Numerical Investigation of the Use of Machinery Low Viscosity Working Fluids as Lubricants in EHL Point Contacts

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Abstract

This work proposes a numerical investigation of the potential use of machinery working fluids as lubricants in contacts operating under an elastohydrodynamic (EHD) regime. These fluids are usually of very low viscosity and pressure-viscosity dependence. This is why, unmixed with oil they have been of little interest for the tribological community. Hence, their rheological properties are poorly known. In fact, these are restricted to a narrow range of conditions compared to the range of interest in EHL applications. This is why some measurements are carried out in order to determine both the viscosity and density of these uncommon lubricants. Besides, their viscosity being low, high mean entrainment speeds are required for a sufficiently thick lubricant film to build-up. This leads to important thermal dissipation within the contact. Thermal effects are included in the analysis in order to have an as accurate as possible estimation of film thicknesses and friction coefficients in these contacts. Results are discussed in the light of the peculiar properties of machinery low viscosity working fluids.

1. Introduction

Nowadays, many machinery operate with two fluids, each having a different function. The first one is the lubricant, while the second, usually known as the "working fluid", can have different functions depending on the type of machine. It could be a heat transfer fluid in heat pumps or refrigeration systems for example, or a combustion fluid such as fuel in combustion engines or cryogenic liquids for rocket propulsion engines. This work aims to investigate the potential use of working fluids as lubricants in elastohydrodynamic (EHD) contacts. The use of these fluids as lubricants offers two main advantages. The first is related to the economical issue of energy saving which is also more and more considered nowadays as an environmental issue. As a matter of fact, the energy dissipation in a lubricated contact due to friction forces increases with the viscosity of the lubricant. Working fluids in many cases have a low viscosity and also a low viscosity-pressure dependence, hence they are expected to lead to reduced frictional dissipation in lubricated contacts. Fox [1] shows that a 4% economy of fuel consumption in a Diesel engine could be reached by simply reducing the viscosity of the lubricant. Clearly, the contact conditions become more severe and the wear rates of the contacting surfaces increase. Nevertheless, a good functioning of the system can be achieved by applying a surface treatment to the contacting bodies in such a way to increase their resistance to wear. Another alternative to avoid surface degradation consists in adding some anti-wear additives to the lubricant. Hence, in order to complete the current work, an additional question is to be addressed regarding the potential reduction in the bearing's life because of the reduced lubricant film thicknesses that are likely to be encountered. In fact, one has to wonder about the relative importance of reducing energy dissipation compared to the reduction in the component's life. This question reaches beyond the scope of this work and shall not be addressed here. The second advantage is related to the reduction in size, weight and complexity of machines operating with two fluids. In fact, for a good functioning of such systems, it is highly preferable that the two fluids do not mix. This is why these machines are generally designed with two separate circulation systems, one for each fluid. Not only does this make their design and maintenance more complicated, but it also leads to an increase in their size and weight. Thus, it would be interesting to have one single fluid fulfill the two different functions inside the system. This would allow an easier design and maintenance of the machine that would include only one circulation system. Knowing that a lubricant could almost never replace the working fluid, sometimes the only solution would be to use the working fluid as a lubricant.

In this paper, the label of Low Viscosity Working Fluids (LVWF) is attained to fluids with viscosities of the order of 10^{-4} Pa.s under ambient pressure. Compared to water that has a viscosity of 10^{-3} Pa.s, or air that has a viscosity of 10^{-5} Pa.s, working fluids have a range of viscosities varying between those of air and water. Such fluids have been of little interest for the tribological community. Hence, their rheological properties are poorly known. In fact, these are restricted to a narrow range of pressure conditions compared to the range of interest in EHD applications. In this work, some measurements are carried out in order to determine both the viscosity and density of these uncommon lubricants. To find out whether it is possible or not to use LVWF as lubricants, this paper offers a numerical investigation of EHD point contacts lubricated with such fluids. The viscosity of LVWF being low, high mean entrainment speeds are likely to be required for a sufficiently thick lubricant film to build-up. This would lead to important thermal dissipation within the contact. Thermal effects are included in the analysis in order to have an as accurate as possible prediction of the contact's behavior under these conditions. Two families of working fluids are of interest in this work: refrigerants and combustion fluids. Film thicknesses and friction coefficients are analyzed for EHD point contacts lubricated with typical fluids of the two types in order to determine the range of operating conditions under which these contacts can operate "safely".

2. What about inertial effects?

Generally, in liquid flows, when low viscosity fluids are considered or also when high velocities take place, inertial effects might become important and cannot be considered as negligible anymore. The appearance of these effects is characterized by high values of the reduced Reynolds number associated to the flow. The latter expresses the ratio of inertia forces over viscous forces and is written in general under the following form:

$$\mathrm{Re}^* = \frac{Inertia\ \ forces}{Viscous\ \ forces} = \frac{\rho\, V_c\, L_c^2}{\mu\, L} \tag{1}$$

Where:

ρ : Fluid's density	V_c : Characteristic velocity of the flow
μ : Fluid's viscosity	L : Contact characteristic length ($L = 2a$)
a : Hertzian contact radius	L_c : Characteristic length of the flow

When the value of Reynolds' number is below a given critical value ($\mathrm{Re}^* < \mathrm{Re}_c$), the corresponding flow is considered laminar and inertial effects negligible. However, when the Reynolds' number exceeds the critical value, inertial effects become important and these have a significant influence on the flow characteristics. If it is further increased, turbulent regime may appear beyond a certain limit.

Inertial effects are rarely encountered in tribological applications because of the "relatively" small characteristic lengths (film thicknesses) of the corresponding flows and the generally high viscosity of classical lubricants. However, in journal bearings lubricated by a low viscosity lubricant or also in gas bearings working under high speed operating conditions inertial effects might become important and a turbulent regime might even arise [2]. In such cases, the simplifying assumptions of the classical lubrication theory are not valid anymore (pressure is not constant over the film thickness, the normal velocity becomes of the same order of magnitude as the rest of the velocity components …). Hence, the classical Reynolds equation is not representative of the lubricant flow within the contact anymore. The latter has to be replaced by the generalized Reynolds equation with inertia effects or even the Navier-Stokes equations if a turbulent regime is encountered. Since very low viscosity fluids are considered in this work, one might wonder if inertia effects have to be considered or not, especially that high speed operating conditions are probably required in order to have a sufficient film thickness that prevents direct contact between surface asperities. In order to clarify this point, an order of magnitude analysis is carried out for a typical EHD point contact lubricated with a working fluid. The characteristic length of the flow which corresponds to the film thickness is considered to be 100 nm, the characteristic velocity which in this case corresponds to the mean entrainment speed is taken as 10 m/s. Typical values for the zero-pressure density and viscosity of working fluids are 1000 Kg/m^3 and 10^{-4} Pa.s. The contact's characteristic length is considered to be 0.1mm. The value of the critical Reynolds' number for a flow between two parallel plates $\mathrm{Re}_c \approx 2000-2500$. This is the closest configuration to a typical EHL circular contact. Thus the reduced Reynolds' number that corresponds to an EHD point contact lubricated with a typical working fluid is estimated as follows:

$$\mathrm{Re}^* = \frac{\rho\, V_c\, L_c^2}{\mu L} = \frac{1000 \times 10 \times \left(10^{-7}\right)^2}{10^{-4} \times 10^{-4}} = 0.01 \ll \mathrm{Re}_c \approx 2000-2500$$

Note that the "safer" zero-pressure value of the viscosity has been used in this case. The latter is far smaller than the actual value that is encountered within the contact (especially in the central area). This way, the analysis is valid for both the inlet and central regions of the contact. It is clear that the value of the typical Reynolds' number is far smaller than the critical value. Hence, the flow within the corresponding conjunctions can be considered laminar and inertial effects negligible. The classical lubrication theory remains valid in this case.

