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How to design a centrifugal pump with constant power consumption for all flow rates

Marcus Beck[1], Paul Uwe Thamsen*[1]

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Abstract

This paper demonstrates a new hydraulic design of a centrifugal pump for drum pump application with constant power consumption for all flow rates. The advantage of constant power is a smaller electrical motor size and also constant speed for all operation points. Thus the cooling effect remains effective and the noise emission is low. This approach is quite new, because usually the hydraulic design focuses on best efficiency. Different design parameters are systematically investigated, which results in approximately eleven different semi-axial type impellers combined with a new diffuser. All impellers are tested in a special test stand which allows a detailed loss analysis to determine the pump efficiency. The results are displayed and discussed, which help to extract some guidance for design of such pumps.

Keywords

Constant power consumption — Semi-axial — Design — Experiment — Centrifugal Pump

[1] *Department of Fluid System Dynamics, Technische Universitaet Berlin, Berlin, Germany*

***Corresponding author**: paul-uwe.thamsen@tu-berlin.de

INTRODUCTION

The power consumption of centrifugal pumps depends mainly on the type of impeller. Typically it increases with the flow rate of radial impellers and it decreases with the flow rate of axial impellers. However, as shown in Figure 1, each type of impeller shows a significant maximal power consumption, which defines the adequate motor power for the driver.

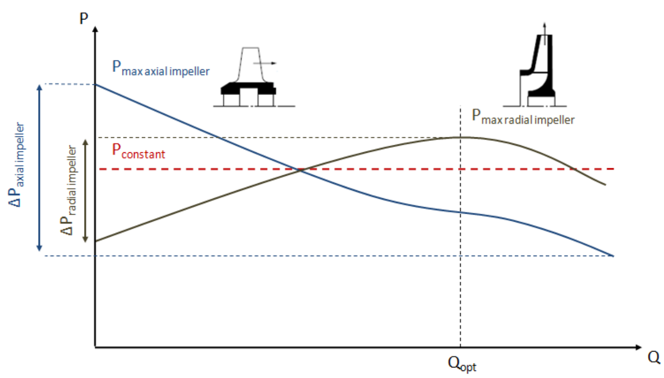

Figure 1. Schematic diagram of typical power consumption curves for axial and radial impeller pumps

Constant power consumption for a centrifugal pump is a new approach for impeller design because until today design of centrifugal pumps is focused on high efficiency performance. Additional challenge for this project was the application of a drum pump, which is a relative small pump with an impeller diameter of about 38 mm. It is known, that the design method for such small pump deviates significantly from the standard design process, whereas the slip factor is much higher.

Liu, Nishi and Yoshida [1] report about new semi-axial impeller designs for mini pumps and *Bischoff* [2] presents the design process for small cooling pumps. Mini pumps have limited efficiency, so other priorities like small motor size are interesting too.

Semi-axial impellers already show relative horizontal power consumption. *Gülich* [3] presents a diagram in his book, which recommends a specific speed of about n_q = 100 as a good starting point (Figure 2).

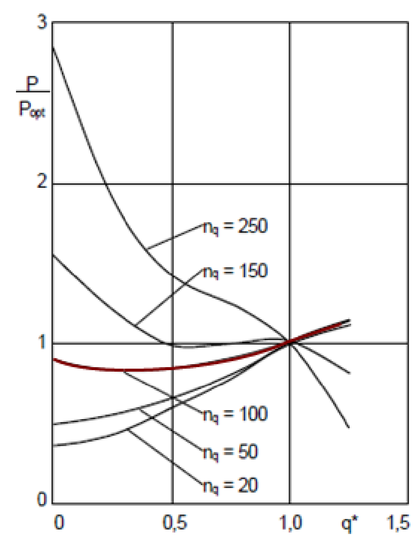

Figure 2. Diagram of theoretical power curves for different specific speed numbers

The onset and behaviour of recirculation at part load has a tremendous influence on the Head generation as well as the power consumption at part load for semi-axial pumps. This is reported in *Troskolanski et. al.* [4] and also in *Gülich* [3]. The later introduces a recirculation power as a power loss that is demonstrated in Figure 4.

