Design optimization of a low specific speed centrifugal pump with an unshrouded impeller for cryogenic liquid flow

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Abstract. Rocket engines require centrifugal pumps used in its propellant feeding system to be smaller and lighter to increase the final speed of payloads, and be produced with lower costs. Reducing the number of stages and removing shroud rotating wall make pumps lighter and easier to manufacture. Therefore, these pumps should be designed with higher head to reduce the number of stages and have an unshrouded impeller. In this study, various shapes of shrouded and unshrouded impellers with the same meridional plane, that have various blade angle distributions, and splitter blade shapes, have been analysed by a computational fluid dynamics (CFD) approach. Based on this survey, the impeller blade shape which demonstrates the highest head and efficiency has been designed considering the internal flow of the pump. As a result, it was observed that unshrouded impellers had the loss generation structure due to tip leakage flow which was different from shrouded impellers had. Based on the above, a design method considering the tip leakage flow of unshrouded impeller was suggested.

1. Introduction

Cryogenic centrifugal turbo pumps for rocket engines have one of the largest output relative to its size. They are operated with high rotation speed to pressurize the propellant above the combustion pressure, and feeds it to the combustion chamber. Rocket engines require turbo pumps to be smaller and lighter to increase the final speed of its payloads. Moreover, due to recent intense competition of commercial orbital transportation, the pumps are required be produced with lower cost because these are used only once or a few times. For reducing the number of stages cut the weight, cost and time to manufacturing of components, and it is easier to manufacture unshrouded impellers by machine-cutting than shrouded impellers by casting. Therefore, these pumps should be designed with higher head to reduce the number of stages, and with unshrouded impellers.

Senoo [1] measured the outlet flow velocity of an unshrouded impeller type centrifugal compressor. Based on the difference of the slip factor influencing the tip clearance, their loss structure caused by leakage flow rate was clarified. Recent reports said that the tip clearance flow changes mainstream and affect the performance or stability of impellers [2][3][4][5]. Kaneko et al. [6] reported in the survey by a computational fluid dynamics (CFD) approach that tip leakage vortex causes the loss due to its blockage effect, interference with splitter blade and static pressure reduction. In addition, unshrouded centrifugal pumps with higher head have lower specific speed, hence its internal flow is more affected by tip leakage flow compared to pumps with higher specific speed because they have lower blade height. Therefore, it is expected that the inner flow pattern of unshrouded impeller with lower specific
speed is different from that of unshrouded, and the adequate design method for unshrouded impellers is different from that for shrouded impellers. The purpose of this survey is to suggest an adequate design method for unshrouded impellers by comparing various shapes of shrouded and unshrouded impellers with the same meridional plane, that have various blade angle distributions, and splitter blade shapes, using CFD.

2. Numerical method

2.1. Pump model
The centrifugal pump model under researched is a liquid hydrogen turbo pump designed for rocket engines, which has axial inducer, centrifugal impeller and radial vaned diffuser. Table 1 shows the details of the hydrogen turbo pump impeller. This impeller is given prewhirl from the axial inducer. In this survey, only the impeller was researched.

**Table 1.** The details of the hydrogen turbo pump impeller.

| Items          | Values                  |
|----------------|-------------------------|
| Specific speed | $143.2 \text{ min}^{-1}, \text{m}^3/\text{min}, \text{m}$ |
| Flow Coefficient $\phi$ | 0.0883                  |
| Reynolds Number | $1.078 \times 10^9$     |
| $D_1 / D_2$    | 0.439                   |
| $b_c / b_2$    | 0.0342                  |
| Number of Blade | Full:10, Splitter:10   |

2.2. Parametric Study setup
In order to research the influence of the impeller shape on its performance, the following parametric study was carried out.

