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Numerical modelling of flow phenomena in a pump with a multi-piped impeller

Janusz Skrzypacz *

Wroclaw University of Technology, Faculty of Mechanical and Power Engineering, Wybrzeże Wyspiańskiego 27, 50-370 Wrocław, Poland

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ABSTRACT

The multi-piped impeller (MPI) is a completely new, patented approach to the design of the impellers of pumps operating with a very low specific speed $n_q < 10$. Such a construction is a development of the concept of a pumping disc with drilled holes (a drilled impeller). In order to identify the flow phenomena in such an impeller, numerical calculations were done taking into account the influence of the grid size and turbulence model on the accuracy and time of the calculations. The numerical model was validated by comparison with experimental results. The calculation results were presented and discussed.

1. Introduction

When it comes to the specific speed $n_q < 10$ and the relatively low capacity ($Q < 10\,m^3/h$), with regard to low efficiency the construction of rotodynamic pump impellers requires a non-standard design approach. Nowadays, it is extremely difficult to find out any information on the design, construction or operation of rotodynamic pumps in relation to the specific speed $n_q < 10$. Only certain limited information [1,2] and a small number of research results – e.g., [3–6] – are available. Patent databases, where one can find interesting solutions to the construction of pump impellers, operating at, e.g., $n_q < 10$ [7–10], turn out to be a useful source. One such solution covers a multi-piped impeller patented by the author of Ref. [11].

Multi-piped impellers (the name is suggested by the author) [11] develop the concept of pumping discs with drilled holes [1,2] used in rotodynamic pumps with the specific speed $n_q < 10$. Such pumps are widely applied in the chemicals industry, machine lubrication systems, liquid gas technology and firefighting, among others.

From the point of view of the liquid flow through the impeller flow channels, both constructions – a multi-piped impeller and a drilled impeller – are identical (Fig. 1). The essential difference in operation results from the different phenomena at the external flow around both of the impellers. In the drilled impeller, the external flow around the body is accompanied with the friction of the rotating disks of the impeller and the liquid, which results in power losses proportional to the fifth power of the outer diameter of the impeller and the third power of the angular speed (d^5 and ω^3) [1].

In the case of multi-piped impellers, the external flow around the pipes generates a drag force, the value of which can be theoretically expressed with the following relationship [12]:

$$P_x = C_x \frac{\rho u^2}{2} A \qquad (1)$$

The value of the real drag force will be different due to the mutual interaction of individual pipes, influencing the speed profile of the liquid flow around the impeller flow channels. However, it seems that some of the power necessary to overcome the drag force at a specific rotational speed will cause an increase in the liquid circulation inside the pump and, hence, an increase in the total head. Initial experimental tests revealed that a multi-piped impeller generates a 30% higher head as compared to a drilled impeller with identically sized flow channels at a comparable efficiency level. When designing a multi-piped impeller for working parameters corresponding to a drilled impeller (Q, H), the surplus of the head allows for a certain optimization of the multi-piped impeller flow geometry.

Another advantage of multi-piped impellers is that they are easy to make. The flow channels in the form of pipes can be freely shaped and then connected with the impeller hub with any technology (e.g., thread – Fig. 1, welding – Fig. 11, etc.).

In order to design multi-piped impellers in a sensible and conscious way, one should become familiar with the flow phenomena occurring with the liquid flow in and around such a construction.

* Tel.: +48 071 320 48 25.
E-mail address: janusz.skrzypacz@pwr.wroc.pl

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Nomenclature

A	area of cross-section (m^2)
H	pump total head (m)
M_i	momentum on the internal surfaces of the impeller flow channels (nm)
M_o	momentum on the external surfaces of the impeller flow channels (nm)
M_t	total momentum on the surface of the rotating impeller flow channels (nm)
n	rotational speed (rpm)
n_q	kinematic specific speed ($n_q = nQ^{0.5}/H^{0.75}$) (rpm)
p_{co}	total pressure in the pump outlet section (Pa)
p_{ci}	total pressure in the impeller inlet section (Pa)
P_h	hydraulic power (W)
P_w	power on a pump shaft (W)
P_x	drag force (N)
Q	flow rate (m^3/s)
$\bar{u}$	vector of componential velocities (u, v, w) (m/s)
u, v, w	velocity in directions X, Y, Z (m/s)
u', v', w'	velocity fluctuations (m/s)
U, V, W	average velocities in directions X, Y, Z (m/s)
η_h	hydraulic efficiency
η_v	volumetric efficiency
η_m	mechanical efficiency
η	total efficiency
ρ	density (kg/m^3)

