ISSN 1392–1320 MATERIALS SCIENCE (MEDŽIAGOTYRA). Vol. 9, No. 1. 2003

Complementary Dynamic Tests for Judgement of Rotary Shaft Lip Type Seals

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Received 30 October 2002; accepted 04 February 2003

A comparison test of two rotary shaft seals variants of a gear pump is presented. The test parameters are chosen in a way, that differences on dynamical qualities as well as on wear behaviour can be judged. Two different tests have been performed. At the first test, severe test conditions have been set, partly based on the material limits, in order to reduce the test time. At the second test the test parameters are closer to the actual practical conditions. The results are discussed and further assessment criterions are set. The tests show that an unequivocal decision can be taken in favour of a variant. The bench test shows to be a suitable method for comparing two different rotary shaft seal variants. For these tests, the existence of a comparison seal is important, so that for a new construction the test can only be used as a complement.

Keywords: rotary lip seal, shaft, radial force, leakage.

1. INTRODUCTION

The task of producing economic machines and units requires the consideration of each machine element. The question often arises, if it is feasible to replace an element by other variant or by apparently the same element of other manufacturer.

At first glance, this task doesn't seem to be complicated, but depending on the machine element it may mean a considerable deal of research. This is especially the case in radial shaft seals [1 – 3].

Shaft seals are very complex machine elements, not necessarily due to the geometry or the amount of parts, but because of the behaviour at different operating conditions and the not yet totally known way of working [4 – 6].

Even at apparent identical shaft seals of different manufacturers, we cannot suppose that they will have the same behaviour at work.

Taking on account these facts, the convenience of a test of comparison in the event of changing the sealing type or manufacturer is evident.

2. TEST CONDITIONS

2.1. Fixing the test and the test temperature

The first decision to take is, if seal testing will be made direct in the machine or on a test bench. Bench testing is often convenient because of timesaving and lower costs.

The question arises, if the results obtained on a test bench, comparing the measurements of the old and the new seal variant, are revealing enough.

A test of comparison for two seal types is carried out on a test bench and the results are analysed and presented.

For lying down the test conditions, the working conditions in practical operation are taken into account and not the operational limits of the components.

Exemplary the test temperature has been worked out. If the maximal temperature in practical operation is 80 °C and the operational limits of the two seal types are 100°C and 120 °C, it is evident that only temperatures under 100 °C should be applied. The fact, that one seal variant can resist higher temperatures than the other is meaningless for the case we want to test. However, the temperature range between 80 °C and 100 °C should be used to accelerate the aging process of the seal.

Fixing the test conditions involves dealing with several contrasting processes. On the one hand we want a short test time, but on the other hand the test parameters must not be to far from the actual practical conditions [5].

2.2. Example: Gear Pump

A gear pump has been equipped several years with a rotary shaft seal variant. This working variant shall be replaced by an economic variant from other manufacturer due to cost reasons.

The task consists in comparing both seal variants at a test bench, thus settling the question, if the new variant has the same or better qualities as the original one.

2.3. Fixing the test conditions

In this case not only a comparison of both seal variants will be made, but also analysing the effects of different test conditions on the qualitative information output of the test is needed.

Two different tests have been carried out. One test has been performed with more adverse test conditions. Here the test temperature was set according to the material limits, and in addition the dynamic loadings of the seal have been set higher than the expected in the actual practical conditions (Table 1).

Table 1. Parameter of the first test

Parameter	Unit	Value
rotational speed (n) and time	min^{-1}	500, 1000, 2000, 3000 each 250 h starting with 500
lower test pressure	bar	1 absolute
higher test pressure	bar	4 absolute
time at higher pressure	s	5
time at lower pressure	s	5
static eccentricity	mm	0.15
dynamic eccentricity	mm	0.18

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Continuation of **Table 1**

test medium		hydraulic oil Shell Tellus Oil 46
test temperature		according to DIN 3761 Part 10, Part 2; maximal test temp. 110 °C, middle test temp. 90 °C
combination of rotational speed, temperature and time		according to DIN 3761 Part 10; A test cycle is 24 h: 14 h at 90 °C and n 6 h at 110 °C and n 4 h shaft rest period and cooling to room temperature

The second test was modelled on the actual expected practical conditions, setting the test parameters a bit stronger in order to achieve an aging acceleration. A crucial difference between the test parameters (Table 2) are the different cycles on the rotational speeds.