2.2.1. Blade angle distribution
Blade shape of impeller which directly affect the internal flow pattern of pumps is determined by distribution of blade angle from circumferential direction. In this survey, 7 distributions of blade angle showed by Figure 2 were prepared. “Neutral” distribution has a constant increase rate of blade angle from LE to TE. “Front”, “Mid” and “Aft” distributions have a maximum increase rate of blade angle at first, middle and end of blade meridional area, respectively. “Front2”, “Mid2” and “Aft2” distribution have emphasized blade angle at first, middle and end of blade meridional area, respectively. These distributions are expected to increase the blade loading at each modified area. Impellers were made based on “Neutral” distribution with 70[\%] length splitters, and the impeller with each distribution at hub side, shroud side and both side of span wise location were prepared.

2.2.2. Splitter blade length
Splitter blades rectify the interflow of full blades and support loading of the full blades. For splitter blades to work, a sufficient length ratio to the full blades is required. In this survey, 3 types of splitter blade with 50[\%], 60[\%], and 70[\%] length of the “Neutral” full blades were prepared.

2.2.3. Shrouded and Unshrouded
To clarify the effect of tip leakage flow, shrouded and unshrouded impellers of “Neutral” shape were prepared.
2.3. CFD set up
The numerical analysis of the different blade shapes was performed using ANSYS CFX 17.2 solving Reynolds-Averaged Navier-Stokes (RANS) equations and Shear Stress Transport (SST) k-omega model, which are practical for computational fluid analysis of centrifugal pumps [7][8]. The predictability of this computational method for macroscopic performance of pumps has been confirmed experimentally using a centrifugal room temperature air compressor which has the meridional plane similar to the currently researched turbo pump [9]. Structural grids with about 4 million nodes as shown in Figure 3 were created for each impeller. The properties of hydrogen were assumed to be uncompressible, the density to be 70\( \frac{kg}{m^3} \), and the viscosity to be 11.6\( \times 10^{-6} [Pa \cdot s] \). Since only the impeller is under researched in this survey, the estimated velocity distribution of the inducer outflow was extrapolated at the inlet of the impeller.

2.4. Performance evaluation method
The efficiency of the impeller is evaluated by the ratio of output head and input torque. In this survey, the values of efficiency were calculated by the increase of total pressure from the inlet to the outlet.

\[
\eta = \frac{\rho g Q \Delta H}{\omega T} \quad (1)
\]

The head of the impeller is evaluated by increase in pressure or angular momentum from inlet to outlet of the impeller. In this survey, 3 evaluation methods shown below were used.
2.4.1. **Velocity head**

The values of angular momentum on the control surface is calculated as the product of the averaged circumferential velocities of the blade and fluid. Then the difference between inlet and outlet is computed. When the slip of the outlet could be ignored, this method is generally referred to as Euler’s head considering the slip factor.

\[ \Delta H = \frac{\Delta L}{\rho g Q} = \frac{\omega}{g} (r_2 v_2 \cos \alpha_2 - r_1 v_1 \cos \alpha_1) \]

In this method, the averaged velocity is calculated using the area average method, which insures that the product of the area of control surface and the average velocity is equal to the flow rate.

\[ \Delta H = \frac{1}{g} \left( \frac{\int u_2 ds}{S} \frac{\int v_{z2} ds}{S} - \frac{\int u_1 ds}{S} \frac{\int v_{z1} ds}{S} \right) \]

2.4.2. **Total pressure head**

The values of total pressure on control surface were calculated using the mass flow weighted average method, which could calculate the amount of energy passing through control surface correctly.

\[ \Delta H = \frac{\Delta P_t}{\rho g} = \frac{1}{\rho g} \left( \frac{\int P_{t2} d\dot{m}}{M} - \frac{\int P_{t1} d\dot{m}}{M} \right) \]

2.4.3. **Angular momentum head**

The values of angular momentum on control surface is calculated by averaging the mass flow rate products of the local circumferential velocities of the blade and fluid.

\[ \Delta H = \frac{\Delta L}{\rho g Q} = \frac{1}{g} \left( \frac{\int u_2 v_{z2} d\dot{m}}{M} - \frac{\int u_1 v_{z1} d\dot{m}}{M} \right) \]

