METHODS OF FRICTION LOSSES MODELING FOR ELASTIC OIL FILM

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Abstract

Different properties that could affect the cooperation between piston and rings, and cylinder bore are characteristic for lube oils present on the market for several years. One of those properties is the oil shear modulus. Since it is hardly ever mentioned in literature the main goal of this study is to present a method defining the interdependence between the lube oil elastic properties and the characteristic features of piston-cylinder set operation against the friction loses of an IC engine.

The paper presents a theoretical basis for the method of determination of shear in oil layer present in a lubricating gap dividing piston ring and cylinder bore, as well as some results of shear calculations.

1. Introduction

Contemporary lubricating oils are well suited for operation in highly loaded combustion engines. Among various characteristic data, their viscous characteristics improves systematic-cally which means that temperature affects the change in viscosity to the less extend.

Such advantageous change results from the process of oil improvement consisting mainly in addition of synthetically obtained polymers.

Some papers hold that because of synthetic and semi synthetic oil technology their properties could be different from those characteristic for Newtonian fluids, i.e. they could adopt properties characteristic for viscoelastic fluids (see Fig. 1).

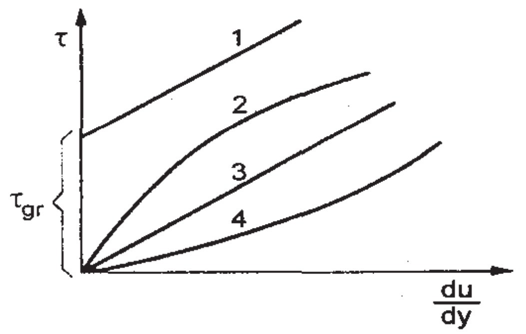

Fig. 1. Yield curves of viscoelastic fluids: 1 – Bingham fluid, 2 – viscoplastic fluid, 3 – Newtonian fluid, 4 – dilatation fluid

Several mathematic models of viscoelastic fluid have been developed. The most popular one is that formulated by Maxwell [1, 4]

$$\dot{\gamma}=\frac{\tau}{\eta}+\frac{\dot{\tau}}{G}, \tag{1}$$

where: $\dot{\gamma}$ – strain rate,

τ – shear,

η – dynamic viscosity,
G – shear modulus.

Equation (1) embodies the factor G, called a shear modulus. When its value is infinitely high, Maxwell's formula becomes the Hooke's formula, i.e. it describes an ideally elastic body. On the other hand, when it equals to zero, the formula becomes the Newtonian one, i.e. it describes an ideally viscous fluid. As the literature is short of information on the effect of shear modulus G on friction losses at kinematic nodes lubricated with the oils of elastic properties the author has made an attempt to conduct such an evaluation.

The solution consists in definition of shear in oil layer separating lubricated surfaces, i.e. solving the Eq. (1) which is possible using analytic and numerical methods.

2. Analytic method of Maxwell formula solution

For the analytic solution of Maxwell formula it has been assumed that the course of change in velocity of surfaces constraining the lubricating gap relates to the changes in speed of piston in cylinder liner given in following form:

$$u = r \cdot \omega \left(-sin(\omega \cdot t) - \frac{\lambda}{2} \cdot sin(2 \cdot \omega \cdot t) \right), \tag{2}$$

where:
r – crank radius,
ω – crank shaft angular velocity,
λ – crank radius to connecting rod ratio,
t – time.

On the other hand the formula below has been introduced in order to describe the changes in oil layer thickness h

$$h = \frac{1}{g1 + g2 \cdot \sin(\omega \cdot t)}, \tag{3}$$

where $g1$ i $g2$ are the constants, that allow to select a course of oil layer thickness. One of the reasons why so odd form has been chosen was its suitability for solution of Maxwell formula using the Laplace's transformation.

Transforms of the expressions constituting the differential equation have been defined at the beginning stage of solution. Due to limited space, separate stages of solution will not be presented (they are similar to those presented in [2, 3]). The final form of the equation necessary for the definition of shear in oil layer is given by Eq. (4).

