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Analytical Study of Volumetric Scroll Pump for the Cold Moderator System

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Abstract

The volumetric scroll pump has been designed and analyzed numerically to verify the feasibility of the application of volumetric scroll pump with the cold moderator system. The hydraulic analysis of the scroll pump, which was focused in the suction process, was carried out with the Low-Reynolds number k-ε turbulence model. The analytical results show that the scroll pump has small negative relative pressure, and preferable volumetric intake profile. These characteristics might verify the feasibility of the scroll pump for applying to circulate the liquid hydrogen in the cold moderator system. The following advantages would be expected in the application; (1) The pump could be applied with gas, liquid, and mixture (two-phase flow); (2) The minimum pulsation, which provides less vibration, would be expected, since the processes inside the scroll pump occur simultaneously and continuously in a cycle of crank revolution, and it operates at very low speed comparing with the high-speed centrifugal pump; (3) The liquid hydrogen flow rate could be estimated as the ratio of shaft speed.

Introduction

In the spallation target research and development at the Japan Atomic Energy Research Institute (JAERI), an electromagnetic pump (EMP) has been selected to circulate mercury which is a target material working as a spallation neutron source, and a high-speed centrifugal pump would circulate liquid hydrogen in a cold moderator system for classifying cold and thermal neutron [1,2]. At present, in the mercury target system, the electromagnetic pump is planned to replace with a mechanical gear pump, so that mercury flow rate could be estimated due to the volumetric pump feature. Furthermore, for the safety of target system, the multiple functions would be expected when the volumetric pump is applied together with a flow meter.

It would be possible to use a scroll pump as a volumetric liquid pump like a mechanical gear pump, which would give lower flow-induced vibration than the gear pump. The scroll pump has been developed from scroll compressor, which gives high performance and works very well in the application of small capacity refrigerating system and air compressor [3,4]. The scroll compressor has three processes inside i.e., gas suction, compression, and discharge [5]. Unlike the reciprocating (piston type) compressor, the processes inside the scroll compressor occur simultaneously and continuously in a cycle of crank revolution, resulting in a minimum pulsation, which provides less vibration.

In the case that the scroll compressor would be used as the liquid scroll pump, the compression process would be omitted since the fluid to be conveyed is incompressible fluid. Therefore, the scroll wraps of the pump are shorter than that of the compressor. **Figure 1** illustrates the processes inside the scroll pump. As seen in the figure, fluid flows through inlet port into suction chamber in suction process while the fluids in the discharge chamber, which was filled in discharge chamber after the suction process completed in previous cycle, flows through outlet port in discharge process. These processes occur simultaneously and complete the processes within a cycle of crank revolution

In this study, a prototype scroll pump has been modeled and analyzed numerically to predict flow patterns and pressure distributions in the suction process which is one of key processes, with a view for verifying the feasibility of the scroll pump for applying with the liquid circulation system such as the cold moderator system.

Analytical Model

The designed scroll pump configuration is as shown in **Fig. 2**. The coordinate equations of the perimeter curves of the scrolls are as follows:

Fixed scroll:

Inner curve $X = \alpha\cdot(\cos\lambda+(\lambda+\pi-\beta)\cdot\sin\lambda)$

$Y = \alpha\cdot(\sin\lambda-(\lambda+\pi-\beta)\cdot\cos\lambda)$

$0 \le \lambda \le 3\pi$

Outer curve $X = \alpha\cdot(\cos\lambda+(\lambda-\pi)\cdot\sin\lambda)$

$Y = \alpha\cdot(\sin\lambda-(\lambda-\pi)\cdot\cos\lambda)$

$2\pi \le \lambda \le 5\pi$

Orbiting scroll:

Inner curve $X = \alpha\cdot(\cos\lambda+(\lambda-\beta)\cdot\sin\lambda) + \gamma\cdot\cos\theta$

$Y = \alpha\cdot(\sin\lambda-(\lambda-\beta)\cdot\cos\lambda) + \gamma\cdot\sin\theta$

$\pi \le \lambda \le 4\pi$

Outer curve $X = \alpha\cdot(\cos\lambda+\lambda\cdot\sin\lambda) + \gamma\cdot\cos\theta$

$Y = \alpha\cdot(\sin\lambda-\lambda\cdot\cos\lambda) + \gamma\cdot\sin\theta$

$\pi \le \lambda \le 4\pi$

Where α: Scroll coefficient (mm)

β: Discrepancy of starting rolls angle (radian)

γ: Radius of basic circle, $\alpha\cdot(\pi-\beta)$ (radian)

θ: Crank angle (radian)

λ: Position angle (radian)

Adjustment of scroll coefficient, α, would change the scroll pump size in scale while adjustment of discrepancy of starting rolls angle, β, would change the shape of the scroll wraps (i.e., changing the thickness of the scroll wraps). The comparison of the scroll models in different values of β is shown in **Fig. 3**.

