PERFORMANCE ANALYSIS OF COMPACT MULTISTAGE PUMPS MANUFACTURED FROM SHEET METAL

F. Fontana

Department of Industrial Engineering, University of Padua, Padua, Italy,
Federico.fontana.3@phd.unipd.it

ABSTRACT

Cold forming and welding of stainless steel sheets is becoming a leading technology in mass production of multistage, radial flow pumps. The stages of these machines are usually manufactured by resistance spot welding of cylindrical (i.e. 2D) blades on flat surfaces. Such stages commonly feature the so-called "compact design", i.e. the maximum stage diameter is close to the impeller one. Accordingly, an annular chamber connects the impeller and the return channels that drive the flow to the stage outlet. Although competitive in terms of production cost, this technology introduces strong constraints to the fluid dynamic design. Some design methods specifically conceived for compact stages made by moulding or casting technologies are presented in the literature. However, these methods result in blade geometries that are not suited to the present manufacturing technology, due to their complex 3D shape. In the light of this, a dedicated approach is needed for the hydraulic design of compact stages for sheet metal pumps. The present work reports and discusses experimental data on performance and efficiency of several industrial machines, featuring compact stages made from sheet metal. The pumps considered in the study feature a high technological level, as assessed by comparing the measured efficiency with reference values provided by Reg. 547/2012 EC. Nevertheless, the performance measured on these machines does not meet the data suggested in the literature for conventional pumps. Consequently, a theoretical analysis of the compact stage performance is presented in order to explain the difference between traditional and compact machines. In particular, the influence of the dynamic head contribution on the total head rise and on the efficiency of the compact design stages is discussed. The theoretical and experimental results suggest that the design methods for traditional machines are not suitable for compact stages of high efficiency, sheet metal pumps.

KEYWORDS

MULTISTAGE CENTRIFUGAL PUMP, SHEET-METAL PUMP, COMPACT DESIGN PUMP, PUMP PERFORMANCE ANALYSIS

NOMENCLATURE

2	(subscript) referred to the impeller outlet
c	velocity of the fluid
D	diameter
Q	flow rate
g	gravitational acceleration
H	head
imp	(subscript) referred to the impeller
k	disk friction coefficient
MEI	Minimum required Efficiency Index
MS2900	vertical multistage pump, with a nominal speed of 2900 rpm
MSS2900	submersible multistage pump, with a nominal speed of 2900 rpm
n_q	specific speed, (n[1/min]) (Q[m3/s]) $^{0.5}$ (H [m]) $^{-0.75}$

P_{df}	disk friction power
st	(subscript) referred to the static component of the head
u	blade tip velocity
u	(subscript) referred to the tangential direction
ε	degree of reaction
Φ	flow number, $Q/(2\ u_2\ D_2^2)$
η	pump efficiency
η_{hyd}	impeller hydraulic efficiency
η_{mec}	mechanical efficiency
η_v	volumetric efficiency
Ψ	impeller loading factor, c_{u2}/u_2
ζ	head loss

INTRODUCTION

Multistage centrifugal pumps (MCPs) made by cold forming and welding of stainless steel sheets gained significant sales figures around the world, in particular in developed countries and mature markets. This is due to the fact that: (i) MCPs are often the best choice when a high head is required with the constraint of a limited radial dimension (e.g. borehole submersible pumps, high-pressure vertical multistage pumps, etc.); (ii) resistance to corrosion and compatibility with drinking fluids are highly desirable features for any pump; (iii) the use of sheet metal as a raw material allows to optimise the use of stainless steel, which has a relatively high cost in comparison with other common alloys.

The design of non-customized MCPs has often the external diameter of the machine (thus, of the stage), the BEP (best efficiency point) flow rate and the rotational velocity as the inputs, whereas the BEP head results from a compromise between geometrical limitations, efficiency and cost. The latter is affected by both the industrial cost of a single stage and the number of stages required to meet a given performance. In turn, the limitation on the outer diameter strongly encourages an extensive use of the so-called compact design (CD) for values of the specific speed of the stage up to 40. Accordingly, each stage features a centrifugal impeller followed by a centripetal flow passages (i.e. the return channels) that guide the flow to the next stage. The rotating and the fixed vanes are connected by an annular chamber, bounded by the stage casing walls. A schematic view of one CD stage is presented in Fig. 1, where one impeller channel, one return channel and the annular chamber are highlighted. The arrows in the figure show schematically the direction of the main flow.

