Influence of volutes on characteristic curves of centrifugal pumps, 1959

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OF CENTRIFUGAL PUMPS

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Experiments indicate that the characteristic curves of centrifugal pumps can be influenced by the shape of the volute located after the impeller as much as by the construction of the impeller itself. The published research on centrifugal pumps treats almost exclusively a single impeller and the flow conditions in it without consideration of interrelated influence between the impeller and the volute of a real pump. This paper should direct attention to this interaction and thus help to prepare development of a refined design procedure of centrifugal pumps.

Computation of characteristic curves of centrifugal pumps is still uncertain and is more or less satisfactory only if experimental results of similar machines are used. The reason for this uncertainty is that the theory developed until now gives useful predictions about the flow in the runner only for the frictionless flow. The retarded flow is found to a great extent in the centrifugal pump and this fact alone reduces the accuracy of predictions of the discharge head, capacity and efficiency compared to the turbine flow.

The early investigations of the theoretical and experimental type dealt almost exclusively with the impeller of the pump. They increased our knowledge of the flow conditions
in pumps and gave pump manufacturers a valuable basic knowledge. On the other hand the volute of the centrifugal pumps was relatively neglected. This is very strange since the efficiency of centrifugal pumps depends considerably on a possibly fuller transformation of kinetic energy behind the impeller into pressure, Fig. 1 \( ^{(1)} \). The efficiency is the ratio of the useful output of a pump to the applied energy. Specific speed \( n_s \) is the number of revolutions per minute at which a geometrically similar pump with a head of one meter gives exactly one horsepower of useful output. It has been long known in practice that changes on the volute influence not only the so-called starting point but also the whole outline of the characteristic curve. The influence can be as great as that obtained by changing the angle or number of the blades in the impeller. For the demonstrative comparison of the characteristic lines the following non-dimensional coefficients are used:

\[
\text{Pressure coefficient } \psi = \frac{H}{u_2^2/2g} = \frac{H}{D_2^2 n^2} 7150, \\
\text{Discharge coefficient } \varphi = \frac{Q}{D_2^2 b_2 n} 592
\]

where \( H \) is head in meters, \( Q \) is discharge in \( \text{m}^3/\text{h} \), \( n \) revolutions in \( \text{rpm} \), \( g = 9.81 \text{ m/s}^2 \), \( u \) is the circumferential velocity in \( \text{m/s} \), \( D \) is the diameter of the impeller in meters and \( b \)

\(^{(1)} \) Fig. 1 is taken from F. Krisam; Die Grenzen der Verwendbarkeit der Kreiselpumpen (Limits of Applicability of Centrifugal Pumps), Technik, V.3 (1948), p. 306, Fig.3.
is the width of the flow passage in meters. Subscript 2 refers to the discharge of the impeller. In the figures, the attained efficiency is arbitrarily set to 100.

The Influence of the Number of Blades.

Figure 2(2) shows the influence of the number of blades of the impeller on the characteristic curves of the pump for one stage pump with guide vanes and a spiral; the volute, angle and length of the blades were held constant while the number of the blades was varied. The results correspond reasonably well to the ones expected. The pressure coefficient and the efficiency grow with the increasing number of blades. $\psi$ also increases for the point of the best efficiency.

Figure 3 shows experiments for the same pump with unchanged impeller but with a different number of guide vanes. The changes of the highest efficiency, as well as of $\psi$ at the point of the maximum efficiency, are approximately the same as in Fig. 2. The discharge coefficient $\psi$ did not change, contrary to the later conducted experiments. This may depend on the shape of the adjoining spiral in this particular case. It should also be noted, that the increase of efficiency and pressure coefficient does not directly follow from the number of guide vanes.

(2) The test results are from the Exp. Station of the firm Klein, Schanzlin & Becker A. G. in Frankenthal.
The Influence of Guide Ribs and Shape of the Spiral.

Figure 4 shows experiments with the same impeller but with a differently formed spiral. The lower characteristic curve was obtained with a simple spiral and the upper one with a so-called double spiral, Figs. 5 and 6. This double spiral approximately corresponds to the volute with two guide vanes. Also in this case the coefficient remains almost unchanged. The pressure coefficient, however, is constantly higher with only slightly changed efficiency. Both spirals had exactly the same casing and differed only by the middle rib.

