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Design of partial emission type liquid nitrogen pump

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Abstract

High Temperature Superconductor power cable systems are being developed actively to solve the problem of increasing power demand. With increases in the unit length of the High Temperature Superconductor power cable, it is necessary to develop highly efficient and reliable cryogenic pumps to transport the coolant over long distances. Generally, to obtain a high degree of efficiency, the cryogenic pump requires a high pressure rise with a low flow rate, and a partial emission type pump is appropriate considering its low specific speed, which is different from the conventional centrifugal type, full emission type. This paper describes the design of a partial emission pump to circulate subcooled liquid nitrogen. It consists of an impeller, a circular case and a diffuser. The conventional pump and the partial emission pump have different features in the impeller and the discharge flow passage. The partial emission pump uses an impeller with straight radial blades. The emission of working fluid does not occur continuously from all of the impeller channels, and the diffuser allows the flow only from a part of the impeller channels. As the area of the diffuser increases gradually, it converts the dynamic pressure into static pressure while minimizing the loss of total pressure. We used the known numerical method for the optimum design process and made a CFD analysis to verify the theoretical performance.

Keywords : Cryogenic Pump, Partial Emission Pump, Cryogenic Turbomachine

1. INTRODUCTION

There have been many efforts to commercialize High Temperature Superconductor (HTS) power cables to cope with the increasing electric power demand in urban areas and for interconnection between substations by transferring much larger electric power than that of conventional copper cables. To maintain the operation temperature of the HTS power cables, the circulation of subcooled liquid nitrogen is used and the refrigeration system is composed of cryogenic refrigerators, heat exchangers, cryogenic circulation pumps, and so on.

To commercialize the HTS power cable, the reliability of the refrigeration system must be viable [1, 2], and the cryogenic circulation pump is the one of the key components for transferring the coolant over long distances. At this time, cryogenic pumps are imported from foreign companies, and the need for domestic development is increasing due to the import expense and maintenance costs [3].

Among the various types of liquid pumps, a partial emission pump is known to be appropriate for low specific speed pumps, which generate a high pressure head with a low flow rate. As the unit length of the HTS power cable increases, the required pressure head also increases and the specific speed of the pumps decreases. The main difference between a conventional full emission pump, that is, a centrifugal pump, and a partial emission pump is that the partial emission pump discharges the fluid partially through only a partial flow passage, whereas the full emission pump discharges the fluid continuously.

In this paper, we describe our design of a partial emission pump that circulates the subcooled liquid nitrogen considering machinability and efficiency, mainly focusing on the design of the impeller and housing.

2. DESIGN OF THE PARTIAL EMISSION PUMP

2.1. Selection of pump type

To get the highest efficiency of the pump, the type of pump should be selected according to the specific speed (n_s), which is a non-dimensional number from the adiabatic pressure head (H_{ad} [m]), the volumetric flow rate (V [m^3/s]), and the rotation speed (ω [rad/s]) as in (1).

$$n_s = \frac{\omega\sqrt{V}}{\left(gH_{ad}\right)^{3/4}} \quad \text{[4]} \quad \cdots\cdots\cdots\cdots (1)$$

The theoretical pump efficiency is determined by the specific speed and the type of the pump as shown in Fig. 1. At a high specific speed, the full emission pump has a high degree of efficiency, while the positive displacement pump has a high degree of efficiency at a low specific speed. The partial emission pump has the highest degree of efficiency when the speed is between that of the full emission pump and the positive displacement pump. Considering the specific speed of the liquid nitrogen circulation pump for

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HTS power cable applications, the partial emission pump has a higher degree of efficiency than other types of pumps.

The partial emission pump consists of an impeller, a circular housing and a diffuser. Fig. 2 shows a schematic of the structure of the partial emission pump. It has several features that are different from the conventional full emission pump: 1) partially discharged fluid while the full emission pump discharges continuously the coolant that obtains energy from the impeller; 2) straight radial blades [4]; 3) a diffuser at the outlet part to transform dynamic pressure into static pressure, which minimizes the loss of energy.

