Numerical and experimental study of dynamic elastomeric seals behaviour under actively controlled temperature conditions.

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ABSTRACT

Clearance forming between rotating shafts and radial lip seals has been investigated both numerically and experimentally at various temperature conditions. Numerical models of clearance forming has been developed taking in account the effect of temperature, defining the material constituting properties according to empiric WLF function. Experiments have been performed with a special apparatus that allows control of lubricant temperature and level at each shaft speed . Results agree with simulations and give useful indications for pratical applications.

NOMENCLATURE

τ	relaxation time	p_d	dynamic stress
Ψ	phase displacement	p_s	static stress
ω	shaft speed	p_t	total stress
R_a	surface roughness	t	time
T_g	glass transition temperature	x_d	dynamic strain
e_d	dynamic eccentricity	x_s	static strain
e_s	static eccentricity	x_t	total strain

1. INTRODUCTION

Radial lip seals for rotating shafts are widely used to retain oil and exclude contaminants, often shielding bearings or journal boxes for rotating shafts. They are cheap, compact and can be used also in presence of vibrations or little radial runout.
Many types of seals are available, differing in material and geometry in order to be suitable for several operating conditions. In spite of this variety, they are always built up by three fundamental elements: an elastomeric ring, a metallic stiffening insert and a garter spring (see figure 1).

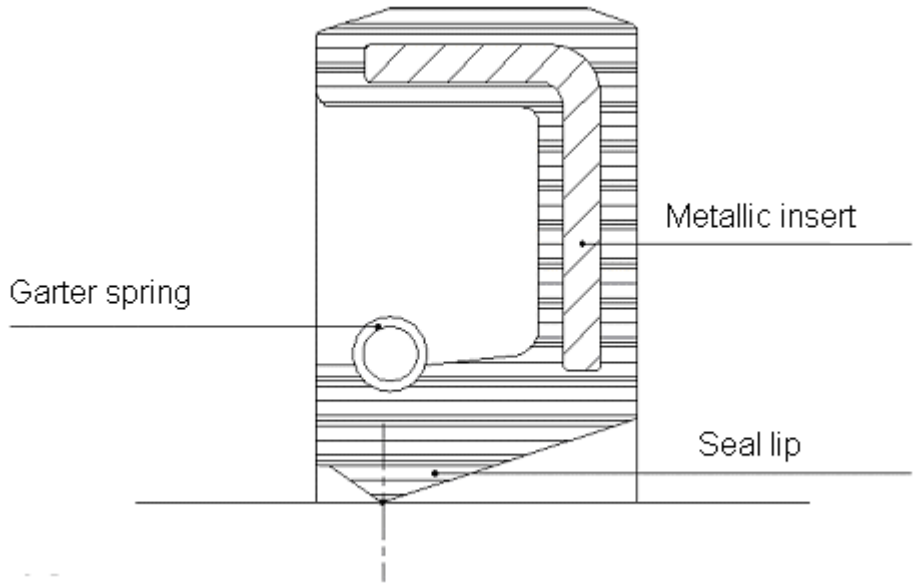

Figure 1: Section of radial lip seal.

The ring ends, at shaft interface, with a lip; the garter spring stiffens the lip and assures a better uniformity of the bearing pressure between seal and shaft on varying working conditions. Rotary lip seals are very simple mechanical system components that have highly sophisticated and complex operating mechanisms. Investigations on the sealing and lubrication of radial lip seals have a long tradition (1), (2), (3), involving different disciplines including hydrodynamics, materials science and tribology. Their behaviour has been explained with the influence of temperature (4) and visco-elastohydrodinamic lubrication (5), (6), (7), but such explanations do not completely justify experimental results that show several stages of the leakage rate at different shaft rotating speed (8) (9) (10).

Preceding works have shown many experimental results achieved both in working conditions and with simulation apparatus by means of several transducers: cameras, torque meters, accelerometers (for *Frequency Response Function analysis*), thermocouples, flowmeter and, recently, a strain gauge transducer mounted on a machine that simulates actual operating conditions (10), (11).

Available experimental data also made possible to start implementing a numerical simulation of the phenomenon with Finite Elements Method. In (11) is described a first approach where dynamic seals behaviour was approximated by a non-linear Mooney-Rivlin model. This method was probed to be suitable for high frequencies rotating stress and improved in (12).
In order to reach a better synthesis of the tests results and the theoretical studies, we procedeed with experimental and numerical tests aimed at simulating temperature effects in various working conditions.

2. NUMERICAL ANALYSIS

In (10) and (11) has been described a FEM seal model obtained as solid of rotation from a simplified section and its reliability has been validated by comparisons of experimental results of Frequency Response Function analysis. A further development (12) took account of temperature effects on material constitutive law and has been resumed in this paper.