Fig. 2 shows courses of shear τ for exemplary set of values that differ one from another by the value of shear modulus G. Value of quantities used in calculations are given below the figure. They begin with the value of $t = 0$ and end with values different from zero at the end of full engine cycle (except for the curve 1 for which $G = \infty$). This is the result of units in Eq.(4) that comprise expression $\exp(G/\eta{*}t)$ which equal to zero after t time (or an increase in crank shaft angle).

$$\tau = G \cdot r \cdot \omega^2 \left\{ \begin{array}{l} \dfrac{g1}{\left(\frac{G}{\eta}\right)^2 + \omega^2}\left[-\dfrac{G}{\omega \cdot \eta}\sin(\omega \cdot t) + \cos(\omega \cdot t) - e^{-\frac{G}{\eta}t}\right] + \\ \dfrac{g2}{\left(\frac{G}{\eta}\right)^2 + (2 \cdot \omega)^2}\left[\left(\dfrac{G}{2 \cdot \omega \cdot \eta}\right)\cos(2 \cdot \omega \cdot t) + \sin(2 \cdot \omega \cdot t) + \dfrac{2 \cdot \eta \cdot \omega}{G}e^{-\frac{G}{\eta}t} - \dfrac{\left(\frac{G}{\eta}\right)^2 + (2 \cdot \omega)^2}{2 \cdot \omega \cdot \frac{G}{\eta}}\right] \\ -0.25 \cdot \lambda \left[\begin{array}{l} \dfrac{4 \cdot g1}{\left(\frac{G}{\eta}\right)^2 + (2 \cdot \omega)^2}\left[\left(\dfrac{G}{2 \cdot \omega \cdot \eta}\right)\sin(2 \cdot \omega \cdot t) - \cos(2 \cdot \omega \cdot t) + e^{-\frac{G}{\eta}t}\right] + \\ \dfrac{g2}{\left(\frac{G}{\eta}\right)^2 + \omega^2}\left[\sin(\omega \cdot t) + \dfrac{G}{\omega \cdot \eta}\cos(\omega \cdot t) - \dfrac{G}{\omega \cdot \eta}e^{-\frac{G}{\eta}t}\right] - \\ \dfrac{g2}{\left(\frac{G}{\eta}\right)^2 + (3 \cdot \omega)^2}\left[3 \cdot \sin(3 \cdot \omega \cdot t) + \dfrac{G}{\eta \cdot \omega}\cos(3 \cdot \omega \cdot t) + -\dfrac{G}{\eta \cdot \omega}e^{-\frac{G}{\eta}t}\right] \end{array} \right] \end{array} \right\} \tag{4}$$