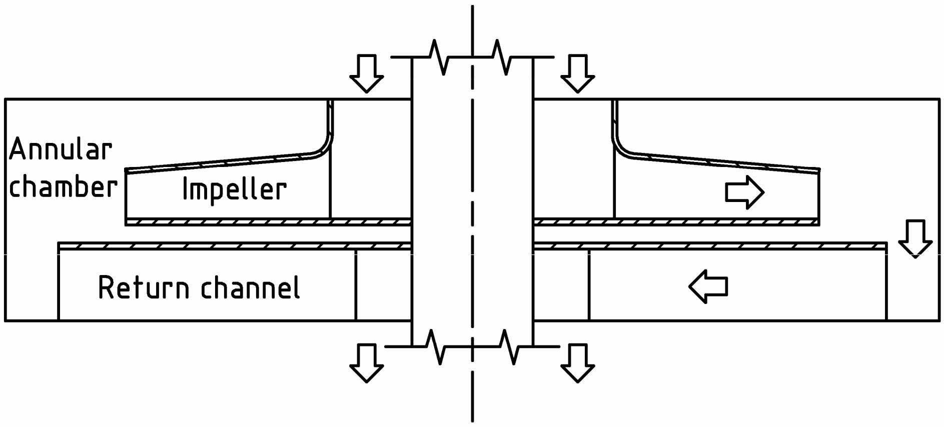

Fig. 1: Schematic view of a compact design stage, with the main flow path indicated by arrows.

The head loss in the annular chamber was studied in a previous work (Fontana, 2017), for several stage designs. It was found that it depends on: (i) the dynamic head at the impeller outlet;

(ii) the geometrical expansion ratio, i.e. the ratio between the impeller blade height and the annular chamber height; (iii) the ratio of the axial distance between the impeller outlet and the return channels inlet, to the impeller outer diameter. A correlation to evaluate the head loss of different stage geometries was determined for the nominal range of operation.

This head loss impacts the efficiency of CD stages, while it does not affect the stages featuring traditional shapes. Despite of their different features, CD and traditional pumps are considered altogether in the European regulation Reg. 547/2012 EC, and the same efficiency thresholds are prescribed for both types. For the development of this regulation, data on the efficiency of machines by several manufacturers were clustered into 8 classes for each external configuration of the pump. For each class, the minimum required efficiency at BEP is provided as a function of the specific speed and of the nominal flow rate (see Falkner, 2008). The efficiency curves corresponding to the highest class (MEI 0.8) are used as reference in this work and they are plotted in Fig. 2 by solid lines, for: left side, vertical multistage pumps (called MS 2900 in Falkner, 2008); and right side, submersible borehole pumps (called MSS 2900 in Falkner, 2008), respectively. The same figure shows also the current legal threshold as per Reg. 547/2012 EC, i.e. the MEI 0.4 class (broken lines with cross markers in the figures). The data provided by Gülich, 2008 for multistage pumps are reported as well, by broken lines and filled markers.

According to Fig. 2, the trends of the curves provided by Gülich, 2008 and Falkner, 2008 agree quite well for nq up to 50. For MS 2900, the data in Gülich, 2008 are well above MEI 0.8 threshold for the large sized pumps, whereas they drop slightly below MEI 0.4 threshold for the small sized pumps. Therefore, none of the pumps featuring a BEP flow rate lesser or equal to 18 m^3/h fulfils the legal requirements according to Gülich, 2008.