Table 2. Parameter of the second test

Parameter	Unit	Value		
rotational speed (n) and time	min^{-1}	3000, 2000, 1000, each 24 h starting with 3000, 10 weeks		
lower test pressure	bar	1 absolute		
higher test pressure	bar	4 absolute		
time at higher pressure	s	3.5 – 3.6		
time at lower pressure	s	3.5 – 3.6		
		4 shafts	2 shafts	2 shafts
static eccentricity	mm	0.24	0.27	0.33
dynamic eccentricity	mm	0.09	0.06	0.00-0.01
test medium		hydraulic oil Shell Tellus Oil 46		
test temperature		according to DIN 3761 Part 10, Part 2; maximal test temp. 110 °C, middle test temp. 90 °C		
combination of rotational speed, temperature and time		according to DIN 3761 Part 10; A test cycle is 24 h 14 h at 80°C and *n* 6 h at 100°C and *n* 4 h shaft rest period and cooling to room temperature		

2.4. The test bench

The test bench is shown in Figure 1 and 2.

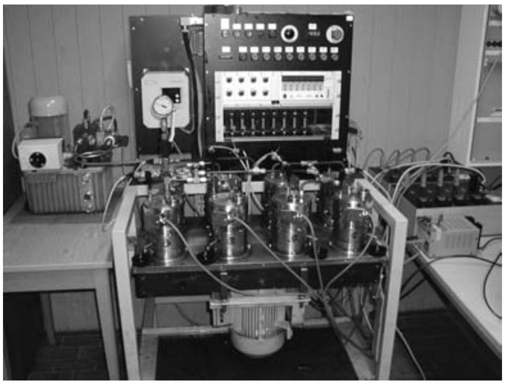

Fig. 1. Front view of the test bench. The wear test bench has 8 work places. The pressure unit is on the left, on the right site is the heating device at the back and the control unit in front. The test chambers are heated with heating collars

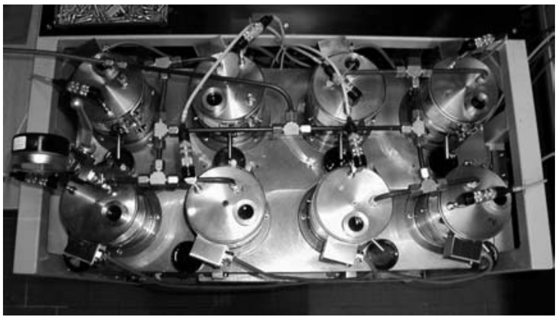

Fig. 2. Top view of the test bench. All test chambers are connected with hydraulic ducts that are centrally communicated with the pressure unit. The chambers are completely filled with oil. There is no oil exchange between the chambers. The pressure unit only serves to build up pressure in the chambers. The leakage is collected in the black vessels and weighed out

The rotary shaft seals are pressed in the housing. The oil temperature is controlled by means of thermoelements and external heating collars. The oil pressure is induced by the pressure unit. The load alternation is managed by a control module that controls the pressure unit. The leakage is collected underneath the seal by a collecting plate and lead to a vessel. The collecting plates have been moistened with oil before the test. The position of the shafts is vertical. To determinate the quantity of oil that remains in the way to the collecting vessels, or rather to establish the precision of the results, one gram of oil has been placed on the collecting plate during five consecutive days. The collecting vessel is weighed with a time delay of 24 hours. The remaining amount of oil in the way to the vessel is 5 g. This accuracy test has been carried out at the actual test temperature [5].

2.5. Carrying out the test

The radial force F_R of each seal is measured before the test by means of the digital radial force measuring instrument of DIN 3761 (Radiameter).

On the first test the radial force has been recorded for 30 minutes, on the following tests only for 25 min. Afterwards the seals are fitted on the testing bench and the test is performed according to the schedule. The leakage is periodically weighed and recorded.