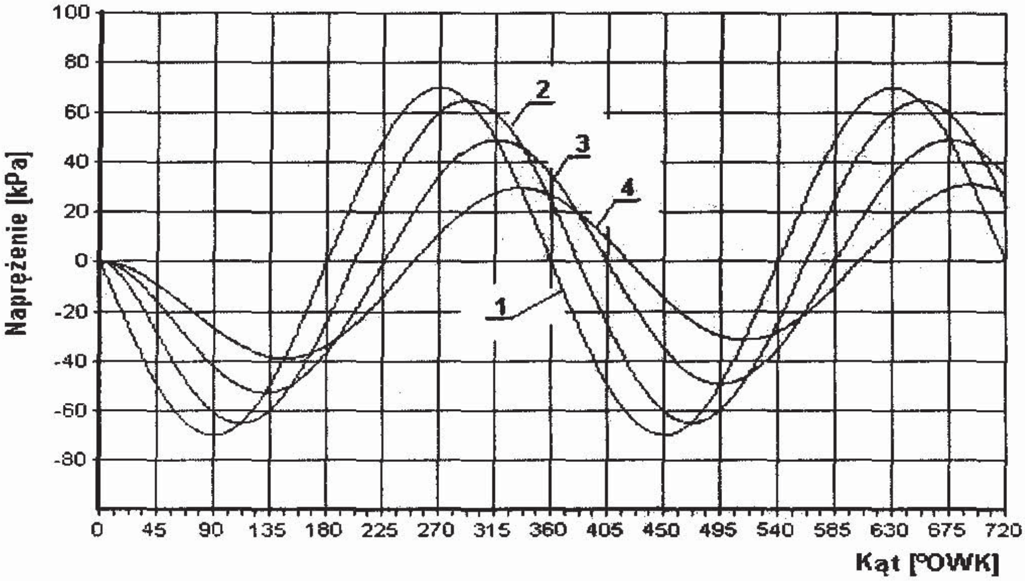

Fig. 2. Changes in shear value vs. crank angle for selected values of oil shear modulus G; 1 – G =∞, 2 – G = 5 Pa, 3 – G = 2 Pa, 4 – G = 1 Pa and for η = 0,02 Pas, ω = 100 rad/s, r = 0,035 m, h = 1 μm

Even for very low values of G, the values of expressions containing the $\exp(G/\eta{*}t)$ component decrease very quickly. For the data characteristic for lube oils used in combustion engines (G higher than 5 Pa) and velocity of $\omega = 100$ rad/s the time of significant effect of these modules on the course shear value does not exceed 50 °CA.

3. Numeric solution of the Maxwell formula

The Maxwell formula could be solved using numerical methods. Different forms of the formula are obtained depending on the differential quotient applied for transformation. For example, using a unilateral differential quotient the shear in oil layer limited with arbitrary located surfaces becomes of the following form (i is a point of computation cycle):

$$\tau_{i+1} = \tau_i\left(1-\frac{G}{\eta}\Delta t\right)+G\cdot\Delta t\left(\frac{u}{h_r}\right)_i-\frac{G\cdot\Delta t}{2\eta}(h_k)_i\left(\frac{p_{r+1}-p_{r-1}}{2\cdot\Delta x}\right)_i . \qquad (5)$$

where r is any point on a surface limiting the lubricating gap between points 1 and $m+1$ (the height of gap has been divided into m fractions).

Definition of the value of expression (5) for consecutive points of computation cycle resulted with the set of n equations that allow (after some transformations) to obtain formula suitable for computation of shear at any point of computation cycle, as the function of the shear at the beginning of calculations, i.e. for $i = 1$.

$$\tau_i = \tau_1\cdot a^{i-1}+b\cdot\sum_{k=1}^{i-1}\left[a^{k-1}\cdot\left(\frac{u}{h_r}\right)_{i-k}\right]-c\cdot\sum_{k=1}^{i-1}\left[a^{k-1}\cdot(h_r)_{i-k}\cdot\left(\frac{p_{r+1}-p_{r-1}}{2\cdot\Delta x}\right)_{i-k}\right], \qquad (6)$$

$$\tau_1 = \frac{G\cdot\Delta t\cdot\sum_{k=1}^{n}\left[a^{k-1}\cdot\left(\frac{u}{h_r}\right)_{n-k+1}\right]-\frac{G\cdot\Delta t}{2\eta}\cdot\sum_{k=1}^{n}\left[a^{k-1}\cdot(h_r)_{n-k+1}\cdot\left(\frac{p_{r+1}-p_{r-1}}{2\cdot\Delta x}\right)_{n-k+1}\right]}{1-\left(1-\frac{G}{\eta}\Delta t\right)^n} . \qquad (7)$$

Knowledge of these values allows to calculate the shear at any point of engine operational cycle, definitely at the chosen point of lubricating gap r.

For the construction of computational program it has been assumed that the u speed of gap limiting surface corresponds to the speed of piston in cylinder (see Eq. 2), while the course of oil layer thickness is described by the expression below instead of the Eq. 3.

$$h = h_o + A_m\cdot\sqrt{\left|-sin(\omega\cdot t)-\frac{\lambda}{2}\cdot sin(2\cdot\omega\cdot t)\right|}, \qquad (8)$$

where:

h_o – minimum oil layer thickness,

A_m – amplitude of changes in that thickness.

