Mechanical Frontal Seals Used in Centrifugal Pumps - From Theory to Experiment

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Abstract: *The paper includes a theoretical approach to the frontal mechanical seals with rubber bellows, used in the conveyance of fluids in centrifugal pumps, in terms of friction coefficients, balancing and possible fluid losses. The study also presents an experimental study for a mechanical seal with rubber bellows and a shaft diameter of 35 mm, operating at different speeds at the pump shaft and different pressures of the transported fluid. At higher pressure the seal is balanced and at lower pressures is discharged.*

Keywords: *Balanced seal, mechanical seal, friction coefficients, fluid pressure*

1. Introduction

Seals are landmarks of centrifugal pumps that have the role to prevent the flow of the fluid transported from inside the hydraulic machine to the outside. They are mounted on the drive shaft, between the housing towards the bearings, preventing fluid leaks to the outside.
Seals can be: - soft seal, cord-type, allowing a slight dripping of the fluid and used only for transporting water,
- mechanical seal, which can ensure seals up to pressures of 450 bars and peripheral rotation speeds of maximum 60 m/s.
A category very often used in centrifugal pumps are mechanical seals with rubber bellows, which are analyzed in this study. The advantages of frontal mechanical seals, according to the Eagle Burgmann catalog [1] are:
- protection of the shaft along the entire length of the seal
- protection of the sealing face by the special design of the bellows
- they have a high capacity to take over axial displacements
- high flexibility through the wide range of materials.

The most frequently used unbalanced frontal mechanical seals are the MG1 and the RMG12 version for hot water centrifugal pumps. In the experimental part I will use this sealing model with rubber bellows - fig. 1 [1], which can be mounted on shafts with diameters d1=10...100 mm, works at pressures of 16 bar, temperatures t = -20 °C … +140 °C, peripheral speed of 10 m/s and accepts an axial displacement of ±2.0 mm.
Materials used: for the mobile ring of the seal - carbon graphite antimony impregnated, or carbon graphite resin impregnated, or silicon carbide; for the fixed ring – silicon carbide, tungsten carbide. Bellows is made of elastomers as: NBR, EPDM, FKM, HNBR, and the mechanical components are from CrNiMo steel or Hastelloy, mentioned by [1].
The reference standard for this type of seal is EN 12756. The catalog states that the front seals MG1 can also be used as a multiple seal in tandem or in a back-to-back arrangement, depending on the conditions in the installations.
The applications of these FMS seals are manifold: water supply, water and wastewater pumps, circulating pumps, submersible pumps, chemical standard pumps, oil industry, paper industry, food technology etc.,

Fig. 1. Mechanical seal model MG1 [1]

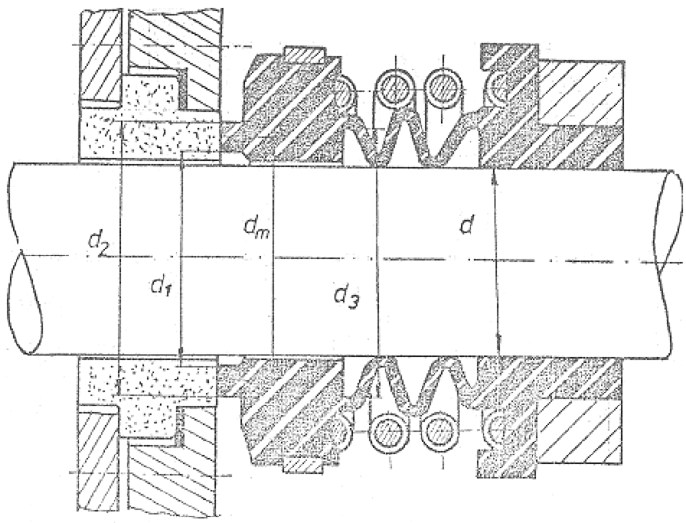

Fig. 2. Frontal mechanical seal with rubber bellows [2]

2. Theoretical approach

In the analysis of the correct functioning of a frontal mechanical seal, it is important to determine the coefficient of friction between the sealing rings and the flow rate of possible leaks, these depending on the shaft speed and the pressure of the transported fluid.
In fig. 3 you can see the distribution of pressures on the rings of an FMS and the way to achieve the balance of forces.
It was noted: p_f – fluid pressure,
d_1, d_2, d_3 – characteristic diameters,
p_{at} – atmospheric pressure,
$F_p, F_{p'}$ – pressure forces on the rings of seal
F_r – force from the spring,
F_f – friction force.