This study aims at modelling a flow in a pump with a multi-piped impeller using CFD and verifying a numerical model against experimental data. The verified numerical model will be used for identifying the flow phenomena in such a construction and for further research aimed at determining the impact of geometrical features of such an impeller on the operating parameters and efficiency of the process of transferring liquid energy.

2. Test rig

With a view to verifying the results of the numerical calculations, a specialized test rig – presented in Fig. 2 – was designed and constructed. The main element of the test rig includes a pump (Fig. 3) with a special design allowing for the quick replacement of the tested impellers while maintaining the repeatability of the measurement results.

The pump is supplied from a closed tank where – depending upon the needs – one can generate overpressure or negative pressure and control the medium level and temperature.

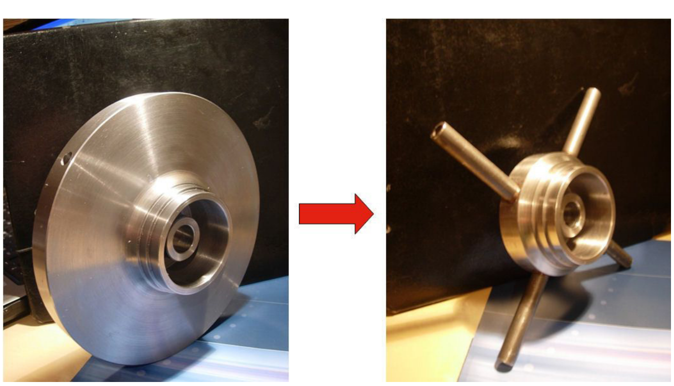

Fig. 1. A drilled impeller pump and a multi-piped impeller pump.

Fig. 2. View of the test rig.

The pump capacity can be adjusted with a ball control valve, namely MARS 88V with an IntrOM OM-1 electric drive.

The measuring instruments whose parameters are presented in Table 1 were used for measuring the specific values.

The characteristics of the tested impeller were measured in a fully automated way, according to the recommendations given in [13]. The measurement process is controlled by a computer and dedicated software, whose interface is presented in Fig. 4. The user can set the following parameters:

- Operating range of the control valve: minimum (Fig. 4, position 1) and maximum opening (Fig. 4, position 2), expressed in %.
- Percent stroke of the control valve (Fig. 4, position 3).
- Flow calming time (Fig. 4, position 4).
- Time between subsequent samplings (Fig. 4, position 5).
- Number of samples in one measurement point (Fig. 4, position 6).
- Rotational speed, basic (Fig. 4, position 7).
- Maximum engine power – when it is exceeded, the system will switch off automatically (Fig. 4, position 8).

The measurement was conducted from a complete opening of the control valve to its closing, and then from closing to the fully open position. This is why there is a double line on the diagrams presenting the characteristics (Fig. 6). The measurement results and the estimation of the measurement uncertainty are carried out according to the formulae given in [13].

The construction presented in Fig. 5 is the basic impeller. The impeller works with a constant cross-section stator, whose outlet ends with an outlet diffuser. The results of measuring the performance curves of the base construction are presented in Fig. 6.

Table 1
Measuring instruments.

No	Measuring instrument	Range	Accuracy class
1	Electromagnetic flowmeter Arkon MAGS1-ST DN25 PN 40	0.18–17.67 m^3/h (0.1–10 m/s)	0.2%
2	Pressure gauge (suction) FUJI FKP 01	−0.7 to 0.5 bar	0.1%
3	Pressure gauge (discharge) FUJI FKP 03	0–30 bar	0.1%
4	Active power transducer METROL PP73	0–3000 W	0.3%
5	Temperature transducer FLEXTOP	0–50 °C	±0.9 °C