepalo klampa įtakoja mažai. Autoriai daro išvadą, kad tokių kompresorių trinties nuostolius stipriau lemia ribinis tepimas nei hidrodinaminis tepimas. Naudojant žemesnės klampos tepalus, ribinio tepimo trinties nuostolių dalis didėja. Šaldymo kompresorių tepale ištirpsta daug šaldymo agento. Tai padidina kavitacijos įtaką tepalo plėvelės nešančiajai gebai. Problemą dar padidina didėjantis skirtumas tarp kompresoriaus bandymo sąlygų ir darbo šaldytuve sąlygų.

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EXPERIMENTAL INVESTIGATIONS AND ANALYSIS OF COMPRESSOR'S FRICTIONAL LOSSES

S u m m a r y

In this article the results of experimental investigation of household refrigerating compressor's friction losses are summarized and analysed. The main objective of the research was achieving higher compressors efficiency. The research revealed that for modern compressors the oils viscosity has low influence on friction losses. The authors draw conclusion that friction losses of such compressors are more determined by boundary lubrication than by hydrodynamic lubrication. With lower viscosity oils part of boundary friction losses is increasing. The oils of refrigerating compressors holds big amount of dissolved refrigerant. This increases the influence of cavitation on load-carrying capacity of oils film. The problem is further complicated by increasing difference between compressors test conditions and its operation conditions in refrigerator.

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ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ И АНАЛИЗ ПОТЕРЬ НА ТРЕНИЕ КОМПРЕССОРА

Р е з ю м е

В статье обобщены и проанализированы результаты экспериментального исследования потерь на трение бытовых холодильных компрессоров. Основная цель работы было повышение эффективности компрессора. Исследование показало, что вязкость масла слабо влияет на потери на трение в современных компрессорах. Авторы делают заключение, что потери на трение в таких компрессорах определяются больше граничным, чем гидродинамическим режимом смазки. С уменьшением вязкости масла, часть потерь на граничное трение возрастает. В масле холодильных компрессоров растворяется большое количество хладагента. Это увеличивает влияние кавитации на несущую способность масляной пленки. Проблему усугубляет увеличивающаяся разница между условиями испытания компрессоров и условиями их работы в холодильнике.

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