TRIBOLOGICAL BEHAVIOUR OF POLYALPHAOLEFINS: WEAR AND ROLLING CONTACT FATIGUE TESTS

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SUMMARY

Polyalphaolefin fluids are gaining rapid acceptance as high-performance lubricants and functional fluids because they have certain inherent, and highly desirable, characteristics relative to mineral oils. One of these characteristics is their low toxicity. It combined with excellent viscometrics and lubricity, have made low-viscosity PAO fluids an important component in lubricant formulations.

Typical data found in product specifications for lubricants are the kinematic viscosity and the viscosity index. These values do not give enough information to choose the optimum lubricant for a lubricated contact. In mechanical systems take place rolling, sliding and rolling/sliding contacts, therefore lubricants have to work the best possible in these operation conditions.

In this study are experimentally determined the L_{50}, L_{10} and Weibull´s slope (α) of polyalphaolefins with two different viscosities. This test was made on a four-ball machine (Stanhope Seta). Wear test also was made on a four-ball tester (Roxana) in order to measure the wear scar diameter (WSD), and the flash temperature parameter (FTP). Lubricants were identified through infrared spectroscopy, and ball´s pittings were observed with SEM.

Keywords: Polyalphaolefins, rolling contact fatigue, wear, SEM, four-ball machine.

1. INTRODUCTION

The polyalphaolefin (PAO) fluids are the first and foremost synthetic hydrocarbons in commercial importance [1]. This commercial importance is due to their wide operational temperature range, good viscometrics (high viscosity index), thermal stability, hydrolytic stability, compatibility with mineral oils, low toxicity, good to moderate relative biodegradability, etc. [2].

Typical data found in product specifications for lubricants are the kinematic viscosity and the viscosity index. These values do not give enough information to choose the optimum lubricant for a lubricated contact [3]. In mec7hanical systems take place rolling, sliding and rolling/sliding contacts, therefore lubricants have to work the best possible in these operation conditions. In order to avoid wear, it is necessary to keep the lubricant film thick enough to prevent any contact between the surfaces. Friction should be minimised in order to obtain as high efficiency as possible and pressure peaks should be as small as possible to avoid large sub-surface stresses and thus avoid pitting fatigue [3]. An equation that describe the influence of some parameters in lubricant film is the Hamrock-Dowson one (Equation 1) for elliptical conjunctions.

$$h_c = 2.69\, u^{0.67} \cdot \eta_0^{0.67} \cdot E^{'-0.073} \cdot \alpha^{0.53} \cdot w^{-0.067} \cdot R_x^{0.464} \left(1 - 0.61 e^{-0.73 k}\right) \tag{1}$$

This paper present the results of current studies about the general tribological behaviour of polyalphaolefins. Measurements of wear scar diameter, rolling contact fatigue, flash temperature parameter (FTP) were made. Lubricants were identified through infrared spectroscopy, and ball´s pittings were observed with SEM.

2. EXPERIMENTAL

2.1 Apparatus

The four ball friction apparatus used in this study was a Falex Roxana machine and tests were made using the ASTM standard D 4172-88. The rolling contact fatigue tests took place in a Stanhope Seta four ball machine and the standard used was the IP-300/82. The ball test material was AISI 52100 steel, 12.7 mm. in diameter, with a hardness of about 65 HRC and its chemical composition can be observed on Table 1.

Chemical Composition (wt. %)	
C	0.98 – 1.1
Si	0.15 – 0.30
Mn	0.25 – 0.45
P	0.025
S	0.025
Cr	1.30 – 1.60

Table 1: Chemical Composition (wt. %) of the AISI 52100 steel ball material.

2.2 Oil samples

The viscosities of polyalphaolefins used in this study are shown in Table 2.

Code	Viscosity (cSt)	
	40ºC	100ºC
PAO 6	31.0	5.90
PAO X	55.0	8.70

Table 2: Viscosities of Oils.

Both oil samples were identified through infrared spectroscopy and the spectrum were the same of the spectra given by Sadtler Research Laboratories, Inc., for polyolefins. The spectrometer used was the FT-IR PARAGON 1000 of PERKIN ELMER.

