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OPTIMISING THE SELECTION OF WORKING MACHINE GEAR LUBRICANT OIL VISCOSITY TO MAXIMISE GEAR MESHING DURABILITY

Abstract. The paper presents a systematised approach to the selection of optimum viscosity of gear lubricant for drives of working machines used in civilian and military areas. Traditional literature procedures are characterised for determining the required oil viscosity according to the hydrodynamic theory (HD theory), and procedures based on the theory of elastohydrodynamic lubrication (EHD theory) are also presented. Nomograms developed by the authors allow to determine the dimensionless friction parameter resulting from the physical nature of the frictional contact in the lubricated gear meshing. The relationship between this parameter and the main types of tribological wear (abrasion, contact fatigue and seizure) of the gear teeth is demonstrated.

Keywords: working machine drives, gears, lubrication, tribological life of gear meshing.

1. INTRODUCTION

The problem of gear lubrication of stationary and mobile working machines, in spite of the on-going cognitive and utilitarian research, particularly that pertaining to optimising the selection of viscosity class of the required lubricating oil, must be regarded as open at the stages of design, manufacture and operation of the mentioned gear units. At OBRUM that problem is one of the prevailing directions of cognitive, research and development and implementation work which has been running for more than 20 years (e.g. [4, 15, 28, 29]).

The main problem is that optimisation of gear lubrication must take into account many "poorly" defined criteria or baseline data which represent incomplete and fuzzy sets. Under such circumstances actions are based mainly on recommendations that are a compromise between knowledge derived from theoretical, experimental, laboratory and utilitarian research and from practical experiments. The necessity of such compromise results from the differences between theoretical-experimental (cognitive) and utilitarian (practical) approach.

In the *theoretical and experimental approach*, the essential criterion is based on the formulation, which in short can be defined as the physical essence of lubrication, meant as effective separation of two mating surfaces with a film of lubricant in order to replace detrimental external friction of the contacting parts with friction that occurs partially (mixed friction) or completely (fluid friction as a transient state - boundary friction) inside the lubricant film. This indirectly corresponds to the criterion of limited or complete elimination of tribological wear of the outer layer of frictional elements.

In the *utilitarian-practical approach*, the decisive criteria take into account the assessment of such parameters as: durability (in more general terms: reliability), thermal state, vibroacoustic state, safety of the gear transmission used in the drive of a machine. In this approach we deal with the physicochemical essence of friction contact and the effects thereof in the micro (or even nano) scale. Such approach results in formulating guidelines

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(recommendations) for operating a machine in accordance with Tactical and Technical Objectives (in the case of, for instance, a military mobile machine) or in accordance with manufacturer's technical documentation (of, for instance, a mining shearer or excavator).

As so far there are no direct correlations between the theoretical-experimental results obtained (usually transferred into practice by applying the theory of physical similarity [7, 8]), and hence the previous remark that the compromise required consists not in finding an "absolutely" optimal criterion, but rather in approximating the optimum. The state of this approximation depends to a great extent on the adopted base and on methods of formulating recommendations ranging from the most simple (classic), based on hydrodynamic lubrication theory, to the advanced ones based on the results of model tests. Contemporary approach takes into account the thermo-elastohydrodynamic lubrication theory and is supported by experimental research [1, 2].

2. CLASSIC PROCEDURES OF SELECTING VISCOSITY OF GEAR LUBRICANT OILS

Classic (historically the oldest) methods of selecting gear lubricant viscosity are based on conclusions drawn from the one-dimensional form of the Reynolds equation (1886) written as

$$\frac{dp}{dx} = 6 \cdot \eta \cdot v \cdot \frac{h - h_0}{h^3} \tag{1}$$

Equation (1) is interpreted in the following way: pressure increase dp/dx along the constricting lubrication gap (h-h_0) depends on the product (adopted as constant) of dynamic viscosity η and relative velocity v of the mating parts. Since both η and v appear in the first power, it is justifiable to say that low peripheral velocity on the wheel diameter requires a higher viscosity than at higher speeds. This forms basis for setting up tables for determining the required lubricant viscosity (e.g. [13]).

The relatively wide range of the required lubricant viscosity is due to the following:

- the gear train is usually of multistage type,
- peripheral velocity of the gears is not constant during transmission operation.

An approximate graph that illustrates the relationship between the required kinematic viscosity and velocity (Fig. 1) can be plotted based on a table of recommendations [13].

