CENTRIFUGAL PUMP IN TURBINE MODE FOR SMALL HYDROPOWER

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Abstract
In order to minimize investment costs when setting up small hydropower plants, pumps in reverse turbine mode are used in some cases. However, small pump units in turbine mode show decrease of overall efficiency compared to pump efficiency. The article describes innovative design changes, which reduce this negative effect. The proposed changes were verified in a hydraulic testing circuit with the following results: modifications of the existing pump design showed an increase in the efficiency by several percent, significantly increased power output and shaft torque. The best results were achieved with a new turbine runner geometry: efficiency increased by 12,5%, torque by 48% and power output by nearly 29%.

Key words: efficiency, hydraulic profile, modification, pump as turbine, runner blade.

INTRODUCTION

Many localities with low potential of hydropower resources remain unexploited. Frequent reasons include high initial costs with long payback period for investors. In terms of investment in small hydropower plants the most expensive part is water engine (turbine). Thus, there is a need for such technical solutions that would be acceptable to investors. One possibility is the use of hydrodynamic pumps as turbines (PAT). The free flow space between the blades of runner of hydrodynamic engines enables reversibility of energy transformation. In pumps, input mechanical energy changes into hydraulic energy, whereas in turbines incoming hydraulic energy changes into mechanical energy. BLÁHA, ET AL. (2012) given the offer on the pumps market, these relatively inexpensive and reliable devices, that are even in terms of maintenance and service often more advantageous than conventional turbine supplied "on demand", should be taken into consideration. Some practical experiences with PAT are known and have been described in reference literature SIGH ET AL. (2012); ALATORE (1994); DERAKHSHAN ET AL. (2008); NAUTIYAL ET AL. (2011); RAMAN ET AL. (2013); POCHYLÝ ET AL. (2013).

In some cases, when the pump is operated in PAT mode, efficiency is an issue. In case of large machines, the efficiency of both pump and turbine mode is the same or comparable and therefore could be predicted from the known pump efficiency. However, small pumps have lower efficiency in turbine mode. Moreover, the rate of decrease in efficiency can be different for different engines. The question therefore is how to determine efficiency, or rather, how to eliminate its decrease. Some simple improvements of pumps for turbine mode have already been tested and are described in SIGH (2005) and SIGH ET AL. (2012), however more detailed information on this issue in the research literature is still missing.

MATERIALS AND METHODS

For the research of pumps operating in turbine mode, a single-stage centrifugal pump with spiral casing was selected in order to verify its specific design modifications. The pump outline including its characteristics is shown in Fig. 1. This is a specific low-speed pump with specific speed n_{qp} = 24 min^{-1}, runner outer diameter D_1 = 132 mm. The optimum in pump mode for pump speed n_p = 1450 min^{-1} corresponds to: flowrate Q_p = 3.54 l.s^{-1}; water head H_p = 5.85 m; power input P_p = 0.33 kW; overall efficiency η_p = 62.5 % (see Fig. 1).

The essence of individual improvements made on the pump consists mainly in reducing hydraulic losses. These modifications include improvement of the runner geometry and of immediately connected zones, i.e. the zones of input, flow and output of fluid from the runner. For example, for a pump with similar specific speed the flow analysis in both pump and turbine mode is provided in SEDLÁŘ ET AL. (2009).