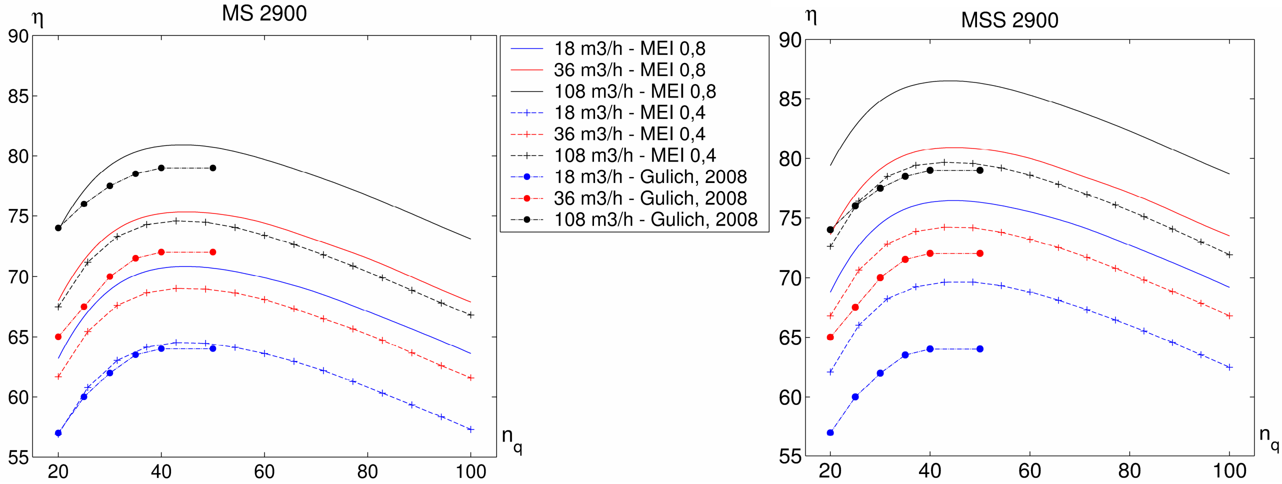

Fig. 2: Curves of efficiency versus specific speed, at different BEP flow rates, according to MEI 0.4 and MEI 0.8 thresholds (see Falkner, 2008) and according to typical values suggested in the literature (Gülich, 2008). Left side: multistage vertical pumps (MS 2900); right side: submersible multistage pumps (MSS 2900).

For MSS 2900 the gap between the two references is even more evident. In particular, according to the data from Gülich, 2008, only large sized, low specific speed pumps fulfill the regulation.

The previous comments suggest that:

1) Vertical multistage pumps meet the current legal requirements regularly only for large sized machines. In turn, small sized multistage pumps need some efforts in terms of industrial research to meet the legal requirements.
2) Submersible borehole pumps need a high investment in product development to reach the legal threshold, in particular when dealing with small sized machines.

In the light of this, special efforts to develop design approaches dedicated to CD stages of MCPs seem necessary. The literature suggests some design approaches for CD stages made by moulding or casting. However, this guidelines are not applicable to sheet metal pumps, due to technological and economical feasibility. Therefore, the present study investigates the performance and the efficiency of MCPs made by sheet metal forming, with a special interest on CD machines. In details, firstly the paper reports an overview of the literature on the performance of diffuser pumps, that are used as a reference to evaluate the performance of MCPs. Then, it presents and discusses experimental data on efficiency and performance of multistage pumps made from sheet metal and featuring either a CD or a diffuser design. Finally, it suggests a theoretical model that explains the experimental results obtained from CD machines.

Literature overview on performance of diffuser pumps

To assess the attainable performance for the stages of multistage machines and to support the analysis of CD stages, an overview of the literature on diffuser pumps was performed. The analysis included single- and multi-stage pumps featuring a diffuser design, either followed by a volute or not. Table 1 lists the data sources used in this study and associates them with a code. When possible, the dimensionless head coefficient, $gH/(u_2)^2$, at either the diffuser outlet section or the stage outlet section was considered (M1, M2, M3, M4, M10, M11); otherwise the head coefficient of the entire pump was plotted (M5, M6, M12, M13, M14, M15, M16). Industrial machines were preferred, although some pumps specifically conceived for research purposes were included (M7, M8). As a term of comparison, the optimal BEP head coefficient for volute and diffuser pumps according to Gülich, 2008 was considered. The lines in Fig. 3 bound the ± 10% scatter band around the optimsl value suggested by the latter.

Table 1: references for diffuser pump performance

Code	Reference	Code	Reference
M1	Eisele et al., 1997	M9	Sloteman, 1989
M2	Masafumi et al., 2008	M10	Wang and Tsukamoto, 2003
M3	Aysheshim, 2001	M11	Yoshida et al., 1990
M4	Stel et al., 2013	M12	Miyabe et al., 2009
M5	Sano et al., 2002	M13	Pei et al., 2014
M6	Barrand et al., 1985	M14	Dyagilev et al., 1977
M7	Arndt et al., 1990	M15	Tsukamoto et al., 1995
M8	Ubaldi et al., 1998	M16	El Hajem et al., 1998
G +10%	Gülich, 2008 (+10%)	G -10%	Gülich, 2008 (-10%)

The values of the total head coefficient from the 16 references are reported on the left side of Fig. 3 (markers). The graph shows that the most part of the machines analysed falls within the band suggested by Gülich, 2008 (the only markers that are well outside of the suggested range correspond to machines designed for research purposes). Consequently, the latter is confirmed to be a best practice for diffuser pumps and, in particular, for the multistage ones.