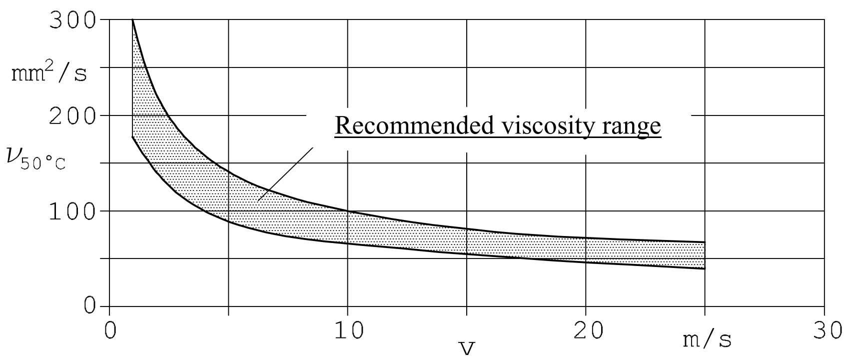

Fig. 1. Kinematic viscosity (expressed in mm²/s) at oil temperature of 50°C for selected peripheral velocities (expressed in m/s) of the first stage gear

It should be noted that according to the current ISO 4448 classification viscosities of industrial oils are referred to 40°C, therefore the kinematic viscosity coefficients shown in

Figure 1 should be corrected in accordance with Figure 2 plotted for mineral oils with viscosity index of 90.

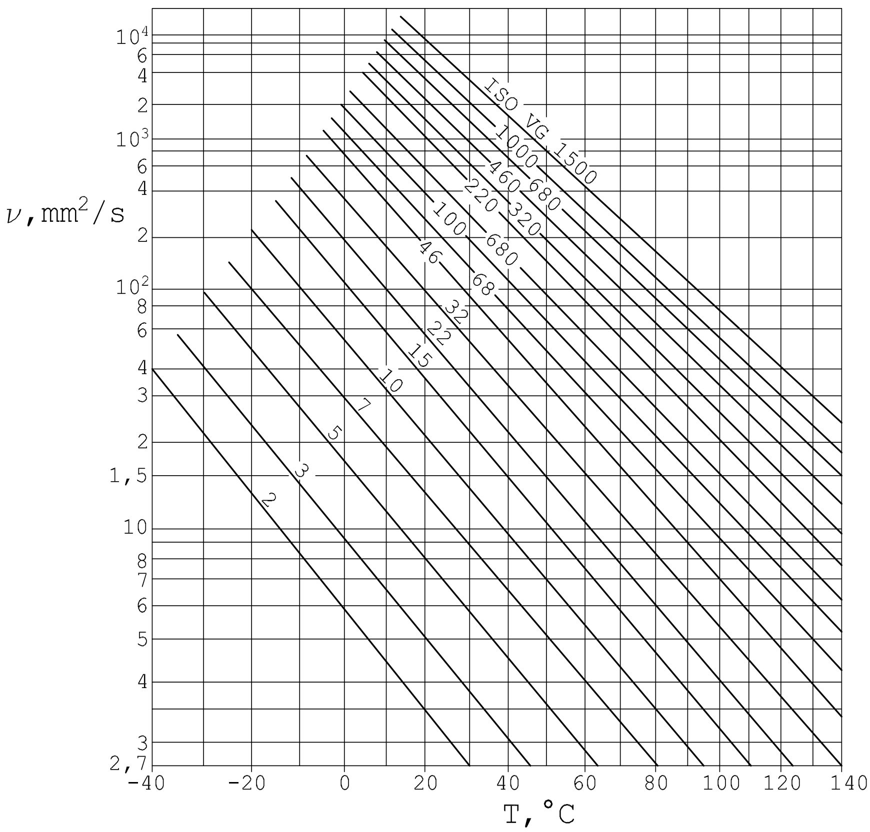

Fig. 2. Viscosity of mineral oils with viscosity index 90 vs. operating temperature. Note: VG 2 ... VG 1500 refer to oil viscosity class according to ISO 4448

The nomogram according to DIN 51519 [11], shown in Figure 3, is a certain development and upgrade of the problem of classic selection of gear oil viscosity class.

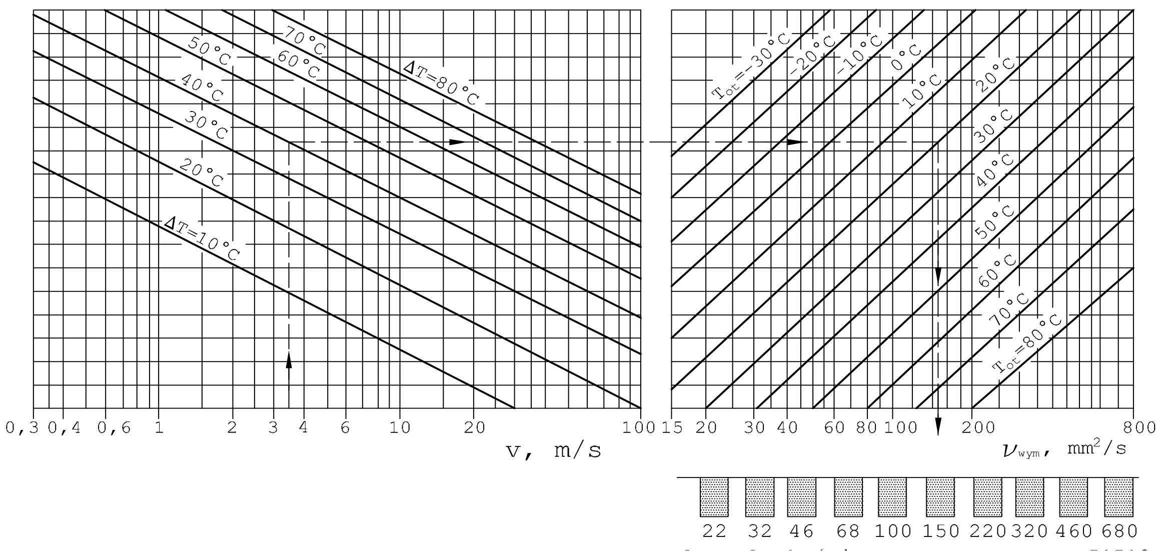

Fig. 3. Nomogram for determining the required viscosity class of gear oil according to DIN 51519 [2, 11]

