DESIGN AND TESTING OF PISTONS AND CUPS FOR LARGE HYDROSTATIC PUMPS AND MOTORS

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ABSTRACT

A new transmission is being designed for a next generation of large, offshore wind turbines, based on floating cup pumps and motors. The machines have a fixed displacement of around five liter per revolution. The objective of this study is to design, manufacture and test the pistons and cups of these machines. To this end, a new test bench has been designed and build, to measure the leakage and friction of the pistons up to a pressure level of 350 bar. Several sets of pistons and cups have been tested against a reference set which was proven to have very little friction at rated and peak operating conditions. The leakage between the pistons and cups was measured at different piston positions at stationary conditions. The friction between the piston and the cup has been measured continuously. From the tests it can be concluded that the friction force is below 0,01% of the piston force. The leakage losses are less than 0,5% of the total effective flow output.

Keywords: Floating Cup, Transmission, Turbine

1. INTRODUCTION

The Dutch company Hydrautrans is developing a transmission for large scale wind turbines. The target of the new design is to combine high efficiency, reduced nacelle weight, increased lifetime, and reduced maintenance costs. A key element of this transmission is a 4919 cc hydrostatic pump or motor that uses floating cup technology [1]. The pistons and cups were considered to be the most critical components in this design. To test these core components, a separate test bench has been designed and build. This new test bench needed to give evidence of the low friction and wear behavior. The test results are also expected to be a good indication of the efficiency of the complete hydrostatic drivetrain components.

2. DESCRIPTION OF THE TEST BENCH

With a diameter of 66 mm, the dimensions of the pistons and cups in the 4919 cc pump are much larger than the dimensions of any floating cup pump tested before. At this size, the deformation of the components as a result of the oil pressure will be much larger than with smaller components. **Figure 1** shows an exaggerated illustration of how the two components will expand. The figure shows that the cup will expand uniformly, while the piston, which is positioned at an angle, will deform to an oval

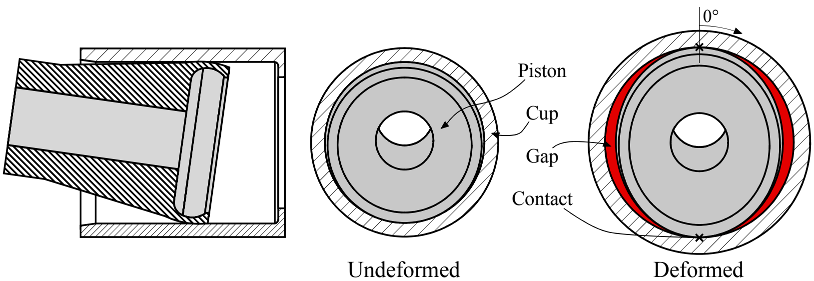

Figure 1: Deformation of the piston and the cup as a result of the oil pressure.

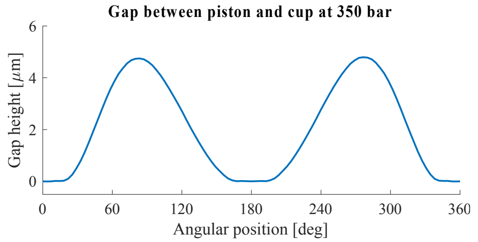

Figure 2: Simulated gap between piston and cup at an oil pressure of 350 bar (0° corresponds with definition in **Figure 1**).

shape. **Figure 2** shows the distance between the two components from a contact simulation of the two components with an oil pressure of 350 bar.

Friction forces

As a result of the oval expansion, the friction between the piston and cup will increase where the piston expands more than the cup (points indicated with ‘x’ in **Figure 1**, and gap height = 0 in **Figure 2**). The friction was expected to be less than 100 N. Since this is merely 0.08% of the hydrostatic forces on the piston at a pressure of 350 bar, the main design challenge was to isolate the friction from the other forces. This was realized by having two sets of pistons and cups being applied in opposite directions, thereby balancing the main piston forces.

