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# Experimental evaluation of hydraulic design modifications of radial centrifugal pumps

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**Abstract.** In the field of small hydropower, pump units in turbine mode (PAT) are frequently used as alternative to conventional turbines. In order to maximize their operation efficiency, it is possible to optimize the performances of these engines through various simple innovative modifications which relate mainly to the geometry of the flow parts. This paper deals with the results of several successful modifications verified on one such engine. While various simple modifications led to the increase of overall efficiency and power output by few percentages, power output increased by nearly 30% with the innovated runner blades geometry. The modifications also had positive effects on the pump's operation, with significant increases in flow rate, head and total efficiency.

**Key words:** pump as turbine (PAT), efficiency, inovation, turbine mode, pump mode.

## INTRODUCTION

One major driving force for innovations has been the effort to improve the efficiency of machines in operation, a trend which in recent time has been supported by EU legislation (Directive 2009/125/EC). Regulations focused specifically on hydraulic machines (Commission regulation (EU) No 547/2012) specify a so called minimum energy efficiency index (MEI) of at least 0.1 which means in practice that 10% of pumps with the lowest efficiency must be withdrawn from the EU market. From January 1, 2015, this index must be at least 0.4, which means it will be necessary to replace 40% of the pumps on the European market (Nevěřil, 2012).

Centrifugal pumps are often used in hydropower as alternative to conventional water engines due to their reversibility – their ability to convert hydraulic energy to mechanical energy. Such applications are financially highly attractive alternatives to conventional turbines. Properties of pumps in turbine mode are described in literature (Alatore 1994; Derakhshan, et al. 2008; Sedlář 2009; Bláha 2011; Nautiyal, et al. 2011; Singh & Nestmann, 2011; Pochylý, et al. 2013; Raman, et al. 2013). Since smaller pumps usually have lower efficiency, the issue of their optimization in turbine mode can be targeted. Although some modifications have been tested and described (Gülich 2003; Singh 2005; Singh, et al. 2012), results of detailed studies on the subject are still missing in literature. This article follows on research published in (Poláková & Polák, 2016).

## MATERIALS AND METHODS

### Pump modifications and verification tests

Based on above mentioned existing researches and own hydraulic calculations, innovative modifications for pump optimization were proposed. They are designed primarily to increase the efficiency of the turbine pump operation but, at the same time not to reduce the effectiveness of the pumping operation. The concept of the modifications is based on the theory of fluid flow through radial hydrodynamic machines according to Hýbl (1928), Nechleba (1962), Nechleba (1966), Melichar et al. (1998), Munson et al. (2006), Ulrych (2007), Polák & Polak (2010). Other minor modifications resulted from technical documentation and the experiences described in Singh (2005), Polák (2013), Poláková & Polák (2016). Modifications were designed to be, in terms of feasibility as simple as possible while being at the same timeapplicable to already existing pumps. Summary description of the two 'most successful' modifications is given below in Table 1.

**Table 1.** Summary of tested variations

| Variation | Description |
|---|---|
| A | Original unmodified pump – in characteristics indicated by dashed lines. |
| B | Pump with modified impeller and modified spiral casing – in characteristics indicated by full lines. Modifications consisted in arrangements that led to reduction of local and friction losses in impeller and spiral casing |
| C | Pump with new impeller blades geometry and modified spiral casing – in characteristics indicated by double lines. New geometry of the blades was designed on the base of the theory of fluid flow in specific low-speed hydrodynamic machines. Spiral casing modification is the same as in variation B – minimization of friction and local losses |

For experimental verification of the proposed modifications, a single-stage centrifugal pump with spiral casing was selected. The pump's scheme, including its parameters as provided by the manufacturer is shown in Fig. 1.