This nomogram is used to determine the oil viscosity class according to ISO as a function of the gear speed *v* (m/s), with account taken of the operating temperature of the transmission calculated as the sum of ambient temperature T_{amb} (°C) and assumed oil temperature rise ΔT (°C) under steady operating conditions.

The classic procedure of gear lubricant oil selection, shortly described here, is commonly used by European manufacturers of gear transmissions. However, taking into account the current state of the art of tribology with respect to the gear meshing, this procedure should be regarded as approximate (preliminary), the results of which must be verified due to the criteria resulting from the minimisation of the tribological wear (improved durability) of the gears and bearings.

3. OPTIMISATION OF THE SELECTION OF GEAR OIL VISCOSITY CLASS BASED ON THE ELASTOHYDRODYNAMIC LUBRICATION THEORY

In the 1960s new fundamental works appeared, related to both new understanding of the physical essence of the external friction phenomenon in solids [3], as well as to the modification of the Reynolds equation to take into account the deformability of the contact zone and the variation of oil viscosity in that zone under increased pressure [6]. That modification assumed the name of the theory of elastohydrodynamic lubrication (EHD). Based on this theory of contact of conformal surfaces, formulas were derived for describing the contact of involute teeth [2, 11, 19] enabling defining the EHD oil film thickness to identify the friction conditions in the gear mesh (fluid, mixed, boundary).

The applicable procedures (methods) for determining the required oil viscosity for a given gear transmission can be graphically depicted as in Figure 4.

