Experimental investigation of the rim-lip-seal of a double regulated STRAFLO Kaplan-turbine under extreme conditions

Schiffer J.*, Benigni H., Jaberg H.
Institute for Hydraulic Fluidmachinery, TU-Graz
Kopernikusgasse 24/IV
8010 Graz, AUSTRIA

Wessiak M., Mayrhuber J.
Verbund Hydro Power AG
Europaplatz 2
1150 Vienna, AUSTRIA

* Corresponding author: juergen.schiffer@tugraz.at

Introduction

The STRAight-FLOw- or STRAFLO-turbine stands for a Kaplan-based turbine-concept where – similar to conventional bulb turbines – the flow passes in horizontal direction. The main difference between a STRAFLO-turbine and a conventional bulb turbine is that the generator is mounted to an external outer ring connecting the tips of the turbine blades instead of being located in the bulb of the machine. On the one hand, this constructive feature results in a number of structural advantages as well as in an increased flow cross-section. On the other hand, it requires an alternative solution for the sealing between the rotating external runner ring and the generator cavity, which is closely connected to several challenging problems. As fig. 1 [1], showing the water inlet (1), the guide vanes, (2), the rotor blades (3), the ring generator (4), the seals (5), and draft tube (6), demonstrates by way of example the STRAFLO-turbine concept requires one sealing system which is placed on the high pressure side and one on the low pressure side of the rotating external ring.

The very first STRAFLO-units were installed in Europe during the 1940ies. These units were small in size having a diameter of only one or two meters and a few megawatts in capacity. The realization of larger sizes was restricted by the use of an appropriate mechanical seal between the water passage and the generator cavity. [2]

Especially in the case of single regulated turbines with variable guidevane position and fixed runner blades the STRAFLO-concept has proved its worth. This may be exemplarily shown with the operational experience drawn from the worldwide largest STRAFLO-based power plant located in Laufenburg (Germany, river Rhein) having a total number of ten units and a maximum capacity of 100 MW. The power station is in operation since 1987 and is running with a higher power output and a higher durability of the sealing system than usually expected. [3]

However, the regulation of the turbine-blades for large units brings additional challenges. Large outer diameters result in a comparatively high circumferential velocity, and the runner blades cause rotating pressure peaks loading the sealing. Furthermore, the axial load of the machine yields an axial displacement of the runner in the order of a few millimetres. Taking into account all these facts makes it almost indispensable to refer to a flexible lip-seal. Additionally, flood waters lead to increased bed load, and the sand carried with the flow is forced into the sealing area by the high centrifugal forces present. According to experience, this scenario yields exceedingly fast damage of the sealing.

In order to cope with those challenges revised concepts of the STRAFLO-turbine were presented in the late 1980ies. The sealing-system was adapted to the special requirements connected with the double regulation of the turbine. It consists of a flexible annular lip seal made of rubber or similar material which is pressed against a comparably hard counterpart made of ceramic. An example of this construction is shown with fig. 2.

Finally the first double regulated STRAFLO-project was realized at HPP Weinzödl (AUSTRIA, river Mur) operated by VERBUND Hydro Power AG. Since 1982 two parallel STRAFLO-Turbines with a diameter of D=3700mm and a capacity of 8 MW per unit are in operation. [4]

Although the new sealing concept was experimentally tested before taking it into operation it caused frequent problems, especially during the occurrence of flood waters. Since the very first year of operation sealing system and materials were changed several times. Anyway, single sealing rings had to be changed at least once a year due to massive abrasive wear. The subsequent breakdown of the machine causes production downtime and thus considerable financial damage.

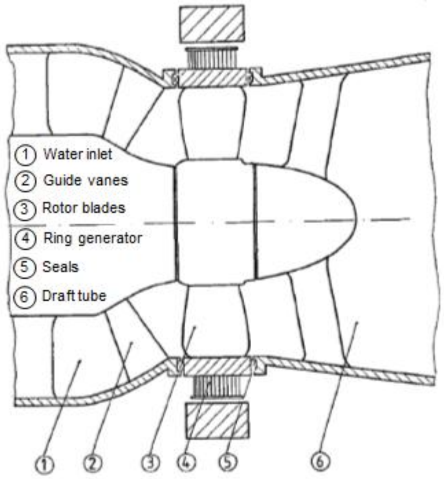

Fig. 1. Cross section through a STRAFLO-turbine [1]

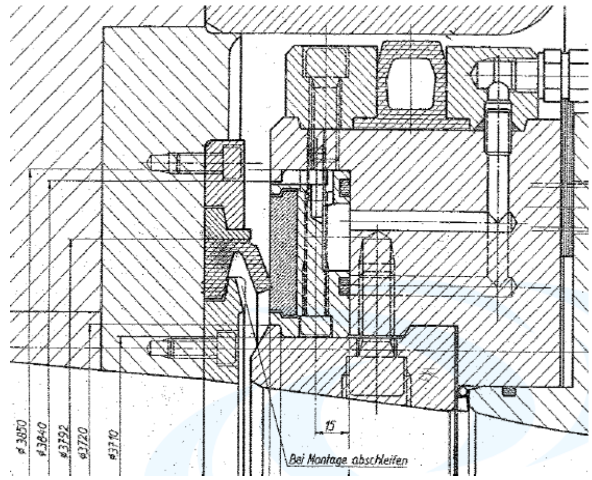

Fig. 2. Drawing of the latest sealing system of HPP Weinzödl

The present publication presents a comprehensive experimental investigation of the rim-lip-seal of the STRAFLO-turbine located at Weinzödl, having the highest capacity amongst all double regulated STRAFLO applications worldwide. The main objective of the study was, to investigate the operation of the lip-seal used in the HPP under repeatable conditions as close as possible to reality. For this purpose, a test-rig was designed and built up in the laboratory of the Institute for Hydraulic Fluidmachinery at Graz University of Technology. It was designed for the use of various other types of ring-shaped seals (e.g. shaft sealings), for long term tests with rim speeds up to 30 m/s, diameters up to 0.38 m, hydraulic heads up to at least 30m and with the opportunity for the dosage of solid particles in the test water. The final test rig design allows for the systematic investigation of any kind of lip-seals under the same conditions as those, present in the original installation with the possibility to vary all operational parameters of relevance for friction losses and wear.