The assumed form of equation describing the changes in oil layer thickness is an approximation of relevant function where the wedge effect has been taken into account.

4. Computer program for determination of shear in oil layer

Construction of computer simulation program was necessary for determination of shear in oil layer employing dependencies defined earlier. This program has been constructed using the Delphi programming language.

Fig. 3 presents a graphic interface of the program visible on the screen when program is executed. Its form allows to introduce quantities describing:

- technical data of the analyzed oil (dynamic viscosity, shear modulus, minimum oil layer, amplitude of oil layer thickness changes),

- engine construction and conditions of its run (crank radius, bore, crank radius to connecting rod length ratio, angular speed),

as well as compression ring design.

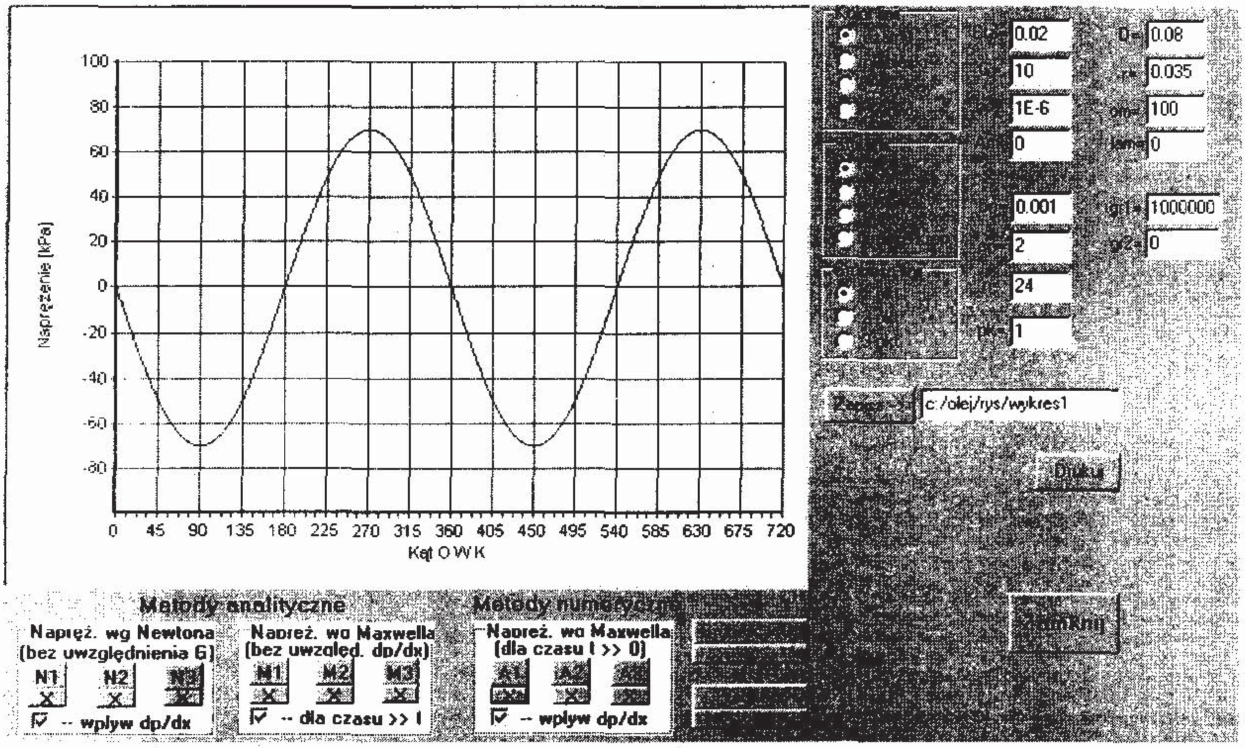

Fig. 3. The view of program graphic interface

The control panel has keys initiating the execution of computations of oil layer shear. They are divided into groups according to the method of computations (analytical or numerical) and the model employed (Newtonian or Maxwell). Moreover, the computation carried out using Maxwell's model and analytical method gives the possibility of taking into account or neglecting the initial phase of shear course (option $t >> 0$). If planes limiting the oil layer are not parallel, activation of the dp/dx option is necessary [4].