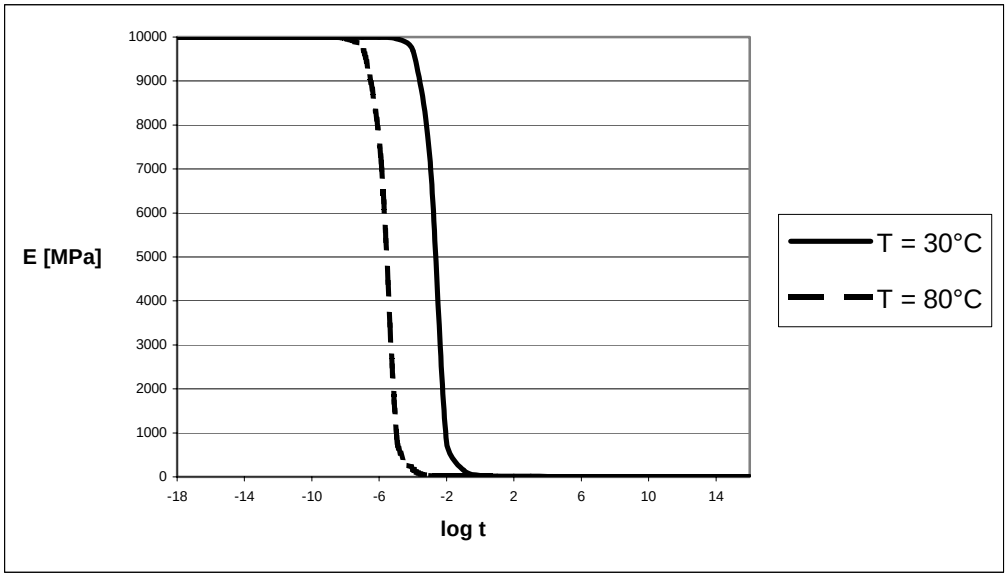

Figure 2: Relaxation curve shift.

The approach is based on repeating the simulation at various elastomer temperature, so *Prony series*, that describes material constituting properties, changes according to empiric *Williams - Landell - Ferry* shift function, that gives the glass transition frequency shift. When stress frequency reaches the transition zone, the seal lip cannot follow shaft movement and clearance happens.

In detail, simulations exploit the ability of ABAQUS software to simulate varying boundary conditions applying to the seal lip a static strain x_s and a sinusoidal one, $x_d(t)$, obtaining altogether:

$$x_t(t) = x_s + x_d(t) = x_s + |x_d| \cos \omega t$$

The corresponding total stress results are:

$$p_t(t, y) = p_s(y) + p_d(t, y) = p_s(y) + |p_d(y)| \cos(\omega t + \Psi)$$

So is possible to suppose that the detachment happens if, in i^{th} node, where the stress is maximum, dynamic stress become greater than the static one, giving:

$$p_t(t, y_i) < 0$$

Physically this happen near to the glass transition of the elastomer, that cause a quick increase in rubber stiffness. Table 1 shows the *Prony* series coefficients used at reference temperature of 303 K, while the glass transition temperature has been estimated as T_g.=253 K. Curves in figure 2 exemplify as a relaxation curves shifts passing from reference temperature to working temperature (353 K).

The geometric modeling has been made up both with a simplified 2D model (see figure 3) and with a 3D model (see figure 4). The first has been useful to quickly adjusting conditions and to verify results comparing with (5), while the 3D model is more realistic and takes in account also the tangential stress that the shaft generates on the contact surface.

Table 1: Prony series coefficients

$g_1 = 0{,}9504$	$\tau_1 = 0{,}003$ [s]
$g_2 = 0{,}045$	$\tau_2 = 0{,}072$ [s]
$g_3 = 0{,}003$	$\tau_3 = 1{,}01$ [s]
$g_4 = 0{,}0003$	$\tau_4 = 19{,}9$ [s]
$g_5 = 0{,}00013$	$\tau_5 = 367$ [s]
$g_6 = 0{,}00011975$	$\tau_6 = 3606$ [s]
$g_7 = 0{,}0000828$	$\tau_7 = 47347$ [s]

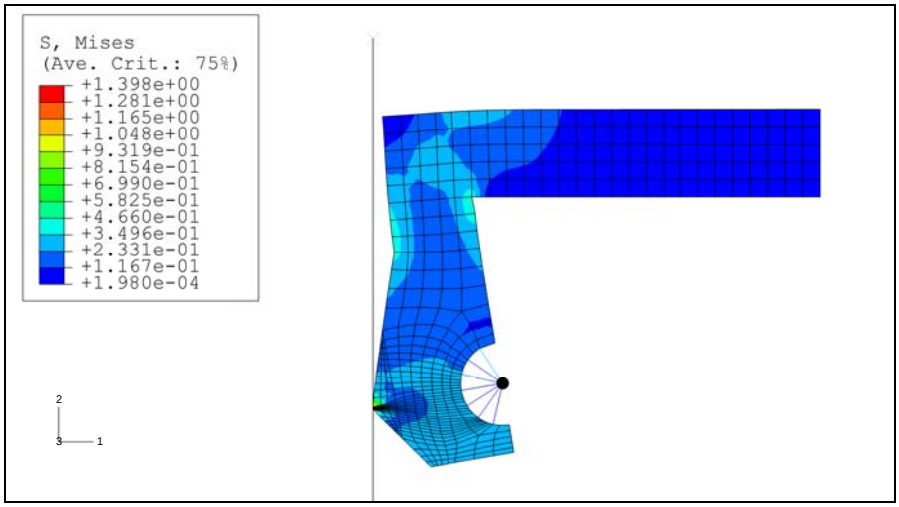

Figure 3: Von Mises stress in bidimensional model.