The main objectives of the experimental study were:

- Experimental determination of leakage and friction losses caused by the sealing
- Sensitivity analysis regarding variable axial displacement of the external rim and variable pressure loading
- Determination of the influence of various sealing materials on leakage and friction losses
- Investigation of the behaviour under excessive sand contamination
- Optimization of the sealing concept with respect to the minimization of abrasive wear

1. Modelling of the sealing system as a prerequisite for the experimental investigation

In order to carry out a systematic experimental investigation of the original sealing system having an outer sealing diameter of approximately D=4000mm a scaling of the problem was initially required. A scale of 1:10 was chosen based on comprehensive FEM-simulations carried out to find the permissible minimum model scale. It was found that a reduction of the outer sealing diameter below D=400mm yields a bending stiffness which is completely different to that of the original seal ring.

While the scaling was only applied to the circumference of the sealing ring the cross section of the rim-lip-seal was retained. Consequently the experimentally determined leakage and friction-losses can be converted to the original machine with a factor of 10. Additionally the reduced diameter contributes to ease the manufacturing process of the test-rig and keeps the friction losses caused by the sealing system at a reasonable low level.

The prerequisite for the simple conversion of measurement results is that the specific friction power of the model represents the situation in the original machine. The specific friction power is defined as:

$$P_{FR-specific} = F_{FR-specific} \cdot u = \left(F_{N-specific} \cdot \mu\right) \cdot \left(D \cdot \pi \cdot n\right)$$

containing the specific contact force $F_{N\text{-}specific,}$ the friction coefficient μ, the diameter D and the speed n.

In order to adapt the circumferential velocity u of the model to the original turbine the rotational speed n has to be increased from n_{orig}=150 rpm to n_{model}=1500 rpm by using the geometrical scaling factor of 10.

Compared to the original turbine, the higher centrifugal force effects on the model resulting from the increased speed as well as from the changed stiffness of the sealing ring caused by the scaling of the sealing system consequently induce a higher contact force of the lip seal. Thus, comprehensive FEM-simulations were carried out in order to calculate test pressure values that cause a specific contact force $F_{N\text{-specific}}$ and consequently a specific friction force $F_{FR\text{-specific}}$ that are comparable with the situation in HPP Weinzödl. An exemplary result of these simulations is shown with the diagram in fig. 3, showing the contact pressure of the lip-seal plotted against the contact length x. The specific contact force $F_{N\text{-specific}}$ is represented by the area below the curve.

The FEM-simulations were carried out for the model scale as well as for the original turbine. This procedure allows the adjustment of the experiment to particular operation points of the original sealing system. As an example a relative seal pressure of 1.7 bar and an axial displacement of the original lip seal of Δx=4.5 mm is simulated with a displacement of Δx=4.5 mm and a relative test pressure of p_{test}=0.7 bar on the test-rig. The test pressure is measured on the level of the rotation axis of the seal ring and is individually adjusted by a ball valve at the outlet of the test drum. The pressure correction of Δp=1.0bar is composed of Δp=0.9 bar due to the increased rotational speed and Δp=0.1 bar due to the changed stiffness of the seal ring.

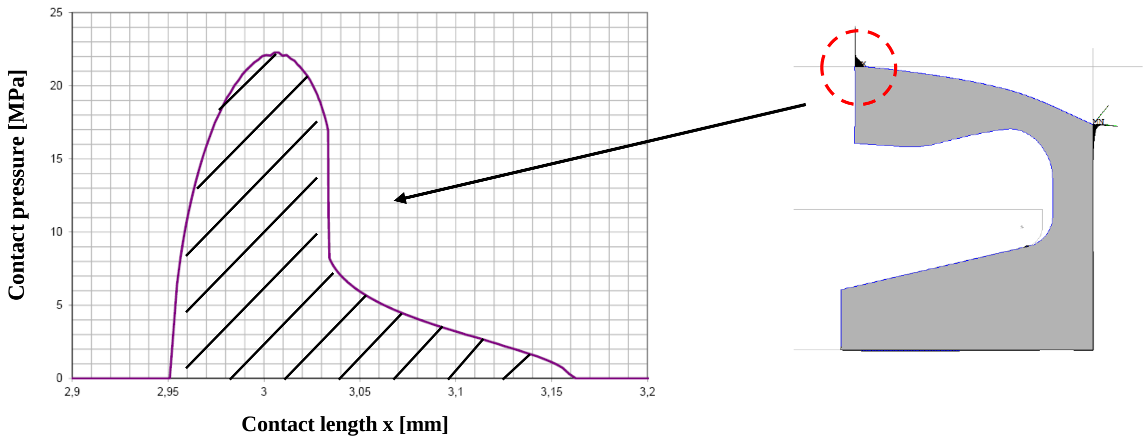

Fig. 3. Contact pressure plotted above the contact length as an exemplary result of the FEM-simulations

2. Development of an appropriate test-rig design

In order to investigate the sealing system of the STRAFLO-turbine in an experimental way a test-rig concept is required allowing a rotational movement of the lip-seal in respect to a stationary counterpart with the additional possibility to adjust the axial position of the lip-seal during the operation of the test-rig.