The radial force is measured again after the test. For disassembly it is necessary to wait, till the test chambers reach room temperature. The seals are then removed together with the test shaft. Immediately after removing the test shaft the seal is inserted on the plug gauge of the Radiameter and radial force is measured.

After this measurement the seals are stored without the shafts in an oil-bath, and after two days a new measurement of the radial force is made.

After this measurement the spiral spring is removed and the seals placed again in oil for one day. The next measurement is made.

3. OUTPUT OF THE INVESTIGATION

3.1. First test

The tightened-up test conditions of Table 1 are set. The failure criterion is set for a leakage of 20 g.

3.1.1. Leakage

The type A seals have failed already after 3 cycles (3×24 h). There was not significant attrition on these seals.

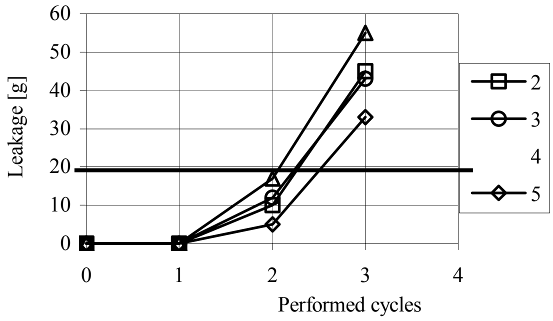

Fig. 3. Progression of leakage until removal – Type A

Figure 3 shows clearly that all seals of one type failed almost at the same time. It can be supposed that all seals of a type failed due to the same cause. Taking in consideration the leakage values, variant A has a clearly inferior performance to variant B.

Next the radial force at different times is analysed.

3.1.2. Radial force

The radial force of 7 seals of type A and 6 of type B has been measured.

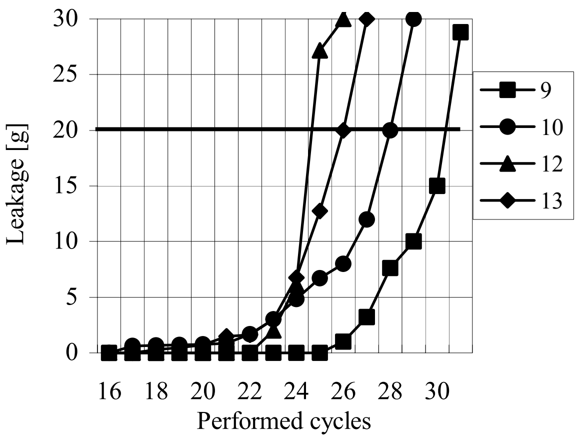

Fig. 4. Progression of leakage until the 20 g limit – Type B

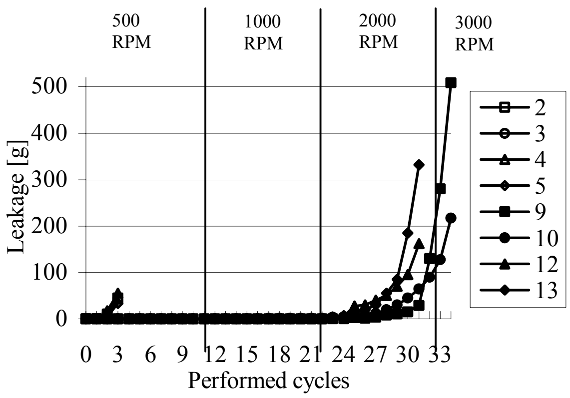

Fig. 5. Leakage over the entire test time – Type A and Type B

Figure 4 shows distinctly the difference between the two variants. The radial force of type A (seals 1 to 7) is clearly under type B (seals 8 to 13) and the values are less spread. The following listed seals are used on the test: Type A: seals 2, 3, 4, 5; Type B: seals 9, 10, 12, 13.

Figure 5 shows the radial force that acts on the shaft on the moment of failure (exceeding of the 20 g limit).

Both variants have at this moment similar radial forces. It is true that the radial force of variant B is a bit higher than A, but we have to take on account that this type has been working at a rotational speed of 2000 rpm supporting higher dynamical loadings. At a speed of 500 rpm the variant B would have been working even longer and the radial forces would be even closer.

