Effect of vane passage modification on performance and cavitation characteristics of a centrifugal pump

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Experimental work involving the modification of an impeller with straight front shroud to curved one to satisfy the condition of uniform variation of velocities in the vane channel is attempted. Inspection of vanes and vane channels using precise and accurate instrument will form an important step in the manufacture and realisation of an impeller. A method of such inspection procedure for vanes and vane channels is presented. The values of linear correlation coefficient of vanes and vane channels are used for evaluating the variations from one channel to another. Performance and cavitation characteristics of the pumps have been obtained experimentally at various discharges and speeds of nine bladed single stage centrifugal pump of shape number 95 using straight and curved front shrouded impellers. The results for two impellers have been compared.

The accuracy of manufacture of centrifugal pump impellers determine the closeness of the impeller vane course with the design. It is essential to achieve the expected variation of velocity along the vane channels. An attempt is made here to evaluate the loss in effectiveness of the pump with straight shrouded impeller. The impeller was first fabricated with straight front shroud and is identified as IMP4. Experiments were done with this impeller to evaluate its performance and cavitation characteristics. This impeller was later modified as one having curved front shroud in such a way that the vane passage area matches with the desired velocity variation in the channels. The modified impeller is designated as IMP5. Fig. 1 shows the flow passages of IMP4 and IMP5 in meridional view.

Fig. 1 – Flow passages of IMP4 and IMP5 in meridional view
linear correlation coefficient of all vanes on each side and of the vane channels. With these coefficients it was possible to correlate the accuracy of the vanes and vane channels of the impeller with the design values.

The impeller with straight front shroud (IMP4) and the modified one with curved front shroud (IMP5) were tested in the same pump casing and in the same test set up.

The performance curves of both impellers were plotted at various speeds and compared. Analysis of the performance curves of the two impellers were done to study the effect of the vane passage modification.

The complete cavitation characteristics of these two impellers were compared to study the effect of vane passage modification on cavitation inception. This result will be very useful for pump designers and manufacturers especially when they decide upon the shroud shape and fabrication techniques of the impellers.

Fabrication and Inspection of Impellers

The impeller was designed to be made in 3 main parts, viz., rear shroud, vanes and front shroud and assembled by means of a number of stainless steel screws. The impeller front shroud was made of acrylic plastic material to enable flow visualisation. The vanes were made from brass flats of 4.5 mm thickness. The flats were first cut approximately to the length of the vanes and bent to the curvature in a plate bending machine. The front shroud and the mating vane surface were straight taper for most of the portion with a radius of 10 mm at entry for IMP4. The taper angle was selected in such a way that the required front shroud profile as per design, i.e., IMP5, can be obtained by copy turning of IMP4. The variation of impeller width as a function of impeller radius of both the impellers is given in Table 1. The front surface of the vanes fitted on the rear shroud which is corresponding to the front shroud meeting the vanes, had to match with the inner surface of the front shroud. The surfaces of the acrylic front shroud were also copy turned with templates. The holes required for fixing the front shroud onto the vanes were drilled and tapped with the help of an index table.

Ferranti 3-dimensional coordinate measuring machine model 6100, was used for measurements. It is suitable for inspection of castings, small machined parts and printed circuit boards using solid or electronic probes with smallest unit of measurement of 0.001 mm.

For measurements, the probe was traversed along the vane. The vanes and vane channels are designated as A, B, C, etc. and AB, BC, CD, etc., respectively. The probe was moved along the vanes on pressure as well as suction side and the coordinates were measured at different arbitrary intervals.

The X and Y coordinates were used for plotting the measured values and is reproduced in Fig. 2a together with the meridional view in Fig. 2b.

Calculation of Linear Correlation Coefficients

$R$ and $\theta$ coordinates were used to find the linear correlation coefficient of each vane and on both sides.