In the partial emission pump, the coolant rotates in the circular housing and a perfect forced vortex is generated; the rotational movement is relatively stronger than other velocity components and the other components of absolute liquid velocity are negligible. In other words, a small amount of the coolant is discharged compared with the volume of the rotating coolant [7].

The straight radial blades of the impeller are more appropriate than conventional curved blades. Fig. 3 shows the characteristics for varying the blade outlet angle. The x-axis represents the flow rate and the y-axis represents the head and power according to the exit angle. When considering the head and power at a low flow rate, it is clear that a straight radial blade is the most efficient. Therefore, we chose straight radial blades.

2.2. Design specifications

To commercialize the HTS power cable, the unit length must be increased, and the pump must transport the coolant over long distances. Table 1 shows the design specifications of the pump we used to transport the subcooled liquid nitrogen over 2 km of cable.

As in all rotating machines, the efficiency and reliability are decreased by friction, abrasion and fatigue stress. Thus, this system requires periodic maintenance. However, reoperation for maintenance causes significant economic losses, so the system should be designed for reliability and efficiency.

TABLE 1
DESIGN SPECIFICATIONS.

Fluid	Subcooled liquid nitrogen
Target efficiency	65 %
Target differential pressure	> 6 bar
Target mass flow rate	> 0.4 kg/s
Rotating speed	9000 rpm

2.3 Shape design

The pump was designed according to the specific speed – specific diameter diagram as shown in Fig. 4. The theoretical efficiency of a partial emission pump is about 30-65 % and has a maximum efficiency at a specific speed of around 0.15. The specific speed of the partial emission pump is calculated in equation (2), which is expressed as parameters of the partial emission. The two specific speed values, designed by different parameters, are almost the same. Therefore, it is irrelevant to design the system according to equation (1).

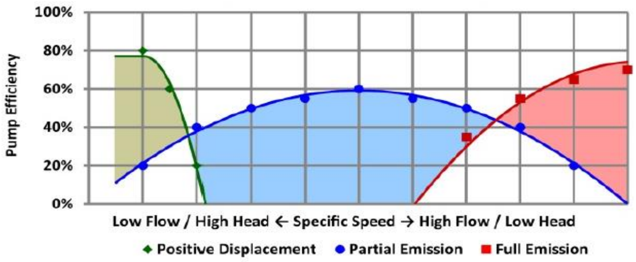

Fig. 1. Pump efficiency – specific speed diagram [5].

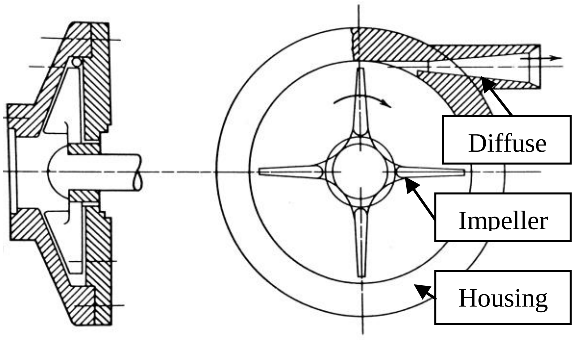

Fig. 2. Schematic of the structure of a partial emission pump [4].

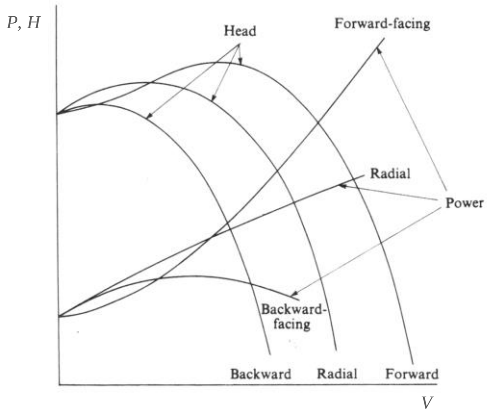

Fig. 3. Characteristics for varying the blade outlet angle [6].