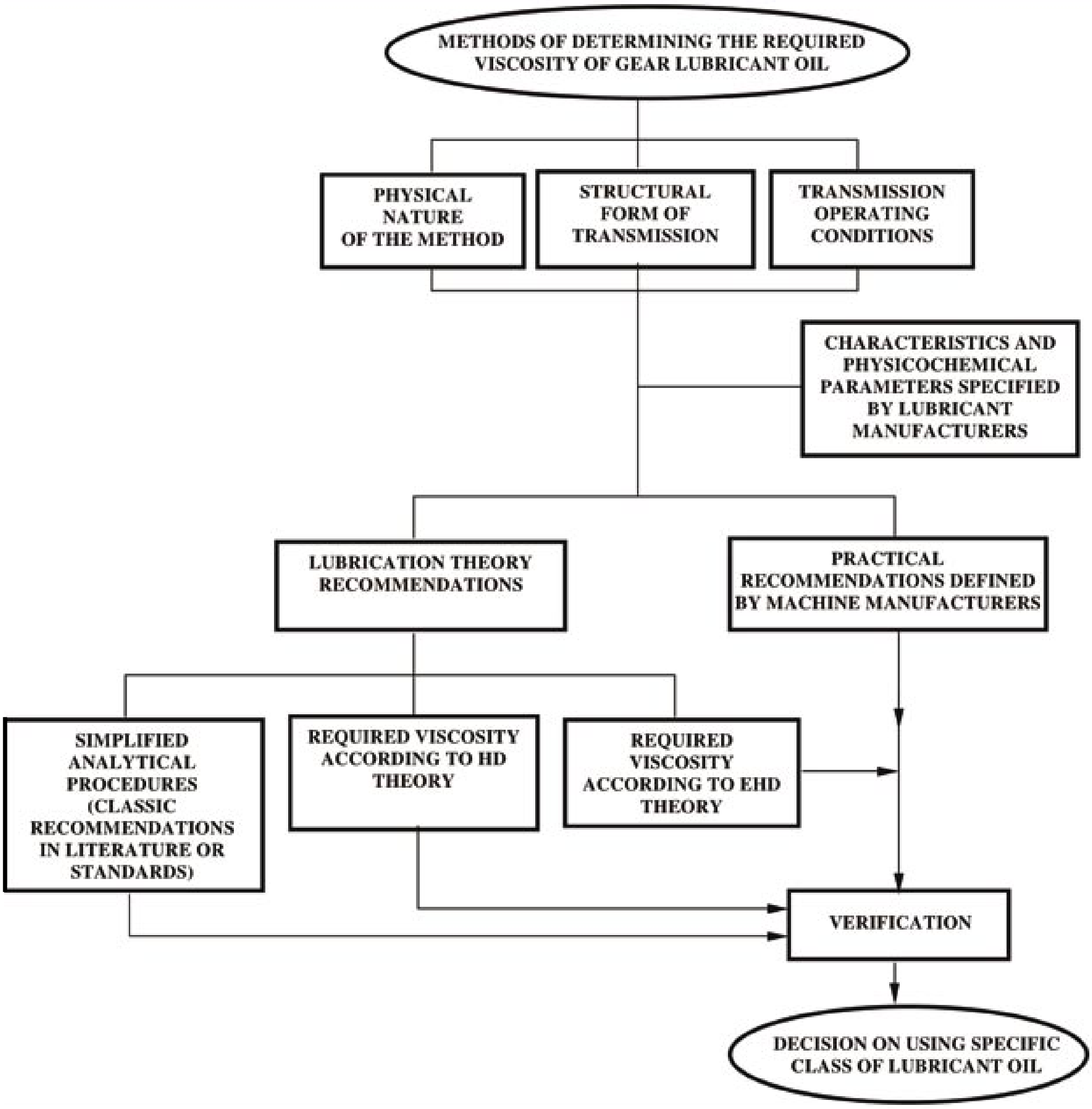

Fig. 4. Graphical representation of procedures for selecting optimum viscosity of lubricant oil for cylindrical gear

The algorithm of the possible procedure of selecting oil viscosity shown in Figure 4, indicates that:

- the use of lubricant oil of specified viscosity class should be accordant with the recommendations given in the technical documentation (TD) of the respective working machine. These recommendations should not be accepted indiscriminately, and at the present stage of development of tribological engineering there is a sufficient basic knowledge allowing for the verification of these recommendations using literature data, particularly lubrication theories;
- currently the basis for accepting a verification procedure is the EHD theory that takes into account the change of rheological parameters of the oil as a function of pressure and temperature in the meshing contact zone and of the modification of the shape of the lubrication gap resulting from deformation of the mating teeth under the transferred contact load. This method, which applies the EHD theory, consists primarily in the determination of the minimum

thickness of the elastohydrodynamic film of oil in the mesh, which is done using the Dowson-Higginson equation [5] written in the following non-dimensional form (see Fig. 5):

$$S = 1.6 \cdot Q^{-0.013} \cdot M^{0.6} \cdot H^{0.7} \tag{2}$$

or in an expanded form:

$$\frac{h_{0,min}}{\rho} = 1.6 \left(\frac{Q_1 \cdot d_1}{E \cdot \rho}\right)^{-0.013} \cdot (\alpha \cdot E)^{0.6} \cdot \left(\frac{\eta \cdot v}{E \cdot \rho}\right)^{0.7} \tag{3}$$

Equations (2) and (3) may be interpreted in the following way: the relative minimum lubricating film thickness S in the mesh is determined by the criterion of the contact deformability of the active faces of the teeth Q, the generalised "material" criterion M and the criterion of the contact load transfer by the elastohydrodynamic film of oil with the dynamic viscosity parameter η resulting from pressure, temperature and strain in the contact zone (see Fig. 5).

Symbols in formula (3) have the following meanings [10, 13]:

$h_{o,min}$ - minimum thickness of lubricating film between mating tooth profiles,
ρ - effective radius of tooth curvature, usually referred to the pitch circle of the mating driving gear ρ_1 (pinion) and driven gear ρ_2

$$\rho = \frac{\rho_1 \cdot \rho_2}{\rho_1 + \rho_2}$$

Q_1 - specific load on pinion

$$Q_1 = \frac{F_n}{b \cdot d_1}$$

where:
F_n – normal load on meshing,
b – length of teeth contact line,
d_1 – pitch (nominal) diameter of driving gear,

E - effective modulus of elasticity of the driving gear material E_1 and of the driven gear material E_2 (assuming that gears are made of steel, then Poisson's ratio $\upsilon_1 = \upsilon_2 = 0.3$)

$$E = \frac{2E_1 \cdot E_2}{E_1 + E_2}$$

α - oil viscosity variation factor

$$\eta_p = \eta_0 \cdot e^{\alpha p}$$

where:
η_p, η_o – dynamic viscosity coefficients, respectively: at pressure p in contact zone and atmospheric pressure p_0,
e – base of natural log,
α – factor of oil viscosity variation with pressure change, dependent on type and structure of lubricant; for mineral oil $\alpha = (1 \ldots 3)\cdot 10^{-8}$ m^2/N,
$\eta = \eta_{p,T}$ – viscosity at pressure p and assumed operating temperature of transmission T,

v - sum of peripheral velocities of pinion v_1 and gear v_2 along the path of contact

$$v = v_1 + v_2$$

whereas for the peripheral velocity at pitch diameter $v = 2v_1$.