To investigate further the stage performance, the right side of Fig. 3 reports the static head at the impeller outlet for the same references (where available). The graph shows that the dimensionless static head coefficient ranges between 0.34 and 0.39 when the specific speed ranges between 20 and 40. The difference between corresponding markers in the two graphs is proportional to the dynamic head at the impeller outlet.

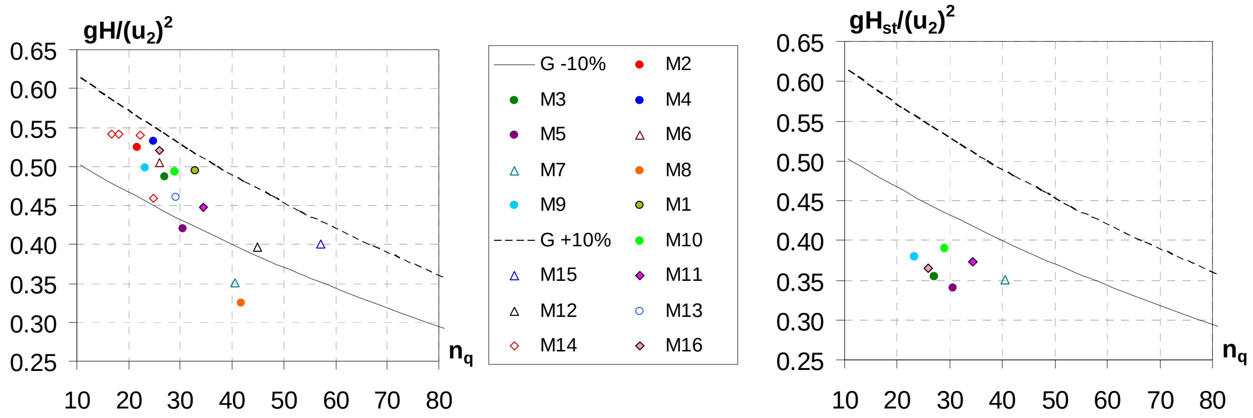

Fig. 3: left side, dimensionless head coefficient versus specific speed; right side, dimensionless static head coefficient at the impeller outlet. Data from references in Table 1. Lines represents the ± 10% band around the optimal curve suggested by Gülich, 2008.

EXPERIMENTS

Experimental methodology

9 submersible borehole pumps and 20 vertical multistage pumps produced by several manufacturers were investigated. 15 machines feature CD stages. The BEP flow rates ranged between 1 and 100 m^3/h at 2900 rpm, with numbers of stages between 3 and 18.

Performance and efficiency were measured in ISO 9906 compliant, closed-loop, test rigs installed in an industrial test facility. The test rigs comprise a large capacity vessel, suction and delivery piping, valves to control the flow rate on the delivery side and electro-magnetic flow meters. The pumps were driven by electric motors at a constant speed. The torque and rotational velocity at the pump shaft were measured. A suction and a delivery pressure transducers were used for the vertical multistage pumps. On the contrary, only the latter was used for the submersible pumps, because the total pressure at the inlet of these pumps was the atmospheric pressure. The measuring uncertainty was 0.3% for performance and 0.5% for efficiency.

For each duty point, 500 samples of volumetric flow rate, pump head, inlet temperature, torque and angular velocity were collected in 10 seconds and then time averaged. A total of 20 to 25 duty points spanning the operational range of each machine was measured.

Experimental results

Analysis of Efficiency

Fig. 4 shows the ratio between the measured values of efficiency and the reference values provided by Falkner, 2008 for MS 2900 pump type (i.e. vertical multistage pumps), as a function of the specific speed. The data are represented as bubbles, featuring an area proportional to the BEP flow rate. The graph shows that the pumps in the sample can be considered as representative of the best available technology, since most of the them belongs to the MEI 0.8 class Falkner, 2008, while the remaining part is close to the MEI 0.8 threshold. It is noted that when both the specific speed and the flow rate are small, a larger fraction of machines reaches high values of relative efficiency. This suggests that in this region the reference curves according to Falkner, 2008 under-estimate the actual efficiency of sheet metal pumps.

Moreover, in the Introduction it was highlighted that the values suggested in Gülich, 2008 are, on average, below the threshold fixed by Falkner, 2008. Thus, as first general conclusion it can be stated that either the scientific literature is even more pessimistic with reference to multistage pumps or stainless steel, sheet metal pumps feature a higher efficiency compared to other types of multistage pumps.