Linear correlation coefficient $= \frac{\sum(x \cdot y)}{\sum x^2 \cdot \sum y^2}$

### Table 1 - Comparison of impeller width

<table>
<thead>
<tr>
<th>Radius mm</th>
<th>IMP4 Width mm</th>
<th>IMP5 Width mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>80</td>
<td>38.80</td>
<td>37.00</td>
</tr>
<tr>
<td>90</td>
<td>34.90</td>
<td>33.00</td>
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<tr>
<td>100</td>
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<td>29.50</td>
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<tr>
<td>120</td>
<td>29.80</td>
<td>28.00</td>
</tr>
<tr>
<td>130</td>
<td>28.20</td>
<td>27.00</td>
</tr>
<tr>
<td>140</td>
<td>26.50</td>
<td>26.00</td>
</tr>
</tbody>
</table>
The linear correlation coefficients of all vanes and vane channels were found out for comparison of all vanes and vane channels. Tables 2 and 3 give the values of the linear correlation coefficient obtained for the vanes and vane channels, respectively. The percentage deviation with respect to design was found to be 0.3% for the vanes (with a standard deviation of 0.00280227) and for the vane channels the percentage deviation with respect to arithmetic mean was found to be 0.4% (with a standard deviation of 0.00388588). Thus, by determining the linear correlation coefficients, variations from one channel to another and variations in the vane course of the vanes could be found out. Correction of the variations is now possible so that the fabricated impeller matches with the design.

Experimental Procedure

The experimental set up has the following arrangements for measurements of various quantities. Speed was measured by an inductive pick up which was positioned in proximity to the coupling with slots around its periphery and an impulse counter. Torque was measured by the twist in the calibrated torsion bar due to the torque transmitted by the dynamometer. The twist was optically amplified and projected on a ground glass screen. Discharge was measured by a calibrated venturimeter of 150 mm nominal diameter and an area ratio of 0.411. The head of the pump was determined by measuring the suction and delivery pressures. The suction reading was also used for calculating the Net Positive Suction Head (NPSH).

The performance of the pump was initially determined at five different speeds ensuring cavitation-free operation. Cavitation test was conducted at a constant speed and discharge by increasing the static suction lift, i.e., decreasing NPSH and was continued till either full cavitation or lowest water level in the sump without loss of priming was reached. After completing the experiments with IMP4, it was machined by copy turning of the front shroud mating surfaces to obtain IMP5. The experiments were repeated with IMP5.

Results and Discussion

Non-cavitating performance – The head (H) versus discharge (Q) curves of IMP4 and IMP5 were drawn by curve fitting on the basis of the results of regression analysis and are shown in Figs 3 and 4, respectively. Iso-efficiency curves are also
The values of speed, head, discharge and efficiency of IMP4 and IMP5 are compared in Table 4. It was observed that the modification did not alter the best speed of operation. This agrees with the results of Ramachandran and Ravikumar. The best efficiency point (bep) has shifted to lower discharge and values of Head and efficiency at best efficiency point were reduced for IMP5. The flow coefficient \(C_{2m}/u_2\) and head coefficient \(H/(u_2^2/2g)\) calculated at best efficiency point of IMP4 and IMP5 are also given in Table 4. It is to be noted that the head coefficient and flow coefficient at best efficiency points were reduced due to the modification of the impeller vane passage.

The inlet and outlet velocity triangles are not the same for IMP4 and IMP5 due to the geometrical changes indicated in Table 1. For example, at design speed (1500 rpm), the meridional component of velocity just after the impeller outlet \(C_{3m}\) was lower for IMP5 and thus \(C_{2m}\) has also decreased. Due to this, peripheral component of absolute velocity at outlet \(C_{2a}\) also reduced leading to decrease in specific hydroenergy because the value of peripheral velocity at outlet remains constant. Hence, the discharge and head developed had reduced when compared with that of IMP4.