Additional boundary conditions for the test-rig design were the possibility for long term tests, the adjustment of rim speeds up to 30 m/s and hydraulic heads up to at least 30m, the use of sealing diameters up to 0.40 m, the dosage of solid particles in the water and the opportunity to vary all operational parameters of relevance for friction losses and wear. Finally a customizable test-rig design was created which may be also used for various other types of ring-shaped seals (e.g. shaft sealings). While fig. 4 shows the 3D-CAD-model of the test-rig with a section through the sealing system and its surrounding fig. 5 shows a detail view of the STRAFLO rim-lip-seal.

As shown in fig. 5 the tested lip seal ring (1) is mounted to a rotating disc (2) by means of segmented clamping elements (3). The stationary counterpart of the sealing system is made of ceramic (4) and, similar to the clamping ring, manufactured in segments. At the joints of the single ceramic elements slanted edges ensure the development of a leakage flow between the lip-seal and its contact surface required for the lubrication and cooling of the sealing system. The single ceramic elements are clamped to a supporting ring (5) by means of separate clamping elements (6). The trick here was to press the ceramic elements against two sealing rings with a different diameter (7) while avoiding cracks in the shoulders of the brittle elements. A comparison of fig.5 and fig.2 shows that the experimental setup was designed close to reality.

The supporting ring bearing the ceramic elements is connected to the test drum which is slidable mounted to two guide bars located in parallel to the rotational axis of the sealing at the right and left handside of the test rig. With the help of two spindles the axial position of the test drum can be individually changed in respect to the rotating sealing ring. The distance between the "zero position" of the lip seal ring and the ceramic elements is measured with the help of an electrical distance sensor located outside the sealing system (8).

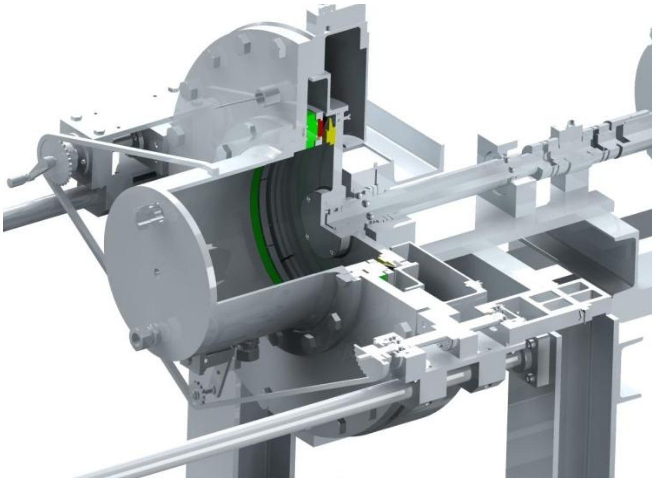

Fig. 4. 3D-CAD-model of the test-rig

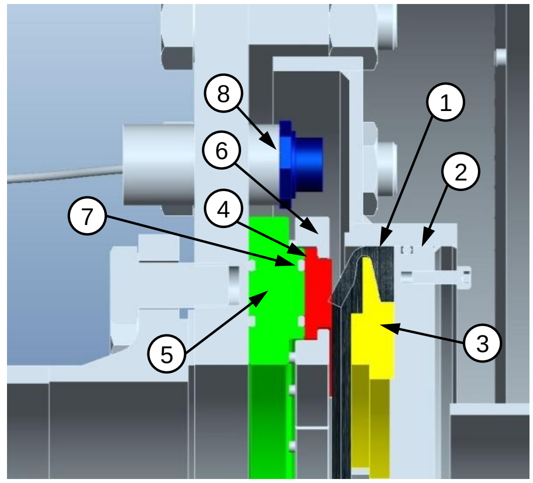

Fig. 5. Detail view of the sealing region

The leakage water passing the sealing system is collected in a basin located on an electrical scale just below the test rig. Every ten minutes a solenoid valve opens and the basin is automatically emptied. In this way the time-dependent mass of leakage water in the basin is recorded and finally the leakage can be calculated.

Due to the fact that the meridional velocity in axial turbines is low compared to the circumferential velocity, a test-rig without a fully axial flow through the sealing system was designed. Instead, the test water enters the test drum through a pipe located at the bottom face side of the test drum and leaves through a pipe located at the top face side. A ball valve at the outlet was installed to individually adjust the test pressure inside the test drum. The water supply is provided by a circulation pump located in a water tank used for the storage of the test water.

A front view of the described test rig is shown with following figures which also show that two test rig units were installed in parallel in order to speed up the test procedure.

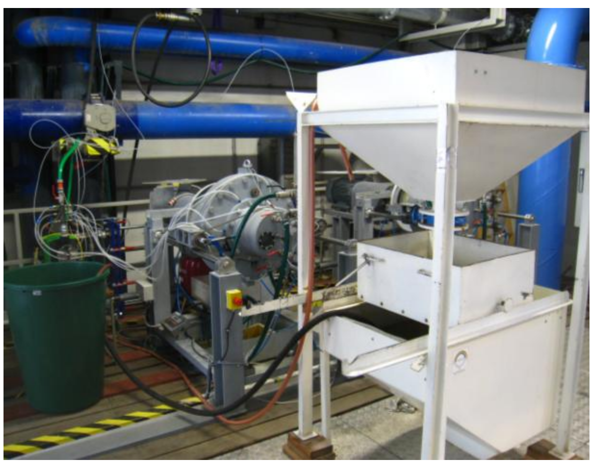

Fig. 6. Picture 1 of the final test rig

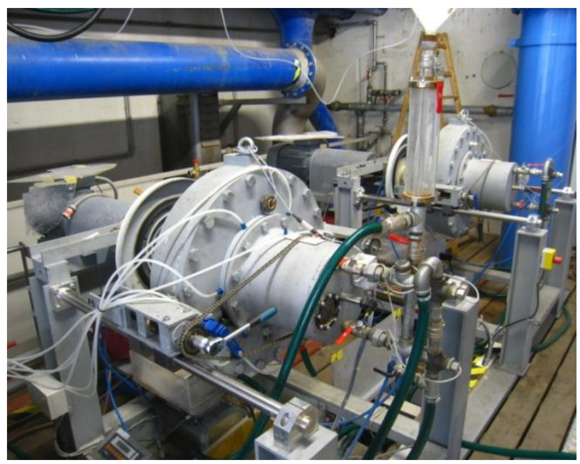

Fig. 7. Picture 2 of the final test rig

In addition to the demand for stable operation conditions on the test rig also the temperature of the test water had to be kept at constant level. Due to the heat production caused by friction, which was initially estimated to be in the order of P_{FR}=15 kW, an additional cooling system had to be installed.