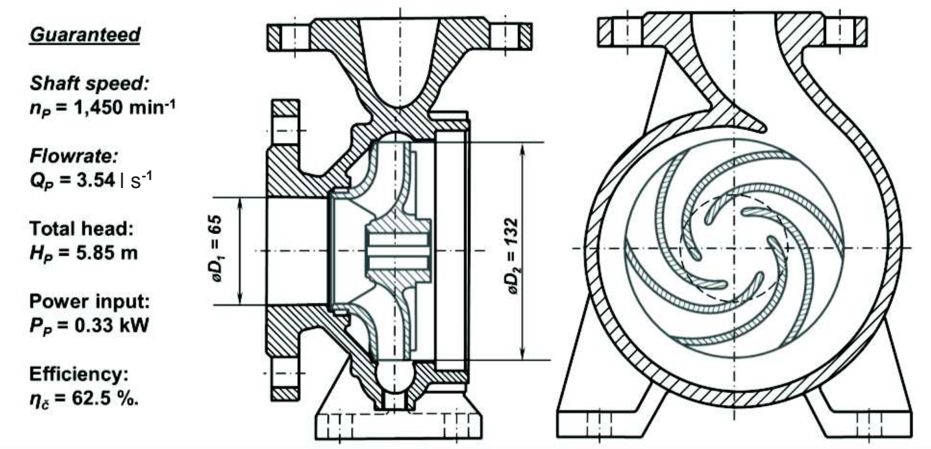


**Figure 1.** Centrifugal pump for experimental tests and original pump parameters.

Verification tests were conducted on a hydraulic circuit in the Fluid Mechanics laboratory at the Faculty of Engineering, Czech University of Life Sciences Prague. The circuit diagram is shown in Fig. 2. In the first stage characteristics of the pump and its modifications in the turbine mode were measured. Subsequently the effect of the modifications in the pump operation was verified.

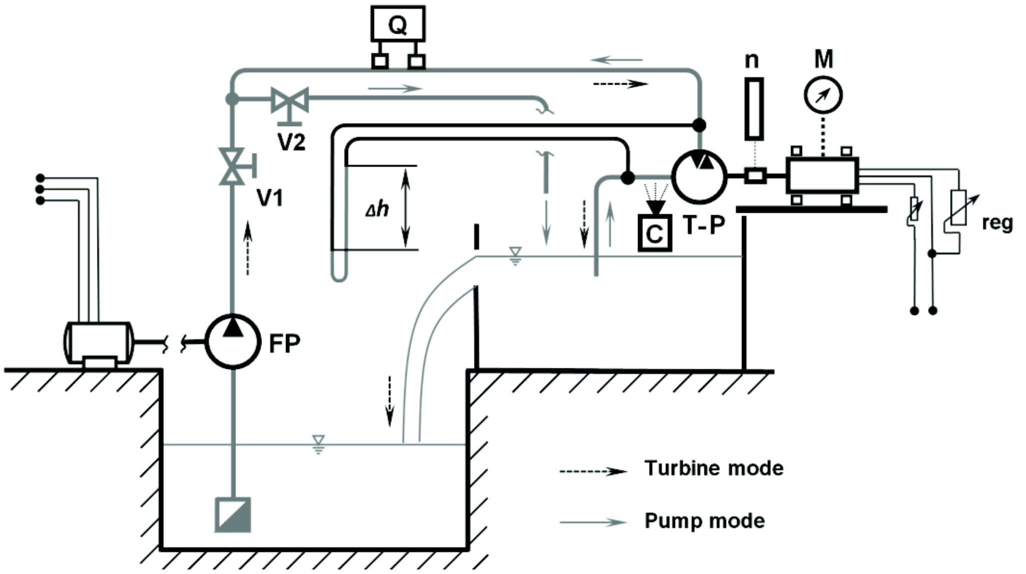


**Figure 2.** Hydraulic circuit scheme for testing turbines/pumps: Q – flowmeter; FP – feed pump (for turbine mode); T–P tested pump/turbine; V1, V2 – control valves; M – dynamometer, *n* – revolution counter, reg – load/drive control of the turbine/pump.

The testing circuit consisted of a set of two reservoirs with pipes and control and measuring elements. With this setting the tested pump (T-P) was measured in turbine mode – by closing valve V2 while regulating valve V1, the water flows in the direction of dashed arrows, while the feeding pump (FP) creates the hydro-technical potential for the turbine. After rearranging the valves V1and V2, the machine (T-P) was tested in the pumping operation on the same circuit – by closing the valve V1 and controlling valve V2 the water flows in the direction of gray arrows. The dynamometer (M) with continuous revolutions control allows operation in motor and braking mode. The dynamometer has cylindrical stator. It is placed on a bed allowing a slight rotation, which enables a measurement of the reaction torque at load. Dynamometer is a DC machine connected to continuous resistance load control unit. The water flow was measured using an ultrasonic flowmeter, Q (the Siemens SITRANS FUP1010). Specific energy of the turbine was determined on the basis of flow rate, $Q_T$ and the value of differential pressure of mercury manometer, $\Delta$h connected to the pipe by collecting probes according to (ČSN EN ISO 9906; Hodák 1982), as detailed in Fig. 4. The turbine shaft speed was measured using an infrared sensor (TESTO 465).

