INFLUENCE OF LUBRICANTS ON THE PERFORMANCE OF GEARS

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Abstract

The load carrying capacity, life span and reliability of gears enormously depends on the effectiveness of lubrication. Choosing suitable lubricants and lubrication technology the optimal lubrication state can be achieved ensuring the longest life span and higher operational reliability of gears. In practice generally elastohydrodynamic lubrication is preferred by spur gears where the viscosity and pressure-viscosity coefficient of lubricants play an important role to reach the necessary oil film thickness between the gear teeth to ensure the best performance.

With increasing the oil viscosity and the pressure-viscosity coefficient the oil film thickness and the reliability of operation would be higher, but also the friction loss and the temperature of operation will increase. So at choosing the viscosity of lubricants it is necessary to take into consideration the contradictory requirements of the low friction and thick oil film.

INTRODUCTION

It is well know, that at damages of machines higher lubricating oil viscosities are oft suggested to improve their performance. Choosing lubricants for rolling bearings or gears for their heavy-loaded applications a higher oil viscosity is oft suggested. In many cases design subscriptions prescribe to use lubricants of higher viscosity to increase the load carrying capacity of mating machine parts as tooth gears, worm gears, sliding bearings, rolling bearings. However, higher oil viscosity not only increased the load carrying capacity but, at the same time, it leads to higher heat generation causing excessive heat and higher temperature, intensive wear or severe damages of the mating surfaces. Therefore it is necessary to find an optimum solution at choosing the viscosity grade of lubricants intended to lubricate machines, especially heavy-duty gears.

EFFECT OF VISCOSITY ON THE OIL FILM THICKNESS OF GEARS

Theoretically the loaded teeth of gears contact in a small, localized area due to the elastic deformation of the teeth, where high pressures develop making difficult that continuous lubricant films fully separate the contacting surfaces. Such continuous oil films are necessary to maintain low friction losses, to hinder the wear and the scoring of the mating surfaces and to reach high load carrying capacities of gears. High oil viscosity and adequate large contact area are necessary to develop an oil film being able to trust the applied load. The elastohydrodynamic theory of lubrication describes the laws of the formation such a continuous lubricating oil films, according to which the main factors influencing the thickness of fluid lubricant films between the teeth are the viscosity, and the viscosity-pressure coefficient of lubricant, the sweeping velocity, the geometry and also the mechanical properties of the contacting surfaces.

Using the formula of Dowson and Higginson [1] the lubricant film thickness between the teeth of spur gears was determined for a given drive. The results of calculation are presented in Fig. 1- 4. Increasing the gear ratio of drives decreased the film thickness, therefore the chosen gear ratio in the calculation was 5, which is generally the highest value used at

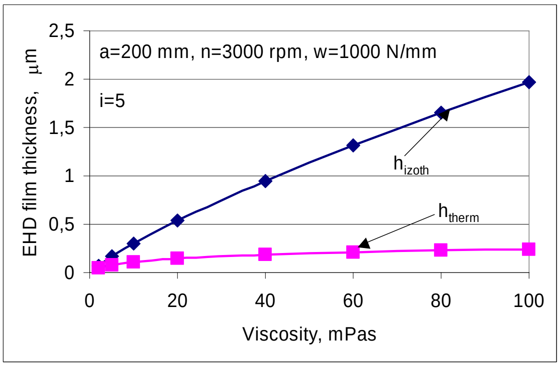

Fig. 1. Calculated EHL film thickness between teeth using isothermal and thermal EHD theory

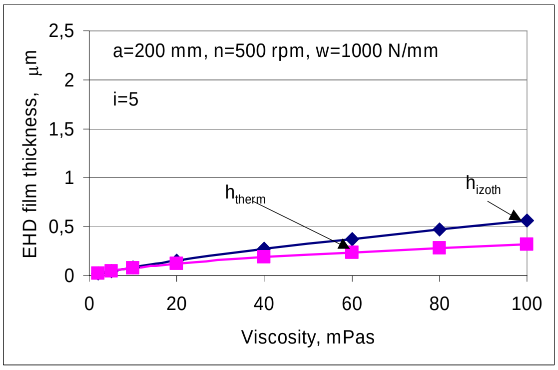

Fig. 2. Calculated EHL film thickness between teeth using isothermal and thermal EHD theory

industrial gears. Raising the viscosity of lubricants and the rotation speed increases the oil film thickness, as can be seen in these Figures.