The relationship between intake volume and scroll coefficient, α, is shown in **Fig. 4**, while the relationship between intake volume and discrepancy of starting rolls angle, β, is shown in **Fig. 5**. As seen in **Fig. 4** and **Fig. 5**, it could be confirm that scroll pump can be designed and configure before applying to the circulation system to meet the flow rate condition of the system.

In the study, the analytical model was considered to have an appropriate pump size of 300 mm in diameter and 50 mm in depth. The scroll coefficient, α, equals to 10 mm, the discrepancy of starting rolls angle, β, equals to 0.5π radian, the suction port has its diameter of 30 mm and its center at X=90 mm and Y=-90 mm, while the discharge port has its diameter of 25 mm and its center at the origin. The inlet and outlet pipes have the same diameter size as the connecting ports with each of 50 mm in length.The thermal-hydraulic analytical code, STAR-CD, was used for predicting the flow properties inside the pump. In this study, the three-dimensional fluid cells in moving region have been generated by using the surface extrusion technique. The vertices and splines input data of the scroll curves on the x-y plane were calculated from the equations described above using zero degree crank position, $\theta = 0$. Extruding the curves in z direction generated the first primitive shells of the scroll surfaces then extruding the orbiting shell in the moving direction generated three-dimensional cell layers. During an extrusion, new sets of vertices were generated based on shell vertices in the current shell set and on a given vertex number offset i.e., the new vertex number equals to the vertex number of the current shell set plus the given vertex number offset. This extrusion process was done repeatedly to complete one cycle of revolution (360 degree of crank angle) so that the model consists of entire overlapped cells. The present analyses have been performed for 24 steps of 15 degree. The overlapped cells must be managed to activate or deactivate along with their position changed at each step of calculation. The resulting cells in the first and the last steps in two- and three-dimension are shown in **Fig. 6**. Note here that the present calculation was performed only in the suction process.

The analyses were carried out under water and liquid hydrogen flow conditions as so that the fluid cells were applied for both of them. The flows are assumed to be turbulent and driven entirely by the motion of the orbiting scroll wrap. The Low-Reynolds number k-ε turbulence model was applied to determine the Reynolds stress and turbulent scalar fluxes. The port boundary pressures are held constant at atmospheric pressure boundary condition throughout the computation. The velocity magnitude of the orbiting scroll equals to 1.645 m/s according to the shaft speed of 1000 rpm. The illustrations of the boundary conditions are also shown in **Fig. 6**.

Results and Discussions

The intake volume and the corresponding angle are plotted in **Fig. 7**. The intake volume profile is almost linear and the fluid is induced into the suction chamber continuously so that the scroll pump suction process should be smooth as mentioned.

The analytical results of velocity and relative pressure distributions in the suction process are plotted on the center surface, $z = 25$ mm, in **Fig. 8** and **Fig. 9** respectively. The results are illustrated at 0, 90, 180, and 270 degree corresponding to **Fig. 1**. The relative pressure indicates the pressure difference against the inlet pressure.

Figure 8 shows how the orbiting scroll drives the flow inside the pump. The fluid in the region where the orbiting scroll moves toward has a high velocity. The high velocity fluid flows against the stationary wall, when the orbiting scroll moves close to the wall, causes the recirculation flows. It can be seen also in **Fig. 9** that the higher relative pressure is in the corresponding region to the region that the recirculation flows occurs.

As seen in **Fig. 9**, the low relative pressure occurs in the region that the moving scroll moves from while the high relative pressure occurs in the region that the moving scroll moves to. The negative relative pressure normally occurs inside the suction chamber i.e., in the pocket between the orbiting and the fixed wrap. However, when the suction process is almost completed as shown in **Fig. 9 (1d)** and **(2d)**, high-pressure would occur in the pocket where the concave face of the orbiting wrap presses the fluid inside. Furthermore, inside the high-pressure pocket, it can be seen in **Fig. 8** that the recirculation flow always occurs in the corner region, and it causes the considerably high pressure in the corner. This effect increases significantly when the orbiting wrap is nearly close to the fixed one. We will study this effect in more details in the next step.