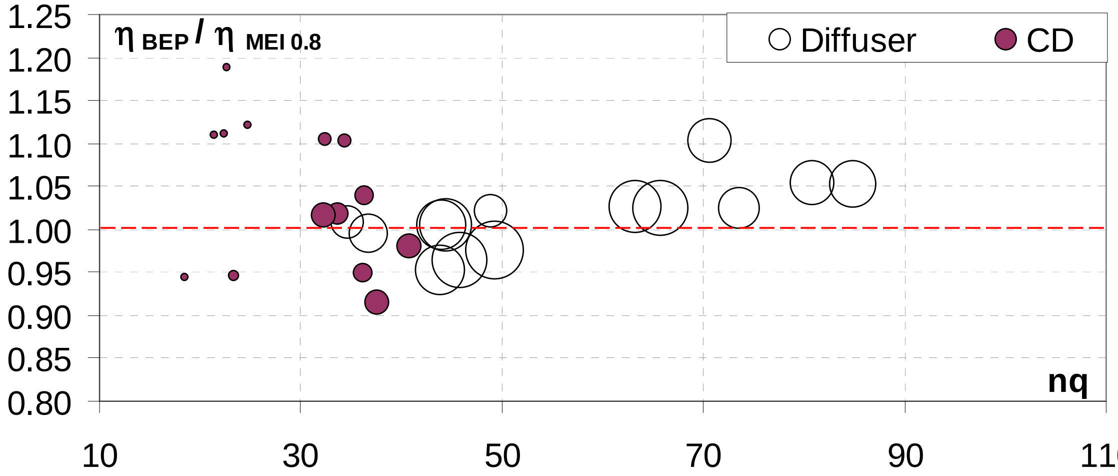

Fig. 4: ratio between the BEP measured efficiency and the corresponding MEI 0.8 value for MS 2900 according to Falkner, 2008. The bubble area is proportional to the BEP flow rate. Filled markers: compact design pumps; empty markers: diffuser pumps; red broken line: MEI 0.8 threshold.

<u>*Analysis of Performance*</u>

Fig. 5 compares the measured values of the BEP dimensionless head coefficient with the reference curves according to Gülich, 2008. MCPs featuring a diffuser design (empty triangles in figure) follow pretty strictly the literature. In contrast, the CD pumps (circles in figure) form a cloud of points centred on a value of head coefficient equal to 0.37, with the most part of the data laying between 0.33 and 0.42. No influence of the specific speed is evident among CD pumps. Accordingly, a significant difference from the curves in the literature (20% or more) is found for the smaller values of the specific speed (see also Fig. 3, left side), while the gap closes for n_q around 40. Thus, the issue regards mainly the so-called “slow” machines, which are supposed to operate with high values of the impeller loading factor, Ψ (Gülich, 2008). This experimental evidence cannot be ascribed to the poor quality of the machines considered because, as already discussed, the sample of pumps in this study represents the best available technology in the field. Therefore, this is supposed to be a specific feature of the CD stages, due to the annular chamber, which is the main difference from other multistage pumps.

It is worth noting that the CD pumps in Fig. 5 gather in a region that overlaps to that of the static head coefficients at the impeller outlet according to the right side of Fig. 3. This suggests that the useful head of CD stages consists only in the static contribution, because of the limited recovery of the dynamic head.

THEORETICAL ANALYSIS

To support the analysis of the experimental results, a theoretical model of the hydraulic behaviour of the compact design stages was developed. The theoretical model aims at highlighting the influence of the impeller loading factor and of the dynamic head contribution on the pump efficiency. The model considers a stage having a fixed geometry apart for the blade angle at the impeller outlet, so that the impeller can provide different values of specific work by changing this feature. This is a simplification to the problem of the constrained design of a family of multistage pumps (see the Introduction).