3. Numerical model

The numerical model employed in this work is described in details in [3]-[6]. In this section, only the main features are recalled. The model is based on a finite element fully-coupled resolution of the EHD equations: Reynolds, linear elasticity and load balance equations. The latter are solved simultaneously providing robust and fast converging solutions. Since inertial effects are negligible, the classical Reynolds equation is employed. Special formulations are used in order to stabilize the solution of Reynolds equation under high loads. All considered fluids are assumed to have a Newtonian behaviour under the range of operating conditions considered in this work. Since high mean entrainment speeds are expected to be needed for a sufficient film thickness to build-up, thermal effects are likely to be significant. The temperature distribution in the contact is obtained by solving the 3D energy equation in the lubricant film and solid bodies. An iterative procedure is applied between the respective solutions of the EHD and thermal problems as described in [3]-[6] until a converged solution is obtained.

4. Is it possible to use working fluids as lubricants?

As mentioned previously, the use of working fluids as lubricants presents two important aspects in engineering applications. First, due to their low viscosity, they lead to reduced energy dissipation by friction in lubricated contacts. Second, from a practical point of view, it is much easier to design and maintain machines operating with only a single fluid serving as the lubricant and working fluid. However, because of the very low viscosity of these fluids, thin lubricant film layers are expected to be built up in the contact area. The aim of this work is to investigate whether it is possible or not to lubricate "correctly" with such fluids and under what range of operating conditions.

Two categories of working fluids are of interest for this work: refrigerants and combustion fluids. Two typical fluids from each category are selected to run numerical tests in EHD point contacts. It is considered that for film thicknesses greater than 5 nm, the theory of continuum mechanics used in this work is valid. In fact, Granick [7] noticed that for *n*-dodecane, which has a similar molecular size and weight compared to the working fluids considered in this work, the bulk properties of the lubricant flow were the same as those predicted by a continuum approach down to 4 nm. This is confirmed by the observations of Georges et al. [8] who showed that this was true for *n*-dodecane and *n*-hexadecane down to 2.5 and 4.4 nm respectively. In EHL applications, the same observations were made by Guangteng and Spikes [9][10] and also Matsuoka and Kato [11][12] who showed that the continuum approach is valid down to approximately 10 molecular layers. When thinner films are considered, the analysis becomes more complex since it involves additional parameters that are neglected by the continuum approach such as surface tensions, solvation forces, physical and chemical reactions, etc. In this case, alternative methods are introduced to study such contacts e.g. Tichy [13][14]. Hence, the reader should be aware that the less than 5 nm film thicknesses that are shown next have no physical relevance. Moreover, even the best finished surfaces have a surface roughness of a few nanometers and therefore a direct contact between asperities is likely to occur. Thus, a simple EHL analysis would not stand anymore and a "mixed lubrication" analysis would have to be considered when film thicknesses fall below the 5 nm limit.

In order to answer the question that is addressed in the title of this section, it is considered that a minimum surface separation of 10 nm is sufficient to ensure a reasonably "safe" functioning of a high-quality ball bearing.

4.1 Refrigerants

Two refrigerants are selected for testing in this section; these shall be referred as "Fluid A" and "Fluid B". For "Fluid A", the ambient temperature is considered to be T_0=10°C at which the zero-pressure viscosity is μ_0=0.465mPa.s and the equivalent pressure-viscosity coefficient (defined in [15] as the "reciprocal asymptotic isoviscous pressure coefficient") α^*=4.36GPa^{-1} whereas for "Fluid B", T_0=-0.5°C, μ_0=0.268mPa.s and α^*=2.55GPa^{-1}. Note that both refrigerants have low viscosities and pressure-viscosity dependencies. Also note that the ambient temperature for "Fluid B" is required to be at least as low as -0.5°C in order to maintain a liquid phase and prevent its ambient pressure evaporation. Film thickness numerical tests are run for steel-glass point contacts under pure rolling regime

Remark: In refrigeration systems, no glass-steel contacts can be found. The choice of this solid material combination was solely based on an attempt of experimental verification of the numerical results proposed in this paper. As a matter of fact, the authors carried out some experiments to validate their results, but handling refrigerants in a confined apparatus turned out to be even more difficult than expected and only a few data points were possible to obtain [3]. These allowed validation of the numerical results provided in this work; however the authors believe that a few data points are not enough to be presented in this paper.

Figures 1 and 2 show the central and minimum film thickness curves respectively as a function of the mean entrainment speed u_m for both fluids under pure rolling conditions and for different values of the normal load. The range of operating conditions covered in these figures is $M \in [20-17000]$ & $L \in [0.74-1.85]$ for "Fluid A" and $M \in [7-5600]$ & $L \in [0.2-0.6]$ for "Fluid B" (M and L are the Moes [16] dimensionless load and material properties parameters respectively).

Remark: Note that in practice, the 250N load case considered here corresponds to a numerical test and can never be realized on an experimental apparatus because glass would not withstand such a load.

Figures 1 and 2 clearly show that, in general, high speed operating conditions (up to 10 or 20 m/s) are required for a "safe" film separation especially for "Fluid B" which has a lower viscosity than "Fluid A". Moreover, as the load is increased, the minimum required entrainment speed increases. Also note that the film thicknesses generated by "Fluid A" are generally higher than those by "Fluid B". This is to be expected since the former has a higher viscosity than the latter.

