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Review Paper (Invited)

J-Groove Technique for Suppressing Various Anomalous Flow Phenomena in Turbomachines

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Abstract

In operating a turbomachine at off-design conditions various instabilities caused by anomalous flow phenomena occur and sometimes lead to the damage of a turbomachine. In order to avoid these phenomena various devices characteristic to each phenomenon have been developed, however they make turbomachines large-sized and cause efficiency drop. The present author has developed a very simple and innovative device, termed "J-groove," of suppressing various anomalous flow phenomena commonly by controlling the angular momentum of the main flow. It has been revealed that J-groove makes an operation of a turbomachine stable in all flow range, causes little efficiency drop, and can be easily applied to an existing machine. Here is reviewed totally the results of suppressing various anomalous flow phenomena in turbomachines.

Keywords: J-Groove, Anomalous Flow Phenomenon, Surge, Cavitation, Performance-curve Instability

1. Introduction

In operating a turbomachine at off-design conditions there occur various instabilities caused by anomalous flow phenomena such as surge, rotating stall, cavitation, abnormal axial thrust, impeller whirl and so on. These anomalous phenomena often cause vibration and noise and sometimes lead to the damage of turbomachines. To avoid these phenomena various devices characteristic to each phenomenon have been developed so far[1]. However they make turbomachines large-sized and cause efficiency drop.

The present author has found a very simple and innovative device to control and suppress a flow rotation considerably[2]. This device is only to install many shallow grooves on the inner casing wall of a turbomachine in the direction of pressure rise. As a turbomachine is principally an energy converter utilizing flow rotation, this device has a possibility of controlling the internal flow drastically. Thus the present device has been applied to various anomalous flow phenomena in many types of turbomachine and has succeeded in commonly suppressing various instabilities by controlling an angular momentum of the main flow.

The present device, termed "J-groove," shows variously different effects depending on the place of installation. When installed near the entrance of an impeller, J-groove has succeeded in suppressing the whirl of inlet flow, which made the performance-curve instability characterized by a rising head-capacity curve very stable. Thus J-groove can suppress surge without dropping the maximum efficiency in mixed flow and axial flow pumps[3][4]. When installed at the entrance of diffuser of a centrifugal flow type, rotating stall in a vaned and a vaneless diffusers has been perfectly suppressed in the entire flow range and the pressure fluctuation has disappeared[5][6]. Moreover it has improved cavitation performance[7] and reduced axial thrust largely[8]

2. What happens by J-Groove?

In order to elucidate the mechanism of suppressing the flow rotation by J-groove, we consider the whirling flow along a rotating disk enclosed in a cylindrical casing shown in Fig. 1.

The flow field is composed of three parts, the boundary layers on the rotating disk and the stationary casing wall and the core region between the boundary layers. In the core region, the centrifugal force balances with radial pressure gradient, and the flow rotates at the velocity about a half of the disk rotation, $U = r\omega/2$. In the disk boundary layer the centrifugal force is larger than the pressure gradient, which causes the radial outward secondary flow, while in the boundary layer of the stationary wall the radial inward secondary flow is induced.

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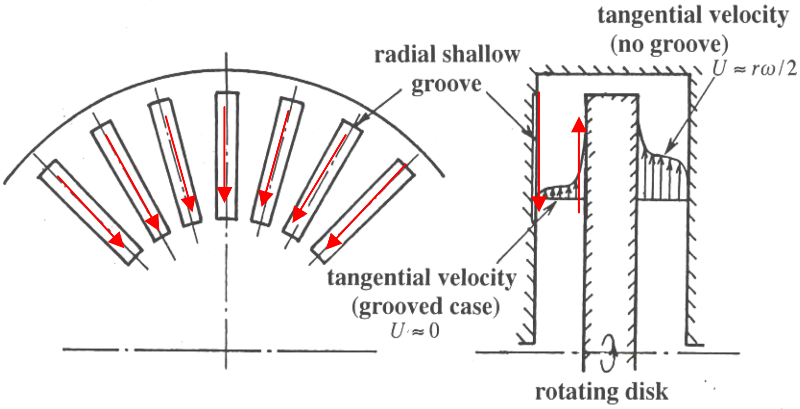


Fig. 1 Flow along a rotating disk enclosed in a cylindrical casing and radial grooves

When many radial shallow grooves are installed on the casing wall in this flow field as shown in Fig.1, the strong inward jet, shown by the red allows in the left figure, is induced in the grooves due to the pressure gradient. This inward jet in the radial grooves increases the radial outward secondary flow in the disk boundary layer to satisfy the continuity equation. The high velocity fluid at the outer radii loses large angular momentum when entering into the radial grooves, and the increased radial outward secondary flow in the disk boundary layer takes small angular momentum (low rotating velocity) at the inner radii to the outer radii. These two secondary flows decelerate the flow rotation of the core region considerably and the rotation of all flow field becomes almost zero, $U \approx 0$. Thus the pressure distribution along the disk becomes almost uniform.

It is, then, predicted that the idea of utilizing this mechanism could be one possible way of controlling and suppressing several anomalous flow phenomena caused by a whirl flow, such as rotating stall in a vaneless or a vaned diffuser, performance-curve instability characterized by positive slope of head-capacity curve, rotating cavitation, draft tube surge and so on. In all these cases, the grooves should be installed in the direction of pressure rise so that the reverse flow is induced in the grooves.

3. Suppression of Surge in Pumps

Performance-curve instability characterized by a positive slope of head-capacity curve sometimes causes a severe pressure oscillation, called surge, and is an especially serious problem in a high specific speed turbomachine such as an axial flow type and a mixed flow type. The present author has revealed that the performance-curve instability in a mixed flow pump is caused by a sudden drop of theoretical head due to a strong whirl of the reverse flow at an impeller inlet[9].

To suppress this instability an active control using jet injection in the counter rotating direction of the impeller has been found effective[10], but this requires complicated mechanisms and utilizes additional machinery that eventually decrease the overall efficiency and reliability. The present author applied J-groove to suppress surge of the mixed flow pump and the axial flow pump, of which results are summarized below.

3.1 Performance-curve Instability in Mixed Flow Pumps[3][11]

To suppress the performance-curve instability and surge of a mixed flow pump, J-groove was applied to the mixed flow pump shown in Fig. 2, of which specific speed *Ns*=830 [$m, m^3/min, min^{-1}$] or the non-dimensional specific speed 0.33. J-groove was installed at the impeller inlet region on the inner casing wall to the axial direction, as the pressure rises in the axial direction.

The change of head-capacity curve $\psi - \phi$ is illustrated in Fig. 3. The original curve shows clear performance-curve instability with a sudden drop of head at the flow-rate 61% of the BEP, however the J-groove with the optimum dimensions has succeeded in making the performance curve stable for the entire flow range without dropping the maximum efficiency.

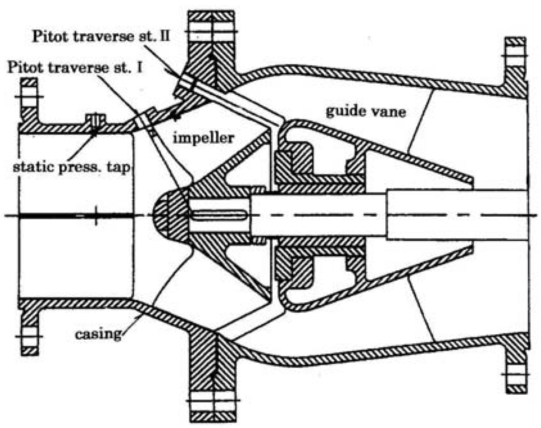


Fig. 2 Mixed flow pump tested(*Ns*=830)

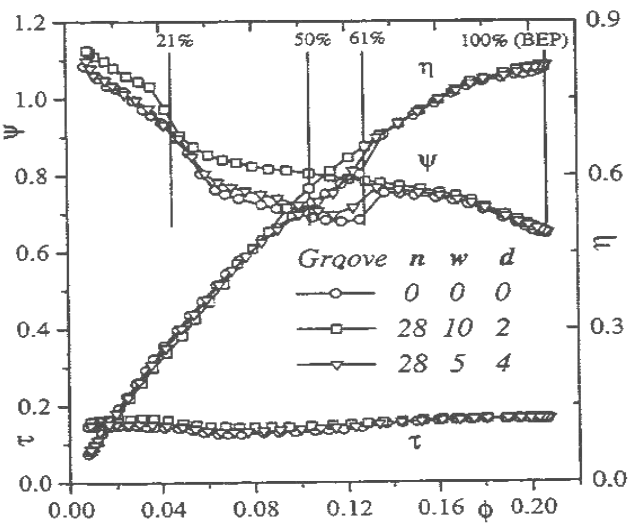


Fig. 3 Comparison of performance curves

It has also been made clear that the J-groove location has key importance, that the grooves are to be installed from the impeller leading edge into the impeller channel, and that wide and shallow grooves are recommended.