A schematic drawing of the designed test rig, showing all the main parts and aggregates required for the operation, is shown with fig. 8. The schematic drawing also contains the location of the measurement equipment installed for the systematic investigation of the sealing system. The recorded measurement categories as well as the installed measurement equipment are summarized in table 1.

Table 1. Recorded measurement categories and installed measurement equipment

Measurement category	Measurement instrument	Manufacturer
Rotational speed n	Torque measurement flange	HBM
Friction torque M_{FR}	Torque measurement flange	HBM
Electrical drive power $P_{Electr.}$	Frequency converter	DANFOSS
Leakage m_{leak}	Basin with solenoid valve and electrical scale	HENK MAAS
Axial displacement of the lip seal Δx	Electrical distance sensor	PULSOTRONIC
Test pressure p_{test}	Absolute pressure transducer	KELLER
Water temperature T at in- and outlet	Temperature sensor	SAUTER

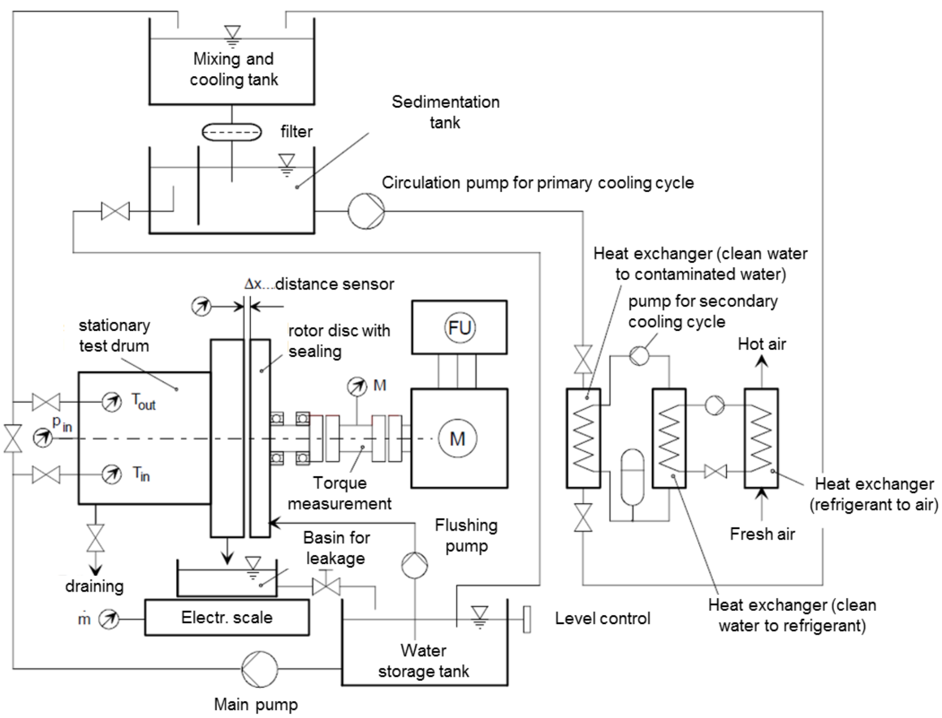

Fig. 8. Schematic drawing of the test rig

3. Experimental programme:

In order to carry out a systematic experimental investigation of the STRAFLO rim-lip-seal used for HPP Weinzödl the test programme was divided into following stages:

- Stage 1: Short-term clean water tests: The purpose of this investigation was to determine the influence of a variable test-pressure and a variable axial displacement of the lip seal on friction power and leakage. In this stage three different sealing materials were tested and compared. Each operation point was tested for 1.5 hours.
- Stage 2: Long-term clean water tests: Particular operation points tested in course of the short-term tests were picked out for a long-term investigation. For each operation point a testing time of 350 hours was used. Every 50 hours the test rig was opened for a visual evaluation of wear on the surface of the lip seal.
- Stage 3: Short-term tests with excessive sand contamination: The operation points chosen for the long-term tests were additionally tested with excessive sand contamination. The tests were aimed at determining the influence of sand contamination on friction power, leakage and abrasive wear of the lip-seal.
- Stage 4: Prevention from sand contamination of the sealing region by means of constructive changes: Various constructive changes were realized to avoid the abrasive wear of the lip-seal due to massive sand contamination. Additionally a flushing device was installed to keep the sand away from the sealing region.

Based on the fact that the sealing pressure as well as the displacement of the lip seal in respect to the ceramic elements varies in the original STRAFLO-turbine, a matrix containing several operation points to be investigated was defined to guarantee the experimental investigation of an operation range that is comparable to the operational situation in HPP Weinzödl. The variation of operational parameters on the test rig which results in the definition of 12 different measurement points is summarized in table 2.