3. **Results of parametric study**

3.1. **Overall performance rating**

Figure 4 shows the efficiency values of each unshrouded impeller, and Figure 5 shows the head coefficient values by 3 methods and the converted head coefficient values using the formula of (6) from input torque of each unshrouded impeller. The formula (7) was used for the coefficients.

\[ T_H = \frac{\omega}{\rho g Q} T \]

\[ \psi = \frac{\Delta P_t}{p u_z^2} \]

Since the values of velocity head coefficient defined in section 2.4.1 clearly exceeded the input torque coefficient, it was confirmed that the evaluation method with the velocity head is not effective. The trend of angular momentum is similar to that of input torque, which is different to that of the total pressure. It appears that the energy of input torque increased the angular momentum, and the angular momentum increases the total pressure involving diffusing loss.

The impeller with the “Front2” angle distribution had the highest efficiency than all other angle distributions, and “Neutral” impellers with 50, 60 and 70[\%] splitters had almost same efficiency as “Front2”. “Mid” impeller realizes remarkably low efficiency, due to lower total pressure increase.

Focused on the only half-side changed impeller, the efficiency trend of the shroud-side changed impeller is similar to that of the both-side changed. It suggests that the efficiency of impeller depends on the blade shape at shroud side.
Figure 4. The efficiency values of each unshrouded impeller.

Figure 5. The head coefficient values and the converted head coefficient values from input torque of each unshrouded impeller.

3.2. Shrouded and Unshrouded

In order to clarify the loss region in impellers, loss coefficient was defined by the following equation.

\[ I_R = \frac{p}{\rho} + \frac{w^2}{2} - \frac{(r_2 \omega)^2}{2} \]  \hspace{1cm} (8)

\[ \zeta = \frac{I_R - I_{R_1}}{(r_2 \omega)^2/2} \]  \hspace{1cm} (9)

Figure 6 shows the streamlines through the large loss coefficient region in “Neutral” shrouded and unshrouded impeller. The streamlines of the shrouded impeller show that the separation vortex with backflow at suction side of blades closed flow paths between blade to blade. The loss was mainly caused by the back flow of vortex. On the other hand, the streamline of the unshrouded impeller show
that the loss is mainly caused by the static pressure reduction at tip leakage vortex core, and the tip leakage flow prevent the formation of strong vortex in the separate region at suction side of blades. Thus, since the factor that governs the flow pattern of the unshrouded impellers are different from that of shrouded impellers, it is confirmed that the adequate design method is needed for unshrouded impellers.

Figure 6. The streamlines through the large loss coefficient region in shrouded impeller.

3.3. Splitter blade length

Figure 7 shows the magnitude of torque on the main blades and splitter blades. It appears that longer splitters have a larger torque. Figure 8 and 9 show the velocity map on the 50[%] span faces of “Neutral” impellers with 50[%] and 70[%] splitter lengths. For the 50[%] splitters, the main stream from the inlet went to the suction side of the splitter because the blockage region caused by separation closed at pressure side of splitters. On the other hand, for the 70[%] splitters, the main stream was parted to pressure side and suction side of splitter, because the blockage region did not close at pressure side of splitters. Hence, the amount of fluid loading on 50[%] splitters was smaller than that of 70[%] ones. Moreover, in the impeller with “Front” and “Front2” blade angle distribution, the separation started upstream, then the blockage region closed at pressure side of splitters. In this way, the blade load on the splitters decreased.

Figure 7. The magnitude of torque on main blades and splitter blades.
3.4. Blade angle distribution