2.3 Calculated parameters and test conditions

The calculated parameters in the wear tests were the scar wear diameter (WSD) and the flash temperature parameter (FTP). The WSD was measured using a microscope according to ASTM 4172-88. The FTP was calculated with the formula:

$$FTP = W/d^{1.4} \qquad (2)$$

where W is the load in kilograms, and d is the mean wear scar diameter (WSD) in millimetres at this load. This single number express the critical flash temperature above which a given lubricant will fail under given conditions.

The calculated parameters in the rolling contact fatigue tests were the L_{10}, L_{50}, and Weibull's slope (α). The behaviour load-fatigue life was determined through tests with three different loads (from 500 to 700 kg.) and the same velocity (1470 rev·min^{-1}). The test runs were finished when pitting occurred in any ball. The results were processed using the Weibull's theory.

Wear scar, worn surfaces and pitting fatigue were observed using a scanning electronic microscope (SEM) after test runs.

3. RESULTS AND DISCUSSION

3.1 Rolling contact fatigue

The results of rolling contact fatigue test can be observed in Figure 1. The rolling contact fatigue life decrease with the increase of load. If the Hamrock-Dowson equation is checked, the higher load the thinner oil film thickness in the contact and the asperities interaction is higher too. These phenomena increase the stress below the contact and pitting occur easier.
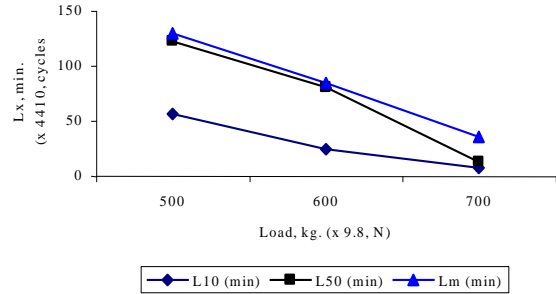

Figure 1: Results of Rolling Contact Fatigue Test in PAO 6.

In the Figure 2 appear the temperature behaviour during the rolling contact fatigue test, it is appreciated that the higher load the higher test temperature. This temperature increase due to load increase, diminish the oil viscosity and the film thickness become thinner too.
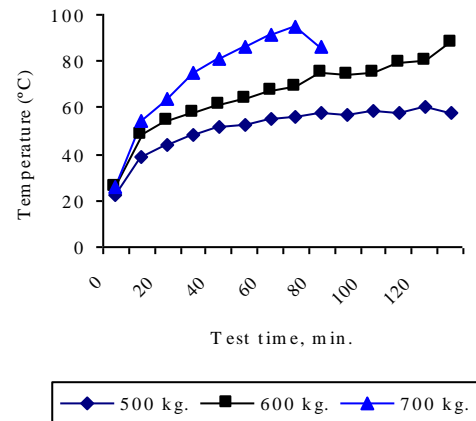

Figure 2: Temperature Behaviour of Rolling Contact Fatigue Test in PAO 6.

The PAO X´s results in rolling contact fatigue life and temperature behaviour are, in general, the same of PAO 6, Figures 3 and 4.
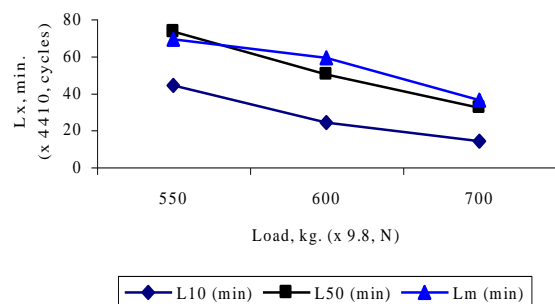

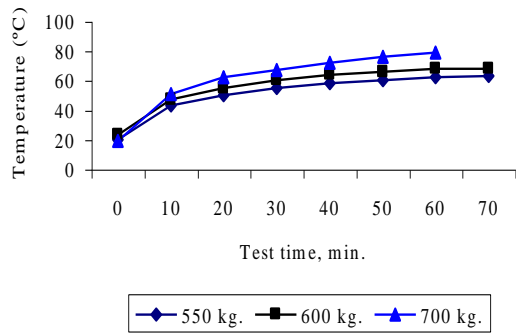*Figure 3: Results of Rolling Contact Fatigue Test in PAO X.*

Figure 4: Temperature Behaviour of Rolling Contact Fatigue Test in PAO X.