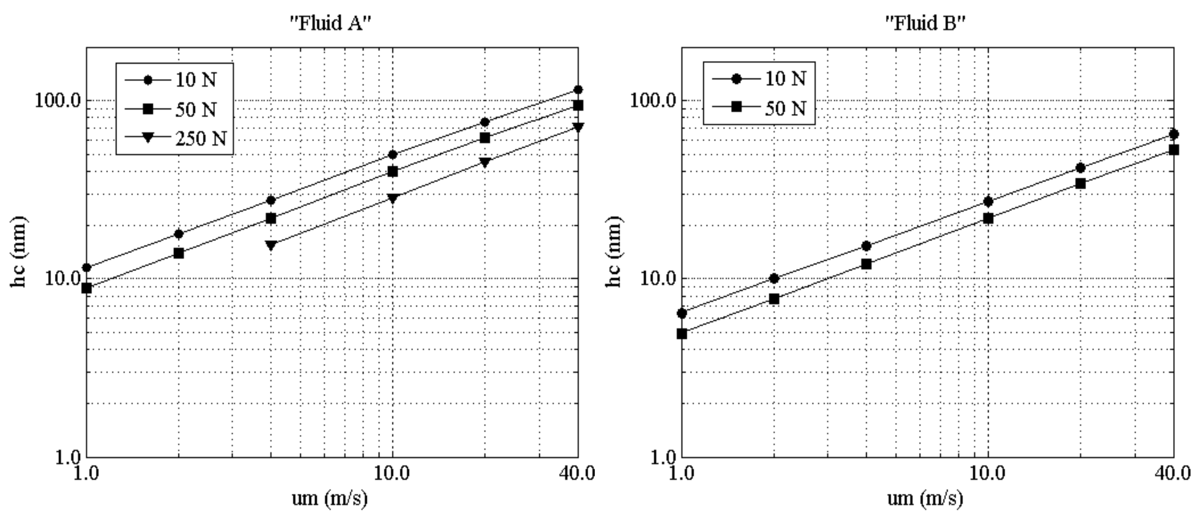

Figure 1: Numerical results of central film thickness curves for a steel-glass contact on a log-log scale as a function of the mean entrainment speed for "Fluid A" (left, T_0=10ºC) and "Fluid B" (right, T_0=-0.5ºC) (F=10 N / p_h=0.36 GPa, F=50 N / p_h=0.61 GPa, F=250 N / p_h=1.05 GPa)

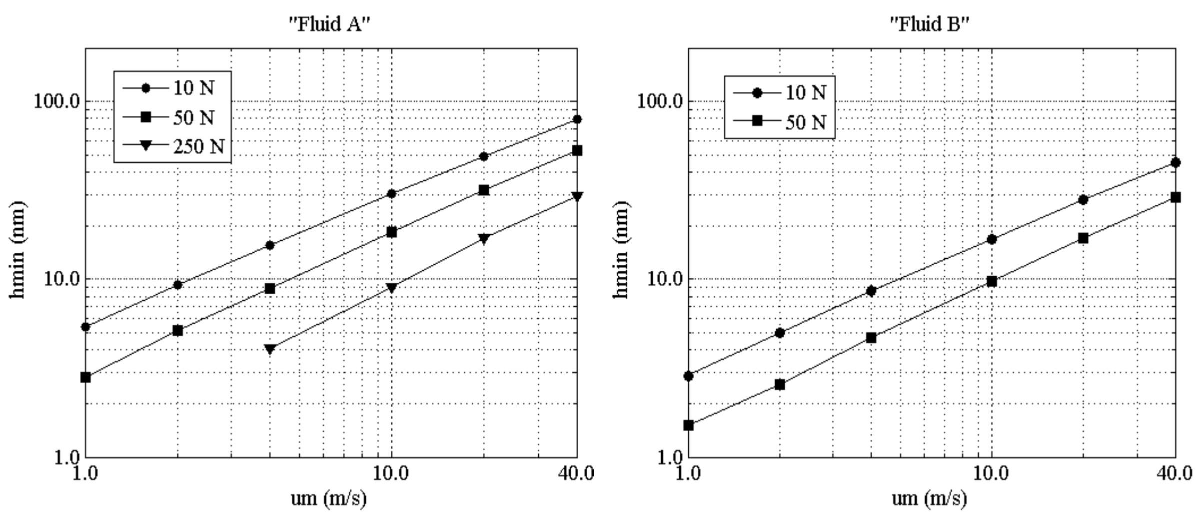

Figure 2: Numerical results of minimum film thickness curves for a steel-glass contact on a log-log scale as a function of the mean entrainment speed for "Fluid A" (left, T_0=10ºC) and "Fluid B" (right, T_0=-0.5ºC) (F=10 N / p_h=0.36 GPa, F=50 N / p_h=0.61 GPa, F=250 N / p_h=1.05 GPa)

Because of the small viscosity value, weak viscosity-pressure dependence and fairly high loads that are considered in these tests, one might expect that the *elastic-isoviscous* asymptote that was introduced in [16] for cases of large values of M and L=0 would be appropriate to estimate the film thickness generated by these fluids in EHD contacts. This formula was developed for central film thickness and has the following mathematical expression:

$$h_c = 1.96\, R \sqrt{2U}\, M^{-1/9} \tag{2}$$

Where U is the dimensionless Hamrock & Dowson [17] speed parameter. Figure 3 shows the central film thickness curves shown previously along with the *elastic-isoviscous* asymptote provided in equation (2) for both "Fluid A" and "Fluid B".

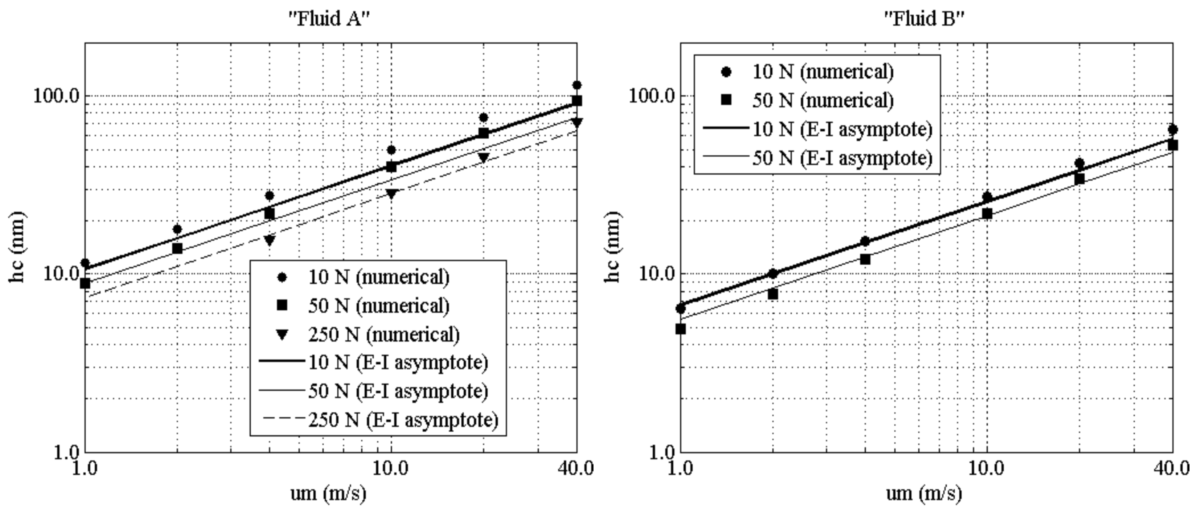

Figure 3: Comparison of central film thickness curves for steel-glass contacts under pure-rolling regime predicted by numerical resolution and the *elastic-isoviscous* asymptote for "Fluid A" (left, T_0=10ºC) and "Fluid B" (right, T_0=-0.5ºC) (F=10 N / p_h=0.36 GPa, F=50 N / p_h=0.61 GPa, F=250 N / p_h=1.05 GPa)