Higher speed and viscosity cause higher temperature in the lubricant film decreasing the effective viscosity and so the oil film thickness. This effect is proved by the calculated results of film thickness using the isothermal EHL theory (h_{izoth}, where the temperature and the viscosity is constant), and the thermal EHL theory (h_{therm}, where the effect of increasing temperature is taken into consideration). As can be seen, at lower rotation speed of 500 rpm, there is little difference between the isothermal and thermal film thickness. At higher rotation speed of 3000 rpm, however, the thermal film thickness is much lower than the isothermal one. Comparing the diagrams of Fig. 1 and Fig. 3 shows that, in the higher viscosity range, the thermal lubricant film thickness is lower at rotation speed of 3000 rpm than at rotation speed of 500 rpm proving the intensive increasing of the oil film temperature at higher speed, and causing the ineffectiveness of the higher oil viscosity.

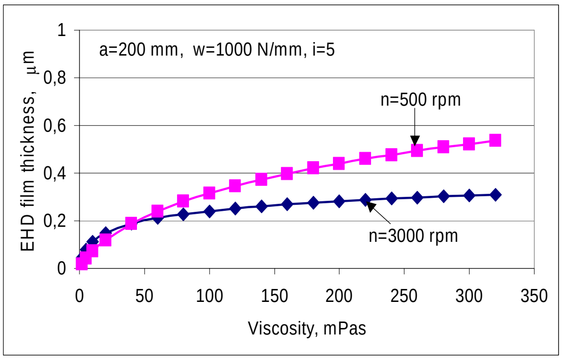

Fig. 3. Calculated thermal EHD film thickness between teeth using thermal EHD theory at different rotational speed

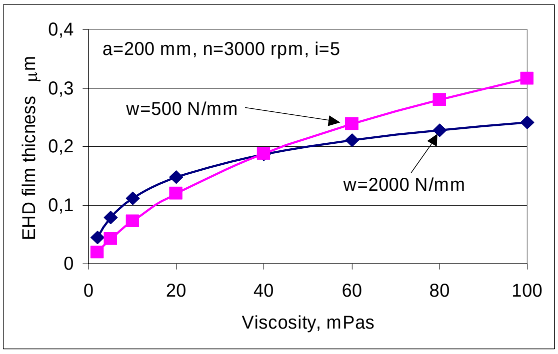

Fig. 4. Calculated thermal EHD film thickness between teeth using thermal EHD theory at different load level

It is interesting to see, that the applied load between the teeth has not too large effect on the EHL film thickness (Fig. 4), and on the coefficient of friction (on the effectiveness of gears) as it will be presented in the following Figures showing the calculated coefficient of friction and efficiency of gears using different methods presented in the technical literature.

At heavy duty applications there are suggestions to use partial synthetic gear oils, or full synthetic gear oils while they have some beneficial properties comparing to mineral oils: their

Viscosity Index is higher, molecule structure is more stabile and some types have good friction properties. Synthetic gear oils, which are generally applied in machinery, are synthetic hydrocarbons (PAO) (polyalpha olefines), polyalkylene glycols (PAG) and polyesters (PEO). But calculations of the EHL film thickness between the mating teeth of gears revealed, that some synthetic oil gave lower film thickness than the mineral oils (MO), at relative low temperatures. As can be seen in Fig. 5 the polyalpha olefin results lower EHL film thickness below 80 °C, the polyalkylene glycol below 60 °C.

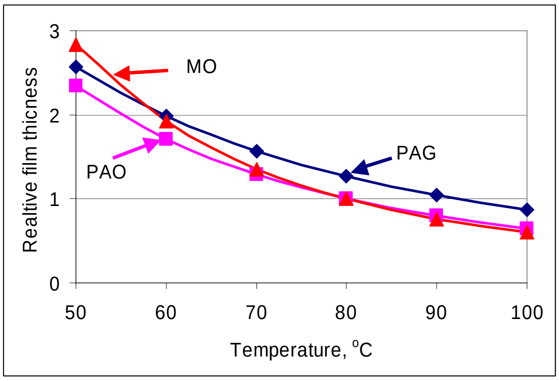

Fig. 5. Calculated EHD film thickness between gear teeth using different lubricants (MO-mineral oil, PAO - polyalpha olefin, PAG - polyalkylene glycol)

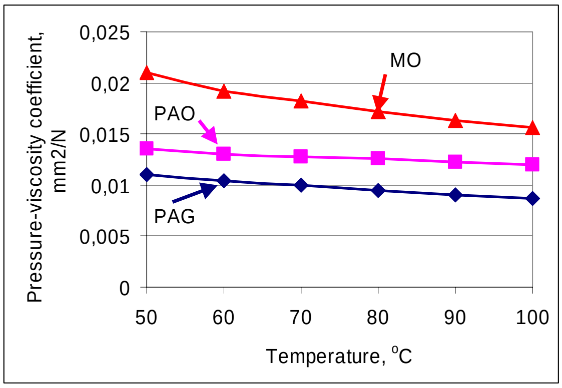

Fig. 6. Variation of the pressure-viscosity coefficients of different oils in function of temperature (MO-mineral oil, PAO - polyalpha olefin, PAG -polyalkylene glycol)