Table 2. Summary of operation points for the experimental investigation on the test rig

Test pressure $P_{abs-test}$ [bar]	**Axial displacement Δx of the sealing in respect to the ceramic ring [mm]**			
	0	2.5	4.5	6.5
1.2	MP 1	MP 4	MP 7	MP 10
1.7	MP 2	MP 5	MP 8	MP 11
2.2	MP 3	MP 6	MP 9	MP 12

As initially mentioned 3 different materials for the rim-lip-seal were tested with a stationary counterpart made of ceramic. The material data of the single components are summarized in table 3.

Table 3. Summary of investigated lip-seal materials

Component	**Material**	**Abbreviation**
Lip seal ring	Acrylnitril-Butadien-Kautschuk NBR	NBR, soft
Lip seal ring	Acrylnitril-Butadien-Kautschuk NBR	NBR, stiff
Lip seal ring	Self-lubricating molded polyurethane PUR	PUR, extra stiff
Ceramic ring	Siliziumcarbid SiSiC	ROCAR SiC

4. Results

The results of the previously described test stages are summarized with the following sub-chapters. It will be shown that detected friction power and leakage is strongly influenced by the choice of the seal material. Additionally the results show that apart from run-in grooves no signs of abrasive wear could be found on the surface of the lip-seal during the short- and long-term clean water tests. Finally the dosage of sand yields excessive abrasive wear which is a proof that the state of damage known from HPP Weinzödl can be simulated close to reality.

4.1 Short-term clean water tests

To gain a clear understanding of the behaviour of the sealing system under variable operating conditions short-term tests with clean water were carried out. In course of this test phase a test time of 1.5 hours was used for each of the twelve operation points summarized in table 2. Following figures present the results achieved with three different sealing materials. While fig. 9 shows the friction power $P_{friction}$ as function of test pressure and sealing material fig. 10 presents the detected leakage. The present friction power range and leakage range at constant test pressure results from the variation of the axial displacement of the rotating lip-seal in respect to the ceramic counterpart.

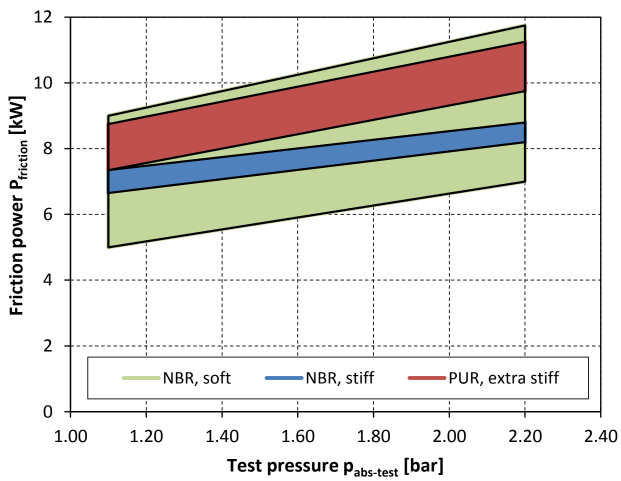

Fig. 9. $P_{Friction}$ as function of test pressure and material

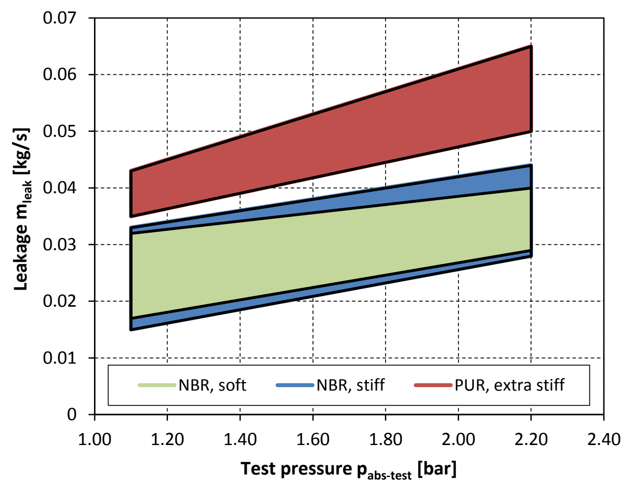

Fig. 10. Leakage as function of test pressure and material

It turns out that the lowest friction losses can be detected for the stiff and soft NBR sealing, whereas the soft NBR sealing shows a higher sensitivity to changes of the axial displacement. The average of all measured friction power values accounts for $P_{Friction}$=7.5 kW for the NBR sealing and for $P_{Friction}$=9.0 kW for the PUR sealing.

Also as far as it concerns the leakage losses the NBR sealing versions show a better behaviour. While the mean leakage accounts for m_{Leak}=0.03 kg/s for the NBR sealings the average leakage measured with the PUR sealing accounts for m_{Leak}=0.045 kg/s.

These findings consequently enable a systematic explanation of the function of the sealing system. Referring to the same operation point the use of the extra stiff sealing ring made of PUR results in a smaller contact surface than compared to the soft NBR sealing versions. Consequently the contact force of a stiff lip-seal version is higher than compared to a soft version. The development of a lubrication film between the rotating sealing ring and its counterpart made of ceramic is suppressed and an increased friction power can be detected.

4.2 Long-term clean water tests:

In order to evaluate the running-in behaviour of the investigated sealing system long-term clean water tests were carried out at particular operation points already investigated in course of the previous test phase. An exemplary result of the long-term tests is shown with fig. 11 presenting the development of friction power and leakage of the stiff NBR sealing at measurement point MP 8 (see table 2) over a period of 350 hours.