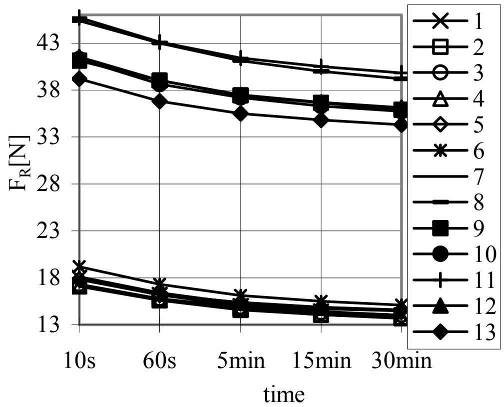

Fig. 6. Radial force of the new seals – Type A and B

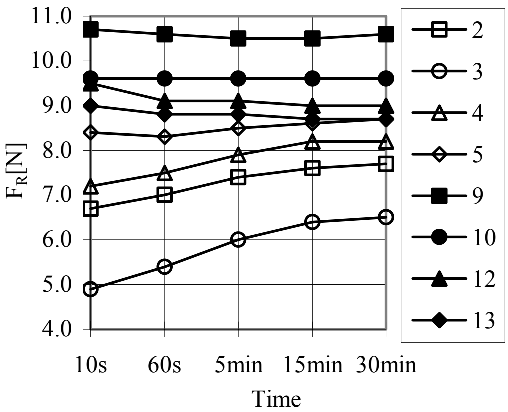

Fig. 7. Radial force immediately after removing the shaft

Figures 6 and 7 show that the spring contribution on radial force is much higher on type B than on type A. On variant A the spring contribution is almost negligible. This is also the reason, why, compared with type B, the radial force of type A is lower on Figure 5 and higher on Figure 6.

3.1.3. Discussion

Besides an only insignificant wear, there are almost no changes on type A. We can suppose, that the seal could not follow the dynamic eccentricity of the shaft. The reason for this is the altogether weaker radial force of type A, having both seals similar geometry, as well as the insignificant spring contribution on type A.

On type B there are clear wear traces due to longer test time. At failure moment the spring contribution on radial force is about one third of the total (Figure 6 and 7).

Since the spring contribution is higher on type B, we have to assume that the difference on the radial force whilst working is bigger than the shown on Figure 5. With rising temperatures the radial force will decrease more on type A than on type B.

On the first test, there is no cyclic change between the lower and the higher rotational speeds. It starts with the lowest speed and after 250 h the following is set. Under these conditions shaft speed and seal wear increases simultaneously and it is difficult to differentiate between failure due to wear or due to the seal lip incapacity of following the dynamic rotating shaft. For this reason, the rotational speeds on the second test schedule have been changed. Now the speed changes daily between the high and the low value, so that it is possible to differentiate between the two failure reasons: wear or dynamical deficiency.

3.1.4. Modified variant A seal

Due to the premature failure, the variant A has been modified and tested again at the strong test conditions of the first test schedule. Eight seals have been tested. Figure 8 and 9 show clearly a large increasing of the radial force due to the modifications.

