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ABSTRACT

The Hydraulic Pressure Machine (HPM) is an energy converter to exploit head differences between 0.5 and 2.5 m in small streams and irrigation canals. The HPM looks similar to a classic breast shot water wheel but has a smaller number of blades, a relatively large central hub and the wheel runs at variable speeds (2 to 12 min^{-1}). Preliminary results show that the HPM is an economically and ecologically viable technology for small hydropower generation. The clearance gap between the blade tip and the shroud at the bottom of the wheel is very important regarding power losses. A theoretical approach has been developed which considers a stationary wheel to quantify the leakage losses. However, no validation of this theory has been done. The goals of this research are to quantify the leakage at operating condition and to improve the HPM blade design to further reduce gap losses. Thus a large scale physical model is tested at laboratory conditions. The HPM model is 1.1 m in diameter, 0.8 m wide and has 12 flat blades. Variable blade tips machined from steel and EPDM rubber are investigated with gap sizes of 1, 5 and 10 mm. The physical model results show that the flow rate passing the wheel during operation is approximately one third of the flow rate calculated by the theoretical approach. The variation of gap sizes reveal the importance of small clearance gaps to reach high efficiencies.

Keywords: *Small Hydropower, Gap Loss, Hydraulic Pressure Machine*

1. INTRODUCTION

The history of water wheels goes back to antiquity. Scoop wheels are known for irrigation purpose at the Euphrates in Mesopotamia since 1200 B.C. (Wölfel 1987). Also at the Huáng Hé in China and Nile in Egypt, large wheels were used for water conveyance and to grind grain (Mosonyi 1963). Subsequently, water wheels were distributed widely and were optimized for a variety of purposes (e.g. mills, mines, metal working). The 19th century was the time of prosperity for the technology and it was significantly improved by several technological enhancements.

During this time the Swiss engineer Walter Zuppinger developed two types of water wheels: The well-known *Zuppinger wheel,* which is a special type of an undershot water wheel, and a different wheel with a closed hub (Delabar and Dingler 1867). To the authors' knowledge Figure 1 from the 1860s is the first illustration showing a water wheel with a closed hub. In the following text, these kinds of wheels are called Hydraulic Pressure Machines (HPM).

The HPM looks similar to a typical breast shot water wheel but has some significant differences. Special features of the technology are the fewer number of blades (around 12 to 24 – unlike traditional water wheels with around 36 to 48 blades) and a relatively large central and closed hub (measuring 1/3 of the outer diameter) to dam the upstream water level (see Figure 1 and Figure 2). For optimal power extraction the upstream water level is kept at the top of the hub. Therefore, the wheel runs at variable speeds in response to naturally changing flow.

The technology developed by Zuppinger was ignored completely during industrialization and was never applied for a real installation. But the technology was rediscovered in the beginning of the 21st century and investigated in detail by the University of Southampton (UK), the Darmstadt University of Technology (DE) and the University of Architecture, Civil Engineering and Geodesy in Sofia (BG) (Brinnich 2001, Senior et al. 2010, Bozhinova et al. 2012, Müller et al. 2012).

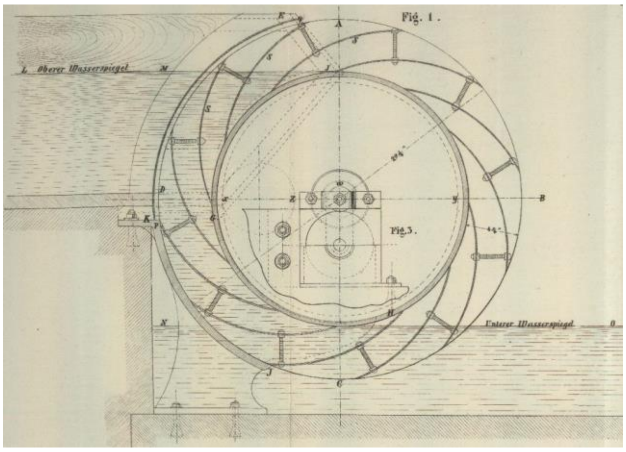

Figure 1. Hydraulic Pressure Machine (Delabar and Dingler 1867)