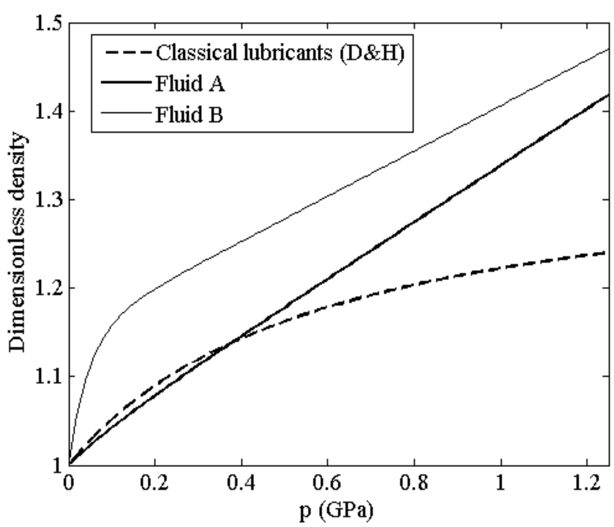

Figure 4: Compressibility of "Fluid A" and "Fluid B" compared to classical lubricants

The *elastic-isoviscous* asymptote fits reasonably well the numerical results especially for “Fluid B”. However, for “Fluid A”, the fit is less accurate. This is because the latter has a higher pressure-viscosity coefficient and is thus further from the *isoviscous* extreme. Moreover, Figure 4 shows the unusual density-pressure dependence of these fluids with respect to classical lubricants which often obey reasonably well the Dowson & Higginson [18] relationship. This unusual density-pressure dependence leads to unusual film thicknesses that cannot be predicted by classical formulae. In fact, if for the same operating conditions a fictive less viscosity-pressure dependent fluid (“Fluid C”) is considered with α^*=0.5GPa^{-1}, μ_0=0.5mPa.s and a Dowson & Higginson-like compressibility, the fit between numerical results and the *elastic-isoviscous* asymptote would be much more accurate as can be seen in Figure 5. Hence some specific formulae have to be developed to predict central and minimum film thickness in EHD contacts lubricated by these refrigerants. Because of the different compressibility of “Fluid A” and “Fluid B”, specific formulae have to be introduced for each fluid.

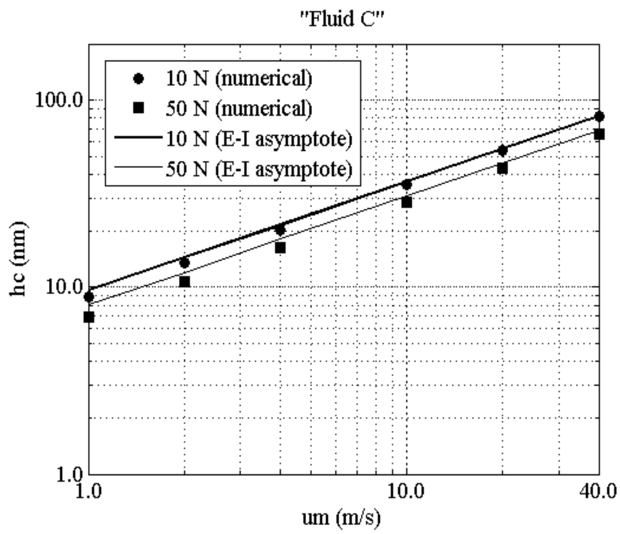

Figure 5: Comparison of central film thickness curves for steel-glass contacts under pure-rolling regime predicted by TEHL numerical resolution and the *elastic-isoviscous* asymptote for “Fluid C” (F=10 N / p_h=0.36 GPa, F=50 N / p_h=0.61 GPa)

The numerical results for “Fluid A” and “Fluid B” were fitted to the following mathematical expressions for central and minimum film thickness:

$$\text{"Fluid A"}: \begin{cases} h_c = 2.9401\ R\sqrt{2U}\ M^{-0.1541}L^{0.0526} \\ h_{\min} = 3.7607\ R\sqrt{2U}\ M^{-0.3155}L^{0.0013} \end{cases}$$

$$\text{"Fluid B"}: \begin{cases} h_c = 2.6815\ R\sqrt{2U}\ M^{-0.1405}L^{0.1087} \\ h_{\min} = 4.1150\ R\sqrt{2U}\ M^{-0.3255}L^{0.0568} \end{cases} \quad (3)$$

The fit between the previous analytical formulae and full numerical results for both fluids is shown in Figures 6 and 7:

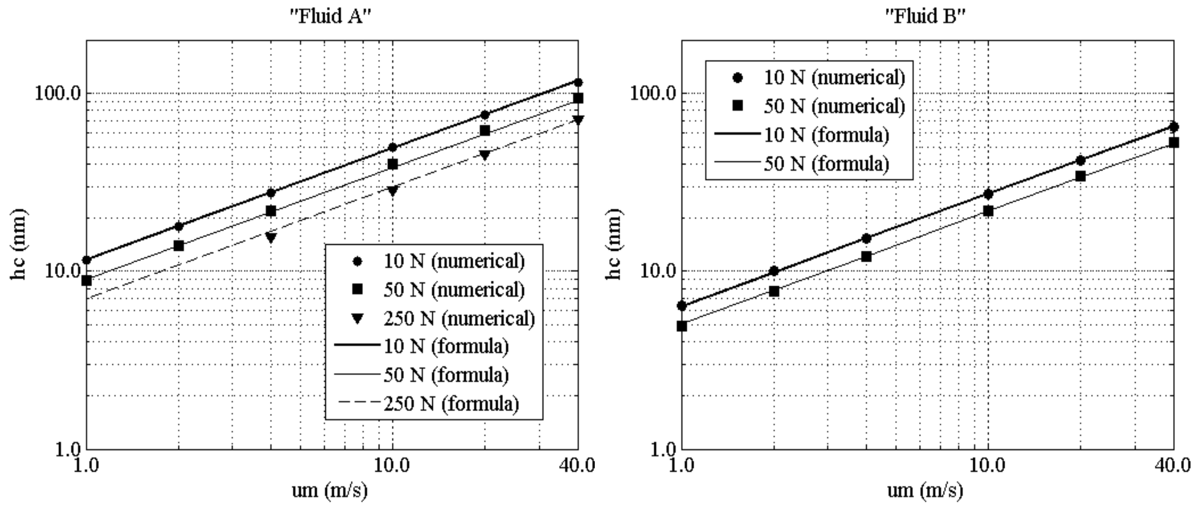

Figure 6: Comparison of central film thickness curves for steel-glass contacts under pure-rolling regime predicted by TEHL numerical resolution and analytical formulae (3) for "Fluid A" (left, T_0=10ºC) and "Fluid B" (right, T_0=-0.5ºC) (F=10 N / p_h=0.36 GPa, F=50 N / p_h=0.61 GPa, F=250 N / p_h=1.05 GPa)