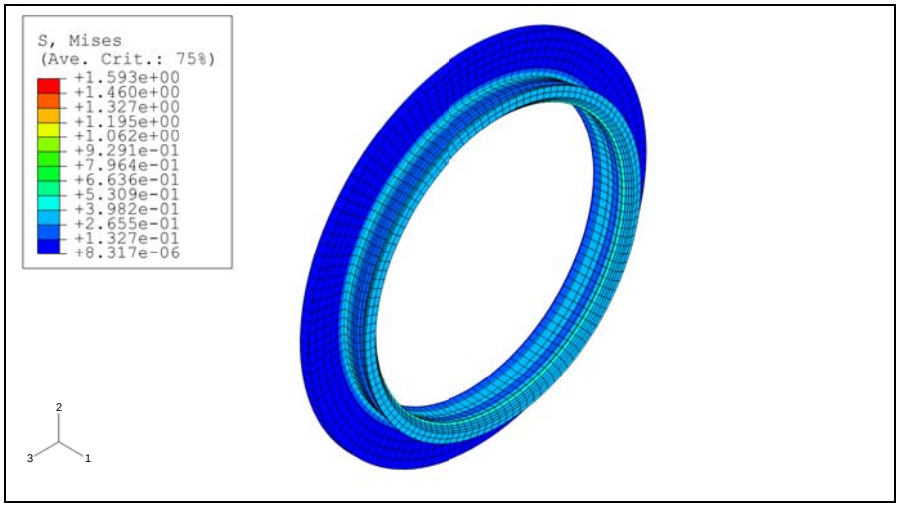

Figure 4: Von Mises stress in tridimensional model

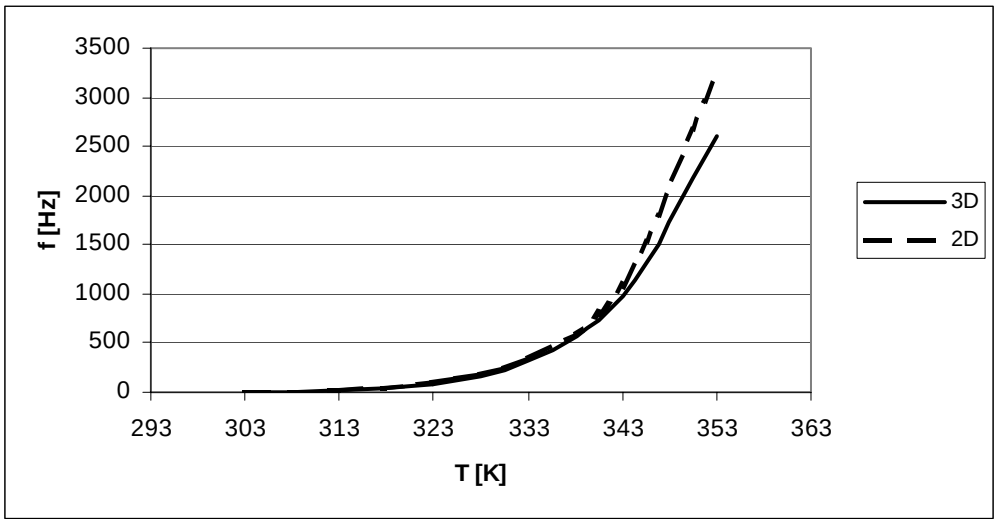

Figure 5: Comparison of clearance frequencies in 2D and 3D models.

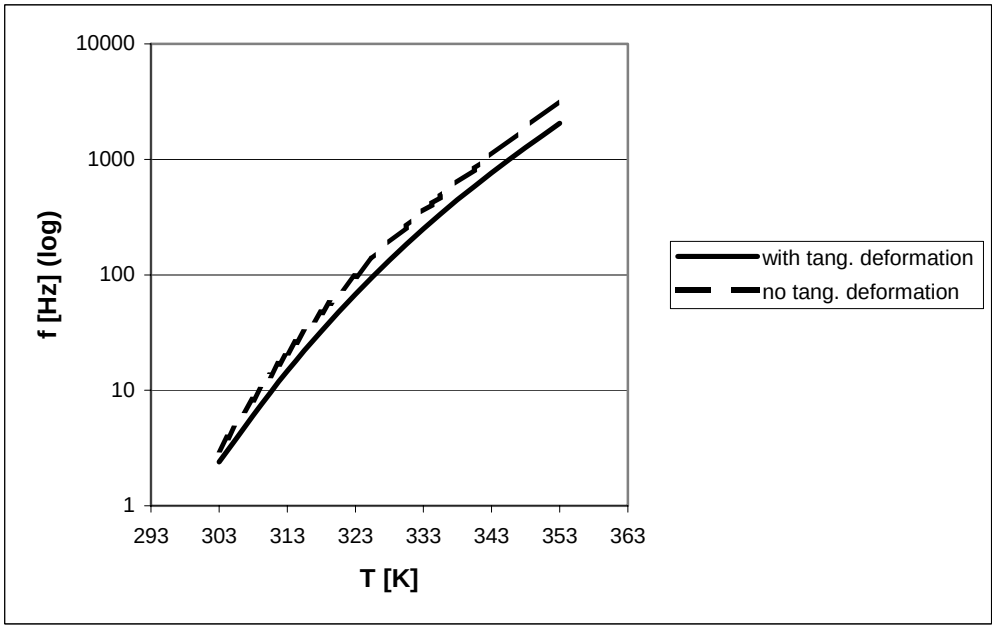

Figure 6: Effect of tangential stress

Bidimensional and tridimensional models agree on clearance frequencies, as curves in figure 5 show. Figure 6 highlights the contribution of tangential stress to reaching the clearance forming.

3. TEST RIG

The test rig is made up of three subsystems: a rotating shaft machine that allows a repeat of actual working conditions, an oil circuit that controls the temperature and the level of the fluid and several measuring and data acquisition instruments devoted to observe the presence of clearances between the shaft surface and the seal lip.