The HPM is run by the hydraulic pressure difference between upstream and downstream water levels (Senior 2009). Experimental results indicate that the technology is suitable for power extraction in sites with small head differences (0.5 to 2.5 m) (Senior et al. 2007). Furthermore, it was found that one of the main benefits of the HPM compared to other hydraulic machines like traditional water wheels and turbines, is that the wheel allows for good continuity for aquatic life (e.g. fish) and bed load (Senior et al. 2010). This advantage is due to the slow rotational speed (2 to 12min^{-1}) and the small number of blades, which creates a relatively large space between two blades. Efficiency rates from 60 to up to 80 % were achieved (Delabar and Dingler 1867, Senior et al. 2010, Müller et al. 2012, Paudel et al. 2013).

The research conducted to date revealed that gap losses are of high importance for the HPM technology due to the large proportion of the total energy losses within the converter (Linton 2013). Gap losses can be described as the flow rate passing the wheel between the side of blade and the housing or the blade tip and the shroud (see Figure 2). In literature, gap losses are given from 2 to 12 % of the total flow rate (Brinnich 2001, Müller et al. 2012). To further investigate the quantity of gap losses, a theory was developed and different bottom clearance gap sizes are investigated within a physical model in this study. The main goal is to compare measured data from a physical model with results from the previously developed theory.

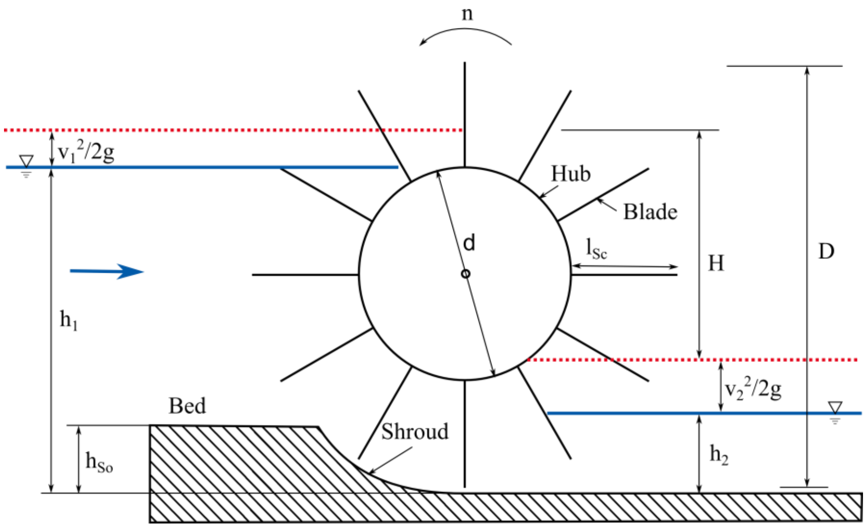

Figure 2. Functional principle of a Hydraulic Pressure Machine

2. THEORETICAL APPROACH

To quantify the gap loss flow rate (Q_L) a theoretical approach was developed (Senior 2009). The theory considers the pressure difference or rather the velocity by reason of the pressure difference at the gap, the corresponding flow area of the gap (width of the gap and length or height of the blade) and an empirically determined flow coefficient (μ) that depends on the geometry of the blade tip or the blade tip material.

$$Q_L = \textit{flow velocity in the gap x gap area x flow coefficient} \tag{1}$$

To simplify calculations of the gap losses the surrounding gaps of the blade are divided into three characteristic sections as follows (see also Figure 3):

- Q_{L1}: Leakage loss between blade tip and shroud with the bottom gap size g_b and the blade width W
- Q_{L2}: Leakage loss between blade side and housing below h_2 with gap size on the side g_s
- Q_{L3}: Leakage loss between blade side and housing above h_2 with gap size on the side g_s