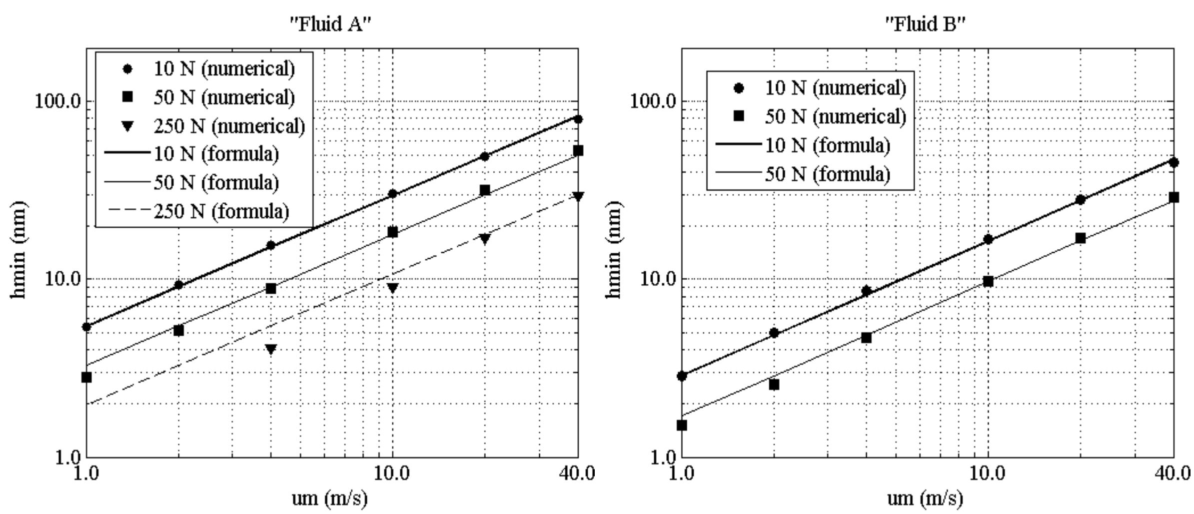

Figure 7: Comparison of minimum film thickness curves for steel-glass contacts under pure-rolling regime predicted by TEHL numerical resolution and analytical formulae (3) for "Fluid A" (left, T_0=10ºC) and "Fluid B" (right, T_0=-0.5ºC) (F=10 N / p_h=0.36 GPa, F=50 N / p_h=0.61 GPa, F=250 N / p_h=1.05 GPa)

A fairly good agreement is obtained over the considered range of operating conditions between numerical results and the specific analytical formulae for the two considered refrigerants. Finally, as was pointed out earlier, an attractive feature in the use of LVWF as lubricants is the economical aspect. In fact, due to the low viscosity of these fluids, frictional losses in the contact area are much smaller than those generated by classical lubricants. In order to quantify this, traction coefficients corresponding to rolling-sliding steel-steel contacts are shown in Figure 8 as a function of the slide-to-roll ratio (SRR).

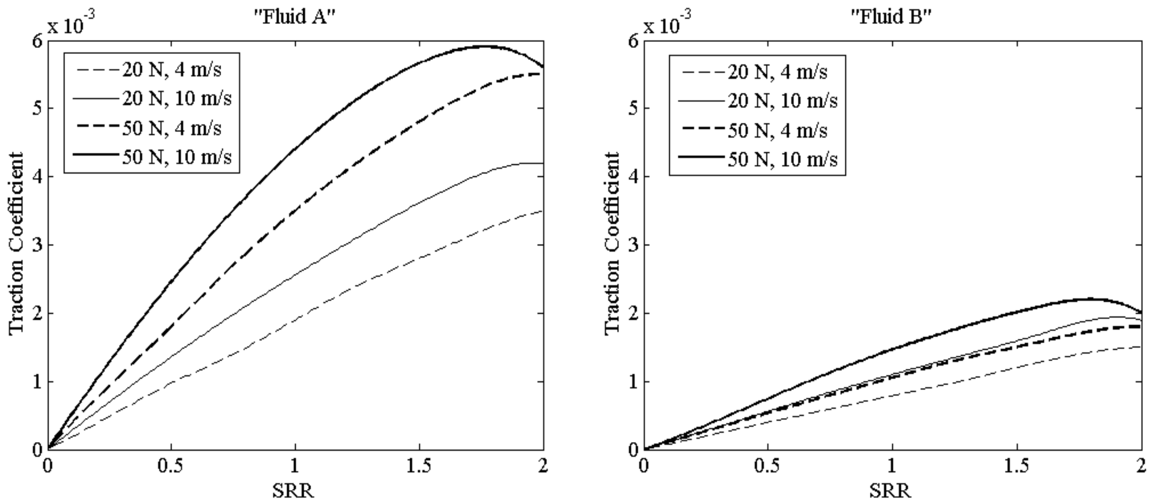

Figure 8: Numerical results of traction curves for “Fluid A” (left, T_0=10ºC) and “Fluid B” (right, T_0=-0.5ºC) for steel-steel contacts working under rolling-sliding conditions (F=20 N / p_h=0.68 GPa, F=50 N / p_h=0.93 GPa)

Note the relatively small values of the friction coefficients generated by the use of both fluids even under pure sliding conditions (SRR=2). The latter never exceed 6 ‰. This value is in most cases 10 times smaller than what would be obtained with a classical lubricant. Since energy dissipation by friction within a mechanical system is proportional to the friction coefficient, then the former is expected to be reduced by a factor of 10 when a refrigerant is used as both the working fluid and the lubricant.

Finally, it is important to note that, due to the low viscosity and viscosity-pressure dependence of the considered refrigerants; thermal effects are barely noticeable on film thickness whereas they are more important on friction. As a matter of fact, film thickness curves are straight lines on the log-log scale which is characteristic of an isothermal regime. No film thickness reduction can be noticed even at very high mean entrainment speed. On the other hand, friction is more affected by thermal effects, especially at high sliding velocities where a reduction in friction coefficients takes place as a function of SRR. This implies that temperature rise is negligible in the inlet area of the contact (where film thickness is known to build-up) whereas it is more important in the central area of the contact which dominates friction behaviour (viscosity values become significant due to the high pressures that are encountered in this area).

4.2 Combustion fluids

As for refrigerants, it is often highly desirable that combustion fluids and lubricants do not mix. For instance, in an internal combustion engine, the fraction of lubricating oil that remains in the combustion chamber during combustion leads to undesired polluting emissions [19]. In this section, the use of combustion fluids as lubricants is investigated. For this purpose two typical combustion fluids are selected, a kerosene referred as “Fluid D” and a Diesel fuel referred as “Fluid E”. These fluids are expected to operate under high temperatures, this is why for “Fluid D” the ambient temperature is considered to be T_0=100ºC at which the zero-pressure viscosity is μ_0=0.642mPa.s and the equivalent pressure-viscosity coefficient α^*=5.55GPa^{-1} whereas for

"Fluid E", T_0=150^{o}C, μ_0=0.707mPa.s and α^*=5.82GPa^{-1}. Again note that both fluids have low viscosities and pressure-viscosity dependencies under the operating conditions considered here. Film thickness tests are run for steel-steel point contacts under pure rolling regime. Figures 9 and 10 show the central and minimum film thickness curves respectively as a function of the mean entrainment speed u_m for both fluids under pure rolling conditions and different load values. The range of operating conditions covered in these figures is $M \in [53-12500]$ & $L \in [1-2.8]$ for "Fluid D" and $M \in [50-11600]$ & $L \in [1.1-3.0]$ for "Fluid E".