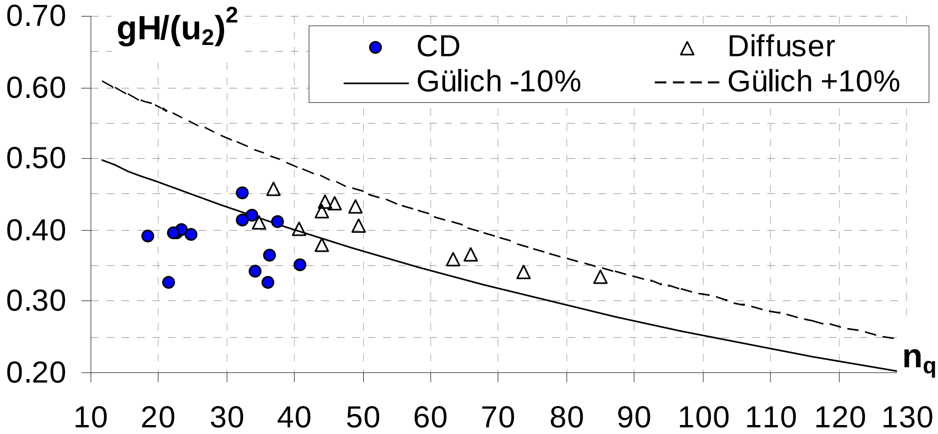

Fig. 5: experimental values of the head coefficient at BEP (markers). Lines represents the ± 10% band around the values suggested in Gülich, 2008. Filled markers: compact design pumps; empty markers: diffuser pumps.

Under the usual assumption of swirl-less inlet flow, the total power provided by the impeller to the fluid equals: $\rho Q_{imp}(c_{u2}u_2)+P_{df}$, where P_{df} is the disk friction power and the other term is the power provided to the flow by the impeller channels. According to the dimensional analysis (see the literature review in Gülich, 2003), the disk friction power can be modelled as $P_{df}=\rho\cdot k\cdot u_2{}^3\cdot D_2{}^2$.

The useful power equals ρQgH , which becomes $\eta_v\cdot\rho Q_{imp}gH$ after introducing the volumetric efficiency. In the former, the rise in useful specific energy, ρQgH , is determined from the impeller specific work, $c_{u2}u_2$, after subtraction of energy dissipated in the stage (inlet and outlet losses are neglected here). To account for the head loss in the impeller, the impeller specific work is multiplied by the impeller hydraulic efficiency, η_{hyd}. Thus, the rise in useful specific energy between the impeller inlet and outlet sections becomes $\eta_{hyd}\cdot c_{u2}u_2$. The head losses in the annular chamber and in the return channels have to be subtracted from this quantity in order to obtain the rise in useful specific energy.

The head loss in the annular chamber can be represented as a loss coefficient multiplied by the dynamic head at the impeller outlet (Fontana, 2017). The influence of the geometry is neglected in the present analysis, since the stage casing geometry was assumed to be fixed. The return channel loss is partly due the fraction of the dynamic head that is not recovered into static pressure (i.e. the diffusion loss), and partly to wall friction and other loss mechanisms. Only the first contribution is considered in the following. Accordingly, the total loss through the annular chamber and the return channels is proportional to the dynamic head at the impeller outlet by means of a loss coefficient, ζ. The dynamic head at the impeller outlet can be calculated as $(1-\varepsilon)\cdot c_{u2}u_2$, where ε is the degree of reaction. Accordingly, the rise in useful energy through the pump reads:

$$gH=\eta_{hyd}\cdot c_{u2}u_2-\zeta\cdot(1-\varepsilon)\cdot c_{u2}u_2=\left[\eta_{hyd}-\zeta\cdot(1-\varepsilon)\right]\cdot c_{u2}u_2\,. \tag{1}$$

Consequently, the pump efficiency reads:

$$\eta=\eta_{mec}\cdot\frac{\eta_v\cdot\rho Q_{imp}gH}{\rho Q_{imp}(c_{u2}u_2)+P_{df}}=\eta_{mec}\eta_v\cdot\frac{gH}{(c_{u2}u_2)+{P_{df}}/{\rho Q_{imp}}}=$$

$$= \eta_{mec}\eta_v \cdot \frac{\lfloor \eta_{hyd} - \zeta \cdot (1-\varepsilon) \rfloor \cdot c_{u2} u_2}{(c_{u2} u_2) + \dfrac{k \cdot u_2^{\;3} \cdot D_2^{\;2}}{Q_{imp}}} \tag{2}$$

where η_{mec} is the pump mechanical efficiency (see Gülich, 2008). In order to highlight the impeller loading factor in Eq. 2, both the numerator and denominator are divided by the square of the blade tip velocity. Accordingly, the pump efficiency reads:

$$\eta = \eta_{mec}\eta_v \cdot \frac{\lfloor \eta_{hyd} - \zeta \cdot (1-\varepsilon) \rfloor \cdot \psi}{\psi + \dfrac{k \cdot u_2 \cdot D_2^{\;2} \cdot \eta_v}{Q}} = \eta_{mec}\eta_v \cdot \frac{\lfloor \eta_{hyd} - \zeta \cdot (1-\varepsilon) \rfloor \cdot \psi}{\psi + \dfrac{k \cdot \eta_v}{2 \cdot \Phi}}, \tag{3}$$

where Φ is the dimensionless flow number.

