Discussion on Paper VI (iii) by W.J. Bartz: "Importance of the effective viscosity on the behavi1our of internal combustion engines"

although I have made no experimental check, that the patchy films form at the summits of the higher asperities and that they are raised above the metal. I also believe that in a general way they contribute to anti-wear behaviour and load-carrying.

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Dr. R.W. Wilson (Shell Research Limited, Thornton Research Centre, U.K.). Dr. Summers-Smith raised the crucial question of matching the lubricant properties and supply to the service requirements. Experiences with North Sea gas compressors highlighted in the paper by Cox and Gainey (Paper III(iv)) are relevant. Mineral oil lubricants were not satisfactory, the gas under compression, methane, dissolved in the lubricant, reducing the viscosity and in addition the lubricant dissolved in the compressed gas. Hence the lubricant is carried down into the oil producing formations and there it clogs the porous structure, reducing crude output. So increasing the lubricant supply was not an acceptable solution to the reservoir engineers. The very satisfactory answer to the problem was to use a synthetic lubricant with a low solubility for and in the gas which is, in addition water soluble. So if it is carried over into the producing formation it does no harm.

Reply by Dr. J.D. Summers-Smith (Imperial Chemical Industries Ltd., Billingham, U.K.). I am grateful to Dr. Wilson for pointing out the limitation of mineral hydrocarbon oils in lubricating high pressure methane compressors. Hopefully the empirical equations given in the paper will enable the operators to obtain an estimate of both the thinning effect of gas solution and the oil take up by the gas so that a decision can be made as to whether a mineral oil is acceptable or whether a different solution, such as the use of a a synthetic lubricant as indicated or possibly the use of a non-lubricated compressor, should be sought.

Mr. F.D. Gainey (Shell Research Limited, Thornton Research Centre, U.K.). The equation presented in the paper for the enhancement factor for the vapourisation of the lubricant, takes no account of the molecular weight of the gas being compressed - could the author comment on this please?

Reply by Dr. J.D. Summers-Smith (Imperial Chemical Industries Ltd., Billingham, U.K.). Although the equation for vapour pressure enhancement does not specifically take the molecular weight of the gas into account, this is in effect covered in the enthalpy values that are those of the gas being compressed. For mixtures of gases enhancement factors can be calculated for each component at its partial pressure and a composite enhancement factor derived as shown in the paper.

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Professor A. Cameron (Imperial College of Science and Technology, London, U.K.). Could Dr. Bartz give the details of the base oil used in his polymer loaded oil 3, viz viscosities and if possible pressure-viscosity data, polymer composition and percentage?

Reply by Professor Dr.-Ing. W.J. Bartz (Technische Akademie Esslingen, Federal Republic of Germany). The following test oils were used (see Figure 6 of Paper VI iii).

(a) Newtonian oil Commercially available SAE 20W-50 oil in the form of synthetic hydrocarbons without any polymers.

(b) Non-Newtonian oil Commercially available SAE 10W-50 oil (containing polystyrene butadiene copolymers as VI improvers in a mineral base oil) plus 6.1% of a cylinder oil in order to meet the SAE 20W-50 viscosity-temperature behaviour. Viscosity data including pressure-viscosity of the base oil components have not been determined.

Ir. J.J. van Leeuwen (Eindhoven University of Technology, Netherlands). The authors should be commended for his work on non-Newtonian fluids in stationary and dynamically loaded bearings. This paper shows the importance of shear rate effects on the journal orbit, through the shear rate dependent viscosity. Pressure and temperature effects on the viscosity are taken into account. I understand that the temperature is constant through the oil film. If so, the temperature effect can be as pronounced as the shear rate effect, if only a mistake of, say 10°C, is made in estimating the temperature. How can the designer know which temperature he should choose? Or does he have to take recourse to a temperature measurement in a test rig?

It would be a very welcome addition to the literature if the author can publish a subsequent paper on measurements in dynamically loaded contacts, which will give his calculations a solid base.

Reply by Professor Dr.-Ing. W.J. Bartz (Technische Akademie Esslingen, Federal Republic of Germany). It was the aim of the investigation to show the combined influences of temperature, pressure as well as shear rate on the behaviour of lubricating oils, especially non-Newtonian oils in dynamically loaded bearings. In the first step indeed operational conditions have been chosen which result in almost constant temperatures in the bearing. It is correct that for accurate calculations during the design process the temperature distribution obtained by frictional considerations and following another iteration process has to be known. Measurements in a test rig seem not to be sufficient.

Actual measurements in a driven dynamically loaded bearing in general confirmed the mathematical considerations and will be published in another paper.