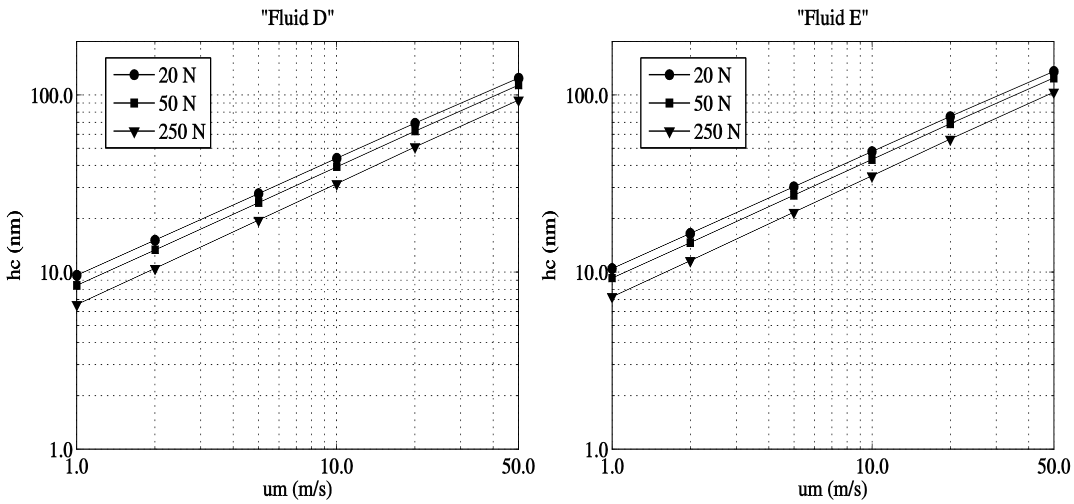

Figure 9: Numerical results of central film thickness curves for a steel-steel contact on a log-log scale as a function of the mean entrainment speed for "Fluid D" (left, T_0=100ºC) and "Fluid E" (right, T_0=150ºC) (F=20 N / p_h=0.68 GPa, F=50 N / p_h=0.93 GPa, F=250 N / p_h=1.59 GPa)

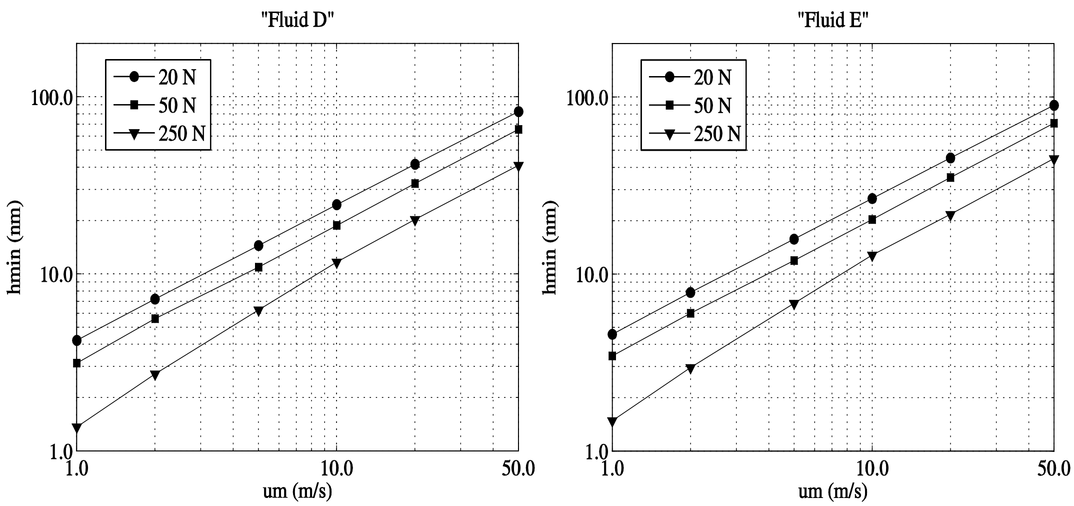

Figure 10: Numerical results of minimum film thickness curves for a steel-steel contact on a log-log scale as a function of the mean entrainment speed for "Fluid D" (left, T_0=100ºC) and "Fluid E" (right, T_0=150ºC) (F=20 N / p_h=0.68 GPa, F=50 N / p_h=0.93 GPa, F=250 N / p_h=1.59 GPa)

As for refrigerants, it is clear that for both combustion fluids considered here, relatively high mean entrainment speeds (>10m/s) are required for a reasonably "safe" fluid film separation, especially under high loads. “Fluid D” and “Fluid E” have a relatively close viscosity and pressure-viscosity dependence to those of the two refrigerants considered earlier. Besides, their compressibility also deviates significantly from that of classical lubricants as shown in Figure 11 and is also quite different from that of “Fluid A” and “Fluid B”. Hence, specific film thickness formulae are also required for the combustion fluids considered in this section.

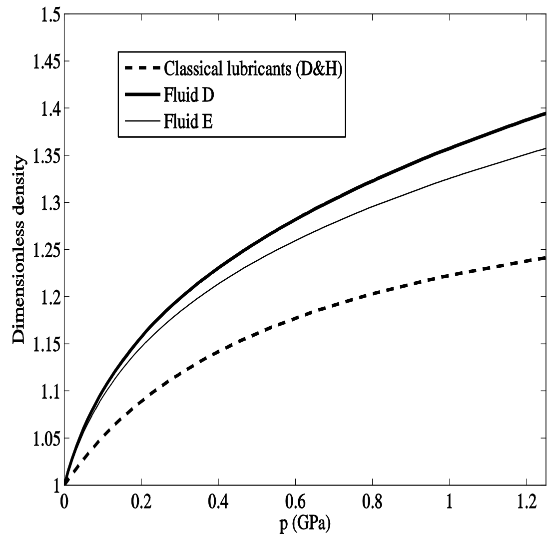

Figure 11: Compressibility of “Fluid D” and “Fluid E” compared to classical lubricants

The following film thickness formulae were derived for the central and minimum film thicknesses for both “Fluid D” and “Fluid E” as a function of the Moes dimensionless parameters:

$$\text{"Fluid D" : } \begin{cases} h_c = 2.6528\ R\sqrt{2U}\ M^{-0.1261} L^{0.2807} \\ h_{\min} = 3.8251\ R\sqrt{2U}\ M^{-0.2988} L^{0.1972} \end{cases}$$
$$\text{"Fluid E" : } \begin{cases} h_c = 2.5828\, R\sqrt{2U}\ M^{-0.1209} L^{0.2944} \\ h_{\min} = 3.8276\ R\sqrt{2U}\ M^{-0.2986} L^{0.1939} \end{cases} \tag{4}$$

Note that, although the viscosity and pressure-viscosity dependence of the refrigerants and combustion fluids considered here are relatively close, their film thickness formulae are completely different because of their relatively different density-pressure dependence. As stated earlier, the unconventional compressibility of LVWF has an important effect on film formation in EHL contacts lubricated with these fluids.

