

THE DEVELOPMENT OF THE ROTARY HYDRAULIC PRESSURE MACHINE

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ABSTRACT

There currently exists a demand for economical hydro-electric machinery that operates at low heads and moderate flows; sites where the drop from a higher water level to a lower one is quite small, in the order of 1-3 m, with flows between 5 m³/s and 15 m³/s. The economics which determine a machine's viability are a function of its size and complexity, its efficiency characteristics and its power output. This paper analyses the *Staudruckmaschine* (SDM) in detail using scale model tests. These tests found that the SDM had many deficiencies resulting in unnecessarily low efficiencies and flow capacity, which is the volume of water which the wheel can process per second per metre width. The scale model tests also aided the development of a new detailed theory of operation for wheels operated by hydraulic pressure. Significant modifications are then proposed for the SDM to address its deficiencies. These revise its channel design, blade shape, and support structure. The result is effectively a new machine, the *Rotary Hydraulic Pressure Machine* (RHPM), which is expected to have over 20 % higher efficiency, greater flow capacity and smoother operation than the SDM. The RHPM can be built at any site; however it is especially suitable for incorporation into existing weir structures with minimal modifications being required to the weir structure itself. Furthermore, the RHPM is unlike most conventional rotodynamic machines as it is symmetrical and therefore fully reversible. This property means it also has potential application in tidal flow environments.

Keywords: Rotary Hydraulic Pressure Machine, Staudruckmaschine, Low Head, Hydropower

1 INTRODUCTION

The European Commission has set a target which requires 12 % of the total electricity production to come from renewable sources by 2010. Within Europe, over 70 % of the theoretically available hydropower output is already exploited (Giesecke & Mosonyi 2005). Of this, the development of hydropower over 1 MW installed capacity is virtually complete and what remains are sites with head falls between 1 m and 3 m. Until now this sector of hydropower has generally been considered un-exploitable and ignored by many resource surveys, e.g. ETSU (1982). An indication of the number of potential sites was made by NRW (2005) which counted the number of weir structures in a German federal state with an area of 1,000 km². This found 1,304 potential sites with heads between 0.2 m and 1.0 m which would suggest that tens of thousands of sites must exist throughout Europe. The majority of these sites have flow rates too small to make use of conventional turbines economically. This fact has been realised and many manufacturers are now developing new technologies to address this. The working principles range from kinetic energy conversion, the creation and utilization of vortex flow, siphon-action,

wave energy conversion for re-conversion into higher head differences, ‘classic’ stream wheels and a rotary hydraulic pressure machine. A study of these machines by Wiemann (2006) found most of these to have insufficient efficiency to be economical. This concluded that the technology with the most potential so far was the *Staudruckmaschine* which had the greatest efficiency and would have the least ecological impact.

2 THE STAUDRUCKMASCHINE

The *Staudruckmaschine* (SDM) shown in Figure 1 (comprising two symmetrical units) was patented in 2001 (Brinnich) as an attempt to exploit sites with low heads and moderate flows, that is with 1-3 m head and flows from 5 m³/s to 15 m³/s (assuming a 5 m wide wheel). *Staudruck* translates as dam-effect, and a definition is proposed:

A Dam-effect waterwheel restrains the flow of water thus maintaining the upstream water level, without the use of a sluice gate

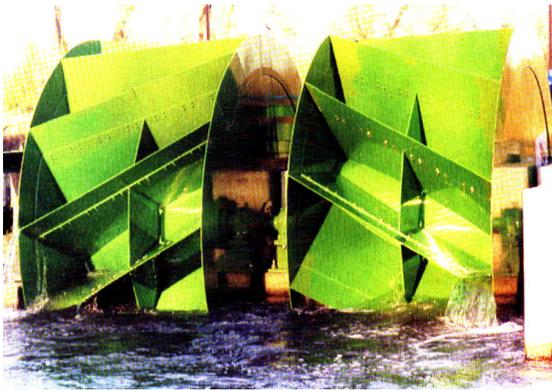


Fig.1. The *Staudruckmaschine*, SDM (Brinnich, 2001)