For sake of simplicity, in the following it is assumed that the mechanical and volumetric efficiencies are independent on Ψ. Moreover, since their value is usually high near BEP, both efficiencies are assumed equal to unity. Therefore, eq. (3) becomes:

$$\eta = \frac{\lfloor \eta_{hyd} - \zeta \cdot (1-\varepsilon) \rfloor \cdot \psi}{\psi + \dfrac{k}{2 \cdot \Phi}}. \tag{4}$$

The value of k depends on Ψ because of: (i) variations in the leakage flow rate; (ii) variations in the momentum of the rotating core of fluid (see Gülich, 2003). However, both dependences are considered as secondary effects and are neglected in the following. Note that k depends also on the Reynolds number and on the geometry of the impeller side rooms. Nevertheless, a constant value of k is considered hereinafter, because the geometry, the rotational velocity and the viscosity of the fluid are fixed in the model.

According to Gülich, 2008, the impeller hydraulic efficiency changes only a few points percent for n_q between 27 and 80. Thus, η_{hyd} is considered as a constant in the present study. It is worth to note that eq. (4) provides only an indication of the attainable efficiency, but it cannot be directly compared with measured values, due to the many assumptions and simplifications considered in this study. The latter are partly due to the lack of accurate data regarding all of the loss mechanisms involved, and partly to a modelling choice. In fact, the aim of this model is to highlight the main fluid-dynamic features of CD stages. For this purpose, a simpler analytical model is advantageous. Nonetheless, a more accurate model can be obtained by removing these simplifications and replacing the parameters which are assumed to be constant with appropriate correlations.

The degree of reaction and the impeller loading factor are strictly linked one each other. In fact, for swirl-less inlet velocity and constant meridional velocity, the degree of reaction is:

$$\varepsilon = \frac{c_{u2} u_2 - 0.5 c_{u2}^{\;2}}{c_{u2} u_2} = 1 - 0.5 \cdot \psi.$$

Therefore, eq. (4) can be solved in terms of one of the two quantities, when the appropriate values of k, η_{hyd} , and Φ are provided. In the following, the equations are solved with reference to impeller loading factor, in order to evaluate the effect of this design parameter on the performance and efficiency of the stage.

Theoretical results

The assumptions considered in the following are: k= 9.19E-4, η_{hyd} =0.90, Φ=7.32E-2. The values of k and Φ correspond to a real test case and the choice of the hydraulic efficiency is based on the results of numerical simulations.

For the choice of ζ it is observed that: (i) according to Fontana, 2017, the head loss coefficient for the annular chamber ranges between 0.2 and 0.8; (ii) the recovery of the dynamic head in the return channels is very slight (see for instance Zhou et al., 2012); (iii) the experimental results previously shown suggest that the dynamic head at the impeller outlet is completely lost. Consequently, the value of ζ is expected to range between 0.5 and 1.

Fig. 6 reports the model results corresponding to three values of ζ, namely: 0.35, 0.75 and 1. It is worth to note that Fig. 6 can be used to evaluate the effects of a change in a single design parameter (i.e., the impeller loading factor) once all of the other design parameters have been fixed. Accordingly, it does not allow to compare different designs or specific speeds, such as Fig. 2 and 3 do. The left side of the figure shows the predictions of efficiency according to eq. (4), as a function of Ψ. The graphs highlights that, for the smallest value of ζ, the curve is rather flat for values of the impeller loading factor in the range between 0.3 and 0.7. The optimal efficiency is reached at a value of Ψ between 0.4 and 0.5, that is consistent with the literature on radial flow impellers (see Gülich, 2008). As already noted, a comparison between these results and experiments in terms of absolute figures is not appropriate, due to the simplifications considered in the model.