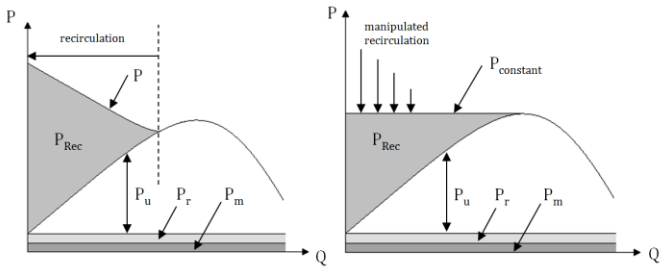

Figure 4. Power curve separated in power shares: recirculation loss, hydraulic power, friction loss and mechanical loss

The left sides of the diagrams show the large influence of part load recirculation on the power consumption. So it is obvious to focus on this effect to design the impeller for constant power consumption. There are only few aspects reported to influence recirculation flow:

- decreasing impeller inlet diameter d_s
- decreasing inlet cross section d_s/d_{1i}
- generate rotation before inlet
- increasing blade inlet diameter d_{1i}
- changing inlet cross section of diffuser

Because of its complex 3d-characteristic until today recirculation flow and the influence on power curves is not completely understand.

1. PUMP DESIGN

Standard impellers of drum pumps are typically radial for low flow and higher pressure or axial for large flow and moderate head. The electrical motor is selected due to maximum power consumption of the pump, which is for both impeller types significantly above the required consumption for the duty point. Additionally the speed of the motor follows the power consumption curve, so at high motor load, the speed of the motor and moreover the speed of the cooling fan is low. This affects the cooling of the motor and its lifetime.

The constant power consumption requires obviously a semi-axial design, which also gives the advantage of only one universal impeller design for all required operation points. Based on the relative large axial tolerance of the long shaft of a drum pump, the impeller gap needs also a large tolerance. Last, the inlet and outlet of the semi-axial impeller should be in axial direction, because the complete pump is mounted in a relatively simple pipe as pump housing. Based on the said boundary conditions, the best impeller design in meridional sketch is shown in Figure 5. It results in a semi-axial impeller with open axial inflow and diagonal outflow into a bladed diffuser. The impeller has a rotating inflow contour and a constant axial gap.

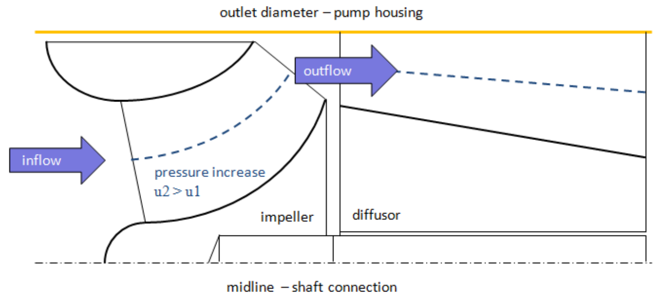

Figure 5. basic concept for the new impeller design

To influence the recirculation power of the impeller to reach nearly constant power consumption a variation of a number of design parameters is necessary. Comparable to Figure 6 the following design parameters are varied:

- number of blades z,
- inlet inner diameter d_{1i},
- suction mouth diameter d_s and
- outlet width b_2.

Additionally the influence of backup blades is investigated. Other geometric parameters like outlet diameter or blade angels of impeller and diffuser remain constant.

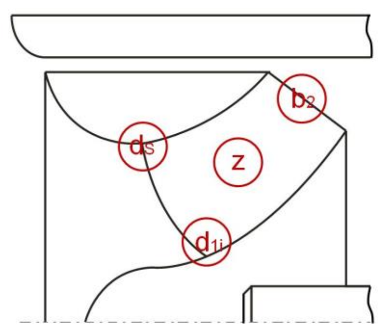

Figure 6. Overview of investigated design parameters

The design point is illustrated in Figure 7 and the specific speed number is n_q = 90.