2.1 Equations

To calculate the coefficient of friction μ of a frontal mechanical seal, important in the tribological analysis of the phenomenon of Friction - Lubrication - Wear [3-6], it starts from the balance of forces:

$$F_r + F_p = F_f + F_p^{'} \quad (1)$$

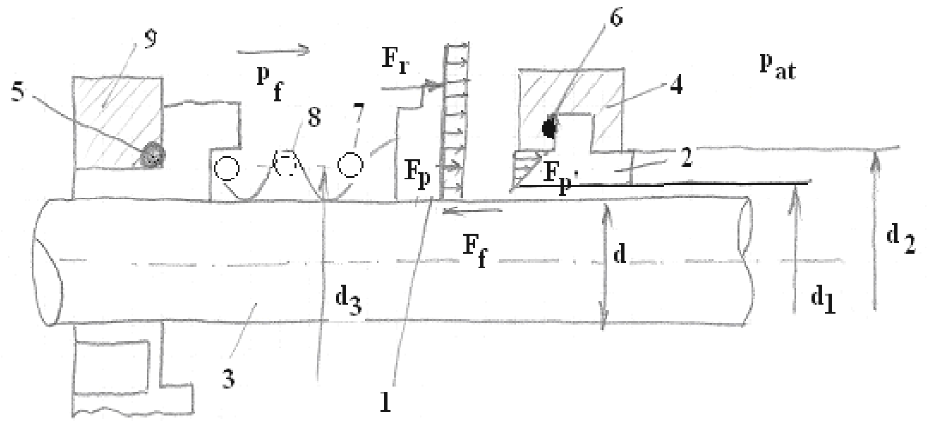

Fig. 3. Pressure distribution on the front mechanical seal rings 1- mobile ring, 2- fixed ring in the housing, 3- shaft, 4 - housing, 5, 6 - O-rings, 7 - spring, 8 - rubber bellows [2]

- axial force exerted on the contact between the two rings by the force of the spring $F_r = K_r f_o$ = the multiplication of the spring constant and its mounting arrow, the force due to the pressure of the fluid to be sealed, which is opposed by the force due to the pressure of the fluid, assumed in linear variation, flowing through the gap between the two rings, equal to

$$F_p^{'} = \frac{\pi}{4}\left(d_2^2 - d_1^2\right)\cdot\frac{p_f}{2} \qquad F_p = \frac{\pi}{4}\left(d_2^2 - d_3^2\right)\cdot p_f \qquad F_f = \mu N \qquad F_r = 150\,N \tag{2}$$

$$\mu N = F_r - \frac{p_f}{2}(d_2^2 - d_1^2)\frac{\pi}{4} + p_f(d_2^2 - d_3^2)\frac{\pi}{4} \tag{3}$$

This relative to the contact surface between the two rings gives us the pressure in the fluid film, called sliding pressure p_g, compared with the fluid pressure p_f.

$$p_g = \frac{N}{\frac{\pi}{4}(d_2^2 - d_1^2)} \geq\leq p_f \tag{4}$$

and allows us to determine if the seal is loaded, balanced, or unloaded, thus:
$p_g = p_f$ the seal is balanced otherwise the seal is unbalanced i.e.
$p_g < p_f$ the seal is loaded,
$p_g > p_f$ the seal is unloaded.
d_1 – the starting diameter of the seal, d_2 – the ended diameter of the seal, d_3 –the spring diameter (bellows).
To determine the friction losses of the sealing rings, from the power absorbed by the direct current electric motor, the losses in the motor rotor windings are subtracted P_i, current losses through ventilation, hysteresis are reduced, due to magnetizations P_0', mechanical losses are reduced P_M, resulting the power dissipated by friction in the sealing rings P_{fr}.