Velocity and pressure measurements have revealed the mechanism of suppressing the instability as follows; Groove flow flows against the main flow and mixes with the whirling flow near the impeller inlet tip region, which reduces the whirl strength and the region of reverse flow in the critical flow and low flow range. Because the onset of inlet whirl causes a sudden drop of the pump theoretical head and thus causes performance-curve instability, the reduction of the whirl strength and region of the reverse flow by J-groove makes the performance-curve instability disappear. The present method utilizes absorption of the angular momentum by the groove reverse flow owing to mixing with the whirling flow.

3.2 Performance-Instability in Axial Flow Pumps[4]

In an axial flow impeller the pressure-rise in an impeller channel is so small compared with that of a mixed flow impeller that it is difficult to attain enough groove flow by use of usual type J-groove. Though the perfect suppression of performance-curve instability in an axial flow pump was reported[12] by use of J-groove, the groove dimension was so large that the efficiency must have dropped largely. Thus a new type J-groove has been developed in order to attain perfect suppression of the performance curve instability without dropping the maximum efficiency.

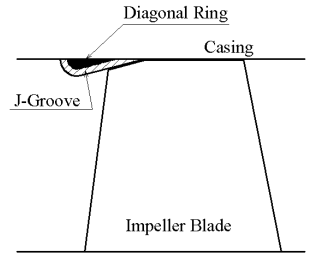


Fig. 4 New-type J-groove

The new type J-groove utilizes centrifugal force effect besides the blade work by making the groove shape diagonal as shown in Fig. 4. The impeller tip is cut diagonal near the leading edge and the diagonal ring with many grooves installed on the inner wall is inserted just before the impeller inlet. In order to examine the effect of J-groove on the performance-curve instability, the experiment was conducted by using an axial flow pump, shown in Fig. 5, of which specific speed *Ns*=2000 [$m, m^3 / min, min^{-1}$] or the non-dimensional specific speed 0.80.

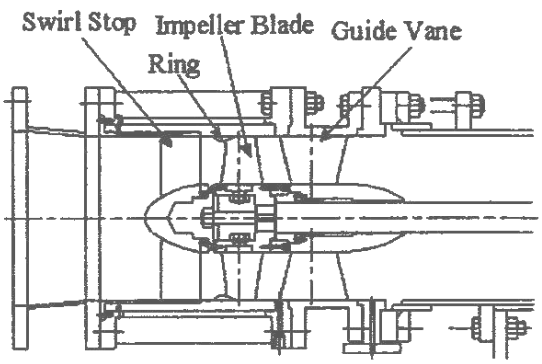


Fig. 5 Axial flow pump tested (*Ns*=2000)

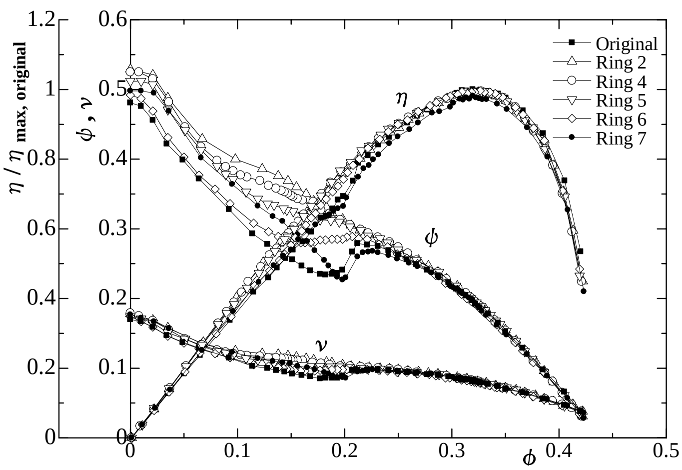


Fig. 6 Comparison of performance curves

The depth of groove d was varied with the number n and the width w of the grooves kept fixed, that is n=40 and w=15mm respectively except for the "Ring 5(n=20)," while the impeller radius r_2=140mm. The height of diagonal ring is equal to the groove depth d. The case of non-diagonal ring without impeller tip cut is also compared as "Ring 7".

The performance curves are compared in Fig. 6, in which the original curve shows a considerable instability. The depth of groove d was varied to 7mm (Ring 1), 5mm(Ring 2 and Ring 7), 3mm(Ring 4 and Ring 5) and 1.5mm(Ring 6). It is clearly shown that the diagonal ring with a new type J-groove can make the performance-curve instability disappear perfectly in the entire flow range without dropping the maximum efficiency, when the groove depth is in the range of $3 \le d \le 7$ [mm]. It is also recognized that the diagonal shape of the ring with impeller tip cut has an important role in stabilizing the performance curve.

4. Suppression of Rotating Stall

A vaned or a vaneless diffuser is installed at the downstream of an impeller of a centrifugal type or a mixed flow type, and recovers pressure by decreasing high fluid velocity. When a rotating stall initiates here in the low flow range, a periodical pressure fluctuation works on an impeller and a diffuser channels, resulting in the damage of an impeller or diffuser vanes. Thus, many studies have been performed so far to reveal the mechanism and suppress a rotating stall in a vaneless and a vaned diffusers, such as a casing treatment and an active control technique[1]. However, they require complicated mechanisms, and it is as yet a strong requirement to find a simple method of suppressing a rotating stall.

The J-groove technique is most suitable to suppress these phenomena, as it increases the flow-rate of the J-groove region due to the groove reverse flow and decelerates a fluid whirl velocity. Thus the present author has applied J-groove to a vaneless diffuser and a vaned diffuser.

4.1 Rotating Stall in Vaneless Diffuser[5]

When the flow-rate was decreased in the parallel-wall vaneless diffuser channel installed around a centrifugal impeller shown in Fig. 7, the flow angle became smaller, partial separation of flow occured in the boundary layer and many types of pressure fluctuation, shown in Fig. 8 caused by a rotating stall were observed depending on the sectional averaged flow angle α .

A periodical pressure fluctuation initiates at the flow angle of $\alpha = 26^\circ$, and this pressure fluctuation grows to a clear sinusoidal oscillation as shown in Fig. 8(a) with two cells. With further decrease in the flow angle both the frequency and the amplitude increase and the oscillation changes from a sinusoidal wave to a triangular wave as shown in Figs. 8(b). At about $\alpha = 12^\circ$ another oscillation with one-cell and much lower frequency is imposed on the two-cell oscillation and grows rapidly with a decrease of α as shown in Figs. 8(c) and (d). The pressure fluctuation takes the maximum at about $\alpha = 8^\circ$, and the amplitude of one-cell oscillation amounts to about twice the two-cell oscillation.

The behavior of the above described rotating stall seems to be much different from that previously reported.[1][13] This is because the parallel-wall diffuser used here is large enough to realize various types of rotating stall. Rotating stall in a large-sized vaneless diffuser shows many different patterns depending on the flow angle, as the region of the reverse flow induced near the wall hardly expand to the diffuser outlet. On the contrary, in a small-sized diffuser the flow pattern of rotating stall changes little with the change in flow angle.