These can be explained with the lower pressure-viscosity coefficient presented in Fig. 6. But the film thickness at application of synthetic oils will increase at higher temperature owing to the higher viscosity indexes, which ensures higher dynamic viscosity at higher temperature for synthetic oils comparing to mineral oil (Fig. 7).

These phenomena suggested using synthetic oils to lubricate gears only at temperatures above 80 °C. At the same time other beneficial properties of synthetic oils made it possible to use them in a much wider application conditions, as it is proved recently in many fields.

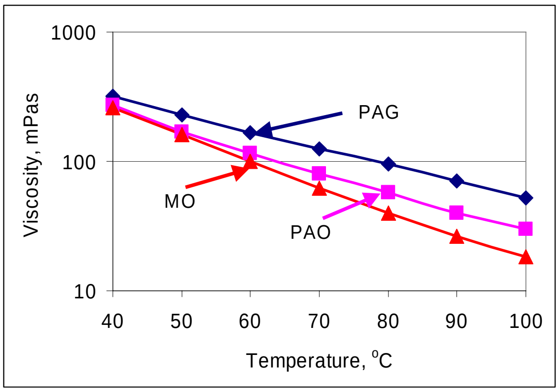

Fig. 7 Variation of the dynamic viscosity of different oils in function of temperature (MO-mineral oil, PAO - polyalpha olefin, PAG - polyalkylene glycol)

EFFECT OF VISCOSITY ON THE EFFICIENCY OF GEARS

Using gear oils with optimum selected base oils, additives, VI-improvers and viscosities friction power losses can be decreased resulting in increasing efficiency. The gear oil influence on these losses not only can be obtained for high load and high-speed applications but also under partial load and idling conditions [5].

The increase of EHL film thickness leads to decrease the coefficient of friction and to raise the efficiency of gears as can be seen in Figures 8-11. Figures 8 shows the coefficient of friction between the teeth of gears, calculated by formula suggested in [2]. This formula takes into consideration only the viscosity of lubricants at the bulk temperature of gears, the relative radius of curvature at the pitch point, and the entraining velocity at the pitch line.

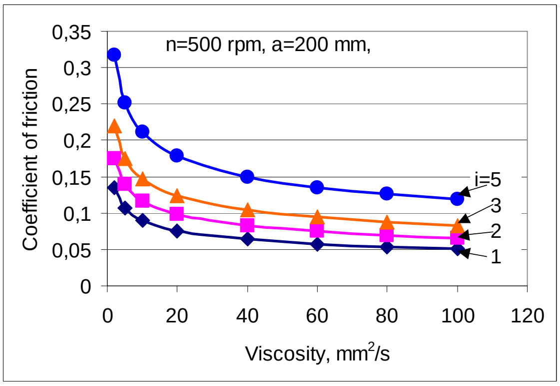

Fig. 8. Coefficient of friction of contacting teeth in function of lubricant viscosity at different gear ratio

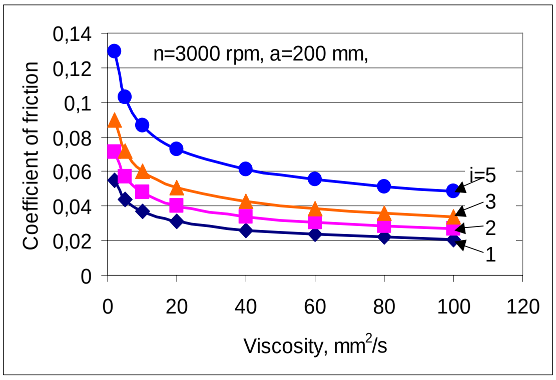

Fig. 9. Coefficient of friction of contacting teeth in function of lubricant viscosity at different gear ratio

It can be seen in these figures that increasing the viscosity of lubricant decreases the coefficient of friction. It also can be stated that at larger gear ratio the coefficient of friction is higher, because the entraining velocity is lower owing to the smaller rolling diameter of the pinion.

The coefficient of friction is much lower at higher rotation speed of 3000 rpm (Fig. 9) than at lower rotation speed of 500 rpm (Fig. 5) proving the beneficial effect of elasto-hydrodynamic lubrication. Accordance with the coefficient of friction the efficiency of gears rising with increasing the viscosity of lubricant and the rotation speed, as the diagrams presented in Figures 10-11 prove them.