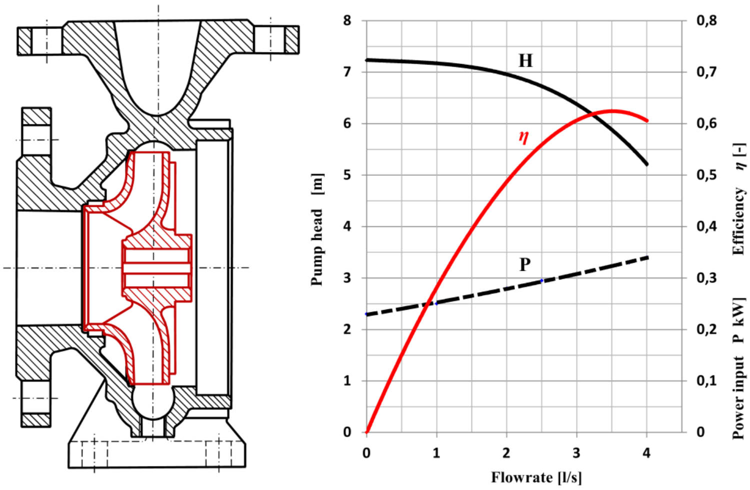

Fig. 1. – Centrifugal pump for testing in turbine mode

The idea for individual construction improvements, mainly the modification of the geometry of runner blades, results from the theory of radial Francis turbine design according to MUNSON ET AL. (2006); NECH LEBA (1962); NECHLEBA ET AL. (1966); ULRYCH (2007); HÝBL (1928); MELICHAR ET AL. (1998); POLÁK ET AL. (2010). The results served as basis for improvements, respectively for manufacturing a prototype, mainly the runners. Additional improvements relating to the reduction of hydraulic losses during reverse flow through the pump resulted from the technical documentation and experience described in SIGH (2005) and POLÁK (2013). All proposed design improvements were subsequently tested experimentally on the hydraulic circuit in the Laboratory of Fluid Mechanics at the Faculty of Engineering (Czech University of Life Sciences Prague). Fig. 2 shows the hydraulic circuit outline for pump testing in turbine mode.

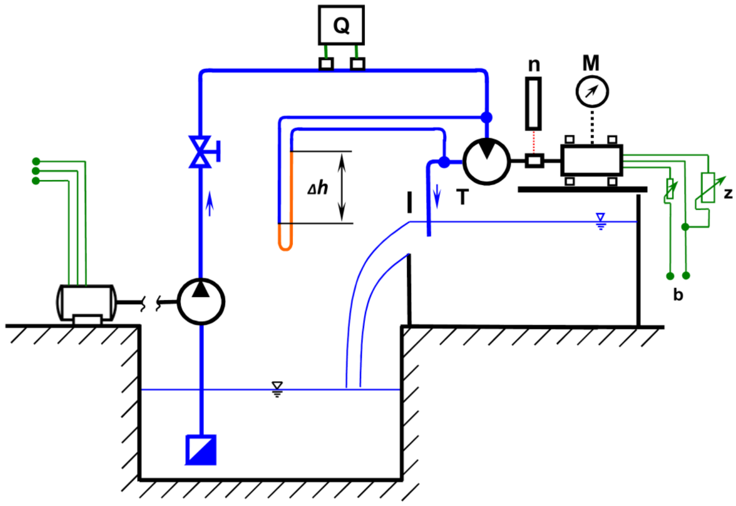

Fig. 2. – Hydraulic circuit for testing of pumps in turbine mode
Q – flowmeter, T – pump in turbine mode, M – dynamometer (shaft torque measuring), n – revolution counter, b – excitation of dynamometer, z – resistance load of dynamometer.

The testing circuit consists of a water reservoir with a pump that generates hydraulic potential for the tested turbine (T), or more precisely pump in turbine mode. The turbine is connected to the dynamometer with a stator placed on a bed allowing a slight rotation, which enables a measurement of the reaction torque (M) at load. Dynamometer is a DC engine connected to continuous resistance load control unit (z). The flow of water through the turbine is measured by the ultrasonic flowmeter Siemens SITRANS

FUP1010 (Q), which was calibrated by using the volumetric tank method, before measurement. The specific energy of the turbine is determined on the basis of the flowrate Q_T and the differential pressure of the U-tube mercury manometer (Δh), connected to the pipe by collecting probes. The scheme for pressure collection for gauging in the 1st precision class is in accordance with the ČSN EN ISO 9906 (2013) and HODÁK (1982). The revolutions of the turbine shaft were measured by a contactless infrared sensor.