The cups are placed back-to-back in a central block that is held in place by a force sensor, while the pistons are connected via a moveable construction around the central block. During experiments, oil is supplied to the pistons and cups under various pressure levels. The pistons can be moved by a linear actuator, while the remaining forces on the central block are being

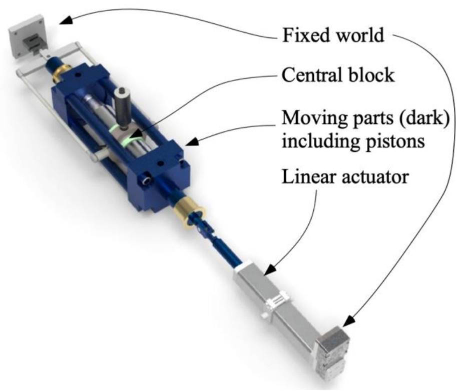

Figure 3: Main components of the test bench.

measured. The main components of the test bench are shown in **Figure 3** and **Figure 4**. Since one of the piston-cup sets will be a reference set with near zero friction, the force that is measured between the central block and the fixed world will be the friction force between the other piston-cup set: the test specimen.

Leakage flow

Another result of the oval expansion of the piston is that a gap forms where the piston expands less than the cup (red areas in **Figure 1**, and gap height > 0 in **Figure 2**). An additional feature of the test bench is that it is designed such that the leak flow at the contact between the piston and cup is separated from the other leakages, as illustrated in the cross-section shown in **Figure 4**. The separated leak flow rate was measured in a separate experiment to exclude the resistance of the flow sensor from the friction force measurements.

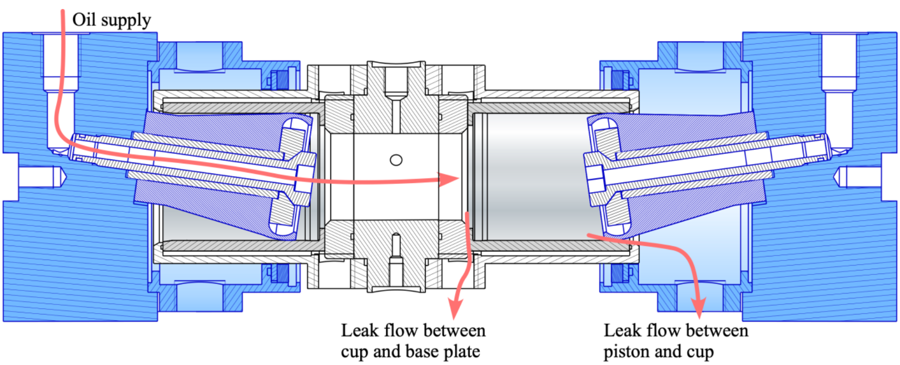

Figure 4: Cross-section of the centre of the test bench (blue parts are moving) with separated leak flow paths.

3. CUPS AND PISTONS

State of the art manufacturing techniques were used to produce two unhardened reference piston-cup sets, and nine hardened piston-cup sets for testing. Each of the components has a diameter of 66 mm, and was manufactured to a precision of less than 3 µm. **Figure 5** shows how the minimum and maximum fitting inside diameter of the cups was measured at five positions. From this figure it can be concluded that the cup had a slightly smaller diameter in the bottom dead center (BDC), which was true for all of the cups. This was an unintentional result from the used production method.

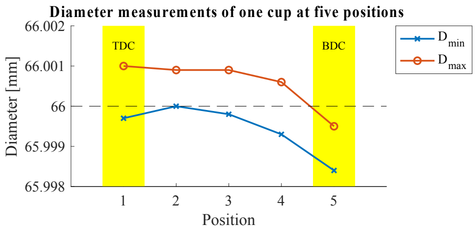

Figure 5: Minimum and maximum fitting diameter measured at five positions in one cup.

Small differences in the diameters of each piston and cup made some sets fit tighter than others. To specify how tightly each piston fits in its cup, the gap between them was defined as the difference between the largest fitting inner diameter of the cup and the smallest fitting diameter around the ball shaped crown of the piston. Since the friction forces will be larger for tighter fitting sets, some of the tightest fits were chosen as the test specimens to measure the worst-case scenario.

By combining pistons and cups from different sets, a total of five sets were selected with decreasing difference in diameter between the piston and the cup: -1.9, -2.7, -3.9, -5.2, and -7.0 µm. Note that all of these sets have a negative diameter difference. This means that the piston crown at some point is larger than the largest fitting inside diameter of the cup. An advantage of negative tolerances is that the deformation of the piston will be less oval, since the expansion of material is limited by the slightly smaller cup. This means the leakage between the piston and cup will decrease, while only slightly increasing the contact friction forces.