It is seen from Table 1 that width of IMP5 is less compared to IMP4. Worster suggests that such a variation in geometry could be related to their performance by constructing casing line and impeller line for a specific pump. Since the casing is not altered in the present situation, the non-dimensional points at best efficiency namely head coefficient and volume flow ratio defined by Worster should lie on a straight line passing through the origin. The average values of the Worster's head coefficient and volume flow ratio of IMP4 are 0.4522 and 0.2803, respectively. The above values are 0.4434 and 0.2568, respectively, for IMP5. These values do not vary on a straight line passing through the origin indicating that the deviation in performance at best efficiency points is possibly not due to impeller-casing interaction.

Further, the variation of cross sectional area as a function of impeller radius is brought out in Fig. 5. The variations in the flow passage has resulted in decrease of head and discharge at corresponding best efficiency points. The decrease in
head and discharge of IMP5 compared to IMP4 is due to the decrease of flow passage in the impeller. However, with all the above drops in discharge and head no drastic reduction in efficiency was noticed for IMP5 possibly because of the uniform variation of velocity from inlet to outlet of the impeller leading to an improved hydraulic efficiency.

Cavitational characteristics—The cavitational characteristics were drawn for different sets of speed and discharge. From the cavitational characteristics NPSH has to be evaluated for various head drops for known values of speed and discharge. The chosen head drops were 0, 0.5, 1, 2, 3, 5 and 10% of the head developed by pump at the operating speed and discharge.

The cavitational characteristics obtained at 1500 rpm for IMP4 and IMP5 are given in Figs 6 and 7, respectively. The values of head and discharge selected for the cavitational tests of IMP4 and IMP5 are not the same. Hence, it is necessary to compare the complete cavitational diagram of each impeller.

The comparison of the values of complete cavitational diagram of IMP4 and IMP5 at 0 and 5% head drops are given in Fig. 8. The NPSH/
NPSH$_{\text{bep}}$ has reduced due to the modification of the impeller vane passage. In other words the head drop occurs at a lower value of NPSH in the case IMP5 as compared to that of IMP4. Hence the pump can operate without cavitation to a slightly more value of suction lift of about 0.5 to 1 m. The improvement of cavitation characteristics is possibly due to uniform meridional velocity along the impeller flow passage.

### Conclusions

The coordinate measurements used for inspection of the vane channels enabled the accuracy of manufacture to be determined in comparison with design values. The checking of vanes and vane channels showed that the variation occurred due to fabrication method and fixing adopted for these impellers to be negligibly small. Similar inspection procedure can be followed for checking scale model pumps and fabrication of core box. In addition, establishment of a method of determining the correlation coefficients enabled the determination of the variations from one vane channel to another. Thus, it will be possible to minimise variations from channel to channel.

The noncavitating performance of the pumps determined at five speeds showed that the best efficiency points were shifted to lower discharge and head for the modified impeller compared to the impeller with straight front shroud.

The best efficiency was more or less the same for both the impellers indicating that head loss has decreased in the modified impeller due to the uniform variation of velocity from inlet to outlet resulting in a possible improvement in hydraulic efficiency.

The cavitation behaviour of IMP5 showed a marked improvement due to more uniform velocity variation in the modified impeller.

The results clearly indicate the necessity of matching the impeller passage consisting of vane and shrouds to obtain optimum performance and cavitation behaviour.

### Nomenclature

- $C$ = absolute velocity of the fluid
- $g$ = acceleration due to gravity
- $H$ = head of the pump
- NPSH = Net Positive Suction Head
- $Q$ = discharge of the pump
- $R$ = coordinate in radial direction
- $\overline{R}$ = arithmetic mean of $R$ values
- $u$ = peripheral velocity of the impeller
- $x$ = normalised value of $R = (R - \overline{R})/\overline{R}$
- $y$ = normalised value of $\theta = (\theta - \overline{\theta})/\overline{\theta}$
- $\theta$ = coordinate in angular direction
- $\overline{\theta}$ = arithmetic mean of $\theta$ values
- $\eta$ = overall efficiency of the pump

### Subscript

- bep = best efficiency point
- $m$ = meridional component of velocity
- $u$ = peripheral component of velocity
- 2 = at impeller outlet, inside the impeller
- 3 = at impeller outlet, outside the impeller

### References