In addition to measured values, flow character in both operating modes was recorded with a CCD camera (C) in transparent flow suction tube. Measured performance parameters were synchronously assessed with camera records and then used to design other optimization. Two images illustrating the flow in turbine and consequently pump mode can be seen in Fig. 3.

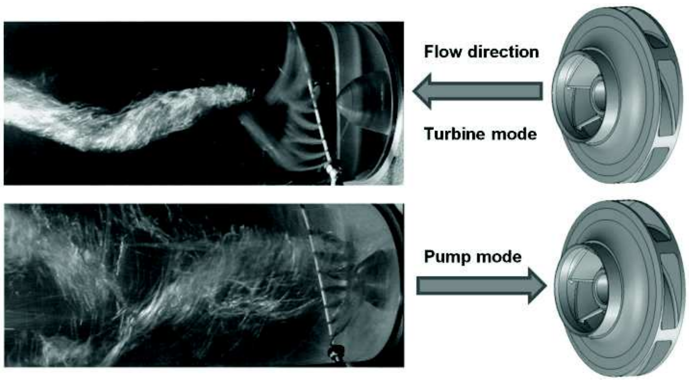


**Figure 3.** Flow inside the suction tube in turbine and pump operation.

Based on the measured values of differential pressure Δh, water flow Q, shaft speed $n$ and torque $M_T$ on dynamometer, other parameters were calculated for the purpose of creating characteristics in both operating modes of the machine.

Power of the water flow transferred to the turbine (or passed on by the pump) is determined from specific energy according to Bernoulli's equation and from measured values by calculation:

$$P_{T-P} = Q \cdot \rho_W \cdot \left[\frac{\rho_{Hg}}{\rho_W} \cdot \Delta h \cdot g + \frac{8 \cdot Q^2}{\pi^2}\left(\frac{1}{d_p^4} - \frac{1}{d_s^4}\right) + g \cdot y\right] \ [W] \tag{1}$$

where: $Q$ – volumetric flow rate [$m^3\ s^{-1}$]; $\rho_{Hg}$ – specific weight of mercury ($\rho_{Hg}$ = 13,540 kg $m^{-3}$ at 20 °C); $\rho_W$ – specific weight of water [kg $m^{-3}$]; $\Delta h$ – measured level difference in U-pipes [m]; $d_p,\ d_s$ – inner diameter of pressure and suction pipe respectively [m $s^{-1}$]; $y$ – vertical distance of zones generating pressure [m] (see Fig. 4).

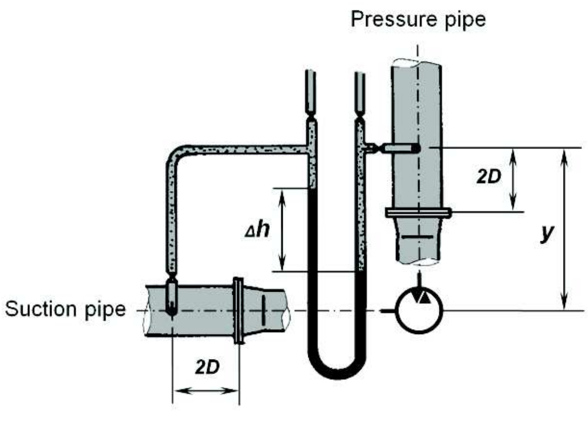


**Figure 4.** Diagram of pressure measurements according to EN ISO 9906.

The mechanical power output/input of the machine is given by torque $M_T$ and shaft speed, n. The overall efficiency of the machine is then expressed by the ratio of mechanical and hydraulic power.