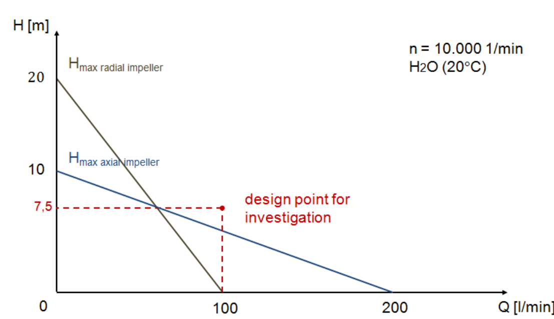

Figure 7. Selected design point

The design process follows the same procedure for all impellers according to Figure 8. It is based on stream line theory with a slip factor, described in standard literature like *Pfleiderer* [5], *Neumann* [6] *Troskolanski* [4] or *Gülich* [3].

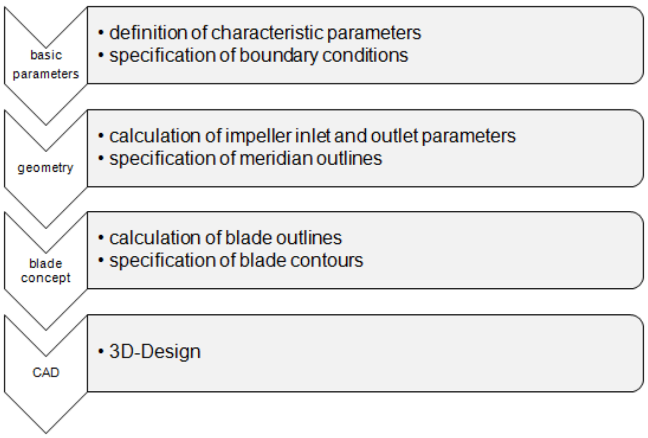

Figure 8. Impeller design procedure

Finally, eleven different impellers were designed and manufactured via rapid prototyping. Figure 9 gives the overview of all manufactured impellers and their diversified characteristics.

Figure 9. Impellers of study
(left: front view without shroud right: meridian view)

All designed impellers and one new diffuser were tested in a special designed closed test loop as shown in Figure 10.
A horizontal acrylic pipe in same dimensions like the drum pump casing is installed in a larger pipe, which is similar to a drum pump application. A long shaft sealed within a 90°-bend and carried by a bearing arrangement is driven by a high speed power unit via a torque sensor. Flow measurement with magnetic inductive flowmeter and pressure transducers on suction and discharge characterize all required performance data from the pump. The measurements of the test stand allow a detailed loss analysis.

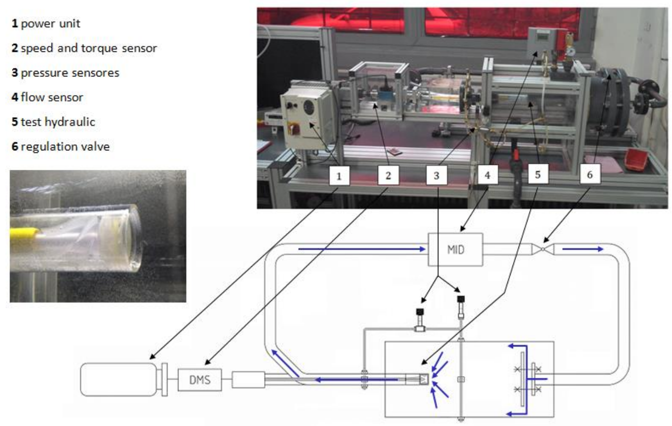

Figure 10. Test stand

Finally, some selected impellers are installed in an additional test stand, which is a real drum pump operation Figure 11 shows the test stand and the new hydraulic on the underside of a drum pump.

Figure 11. Drum pump test stand

2. RESULTS

Impellers are characterised by the head, power at shaft and the pump efficiency. All measurements are taken at the same speed n = 10.000 rpm.