If we observed the L_{10} and L_{50} behaviour of PAO 6 and PAO X on the tested load range, Figures 5 and 6, the L_{10} of PAO X is always better but the L_{50} is only better for loads higher to 650 kg. This phenomenon can be attributed to the dynamic process in the conjunctions. The temperature in the test of 700 kg. was always higher in PAO 6, Figure 7, that situation did thinner the film thickness and higher the asperities interactions and sub-surface stresses and thus a lower rolling contact fatigue life. The temperature in the test of 600 kg. was lower in PAO X until a test time of 20 minutes, which coincide approximately with the L_{10} of both oils, after that time the temperature of PAO X become higher and the viscosity of fluid decreased, as well as the film thickness, and the rolling contact fatigue life to make worse, Figure 8.
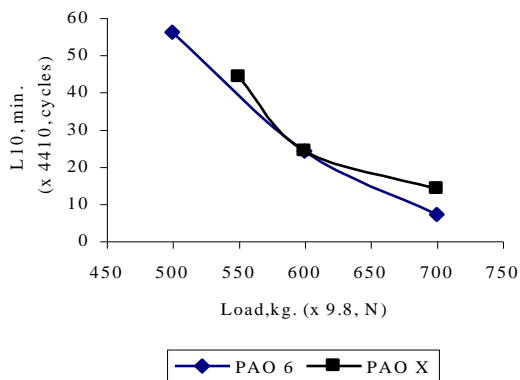

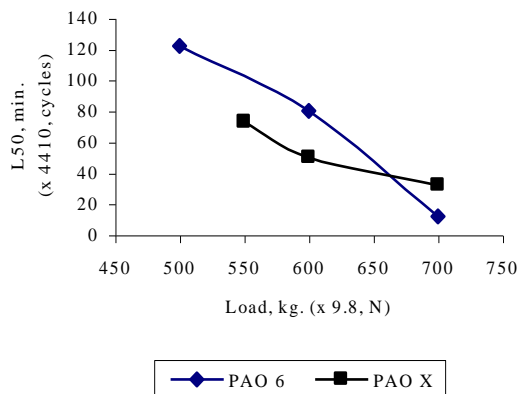*Figure 5: Comparison of L_{10} of PAO 6 and PAO X.*

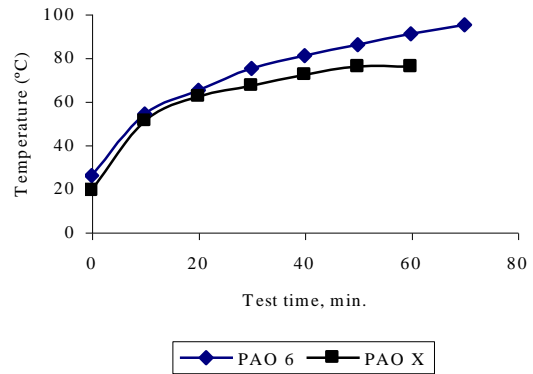*Figure 6: Comparison of L_{50} of PAO 6 and PAO X.*
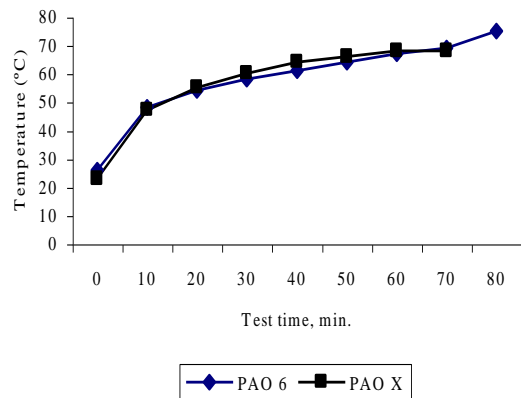

Figure 7: Temperature behaviour in 700 kg. test.

Figure 8: Temperature behaviour in 600 kg. test.

The Weibull's slope (α) for each fatigue test appear on Table 3.