$$P_a = U_A I_A, \quad P_i = f(I_A), \quad P_0' = f(E_o),$$
$$E_o = U_A - r_A I_A, \quad P_M = f(n) = \frac{1}{70}n \tag{5}$$

The friction moment is:

$$M_{fr} = 9550\,\frac{P_{fr}}{n} = F_f \cdot \frac{d_m}{2} = F_f\,\frac{d_1 + d_2}{2},$$

resulting $$F_f = \frac{M_{fr}}{d_m} \tag{6}$$

we will be able to calculate the coefficient of friction between the two sealing rings using equation:

$$\mu = \frac{2M_{fr}}{d_m} \cdot \frac{1}{N}. \tag{7}$$

with reaction force N from equation (3).

2.2 Lubrication conditions

Regarding the losses from the mechanical seal, depending on its values, the friction can be:

- Fluid friction - μ = 0.005
- Mixed friction - μ =0.005-0.03
- friction at the operating limit μ =0.15-0.8
- dry friction - μ > 0.8.

3. Experimental study

The experimental stand of the mechanical seal with the shaft diameter d_1 of 35 mm, from figure 4, consists of shaft, frontal mechanical seal FMS, casing, bearings, coupling with the electric drive.

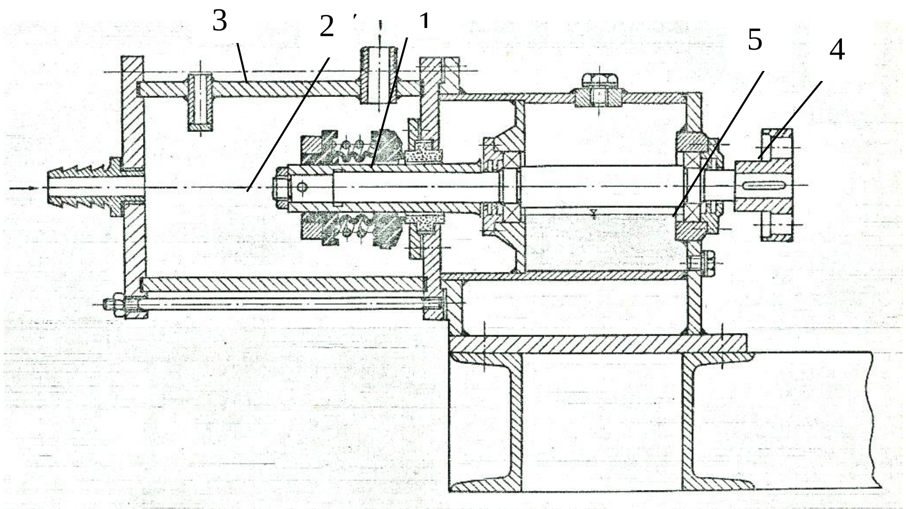

Fig. 4. The test bench of a front seal with a shaft diameter of 35 mm 1-seal, 2-shaft, 3-seal chamber, 4-coupling, 5-oil bearing [2]

Figure 2 shows the mechanical seal in detail, with the fixed ring in the housing, the movable ring on the shaft, the spring, the rubber bellows, the rotating shaft and the housing. Figure 3 shows the distribution of pressures on the frontal mechanical sealing rings and the way to achieve the balance of forces.

The measuring equipment used in these experiments.

Bourdon type manometers are used to measure the pressure to be sealed, having a precision class of 0.6.

The extremely low flow of liquid lost between the sealing rings is measured by the volumetric method, with the help of a graduated cylinder and a timer. For electrical quantities we use ammeter, voltmeter, and tachometer for speed.