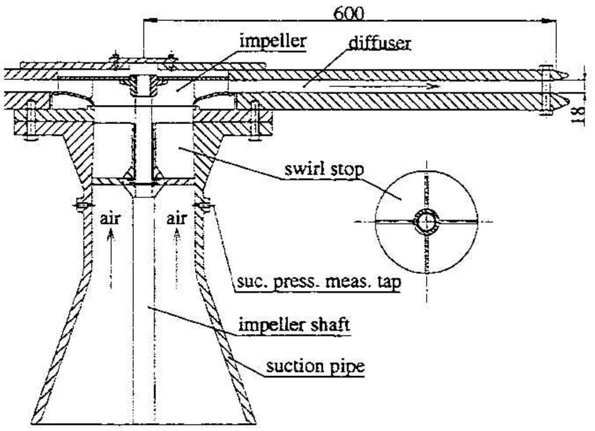


Fig. 7 Parallel-walled vaneless diffuser tested

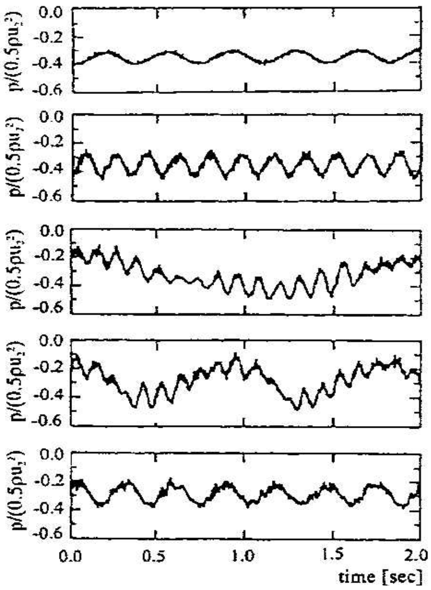


Fig. 8 Pressure fluctuation of rotating stall

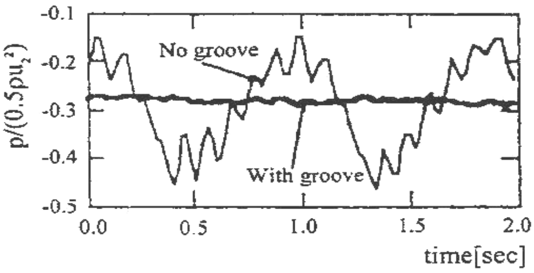


Fig. 9 Remarkable effect of J-groove on suppressing a rotating stall in vaneless diffuser

When J-groove is installed and the number *n* is increased from $n = 4$ with the width and the depth kept fixed, $10^w \times 3^d$, the amplitude of pressure fluctuation decreases rapidly, and the inception of rotating stall is delayed remarkably. But it is not until $n = 32$, when the stall is completely suppressed over the entire flow range. As an example, the wall pressure fluctuation for the groove case and no groove case is compared in Fig. 9. This is the case of the largest pressure fluctuation, that is α=8.4°. It is clearly seen that the periodic pressure fluctuation is perfectly suppressed. It was also revealed that the groove of 3mm depth on one sidewall have almost the same effect of suppressing rotating stall as the groove of 1mm depth on both sidewalls.

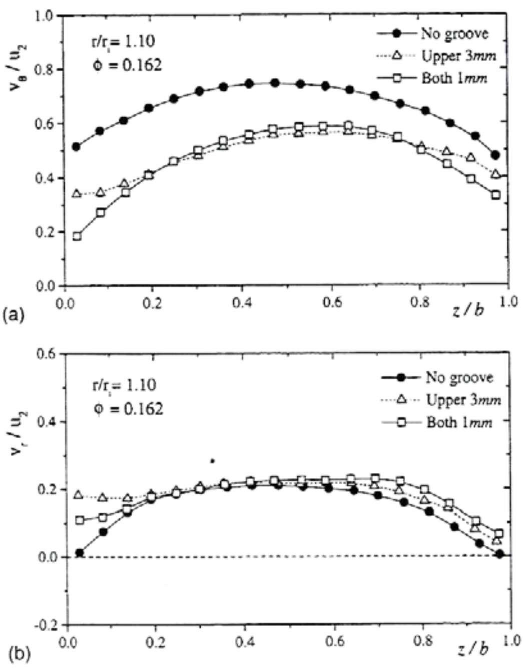


Fig. 10 Velocity distribution in the vaneless diffuser channel measured at r/r_2=1.1
(a)Tangential velocity (b)Radial velocity

In order to reveal the effects of J-groove, the time averaged velocity profiles at the impeller outlet in the parallel-wall diffuser channel are illustrated in Fig. 10, in which v_θ, v_r, u_2 are the tangential velocity, the radial velocity and the impeller tip speed, respectively, and *z/b* is the distance ratio from the upper sidewall. Figure 10 clearly reveals that the tangential velocity is largely and uniformly decreased by J-groove, while the radial velocity is increased especially near the diffuser wall.

Theoretical analysis has revealed that the flow-rate of the reverse flow induced in the grooves amounts up to 30~40% of the main flow, which makes the averaged flow angle increase by about 7 deg. It has also been made clear that the remarkable effects of radial shallow grooves are caused by the following two mechanisms; One is a remarkable decrease of tangential velocity at the diffuser inlet due to the mixing between the main flow and groove flow, and the other is a remarkable increase of radial velocity due to the reverse flow in the grooves. Both effects have almost the same weight of contribution to increase the flow angle. Even if the groove is very shallow such as d=1mm, the increase of flow angle is very large near the wall.

According to the CFD results, the reverse flow region near the wall occupies over the whole surface of the parallel-wall vaneless diffuser, when the rotating stall is perfectly suppressed. Further CFD analysis for the vaneless diffuser of a centrifugal compressor has revealed that the installation of J-groove near the outlet region of the diffuser is much more effective than that near the inlet region, and enlarges the stall margin largely by suppressing a rotating stall, though not confirmed yet experimentally.

4.2 Rotating Stall in Vaned Diffuser[6]

With a decrease of flow-rate in a vaned diffuser, the inlet flow angle into the diffuser vane becomes so small that a partial separation of flow occurs at the vane inlet and a rotating stall is caused. When the rotating stall initiates in the vaned diffuser, a strong and periodical reverse flow flows back into an impeller channel, which causes a sudden drop of pumping head, resulting in a performance-curve instability of a diffuser pump[14].

As a rotating stall is caused by too small flow angle at the diffuser vane inlet, to increase a flow angle is the best way to suppress this anomalous flow phenomena. For this purpose J-groove is suitable as it decelerates a flow rotation. In order to confirm the J-groove effects on suppressing a rotating stall in a vaned diffuser, the test stand shown in Fig. 11 was used. As the space between the impeller outlet and the diffuser inlet is very narrow, the grooves are installed as shown in Fig. 12