The maximum and minimum relative pressures in 24 steps are plotted in **Fig. 10**. In case of water pump, the lowest relative pressure is equal to −106 Pa at 75 degree of crank angle and the highest relative pressure is equal to 1538 Pa at 345 degree of crank angle, while in case of liquid hydrogen the lowest relative pressure is equal to −34.2 Pa at 165 degree of crank angle and the highest relative pressure is equal to 129 Pa at 315 degree of crank angle. These results show that the relative pressure is very small in case of the liquid hydrogen condition in comparison with the water condition.

In one cycle of revolution, the flow is induced into the suction chamber continuously since the volume of the suction chamber increases while the volume of the discharge chamber decreases, see also in **Fig. 1**. The intake volume of the pump was calculated in each step, as the summation of volume of fluid cells excluding the volume of the cells at the first step. In the study, the approximate intake volume per cycle is 690 cm^3. This gives the volume flow rate of 0.69 m^3/min at 1000 rpm of crank speed. We might control the volume flow rate by adjusting the crank speed, for example 0.1 m^3/min at 145 rpm.

Concluding Remarks

The hydraulic analysis of the scroll pump, which was focused in the suction process, was carried out with the Low-Reynolds number k-ε turbulence model. The analytical results show that the scroll pump has small negative relative pressure especially in case of liquid hydrogen, which is the working fluid in a cold moderator system. The scroll pump also has preferable volumetric intake profile. These characteristics might verify the feasibility of the scroll pump for applying to circulate the liquid hydrogen in the cold moderator system. The scroll pump can be designed and configured to meet the average flow rate condition of the system, and can control the fluid flow rate by adjusting the crank speed in operation. The next step to develop the scroll pump is to do analyses on the flow and pressure distributions at the position close to the end of the suction process for more details, especially, in the corner of the high-pressure pocket.

References

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(3) H. Tarada et al, REFRIGERATION Vol.58, No.672, 75-86 (1983).

(4) T. Arata and A. Murayama, REFRIGERATION Vol.62, No.720, 70-75 (1987).

(5) HVAC Systems and Equipment, ASHRAE Handbook, 34.1-34.33 (1996).

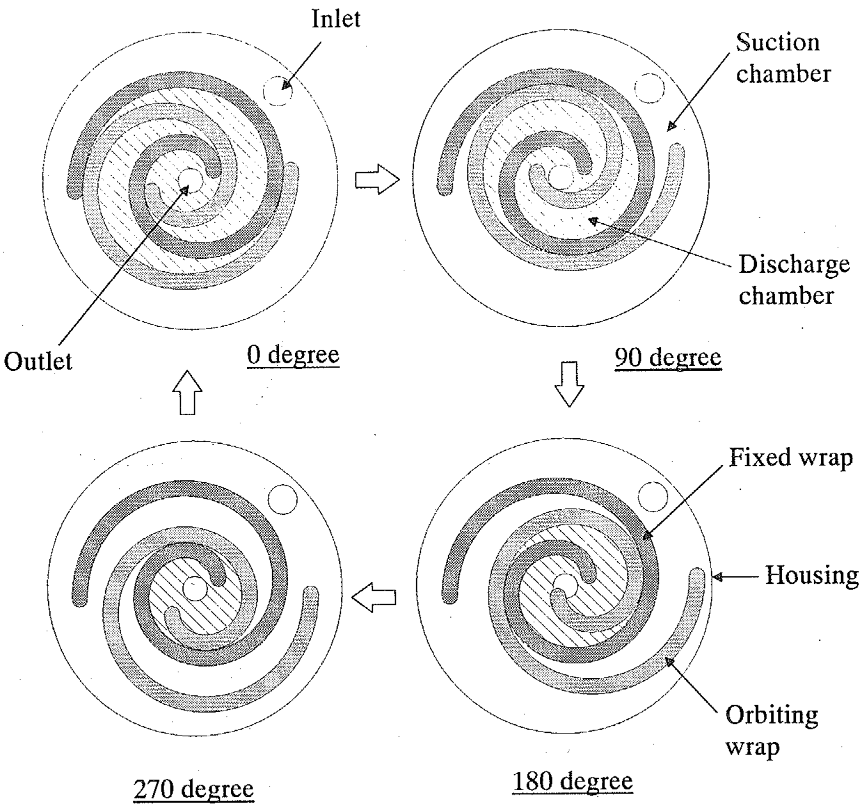
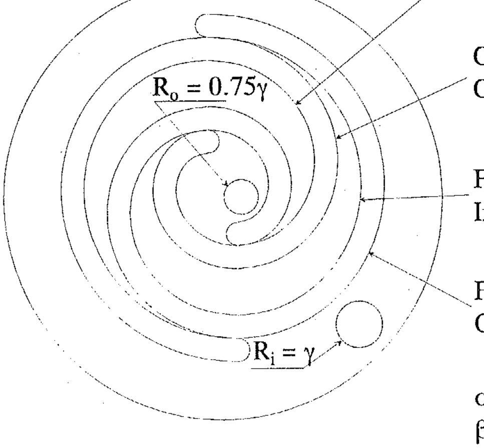