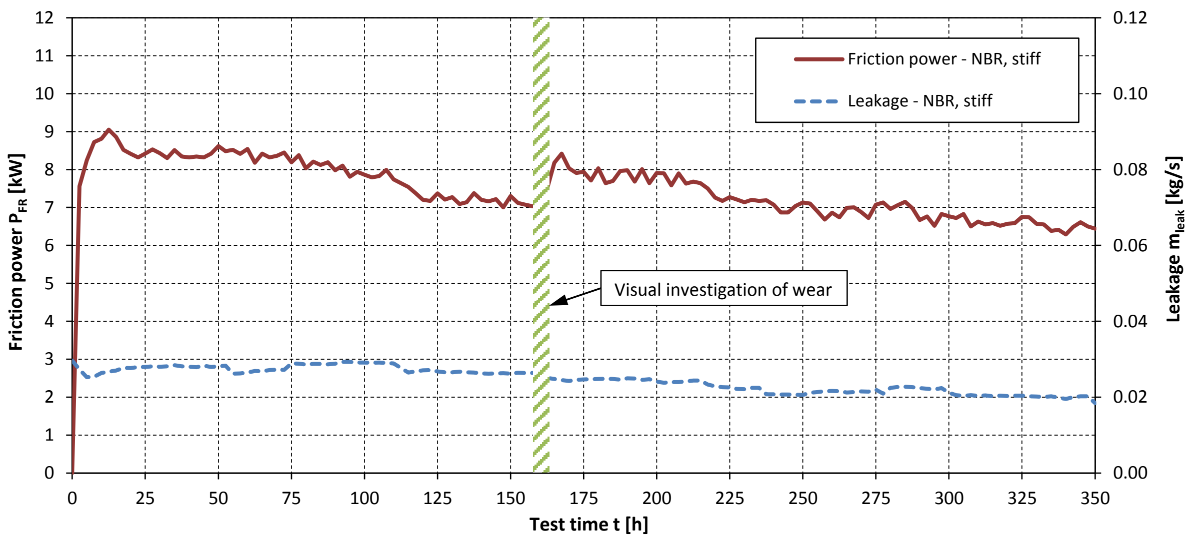

Fig. 11. Long-term test recording of friction power and leakage for MP 8 and NBR, stiff

The results presented in fig. 11 show that a distinctive running-in phase is clearly visible. Within the first 160 hours of operation the friction power dropped from $P_{Friction}$=9.0 kW to $P_{Friction}$=7.0 kW. After an interruption for visual investigations of the sealing ring the long-term tests were continued for another 190 hours. After a total operation time of 350 hours the friction power has dropped to a level of $P_{Friction}$=6.5 kW and the leakage was reduced to m_{Leak}=0.02 kg/s. Apart from run-in grooves no signs of abrasive wear could be found on the surface of the lip-seal which is in accordance with experience accumulated in operating HPP Weinzödl. Without the occurrence of flood waters with increased bed load no abrasive sand particles are forced into the sealing area and the durability of the sealing system is not endangered.

4.3 Short-term tests with excessive sand contamination

To reproduce the massive abrasive wear of the sealing in case of high sand contamination of the flow passing the sealing region a procedure for the dosage of sand was developed on the test rig. For this purpose a sand dosing unit was installed at the test rig just right before the clean test water enters the test drum. The inlet pipe of the test drum leads the mixture of water and sand into the region where the sealing is located. Within a period of 5 minutes the sand is flushed out, separated in the sedimentation tank and a new load of sand has to be added. To accelerate the progress of abrasive wear a dosage of 1 kg sand per minute was used in course of this test phase.

An exemplary result of a test run with an excessive sand dosage every five minutes is shown with fig. 12 presenting the development of friction power and leakage as function of time. A part of the sand added to the test water accumulates in the annulus recess of the V-shaped sealing ring resulting in a higher contact force of the sealing and thus in a higher friction power. A bigger part of the added sand is washed out within a time period of a few minutes and a new dosage of sand needs to be started. Consequently this procedure leads to a gentle increase of the mean friction power and to a tooth-shaped friction power curve.

Furthermore it was observed that the sand passing the sealing region is pulverized and ends up as quartz flour in the leakage basin located at the bottom of the test rig. This finding is in accordance with observations at HPP Weinzödl. Additionally, the fact that a large number of sand particles make their way through the sealing system leads to a slightly increased leakage.

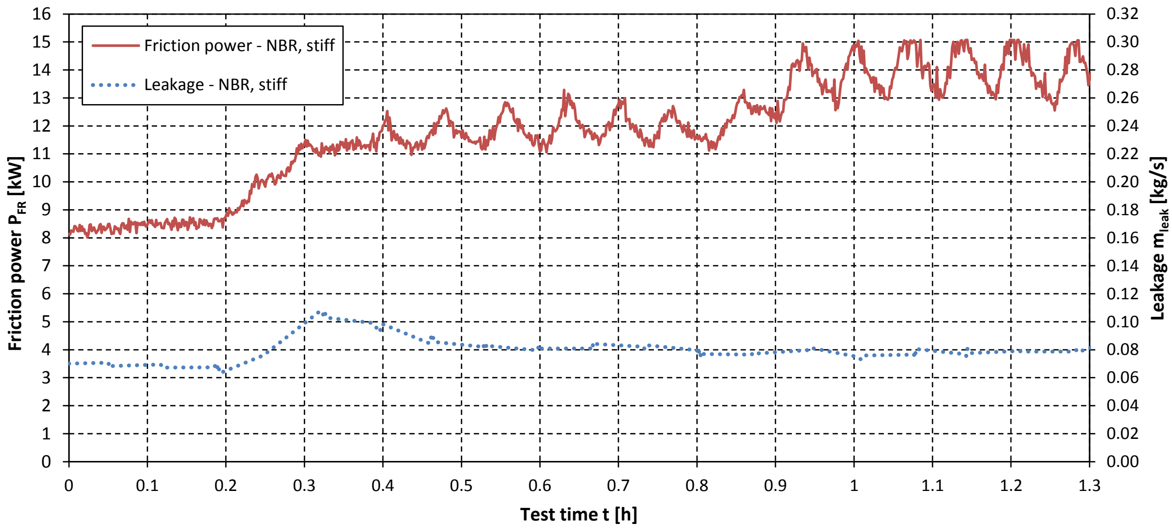

Fig. 12. Development of friction power and leakage in course of tests with excessive sand contamination

The high sand contamination does not only affect friction power and leakage but also the wear of the sealing ring. For the first time massive abrasive wear was detected at the surface of the lip-seal. Within the test time of 1.3 hours several millimetres material thickness was removed. This is demonstrated with fig. 13 and fig. 14 showing a sketch and a photograph of the wear zone on the surface of the lip-seal. The shown damage of the lip-seal is comparable to the damage known from HPP Weinzödl.