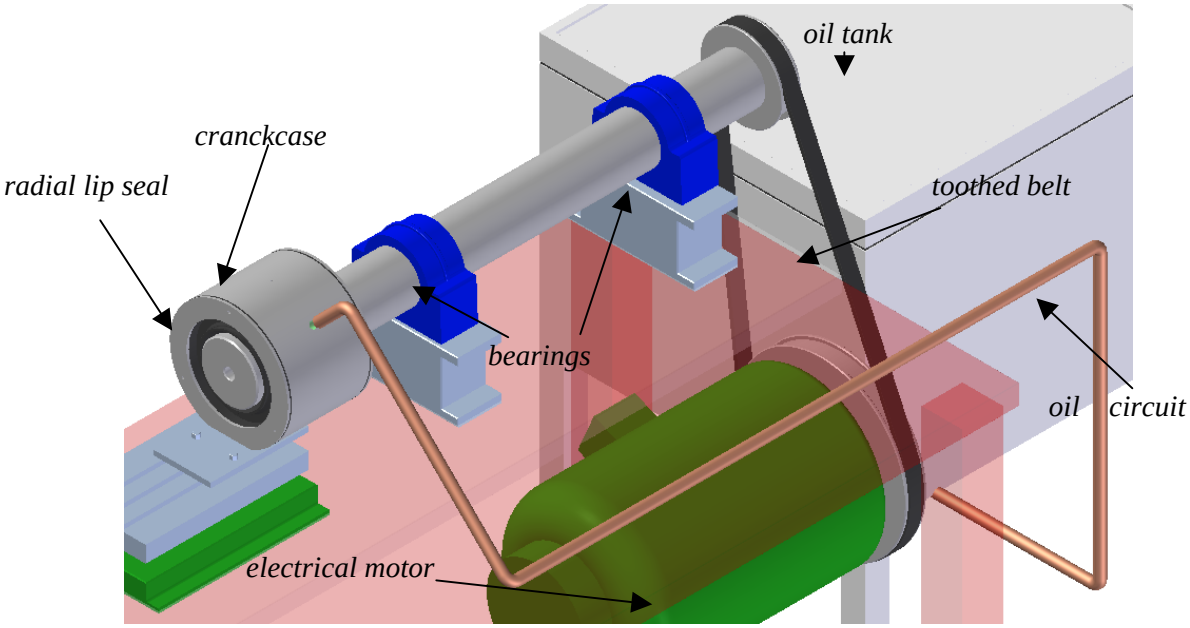

Figure 7: The test machine

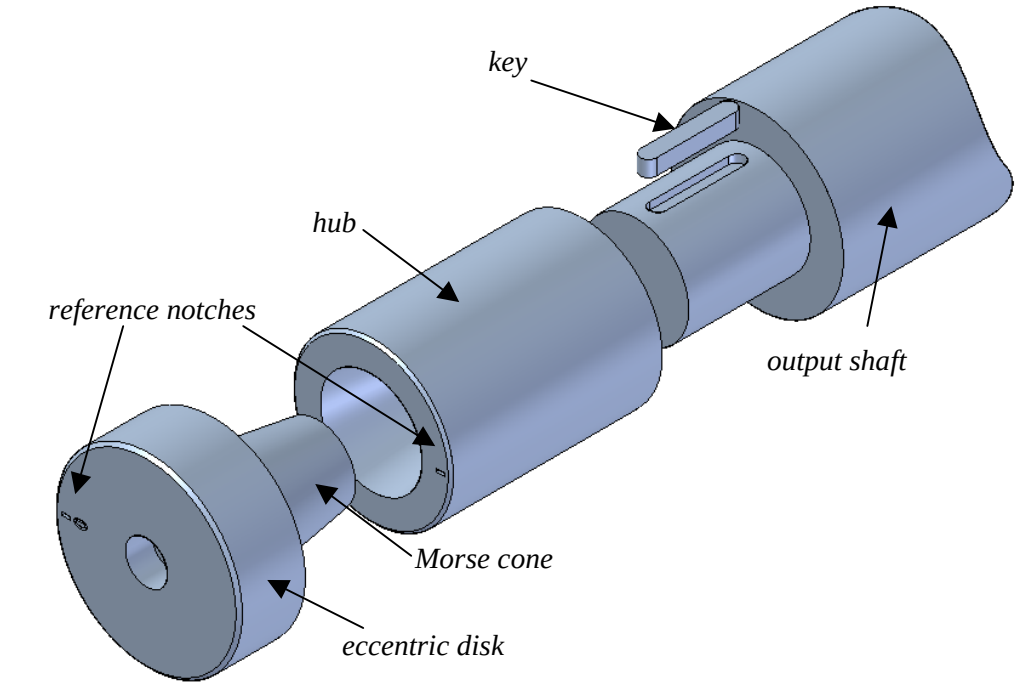

Figure 8: The end of the output shaft

3.1 The machine

The mechanism is mainly constituted by a direct current electrical motor moving a shaft through a toothed belt transmission (see figure 7). The speed is manually controlled and can vary continuously in the 300-8000 rpm range (at the output shaft). At the end, the shaft has a disk of circular profile which is in contact with the test seal. The dynamic eccentricity of this disk can be set to any value between 0 and 0.52 mm by mean of a conical coupling (see figure 8). The disk is polished and has a surface roughness R_a = 0.188 μm, as measured by using a profilometer. The position of the fixed seal housing can also be adjusted, giving the desired static eccentricity.