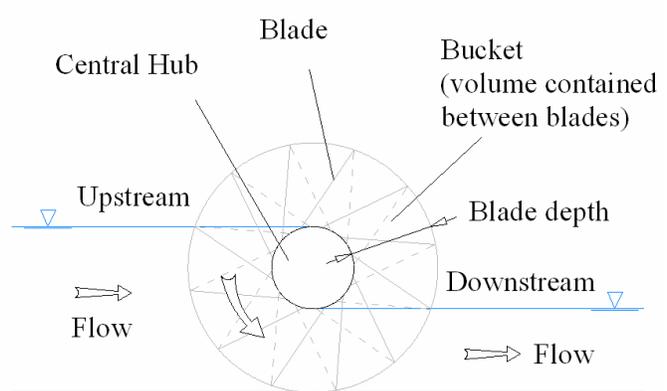


Fig.2. Detail of the SDM

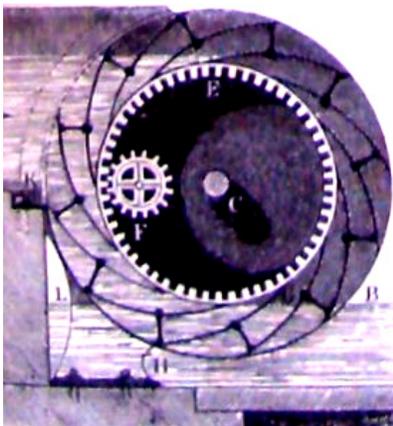


Fig.3. The *Zuppingerad* (Brockhaus 1903)

Referring to Figure 2, the SDM has a central hub the same diameter as the head drop between the upstream and downstream water levels. When compared to traditional waterwheels for equivalent heads, it has a relatively large blade depth, roughly 2/3 of the radius, and the buckets fill completely with water. This means that the flow through the wheel is greater per revolution; however efficiency and flow rate are a function of speed of rotation. This type of wheel geometry is not entirely novel, as the hub’s relation to the water levels was originally conceived as the *Zuppingerad* shown in Figure 3. The unique feature of the SDM is its diagonally mounted blades which enter and exit the water continuously across the width of the wheel as it rotates, instead of at an instant as with traditional waterwheels. This allows the water to run into and out of the buckets more gradually and with less impulse. The blades are supported between two disks.

The inventor claimed that the SDM costs just 10-30 % of an equivalent turbine. Its efficiency would never drop beneath 90 % even when partially filled as the only source of loss was leakage through *air gaps*. This only loss was quantified as being 2-3 % and the maximum flow rate was stated to be 3 m³/s per m width. The explanation indicates that the SDM exploits the energy of the current, similar to a 'classic' stream wheel. The pilot program was considered to be a success and has been shut down as testing is complete. A study by Killer (2003) found that no data was available from the pilot program and access was not granted. It was concluded that the claims did not seem reasonable and that the true efficiency was not known. As a result he concluded that obtaining planning permission for the SDM would be very difficult.

When compared to the other new technologies under development the SDM has several advantages. It is a relatively simple machine operating with just one phase (water) and having a continual rotational motion allows direct energy conversion. Due to its full fronted water entry mechanism it is least likely to have an adverse impact on fish and sediment transport.

3 IDEALISED THEORY OF OPERATION FOR THE SDM

The SDM and its channel bed have geometric similarities with stream wheels and this has often created confusion. It must be noted that no similarities exist in their operation; a stream wheel extracts kinetic energy from fast flowing water by decelerating it. Its blade speed is approximately equal to the velocity of the downstream which is **less** than the upstream velocity. The SDM also has a blade speed approximately equal to the downstream velocity, which unlike the stream wheel is **greater** than the upstream velocity.