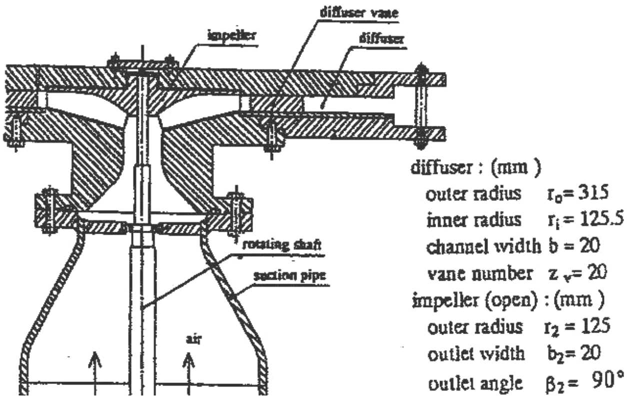


Fig. 11 Test stand of vaned diffuser

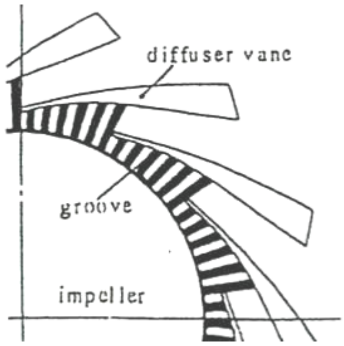


Fig. 12 Installation of J-groove and diffuser vane

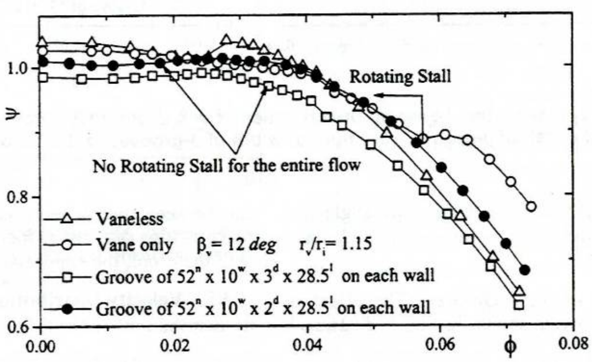
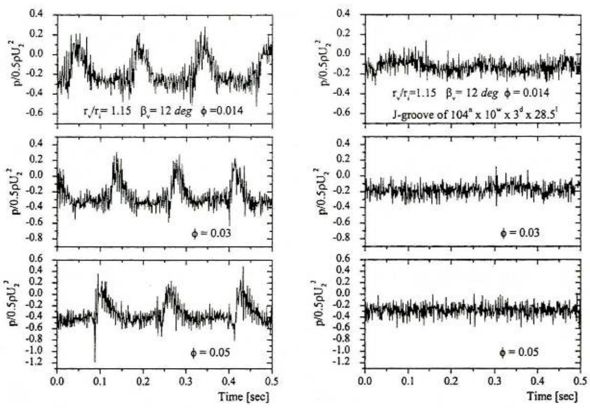


Fig. 13 Change of head curve $\psi-\phi$ by J-groove and range of rotating stall



(a) no groove case (b) grooved case

Fig. 14 Periodic pressure fluctuation

The head coefficient ψ for the groove and the no groove cases are compared in Fig. 13, in which the vaneless diffuser case is also compared. In the vaneless case the periodic pressure fluctuation is observed for the entire flow range, but in the vaned diffuser case the range of rotating stall is seen to become smaller. It is clearly seen that J-grooves of the optimum dimension can suppresses a rotating stall perfectly for the entire flow range. In this case, a large radial pressure gradient due to the strong whirl of the main flow induces radial inward jet in the groove, which creates angular momentum loss when the flow enters into and leavs from the groove. The pressure fluctuations at the impeller outlet for three different flow-rates are compared in Fig. 14. Periodical pressure fluctuation is seen to be perfectly suppressed by J-groove.

5. Suppression of Cavitation

Cavitation phenomenon is a main obstacle in developing a high speed pump. Cavitation causes not only performance drop by passage blockage in an impeller but also driving instability[1] and damages on the blade surfaces of the impeller. To avoid cavitation in the main impeller an inducer is usually used, but various kind of cavitation instability occurs in an inducer. In turbopump inducers for rocket engines the cavitation instabilities such as a rotating cavitation and a cavitation surge[15][16] occur and cause shaft vibrations and blade stress fluctuations.

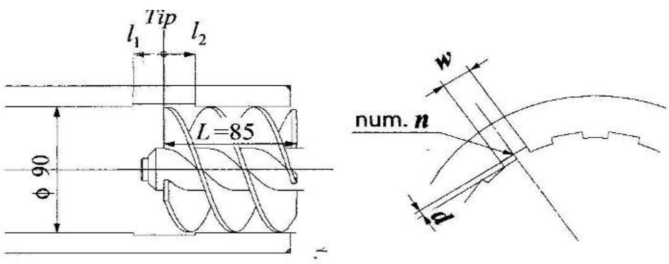
Another important characteristic of J-groove beside the control of flow rotation is to increase the pressure in the low pressure region by carrying the high pressure fluid to the low pressure region through the grooves. Therefore, if this characteristic of increasing the pressure at the low pressure region is applied to cavitation phenomenon, cavitation occurring in the pump might be suppressed. The present author has applied J-groove to suppress cavitation in inducers[17][18] and centrifugal pumps[19],

5.1 Cavitation in Inducer for Industrial Use[17][18]

In order to reveal the J-groove effect on cavitation, J-groove was applied to an inducer for industrial use, as an inducer is usually operated under the most severe cavitation condition. The centrifugal pump with a three-bladed inducer and four types of J-grooves, installed near the leading edge of inducer vane as shown in Fig. 15, have been tested.

The results of cavitation test are compared in Fig. 16, in which the suction specific speed S defined as $S \equiv nQ^{1/2}/(NPSHR)^{3/4}$ is expressed as the non-dimensional form normalized by $S_{nogroove}$ of the no-grooved case. Q is the flow-rate and $NPSHR$ is the required net positive suction head determined using the suction head at 3% head drop point.

Figure 16 clearly reveals that the suction performance is much improved by J-groove over the entire flow range especially in the small flow range, though a slight decrease of maximum efficiency was observed. As the improvement of S is remarkable in the small flow range for the types C and D than A and B, it is deduced that the downstream length of the groove from the blade leading edge is important for suppressing cavitation.



Inducer diameter	89 mm
Tip clearance	0.5 mm
Tip inlet blade angle	17.5 deg.
Blade number	3
Axis blade length	85 mm

Groove	number n	width w [mm]	depth d [mm]	length L [mm] l_1	l_2
A	16	10	2	20	20
B	32	6	2	20	20
C	16	10	2	0	20
D	16	10	2	0	40

Fig. 15 Inducers and J-grooves tested

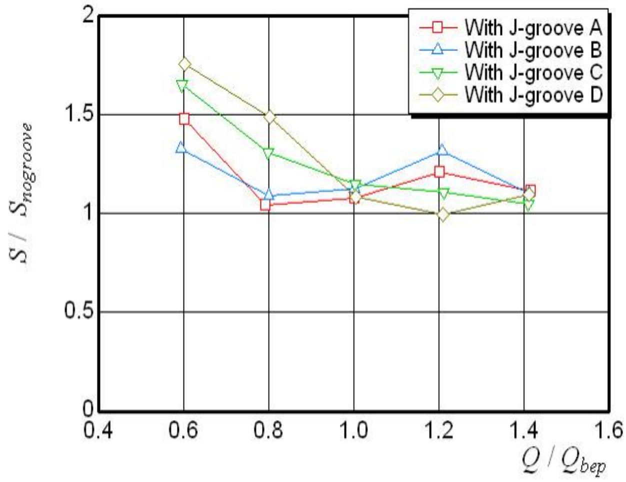
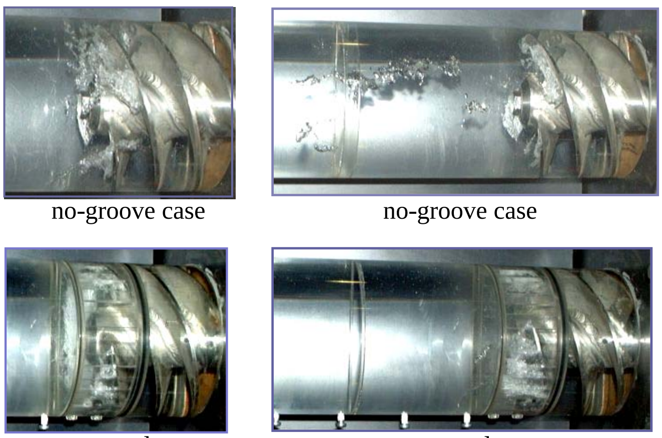


Fig. 16 Improvement of suction performance by J-grooves



no-groove case

no-groove case

grooved case

grooved case

(a) Best efficiency point (b) 60% flow rate of BEP

Fig. 17 Comparison of flow behavior (Groove-A, NPSH=0.72m.)