4. MEASUREMENT DESCRIPTION

Two types of measurements were done:

1. dynamic, measuring the friction forces, and
2. static, measuring the leak flow.

For each piston-cup set, all measurements were done at a pressure level of 50 to 350 bar in steps of 50 bar, using Shell Tellus 46 oil at a measured temperature between 49.5°C and 50.5°C.

4.1. Positioning

The pistons in the design of the full 4919 cc machine will make a stroke of 51.5 mm. The testbench was set up such that the pistons would make a stroke of 54.0 mm, to make sure the extreme positions were included. The position of the piston inside the cup was defined as the distance to the bottom of the cup, with 0 being the top dead center (TDC) and 54.0 mm the BDC. The position was measured at the motor of the linear actuator.

4.2. Friction forces

The friction between the piston and the cup has been determined by measuring the forces acting on the central block. During these measurements, the chambers surrounding the cups to collect the leaked oil were removed on both sides of the test bench to exclude the friction between these chambers and the central block from the measurements.

Figure 6 shows the position profile that was used for the friction tests. the actuator moved the pistons back and forth for two full cycles. This is done at a speed of 0.010 m/s, with an acceleration and deceleration of 0.1 m/s^2 at the turning points, resulting in a frequency of 0.091 Hz. Since the speed of the pistons in the turbine will be much higher, this is considered the worst-case scenario

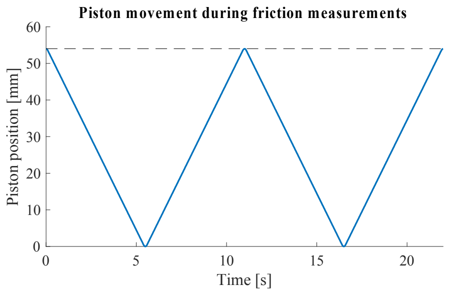

Figure 6: Piston position during the friction tests.

when determining the friction forces, as there is no hydrodynamic friction yet.

Leak flow

The leakage of oil between the piston and the cup has been determined by measuring the flow rate during static measurements at incremental piston positions. In these static measurements, the linear actuator moved the piston block from position 0 to 54.0 mm in steps of 6.0 mm. At each position, the flow sensor output during a period of 10 seconds was stored, the average of which is shown in the results section.

5. MEASUREMENT RESULTS

5.1. Friction force

Reference sets

During a dynamic measurement, the force sensor measures the combined friction of both piston-cup sets. If one of the sets is known to have very little friction, the measured force is more or less equal to the friction of only the other set. To this end, the two reference sets were measured first. **Figure 7** shows the measured forces at four different pressure levels.

The reference force measurements shown in **Figure 7** clearly have an offset at all pressures. This is caused by a slight difference in area between the two piston-cup sets that are placed opposite to each other in the test bench (a difference of 0.01% between the two piston surface areas will already result in 12 N at 350 bar). Assuming that the friction is equal (but switched sign) in both directions, this offset equals the average value of the measured force during the two full cycles. **Figure 8** shows the same results, but after subtracting these average values. This correction seems to remove the offset properly, as the forces are now close to mirrored around the 0 N axis.

The results in **Figure 8** show that while there is very little friction force at lower pressure (up to 8 N), there is almost no friction in the reference sets when operating at higher pressures (less than 2 N for most part of the stroke). This force was found to be negligible with respect to hydrostatic forces acting on the pistons. Please consider that during the measurements with the hardened sets, the measured friction force will be the combined friction of this test subject and one of the reference sets.

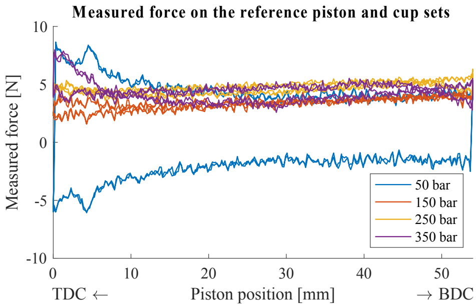

Figure 7: Measured force during dynamic measurement using both reference sets.