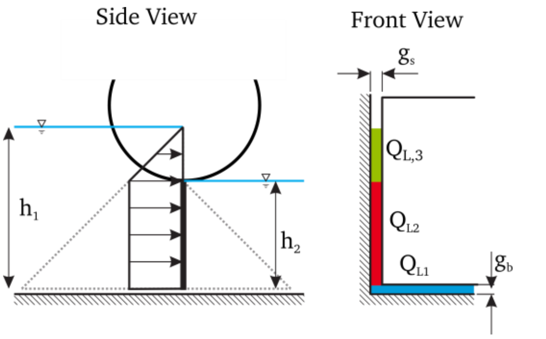

Figure 3. Gap losses sections

Based on Eq. (1) for the three sections three simplified calculation formulas can be derived as follows:

$$Q_{L1} = \sqrt{2\,g\,(h_1 - h_2)}\; g_b W \mu_b \tag{2}$$

$$Q_{L2} = \sqrt{2\,g\,(h_1 - h_2)}\; g_s\, h_2\, \mu_s \tag{3}$$

$$Q_{L3} = \frac{2}{3}\sqrt{2g}\,(h_1 - h_2)^{\frac{3}{2}}\, g_s\, \mu_s \tag{4}$$

The total gap losses can be described as follows:

$$Q_L = Q_{L1} + 2\left(Q_{L2} + Q_{L3}\right) \tag{5}$$

This theoretical approach to calculate the gap losses was initially developed for middle shot water wheels and was transferred to HPM technology (Senior 2009). Regarding the applicable flow coefficient (μ_b, μ_s) the literature on HPM quotes values ranged between 0.6 to 1.0 (Schneider 2016, Senior 2009).

Eq. 2 to Eq. 5 employed to calculate the gap losses are based on basic hydromechanics. The theory takes into consideration flow through a small orifice. The validation of this theory was done to date by measuring the flow rate passing the wheel while the wheel is standing still (Linton 2013). A validation of the previously developed theory on an operating wheel has, to the authors' knowledge, not been conducted.

3. EXPERIMENTAL PROGRAM

The objective of the experimental program is (1) to evaluate the efficiency and power output of the wheel and (2) to measure the flow rate passing the wheel without doing any work under operating conditions. A large scale physical model of a HPM was built and tested under laboratory condition to investigate these two main questions. The model scale corresponds roughly 1:2.5 to a real installation in Bulgaria (Bozhinova et al. 2012).

3.1. Test rig

The model of the HPM wheel was installed in a 20 m long and 3 m wide concrete flume in the hydraulic laboratory of the University of Applied Sciences Darmstadt (D). Figure 4 shows the flume with the tested HPM on the right and parts of a slot fish pass on the left.

The wheel itself has an outer diameter (D) of 1.1 m, a hub diameter (d) of 0.4 m and a width (W) of 0.8 m. 12 flat, 2 mm thick blades from stainless steel are diagonally mounted at the circumference of the hub. The blades are set at an angel of 15° to the shaft of the wheel. The hub is machined from polyethylene. The shaft and load bearing disks (mounted on the shaft) are made of stainless steel. The blades are bolted to the three load bearing disks at the sides and middle of the hub. The wheel is placed over a curved concrete shroud following the outer diameter of the wheel. The inlet and outlet channels are 1.2 m wide (W_1, W_2). Table 1 summarizes the main geometrical details of the tested wheel while Figure 5 shows the wheel in detail. The model itself is designed for maximum flexibility in order to allow further modification of blade geometry, blade number etc. The blade border at the tip and the sides of the blade are exchangeable. This enables tests of different blade tip versions and the variation of the clearance gap (up to 20 mm).