## RESULTS AND DISCUSSION

Before beginning with verification of the individual innovative modifications, the properties of original unmodified machine were tested on the hydraulic circuit – both in turbine and pump mode. Thus obtained characteristics became a basis for assessment of the influence of individual modifications. They were then tested on the same test circuit under the same conditions as the original, unmodified machine – first in turbine and subsequently in pump mode. A number of modifications were experimentally tested in this way but only those with the best influence were selected for the following evaluation. Descriptions of all tested variations are summarized in Table 1. More details of structural design can not be published because they are subjected to law protection of intellectual ownership.

The very measurements in turbine mode were conducted so that the machine was gradually loaded from idle speed up to 900, or 800 rpm. The test results (Figs 5 & 6) illustrate the basic performance characteristics, mainly efficiency, power output, torque, and flow rate in dependence to shaft speed. Optimal operation or best efficiency points (BEP) are indicated for the characteristics of the three variants.

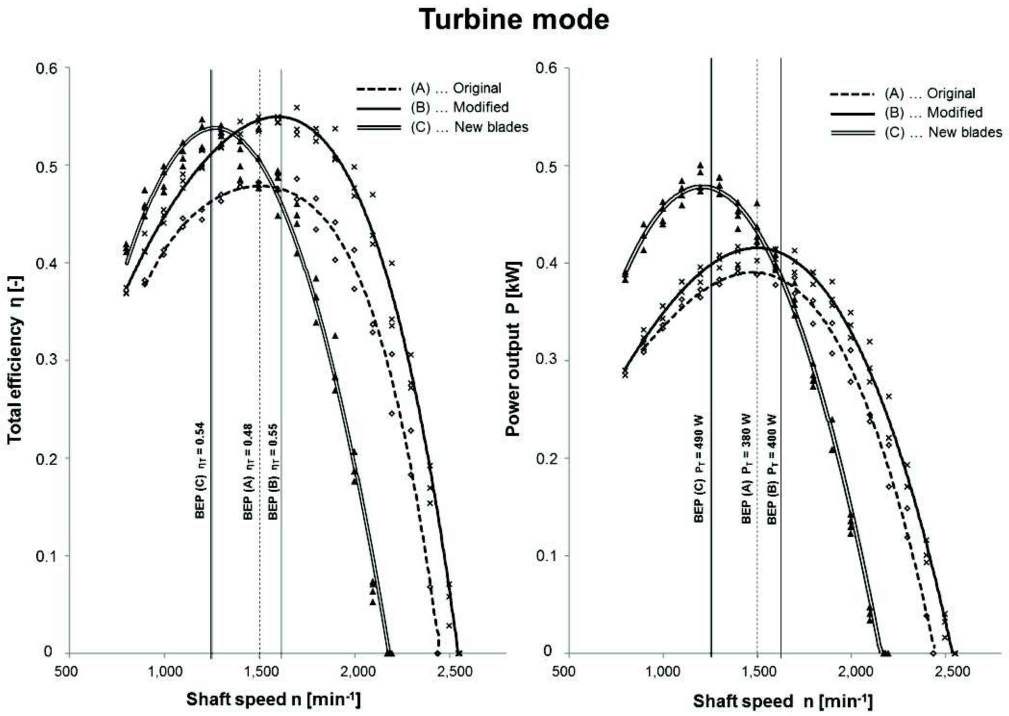


**Figure 5.** Turbine mode – efficiency and power output in dependence to shaft speed.

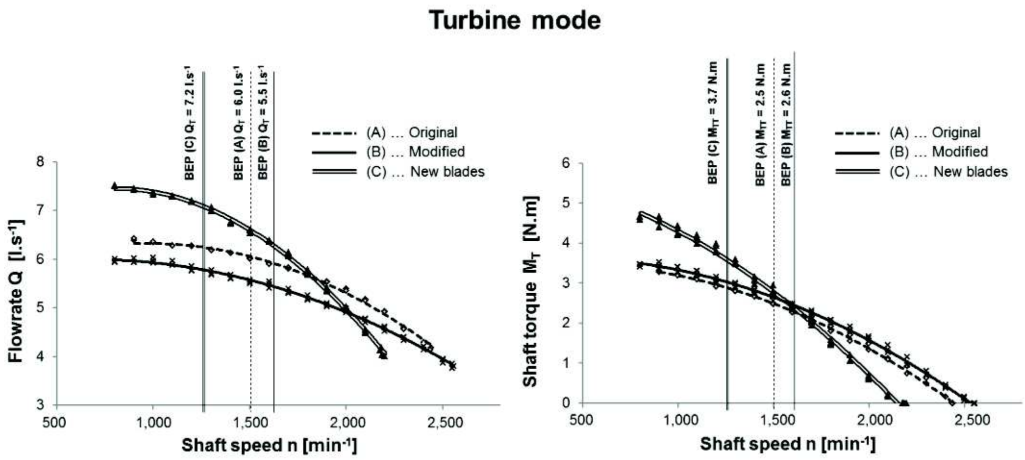


**Figure 6.** Turbine mode – flow rate and shaft torque in dependence to shaft speed.

Properties of above mentioned variants were subsequently verified in pump mode. Measurements were carried out at a constant speed, $n = 1{,}450$ rpm, according to specifications of manufacturer of the original pump. The two mostimportant measured characteristics are shown in Fig. 7, being the dependence of total efficiency and head on flow rate. The charts also indicate corresponding optimum points (BEP).

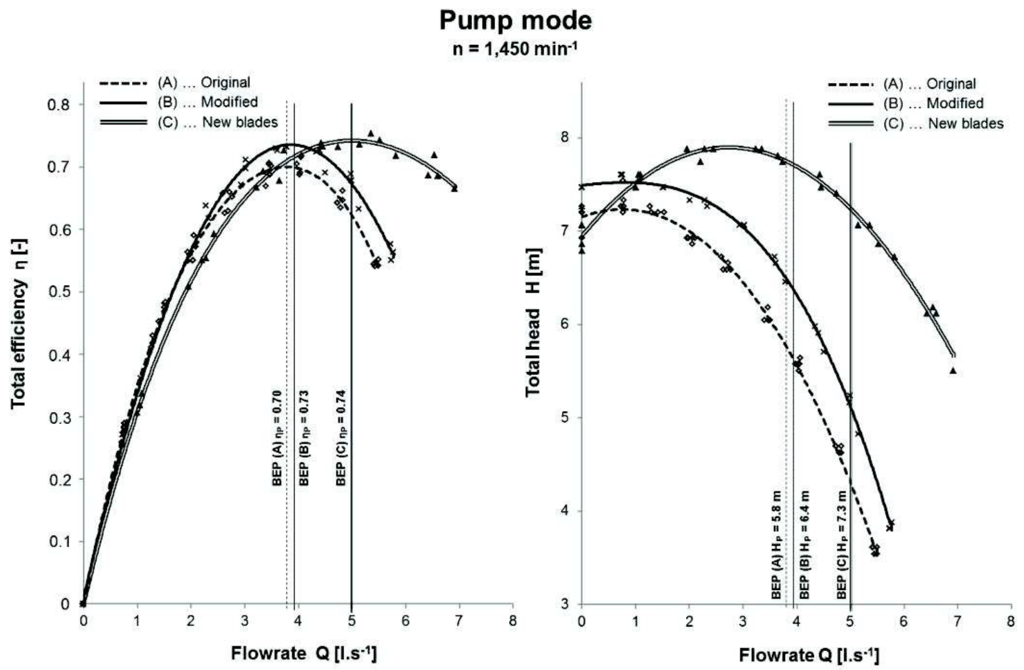


**Figure 7.** Pump mode – efficiency and head in dependence of flowrate.

For quantified evaluation of the proposed innovative modifications, key parameters of the measured characteristics are explicitly summarized in Table 2. Relative increases of monitored values corresponding to optimal operation (BEP) in both turbine and pump modes of operation are shown for the two variants, B and C. Relative increase of the parameters were computed in relation to those of variant A – the unmodified pump.