Figure 10 shows loss coefficient $\zeta$ maps, defined by the formula (9), on 90[%] span faces of the impellers with each blade angle distribution. The maps show that the magnitude of loss caused by tip leakage vortex. The loss of impellers with higher efficient such as “Front2” and “Neutral” is smaller than that of the other impellers. Figure 10 shows the vortex core region of the impellers. It clarified that the tip leakage vortex of “Front2” and “Neutral” impellers was generated from the leading edge, extended in the circumferential direction, and collapsed at the leading edge of the next blade. On the other hand, the vortex of the other impellers was generated from the tip clearance at the middle of blade and extended to the blade to blade path. It caused the vortex loss region to spread to the path. Thus, the efficiency of an unshrouded impeller depends on the generation point and extension direction of the tip leakage vortex.
jet. Then, the stronger tip leakage flow was formed with large loss. In the higher efficient impeller such as “Front2”, there was larger pressure step between pressure side and suction side at the leading edge of impeller. The step generated stronger tip leakage flow at the leading edge, and moved up the meeting line of the leakage back flow and the main stream flow upstream. Thus, the leakage flow was not pushed back to the suction side of the blade which has been passed at tip. Since the leakage flow that had collided with the mainstream flew downstream without being caught in the tip leakage jet, only weak tip leakage vortex was formed. Therefore, a pressure step at the tip of leading edge improved the efficiency of an impeller.

The total pressure head coefficient is valid when evaluating the performance of an impeller, but it contains the effect of static pressure losses. Then, the angular momentum head coefficient of the hub-side changed impellers should be focused. “Front”, “Front2” and “Mid2” had the splitters which were less loaded as mentioned at 5.3. “Aft” and “Aft2” had a large difference in static pressure at trailing edge because of the enhanced blade loading near the trailing edge. The large difference made the flow slip larger than the other impellers. Hence, the impeller with the highest angular momentum head coefficient have “Mid” distribution at the hub side.

**Figure 11.** How the tip vortex is formed in the higher efficient impeller (Front2).

**Figure 12.** How the tip vortex is formed in the lower efficient impeller (Aft).

### 4. Design Optimized Impeller

To realize an unshrouded impeller with optimized head and efficiency, the impeller with “Mid” distribution at the hub side and “Front2” distribution at the shroud side was researched by CFD with the condition shown in the section 2.3. Figure 13 shows the overview of the impeller.

Table 2 shows the performance of this optimized impeller. The velocity map on 50[\%] span faces showed in Figure 14 appears the blockage region caused by the separation at suction side of the main blade did not close the pressure side of the splitter. Figure 15 shows the loss generated by the tip leakage vortex was smaller due to the stronger tip leakage flow at the leading edge. It is confirmed that the design method discussed above is effective to realize the unshrouded impeller with higher head and higher efficiency.

**Figure 13.** The overview of the design optimized impeller.
Table 2. The performance of the optimized impeller.

| Distribution                      | Efficiency | Head coefficient |
|-----------------------------------|------------|------------------|
|                                   |            | Total pressure   | Angular momentum |
| Highest efficiency                |            |                  |                  |
| Shroud Front2                     | 0.9344     | 0.7363           | 0.7718           |
| Highest angular momentum head     | Hub Mid    | 0.9196           | 0.7446           | 0.7927           |
| Highest total pressure head       | Hub Mid2   | 0.9291           | 0.7461           | 0.7846           |
| Optimized                         |            |                  |                  |
| Hub Mid                           | 0.9373     | 0.7574           | 0.7906           |
| Shroud Front2                     |            |                  |                  |

Figure 14. The velocity map on 50[%] span faces of the optimized impellers.

Figure 15. How the tip vortex is formed in the optimized efficient impeller (Optimized).

5. Conclusions
The performance of impellers with various blade shapes was evaluated by a computational fluid dynamics (CFD) approach. As a result, it was clarified that the separation vortex on suction side of blades, which was the main cause of loss in shrouded impellers, was suppressed by mixing the tip leakage flow. The loss in unshrouded impeller was mainly caused by the static pressure reduction at tip leakage vortex core. Moreover, in order to improve the head, the increase of angular momentum should be compared for each impeller. From the above, a design method which realizes the unshrouded impeller with higher head and efficiency as follows were suggested.

1. The longer splitters load larger torque, and increase the head of unshrouded impellers.
2. When the blade angle distribution on shroud side enhances blade load on the front of main blades, the impeller has high efficiency, because stronger tip leakage flow is generated at leading edge, and weakening the tip leakage vortex.
3. The blade angle distribution on hub side which enhances blade load on the middle of main blades increases the head of the impeller, by suppressing the slip at the treading edge.