Table 1 Model parameter summary

Parameter		Dimension
Outer Diameter	D	1.1 m
Hub Diameter	d	0.4 m
Machine Width	W	0.8 m
Width In- and Outlet	W_1, W_2	1.2 m
Number of blades	N	12
Blade angle (to shaft)	β	15°

Figure 4. Photo of test rig from down- to upstream

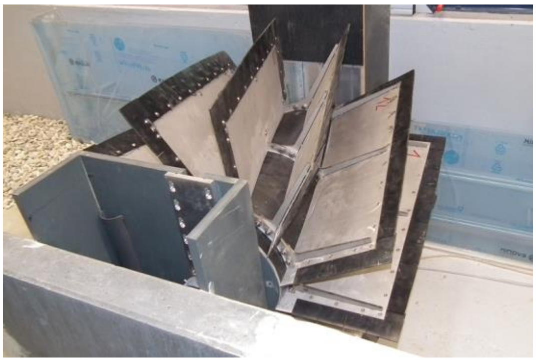

Figure 5. Wheel in detailed view

3.2. Measurement technique

To determine the power output and efficiency of the converter values for water levels up- and downstream (h_1, h_2), flow rate (Q), torque (M) and speed of the wheel (n) were measured. It is important for the operational characteristics of a HPM to keep the upstream water level constant. Thus, a control system is mandatory.

The HPM test setup is electronically braked by a coupled magnetic powder brake FRAT 2002 from Mobac Gmbh. The torque (M) is logged by torquemeter TRS 200 by Liedtke, the rotation (n) by shaft encoder ITD 01 A 4 1024 H NX KR1 S6 by Thalheim and the water levels up and downstream of the wheel (h_1, h_2) by ultrasonic sensor BUS R06K1-XA-12/070-S75G by Balluff. The water levels are measured from above to the water surface. A V notch weir is used for flow rate (Q) measurement (see Figure 4 left, background). Table 2 summarizes the measuring devices used for data acquisition and the corresponding accuracy. The head over the weir is measured via ultrasonic sensor (same make as for the water levels). As a control system for the brake and for data acquisition purpose, a LabVIEW based software in combination with a CompactRIO-9074 from National Instruments is used. A measurement period of 30 s and sampling rate of 30 Hz applies for all collected data.

Table 2 Data acquisition system

Parameter		Make	Model	Accuracy
Water level	h_1, h_2	Balluff	BUS R06K1-XA-12/070-S75G	±0.15 %
Flow rate	Q	Unknown	V notch weir	±0.8 %
Speed	n	Thalheim	ITD 01 A 4 1024 H NX KR1 S6	±0.23 %*
Torque	M	Liedtke	TRS 200	±1 Nm
* speed dependent (example for 8 min^{-1})				

3.3. Analysis

Data analysis is done in accordance with IEC 60193 *Hydraulic turbines, storage pumps and pump-turbines -- Model acceptance tests*. The most important parameter for comparison with other converters is the hydraulic efficiency (η_{Hy}). It is the quotient of the measured mechanical power (P_{Me}) at the output shaft of the wheel and the energy in the flowing water called hydraulic power (P_{Hy}).

The energy in the flowing water can be described by the product of the total energy head difference (H) in m, the flow rate (Q) in m^3s^{-1}, the density of the water ($\rho = 1000\ kgm^{-3}$) and the acceleration of gravity ($g = 9.81\ ms^{-2}$).

$$P_{Hy}=H\,Q\,\rho\,g \tag{6}$$

with

$$H= h_1+\frac{v_1^2}{2g}-\left(h_2+\frac{v_2^2}{2g}\right)=\frac{v_1^2-v_2^2}{2g}+h_1-h_2 \tag{7}$$

The flow velocity up- and downstream of the wheel (v_1, v_2) in ms^{-2} is calculated according to the law of continuity and not measured. The values therefore represent the mean velocity, with bed elevation upstream (h_{So}) and channel width (W_1, W_2) upstream and downstream of the wheel (see Figure 2).

$$v_1=\frac{Q}{W_1\,(h_1-h_{So})} \qquad v_2=\frac{Q}{W_2\,h_2} \tag{8}$$

The mechanical power measured at the shaft of the wheel is described as the product of torque (M) in Nm and the speed of the wheel (n) in s^{-1}.

$$P_{Me}=2\,\pi\,M\,n \tag{9}$$

The hydraulic efficiency is the quotient of mechanical power and hydraulic power.