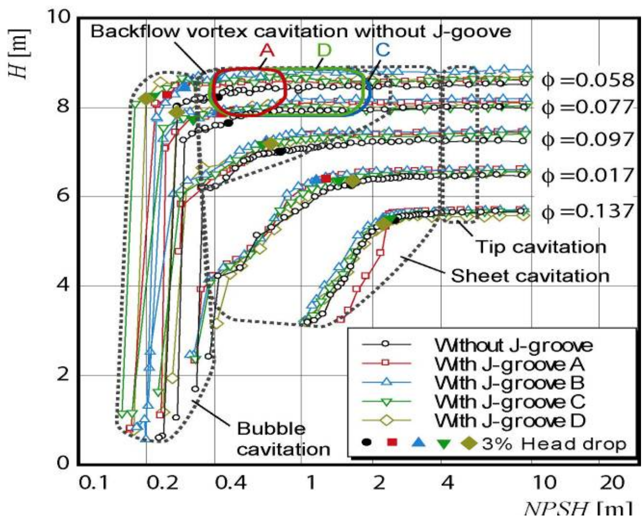


Fig. 18 Occurrence of various types of cavitation

The behavior of cavitating flow is illustrated and compared in Fig. 17 for the case of NPSH= 0.72m, At the best efficiency point shown in Fig 17(a) a spanwise bubble cavitation occurs at the leading edge and the back flow vortex cavitation is observed. With a decrease of flow-rate the reverse flow occurs near the blade leading edge, and the backflow vortex cavitation rotating much slower than the impeller grows at the share layer of the boundary of the reverse flow region. For the grooved case cavitation is confined at the vicinity of the blade leading edge.

With further decrease in flow-rate, the rotating backflow vortex cavitation becomes violent in the no groove case as shown in Fig. 17(b) and forward propagation of cavity into the upstream is clearly shown reaching to about twice of the inducer diameter from the leading edge. For the grooved case, the rotating backflow cavitation is seen to occur little only near the leading edge.

Figure 18 shows the occurrence regions for various types of cavitation. The dotted lines show the cavitating regions in the case of no groove and solid lines with “A,” ”C” and ”D” show the regions of the rotating backflow cavitation in the grooved case for corresponding J-groove. It is clearly recognized that the region of a rotating backflow cavitation is reduced largely by J-groove. In the case of B type J-groove, the rotating backflow cavitation was completely suppressed, which reveals that the upstream length of groove is important for suppressing a rotatig backflow cavitation.. This is attributed to the suppression of inlet reverse flow by J-groove as mentioned above.

Further study has also revealed that the J-groove combined with an inducer with back swept leading edge improves suction performance, and that cavitation surge can be almost suppressed by optimizing the shape of J-groove.

5.2 Cavitation Instabilities in Inducer for Rocket Engine[20][21]

It is very important to suppress cavitation instabilities in turbopump inducers. Since the instabilities are empirically known to be caused by the interaction of tip vortices with next blade, it is expected that J-Groove, which controls the flow near the tip, can be used to suppress cavitation instabilities [20].

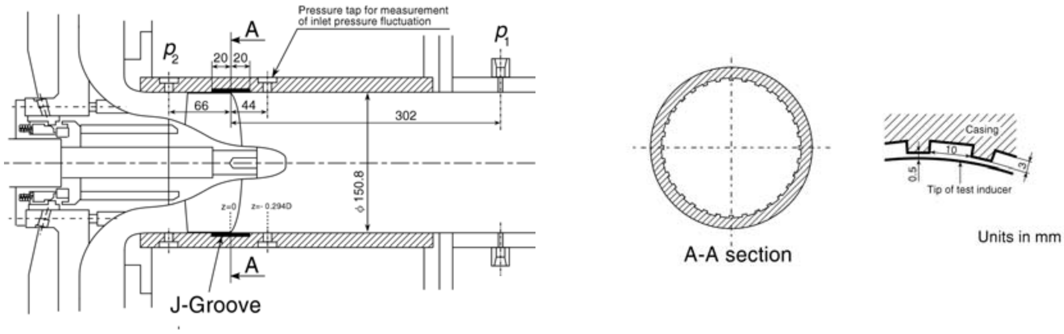
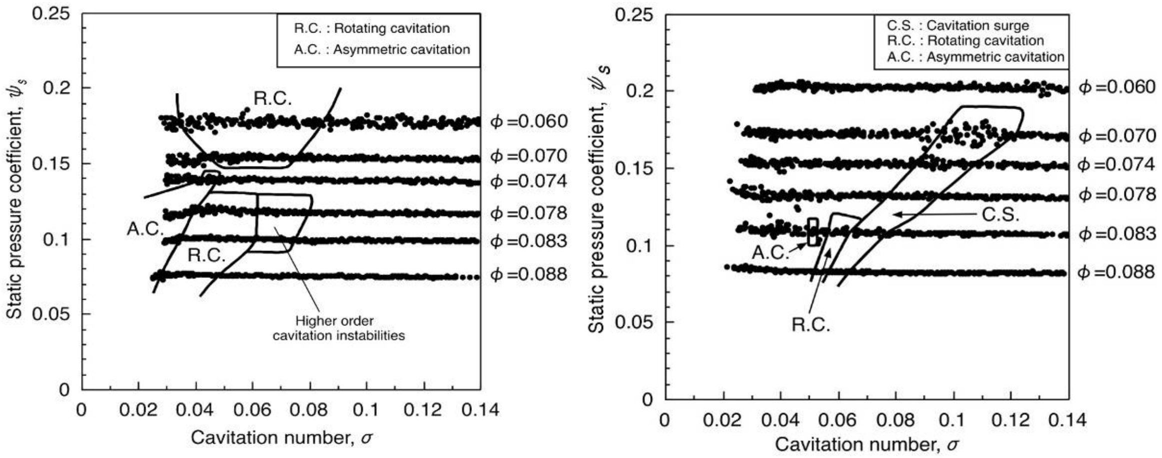


Fig. 19 Inducer tested and J-Groove (20,20)

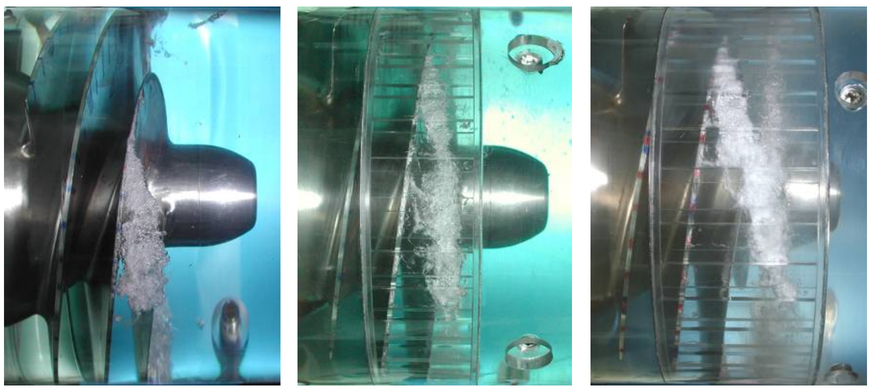


(a) Without grooves (b) With J-Groove(20,20)

Fig. 20 Occurrence regions of various cavitation instabilities

Figure 19 shows the test section and the geometry of J-Groove (20,20) with the axial length 20mm upstream and 20mm downstream. In Figure 20, areas of various cavitation instabilities are shown on the suction performance curves. Figure.20(a) shows the results without J-Groove, and Fig. 20(b) those with J-Groove (20,20). It is shown that rotating cavitation and attached cavitation are successfully suppressed by J-Groove (20,20). However, cavitation surge and rotating cavitation appears at higher cavitation number.