To determine the electrical losses mentioned in equation (5), the calibration diagrams of the 1kW / 3000 rpm alternating current motor, illustrated in fig. 5.

Working Procedure

The start of the stand will not be done dry, but first the liquid will be introduced between the two sealing rings at a certain pressure and then the speed will be gradually increased and the axial force between the rings or the sealing pressure will be increased.

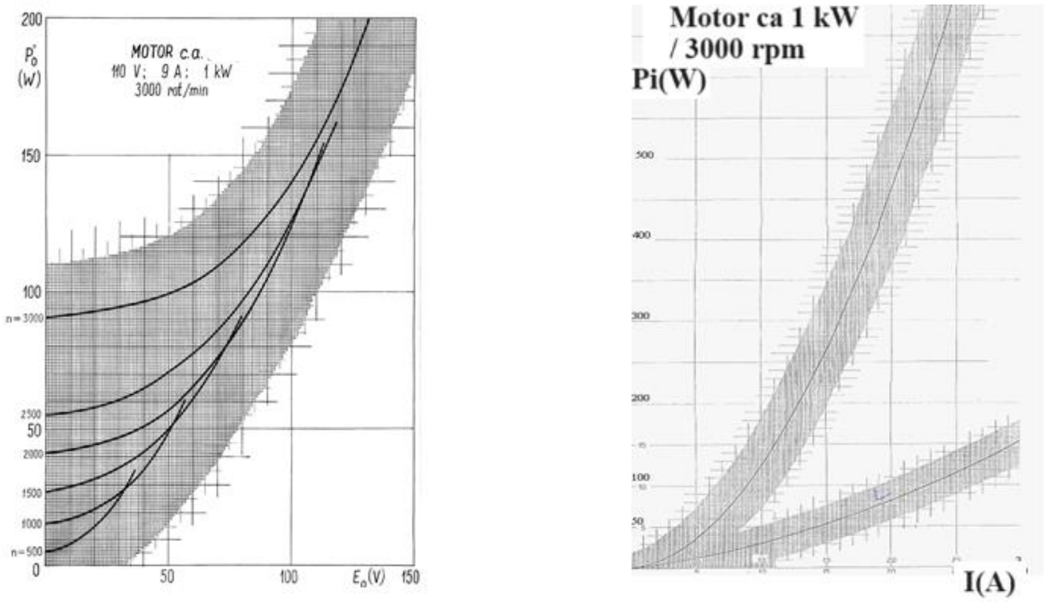

Fig. 5. The curve of electrical losses in copper P_l (W), curve P_o' (n, E_o) for mechanical losses in bearing, for ventilation and hysteresis [2]

Load the seal with fluid pressures of 0.5 bar, 1 bar, 1.5 bar; 2 bar, 2.5 bar; for each pressure the speed is varied, performing measurements at speeds n = 500, 1000, 1500 and 2000 rpm. Some of the experimental data obtained for different sealing pressures at various operating speeds of the sealing will be shown in table no. 1.