Oils	Loads (kg.)			
	500	550	600	700
PAO 6	2.47	-	1.56	3.49
PAO X	-	3.72	2.57	2.27

Table 3: Weibull´s slope (α)

3.2 Wear preventive characteristics and Flash Temperature Parameter of oils.
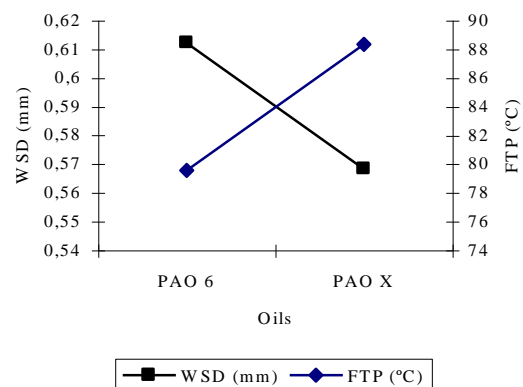

Figure 9: WSD and FTP of PAO 6 and PAO X.

Figure 9 shows how the higher viscosity the lower WSD and the higher FTP. Once again when the viscosity is

increased the oil film thickness is higher and the WSD decrease. That is why the FTP is higher too, however a friction test should be done in order to know if we are saving energy too.

3.3 Wear worn surface characteristics.

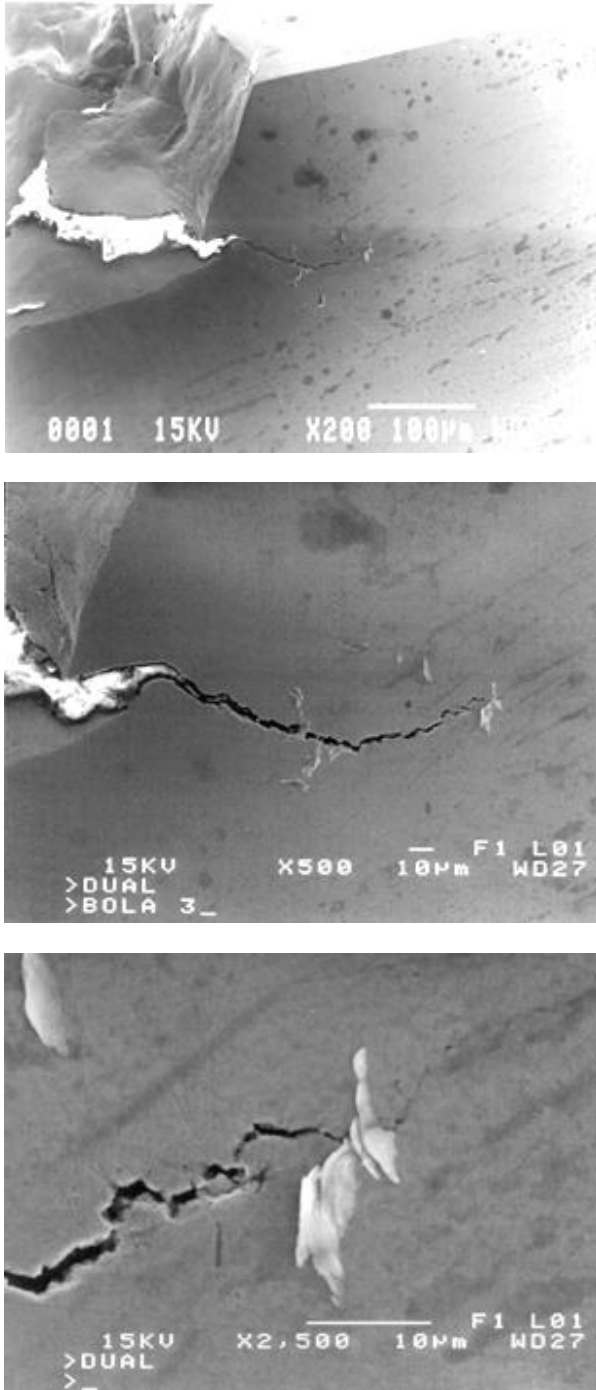The appearance of pitting on test balls are observed on Figures 10 y 11.