$$\eta_{Hy}=\frac{P_{Me}}{P_{Hy}}=\frac{2\,\pi\,M\,n}{H\,Q\,\rho\,g} \tag{10}$$

The uncertainty involved in the derived quantities is calculated according to the constant odd combination method (Moffat 1988). Experimental data contains random (statistical) and bias (systematic or fixed) error. Random error is calculated based on standard deviation of the mean of multiple measurements taken at the same condition (IEC 1999). Fixed error is calculated based on the specification given by the producer of the used measurement equipment. Table 2 summarizes the accuracy of all measurement equipment used. The overall uncertainty for each measured value is calculated as the root sum square of the fixed error and random error (IEC 1999). To calculate the uncertainty of computed values, namely mechanical power output (P_{Me}) and hydraulic efficiency (η_{Hy}), the method of uncertainty propagation is used (Moffat 1988). The overall uncertainty for power output is summarized in Eq. (11) and for the hydraulic efficiency in Eq. (12).

$$\frac{\delta P_{Me}}{P_{Me}} = \pm\sqrt{\left(\frac{\delta M}{M}\right)^2 + \left(\frac{\delta n}{n}\right)^2} \tag{11}$$

$$\frac{\delta \eta_{Hy}}{\eta_{Hy}} = \pm\sqrt{\left(\frac{\sqrt{\delta h_1{}^2 \text{-} \delta h_2{}^2}}{h_1 \text{-} h_2}\right)^2 + \left(\frac{\delta Q}{Q}\right)^2 + \left(\frac{\delta M}{M}\right)^2 + \left(\frac{\delta n}{n}\right)^2} \tag{12}$$

3.4. Test procedure

Two different blade tip materials were chosen for comparison. The interchangeable blade tip spans the entire width of the blade and is approx. 60 mm long. First, a steel blade tip (2 mm thick), representing a conventional blade design as a reference case is analyzed (see Figure 6). Secondly, a combination of EPDM rubber (EPDM/SBR Shore hardness of 80 ±5°, 2 mm thick) and steel (stainless steel, 0.8 mm thick) is tested (see Figure 7). The two materials are layered (rubber, steel, rubber) and glued with adhesive. The blade tip itself is divided into 10 cm wide sections which are partly flexible. Each individual section is reinforced by a middle steel sheet and therefore is sufficiently rigid to resist the hydraulic forces that arise during regular operation. Three sides of each section are flexible enough to allow the blade edge to be flexible against a foreign object (e.g. fish). The main goal of this alternative blade design is to reduce damage to fish passing through the wheel.

To assess the influence of the size of the bottom clearance gap (between blade and shroud) on the efficiency and mechanical power output, three different bottom gap sizes (1, 5 and 10 mm) were investigated. Gap sizes for the two different blade tips, in order to register possible interdependency between blade material and gap size were analyzed.

For all tests the water level upstream is kept at the top of the hub (h_1 = 750 mm) and the downstream water level is kept at the bottom of the hub (h_2 = 350 mm). Earlier research showed that this water level combination (h_1 at the top of the hub and h_2 at the bottom of the hub) results in optimal energy extraction of these machines (Senior 2009). All experiments are conducted with a side gap size of 1 mm and flow rates between 30 and 180 ls^{-1} in 5 and 10 ls^{-1} increments. The variation of flow is done to evaluate whether the properties of these blades and especially of the EPDM blade tip remain constant over the whole band of operation.

Figure 6. Steel blade tip

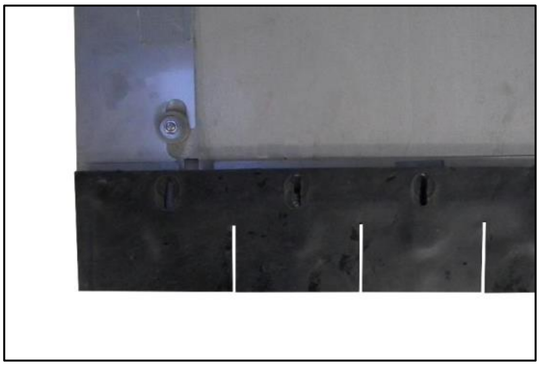

Figure 7. EPDM blade tip

4. RESULTS

4.1. Power output and efficiency

Figure 8 shows the power output of two different blade tips and bottom gap sizes of 1 and 10 mm in comparison. For better clarity, values for 5 mm gap size and error bars are suppressed in the diagram. The curves indicate that there is no significant difference in mechanical power output between the two blade tip versions. It can be concluded that no disadvantages regarding power extraction can be noticed by using alternative EPDM blade tips.