Fig. 1 Process inside a scroll pump



Orbiting scroll: $\pi \leq \lambda \leq 4\pi$

Inner curve $X = \alpha\cdot(\cos\lambda+(\lambda-\beta)\cdot\sin\lambda) + \gamma\cdot\cos\theta$

$Y = \alpha\cdot(\sin\lambda-(\lambda-\beta)\cdot\cos\lambda) + \gamma\cdot\sin\theta$

Orbiting scroll: $\pi \leq \lambda \leq 4\pi$

Outer curve $X = \alpha\cdot(\cos\lambda+\lambda\cdot\sin\lambda) + \gamma\cdot\cos\theta$

$Y = \alpha\cdot(\sin\lambda-\lambda\cdot\cos\lambda) + \gamma\cdot\sin\theta$

Fixed scroll: $0 \leq \lambda \leq 3\pi$

Inner curve $X = \alpha\cdot(\cos\lambda+(\lambda+\pi-\beta)\cdot\sin\lambda)$

$Y = \alpha\cdot(\sin\lambda-(\lambda+\pi-\beta)\cdot\cos\lambda)$

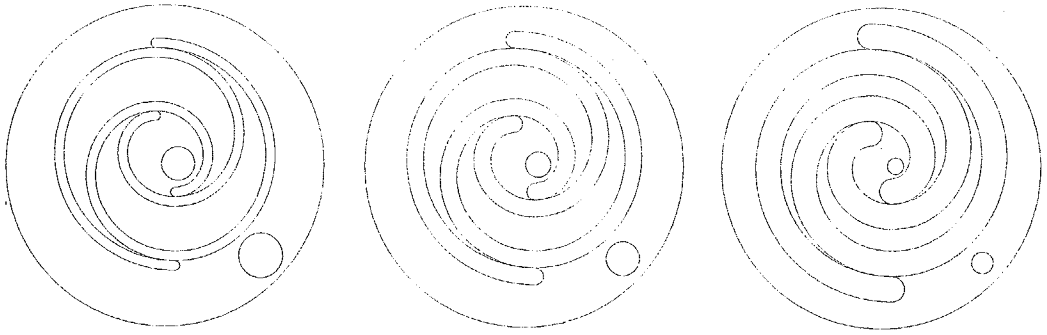
Fixed scroll: $2\pi \leq \lambda \leq 5\pi$

Outer curve $X = \alpha\cdot(\cos\lambda+(\lambda-\pi)\cdot\sin\lambda)$

$Y = \alpha\cdot(\sin\lambda-(\lambda-\pi)\cdot\cos\lambda)$

α: Scroll coefficient (mm)
β: Discrepancy of starting rolls angle (radian)
γ: Radius of basic circle, $\alpha\cdot(\pi-\beta)$ (radian)
θ: Crank angle (radian)
λ: Position angle (radian)