The SDM rotates because of forces generated on the blades as they pass underneath the hub. The largest force is generated by the difference in hydraulic pressure either side of the wheel, resulting from the dissimilar depths of water. This is partially counteracted by a reaction force to the acceleration of the water as it is forced underneath the hub, acting similar to a nozzle. These two forces are assessed below using the following idealised model which illustrates the working principle:

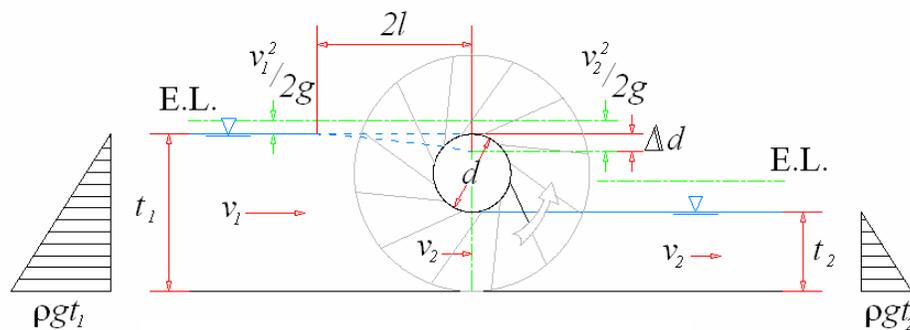


Fig.4. Idealised operation of the SDM

The model makes the following assumptions:

- The bearings are frictionless
- There are no losses as the blades enter and exit the water without influence
- The velocity of the blades as they pass beneath the hub is equal to the downstream water velocity, v_2
- There is no leakage around or underneath the wheel

Force due to hydraulic pressure differential

Referring to Figure 4 the head difference between the upstream and downstream water levels is $t_1 - t_2$. The flow rate Q and thus the speed v_2 are proportional to the rotational speed of the wheel. With the wheel stationary both v_1 and v_2 equal zero. As the wheel starts to rotate v_2 becomes greater than v_1 , and in accordance with continuity we get equation 1:

$$v_2 = \frac{t_1}{t_2} v_1 \quad (1)$$

Assuming that v_2 is achieved beneath the hub, this acceleration of the water leads to a drop in water level, Δh :

$$\Delta h = \frac{v_2^2 - v_1^2}{2g} \quad (2)$$

The force on the blades due to the hydraulic pressure F_P acting on the blade therefore drops ($F_P = \text{Pressure} \times \text{Area of blade } A$):

$$F_P = \rho g (t_1 - t_2 - \Delta h) A \quad (3)$$

Counter-acting acceleration force

The reaction force to the nozzle effect under the hub, F_A is equal to the mass flow rate \times the acceleration. The acceleration is calculated assuming that the known velocity change took place over length l equal to twice the diameter of the hub.

$$F_A = Q \rho \frac{v_2^2 - v_1^2}{2l} \quad (4)$$

Idealised Output Power

The idealised power output is the total force acting on the blade under the hub ($F_P - F_A$) multiplied by its velocity:

$$P_{out,ideal} = \left(A \rho g (t_1 - t_2 - \Delta h) - Q \rho \frac{v_2^2 - v_1^2}{2l} \right) v_2 \quad (5)$$

Efficiency

$$\eta = \frac{P_{out}}{P_{in}} = \frac{P_{out}}{(t_1 - t_2) \rho g Q} \quad (6)$$

The idealised efficiency curve is shown alongside scale model test results in section 5.

4 MODEL TESTING OF THE SDM

For the purposes of comparison, a $\sim 1/10$ scale model of the original SDM machine shown in Figure 1 was constructed for testing in a hydraulic flume. This model is shown in Figure 5. The flume was $12 \times 0.3 \times 0.3$ m in dimensions and had a maximum flow rate of 50 l/sec.