The diagram further demonstrates that with larger gaps the curve shifts on the x axis into higher flow rates. Detailed analysis of the efficiency shows that the efficiency for 1 mm clearance gap with Steel and EPDM blade tips reaches 57.8±1.7% and 57.2±1.6% respectively and for 10 mm clearance gap 49.7±1.4% and 50.6±1.4% respectively. This supports the above made conclusion that there are no disadvantages for EPDM in comparison to metal blade tips regarding power extraction. However these results clearly show that the efficiency of the wheel decreases with larger gap sizes. This is due to the higher gap losses.

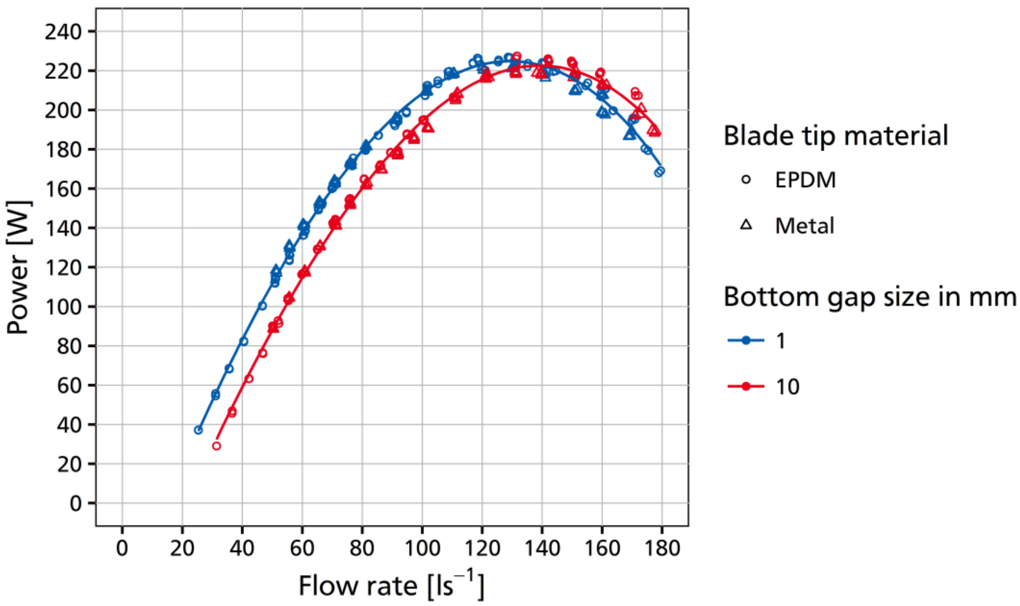

Figure 8. Results of different gap sizes and different blade tip materials on power output

4.2. Gap losses

To determine the gap losses under operation conditions, the values for the speed of the wheel (n) and the flow rate (Q) taken for 1 and 10 mm are displayed in one diagram (see Figure 9). The difference in y-axis intercept of the two displayed linear regression lines represents the difference in bottom gap losses between 1 and 10 mm bottom gap. This approach is only valid due to the fact that at the HPM, the cells between two blades are, in contrast to e.g. water wheels, completely filled up with water during operation.

Based on the parallelism of the two regression lines it can be confirmed that the gap losses are mainly dependent on the gap size (g_b, g_s) and the water levels up- and downstream (h_1, h_2) and independent on the speed of the wheel (n) as described in theory (see Eq. 2 to Eq. 5).