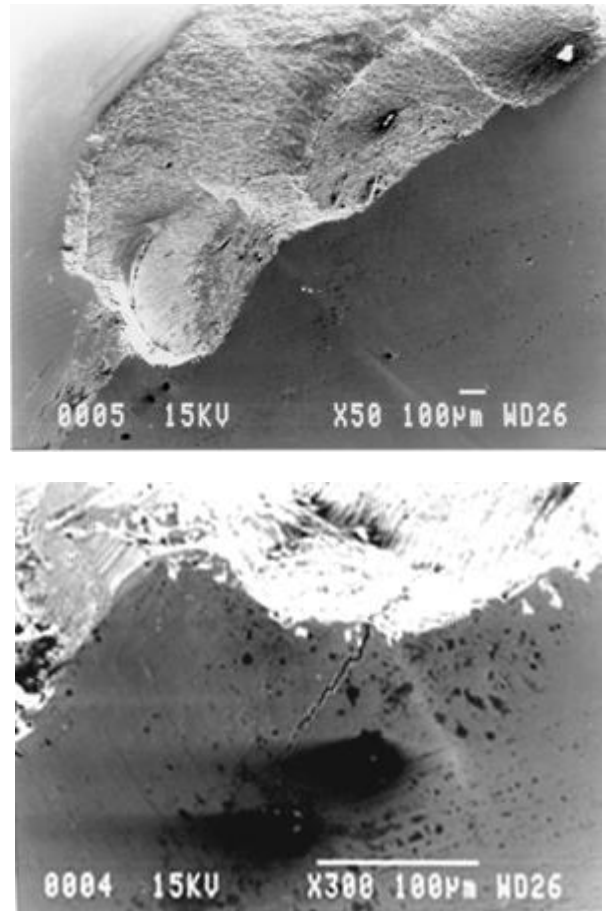*Figure 10: Micrographs showing pitting appearance on a test ball (PAO 6, 600 kg., 58 min.) .*

Figure 11: Micrographs showing pitting appearance on a test ball (PAO X, 600 kg., 61 min.) .

The prime cause of contact fatigue is the high stresses generated by Hertzian contact. As is the case with other forms of fatigue, the three stages of initiation, propagation and fracture are identified. In the initiation stage is originated by a small crack from either a subsurface inclusion or at the surface from wearing contact between the two rolling bodies. Initiation is strongly correlated with shear stress and most crack initiation sites from subsurface inclusions are located close to the subsurface maximum of shear stress for Hertzian contact. The next stage is crack propagation where more rapid growth of the crack occurs leading to the formation of what is in fundamental terms a wear particle. The final stage is fracture where the crack is large enough to allow detachment of the wear particle. Once the wear particle is detached, the rolling contact vibrates since one of the contacting bodies no longer has an accurately circular profile and if rolling speed is high, considerable noise is generated [4]. This phenomenon can be observed on rolling fatigue test (IP 300). Contact fatigue occurs in both lubricated and unlubricated situations but the presence of oil does modify the mechanisms involved. Any oil present inside a rolling contact is subjected to very high pressures of 1 GPa or more. This high pressure enables the oil to separate the contacting surfaces, suppress the surface mode of contact fatigue and thereby prolong the working life of the rolling materials. The same high pressure does however accelerate crack growth by pressurising the

space between the crack surfaces. This theory is the same that Jin and Kang stated [5].

If the Figures above are observed, cracks´s depth are found around 150 μm. From theory of Hertz it can be calculated the depth which maximum stress appear. Wang [6] did this calculus for a four-ball machine and for a load of 600 kg obtained the following results:

p_{max} = 8709 MPa (maximum normal stress).
τ_{max} = 2612 MPa (maximum shear stress).
z = 139 μm (depth of maximum shear stress).

4. CONCLUSIONS

From the results it can be concluded that:
1. The higher load the higher temperature and lower rolling contact fatigue life.
2. The viscosity has an important influence on rolling contact fatigue life but the temperature should not be despised.
3. In the wear tests the results were better when the viscosity was increased.
4. In oils belong to the same family properties as pressure-viscosity coefficient, temperature-viscosity coefficient, compressibility, thermal conductivity and heat capacity per unit volume are approximately equals. In this case the viscosity can become in the principal property.
5. The theory of Hertz is verified for both oils.

5. REFERENCES

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