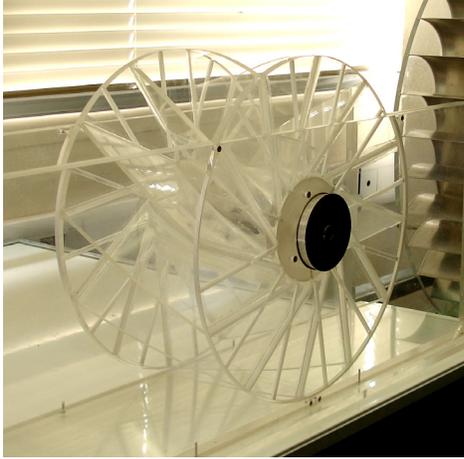


Fig.5. Scale model of the SDM

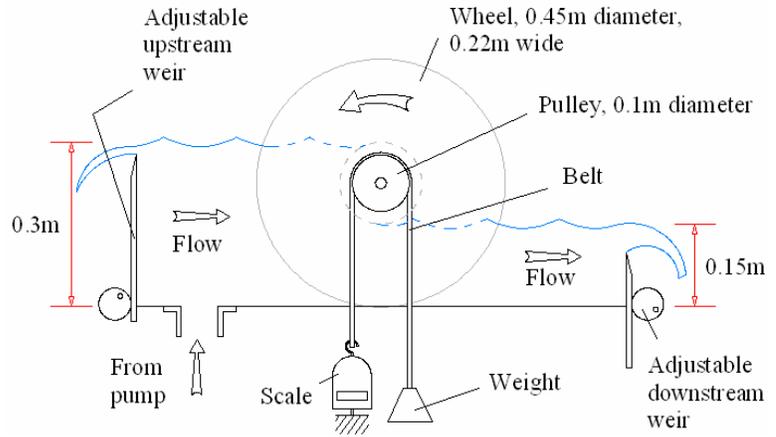


Fig.6. Setup of the scale tests

The experimental setup is depicted in Figure 6. Water from the pump entered the flume at the deeper upstream section, flowed through the SDM and exited via the shallower downstream section. Weights were added to the *Prony brake* power take off system to control the speed of rotation and measure the power output directly. This consisted of a pulley attached to the axel of the wheel, a weight and scale acting as a counter weight. The belt was made stationary, creating a friction force on the pulley equal to the difference between the two weights. Accordingly the measured power output was as follows:

$$P_{out,measured} = (w_1 - w_2) g 2 \pi r f \quad (7)$$

With: w_1 the mass of the weight (kg)
 w_2 the mass of the counter weight (kg)
 r the radius of the pulley (m)
 f the frequency of rotation ($1/s$)

For the SDM, the speed of rotation, flow rate, upstream and downstream levels are four dependent variables, each influencing the other three. To simplify the set up process, weights would be added to the *Prony brake* and the flow rate from the pump would be set artificially high. Any excess flow from the pump which would otherwise have caused the other three variables to change would effectively bypass the SDM model by flowing over the upstream weir. This weir was adjustable in height so that the upstream water level could be set to within 1 mm of its designed level. This was measured using 0.1 m diameter water columns connected to the base of the flume via small diameter tubes. The water in these columns was not influenced by the waves generated by the rotating wheel. An adjustable weir was also used to obtain the designed upstream water level with variable flows. This doubled as a sharp crested measurement weir so that all flow passing through the SDM was accurately measured. The model was supported by two plain nylon bearings to prevent leakage around the wheel. These had more friction than desired so the speed/friction relationship was measured and the power output value was adjusted to eliminate this loss.

5 RESULTS AND ANALYSIS OF THE SDM

This section compares the idealised theory from section 3 to the data obtained from the scale model tests. Disparity is observed and the idealised theory is advanced accordingly. As can be seen in Figure 7 the model tests achieved a maximum efficiency of ~40 % and a maximum flow rate of 12 l/sec. Converted using the Froude scaling laws this is equivalent to 1.6 m³/s per m width for the original 4.25 m diameter wheel, roughly half that claimed by the inventor.