For computations of shear in oil layer of any properties (including elastic ones) the last module of presented program, i.e. the one formulated for the Maxwell model carried out numerically would do. This model embraces all possible computational cases, including those for any elastic properties and those for any tilt of limiting planes (the last does not take the initial phase of computations into consideration, i.e. the $\exp(-G/\eta{*}t)$ module has been omitted. However, other modules were left because they check and verify the obtained results.

5. Exemplary results and conclusions

According to presented earlier relations the value of shear t in oil layer depends on oil properties and on run conditions in lubricating gap. The results presented in this chapter consider the parallel cooperating surfaces, i.e. the oil pressure gradient $dp/dx = 0$. The results for non-parallel surfaces (like piston ring face and bore) will be presented in further publications.

Besides the value of shear modulus, the value of only one quantity affecting the shear in oil layer has been changed in a course of tests. Such procedure allowed for a thorough analysis of the effect of any considered factor on shear in oil layer.

The further part of the paper presents the results of research on the effect of some selected data characteristic for oil and run conditions in lubricating gap on the value and course of shear. It has been initially assumed that the gap is fully filled with oil and the speed of moving surfaces is described by the Eq. (2).

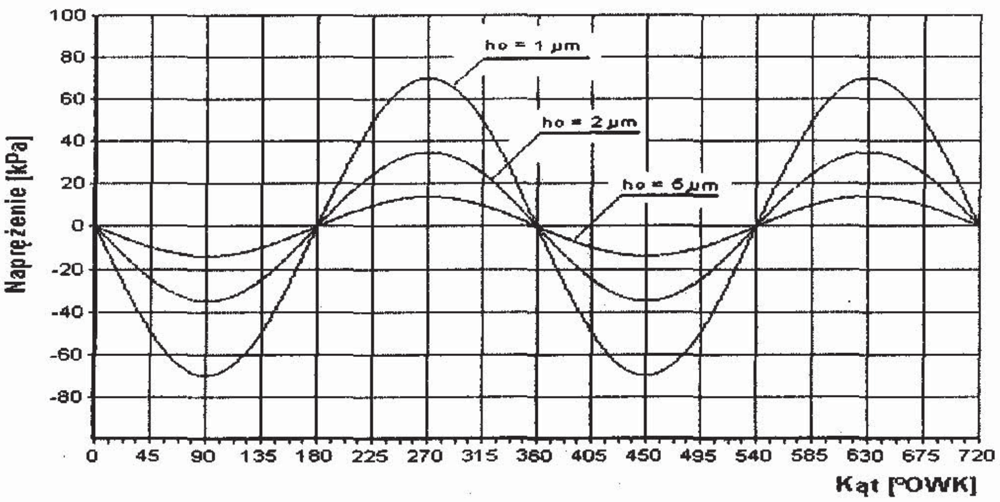

Fig. 4. The change in shear τ value vs. crank angle for selected values of lubricating gap wideness and oil shear modulus $G = 10$ Pa and for $\eta = 0{,}02$ Pas, $\omega = 100$ rad/s, $r = 0{,}035$ m, $\lambda = 0$, $h = 1\,\mu m$

Analyzing the exemplary courses presented in Fig. 4 one can conclude that the changes in lubricating gap wideness h_o are accompanied by the changes in shear τ and phase lag angle φ. An aggregate chart (see Fig. 5) has been prepared on the basis of a number of graphs acquired as a result of series of computations for various values of the shear modulus G.