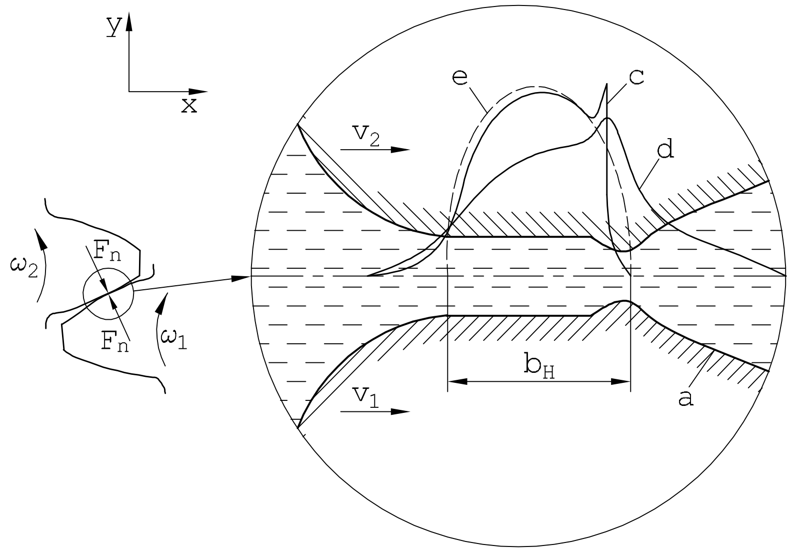

Fig. 5. Shape of the lubrication gap (a), distribution of contact pressures according to Hertz (b), of pressure (c) and of temperature (d) in elastohydrodynamic contact between teeth and involute profile

The minimum oil film thickness in the tooth contact zone ($h_{o,min}$) derived from equation (3) can be referred to the effective roughness parameter R_{ek} of the driving gear (R_{a1}) and of the driven gear (R_{a2}), resulting in the friction parameter λ defined as

$$\lambda = \frac{h_{0,\min}}{R_{ek}} \tag{4}$$

where $R_{ek} = \dfrac{R_{a1} + R_{a2}}{2}$.

If λ exceeds boundary value λ_{gr} (established by experimental testing, usually λ_{gr} = 1.4 ... 2.0), then there is high probability of fluid friction in the meshing, meaning that the condition of expected friction at stable operation is fulfilled.

According to [11], the friction parameter λ can be derived as a function of the peripheral velocity of the driving gear from the graph shown in Figure 6.

Figure 6 shows that high probability of fluid friction in the meshing occurs at peripheral velocities greater than 10 m/s. At lower speeds mixed friction occurs, and boundary friction occurs when starting.

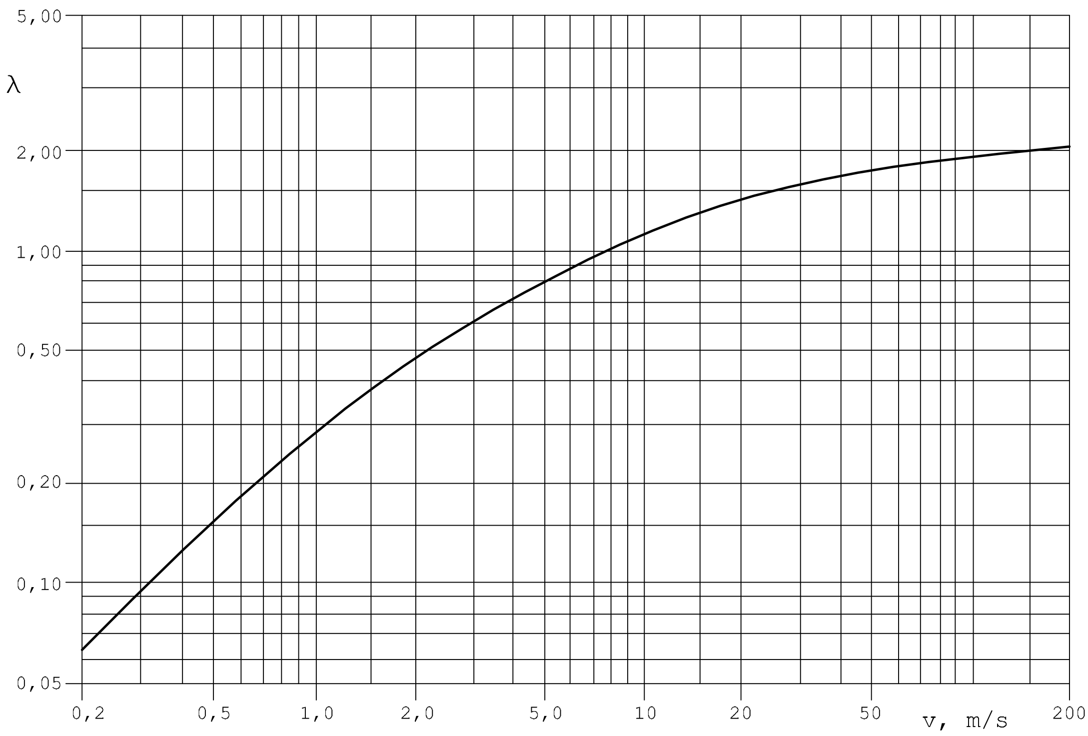

Fig. 6. Friction parameter λ vs. peripheral velocity v of driving gear on pitch diameter

Based on formula (3), a nomogram (Fig. 7) presented in [20] allows to determine the minimum thickness (in the narrowest cross section) of the lubrication gap $h_{0,min}$ depending on the peripheral velocity on the pitch diameter v of the driving gear and on: transmission ratio K_u of the stage, load K_Q, distance between gear axes K_a of the stage and pre-selected viscosity K_υ.