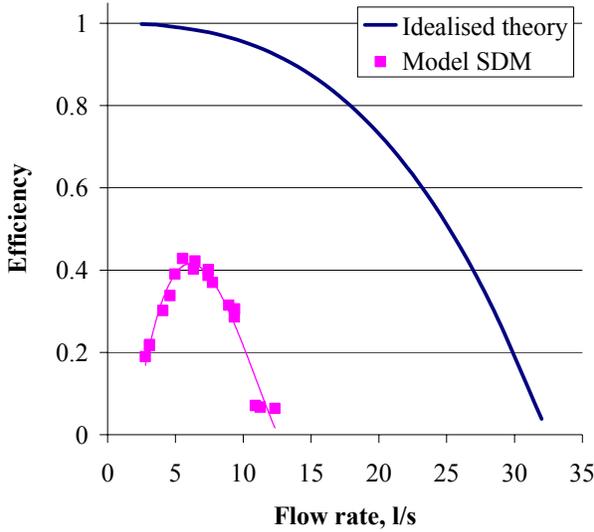


Fig.7. Ideal and scale model efficiency curves

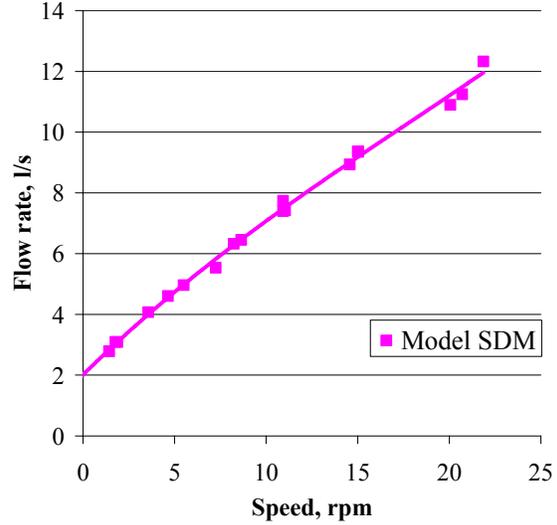


Fig.8. Scale model: flow vs. speed

The idealised theory and the model results shown in Figure 7 do not match well because two of the original assumptions were unrealistic.

The first of these assumptions was that there was no leakage under and around the wheel. Figure 8 indicates that the leakage when the wheel was not rotating was 2 l/sec or 1/6 of Q_{\max} . The driving force behind the leakage is the difference in water levels either side of the wheel. As these levels are maintained at all flow rates the leakage is constant regardless of rotational speed. Therefore the idealised output power in equation 5 can be modified to take this constant leakage into account. See equation 8 which includes the leakage flow term Q_l .

$$P_{out\ modified\ 1} = (F_P - F_A)v_2 - ((t_1 - t_2) \rho g Q_l) \quad (8)$$

Figure 9 has the leakage flow Q_l set to 2 l/sec, which shows the improved correlation between the modified theory and the model results.