Fig. 2 Scroll pump configuration



β = 0.3π β = 0.5π β = 0.7π

Fig. 3 Scroll pump model in different discrepancy of starting roll angles, β

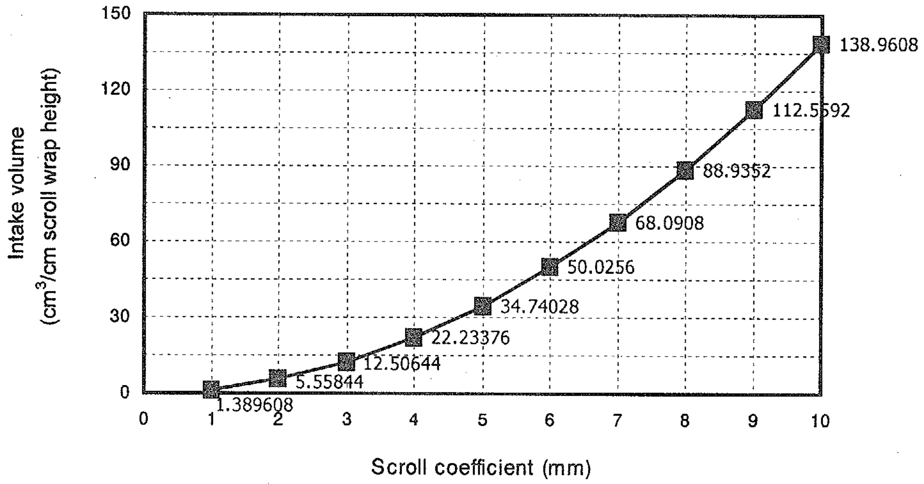


Fig. 4 Relationship between intake volume and scroll coefficient, α

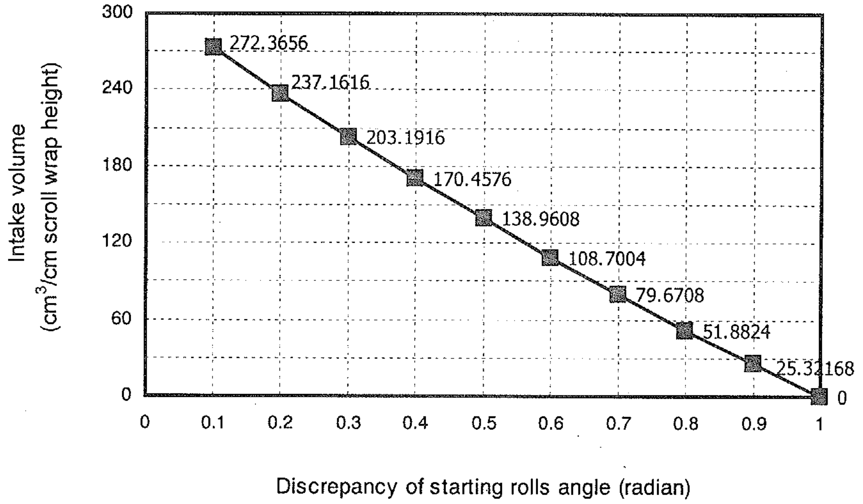
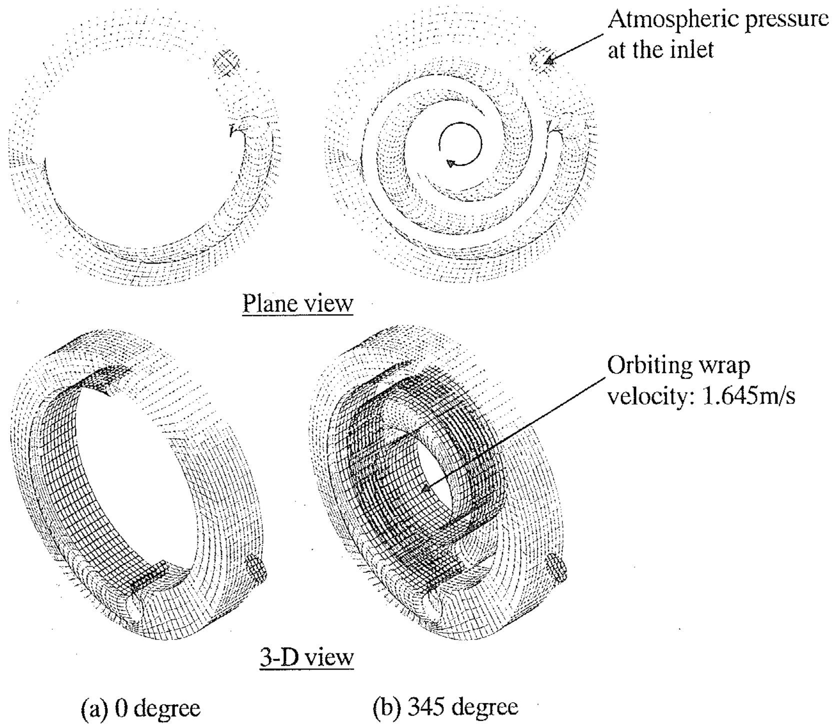