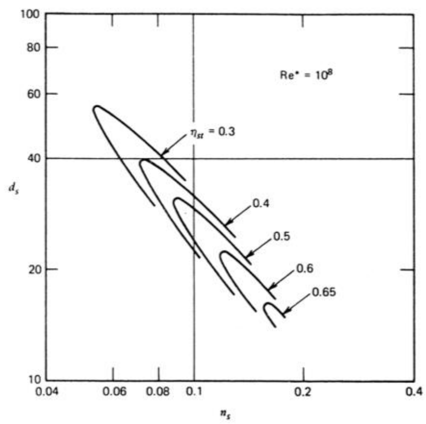

Fig. 4. Specific diameter – specific speed diagram [4].

$$n_s = \frac{\sqrt{\pi\phi^*}\, d_{thr} / D}{q_{ad}^{3/4}}^{[4]} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

$$q_{ad} = \frac{H_{ad}}{u_2^{\,2} / g}^{[4]} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

$$d_{thr} / D = 0.04 \sim 0.055^{[4]} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

where flow factor (ϕ^*) is defined as the ratio of the tip speed (u_2 [m/s]) to the diffuser speed (c_{thr} [m/s]). For (ϕ^*) > 1, the unit chokes in the diffuser [4]. This value is determined as 1 which has a high degree of efficiency, and head coefficient (q_{ad}) is determined as 0.54 by the flow factor in Fig. 5. The ratio of the diffuser throat diameter (d_{thr} [m]) to the impeller diameter (D [m]) is 0.04-0.055 according to the experiment, and it is assumed that the value is 0.055. As the specific speed of the pump that includes these parameters in equation (2) is about 0.155, the partial emission pump efficiency is expected to be about 65 %.

TABLE 2
DESIGN RESULTS.

Specific speed	0.155
Rotating speed	9000 rpm
Mass flow rate	0.4 kg/s
Differential pressure	6 bar
Impeller diameter	77 mm
Blade width of impeller	4.3 mm
Diffuser throat diameter	4.25 mm
Diffuser length	60 mm
Clearance	0.55 mm
Expected efficiency	65 %

The determined diameter of the impeller was 77 mm with the parameters in equation (3). This equation is composed of the previously determined adiabatic head, head coefficient, tip speed and acceleration of gravity. With the previously determined values and rotating speed of the impeller as 9000 rpm, the diameter of the impeller was determined based on the tip speed. The width of the impeller tip was determined according to ratio of the diffuser throat diameter, and the impeller diameter was set as 0.055. This value is slightly larger than the diffuser throat.

The circular housing connects the impeller and the diffuser, thereby forming a flow field of coolant. In a partial emission pump, the gap between the housing and the tip of the impeller is highly important. This is related to the recirculation loss. As the impeller rotates, the differential pressure is generated at both sides of the blade. Due to the differential pressure, the coolant is leaked through the gap, which is the recirculation loss, and it affects performance of the pump. The gap distance is determined by (5).

$$s / b_2 = 0.13^{[4]} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5)$$

where s [m] and b_2 [m] are the clearance and the width of

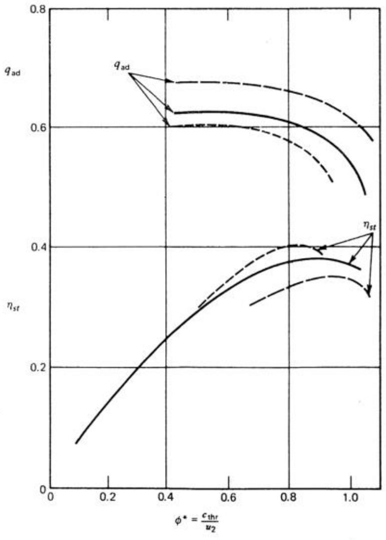

Fig. 5. Head coefficient/efficiency-flow factor diagram [4].

the impeller tip. The diffuser reduces the flow speed and recovers the static pressure head through the increasing cross-sectional area. In this pump, a conical diffuser is used and it can cause energy loss by flow separation [8]. To reduce this loss, a gradual increase in the cross-sectional area is necessary. As the final discharge outlet in the diffuser is determined by the pipes of the system, in consideration of the machinability, the length of the diffuser is designed as 60 mm to minimize the loss caused by the diffuser.