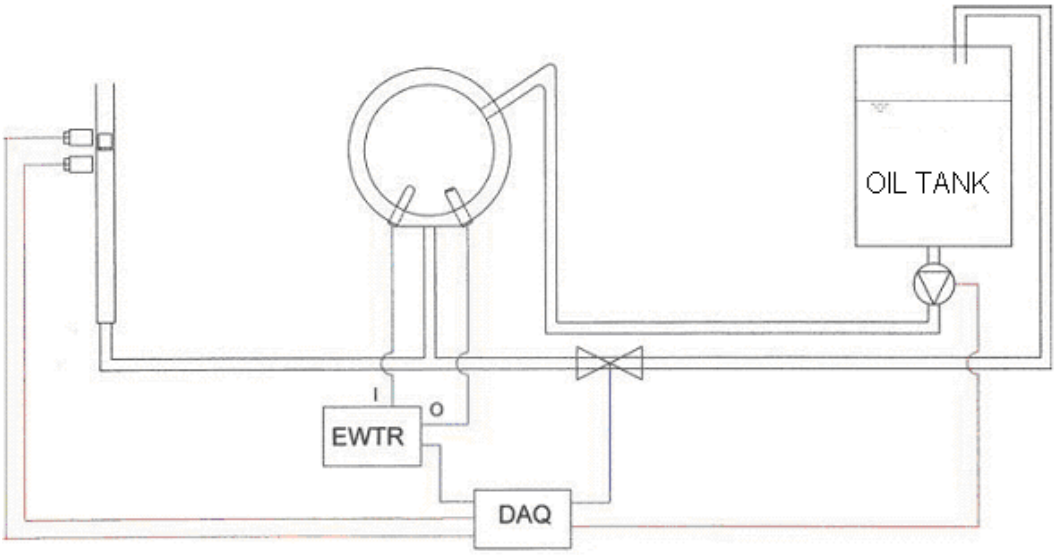

Figure 9: The lubricant circuit

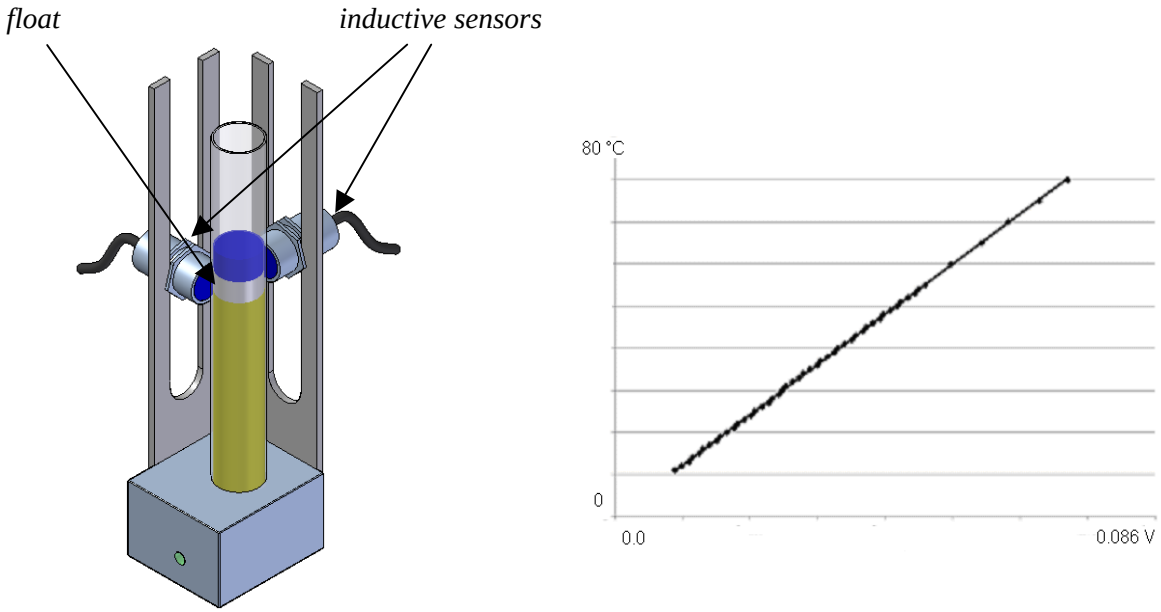

Figure 10: Level sensor and temperature sensor calibration curve.

3.2 The lubricant circuit

This circuit (see figure 9) allows control of both the temperature and the level of the lubricant in the crankcase; as shown in figure 9, it is made up by an oil cooler, a gear pump, a cranckcase heater, temperature and level sensors. These devices and solenoid valves are driven by the same I/O board that is used for data acquiring, so the control software has been written in LabView environment.

In order to ensure stability and robustness in the control loop, the most critical aspect is given by the relative low outflow of the oil, particulary at low temperatures. Since, of course, the cranckcase cannot be pressurized (regarding this matter, the top of the box has been drilled), the fluid must flow only by gravity with very little head, so the limited discharge of warm oil could prevent heat disposal at high speed. The adopted solution consists in mounting an adjustable-with-by-pass flow control valve on pump delivery, partially starting to change the oil in the box on increasing shaft speed, so that the thermic transient remains sufficiently slow in every running condition. In particular, is possible to obtain steady state lubricant running temperature in the 30-70 °C range with the accuracy of ±1 °C.

On the other side, adjusting the lubricant level is necessary for controlling the partially flooded conditions of the contact between the seal and the shaft suface. Seeing that inside the case there is not enough room for sophisticated level sensors, a communicating glass pipe with a float, shown in figure 10, has been adopted to obtain a reliable indicator that can be detected by cheap inductive sensors. The sensor position is easily adjustable tightening them on vertical brackets and so it is possible to set both the maximum and the minimum fluid level with the test rig working.

Oil temperature is measured by a Pt-100 resistor mounted inside the crankcase. Its calibration curve (see figure 10) shows an almost linear behaviour along the whole working range.