Figure 7 gives an especially impressive picture of the influence of the volute. Here the same impeller with three different spirals was investigated. The spirals were developed differently while keeping the same magnitude of pressure drops. Primarily, the end cross sections at discharge of the spirals were kept different in size. The pressure coefficient and capacity coefficient could be changed here considerably without influencing efficiency. The absolute efficiency was (as in the previous experiment - Fig. 4) quite high (84-85%) for these pumps and could hardly be increased considering the size of the pumps. Table 1 gives a comparison of the various coefficients. The subscript "opt" (optimum)* designates here the values at the point of highest efficiency. As seen, the outline of the characteristic

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* Translator's addition
curves depends on the shape of the volute while the relatively high efficiency remains constant. These results immediately give rise to some important questions. What happens, for example, to the "impact-free influx" with this impeller which for \( \psi = 0.095\% \), as well as for \( \psi = 0.152 \), continuously gave good efficiency? Figure 8 shows triangles of the velocity at the entrance of the impeller (suction eye) for three points of the best efficiency. The considerable "impact components" apparently could not appreciably influence the efficiency. Unfortunately, no complete information is given on suction head experiments. At any rate, for spiral II the cavitation coefficient \( \sigma = 0.13 \) was reached without any indication of cavitation at the point of the best efficiency. Even lower value may be used, \( \sigma = 0.13 \) for the obtained specific speed of 147 should be considered rather low as compared with the available curves \( \sigma = f(n_s) \). According to the formula by C. Pfleiderer \(^{(3)}\)

\[
\sigma = \frac{3.89 \times 10^4 \eta_h^2}{365} \left( \frac{n_s}{365} \right)^{4/3}
\]

for example, \( \sigma = 0.153 \), where \( \eta_h \) is hydraulic efficiency. Thus, in spite of the higher capacity coefficient with spiral II and the entrance impact, Fig. 8, the cavitation sensitivity of the impeller was not reduced. One may wonder if the

\(^{(3)}\) Cavitation coefficient \( \sigma \) denotes the lowest suction head for a discharge head of one meter for which no cavitation occurs.

present conception about the entrance impact should not undergo considerable correction. This conclusion will also be strengthened by the experiments on the suction head described below.

Comparison of Spirals and Guide Vanes.

The specific speed of an impeller according to these experiments depends not only on the shape of the blades but also considerably on the type of the volute. This is indicated in Fig. 9 and Table 2 in which impellers of the same discharge diameter $D_2$ and equal discharge width $b_2$ are compared. The blade angle $\beta_2$ at the discharge was between 21 and 30 degrees. The wheel "a" is equipped with a spiral, the wheel "b" - with guide vanes. The opt. discharge coefficient of the impeller "a" is about 50% higher than that of the impeller "b". Impellers "c", "d", and "e" have less extreme ratio of $D_2/b_2$. Impeller "c" has a spiral as a control apparatus. Impeller "d" was investigated with two differently built sets of guide vanes. Impeller "e" had an adjoining spiral besides a purely radial guide vane. The great differences in the discharge coefficient and in the specific speed, despite equal impeller diameters and widths and almost equal discharge angles, are obviously caused by the volute (spiral or guide vanes). In general, the guide vanes reduce the discharge coefficient in comparison with the simple spiral. Both experiments with wheel "d" show, however, that the discharge coefficient of the spiral can be attained also with guide vanes. The outside diameter of the guide vanes
was the same in both cases. Only impellers of approximately the same size with about the same good efficiency were compared. These examples would emphasize what could be seen above: the shape of the impeller specified by dimensions and blade angles does not permit reliable prediction of discharge coefficient, i.e., neither flow rate nor speed.

All the experimental results given previously were obtained with one-stage pumps. The multi-stage pumps with guide vanes and variable blades behave quite similarly, as Fig. 10 indicates. Both curves were obtained with the same impeller but with two different sets of guide vanes whose outside diameters were equal. Whereas absolute efficiency was about the same, the discharge coefficient in one case was 0.07, in the other 0.10. The exact suction head experiments were also conducted on this pump; it was observed: $\sigma = 0.086$ with $n_s = 97$ and $\sigma = 0.048$ with $n_s = 72.5$ while the formula by Pfleiderer gives $\sigma = 0.095$ for $n_s = 97$ and $\sigma = 0.066$ for $n_s = 72.5$. The obtained $\sigma$ values should be considered favorable although as Fig. 11 shows, in both cases there was no "impact-free" inflow. These results, therefore, verify the conclusions of the discussion of Fig. 7.