Figure 21 shows the cavity shapes. Without the groove, the tip cavity extends upstream smoothly and the space between the blade surface and the tip vortex cavitation is filled with cavitation bubbles. However, with J-Groove (20, 20), the tip cavity first extends upstream but it becomes parallel with the upstream edge of the groove. The tip cavities are lifted off from the blade. With J-Groove (20,20), the surge at higher cavitation number occurs when the



(a) Without grooves (σ=0.120, ϕ=0.078) (b) J-Groove(20,20) (σ=0.120, ϕ=0.078) (c) J-Groove (40,20) (σ=0.180, ϕ=0.060)

Fig. 21 Tip cavitation

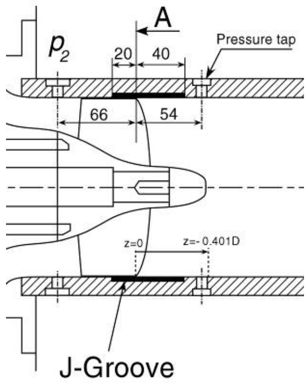


Fig. 22 J-Groove(40,20)

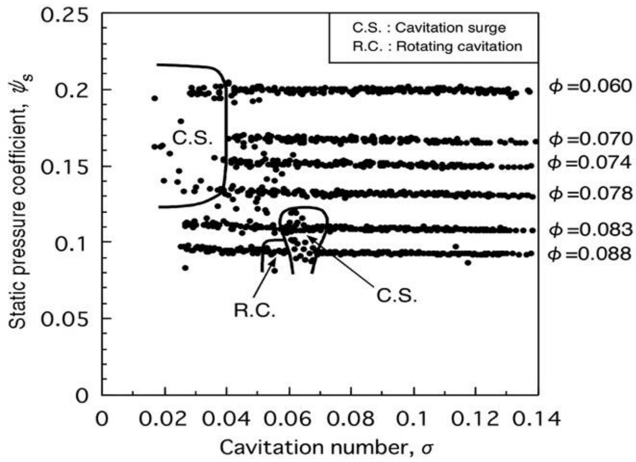


Fig. 23 Occurrence regions of various cavitation instabilities, J-Groove (40,20)

tip vortex interacts with the leading edge of the next blades.

To avoid the cavitation surge at higher cavitation number, J-Groove (40, 20) as shown in Fig.22 was tested. The results are shown in Fig.23. As expected, tip cavitation extends upstream as shown in Fig.21(c), and the cavitation surge at higher cavitation number could be avoided. However, a cavitation surge appeared at smaller cavitation number. At lower cavitation number, the space between the blade and the tip vortex cavity is filled with cavitation and the cavitation surge at smaller cavitation number occurred when the cavitation in the space starts to interact with the leading edge of the next blade.

Thus, it was proved that J-Groove can control the occurrence of cavitation instabilities. The experimental results supported the empirical rule that the cavitation instability occurs when the cavity trailing edge starts to interact with the leading edge of the next blade. Theoretical background was eventually given in [21].

6. Suppression of Draft Tube Surge in Hydro-turbine[22][23]

In the partial load operation of Francis turbine, a vortex rope appears in the draft tube and causes pressure fluctuation. The pressure fluctuation becomes violent, when cavitation is induced in the vortex core as shown in Fig. 24[22]. This anomalous flow phenomena is called as a draft tube surge, and often induces severe power swing in an electric generating system. To alleviate the pressure fluctuation, an active control such as air injection and a passive control such as fins installed in the inlet cone of a draft tube are popular means. However they require additional apparatus and eventually decrease the reliability.

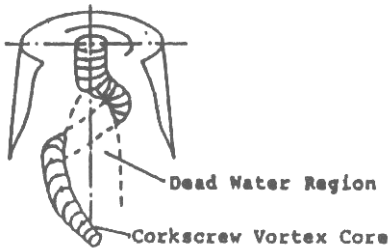


Fig. 24 Draft tube surge

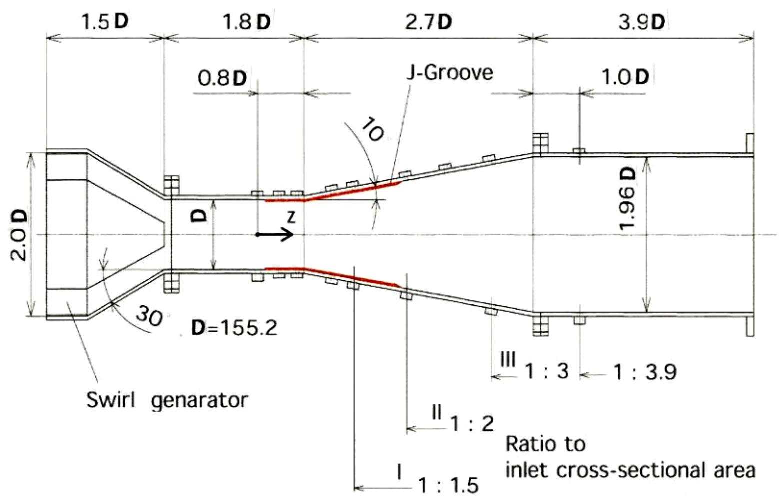


Fig. 25 Conical diffuser and J-groove tested

As a draft tube surge is principally caused by the flow rotation of a runner outlet flow, J-groove technique is suitable to suppress this anomalous flow phenomenon. Thus two kinds of the conical diffusers of different divergent angle $\gamma = 20^\circ$ [23] and 30° [25] have been used as shown in Fig. 25 instead of a draft tube, and rotational flow has been introduced into the test section, where several kinds of grooves are installed in the axial direction.

The time averaged sectional velocity profile in the diffuser channel is shown in Fig.26 for three different swirl strength *m*, which is defined in the upper part of Fig. 27. V_θ and V_z are the tangential and the axial velocity, respectively and V_{z0} is the averaged axial velocity at the diffuser inlet. It is noticed that with an increase of swirl strength the main flow comes to deviate to the diffuser wall especially in the divergent channel, and the flow in the central region becomes reverse. This whirling flow changes the pressure field in a divergent channel drastically.

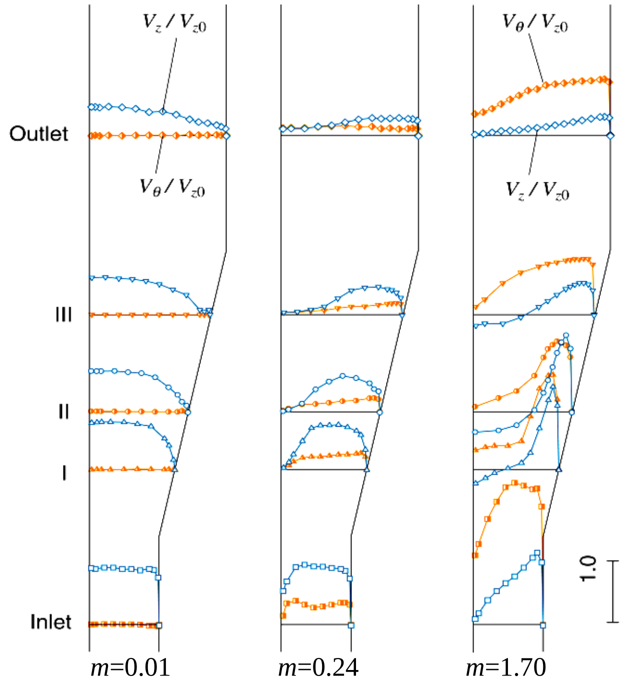


Fig. 26 Velocity distribution in a diffuser channel for different swirl strength ($\gamma = 20^\circ$)

swirl strength: $m \equiv \dfrac{\int v_\theta v_z r^2 dr}{D \int v_z^2 r dr}$ (at diffuser inlet)