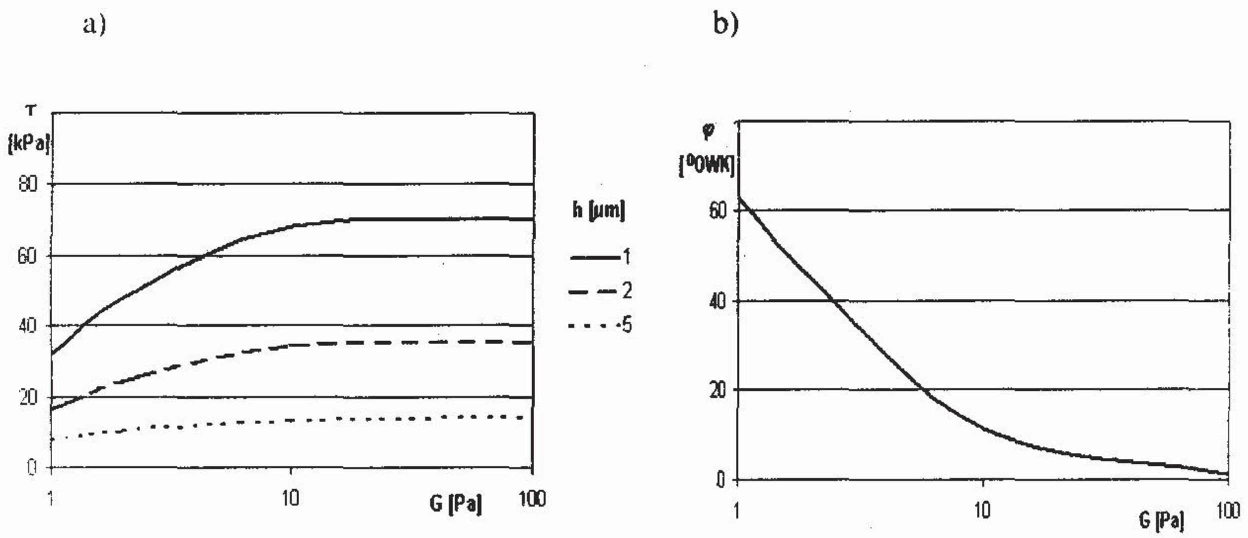

Fig. 5. Dependence of the shear amplitude τ (a) and phase lag angle φ (b) on the shear modulus G for the selected values of oil layer thickness

As it outcomes from the charts presented in Fig. 5 the oil layer shear increases for decreasing lubricating gap wideness and for an increase in shear modulus value. For the analyzed case the limit of an intensive increase in shear value is 10 Pa. The most intensive drop in phase lag angle happens within the same limits and its value does not depend on the layer gap wideness.

An important part of the carried out tests was the evaluation of the effect of oil dynamic viscosity h on the course of oil layer shear for changing values of *G*. The effect of change in viscosity (see Fig. 6) has been evaluated for the gap limited by two parallel surfaces.