Fig. 5 Relationship between intake volume and discrepancy of starting roll angles, β



Atmospheric pressure at the inlet

Orbiting wrap velocity: 1.645m/s

(a) 0 degree

(b) 345 degree

Fig. 6 Computational grid of the scroll pump

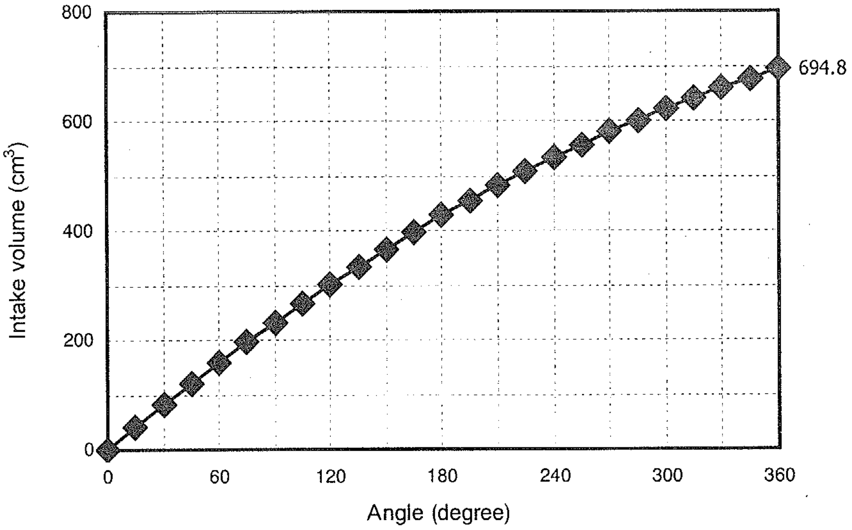
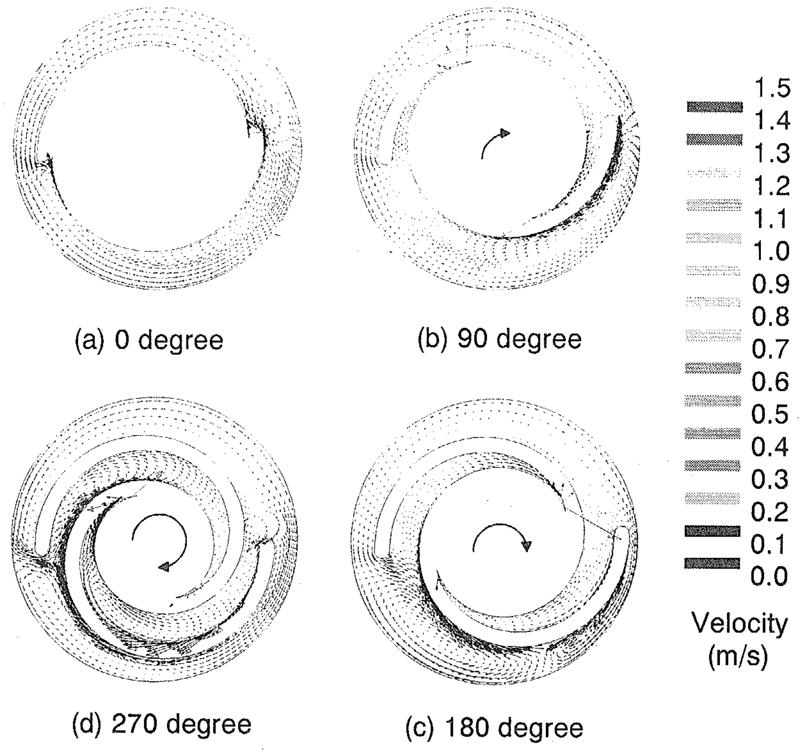