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Nomenclature

\( D \) = Diameter [m] \hspace{1cm} \phi = \text{Flow coefficient} \\
\( b \) = Span wise height [m] \hspace{1cm} \psi = \text{Head coefficient} \\
\( b_c \) = Blade Tip clearance [m] \hspace{1cm} \psi = \text{Head coefficient} \\
\( M \) = Meridional length [%] \hspace{1cm} \eta = \text{Hydraulic efficiency} \\
\( \rho \) = Density [kg/m\(^3\)] \hspace{1cm} \mu = \text{Viscosity [Pa} \cdot \text{s]} \\
\( \mu \) = Viscosity [Pa \cdot s] \hspace{1cm} L = \text{Angular momentum [N} \cdot \text{m} \cdot \text{s]} \\
\( g \) = Acceleration of gravity [m/s\(^2\)] \hspace{1cm} H = \text{Head [m]} \\
\( Q \) = Volume flow rate [m\(^3\)/min] \hspace{1cm} T_H = \text{Torque coefficient} \\
\( r \) = Radius [m] \hspace{1cm} \beta = \text{Blade angle [deg]} \\
\( \omega \) = Angular Speed [rad/s] \hspace{1cm} I_R = \text{Rothalpy [J]} \\
\( T \) = Torque [N} \cdot \text{m]} \hspace{1cm} \frac{p}{\rho} + \frac{w^2}{2} - \frac{(r\omega)^2}{2} \\
\( v \) = Velocity in stationary frame [m/s] \hspace{1cm} \zeta = \text{Loss coefficient} \\
\( w \) = Velocity in rotating frame [m/s] \hspace{1cm} \frac{I_R - I_R^1}{(r_2\omega)^2/2} \\

Subscripts

1 = Inlet of impeller \hspace{1cm} t = \text{Circumferential direction} \\
2 = Outlet of impeller \hspace{1cm} m = \text{Meridional direction} \\

References

[1] Senoo Y 1986 Pressure Losses and Flow Field Distortion Induced by Tip Clearance of Centrifugal and Axial Compressors JSME International Journal 30 pp 375-385 \\
[2] Schleer M, Song S and Abhari R 2008 Clearance Effects on the Onset of Instability in a Centrifugal Compressor. J. Turbomach 130(3), 031002(Apr 01, 2008) \\
[3] Jaatinen-Värrri et al. Centrifugal compressor tip clearance and impeller flow. Journal of Mechanical Science and Technology 30 pp 5029-5040 \\
[4] Bonaiuti D et al. Analysis and Optimization of Transonic Centrifugal Compressor Impellers Using the Design of Experiments Technique. J. Turbomach 128(4), pp 786-797 (Sep 27, 2006) \\
[5] Turunen-Saaresti T and Jaatinen A. Influence of the Different Design Parameters to the Centrifugal Compressor Tip Clearance Loss. J. Turbomach 135(1), 011017 (Oct 30, 2012) \\
[6] Kaneko M and Tsuhita H 2016 Influences of tip leakage flows on flow field at design condition in transonic centrifugal compressor with splitter blade Transactions of the JSME (in Japanese) 82 No 844 \\
[7] Iino M, Tanaka K, Miyagawa K and Okubo T 2003 Numerical analysis of 3d internal flow with unstable phenomena in a centrifugal pump. Proc. the 7th Asian International Conference on Fluid Machinery (Fukuoka, Japan, 2003 October) pp 7-10. \\
[8] Iino M, Tanaka K, Miyagawa K and Okubo T 2003 Numerical simulation of hysteresis on head/discharge characteristics of a centrifugal pump. Proc. the ASME FEDSM 4th ASME_JSME Joint Fluids Engineering Conference (Honolulu HI USA) pp 6-10 \\
[9] Ichinose A, Kinura N, Yoshimura M, Hayashi T, Miyagawa K, Evaluation of The Internal Flow Structure in the Pumps with Diffuser. (in Japanese) Proc. The Lectures in general meeting of Turbomachinery Society of Japan (Toyama, Japan, 2017 September) 7834