Table 1: Experimental results

	p_f	n	q	I_A	U_A	P_a	Pi	Eo (V)	Po	P_M	Pfr	Mfr	Ff	μ	p_g	Obs
	bar	rpm	cm^3/s	(A)	(V)	(W)	(W)		(W)	(W)	(W)	daN cm	daN	(-)	bar	
1	0.5	500	0.6	1.44	24	34.56	10	19.6	12	7.14	5.42	1.035	0.265	0.018	0.35	loded
2	0.5	1000	0.53	1.96	32	62.72	11.2	29.6	28	14.29	9.23	0.882	0.226	0.015	0.3	loded
3	0.5	1500	0.51	2.45	39	95.55	11.6	36.6	30	21.43	32.52	2.071	0.531	0.035	0.34	loded
4	0.5	2000	0.47	2.5	60	150	11.7	56	57	28.57	52.73	2.518	0.646	0.043	0.31	loded
5	0.5	2500	0.43	2.92	83	242.36	15.4	80	80	35.71	111.25	4.250	1.090	0.073	0.47	loded
6	1	500	1.04	2.4	20.5	49.2	11.1	18.22	13	7.14	17.96	3.430	0.879	0.059	0.29	loded
7	1	1000	1.1	2	41	82	8.8	39	35	14.29	23.91	2.284	0.586	0.039	0.24	loded
8	1	1500	1.15	2.1	60	126	11.6	58	60	21.43	32.97	2.099	0.538	0.036	0.31	loded
9	1	2000	1.07	2.5	80	200	15	80	77	28.57	79.43	3.793	0.972	0.065	0.45	loded
10	1	2500	1.02	3	92	276	15.3	85	89	35.71	135.99	5.195	1.332	0.089	0.47	loded
11	1.5	500	1.13	2	21	42	8.8	19.5	15	7.14	11.06	2.112	0.542	0.036	1.24	loded
12	1.5	1000	1.1	2.1	30	63	8.1	28	27	14.29	13.61	1.300	0.333	0.022	1.29	loded
13	1.5	1500	1.07	2.3	51	117.3	11.6	39	34	21.43	50.27	3.201	0.821	0.055	1.45	loded
14	1.5	2000	1.05	2.5	80	200	15.2	67	57	28.57	99.23	4.738	1.215	0.081	1.52	balanced
15	1.5	2500	1.04	2.85	94	267.9	15.4	77	84	35.71	132.79	5.072	1.301	0.087	1.55	balanced
16	2	500	1.33	2.1	22	46.2	8.8	20	15	7.14	15.26	2.914	0.747	0.050	1.89	loaded
17	2	1000	1.33	2	31	62	8.1	29	28	14.29	11.61	1.109	0.284	0.019	1.8	loaded
18	2	1500	1.37	2	40	80	8.3	38	34	21.43	16.27	1.036	0.266	0.018	2	balanced
19	2	2000	1.4	2.4	61	146.4	11	58	56	28.57	50.83	2.427	0.622	0.041	2.1	balanced
20	2	2500	1.44	2.7	80	216	13	77	84	35.71	83.29	3.182	0.816	0.054	2.2	unloaded

4. Analysis of the experimental results

From the analysis of the experimental results, the author found that the pressure between the two rings of the FMS, called sliding pressure, increases as the pressure of the transported fluid also increases.

The mechanical losses only depended on the driving speed, which was predictable. The power dissipated by friction rests constant with the pressure of the transported fluid and increases significantly with the drive speed of the centrifugal pump.

The coefficient of friction between the two rings of the EMF varies both with the pressure of the circulating fluid and with the speed of the shaft of the centrifugal pump. At speeds lower than 1500 rpm, the coefficient of friction between the rings is decreasing, because at speeds above 1500 rpm it becomes increasing, as in fig. 6. In all cases the friction is fluid or mixed.
The flow of leaks that can occur in the frontal mechanical seal increases both with the pressure of the transported fluid and with the speed of the shaft of the hydraulic machine. Comparable results regarding seal leakage can be found in [7].
Regarding the ratio between the fluid pressure and the sliding pressure, the analyzed seal FMS is balanced in few cases, like in Table 1, and in many cases it is loaded.
Based on the experimental results, we plotted the variation graphs of the friction coefficient depending on the speed $\mu = \mu(n)$ resulting curves for each seal fluid pressure. Such a variation of the friction coefficient as a function of speed has the typical form in fig. 6.