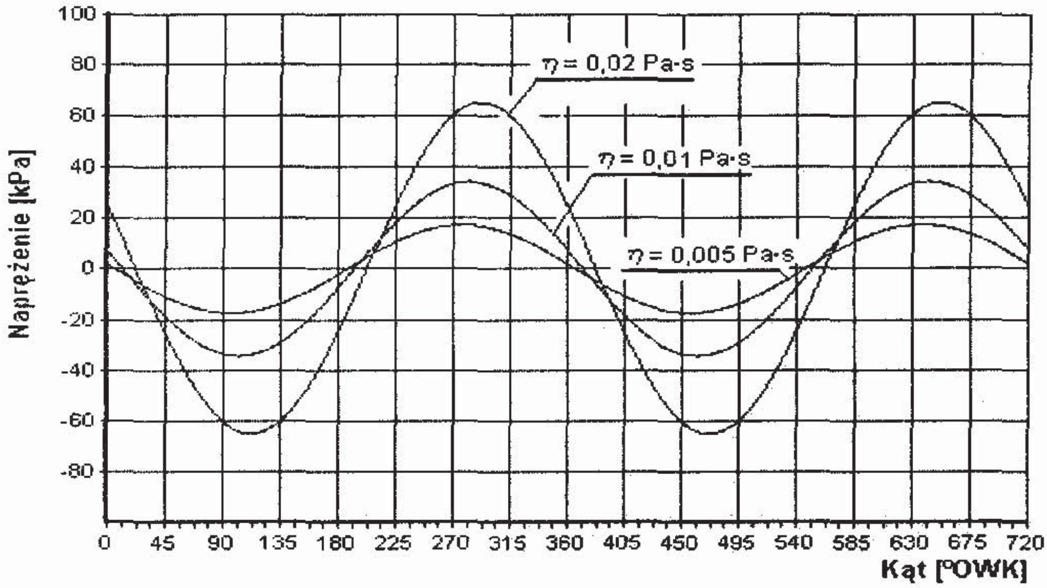

Fig. 6. Changes in shear value τ vs. crank angle φ for selected values of oil dynamic viscosity η and shear modulus G = 100 Pa; ω = 100 rad/s, r = 0,035 m, λ = 0, h = 5μm

As it results from the courses presented in Fig. 6 and from the combined results (see Fig.7), the shear value increases along with the increase in oil viscosity and decreases with an increase in shear modulus value *G*. For the analyzed case the limit of an intensive increase in shear value is *G* = 10 Pa. Opposite to the changes in gap wideness the oil viscosity affects the phase lag angle to the considerable extent.

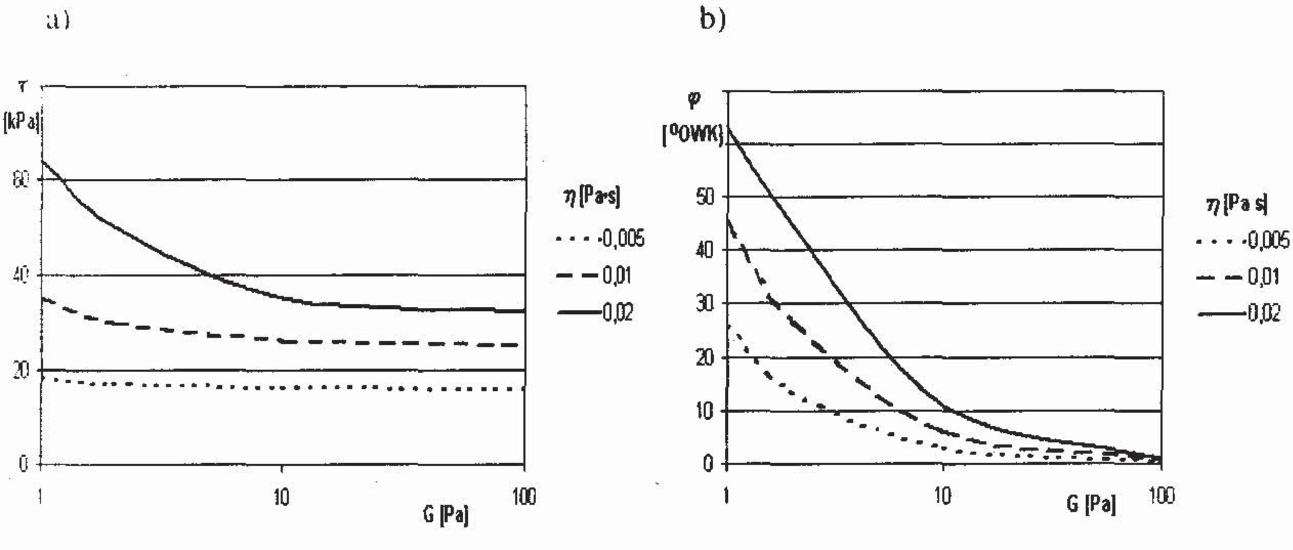

Fig. 7. Effect of shear modulus G on the value of shear τ (a) and on the phase lag angle φ (b) for the selected values of oil dynamic viscosity