**Table 2.** Changes in parameters due to innovative modifications (at BEP)

| Variant | Turbine mode | | | | | Pump mode | | | |
|---|---|---|---|---|---|---|---|---|---|
| | $\Delta\eta_T$ [%] | $\Delta P_T$ [%] | $\Delta M_{TT}$ [%] | $\Delta Q_T$ [%] | $\Delta H_T$ [%] | $\Delta\eta_P$ [%] | $\Delta P_P$ [%] | $\Delta Q_P$ [%] | $\Delta H_P$ [%] |
| B | 14.6 | 5.3 | 4.0 | -8.3 | 2.2 | 4.3 | 9.7 | 2.6 | 10.3 |
| C | 12.5 | 28.9 | 48.0 | 20.0 | -6.7 | 5.7 | 61.3 | 35.5 | 25.9 |

In its original, unmodified state, the centrifugal pump used in this study had a head, $H_P = 5.8$ m, flow rate, $Q_P = 3.8$ $ls^{-1}$, power input, $P_P = 310$ W and efficiency, $\eta_P = 70\%$. The values refer to the optimum operation, i.e. the highest achieved efficiency (BEP). The same unmodified machine, in turbine mode indicated a head, $H_T = 13.5$ m, flow rate, $Q_T = 6.0$ $ls^{-1}$, power output, $P_T = 380$ W and total efficiency, $\eta_T = 48\%$, at optimum point. The pump therefore has a significantly higher efficiency than the manufacturer guarantees. However, if the pump is used in the turbine mode, its efficiency as a machine decreases by 22%. The aim of the innovative modifications was to reduce such a significant decline. Via simple, additional modifications to the existing machine – variant B – a proportional increase in efficiency in the turbine mode was achieved (up to $\Delta\eta_T = 14.6\%$). At the same time, torque increased by $\Delta M_{TT} = 4\%$ and power output of the turbine by $\Delta P_T = 5.3\%$.

When the same pump was fitted with an impeller with new blade geometry, efficiency in the turbine mode increased only by $\Delta\eta_T = 12.5\%$, but the torque increased by $\Delta M_{TT} = 48\%$ and power output almost by $\Delta P_T = 29\%$ (!). Moreover, these parameters were achieved at a lower gradient. The cause of the higher power output was an increased rate of flow through the impeller by as much as $\Delta Q_T = 20\%$.

Manufacturers of pumps are, interested in the influence of modifications in the pump mode. For the modified pump (variant B), a slight increase in flow rate of $\Delta Q_P = 2.6\%$ occurred, while head increased by $\Delta H_P = 10.3\%$. Increase in efficiency of $\Delta\eta_P = 4.3\%$ confirmed positive impact of the innovation also in the pump mode.

The impeller with new blades geometry indicated even a higher increase in performance parameters in the pump mode. Flow rate increased significantly by $\Delta Q_P = 35.5\%$, head by $\Delta H_P = 25.9\%$ and efficiency by $\Delta\eta_P = 5.7\%$. However, given the concept of impeller blades, it can be expected that the impeller with such geometry in the pump mode will have deteriorated cavitation characteristics. This has however not yet been experimentally verified.

## CONCLUSIONS

In this study, two variants to a centrifugal pump were selected following several modifications on the pump's geometric parameters and their performances both as turbine and pump were evaluated and compared with those of the original machine. The original machine's total efficiency as a turbine was 22% below its performance as a pump. As a pump, variant B gave slight increases in flow rate, head and overall efficiency (2.6, 10.3 and 4.3%, respectively) over the original pump. When fitted with an impeller with new blade geometry (variant C) the machine indicated up to 48% increase in torque output and 29% increase in power output over the original machine when tested in the turbine mode. The proposed innovative modifications of the radial centrifugal pump are therefore clearly beneficial for the turbine mode. Due to increased power output of the machine, variant C is especially suitable for this purpose. In variant B, benefits were also demonstrated for the pump mode. In summary, it can be said that the application of these innovations brings to their users benefit in terms of increased efficiency in their use for hydroelectric power. Manufacturers of pumps can then benefit from higher market attractiveness of their products.

## REFERENCES


Alatorre-Frenk, C. 1994. Cost minimization in microhydro systems using pumps-as-turbines, Ph.D. Thesis, University of Warwick.