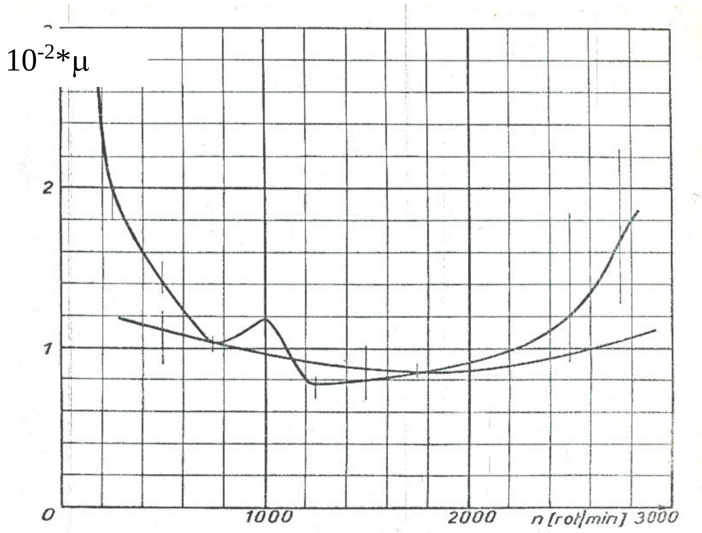

Fig. 6. The variation of the friction coefficient depending on the shaft speed

5. Conclusions

The coefficient of friction between the two rings of the frontal mechanical seal FMS varies both with the pressure of the circulating fluid and with the speed of the shaft of the centrifugal pump. At speeds lower than 1500 rpm, the coefficient of friction between the rings is decreasing, because at speeds above 1500 rpm it becomes increasing. The variation of the friction coefficient depending on the speed is explained by the fact that at low speeds the low friction speed between the two rings does not allow sufficient heating of the graphite ring [8]. It is a poor thermal conductor, leading to its additional superficial expansion compared to its colder depth. This would cause the formation of thermal plumes, creating hydrodynamic lift and therefore the appearance of fluid friction.
On the other hand, at higher speeds, the hydrodynamic frictions caused by the rotation of the mobile ring in the liquid begin to dominate, the so-called ventilation losses, which would require a more hydrodynamic shape of the exterior of the rotating seal while respecting an optimal radial distance from the outer wall of the sealing housing.
The experienced seal being of the maximum pressure 16 bar and the testing being carried out at low pressures, only at the fluid pressure of 2.5 bar, the sliding pressures are equalized with the pressure of the transported fluid, the seal being balanced, at lower pressures the FMS is discharged.

Acknowledgments

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References

[1] EagleBurgmann. "Catalog MG1. Mechanical seals | Mechanical seals for pumps | Elastomer bellows seals." Accessed December 8, 2023. https://www.eagleburgmann.com/en/products.
[2] Budea, Sanda. *Hydraulic measurements and test of hydraulic machines like pumps, fans and blowers / Masurari hidraulice si incercarea unor masini hidraulice din categoria pompe, ventilatoare si suflante.* Bucharest, Printech Publishing House, 2010.
[3] Cazacu, Mircea Dimitrie, and Sanda Budea. *3D fluid flowing of viscous fluids through machines and equipment / Curgeri tridimensionale ale lichidelor vâscoase prin mașini și echipamente.* Bucharest, Printech Publishing House, 2012.
[4] Exarhu, Mihai. *Hydraulic, pneumatic and environmental measurements / Măsurători hidraulice, pneumatice și de mediu.* Bucharest, SC Andor Tipo Publishing House, 2007.
[5] Florea, Julieta, and Valeriu Panaitescu. *Fluid Mechanics / Mecanica Fluidelor.* Bucharest, Didactic and Pedagogical Publishing House, 1979.
[6] Tavoularis, Stavros. *Measurement in Fluid Mechanics*. Cambridge University Press, 2005.
[7] Luo, Yin, Yakun Fan, Yuejiang Han, Weqi Zhang, and Emmanuel Acheaw. "Research on the Dynamic Characteristics of Mechanical Seal under Different Extrusion Fault Degrees." *Processes* 8, no. 9 (2020):1057.
[8] Xiao, Nian, and M. M. Khonsari. "A Review of Mechanical Seals Heat Transfer Augmentation Techniques." *Recent Patents on Mechanical Engineering* 6, no. 2 (2013): 87-96.