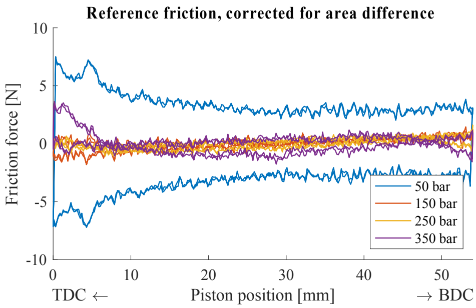

Figure 8: Friction forces of the reference sets after correcting for area differences.

Test specimens

The results of the dynamic measurements using the hardened sets, at a pressure of 50 and 300 bar are shown in **Figure 9**. Similar to the reference results, these results were first corrected by the average value of the measured force to account for the shape differences.

Overall, **Figure 9** shows relatively small amounts of friction during most of the stroke. At 50 bar, the maximum friction of each set was measured when the piston was in BDC. The tightest fitting set (-7.0 µm) measured just under 80 N of friction force under these conditions. This is in accordance with the slightly smaller diameter of the cups shown in **Figure 5**. During most part of the stroke however, the friction was less than 10 N for each set.

At 300 bar, the maximum friction was measured when the piston was in TDC. This side of the cup contains an additional edge on which the cup stands. Due to this extra material, the stiffness of the cup is higher on this side, making

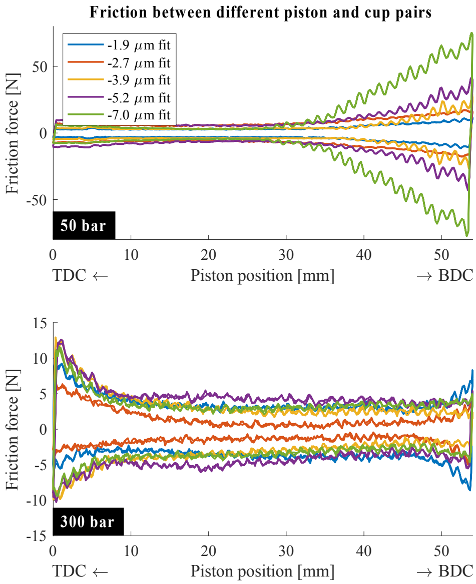

Figure 9: Friction force between the different piston-cup sets at two pressure levels.

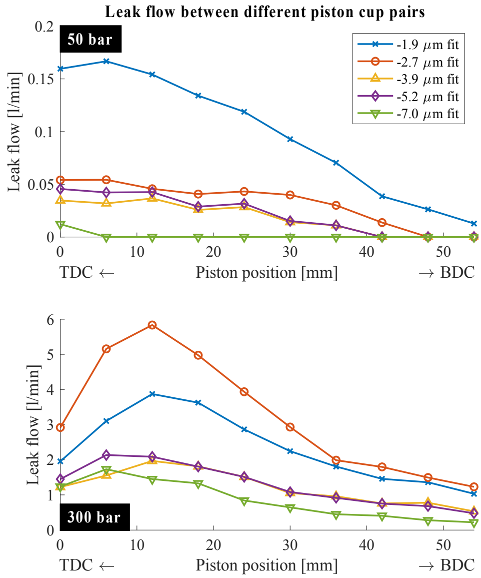

Figure 10: Leak flow between the different piston-cup sets at two pressure levels.

it less sensitive to pressure caused deformations in the TDC. In other words, the cup does not expand as much as the piston does in the TDC, resulting in a contact between the piston and the cup and thus an increased friction force. The maximum measured friction force under these conditions was less than 13 N for all pairs. During most part of the stroke however, the friction was measured to be roughly 5 N or less.

5.2. Leak flow

The results of the static incremental leakage experiments at a pressure of 50 and 300 bar, are shown in **Figure 10**. In this figure, we see that the flow was generally lowest when the piston was in the BDC position, and increased as the pistons moved towards the TDC. The results at 300 bar show that the leak flow was highest at roughly one third of the stroke, and this was found to be true for all sets at pressures above 150 bar.

This leak flow pattern is again in accordance with the shape seen in **Figure 5**, as well as with the results of the friction measurements; at low pressures the cups are tighter near the BDC, and at higher pressures the cups are tighter near the TDC due to different deformation behavior between the piston and the cup.

6. PERFORMANCE PREDICTIONS

To predict the effect of the measured friction and leak flow on the performance of a full 4919 cc pump or motor, the measurement results needed to be scaled up. This was done by accounting for the sinusoidal stroke of a piston, and, in the case of leakage, multiplying by the total number of pistons. These predictions are discussed around 277 bar, which is the nominal operating pressure for the wind turbine application.