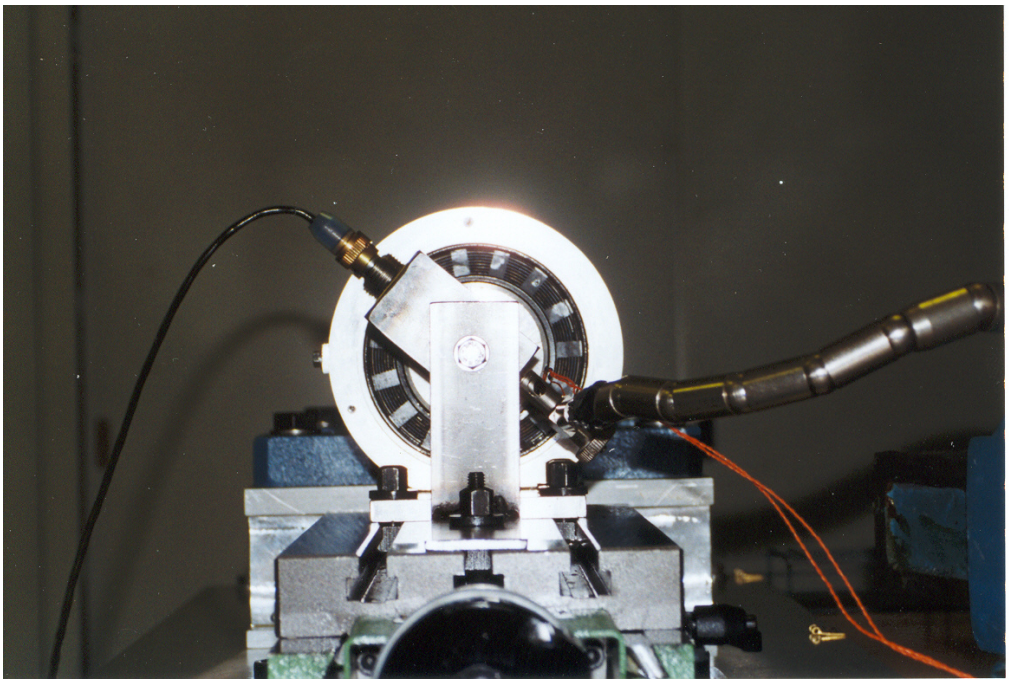

Figure 11: Sensors mounting

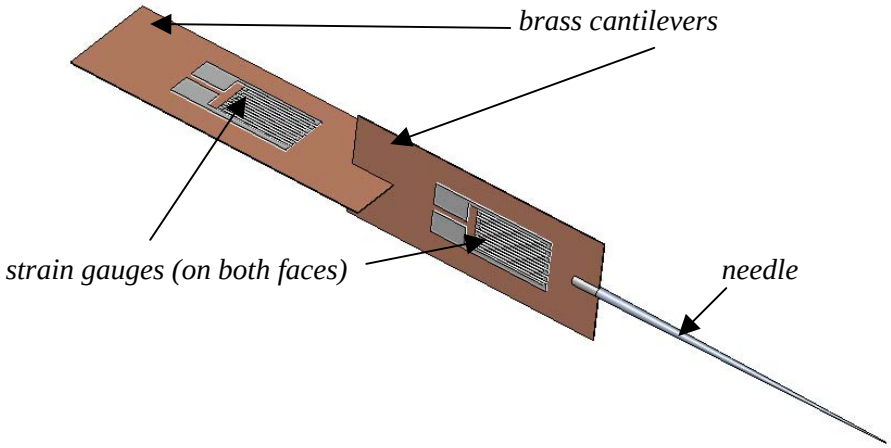

Figure 12: Strain gauge transducer

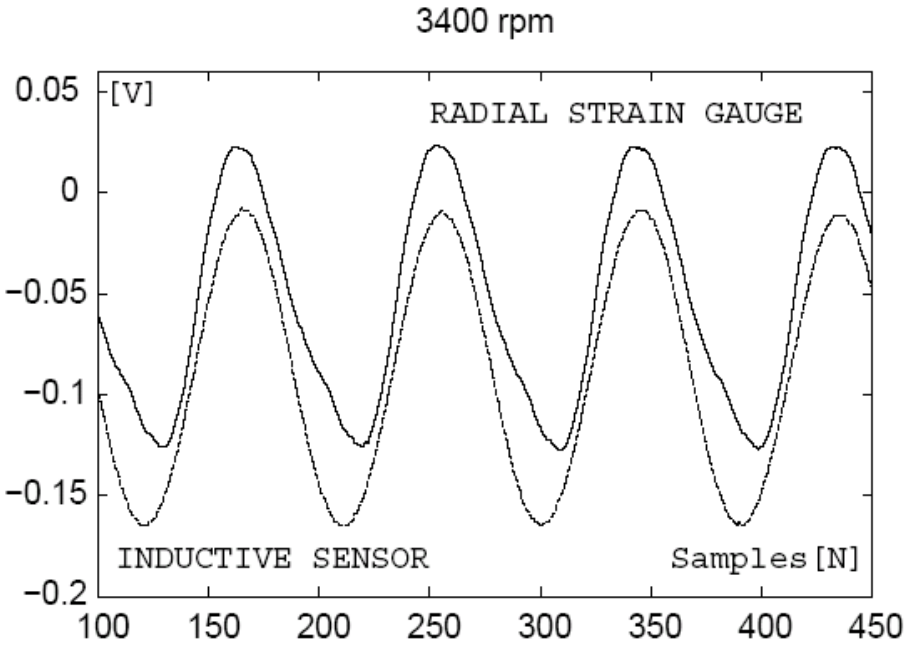

Figure 13: Lip and shaft profile radial displacement

3.3 Measuring Instruments

As figure 11 shows, an inductive sensor collects the surface of the disk profile with which the seal is coupled and hence the radial displacement imposed on the lip. This signal is useful to be compared with the lip position, detected by a strain gauge transducer (see figure 12): when the two signals separate each other (see figure 13) there is a clearance between the lip and the disk.