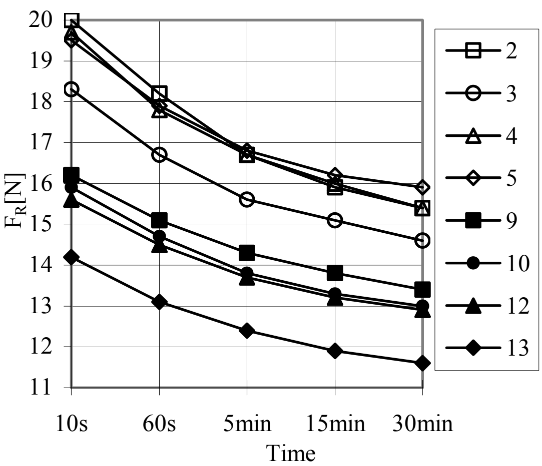

Fig. 8. Radial force two days after removing the shaft – Type A and Type B

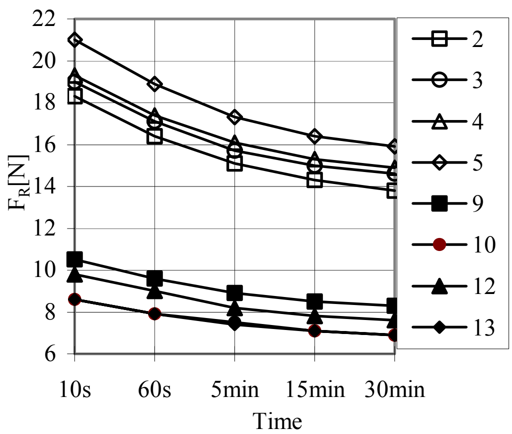

Fig. 9. Radial force one day after removing the spring – Type A and Type B

However, after 3 to 4 days, at a speed of 500 rpm, the seals have again unacceptable leakage values (Figure 10).

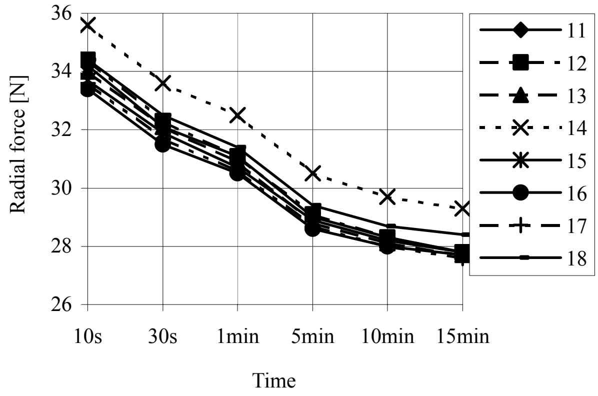

Fig. 10. Radial force of the new seal – Type A modified

Following this test with severe test conditions, a second comparison test (under lighter conditions, Table 2) has been carried out with the modified variant A and the variant B (Figure 11, 12).

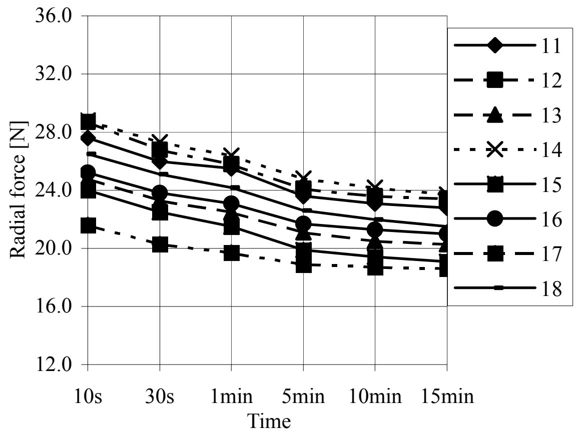

Fig. 11. Radial force two days after removing the shaft – Type A modified

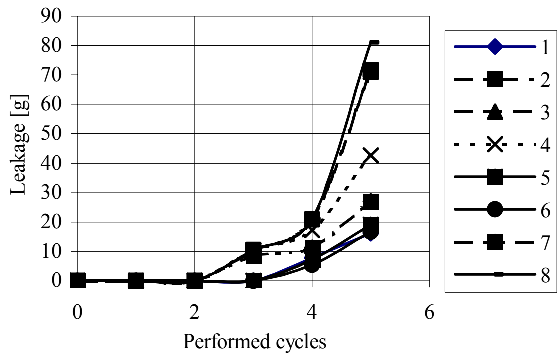

Fig. 12. Progression of leakage –Type A modified- Test parameters Table 1

3.2. Second test

The test parameters listed on Table 2 (more favourable conditions) have been set.

3.2.1. Leakage

Variant A begins to leak relatively early compared with variant B. Even the two seals working with a dynamic eccentricity of 0.06 mm have slight leakage from the beginning on, after 4 weeks the amount of leakage is 8 g and 17 g. (Figures 13, 14).