Fig. 7 Intake volume profile of suction process



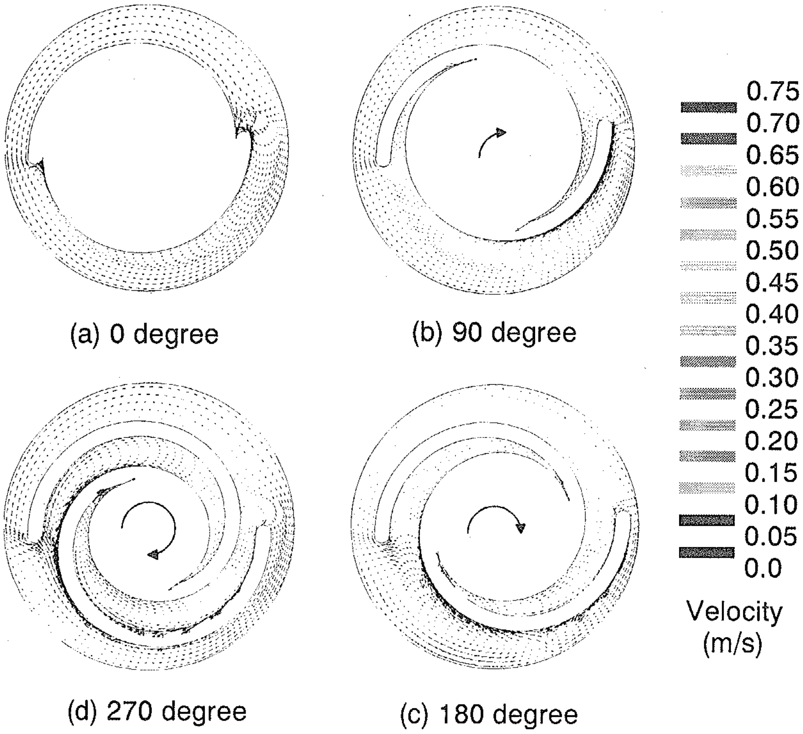
(a) 0 degree

(b) 90 degree

(d) 270 degree

(c) 180 degree

(1) Water



(a) 0 degree

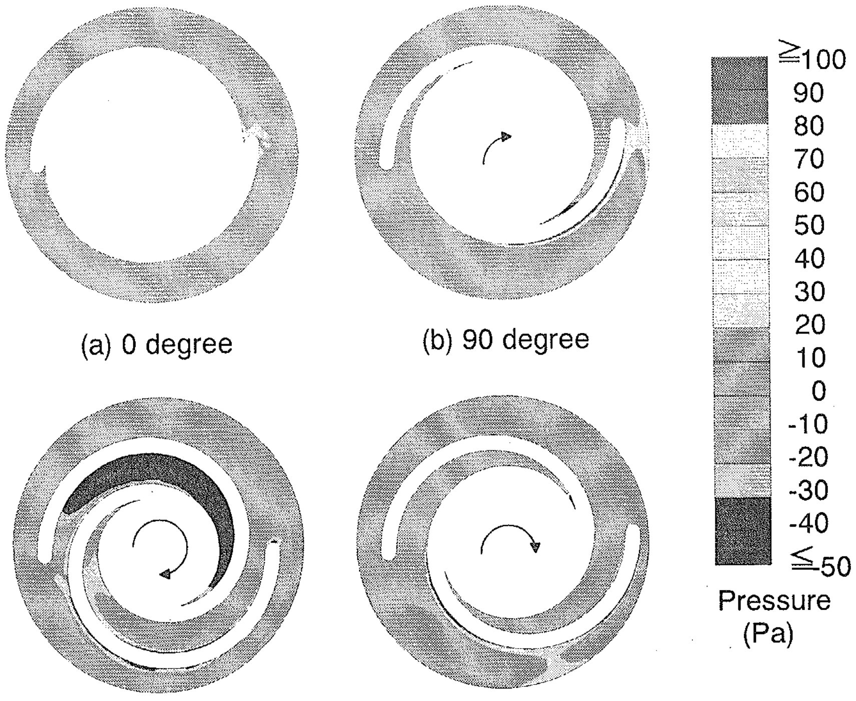
(b) 90 degree

(d) 270 degree

(c) 180 degree

(2) Liquid hydrogen

Fig. 8 Velocity distribution at several orbiting angles