The Influence of Inflow.

Up to now we have shown the possibilities of influencing the characteristic curves of pumps or individual specific numbers by means of controlling discharge flow. Inflow control is seldom used for the pumps of liquids; a possibly whirl-free inflow to the impeller is sought for. Due to reasons
of construction, the entrance elbows often cannot be avoided near the impeller; very often they must be very close to it. Development of initial whirls is very difficult to avoid, especially if there are many elbows near the pump in the inflow pipe. Figure 12 shows how such initial whirls reduce the discharge head and efficiency (solid curves). The initial whirls can be considerably removed by introduction of a correctly located guide rib in the inflow elbow; the discharge head is increased exactly for 10% and the efficiency for 4% (dashed curves). The shape of the inflow elbows is of secondary importance for the whirl-free inflow. The three very different shapes of inflow elbows, Fig. 13, give practically equal characteristic curves for discharge head and for efficiency. Also the cavitation coefficients were the same up to the discharge coefficient of the best efficiency; only for higher discharge coefficients the elbow shape gave worse values, whereas no difference could be determined between shapes "b" and "c". The middle rib serving as a support of the shaft was common to all elbow shapes. Thus, it may be concluded, that if the initial turbulence could be avoided no important influence from the inflow side can be expected on the characteristic curves of pumps.

CONCLUSIONS

The experimental results reported here represent only a selection of especially typical examples of various influences between impeller and other parts of a pump, and they can certainly be verified by other pump manufacturers. The
observations clearly point to a gap in the design methods of centrifugal pumps:

1. The running impeller cannot be considered a unit in itself; the pertaining volute should also be considered. We find an analogy to this in the influence by the suction pipe elbows and the runner of a water turbine. The suction pipe elbow of a turbine has a similar purpose as the volute of a pump, namely, the utilizing of the discharge energy behind the running impeller.

2. The experimental results are difficult to explain by the usual consideration of the "inflow impact". There is much experimental data which indicates that neither high efficiency nor big suction head require so-called "impact-free-inflow". Consideration of the finite blade number requires, according to the theory, a reduction of the entrance angle. The practice, however, makes use of considerable angle increase which (without reducing the efficiency) proved to be an effective means to improve cavitation limit.

Our knowledge of the interaction is still incomplete. The useful assumptions for numerical computations of the influences of the volute are not yet known. Only through experiments, careful and critical collection of practical results can this situation be corrected.
TABLE 1

Comparison of Coefficients at Point of Highest Efficiency for Experiments in Fig. 7.

<table>
<thead>
<tr>
<th>Type of Spiral</th>
<th>Specific Speed $n_s$</th>
<th>Highest Efficiency $\eta_{\text{max}}$</th>
<th>Discharge Coeff. $\gamma_{\text{opt}}$</th>
<th>Pressure Coeff. $\nu_{\text{opt}}$</th>
<th>Ratio of $\nu$ at zero rate to $\gamma_{\text{opt}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spiral I</td>
<td>117</td>
<td>84.5</td>
<td>0.0955</td>
<td>0.98</td>
<td>1.245</td>
</tr>
<tr>
<td>Spiral II</td>
<td>147</td>
<td>84.5</td>
<td>0.120</td>
<td>0.85</td>
<td>1.33</td>
</tr>
<tr>
<td>Spiral III</td>
<td>191</td>
<td>84.5</td>
<td>0.152</td>
<td>0.70</td>
<td>1.46</td>
</tr>
</tbody>
</table>

TABLE 2.

Specific Data for Impellers in Fig. 9.