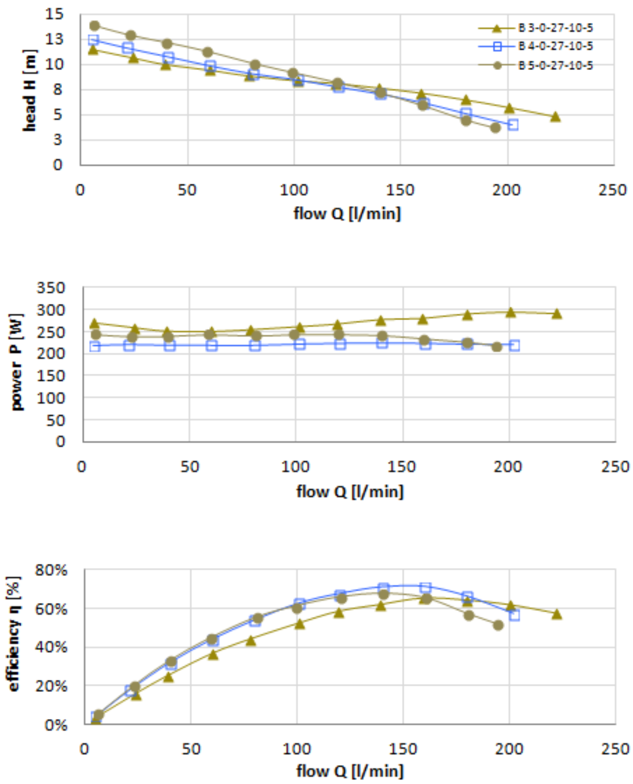

Figure 12. Head, power and efficiency over flow for blade numbers 3, 4 and 5

The comparison of the three blade numbers (Figure 12) represents that nearly constant power consumption could be reached with 4 blades. Power consumption increases with 3 blades and decreases with 5 blades at higher flow rates. Highest efficiency could also be reached with 4 blades. Additionally, it could be pointed out, that the impeller with 3 blades is close to axial and with 5 blades is already close to radial characteristic.

From literature the enormous influence of the blade number is known. That means flow control, vortex areas, friction loss and effects by slip. Together these have oppositional influences so that finally a compromise between all factors allows finding the optimum regarding efficiency and pump power.

In this development, the blade number 4 was selected as the best compromise and all further investigations are done with 4 blades.

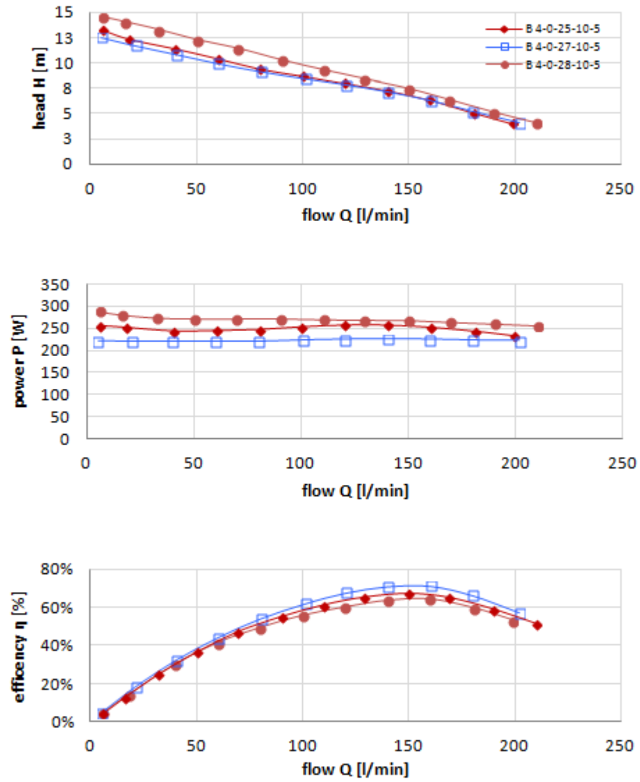

Figure 13. Head, power and efficiency over flow for d_s = 25 mm, 27 mm and 28 mm

The comparison of the three suction mouth diameters (Figure 13) represents that nearly constant power consumption could be reached with all three impellers. The amount of power is approximately 20 W - 50 W less for suction mouth diameter 27 mm than other sizes so there could be reached highest efficiency.