When the lubricant film thickness in the meshing $h_{0,min}$ and the effective roughness of the active faces of the teeth R_{ek} are known, the following condition can be verified:

$$\lambda = \frac{h_{0,\min}}{R_{ek}} \geq \lambda_{wym} \tag{5}$$

where:-

λ_{wym} – required friction parameter determined statistically on the basis of experimental data; usually $\lambda_{wym} = 1.4 \ldots 2.0$, and in the case of heavy duty and high load meshing $\lambda_{wym} = 2.0 \ldots 4.0$.

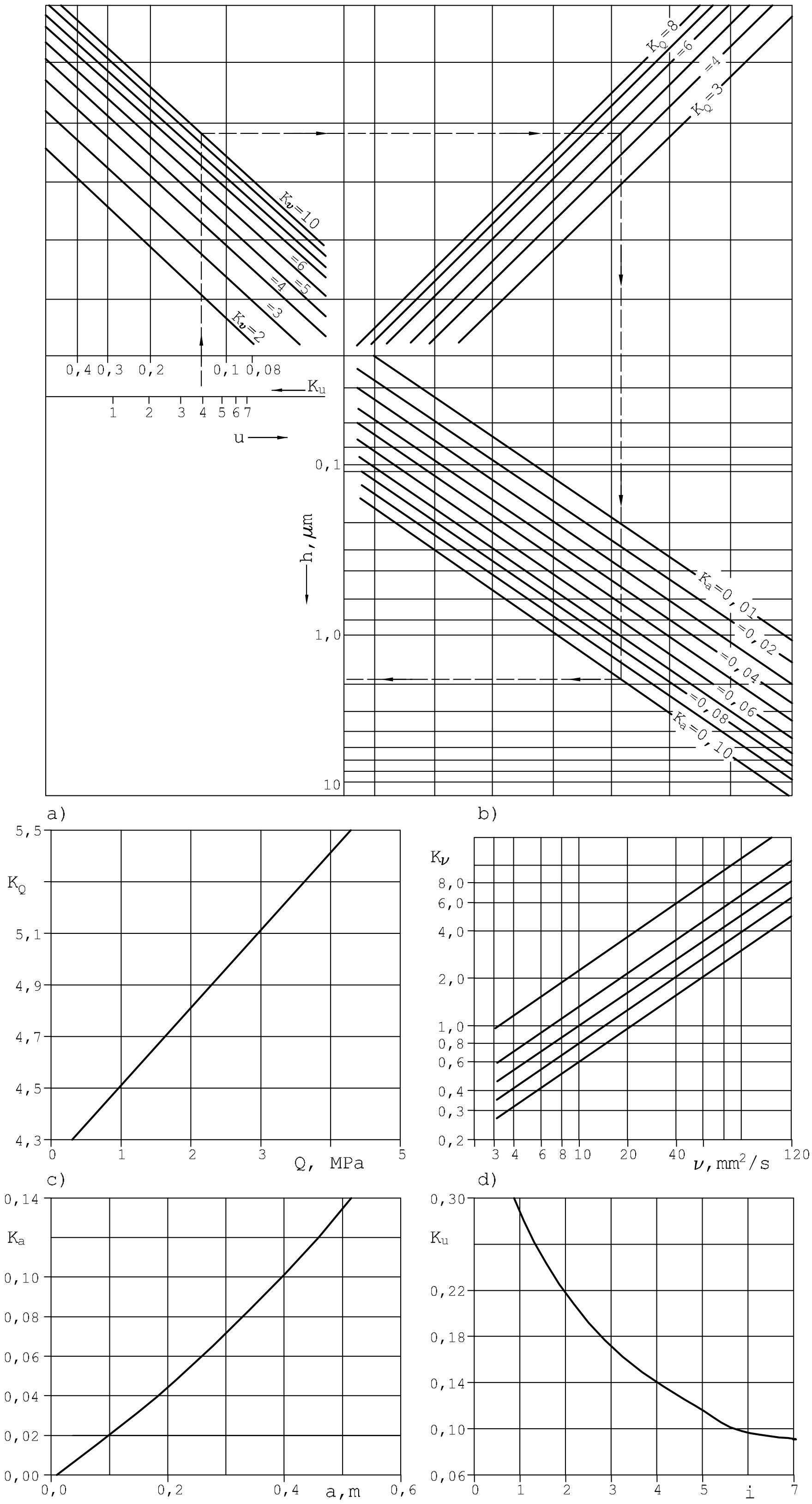

Fig. 7. Nomogram for determining the minimum lubricant film thickness $h_{o,min}$ (μm) based on EHD theory as the function of:

a) gear load factor – K_Q, b) lubricant kinematic viscosity factor – K_υ, c) axes distance factor – K_a, d) transmission ratio factor – K_u. Note: more detailed information is given in [17, 21].