3.5 Flow simulation

In order to verify the performance of the pump for HTS power cable applications, we conducted a flow simulation with an ANSYS 15.0 CFX module using the finite element method. Fig. 6 shows the mesh lattice of the LN2 pump. The lattice was formed of a tetrahedron, a wedge and a pyramid to optimally adjust the shape of the flow field. The pump has three parts: a rotating part and two stationary parts. One stationary part is the diffuser in the flow field, and another is the flow passage to connect the system to the rotor. The rotor is the rotating part that includes the impeller. The number of nodes is 1,852,777 and the number of elements is 6,626,899. Table 3 gives the boundary conditions for the flow simulation.

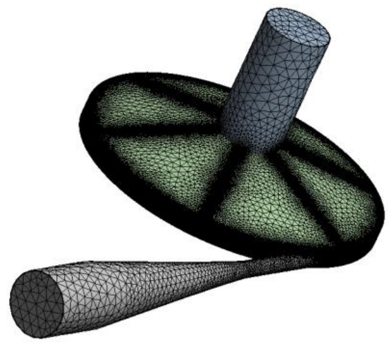

Fig. 6. Mesh of flow field.

TABLE 3
BOUNDARY CONDITIONS OF FLOW SIMULATION.

Coolant	Liquid nitrogen
Rotating speed	9000 rpm
Inlet temperature	70 K
Inlet Total pressure	2 bar
Outlet mass flow rate	0.4 kg/s
Rotor	Impeller
Stator	Housing and Diffuser
Interface connection	Frozen rotor
Shroud wall condition	Counter rotating wall

3. ANALYSIS RESULT

The result of the analysis is shown in Table 4. The designed pump had a high-performance that caused a differential pressure of about 6 bar. The simulation efficiency of this pump was about 60 %. This efficiency was made by considering the assembly and machinability.

The streamline in the fluid field is shown in Fig. 7. Along the rotating impeller, the streamline goes on smoothly from the entrance of pipe to the exit and is discharged through the flow path of the diffuser. Less recirculation from clearance occurs.

Fig. 8 is the distribution of the static pressure plotted in the flow field. After going through the rotor at 2 bar of the inlet pressure, the pressure of the final exit was restored to 8 bar gradually by the diffuser as expected. Thus, we confirmed that the pump can generate the designed differential pressure of 6 bar.

Fig. 9 indicates the distribution of the total pressure in the flow field. It shows that the total pressure gradually increased as the fluid coming from the inlet received energy from the impeller, and the total pressure loss of the fluid discharged through the diffuser was small.

TABLE 4
RESULTS OF FLOW SIMULATION.

Differential pressure	6.26 bar
Input power	528 W
Output power	317 W
Efficiency	60 %

4. CONCLUSION

We designed a pump with partial emission for HTS power cable application. It had sufficient differential pressure to circulate LN2 over a long distance and an efficiency of about 60 %, which is the result of considering machinability and assembly. Based on these results, we will follow with fabrication to develop a LN2 Pump for HTS power cable applications.

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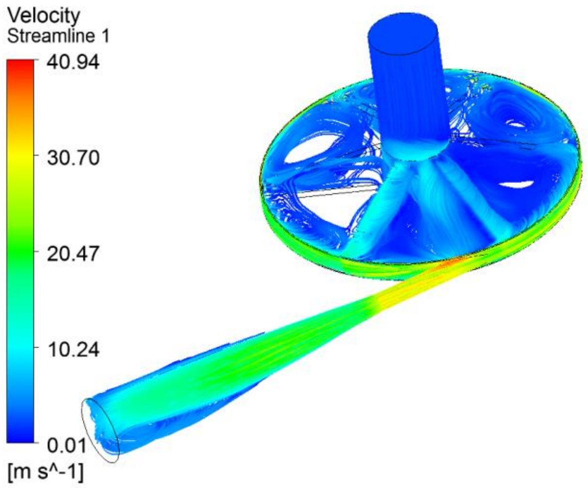

Fig. 7. Streamline of flow field.