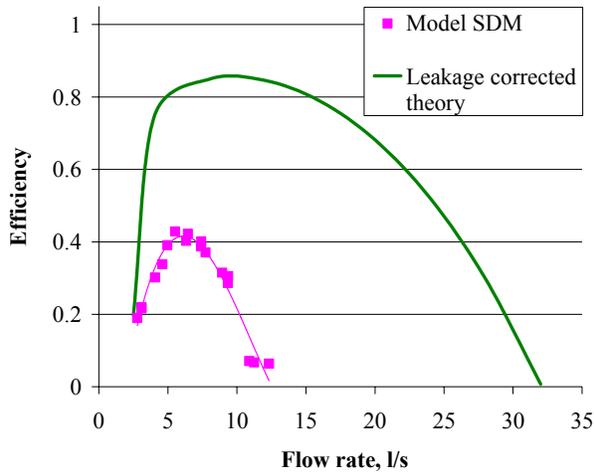


Fig.9. Leakage modified theory and scale model efficiency curves

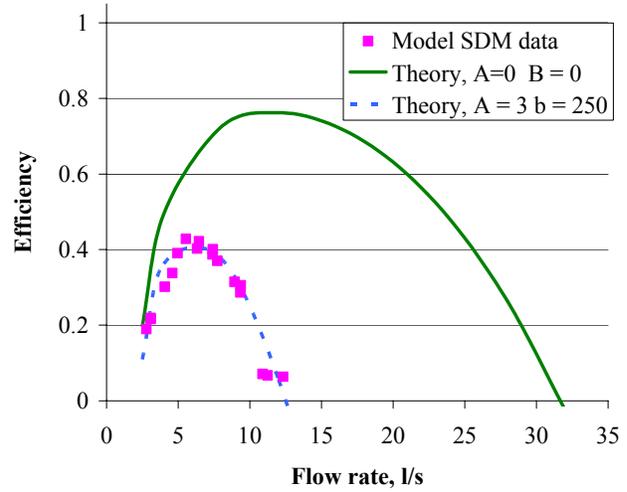


Fig.10. Modified theory adjusted to match scale model data

The second unrealistic assumption was that the blades would enter and exit without losses. In reality the blades would enter and exit with significant losses producing additional counteracting forces to the rotation of the wheel. Some of them were independent of speed (*e.g.* leakage) and others were proportional to speed (*e.g.* turbulent losses). Therefore equation 8 is modified further giving equation 9. This then matches the model tests using a correction factor to the force term:

$$P_{out\ modified\ 2} = (F_P - F_A - (A + Bv_2^2))v_2 - ((t_1 - t_2)\rho g Q_l) \quad (9)$$

A and B are empirical constants, which have been adjusted to the data in Figure 10.

Before describing the blade losses it must be noted that as the SDM has diagonal blades, one side of each bucket always enters and exits before the other. Also, because these diagonal blades consist of flat plates in a single plane (see Figure 1), they do not protrude radially outwards from the hub across the width of the wheel. Instead they appear to lean forward when compared to the radial on one side of the wheel, and backwards on the other, see Figures 11 and 12. This results in different forms of losses on either side of the wheel.

On the side of the wheel where the buckets initially enter and exit the water, losses occurred with the passing of each blade. Figure 11 shows that for every blade, a volume of water i was initially displaced, only to be filled when the blade tip became level with the surface of the water. During this time a pressure force was exerted on the underside of the blade, counteracting rotation. This loss occurred at all speeds of rotation. When the water did enter this volume it ran down the top surface of the blade and impacted the subsequent blade, also counteracting rotation. This loss increased with the speed of rotation. This periodic filling mechanism produced waves in the upstream with an amplitude up to $1/15^{\text{th}}$ of the head difference. At the downstream side of the wheel, a volume of water o was raised with the passing of each blade. This required significant work by the wheel which was lost as soon as the blade tip passed the downstream water level, allowing air to ventilate the bucket. This loss also occurred at all speeds, and is also the main

cause of fast travelling waves at the downstream, and a pulsating rotation that would have negative effects on any power train connected to the SDM.