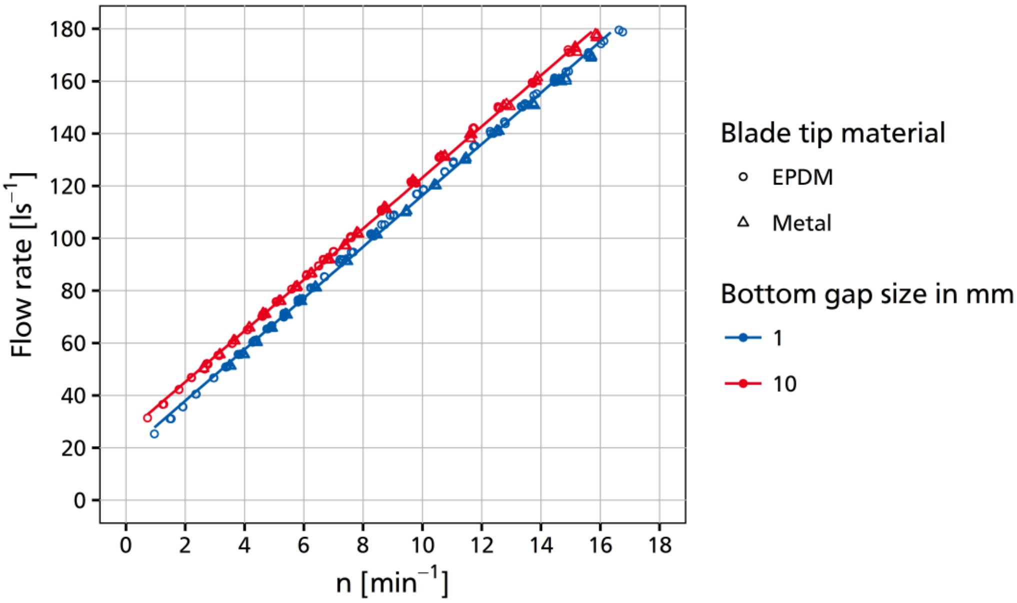

Figure 9. Results of different gap sizes on flow rate and speed of the wheel

Based on the difference in *y*-axis intercept between the two lines, a difference in gap losses of 7.0 ls^{-1} can be determined. Based on this value, Eq. 2 to Eq. 5 and the geometric details of the wheel, the coefficient for the bottom gap (μ_b) can be calculated by rearranging the equations or the use of a goal seek function. For the specific wheel tested the flow coefficient for the bottom gap is approx. $\mu_b = 0.33$. This value is far below the anticipation based on the literature quoting values between 0.6 and 1.0. The physical model results show that the flow rate passing the wheel during operation is approx. one third to half of the flow rate calculated by the theoretical approach taking a standstill wheel into account. This leads to the conclusion that gap losses are responsible for a much smaller proportion of the total energy losses within the converter than expected. That means other effects (inflow, turbulences etc.) have a bigger proportion than expected to date.

5. CONCLUSION

To determine the gap losses between the blade tip and the shroud of a HPM under operating condition, different gap sizes have been investigated in this study. Therefore, a conventional blade design (steel edge) and EPDM rubber as a sandwich construction were investigated. For analysis and comparison, a large scale physical model was built. Both materials reached with 1 mm gap size approx. 58 % hydraulic efficiency and a maximum mechanical power of approx. 220 W. Regarding power extraction and machine behaviour in general, no differences between the two analyzed blade tip materials were found. Results regarding the variation of the bottom gap size showed differences in efficiency of approx. 8 % between 1 and 10 mm. This underlines the importance of minimal gap sizes for high efficiency rates and optimal power extraction. To determine the gap losses (flow rate passing the wheel) a theory based on basic hydromechanics was developed in the past. This theory takes the flow through an orifice into account. The analysis of the results taken from the described physical model showed that the measured gap losses are approx. one third to a half of the flow rate calculated by the theory. This means gap losses have been overestimated and other losses (e.g. inflow, turbulences) underestimated in the past.

6. ACKNOWLEDGMENTS

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