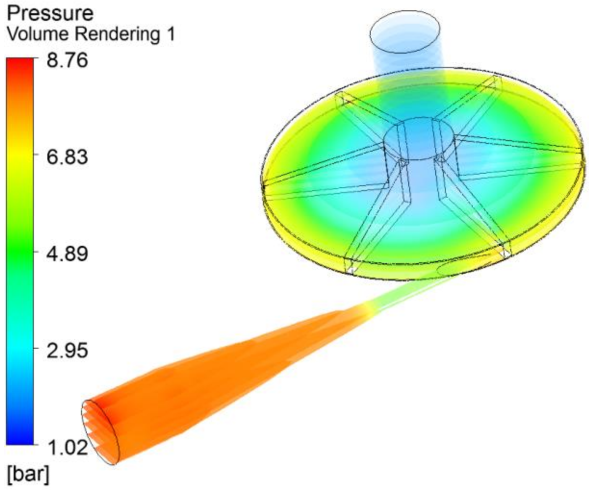

Fig. 8. Static pressure of flow field.

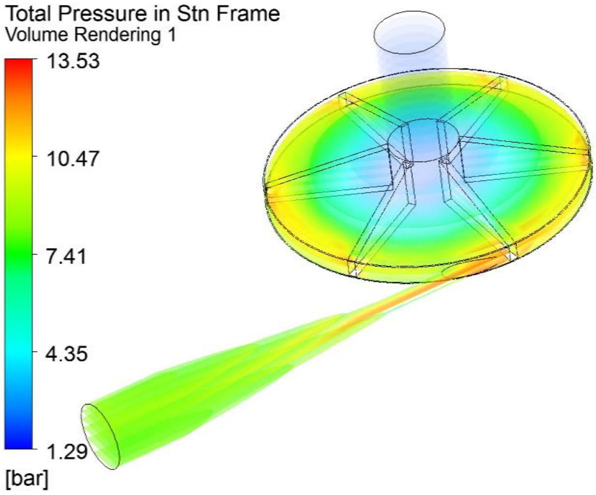

Fig. 9. Total pressure of flow field.

REFERENCES

[1] Jae-Ho Kim, Minwon Park, In-Keun Yu, “Development of Real Time Protective Coordination Algorithm for HTS Power Cable,” *IEEE Trans. Appl. Supercond.*, vol. 25, no. 3, pp. 1-4, 2015.

[2] Masuda, T., Yumura, H., Watanabe, M., “Recent progress of HTS cable project,” *Physica. C*, vol. 468, no. 15/20, pp. 2014-2017, 2008.

[3] Chang-Hyeong Lee, Dong-Min Kim, Hyung-Suk Yang, Seok-Ho Kim, “Conceptual design of cryogenic turbo expander for 10 kW class reverse Brayton refrigerator,” Progress in Superconductivity and Cryogenics, vol. 17, no. 3, pp. 41-46, 2015.

[4] O. E. Balje, *Turbomachines*, JOHN WILEY & SONS, pp. 3-436, 1981.

[5] Design & Manufacture of Specialty Turbomachinery, Barber Nichols

[6] A. T. Sayers, Hydraulic and Compressible Flow Turbomachines, McGraw-Hill, pp. 1-70, 1992.

[7] U. M. Barske, “Development of some unconventional centrifugal pumps.” *Proceedings of the Institution of Mechanical Engineers*, vol. 174, no. 1, pp. 437-461, 1960.

[8] Frank M. White, Fluid Mechanics 7th Edition in SI units, McGraw-Hill, pp. 423-483, 2012.