Analyzing the curves presented in charts (see Fig. 8) one can conclude that for low values of the shear modulus *G* (up to 10 Pa) this increase is accompanied by the rapid increase in shear *τ*. For the case of phase lag angle this dependence is not so univocal. The angle value increases along with the decrease in shear modulus value and the range of angle rapid increase embraces higher and higher values of shear modulus along with the increase in angular speed *ω*.

a) b)

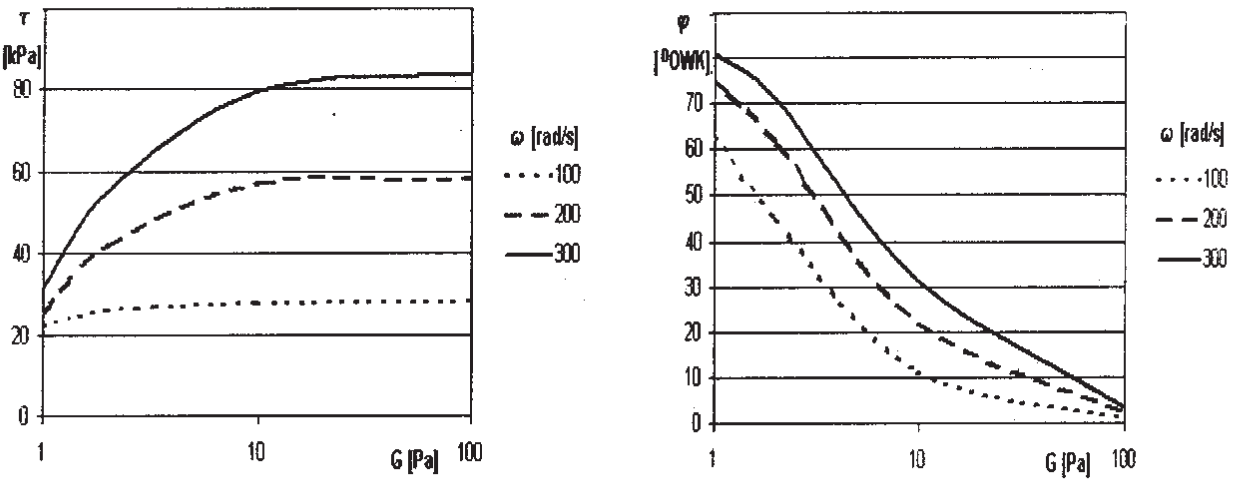

Fig. 8. Effect of shear modulus G on the value of shear τ(a) and on the phase lag angle φ(b) for the selected values of angular velocity ω

The relative speed of surfaces limiting the oil layer affects also the value of shear in oil layer. In a course of another model tests the effect of angular velocity on these values has been analyzed for some values of shear modulus G (Fig. 8).

Exemplary computational results presented in this study relate to the shear in oil layer limited with two parallel planes. The developed computer program allows to extend the analysis on cases of cooperation of arbitrary situated surfaces like for instance piston ring face and bore in order to evaluate friction force and friction loses accompanying the relative movement of these elements.

References

[1] Iskra A.,*Parametry filmu olejowego w węzłach mechanizmu tłokowo-korbowego silnika spalinowego*. Wydawnictwo Politechniki Poznańskiej, Poznań, 2001.

[2] Jakubiak L., Serdecki W., *Badania właściwości sprężystych współczesnych olejów smarujących*. W: 28-th International Scientific Conference on Combustion Engines KONES'2002, September 8 – 11, 2002, Jurata, Poland, s.50-56.

[3] Jakubiak L., Serdecki W., *Metody pomiaru właściwości sprężystych współczesnych olejów smarujących*. W: 28-th International Scientific Conference on Combustion Engines KONES'2002, September 8 – 11, 2002, Jurata, Poland, s.57-63.

[4] Serdecki W., *Badania współpracy elementów układu tłokowo-cylindrowego silnika spalinowego*. Wydawnictwo Politechniki Poznańskiej, Poznań 2002.