Figure 7 shows an example of determining the minimum film thickness for the following gear operation conditions:

– specific load Q_1 = 2.5 MPa,
– transmission ratio of the stage u = 4.0,
– kinematic viscosity of lubricant VG 220 at assumed operating temperature T_{rob} = 60°C is υ_{rob} = 100 mm^2/s,
– distance between gear axes of the stage a = 400 mm.

As shown in Fig. 7, following the direction indicated, the value arrived at is $h_{0,min}$ = 1.9 µm. Assuming that the active faces of teeth are ground ($R_{a1} = R_{a2}$ = 0.32 µm), then for R_{ek} = 0.32 µm the friction parameter is

$$\lambda = \frac{h_{0,min}}{R_{ek}} = \frac{1.9}{0.32} \approx 5.94$$

and thus the condition $\lambda > \lambda_{wzm}$ is satisfied, meaning that during stable operation of the gear transmission there will be fluid friction in the meshing.

4. EFFECT OF LUBRICANT OIL VISCOSITY ON THE TRIBOLOGIC DURABILITY OF THE GEAR TRANSMISSION

The classic theory of isothermal flow of viscous fluid (oil) in a narrow gap (see formula (1)) teaches that with increased viscosity one obtains increased hydrodynamic bearing capacity, and thus durability, and more effective hermeticity of the friction pair, but at the same time increasing resistance to oil flow and agitation, which significantly affects the efficiency (or thermal state) of the node analysed.

As indicated by lubrication engineering [16, 17, 18], the problem of the effect of lubricant oil viscosity on the tribological durability of meshing and bearings should be verified using the generalised form, defined under item 3 of this paper by the friction parameter λ ($\lambda = h_{0,min} / R_{ek}$).

Relationship between the intensity of abrasive wear I_z of the active faces of gears (Fig. 8) and of Hertzian breaking contact stress p_H (i.e. contact fatigue [25]) and the friction parameter λ (Fig. 9) was experimentally determined in the course of authors' studies [16, 18] and on the basis of literature data [26, 27].

Figure 8 indicates that for $\lambda > 1$ the wear intensity is low enough to state that the condition indirectly defines the occurrence of fluid friction in the meshing.

Figure 9 shows, based on authors' studies [25] and on literature data [26, 27], a curve illustrating the relationship between boundary contact stress p_H that causes pitting of cylindrical samples and the friction parameter λ. The figure shows that increase of λ from 0.1 to 1.0 brings about a twofold increase of the pitting-causing contact stress.

Figure 10 presents the relationship between the probability p of teeth seizure and the λ parameter [10]. The figure indicates that for $\lambda > 1$ the probability of seizure is lower than 0.1.