6.1. Friction losses

To overcome the friction force, a small portion of the piston force will no longer be pushing the piston. Therefore, the estimated loss due to friction is quantified by ϵ_f:

$$\epsilon_f = \frac{F_f}{pA} \tag{1}$$

in which F_f is the average friction force during a stroke, p is the pressure level, and A is the surface area of the piston.

The estimated friction loss for the different piston-cup sets is shown in **Figure 11**, which shows the full pressure field, as well as a close-up around the nominal operating pressure. The figure shows that even at low operating pressures, where friction forces were found to be highest, the friction is never more than 0.15% of the piston force. Looking at the losses around the nominal operating pressure, the friction losses are estimated to be less than 0.006% during a full stroke. It can thus be concluded that the friction forces are negligible.

6.2. Leakage losses

Leak flow will reduce the amount of oil displaced by the pump or motor. Therefore, the estimated loss due to leakage is quantified by ϵ_l:

$$\epsilon_l = \frac{z\,Q_l}{n\,V_g} \tag{2}$$

in which z is the number of pistons, Q_l is the average leak flow rate during a stroke of a single piston, n is the rotational speed of the machine, and V_g is the geometrical displacement of the machine per revolution. At nominal operating conditions, the motors will rotate at four times the speed of the pumps, due to a 4:1 transmission ratio. Therefore, a full pump is estimated to have four times as much performance loss as a result of leakage, when compared to a full motor.

The estimated leakage losses for the different piston-cup sets are shown in **Figure 12**. The figure shows these losses both when operated as a pump (left axis) and as a motor (right axis). A pump at nominal operation pressure is expected to have a 1.8% performance loss for the loosest fit, while the tightest fit is expected to lose 0.5%. Although these losses are also not very large, they cannot be neglected like the friction losses.

7. IMPROVED DESIGN

To decrease the expected loss in performance, the design of the piston was slightly altered. Since the deformation behavior of the piston and the cup under higher pressures was found to be an important factor, the new design allowed the piston crown to better follow the deformation of the cup as well as any inconsistencies in the internal diameter. This was realized by removing some material on the inside of the piston crown, as shown in **Figure 13**.

The improved design was tested on the piston of the tightest fitting set. **Figure 14** shows that this improved design reduced the leak flow, especially at higher pressure levels. The expected leakage loss was reduced to around 0.32% at nominal operating pressure for pump operation, and 0.08% for motor operation.

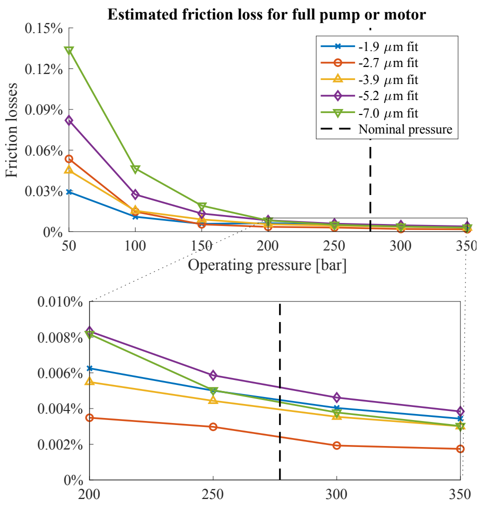

Figure 11: Estimated performance loss due to friction ($\epsilon_f \cdot 100\%$), for a full pump or motor.

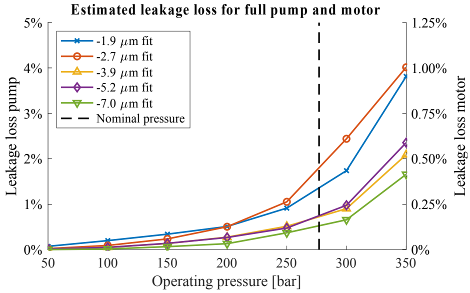

Figure 12: Estimated performance loss due to leakage ($\epsilon_l \cdot 100\%$), for a full pump and motor.

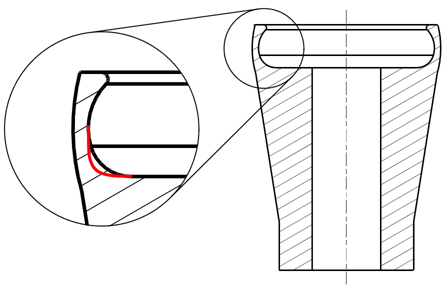

Figure 13: Removed material from the inside of the piston crown to reduce its stiffness.