Fig. 3. – Connection of the turbine in testing circuit

The following parameters were monitored during the turbine tests: differential pressure determined by level difference Δh [m], flowrate Q_T [$m^3.s^{-1}$], turbine shaft speed n_T [min^{-1}] and the shaft torque on the dynamometer M_T [N.m]. From these parameters, additional values necessary for the graphical characteristics of the turbine were then calculated.

Based on the measured values, the specific energy gained by the turbine Y_T [$J.kg^{-1}$] is given by the equation:

$$Y_T = \frac{\rho_{Hg}}{\rho_w} \cdot \Delta h \cdot g + \frac{v_i^2 - v_o^2}{g} + g \cdot y \qquad (1)$$

Where ρ_{Hg} [$kg.m^{-3}$] is mercury density, ρ_w [$kg.m^{-3}$] is water density, v_i, v_o [$m.s^{-1}$] is water velocity in inlet and outlet pipe, respectively and y [m] is vertical distance of zones generating pressure. The power of the fluid gained by the turbine (the power input) P_w [W]is given by:

$$P_W = \rho_W \cdot Q_T \cdot Y_T \qquad (2)$$

The power output of the turbine is determined from the shaft revolutions n_T [min^{-1}] and the shaft torque M_T:

$$P_T = M_T \frac{\pi \cdot n_T}{30} \qquad (3)$$

The overall efficiency of the turbine is:

$$\eta_T = \frac{P_T}{P_W} \qquad (4)$$

RESULTS

The characteristics of the original non-modified pump were first tested in turbine mode in the hydraulic circuit. On the same testing circuit and under the same conditions, different types of modifications of the pump were then tested, their mutual combinations and also new geometries of runner blades. From all tested options the following four modifications were selected for evaluation:

- **I.** Non-modified original pump – illustrated by blue double-line curves in the characteristics on Fig. 4 and Fig. 5.
- **II.** Original pump with modified runner. The modification consisted of changes reducing local and friction losses – yellow curves.
- **III.** Original pump with modified runner (see modification II.), and modified spiral casing. The purpose of the modification was to reduce hydraulic losses – brown curves.
- **IV.** Original pump with original runner, but with modified blades length and modified spiral casing (modification of the spiral casing is the same as in modification III.) – black dashed curves.
- **V.** Pump with new geometry of runner blades and modified spiral casing (modification of the spiral casing is identical to III. and IV.) – green double-line dashed curves.

Measurements were carried out in the way that, by constant opening of the throttle valve, the turbine was gradually loaded from idle speed up to 900 or 800 rpm. In this way, uniform conditions were set up for all tested options so that the results are mutually comparable.

The diagrams in Fig. 3 and 4 provide a summary of the measured characteristics for the above described modifications I – V, i.e. efficiency, power output, shaft torque and flowrate, depending on unit speed:

$$n_{11} = \frac{n_T \cdot D_1}{\sqrt{Y_T}} \qquad (5)$$

Given the possibilities of the testing circuit, where the hydraulic potential for the turbine is created by a centrifugal pump, measurements did not take place under constant water gradient. Therefore the course of the net water gradient depending on unit speed revolutions is in addition also specified here.