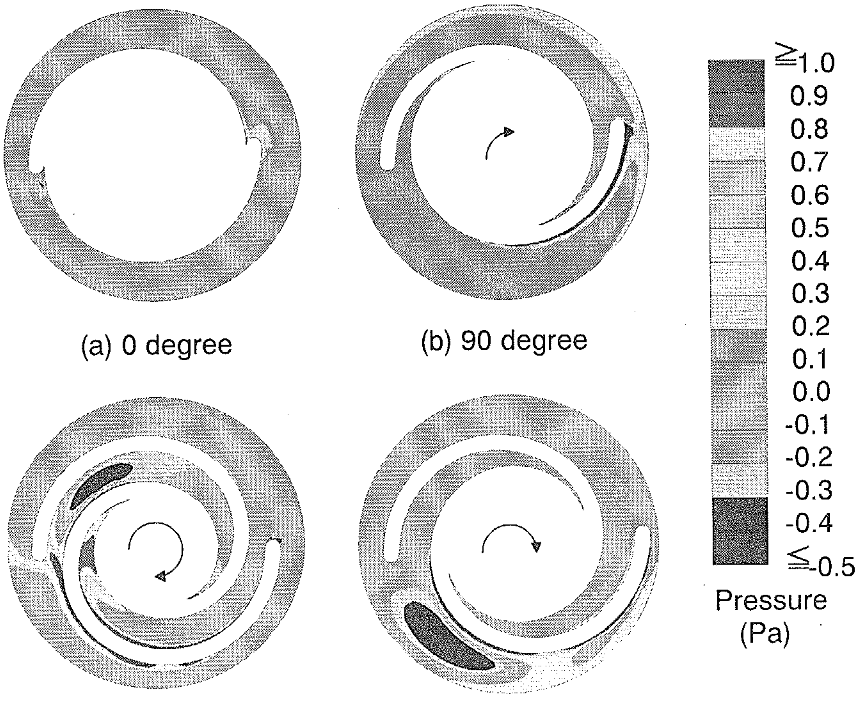
(a) 0 degree

(b) 90 degree

(d) 270 degree

(c) 180 degree

(1) Water



(a) 0 degree

(b) 90 degree

(d) 270 degree

(c) 180 degree

(2) Liquid hydrogen

Fig. 9 Pressure distribution at several orbiting angles

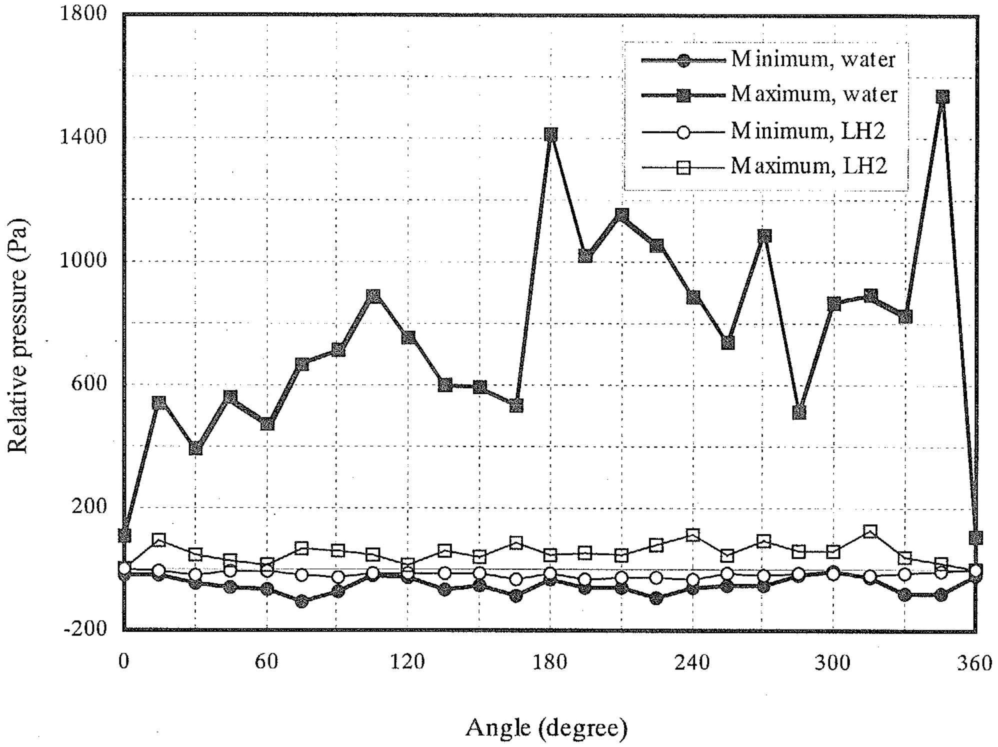


Fig. 10 Relative pressure plot at several orbiting angles