As a matter of fact, if “Fluid D” had been defined with a Dowson & Higginson compressibility while keeping all other parameters unchanged, the film thickness formulae would become:

$$\text{"Fluid D" with D\&H compressibility}: \begin{cases} h_c = 2.5476\, R\sqrt{2U}\, M^{-0.1134} L^{0.3016} \\ h_{\min} = 3.9135 R\sqrt{2U}\, M^{-0.3017} L^{0.1728} \end{cases} \tag{5}$$

The previous film thickness formulae overestimate h_c by an average of 5% and underestimate h_{min} by an average of less than 1% over the considered range of operating conditions. This difference is only due to compressibility since all other parameters were left unchanged. This highlights the importance of these fluids' compressibility on film formation, mostly on central film thickness. However, qualitatively speaking, the pressure and film thickness distributions in contacts lubricated with these unconventional lubricants are quite similar to those observed in oil lubricated contacts as can be seen in Figure 12. The latter shows the dimensionless pressure and film thickness distribution along the central line of a steel-steel contact lubricated with "Fluid D" under an external applied load F=250N and a mean entrainment speed u_m=1m/s (left) and u_m=50m/s (right). The only observation worth noting is that the pressure spike is relatively small because of the low viscosity and pressure-viscosity dependence of "Fluid D".

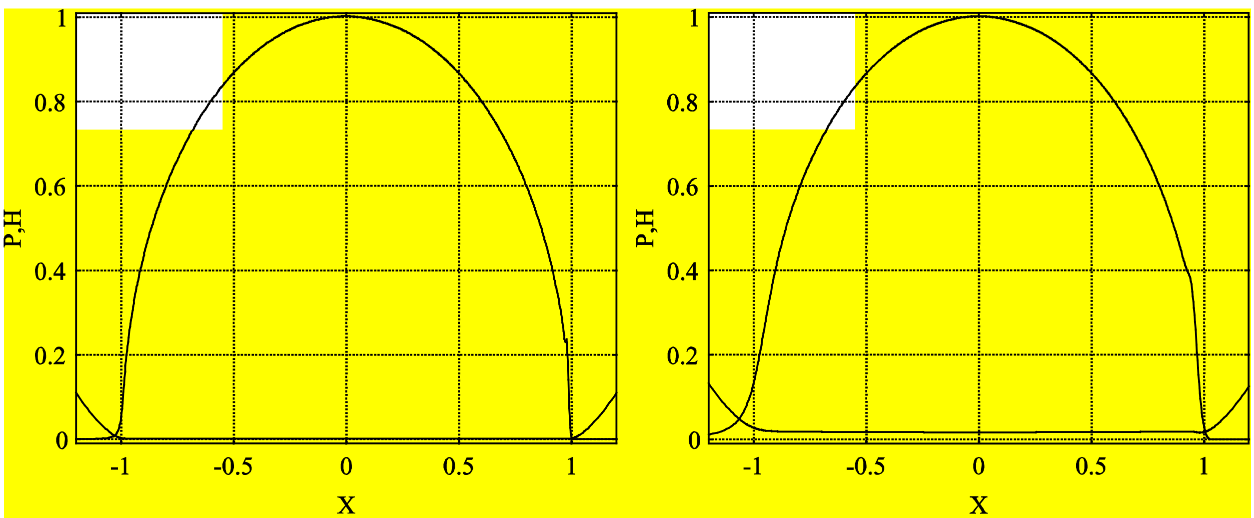

Figure 12: Numerical results of dimensionless pressure and film thickness distributions along the central line of the contact for "Fluid D"(left: F=250N, u_m=1m/s; right: F=250N, u_m=50m/s)

Note that, contrarily to "Fluid A" and "Fluid B", the film thickness formulae (4) of "Fluid D" and "Fluid E" are practically identical. This is to be expected since their density-pressure dependencies are relatively close for the considered operating temperatures. The fit between numerical results and film thickness formulae derived in equation (4) is shown in Figures 13 and 14.

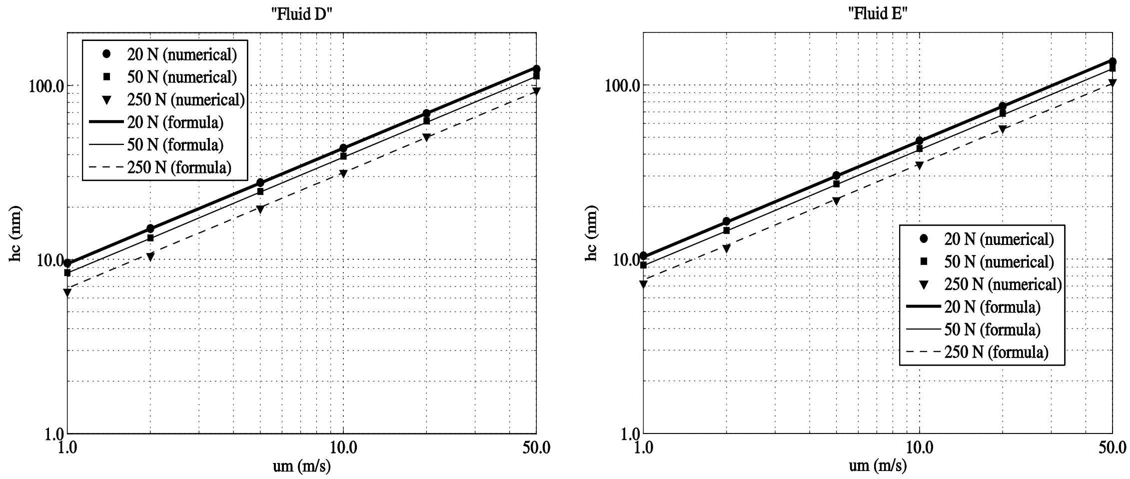

Figure 13: Comparison of central film thickness curves for steel-steel contacts under pure-rolling regime predicted by TEHL numerical resolution and analytical formulae (4) for "Fluid D" (left, T_0=100ºC) and "Fluid E" (right, T_0=150ºC) (F=20 N / p_h=0.68 GPa, F=50 N / p_h=0.93 GPa, F=250 N / p_h=1.59 GPa)