Bláha, J., Melichar, J. & Mosler, P. 2011. Contribution to the use of hydrodynamic pumps as turbines. *Energetika* **5**, 1–5 (in Czech)

Commission regulation (EU) No 547/2012. http://eur-lex.europa.eu/legal-content/EN/TXT/?qid=1486033967314&uri=CELEX:32012R0547. Accessed 10. 1. 2017

ČSN EN ISO 9906: Hydrodynamic pumps – Acceptance tests of hydraulic performance parameters – Degrees accuracy of 1, 2 and 3, ČSNI. (in Czech)

Directive 2009/125/EC. *http://eur-lex.europa.eu/legal-content/EN/TXT/?qid=1486034114 994&uri=CELEX:32009L0125.* Accessed 10. 1. 2017

Derakhshan, S. & Nourbakhsh, A. 2008. Theoretical, numerical and experimental investigation of centrifugal pumps in reverse operation. *Experimental Thermal and Fluid Science* **32**, 1620–1627.

Gülich, J.F. 2003. Effect of Reynolds Number and Surface Roughness on the Efficiency of Centrifugal Pumps. *Journal of Fluid Engineering* **125**(4), 670–679.

Hodák, T., Vodzinský, V., Ondrušek, V. 1982. *Utilization of hydropower (Instructions for exercise).* SVŠT, Bratislava, 1982. (in Slovak)

Hýbl, J. 1928. *Water engines I, II, III*. Czech Technical Institute, Prague. (in Czech)

Melichar, J., Vojtek, J. & Bláha, J. 1998. *Small hydro turbines, construction and operation.* ČVUT, Praha. pp 296. (in Czech)

Munson, B.R., Zouny, D.F. & Okishii, T.H. 2006. *Fundamentals of Fluid Mechanics, Fifth edition*, John Wiley & Sons, Inc., New York.

Nautiyal, H.V., Kumar, A. & Yadav, S. 2011. Experimental investigation of centrifugal pump working as turbine for small hydropower system. *Energy Science and Technology* **1**(1), 79–86.

Nechleba, M. 1962. *Water turbines – their design and accessories*, 2nd Edition. SNTL, Praha. pp 673 (in Czech)

Nechleba, M. & Hušek, J. 1966. *Hydraulic machines*. SNTL, Prague. pp 388 (in Czech)

Nevěřil, J. 2012. Limits of energy demands. *Sigmaprofil* **20**(4), 10. (in Czech)

Pochylý, F., Haluza, M., Rudolf, P., Habán, V., Klas, R., Hudec, M., Himr, D. & Stejskal, J. 2013. Mutual cooperation SIGMA GROUP, Inc. Lutin, SIGMA RDI, Ltd. Lutin and the Department of Fluid Engineering of V. Kaplan FSI in Brno in research pumps. In Proceedings of the Scientific–Technical Conference on ‚Current trends pumping equipment'.Sigma Group a.s., Lutín. (in Czech)

Polák, M. & Polák, V. 2010. Application of the theory of physical similarity in calculation program for the design of hydraulic turbines. *Jemná Mechanika a Optika* **10**, 281–284. (in Czech)

Polák, M. 2013. The issue of energy utilization of low-potential water sources. CULS Prague, Prague 2013. [hab. thesis] (in Czech)

Poláková, Z. &Polák, M. 2016. Centrifugal pump in turbine mode for small hydropower. In Proceedings of the 6th International Conference on Trends in Agricultural Engineering. CULS, Prague. pp 500–505.

Raman, N., Hussein, I., Palanisamy, K. & Foo, B. 2013. An experimental investigation of pump as turbine for micro hydro application. In Proceedings of 4th International Conference on Energy and Environment (ICEE 2013), 5–6 March, Putrajaya, Malaysia. IOP Conference Series: Earth and Environmental Science **16**(1), 012064.

Sedlář, M., Šoukal, J. & Komárek, M. 2009. CFD analysis of middle stage of multistage pump opereting in turbine regime. *Engineering Mechanics* **16**(6), 413–421.

Singh, P. 2005. Optimization of internal hydraulics system design for pumps as turbine with Field Implementation and Evaluation. PhD thesis, Universität Fridericiana, Karlsruhe.

Singh, P. & Nestmann, F. 2011. Internal hydraulic analysis of impeller rounding in centrifugal pumps as turbines. *Experimental Thermal and Fluid Science*, 35, 121–134.

Ulrych, E. 2007. *Applied hydromechanics I – Basics of hydropower*. Powerprint s.r.o., Prague. pp 126. (in Czech)