Measurements show that impeller with d_S = 28 mm has superficial recirculation and the higher circumferential velocity may result in higher shock losses. The impeller with d_S = 25 mm has higher inlet speed, which also tends to higher inlet losses. This means a suction mouth diameter with d_S = 27 mm is the best compromise.

The comparison of the three inlet inner diameters (Figure 14) shows that nearly constant power consumption could be reached with all three impellers. The power consumption of inlet inner diameter d_{1i} = 10 mm is approximately 20 W less resulting in the highest efficiency for this setup.

A displacement of the leading edge into suction mouth or blade channel has direct influence on blade length linked to friction loss, flow deviation and slip.

The measurement shows that the position of leading edge is elected well for inlet diameter d_{1i} = 10 mm.

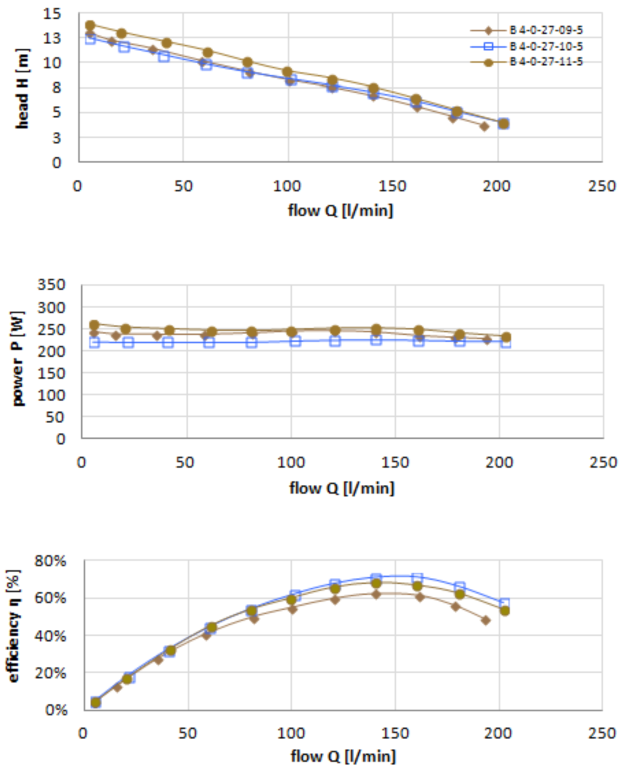

Figure 14. Head, power and efficiency over flow for d_{1i} = 9 mm, 10 mm and 11 mm

The comparison of the impellers with 3 and 4 blades and impellers with additional backup-blades (Figure 15) demonstrates that nearly constant power consumption could be reached by the impellers with backup-blades. The power curves have lower bandwidth over flow than without backup blades and decrease with higher flow rates. Highest efficiency could be reached with 4 blades and without backup-blades.

Backup-blades improve flow control. At the same time lower blade pressure develops linked to minor slip that increased head and tends to bold head curves as recognized in Figure 15. The increase of friction loss as a consequence of more blades limits the flow rate noticeable.

The measurements show that backup-blades improve flow control especially for lower blade numbers and reach constant power consumption. However, the additional blades increase power loss that finally reduces efficiency.

No impeller with backup-blades is preferred because there is still one impeller without backup-blades that has constant power consumption with higher efficiency.