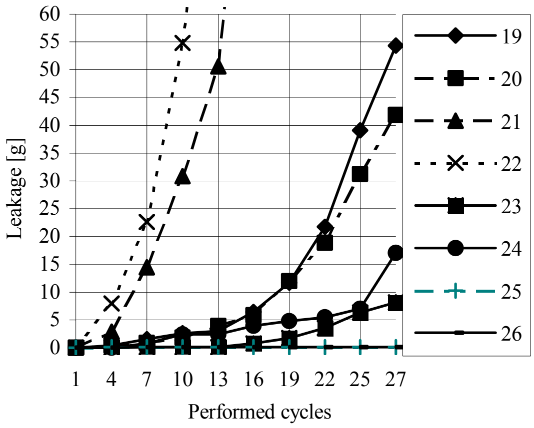

Fig. 13. Leakage progression –Type A modified

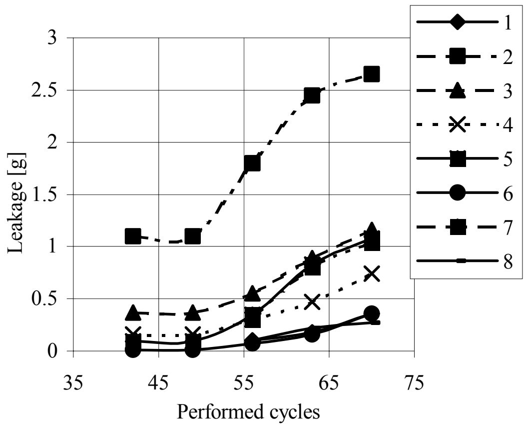

Fig. 14. Leakage progression.-Type B

3.2.2. Radial force

Comparing the absolute leakage values of both variants shown in each figures (Fig. 13 and 14) there are no decisive differences in comparison with the first test. There is still a difference on the spring contribution of the radial force. On the type A the spring contribution is slighter (Figures 15 – 22).