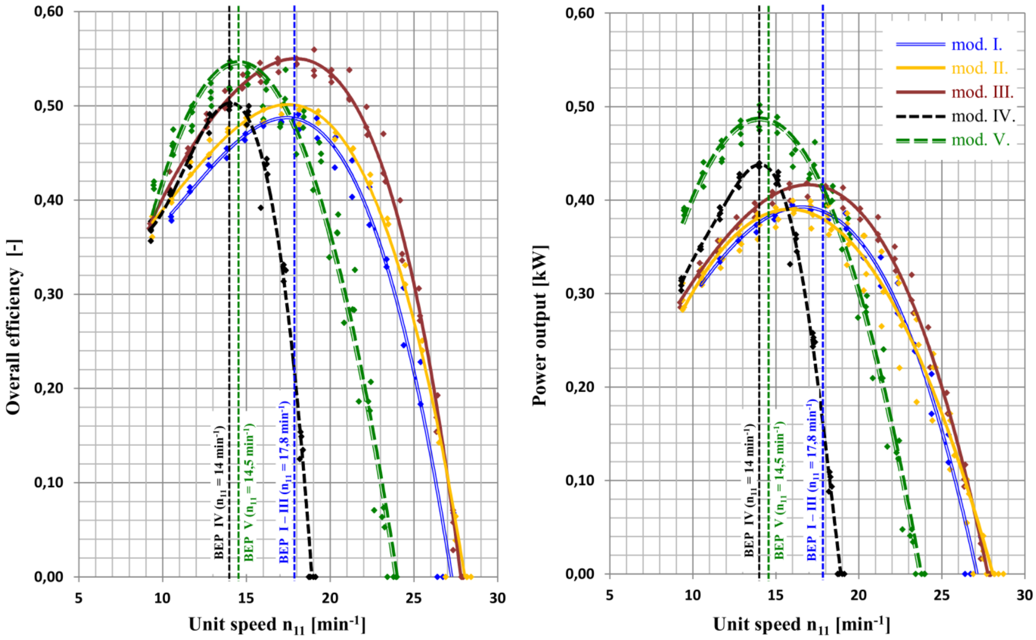

Fig. 4. – Efficiency and power output depending on unit speed

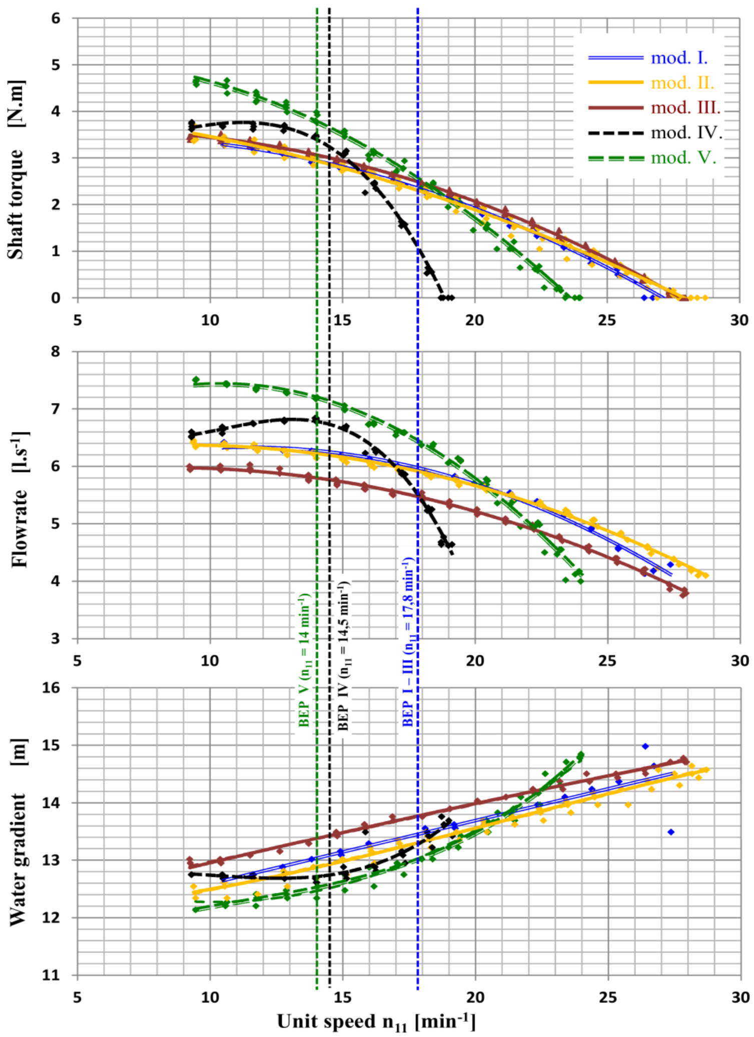

Fig. 5. – Water gradient, flowrate and shaft torque depending on unit speed

On the basis of the above mentioned characteristics, the following table (Tab. 1) explicitly summarizes the results obtained for each modification. These are the parameters of the turbine reached at the highest efficiency – in the characteristics indicated as BEP (Best Efficiency Point).