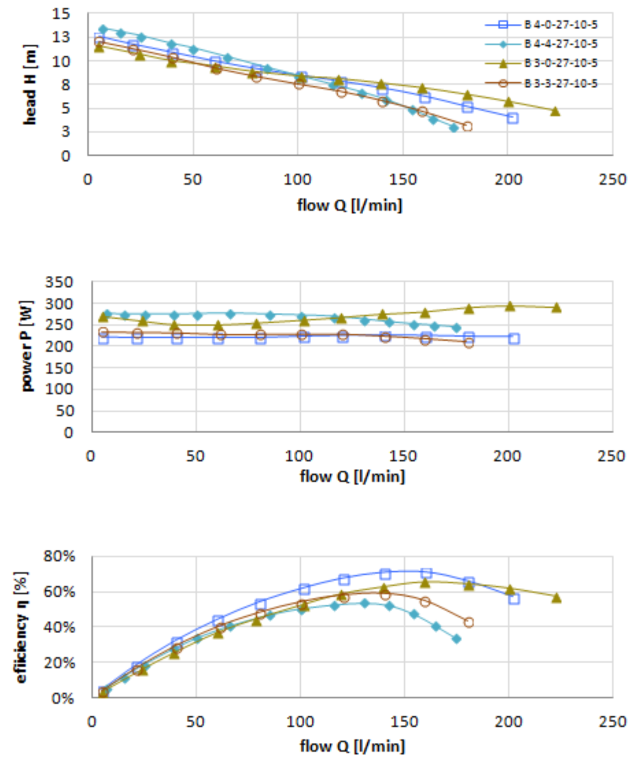

Figure 15. Head, power and efficiency over flow for blades 3 and 4 and backup-blades 3 and 4

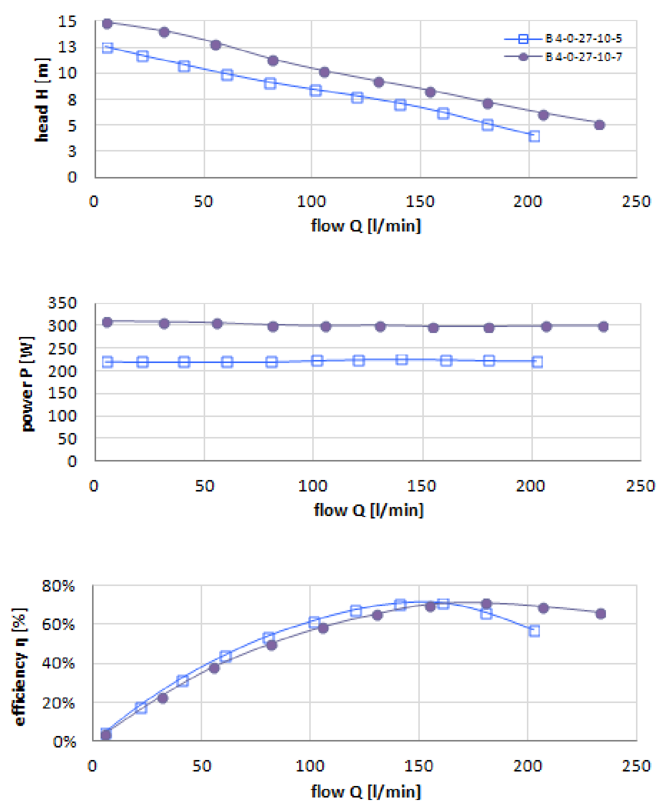

Figure 16. Head, power and efficiency over flow for outlet width 5 mm and 7 mm

The comparison of the impellers with different outlet width (Figure 16) displays for both outlet widths nearly constant power consumption. For outlet width 7 mm, a higher head and in conjunction, the power consumption increased. Highest efficiency could be reached with both impellers unlike highest amount is displaced to higher flow rates for outlet width 7 mm.

The proportion between inlet and outlet cross section is influenced with increasing outlet width to that effect that simultaneous meridian speed is decreasing and rotation speed linked to head is increasing. Through resulting enhanced blade channel flow loss is increased too.
This correlation is also shown in Figure 15.

Altogether there is no direct correlation seen between outlet with and power curve trend. In that case the impeller with outlet width b_2 = 5 mm is preferred because of lower power consumption reaching the required head.