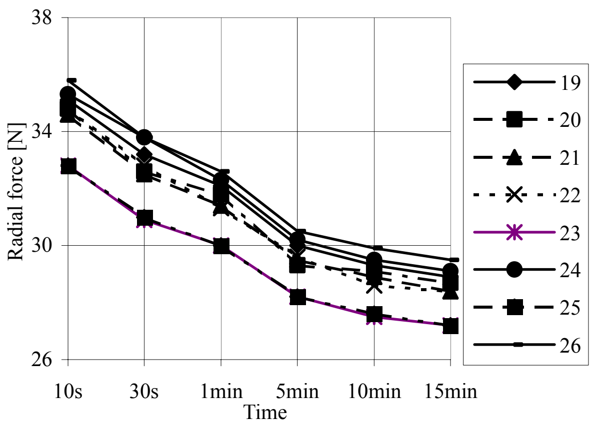

Fig. 15. Radial force of the new seals – Type A modified

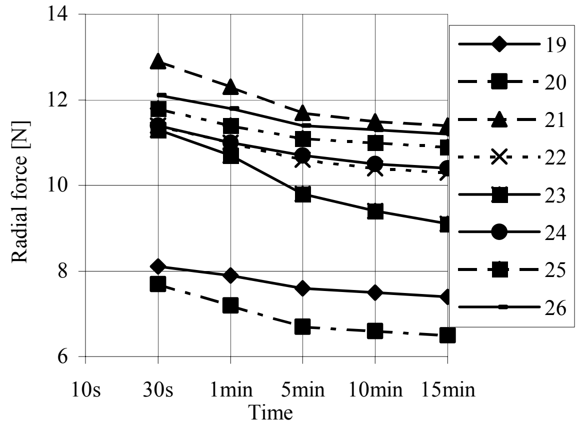

Fig. 16. Radial force immediately after removing

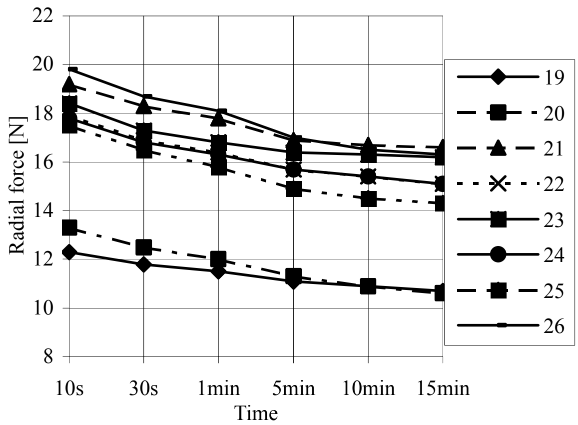

Fig. 17. Radial force two days after removing the shaft – Type A modified

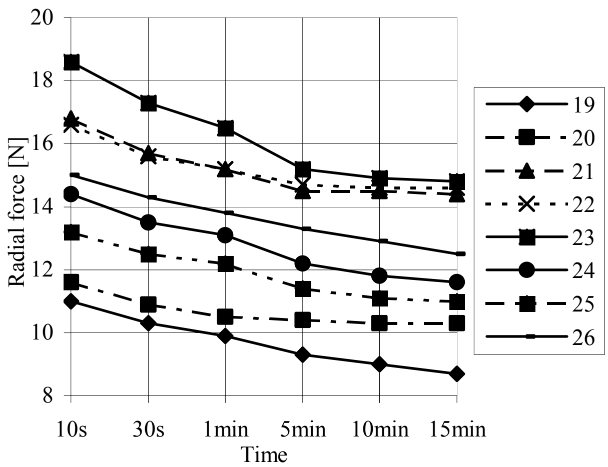

Fig. 18. Radial force one day after removing the spring – Type A modified

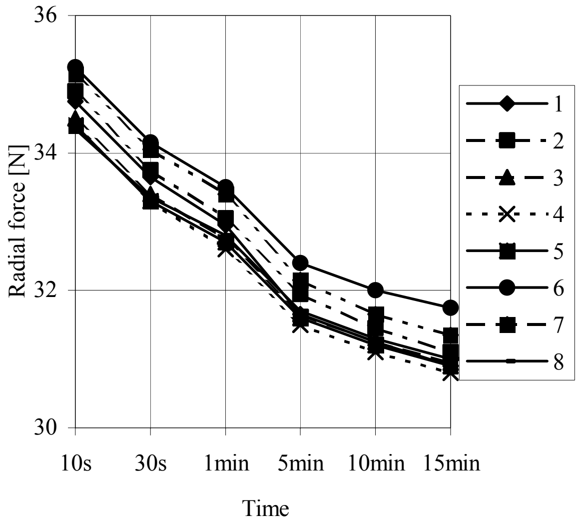

Fig. 19. Radial force of the new seals – Type B

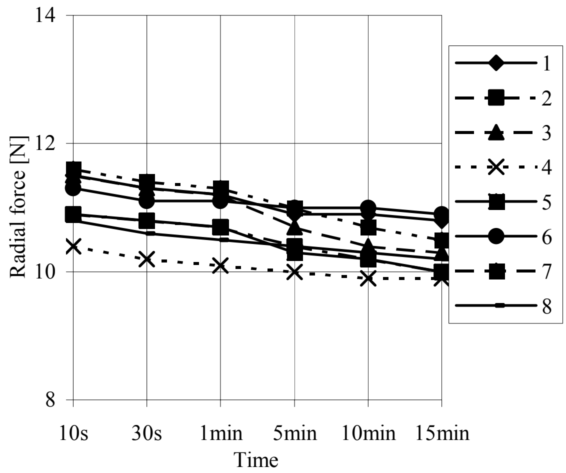

Fig. 20. Radial force immediately after removing the shaft – Type B

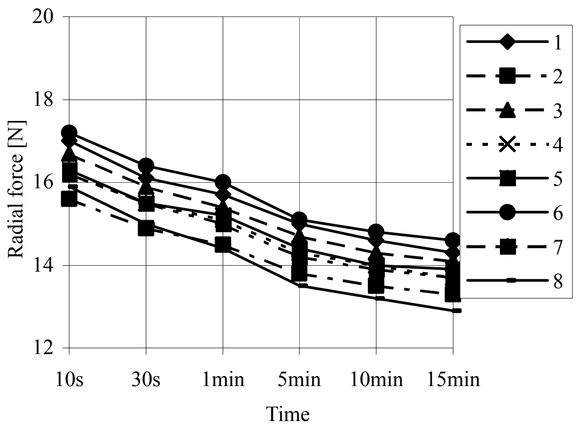

Fig. 21. Radial force two days after removing the shaft – Type B

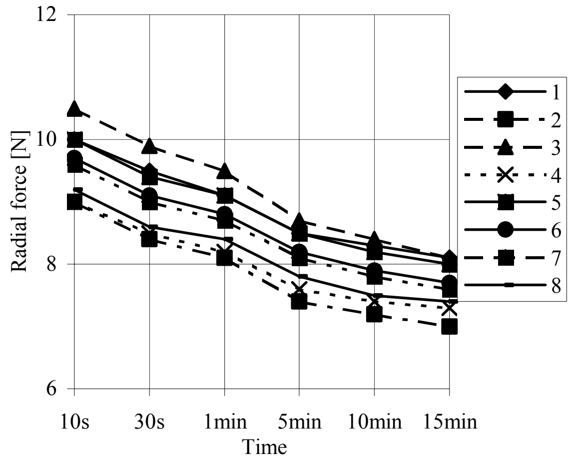

Fig. 22. Radial force one day after removing the spring – Type B

3.3. Discussion of both tests

On the tests it is noticeable that the seals of a variant, working at the same conditions, failed almost at the same time. Further, there are clear differences, between the variants, especially at severe dynamical conditions. although variant A improved its performance it could not equal variant B.

The tests have shown, that the values of radial force measurements give in part a very important clue for the clarification and understanding, however, as seen in variant A after the modification, it can suggest a performance equality that actually does not exist.

Only after the actual test until the failure and the following radial force measurements we have good values for the comparison of different variants.

Table 3. Clues for the test parameters

Parameter		
rotational speed (n) and time	0.5 × n_{max}; n_{max}, changing daily Test duration: Comparison test lasts till the failure of at least one variant. Two different speeds: Makes it possible to differentiate between failure due to wear and failure due to insufficient dynamical properties.	
lower test pressure	usual pressure at practical operation (1bar absolute)	
higher test pressure	maximal working pressure or the permissible pressure of the weaker variant (to test above the allowed pressure of the seal does not make sense)	
time at higher pressure	2.5 s – 3 s	The short times only make sense, if in practical operation the pressure appearance is short-timed as well. The pressure is a crucial load for the amount of energy lead into the sealing zone.
time at lower pressure	2.5 s – 3 s	

Continuation of **Table 3**