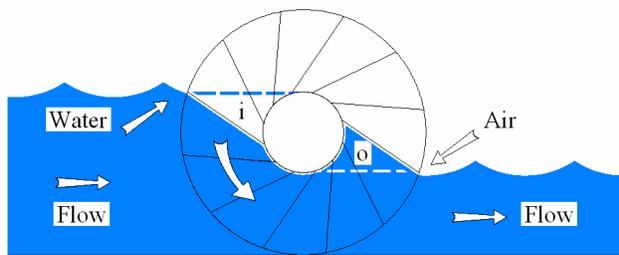


Fig.11. SDM losses 1

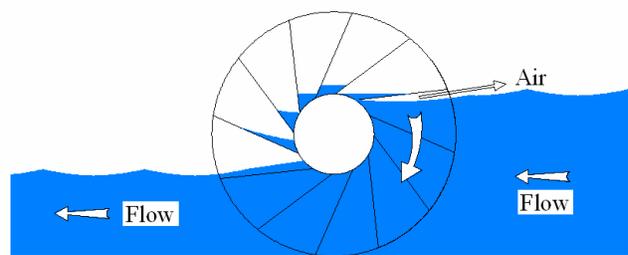


Fig.12. SDM losses 2

On the other side of the wheel where the end of each blade exits and enters the water, two final sources of loss remain. Referring to Figure 12, at the downstream side, a small quantity of water failed to run off the blade. This water was then effectively pumped back into the upstream. At the upstream, as the blades approached the water level and their buckets became fully filled, the angle of the blades being practically parallel to the water surface would eject fast blasts of air. This created turbulence in the water and introduced air into the flow. Both of these problems increased with the speed of the rotation.

All of the above losses were counter productive to the smooth running of the wheel. They not only reduced the efficiency but also the speed of the wheel, thus reducing its flow capacity.

6 THE ROTARY HYDRAULIC PRESSURE MACHINE

Two fundamental modifications have been determined to significantly reduce all of the losses witnessed on the SDM scale model in section 5. The first of these has been assessed and the second has yet to be implemented. The resulting machine, the *Rotary Hydraulic Pressure Machine* (RHPM) is expected to have efficiency and flow characteristics appreciably closer to that predicted by the idealised theory than the SDM.

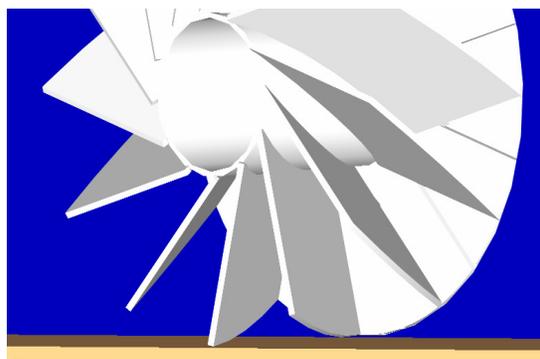


Fig.13. SDM leakage between blades



Fig.14. Shroud used in model tests

The first modification addresses unnecessary leakage underneath the wheel. Figure 13 demonstrates that diagonal blades in combination with a flat channel bed leaves passageways underneath the hub which allow water to flow along the blades without acting upon them. This

was addressed by creating a shroud. The shroud is a curved section of channel bed where the arc is equal in length to the arc between the two extreme corners of a bucket, see Figure 14.

Figure 15 shows that the addition of the shroud reduced the overall leakage by half, and the maximum efficiency of the wheel increased to just under 60 %. The maximum speed of rotation however did not increase as the blade inlet/exit losses are still dominant.