Tab. 1. – Absolute values of the parameters in turbine mode (BEP)

Modification	η_T [%]	P_T [W]	M_T [N.m]	Q_T [l.s^{-1}]	H_T [m]
I.	48	380	2.5	6.0	13.5
II.	50	390	2.4	6.0	13.4
III.	55	400	2.6	5.5	13.8
IV.	50	440	3.5	6.8	12.7
V.	54	490	3.7	7.2	12.6

The following table (Tab. 2) shows clearly the results achieved by each modification and quantifies the increase of the observed parameters. The proportional increase of each parameter is always related to the modification I, i.e. non-modified original pump. The increase is calculated as follows:

$$\Delta x_n = \frac{x_n - x_I}{x_I} \cdot 100 \qquad (6)$$

Where x_n is absolute values (η, P, M_T, Q a H) for options II. up to V and x_I is absolute values (η, P, M_T, Q a H) for modification I.

Tab. 2 - Proportional increase of parameters in turbine mode (BEP).

Modification	$\Delta\eta_T$ [%]	ΔP_T [%]	ΔM_T [%]	ΔQ_T [%]	ΔH_T [%]
II.	4.2	2.6	-4.0	0.0	-0.7
III.	14.6	5.3	4.0	-8.3	2.2
IV.	4.2	15.8	40.0	13.3	-5.9
V.	12.5	28.9	48.0	20.0	-6.7

DISCUSSION AND CONCLUSIONS

The pump used in the tests has in pump mode a maximum efficiency of $\eta_{č}$ = 62.5% (manufacturer's data). However, the same engine operated in reverse turbine mode shows the highest efficiency only η_T = 48%. That means a relative fall of 23.2% related to the pump efficiency. If the pump is modified in order to reduce hydraulic losses during reverse liquid flow, it is possible to achieve a proportional increase in efficiency by $\Delta\eta_T$ = 14.6% (see modification III), the shaft torque increasing by 4% and the turbine output by 5.3%.

For comparison – in a research on this issue in Germany at the University of Karlsruhe [8], comparable modifications with similar type of engine (n_q = 36.4 min^{-1}, D_1 = 174 mm) led to an increase of efficiency by a maximum of $\Delta\eta$ = 4.7% and power output by approx. ΔP = 3.4%. For the remaining seven pumps of similar size and specific speed (n_q = 24.5 ÷ 79.1 min^{-1}) mentioned in [8], the achieved efficiency increase was only about $\Delta\eta$ = 1.3%. It therefore highly depends on the engine individual constructional design, and mainly, on the precision of the performed modifications.

The modifications of the runner blades also give various results. Even though the modification of their length shows efficiency increase by only 4.2% (modification IV), the shaft torque increases by 40% and the power output of the engine by 15.8%. In this respect, the best results are achieved by a runner with completely new blades geometry (modification V). In this case efficiency increases by 12.5%, shaft torque by 48% and the performance by nearly 29%. Moreover, the engine shows an additional plus – it reaches these parameters at lower water gradient in comparison with the other modifications. The reason for this is a greater throughput of the runner which is manifested by an increased flowrate ΔQ_T = 13.3% or 20%.

When deciding about possible modifications of the pump for turbine mode, it is therefore necessary to take into consideration the purpose of the modifications – whether it is primarily to increase efficiency, or if the aim is to maximize energy production. An optimal solution is a combination of both, i.e. partial improvements to reduce hydraulic losses along with a new runner blades design. This option contains the greatest potential – both for the user and in terms of further research.

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