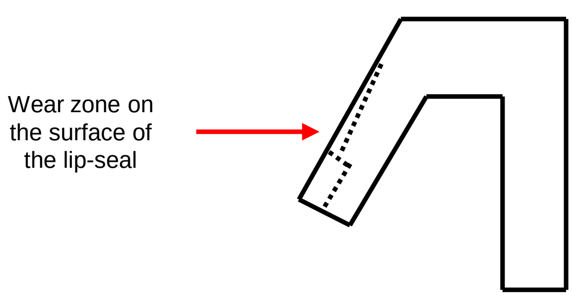

Fig. 13. Sketch showing the wear zone of the lip-seal

Fig. 14. Photograph showing the wear zone of the lip-seal

5. Development of a flushing system for the prevention of sand contamination

To avoid the seal wear due to high sand contamination of the fluid flow a modified design of the sealing region containing an effective flushing system is required. Until this point similar concepts for STRAFLO-turbines have only be used for the STRAFLO tidal power station in Annapolis, CANADA [5], where a hydrostatic seal consisting of several floating sealing elements connected to each other is used as sealing between the rotating external runner ring and the generator cavity.

While the functioning of this sealing system was successfully proven in course of comprehensive experimental investigations (see [1]), the idea of a hydrostatic sealing has the disadvantage that the required construction is rather complicated and the functioning permanently requires pressurized water. Therefore a combination of the conventional lip-seal design and a labyrinth sealing with a flushing device was tested in course of the present study.

For this purpose two additional rings building a labyrinth sealing were inserted to protect the sealing region from direct sand contamination. The adjusted clearance between rotating and stationary parts accounts for s = 1mm in radial direction and depending on the axial displacement between lip-seal and ceramic part s = 1-7 mm in axial direction. Another two additional rings contain the water supply for the flushing system as well as the flushing nozzles which guide the flushing water into the region between the lip-seal and the labyrinth sealing. A sketch of the new sealing system is shown with fig. 15. The experimental investigations showed that the activation of the presented flushing device yields a successful flushing of the sealing region while keeping the total leakage at a reasonable low level.

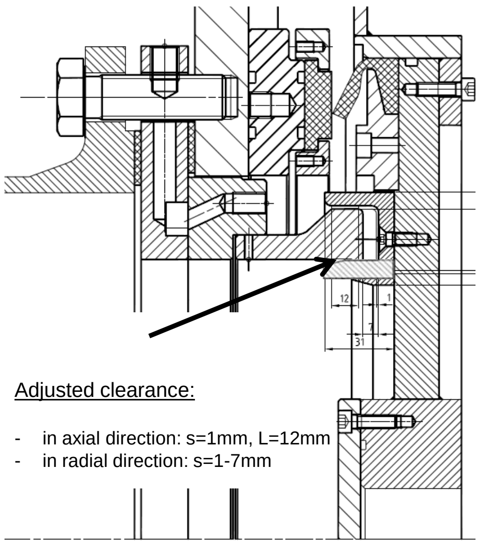

Fig. 15. Cross section through the modified sealing system

Fig. 16. Picture of the stator containing the flushing nozzles

A proof of the functioning of the proposed sealing design is given with the measurement series presented with fig. 17, showing the friction power and leakage as function of the test time for four different test runs with the following test procedure:

- STEP 1: The test rig with inserted labyrinth rings but without flushing and sand dosage was started using a new sealing of the type NBR-soft in combination with a ceramic ring with a moderate degree of wear.
- STEP 2: After one hour of operation the test run was stopped and the flushing device was activated. A flush volume of 0.8 kg/s water was used and the test rig was operated for another hour.
- STEP 3: After another stop of the test rig the sand dosage with a loading of 1 kg/min was taken into operation. The flushing volume was kept at 0.8 kg/s.
- STEP 4: After another hour of operation the flushing volume was reduced to 0.1 kg/s while keeping the sand dosage at a constant level.

First of all the recorded measurement series shows that neither the activation of the flushing device nor the activation of the sand dosage yields an increase in friction power. On the contrary, there is a trend towards decreasing friction power due to the previously detected running-in process (see fig. 11). Additionally it turns out that also the leakage flow is not affected by the activation of flushing and sand dosage.

Finally the reduction of the flushing volume starts to have an impact on the development of the measurement categories. As already shown in fig. 12 the excessive sand contamination causes a strong increase in friction power which is a proof that sand particles find their way into the sealing region. On the other hand the leakage flow drops due to a blockage of the sealing system.

Consequently it turned out that there is a minimum flush volume that is decisive for the functioning of the sealing system. A too low flush volume does not prevent the sealing system from sand contamination and does not show any positive effects.

However, it was proven that the use of a flushing volume of 0.8 kg/s prevents the sealing region from any abrasive wear. The flush volume is around ten times higher than the leakage of the sealing system and has to be supplied by a separate pump system resulting in additional energy consumption. But as the flushing system needs to be only activated in case of flood waters with increased bed load and in case of stopping/starting the units, the total annual over-consumption of energy would be kept quite low.