static eccentricity	Maximal value at practical operation (consisting of a constant and a variable part). The variable part should be treated partially as dynamic eccentricity. To better judge the influence of the static and dynamic eccentricity on the leakage, different combinations of these eccentricities should be set, letting the other parameters unchanged.
dynamic eccentricity	according to practical application
test medium	as in practical application
test temperature	maximal test temperature (T_{max}): 10° above the maximal working temperature (consider material limits) middle test temperature (T_{middle}): 10° below the maximal working temperature
combination of rotational speed, temperature and time	according to DIN 3761 Part 10; A test cycle is 24 h 14 h at T_{middle} and n 6 h at T_{max} and n 4 h shaft rest period and cooling to room temperature

The test is in position to differentiate between failure due to wear and failure due to insufficient dynamic properties.

An other criterion is the run trace width on the seal. This criterion takes effect if both seals fail due to wear. Since variant A failed relatively early and there are different failure reasons, this criterion does not take effect.

The limits of the test parameters shall be set according to practical operation. To reduce test time, some test parameters are intensified. However, material limits of the weaker variant shall be taken into account for intensification.

On Table 3 some clues for test parameters, of the rotary shaft seals that have been dealt with in this project, are exemplary shown.

4. CONCLUSIONS

The dynamical tests have shown, that a good comparison of two variants is possible. Not only a "yes-no" result is obtained. It is also possible to see in which properties one variant is inferior to the other. The dominant loads, expected in the practical operation, will be simulated on the test bench for obtaining accurate results. A relatively sure answer to the question, which variant is more suitable for the operation in the respective machine, seems to be possible. However, it seems to be difficult, if not even impossible, to predict the working life of a specific seal variant in practical operation in a machine.

The tests have also shown, that it is difficult to set an absolute test time. A comparison test will run until the weakest variant fails.

It follows that such a test for the enabling of a single variant is helpful, however, it does not replace the test on the actual machine.

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