In details, the transducer is made up by two brass cantilever located at right angles to each other, of dimensions 25 × 12 mm. On each cantilever are glued two strain gauges, one for each side. At the end of the brass plate there is a steel needle with diameter 0.5 mm.

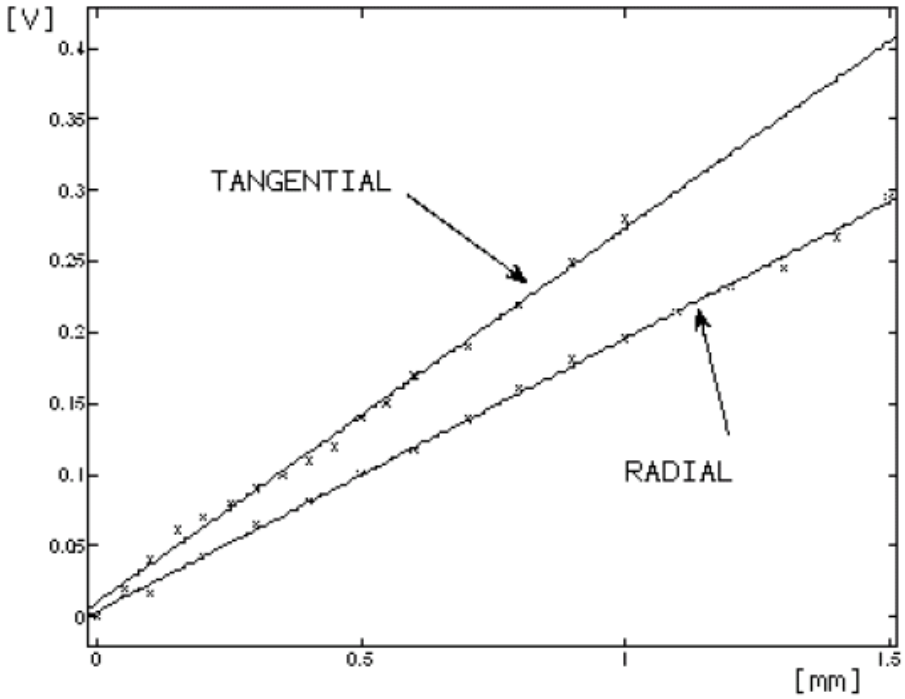

Figure 14: Transducer calibration curves

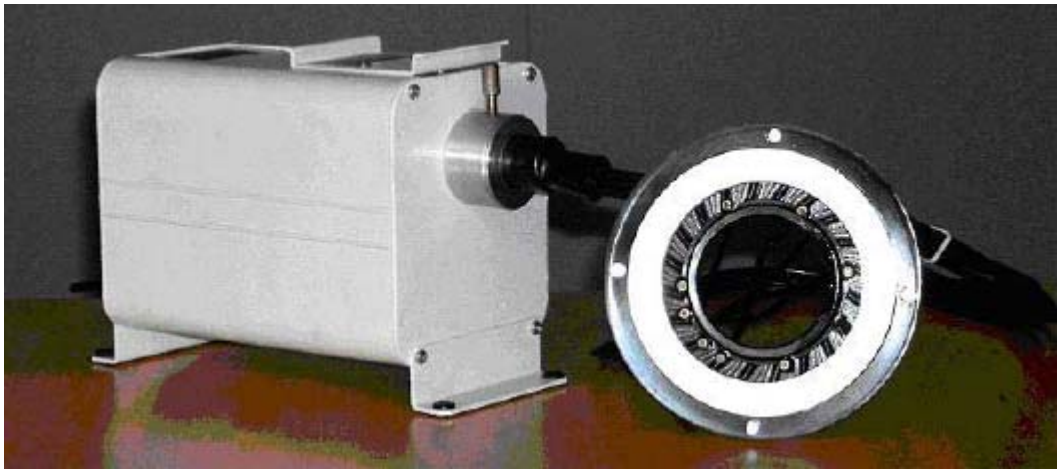

Figure 15: Fiber optic mounted on the back lid

One can note that the inductive sensor and the cantilevers transducer are mounted with a phase difference of 180°, as the two signals may easily superimposed (inverting one of them). The transducer depicted in figure 12 is inserted in the lip, in the zone corresponding to the maximum eccentricity.

The apparatus also includes a signal conditioning amplifier Instrument Division 2120A for the strain gauges, a Kaman measuring systems P3500 for the inductive sensor and a National Instruments PCI-MIO- 16E board that collects both the signals.

Calibration curves of both radial and tangential signals are shown in figure 14. They have been calculated interpolating measurements obtained in 25 equally spaced points in the range of needle tip displacement of 0 ÷ 1.5 mm. Graphs highlight a good linearity in transducer response. A fiber optic lamp (see figure 15) lights the seal from the inside the housing, allowing direct verification, when the measurements point out a clearance, the presence of a gap between the lip and the disk.