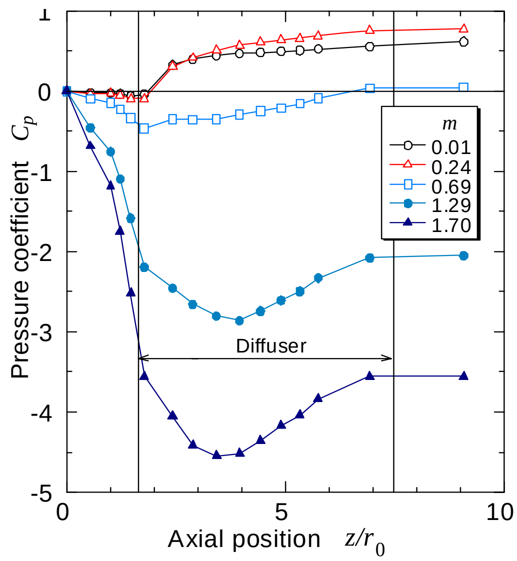


Fig. 27 Pressure distribution along diffuser wall (divergent angle $\gamma = 20^\circ$)

Figure 27 illustrates the pressure distribution along the diffuser wall (r_0 is the inlet pipe radius). When there is no swirl (m=0.01), the divergent channel works as a diffuser and pressure is recovered to the pressure coefficient C_p=0.60. In the weak swirl case of m=0.24, the pressure recovery become more than the no swirl case because of suppression of the flow separation. The swirl strength of Francis turbine at the runner outlet is around m=0.20 at the designed flow-rate. However, with an increase in the swirl strength, the pressure drops considerably in the divergent channel and reaches to C_p=-3.5 for the case of m=1.70. This swirl strength corresponds to the occurrence of the draft tube surge in Francis turbine. In this case the rate of pressure drop is the largest in the inlet straight pipe not in the diffuser channel. This might be because the reverse flow comes back into the central region of the inlet straight pipe and accelerates the flow near the wall.

When J-groove shown in Fig. 28 and Table 1 is installed along the divergent channel, the change of sectional velocity distributions at the outlet of the divergent channel is compared in Fig. 29.

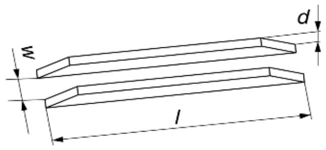


Fig. 28 J-groove used

Table 1 J-groove dimensions

type	w	d	l	n	position z/r_0
A	19	2	116	20	0.5-2.0
D	19	4	156	20	0.3-2.3
E	29-39	4	156	20	4.0-6.0

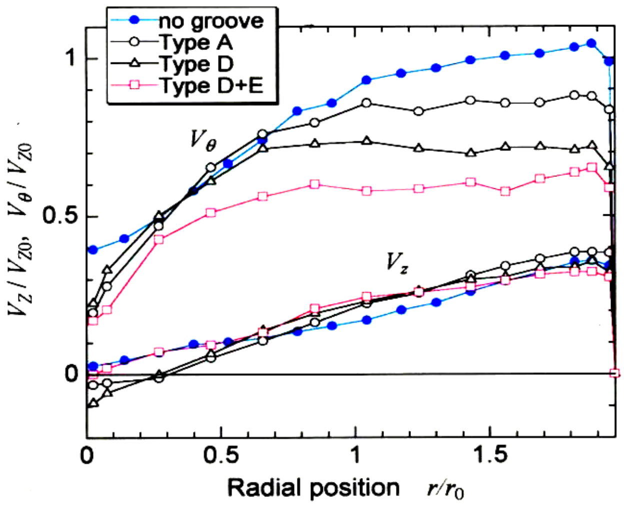


Fig. 29 Change of velocity distribution at diffuser outlet by J-groove for the case of m=1.70($\gamma = 20^\circ$)

Figure 29 clearly reveals that the tangential velocity V_θ is decreased considerably not only near the wall but over the whole section by installing the J-groove near the inlet and outlet of the diffuser channel(Type D+E). In this case the swirl strength was reduced to about 60% of the no groove case. The larger the inlet swirl strength is, the more the outlet swirl is suppressed by J-groove. The pressure fluctuation was also suppressed by 20-40%.

On the other hand, for the case of the diffuser of the divergent angle γ=30°, J-groove could suppress the swirl strength to about 36% of the no groove case for m=1.88. It was also revealed that the additional hydraulic loss created by J-groove is negligible and that a shallow but wide groove is more effective than a deep but narrow groove on suppressing the swirl strength of the main flow.

7. Axial Thrust Control[24]

Axial thrust working on an impeller/runner influences on the reliability and lifetime of a turbomachine largely. Axial thrust is mainly caused by an unbalance between the fluid forces working on the front shroud and the back shroud, and sometimes causes instability such as imposing severe axial vibration on the whole rotating parts including an impeller/runner. In a high head pump-turbine, the fluid force working on one sidewall of a runner amounts up to 10,000 tons, and thus only 1 % of unbalance force amounts to the axial thrust of 100 ton. Thus a perfect balance of axial force is inevitable for attaining high reliability, and various devices have been used, such as a balancing hole, a balancing pipe, a balancing disk, a self-balancing device, and so on. However, these devices cause the efficiency drop and additional equipments cause another instability.

The fluid forces working on the impeller/runner are mainly caused by the pressure distributions in the front and the back spaces of the impeller/runner. In order to control and reduce axial thrust, the J-groove technique is then most suitable, as the fluid rotation and the pressure distribution in the front and the back spaces of an impeller/runner are easily controlled by use of J-groove.

In order to reduce the axial thrust of multistage centrifugal pump shown in Fig. 30, many radial shallow grooves of 1mm in depth were installed on the front casing wall. The specific speed *Ns* of the test pump was 256 [$m, m^3/min, min^{-1}$] or the non-dimensional specific speed 0.10 and impeller radius r_2 was 47mm. Figure 31 shows the comparison of the axial thrust curves for changing the number n of grooves on the front casing wall of each impeller from 0 to 12. It is clearly shown that the axial thrust coefficient C_T decreases considerably and uniformly over the whole flow range with an increase of groove number *n*. In the case of *n*=12, the axial thrust at the best efficiency point($\phi = 0.08$) becomes almost 0 and that at the shut-off point is reduced to a half of the no groove case. The effect of J-groove can be predicted and the optimum design method is also presented[24].

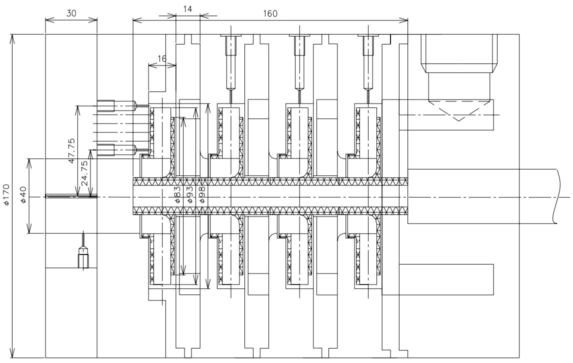


Fig. 30 Centrifugal pump tested

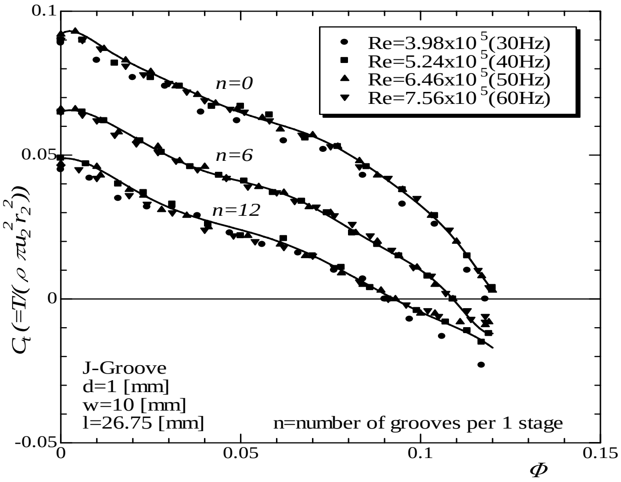


Fig. 31 Effect of J-groove on axial thrust curves

In the self-balance type turbomachines or the balancing equipments, an axial vibration is sometimes caused due to the instability of the axial thrust. In such cases the reduction of axial thrust is inevitable and J-groove technique is best suited. The stiffness of axial thrust against axial movement of rotating parts can also be improved with the J-groove.