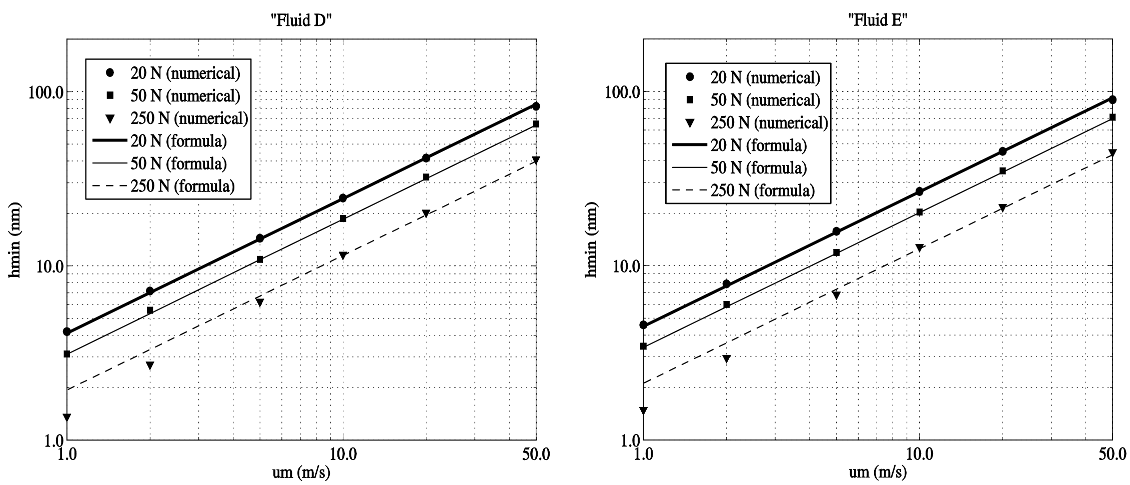

Figure 14: Comparison of minimum film thickness curves for steel-steel contacts under pure-rolling regime predicted by TEHL numerical resolution and analytical formulae (4) for "Fluid D" (left, T_0=100ºC) and "Fluid E" (right, T_0=150ºC) (F=20 N / p_h=0.68 GPa, F=50 N / p_h=0.93 GPa, F=250 N / p_h=1.59 GPa)

Finally, the traction curves in Figure 15 show that also for combustion fluids, friction coefficients never exceed the value of 8‰ for both fluids considered, even under pure sliding conditions. Hence, as for refrigerants, a significantly reduced frictional dissipation is expected when combustion fluids are used as lubricants.

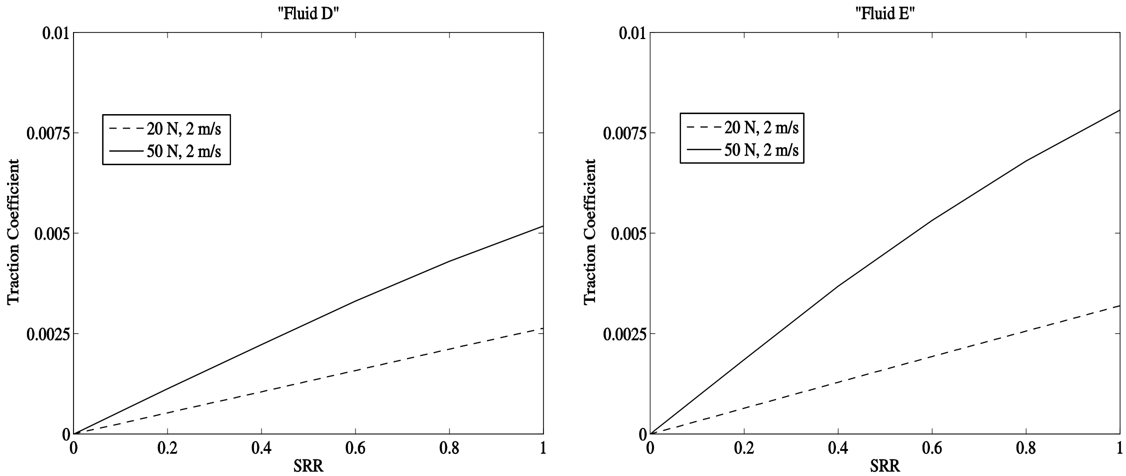

Figure 15: Numerical results of traction curves for "Fluid D" (left, T_0=100ºC) and "Fluid E" (right, T_0=150ºC) for steel-steel contacts working under rolling-sliding conditions (F=20 N / p_h=0.68 GPa, F=50 N / p_h=0.93 GPa)

Note that, as for refrigerants, thermal effects are negligible on film thickness and become more important on the friction behaviour of EHD contacts lubricated with combustion fluids.

5. Conclusion

This work presents a thorough investigation of the potential use of low viscosity machinery working fluids in EHD point contacts. Two families of working fluids were considered: refrigerants and combustion fluids. Two typical fluids of each family were considered and a series of test cases were carried out to compute central and minimum film thicknesses as well as friction coefficients generated by these fluids in EHD contacts under a wide range of operating conditions. As expected high speed regimes (u_m >10m/s) were found to be favourable for a "safe" operation of systems lubricated with such fluids. Specific film thickness formulae were developed for each fluid over the considered range of operating conditions. The importance of these formulae resides in providing an emphasis on film thickness tendencies in contacts lubricated with LVWF rather than providing a recipe for film thickness calculation in these contacts. The relatively high and unusual compressibility of the considered fluids was found to have a significant impact on film thicknesses compared to usual EHL applications where hydrodynamic effects, generated mostly by viscosity, are dominant. A traction analysis showed the important reduction of frictional dissipation in these contacts. The latter would certainly lead to reduced energy consumption in the corresponding mechanical system.

Coming back to the initial question posted in the present paper, whether or not these Low Viscosity Working Fluids can lubricate a contact in EHD regime and at the same time reduce energy losses. From the results of this study it follows that LVWF fluids can be used to lubricate highly loaded nonconformal contacts, if their entrainment speed is of the order of 10 m/s or higher, and the surface roughness is of the order of 10 nm or better.

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Nomenclature

E'	: Reduced elastic modulus (Pa)
F	: External contact load (N)
h_c	: Cental film thickness (m)
h_{min}	: Minimum film thickness (m)
L	: Moes dimensionless material properties parameter
L_c	: Flow characteristic length (m)
M	: Moes dimensionless load parameter
p	: Pressure (Pa)
p_h	: Hertzian contact pressure (Pa)
R	: Equivalent contacting elements' radius (m)
Re	: Reynolds number
Re_c	: Critical Reynolds number
SRR	: Slide-to-Roll Ratio
T_0	: Ambient temperature (^{o}C)
U	: Hamrock & Dowson dimensionless speed parameter
u_m	: Mean entrainment speed (m/s)
V	: Flow velocity (m/s)
α^*	: Equivalent pressure-viscosity coefficient (Pa^{-1})
μ	: Viscosity (Pa.s)
μ_0	: Ambient pressure viscosity (Pa.s)
ρ	: Density (Kg/m^3)
ρ_R	: Reference density under ambient conditions (Kg/m^3)
$\bar{\rho}$	: Dimenionless density

Dimensionless parameters

$$W = \frac{F}{E'R^2} \qquad G = \alpha^* E' \qquad U = \frac{\mu_0 u_m}{E'R}$$

$$M = W\left(2U\right)^{-3/4} \qquad L = G\left(2U\right)^{1/4} \qquad \bar{\rho} = \frac{\rho}{\rho_R}$$

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