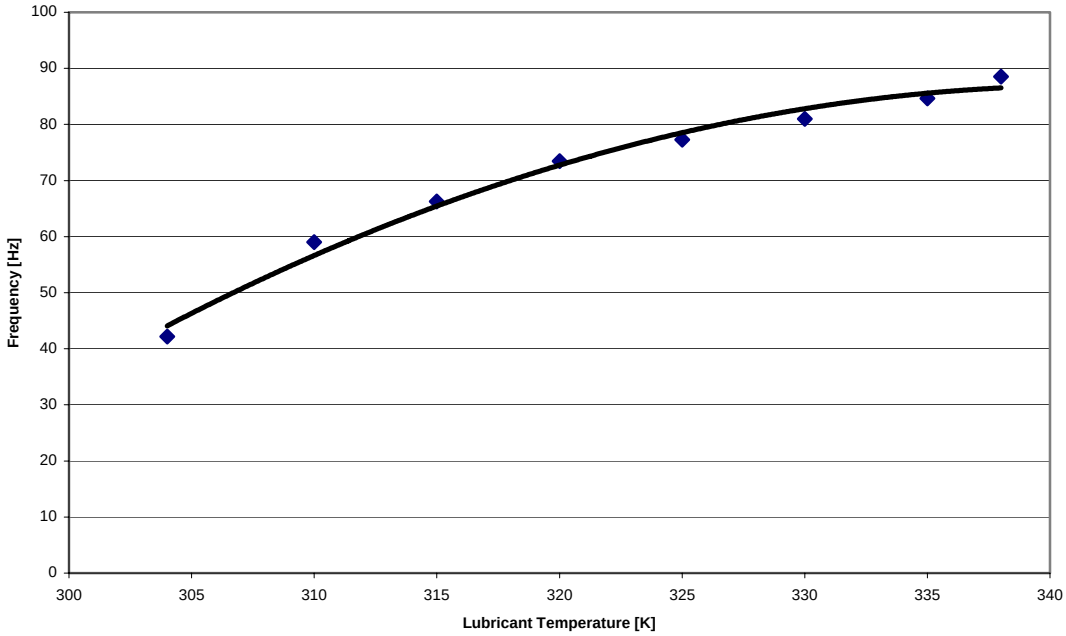

Figure 16: Clearance forming frequency (test measurements, e_d=0,4 mm)

4. RESULTS

The strain gauge transducer allows robust recognition of the presence of clearances between seal lip and shaft, in particular studying the derivative of a function obtained as Fourier series approximation of the acquired signal (12).

It is known (8), (9) that working cycles have a great influence on the seal behaviour. In order to reach seal steady state, five working cycle of one hour at 1000 rpm have been done. Then, data have been collected varying the fluid temperature between 31 and 65 °C with steps of 5 °C and the shaft speed between 500 and 6000 rpm with steps of 100 rpm. Sampling rate vary from 1000 to 6000 Hz. The graph in figure 16 shows the clearance forming frequency observed at various lubricant temperatures. As may be expected, at higher temperature the lip become more able to follow the shaft movement and the clearance occurs at higher frequencies.

Figure 17 shows the simulation results at various dynamic eccentricity values. The meaning of these graphs can be correctly understood referring to hypotheses on which the simulation is based. In particular, the absence of lubricant causes the detachment of the lip at low frequencies, as experimentally proved in (13). Not surprisingly, the comparison with the experimental curve highlights that the pressure in the oil film between seal and shaft greatly improves the sealing effect. Nevertheless, once this difference is taken into account, both graphs show frequency increment at increasing temperatures. Underestimating clearance frequency at low temperatures also causes quite a different shape in the graph trends: while the experimental curve can be approximated by a parabolic curve with decreasing slope, the FEM analysis result follows a 4^{th} degree polynomial with increasing slope.

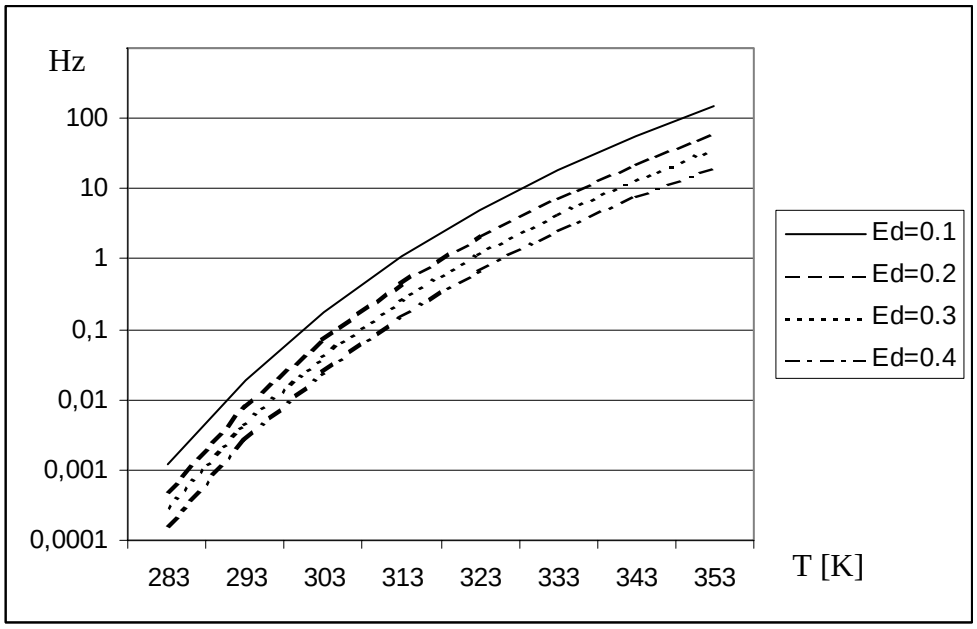

Figure 17: Clearance forming frequencies (FEM simulations)

5. CONCLUSIONS

Clearance forming between rotating shafts and radial lip seals has been investigated both numerically and experimentally at various temperature conditions. In both cases there is a significant influence of the temperature on the clearance forming frequency.

Further studies will include measuring simultaneously the clearance area and the leakage rate, with the aim of evaluating if, on varying temperature, it is predominantly influenced by the clearance size or by the lubricant viscosity.

Results suggest that low temperatures in working machines (e.g. during a transient) should be carefully considered in order to avoid unwanted lubricant leakage, particulary in the presence of large eccentricities.

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