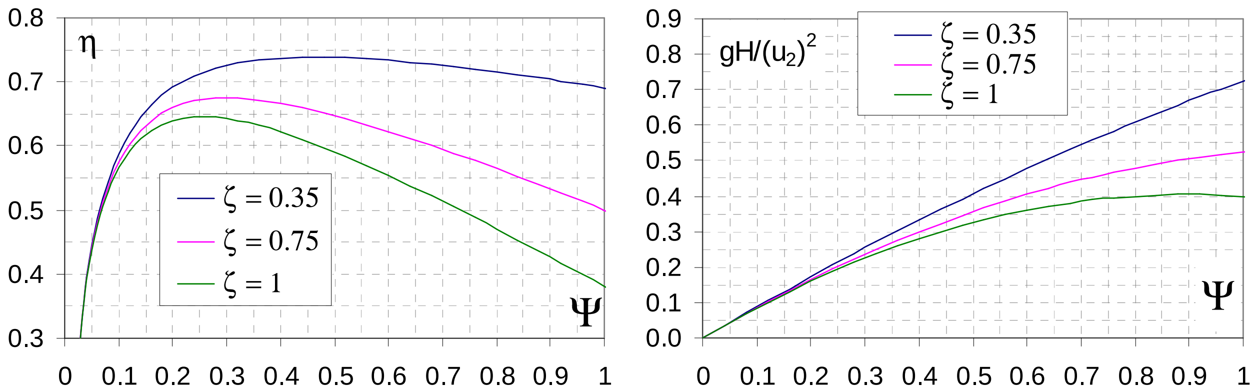

Fig. 6: results of the theoretical model. Left side: efficiency as a function of the impeller loading factor; right side: head coefficient as a function of the impeller loading factor.

In turn, when the value of ζ increases, the efficiency curve decreases faster with increasing Ψ, due to the increasing fraction of the dynamic head, which is dissipated to a large extent. Therefore, the peak becomes more evident and moves toward left, in the range of Ψ between 0.2 and 0.3.

Accordingly, to improve the efficiency of the CD stages (which do not allow for a significant recovery of the dynamic head), the value of Ψ should be selected relatively small. As a consequence, the head coefficient would result below the values in Fig. 3.

The dimensionless head coefficient, based on eq. (1), is shown on the right side of Fig. 6 as a function of Ψ. Clearly, the head decreases with increasing values of ζ. For ζ equal to 0.35, the values of dimensionless head coefficient suggested in Fig. 3 for radial flow machines are obtained when Ψ ranges between 0.45 and 0.6, in agreement with Gülich, 2008. In turn, for higher values of ζ the figure suggests that Ψ should range between 0.4 and 0.5 to obtain the values of head coefficient typical of CD stages (see Fig. 5). On the contrary, if the efficiency was maximised the value of the head coefficient would range between 0.2 and 0.25, that is 25% to 50% below the values reported in Fig. 5. Moreover, the graph shows that for ζ larger than 0.75 it is impossible to achieve the values of head coefficient according to Gülich, 2008, even in case Ψ was increased up to 1.

In the light of this, it can be stated that:

1) For CD stages, it is not worth to consider the values of the dimensionless head coefficient that are suggested for diffuser pumps. In fact, this would result in a sever decrease in efficiency.
2) At present the industrial solutions are the result of a compromise between performance and efficiency, and their efficiency could be raised at the price of a lower head coefficient.

Eq. (3) highlights that the results are influenced by the BEP flow rate. In particular, if no other parameter is changed, an increase in the BEP flow rate results in a decrease in the relative importance of the disk friction contribution. Thus, the dissipation of the dynamic head becomes the leading contribution and smaller values of the impeller loading factor are required to obtain the maximum efficiency. However, the qualitative comments regarding Fig. 6 still hold.

CONCLUSIONS

This work analysed the efficiency and the performance of multistage centrifugal pumps, with focus on compact design machines made by sheet-metal forming and welding. Indeed, the comparison between Reg. 547/2012 EC and the scientific literature showed that multistage pumps still need significant research efforts to meet European laws on efficiency, in particular when dealing with small sized machines.

The paper presented a collection of data of efficiency and performance measured on state-of-art, sheet metal, multistage pumps produced by several manufacturers. The comparison between the measured efficiency and the scientific literature showed that the latter is misleading for sheet metal, multistage pumps. In turn, the comparison on performance suggested that: (i) the values of head coefficient provided for traditional pump types are not suitable for compact design pumps; (ii) the most part of the dynamic head provided by the impeller is dissipated in compact design machines.

In the light of this, compact design, multistage pumps need dedicated design strategies to reach a high efficiency. Accordingly, a conceptual model was used to investigate the influence of the impeller loading factor on the efficiency of compact design stages. The theoretical results suggested that the maximum efficiency is reached for values of the impeller loading factor that are significantly smaller than those suggested for traditional designs. These optimal values are below the reported experimental data for industrial, compact design machines. This suggests that the efficiency of compact design machines could be improved at the price of a decrease in the head coefficient.

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