<table>
<thead>
<tr>
<th>Type</th>
<th>Spiral Vanes</th>
<th>Guide Vanes 1°</th>
<th>Guide Vanes 2°</th>
<th>Spiral and Guide Vanes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Notation in Fig. 9</td>
<td>a</td>
<td>b</td>
<td>c</td>
<td>d₁</td>
</tr>
<tr>
<td>Diameter-Width Ratio at Discharge $D_2/b_2$</td>
<td>24.4</td>
<td>24.4</td>
<td>17</td>
<td>17</td>
</tr>
<tr>
<td>Discharge blade angle $\beta_2$</td>
<td>30°</td>
<td>21°</td>
<td>30°</td>
<td>28°</td>
</tr>
<tr>
<td>Spec. speed $n_s$ rpm</td>
<td>76</td>
<td>61</td>
<td>106</td>
<td>71</td>
</tr>
<tr>
<td>Discharge coeff. $\gamma_{\text{opt}}$</td>
<td>0.0963</td>
<td>0.0673</td>
<td>0.126</td>
<td>0.073</td>
</tr>
<tr>
<td>Pressure coeff. $\nu_{\text{opt}}$</td>
<td>0.93</td>
<td>0.98</td>
<td>1.0</td>
<td>1.12</td>
</tr>
</tbody>
</table>
Fig. 1  EFFICIENCY FOR PUMPS WITH SPIRAL VOLUME WITH OR WITHOUT THE VANES VERSUS SPECIFIC SPEED, \( n_s \).

- a - actual pumps with spiral volume
- b - actual pumps with spiral volume and guide vanes
- c - theoretical curve for pumps with impeller side friction as the only loss according to the formula for theoretical efficiency

\[
\eta_{th} = \frac{n_s^2}{(480 + n_s^2)}.
\]

Fig. 2  INFLUENCE OF NUMBER OF BLADES OF THE IMPELLER ON CHARACTERISTIC CURVES.

Solid, point-dashed and dashed lines stand for impeller with 5, 6, and 7 blades respectively.
Fig. 3 INFLUENCE OF THE NUMBER OF GUIDE VANES

Fig. 4 INFLUENCE OF MIDDLE RIB IN THE SPIRAL ON CHARACTERISTIC CURVES

Solid or dashed curves stand for single or double spiral, respectively, see fig. 5.

$\eta$ - Efficiency $\phi$ - Pressure Coeff.

$\psi$ - Discharge $n_s$ - Spec. Speed Coeff.
Fig. 5 and 6  SHAPE OF VOLUTES FOR CHARACTERISTIC CURVES IN Fig. 4.

Fig. 7  CHARACTERISTIC CURVES FOR AN IMPELLER WITH THREE DIFFERENT VOLUTES (SPIRALS).
Solid, point-dashed, and dashed lines for spirals I, II, and III, respectively.

Fig. 8  DIAGRAMS OF VELOCITY AT IMPELLER INFLOW AT THE HIGHEST EFFICIENCY FOR Fig. 7

Subscripts correspond Fig. 7.
\( u \)  Circumferential velocity
\( C_m \)  Meridional (Radial) component of Absolute velocity
\( w \)  Relative velocity
Subscript 1 denotes inflow (entrance).
Fig. 9 IMPPELLER SHAPES WITH THE SAME DIAMETER/WIDTH RATIO, \( D_2/b_2 \), AT DISCHARGE AND OPTIMUM DATA FROM TABLE 2.

a and c - Impellers with spiral
b, d, and d2 - Impellers with guide vanes
\((d_1\text{ and }d_2\text{ differ only in shape})\)
e - Impeller with spiral and guide vanes

Fig. 10 CHARACTERISTIC CURVES OF THE SAME IMPPELLER WITH A Volute AND GUIDE VANES.

\( \eta \) - efficiency
\( \varphi \) - pressure coeff.
\( \gamma \) - discharge coeff.
\( n_b \) - specific speed in rpm

The dashed line is for \( \eta = 0.07 \),
and the solid line for \( \eta = 0.10 \).

Fig. 11 DIAGRAMS OF VELOCITY AT IMPPELLER INFLOW FOR IMPPELLER IN Fig. 10
Fig. 12 INFLUENCE OF INITIAL TURBULENCE WHEN THE INFLOW IS NOT WHIRL-FREE.

Solid lines indicate cases with and dashed lines without a guide rib in the suction eye.

Fig. 13 INFLUENCE OF DIFFERENT SHAPES OF THE SUCTION ELBOWS.

a, b, c - Elbow shapes  \( \eta \) - Discharge Coeff.
\( \eta \) - Efficiency  \( \sigma \) - Cavitation Coeff.
\( \varphi \) - Pressure Coeff.  \( n_s \) - Specific Speed, rpm