The efficiency of gears was calculated by the formulas published in [3]. The friction loss is lower and so the efficiency of gears is higher at lower gear ratio due to the more beneficial EHL conditions (the larger relative curvature radius and the higher sweeping speed).

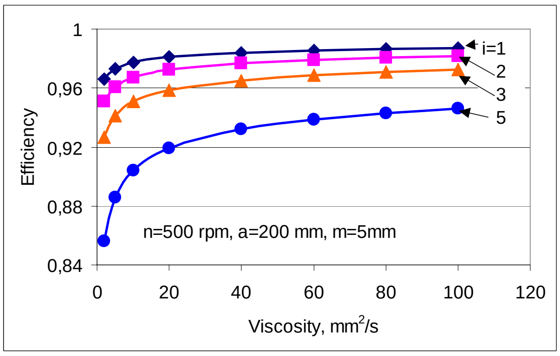

Fig. 10. Efficiency of a gear in function of lubricant viscosity at different gear ratio

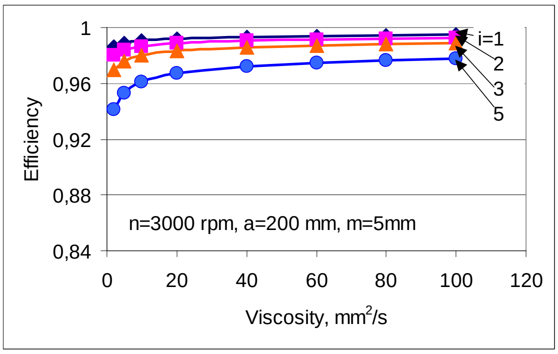

Fig. 11. Efficiency of a gear in function of lubricant viscosity at different gear ratio

The influence of rotation speed on the connection between the coefficient of friction and the lubricant viscosity is presented more suggestively in Figures 9-10.

Up-to date calculation methods of gear friction take into consideration not only viscosity, entraining velocity and relative radius of curvature at pitch point but also the effect of contact pressure between the mating teeth and the surface finish of the contacting surfaces [3]. Using this method for calculation of coefficient of friction it can be determined, that the viscosity of lubricant similarly influences the coefficient of friction between the teeth and the efficiency of gears as at the earlier used method [2] (Compare the diagrams of Figures 8-11 and 14-17). It can be stated from this comparison that the load has little effect on the calculated coefficient of friction values (which is in accordance with the EHL film thickness), and the gear ratio influences not so much the coefficient of friction in the latter calculation method than at the earlier calculation. [3]. This fact shows that entraining velocity has also smaller effect.

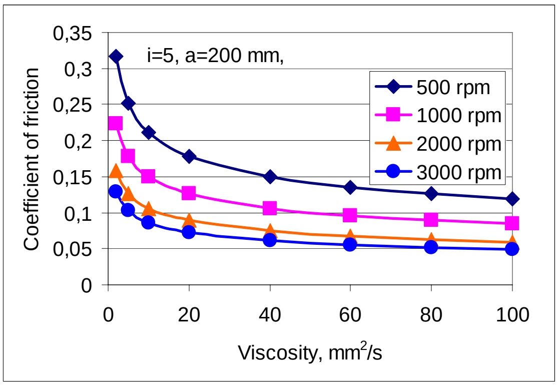

Fig. 12. Coefficient of friction of contacting teeth in function of lubricant viscosity at different rotation speed

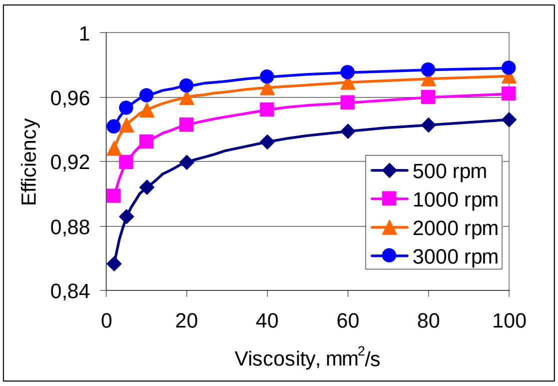

Fig. 13. Efficiency of a gear in function of lubricant viscosity at different rotation speed