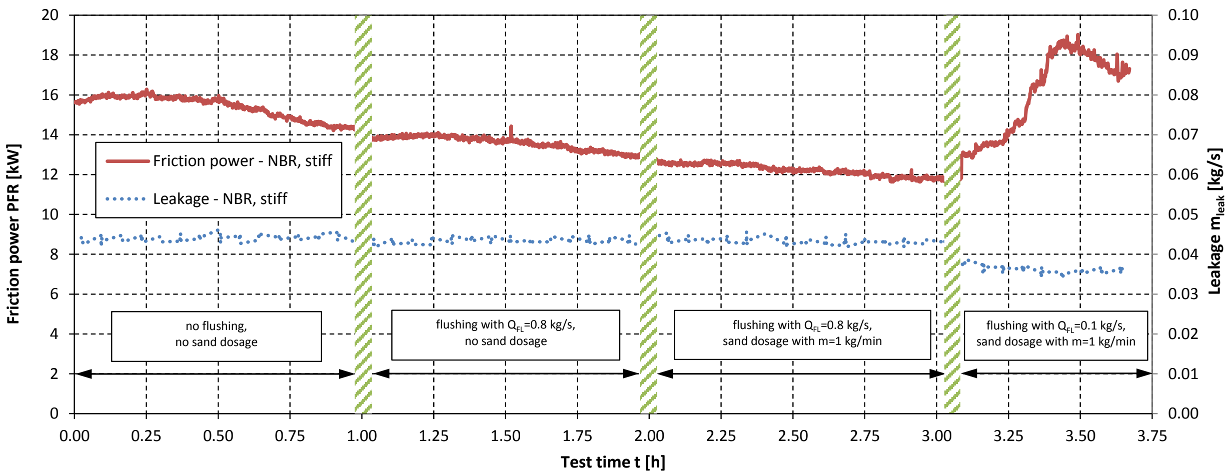

Fig. 17. Friction power and leakage as function of the test time for four different test runs

6. Conclusions

In course of the presented study a test rig for the systematic experimental investigation of the rim-lip-seal of a double regulated STRAFLO Kaplan-turbine was designed and built up at the Institute for Hydraulic Fluidmachinery at Graz University of Technology. With the help of this test rig the magnitude of friction power and leakage caused by the sealing system at HPP Weinzödl was detected for the first time ever. The variation of sealing materials as well as the variation of operational parameters provided important input for the understanding of the behaviour of the sealing system under various conditions.

Additionally the dosage of sand showed that the lip-seal is destroyed within a short period of time resulting from abrasive wear due to excessive contamination of sand. This finding is in accordance with the operational experience from HPP Weinzödl. Quite similar to the original machine excessive sand contamination additionally results in abrasive wear of the ceramic surface. In the case of sand dosage the friction power rises by a factor of 2 to 3.

Based on these findings a modified sealing concept additionally including a labyrinth seal and a flushing device was developed and its functioning was proven on the test rig. It was found that there is a minimum flush volume that is decisive for the successful flushing of the sealing region. The finally presented flushing concept prevents the entering of abrasive sand particles into the sealing region of the machine while keeping the leakage and friction power at a reasonable low value. The concept developed on the test rig is suitable for the implementation in full-scale STRAFLO-units and represents a valuable contribution to solve abrasion problems in the sealing region of STRAFLO-turbines.

References

[1] **Christ, A.:** Hydrostatic seals for large diameters. 9th International Conference of Fluid Sealing, Noordwijkerhout, Netherlands, 1st – 3rd April, 1981.

[2] **Hopper, H.R.; Mayer, H.W.; Severn, B.:** Manitoba HYDRO and STRAFLO Units. 92nd E.I.C. Annual Congress St. John's, Newfoundland. 25th /26th May, 1978.

[3] **Frust, A.:** Ausbau des Rheinkraftwerks Laufenburg – Fünf Jahre Betriebserfahrung mit Straflo-Turbinen. Bulletin SEV/VSE 2/98. S.25-29.

[4] **N., N.:** Mitteilungen. Technische Rundschau Sulzer, 3/1980, S.122-123.

[5] **DeLory, R. P.:** Prototypen-Gezeitenkraftwerk erreicht 99% Verfügbarkeit. Technische Rundschau Sulzer, 1/1987, S. 3-7.

The Authors

Schiffer Jürgen, Dipl.-Ing., studied mechanical engineering at the Graz University of Technology. Since 2008, he is working as scientific assistant at the Institute for Hydraulic Fluidmachinery at the Graz University of Technology where he basically works in the field of computational fluid dynamics for applications in the hydro power and pump industry.

Helmut Benigni, Ass. Prof. Dipl.-Ing. Dr., studies of mechanical engineering at the Graz University of Technology, specialisation in numerical simulation, dissertation in optimisation of hydraulic machines. In post-doctoral position responsible for hydraulic machine simulations by means of CFD-methods, development of different hydraulic designs and of machine configurations. Vice-head of the Institute for Hydraulic Fluidmachinery, Graz University of Technology.

Helmut Jaberg, Prof. Dr.-Ing., studies of aeronautical engineering in Stuttgart, Munich and Southampton, head of a pump development division and a business unit at KSB, certified expert for water hammer and unsteady behaviour of different power plants. Head of the Institute for Hydraulic Fluidmachinery, Graz University of Technology.

Wessiak Meinhard, Dipl.-Ing., studied mechanical engineering at the Graz University of Technology. In 2008 he joined VERBUND Hydro Power AG, Department for mechanical engineering and maintenance. Presently he is working as project manager in refurbishment and new hydropower projects.

Josef Mayrhuber, Dipl.-Ing. Dr.techn., graduated at the Technical University of Graz, Austria in 1987 and achieved his doctoral degree in 1992. He joined VERBUND in 1994 working in various positions. Presently he is head of department for mechanical engineering and maintenance.