The estimated performance loss due to friction as a result of this improved design is shown in **Figure 15**. This figure shows the friction loss reduced significantly at pressures below 200 bar for the improved design. The close-up in **Figure 15** shows that the improved design did introduce some additional friction losses near and above the nominal operation pressure. However, the friction loss is still found to be negligibly small for this tightly fitting set.

The results suggest that the leakage losses can be further reduced by using an even tighter fitting piston-cup combination or removing more material from the inside of the piston crown. The effect of these design changes on the friction and evident wear of the components could be the topic of future research. Durability measurements should furthermore show how the leakage patterns evolve over time.

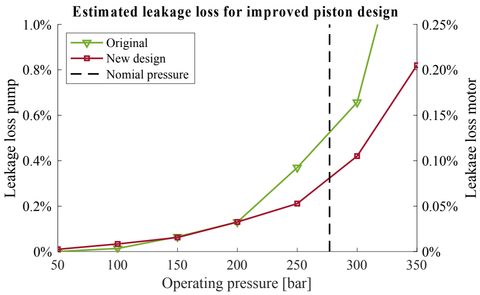

Figure 14: Comparison between expected leakage loss for the -7.0 μm fitting piston of the original design and the improved design.

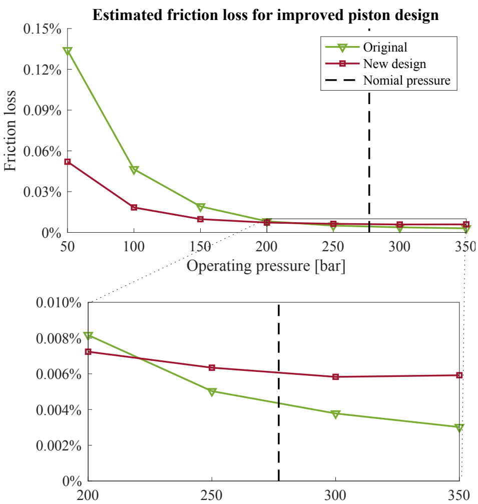

Figure 15: Comparison between expected friction loss for the -7.0 μm fitting piston of the original design and the improved design.

8. CONCLUSION

The new test bench for individual pistons and cups of a 4919 cc floating cup pump and motor was created and used on five sets of pistons and cups. On each of the sets, a series of dynamic and static tests were conducted to measure the friction and the leak flow at the contact between the piston and the cup at different operating conditions. In the dynamic measurements, the piston moved back and forth over more than the length of a full stroke to determine which parts of the stroke will cause most friction during pump or motor operation. In the static measurements, the leak flow was measured at incremental positions of the piston inside the cup to determine how much the leakage is influenced by the different deformations of the two components.

The results show that even for sets with a very tight fit, the friction force between the piston and cup was very low and the performance loss due to friction was found to be negligible. The performance loss due to leakage between the piston and cup however, was not found to be negligible. The leakage was estimated to account for up to 1.8% of the total output flow of a full pump for the loosest fitting piston-cup set and 0.52% for the tightest fitting set.

The friction force and leak flow both depended on the part of the stroke the piston was in. This dependency was found to be partly explained by the shape of the produced cups, and partly by a difference the deformation behavior between the piston and the cup.

To improve the pistons ability to follow the shape of the cup, an improved piston design was proposed. In this design, some material was removed from the inside of the piston crown such that it would become less stiff. The new design was tested on the piston of the tightest fitting set. For this new design, the estimated performance loss due to leakage between the piston and the cup decreased from 0.52% to 0.32%, while the friction losses remained negligible.

Future research could focus on finding the optimum design with respect to leakage and friction, while also measuring the effect of wear by means of durability tests.

NOMENCLATURE

ϵ_f	Performance loss due to friction	[-]
ϵ_l	Performance loss due to leakage	[-]
A	Surface area of piston	[m^2]
F_f	Friction force	[N]
n	Rotational speed	[1/s]
p	Operating pressure	[Pa]
Q_l	Leak flow rate	[m^3/s]
V_g	Displacement volume per rotation	[m^3]
z	Number of pistons	[-]

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