8. Development of J-Groove Pump[25][26]

When J-groove is installed on the inner wall of a stationary casing, various anomalous flow phenomena can be suppressed. Then, what will happen when the J-groove is installed on the rotational wall? As the flow-rate in the grooves are considerably large even if the groove depth is very shallow, the groove should work as a flow channel such as an impeller channel.

Thus the radial grooves were installed on the back surface of a centrifugal impeller of very low specific speed, as shown in Fig. 32[25]. The triangular grooves of very short length were machined at the outer periphery of the impeller.

The performance curves for the three cases, the grooved cases on the front shroud, the grooved case on both shrouds and the no groove case, are compared in Fig. 33. It is recognized that the pumping head is about 10% increased by J-groove on the front shroud and 20% on both shrouds compared with no groove case. The original head-capacity curve shows performance-curve instability, but this instability is much improved by attaching J-groove, though the maximum efficiency dropped by about 3%. The mechanism of rising the pumping head is that the groove flow creates a strong vortex flow with large rotational velocity in both sides of the impeller outlet flow and increases the fluid rotational velocity at the impeller outlet.

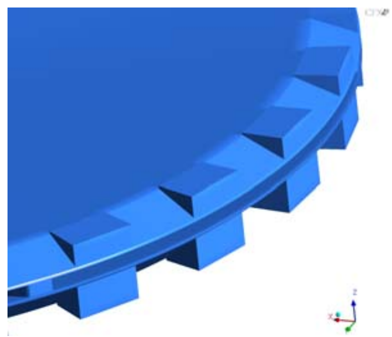


Fig. 32 J-groove at the back of impeller ($24^n \times 13.5^w \times 4.5^d \times 8^l$)

Accordingly, the J-groove technique is suitable not only to passively suppress several instabilities but also to positively create a new type pump. Here is shown an innovative pump, termed "J-groove pump," which is applicable to a very low specific speed range[26]. In the very low specific speed range such as Ns<80 [$m,m^3/min, min^{-1}$], the efficiency of a turbomachine becomes very low and a head-capacity curve tends to be unstable, and then a displacement type fluid machine has long been used.

However, a displacement type has many disadvantages, such as large noise and vibration, high accuracy required in machining every parts, short lifetime and difficulty in speed up. It is still in strong demand to apply turbo-type machine in such very low specific speed range.

The impeller of J-groove pump has many shallow grooves on both sides of a disk as shown in Fig. 34(a), and is enclosed in a circular casing. The CFD calculation results of the internal flow in the impeller channel is illustrated in Fig. 34(b) and the performance curve is illustrated in Fig. 35 for the case of specific speed Ns=60.

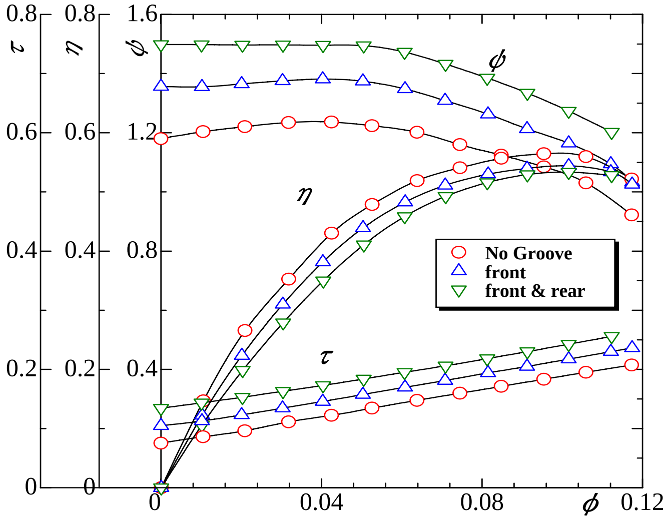
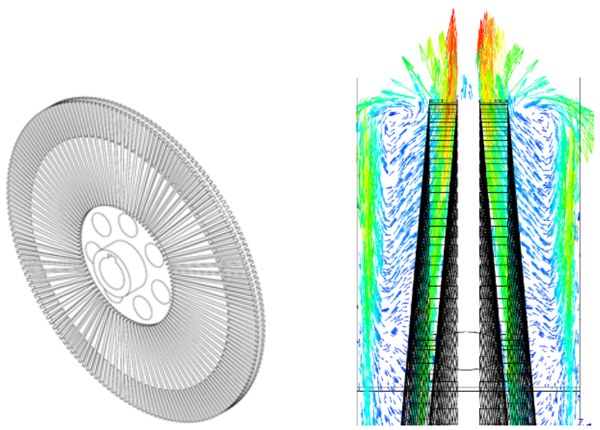


Fig. 33 Comparison of head-capacity curves

The head-capacity curve is seen to be stable in all flow range. As the J-groove pump has no performance instability, no parts for high accuracy in manufacturing and enough stiffness in high speed range, it is applicable to super high pressure range.



(a) Impeller (b) Impeller outlet flow(CFD)

Fig. 34 J-groove pump impeller and outlet flow

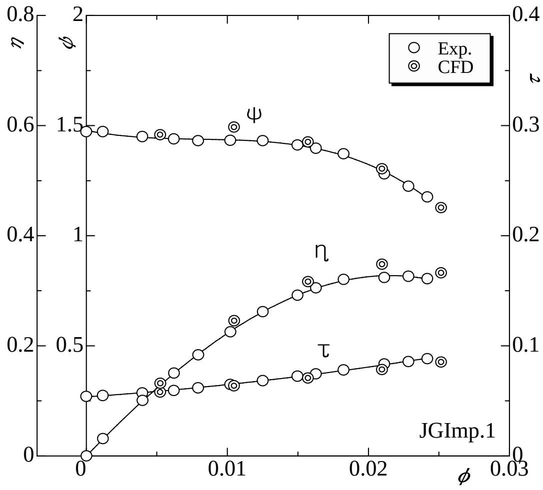


Fig. 35 Performances of J-groove pump

9. Concluding Remarks

The present device is a very simple and clear technique of installing many shallow grooves on the inner casing wall of a turbomachine in the direction of pressure rise. In the grooves is induced strong reverse jet due to the pressure gradient, which suppresses the rotation of main flow considerably. As is shown above, this device is able to suppress many anomalous flow phenomena commonly and conveniently with little drop of the maximum efficiency by installing no special equipment.

The present author wishes that the J-groove technique be further developed and the possibility of controlling and suppressing another anomalous flow phenomena be revealed. For example, in the sliding bearings or in the equipments with a narrow circular gap several instabilities such as impeller whirl or bearing whirl are caused when the lubricating fluid has an inlet whirl. In such cases, the J-groove technique is suitable for stabilizing the instabilities by suppressing the pre-whirl of the lubricant, though not yet confirmed.

Acknowledgments

Professor Tsujimoto of Osaka University took charge of summarizing Chapter 5.2 favorably. I would highly appreciate his cooperation and would like to express my sincere gratitude to Professor Tsujimoto for his help and the long acquaintance.

Nomenclature

C_p	Pressure coefficient (=$p/0.5\rho u_2^2$)	T	Axial thrust [N]
D	Depth of J-Groove [mm]	w	Width of J-Groove [mm]
H	Pumping head[m]	u_2	Tangential velocity of impeller periphery[m/s]
l	Length of J-Groove[mm]	α	Flow angle measured from tangential direction
m	Swirl strength		Divergent angle of diffuser[$deg.$]

n	Number of J-Groove	γ	Flow coefficient
Ns	Specific speed[$m, m^{3/}min, min^{-1}$]	ϕ	Efficiency
p	Pressure[Pa]	η	Fluid Density[kg/m^3]
r_2	Outer radius of impeller or disk[mm]	ρ	Head coefficient($=2gH/u_2^2$)
S	Suction specific speed[$m, m^3/min, min^{-1}$]	ψ	Shaft power coefficient
		τ, υ	

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