Finally impeller B4-0-27-10-5 has been elected as the best result of all impellers.

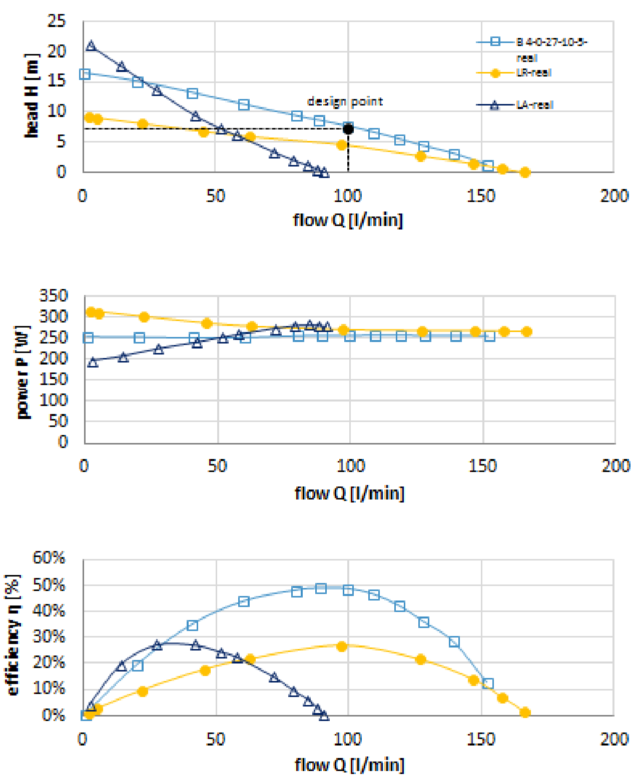

Figure 17. Head, power and efficiency over flow for radial, axial and new semi-axial impeller

Figure 17 shows the results of the measurements in the second test stand in a real drum pump operation and comparison with standard radial and axial impellers.

It should be pointed out, that the rotational speeds of the standard impeller are significantly higher – up to 2.000rpm above the new semi-axial impeller. Thus, the generated noise is also reduced.

The comparison of the curves underlines the tremendous improvement of the new impeller design for the drum pump:

- constant power consumption for all operation points,
- coverage of head-flow requirement radial/axial and
- efficiency increase by more than 20 points.

Overall the new impeller reduces the required motor power significantly and there will be no speed change during operation. Thus the cooling effect remains good for all operation points and the lower speed reduces the noise emission.

3. DISCUSSION

The project demonstrates that it is possible to find an impeller design with nearly constant power consumption for all flow rates.

Regarding literature a semi-axial impeller around n_q = 100 could bring high potential for this investigation n_q = 90 the intention has been reached. Using the standard design process based on the stream line theory was suitable for the special designed semi-axial impellers.

The variation of a number of design parameters shows that complex interactions of several effects influence the power curve so it is necessary to equilibrate them.

It could be shown that for this impeller within a drum pump application blade number has to be four and symmetrical. Unlike the constant power consumption couldn’t be reached. Additional backup-blades are usefully for low blade numbers to generate nearly constant power curve. The disadvantage of backup-blades is the increase of power linked to decreasing efficiency.

The inlet needs an optimization process. This investigation shows that smaller inlets increase head with less efficiency and bigger inlets loose the effect of constant power with less efficiency. The outlet has no influence to reach a constant power curve.

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NOMENCLATUR

BEP	best efficiency point
b_2	outled width in mm
d_{1i}	inled diameter in mm
d_2	outled diameter in mm
d_s	suction mouth diameter in mm
H	head in m
n	rotational speed in 1/s
n_q	specific speed number
Q	flow rate in m³/h
Q_{opt}	flow rate in BEP in m³/h
P	power consumption in W
P_{max}	maximal power consumption in W
P_{opt}	power consumption in BEP in W
q*	specific flow rate
z	number of blades
z_s	number of backup blades
η	efficiency