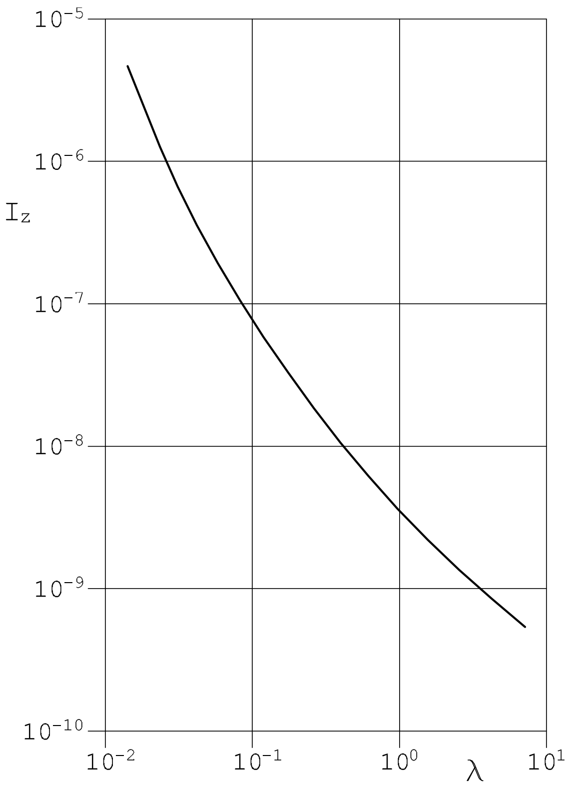

Fig. 8. Intensity of abrasive wear I_z of gears vs. friction parameter λ

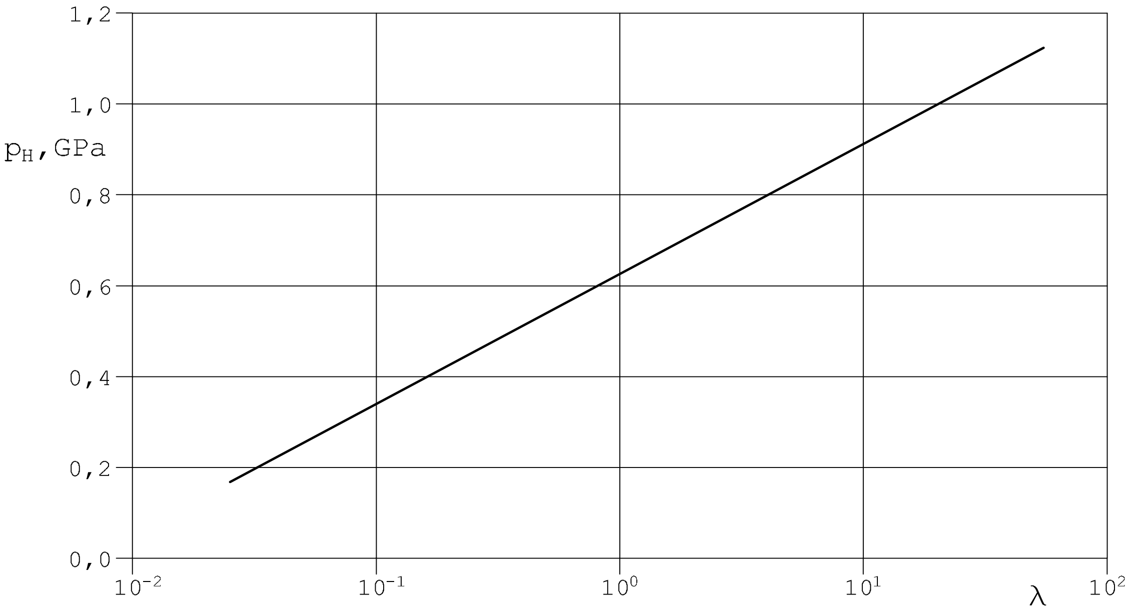

Fig. 9. Pitting-causing boundary contact stress vs. friction parameter λ

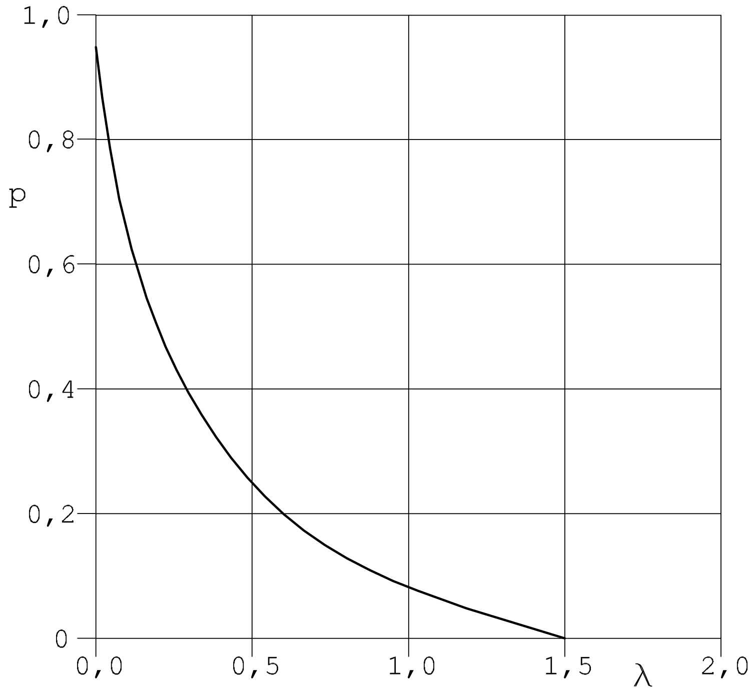

Fig. 10. Probability of teeth seizure *p* vs. friction parameter *λ*

Conclusions drawn from the analysis of figures 8 to 10 justify the adoption of the fluid friction condition in the meshing expressed in the form of relation (5) as the starting point for determining the required viscosity of the lubricant, assuming the required value of λ_{wym} determined from statistical tests of gear operation in drives of specified working machines.

5. SUMMARY AND CONCLUSIONS

As shown in this publication, the selection of the optimum viscosity class of gear lubricant for gear transmissions of working and mobile machines, including those designed for military use, is a broad and complex issue, as illustrated by the algorithm developed and shown in Figure 4. The complexity of this problem lies, on the one hand, in the need to rely on theoretical solutions of the lubrication theory, which are based on model assumptions that diverge strongly from the real situation. On the other hand, it is necessary to take into account the unequivocal identification of the state of operational loads, which are characterised by randomly variable forces in the life cycle of the transmission with a large number of high overloads. Under these circumstances there is a need for a compromise consisting in maximising the use of practical information and in verifying theoretical models.

The following are the main conclusions drawn from the paper:

- determination of the required kinematic viscosity coefficient of oil, which is the main criterion for optimising gear lubrication with regard to the assumed durability of meshing, requires the application of the classic hydrodynamic theory (HL) followed by the verification of the results using the equations of the elastohydrodynamic theory (EHD);
- the parameter conceived by the EHD theory, which characterises the physical nature of rheological fluid flow in a narrow deformable contact zone, is a generalised criterion of similarity of cylindrical gear meshing lubrication conditions.

The paper presented summarises the results of long-term research conducted at OBRUM on the improvement of the durability of gear transmissions in the drives of fast-moving tracked vehicles.

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