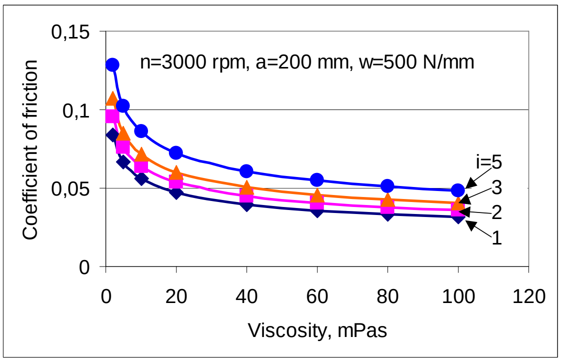

Fig. 14. Coefficient of friction of contacting teeth in function of lubricant viscosity at different gear ratio

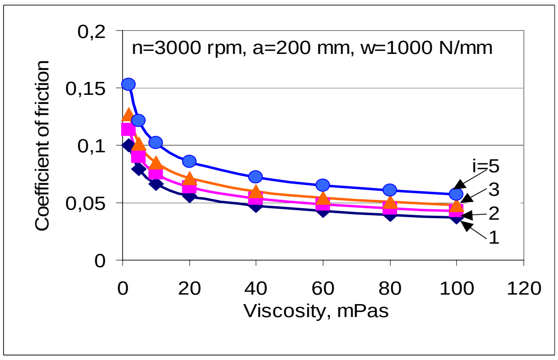

Fig. 15. Coefficient of friction of contacting teeth in function of lubricant viscosity at different gear ratio

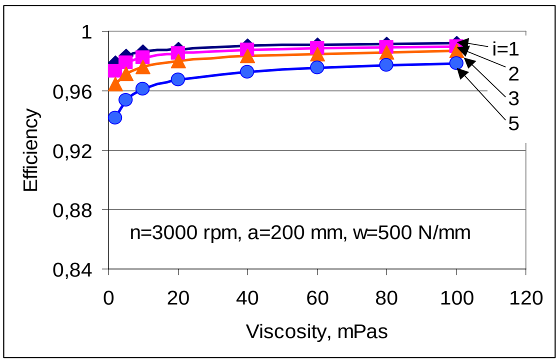

Fig. 16. Efficiency of a gear in function of lubricant viscosity at different gear ratio

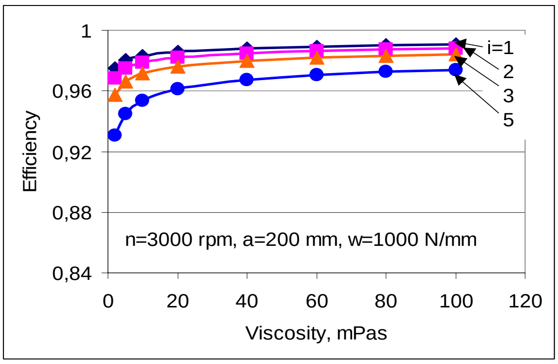

Fig. 17. Efficiency of a gear in function of lubricant viscosity at different gear ratio

The main factor determining the friction loss and the efficiency of gears is the viscosity of lubricants and partly the entraining velocity (the rotation speed) of gears.
At the same time, beside the friction loss between the teeth other power losses also emerged in gearboxes, which can be higher with increasing viscosity. The oil churning loss and also

the bearing friction loss belong to them. Therefore at the evaluations of the efficiency of whole gears, these contradictory effects have to be taken into consideration.

EFFECT OF OTHER PROPERTIES

Naturally, not only viscosities but also other properties of oils influence the efficiency of gears. Thoroughly performed investigations revealed that beside viscosity the molecule structure and additive package also influenced the friction loss of a gear [8]. Among the vegetable oils, synthetic oils and mineral oils a vegetable oil gave the lowest friction loss and the mineral oil led to the highest, in spite of the same viscosity grade.
Additives do not influence the friction behavior of the gears at low and moderate loads except EP additives, which increased it. At high load level the friction modifiers as ZDTP decreases the friction loss, the EP additives increased it, but hindered the galling of the loaded surfaces of teeth [5].
The viscosity of oil and the additives has even larger effect on the wear of gears as it was presented in earlier [5].

CONCLUSIONS

The above-presented results proved that

- The viscosity of the lubricants has enormous effect on the friction loss, efficiency, load carrying capacity and life span of gears.
- Higher viscosity lowers the coefficient of friction and increased the efficiency of gears.
- Lubricating oils of vegetable origin more lowered the friction loss in gear boxes than the synthetic oils, while the mineral oils were the less effective in it.
- At high rotation speed the lubricating oils of high viscosity leads to thinner EHL film thickness than the oils of lower viscosity.
- At lower temperature the synthetic oils results thinner oil film between the gear teeth than the mineral oils.

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