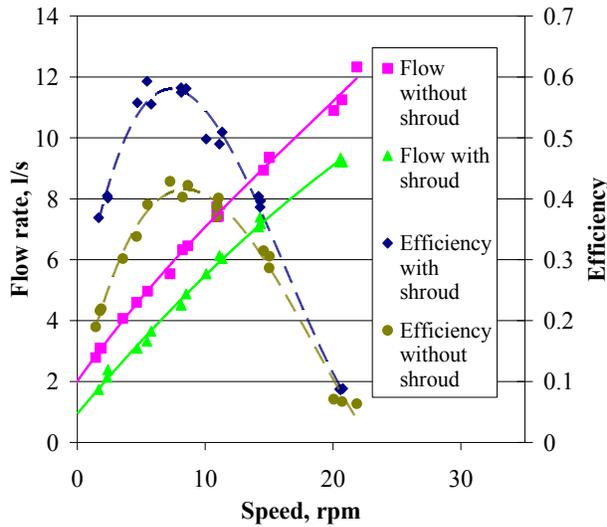


Fig.15. Model flow and efficiency curves with and without shroud

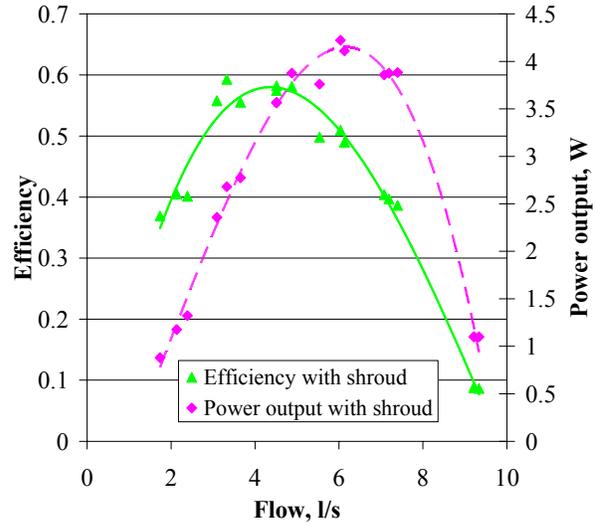
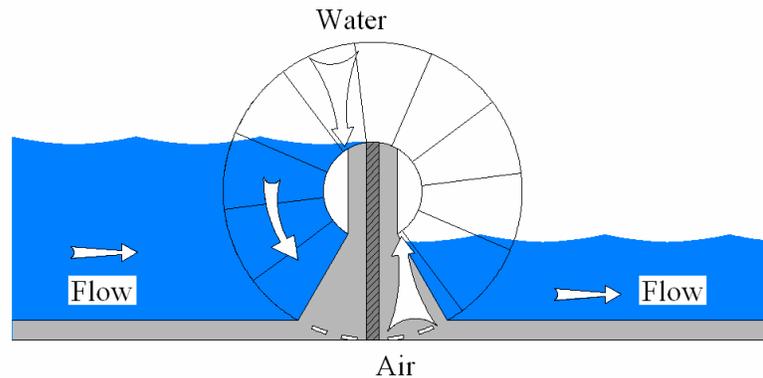


Fig.16. Model efficiency and power output curves with shroud

Figure 16 shows that the power output peaks at a higher flow rate than the efficiency. It is a function of flow and efficiency and can only be extended by addressing the problems highlighted by Figures 13 and 14. The second proposed modification is expected to alleviate these as it allows filling and venting from the sides of the wheel. This will increase the efficiency at all flows and raise the maximum speed of rotation and thus flow capacity. It is shown in Figure 17 and has not yet been implemented. It requires the removal of the side disks seen in Figure 1 and a new wheel support structure. This will be similar to the shroud, preventing leakage through the buckets from their sides. The blades will protrude radially from the hub (and therefore twist along the width of the wheel) in order to minimise the required size of the support structure which must also prevent leakage.

Fig.17. Proposed *Rotary Hydraulic Pressure Machine* with shroud and side filling and venting



7 APPLICATION OF THE ROTARY HYDRAULIC PRESSURE MACHINE

The RHPM is a very simple machine with few components requiring a simple support structure. The flow capacity is greater than with traditional waterwheels and the downstream flow velocity is not as fast as that from most turbines which can cause scouring. The way in which water flows into and out of the machine should not have any adverse effects on sediment transport or fish passage. These points make it suitable for relatively inexpensive installation into existing weir structures. Another unusual feature of the RHPM is that it is symmetrical. This means that it is fully reversible, being able to work in both directions of flow. Such ability could make it suitable for tidal flow scenarios.

8 CONCLUSIONS

The *Staudruckmaschine* does not have the operational characteristics claimed by its inventor. Many sources of loss have been observed using a scale model and two sets of modifications to its design have been determined. The result is a new machine called the *Rotary Hydraulic Pressure Machine*. Compared to the SDM, this is expected to have peak efficiency more than 20 % higher, a greater flow capacity per metre width of wheel, and a smoother less pulsating rotation. A theory has also been proposed to model the operation the SDM and the RHPM, both of which operate because of hydraulic pressure. A market for the RHPM exists as it can be used in scenarios where neither traditional waterwheels nor turbines would be economical. This is because of its simplicity in design, high flow capacity and ability to handle low heads. This market includes opportunities for installation into existing weir structures and use in tidal flow scenarios. Further information is available at www.lowheadhydropower.com.

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