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The Effects of Impeller Vane Roughness and Thickness on the Characteristics of the Mixed-Flow Propeller Pump*

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Effects of the impeller vane roughness and thickness on the flow conditions and hydraulic losses as well as on the overall performances of the mixed-flow propeller pump are presented. The vane roughness has a most harmful influence upon the optimum performances, and also has a marked effect on the peripheral component of the discharge velocity near the impeller tip. The change in hydraulic losses in the impeller due to the variation of vane roughness is equivalent to that of the flat plate. The effects of roughness at various parts on the vane surface are also discussed.

The head-capacity curves with varying vane thickness may be calculated actually on assumption that the capacity varies proportionally to the space area between vanes while the head remains constant.

1. Introduction

There were several experimental investigation on the effects of vane roughness of the radial- and the axial-flow impellers on over-all pump performances⁽¹⁾, but little investigation has been made on its effect on the impeller characteristic and the flow condition. It is expected that the effect of roughness on the pump performance varies with the ratio of the impeller loss to the total hydraulic losses, because the drop of the pump performance with increasing vane roughness is mainly due to the increase of the losses in the impeller. So it may be desirable to study the change in the impeller characteristics (i.e. input head-capacity and hydraulic losses curves) with impeller vane roughness.

In this paper the experimental results about the effects of the impeller vane roughness on the impeller characteristics and the flow conditions at the impeller discharge as well as on the over-all performance of the mixed-flow propeller pump are presented.

The vane roughness mentioned above is equal on every vane surface, but it is not expected that the roughness on some part of each vane surface has the same effect on the performance as the other part of the vane surface. Then the effects of roughness at various parts of the vane surface are also presented.

Moreover it is necessary to obtain the effect of vane thickness which varies usually with the changing vane roughness. A. J. Stepanoff⁽²⁾ said that the vane thickness had no effect on the performance of an axial-flow blower by quoting the experimental result by B. Eckert. However it may be expected in pumps of which impellers generally have large solidities and low vane angles that the pump performance will be affected by the thickness because of marked decrease of the flow area through the impeller by an increased vane roughness. In this paper, therefore, the effect of vane thickness is also presented prior to describing the effect of the vane surface finish.

2. Experimental equipment

Fig.1 shows the sectional elevation of the main part of the test pump which has an open impeller and a diffusion casing. The impeller

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diameters at the inlet and the outlet were 122 and 138 mm, respectively, and the ratio of the hub dia. to the impeller tip dia. at the outlet was 0.5.

The layout of the test circuit is shown in Fig. 2. Water was pumped from the underground tank to the head tank, and sent into the test pump through a 6-inch diameter piping in which a grid was mounted. The level in the head tank was maintained at a height of two meters above the pump axis.

The driving torque was measured mechanically by swinging-casing 5-hp electric dynamometer, whose speed was measured by a tachometer. The overall pressure rise or the total head was measured by a mercury U-tube for the higher

pressure difference and by an inverted water U-tube for the lower. The flow capacity was measured by 90 degree triangular weir. The small L-type static pressure tubes and cobra-type total pressure tubes with yawmeters made of stainless-steel pipe having 0.8 and 1.1 mm diameters were used for internal flow measurement by installing in each place shown in Fig. 1.

The mechanical loss power was measured by driving the impeller hub only in water keeping the same axial thrust as in testing the pump, which could be determined from the strain of the diaphragm warped by axial thrust given by regulating the spring installed in the lid cover as shown in Fig. 1.

The principal design values of the impeller tested are given in Table 1. These are open impellers, and their tip clearances to the casing were kept at a constant value (about 0.2 mm). The principal design values of the diffusion vane used are given in Table 2.

The impeller vane surfaces were finished by means of buffing, filing, casting alone, and coating a piece of cloth or a net on the vane surface, and maximum heights of protrusion of each surface were about 0.003, 0.030, 0.10, 0.20, 0.22, and 0.30 mm, respectively (hereafter these roughnesses will be represented as 3s 30s 100setc.).

Moreover, in order to measure the roughness effect at different places on the vane surface, the method of coating a cloth locally on the vane surface as follows was adopted; (a) either surface on the pressure or the suction surface, (b) either the outer or the inner half surface from the root mean square radius line on the vane (i.e. the tip side or the hub side surface), (c) either leading or trailing surface from the bisector of the vane chord length (i.e. the inlet side and outlet side surface). The rest of the vane surface except the one cloth-coated was painted so as to give the same thickness as the coated cloth.

The method of coating some pieces of cloth was also adopted in changing the vane thickness. In this case, two kinds of vane surface were used, one being rough (200s) as cloth-coated and the other smooth (3s) as painted on the cloth.

3. Effects of impeller vane thickness

The overall pump performances of Impeller M-39 Fo are shown in Fig. 3, when the vane thickness changes, where ϕ is the flow coefficient ($=Q/A_2 r_{2m} \omega$, Q is

Table 1 Principal design values of impellers

Impeller No.		M-39Fo	M-39Co
Inlet angle at mean radius (deg)		23.0	23.0
Outlet angle (deg)	at hub	52.0	80.3
	at mean radius	39.0	39.0
	at tip	32.7	27.0
Number of vanes		5	5
Mean pitch chord ratio	at hub	0.623	0.762
	at mean radius	0.835	0.835
	at tip	0.967	0.843

Table 2 Principal design values of diffusion vane

Development	at hub	at mean radius	at casing wall
Inlet angle (deg)	57.5	51.5	47.3
Outlet angle (deg)	90.0	90.0	90.0
Pitch chord ratio	0.552	0.463	0.328

Fig. 2 Layout of test circuit

Fig. 1 Sectional elevation of test pump

the flow capacity, A the cross area of the flow passage at impeller outlet, r_{2m} the root mean square radius at impeller outlet, ω the angular velocity, ψ the total head coefficient ($=gH/r_{2m}^2\omega^2$, H the total head, g the gravitational acceleration), τ_b the input torque coefficient ($=gT_b/\gamma A_2 r_{2m}^3\omega^2$, T_b the input torque, γ the specific weight), η the overall efficiency, b and l_t the maximum vane thickness and the chord length at the outer edge of impeller, respectively.

It will be seen from Fig. 3 that the thickness has a marked effect on the performance in the higher capacity range, and ψ , τ_b , and η decrease by increasing the thickness, but has very little effect in the lower capacity range where a backflow arises in the impeller inlet. The same result has been obtained for the smooth finish impellers.

In order to show the effects of thickness clearly, the changes in optimum performance by thickness ratio b/l_t are illustrated in Fig. 4, where N_s is the specific speed [$m^3/\min$, m , rpm units], t the mean value of the inlet and the outlet blade pitch, t' the circumferential distance between vanes, and d the minimum space across flow pas-

sage (all the values mentioned above are the ones in the development on the root mean square radius), and the subscript m refers to the optimum performance, and the subscript 0 to the performance of the impeller having minimum thickness vanes ($b/l_t=4.2\%$), respectively.

It is seen from Fig. 4 that ψ_m remains constant but ϕ_m and τ_{bm} decrease linearly by increasing the vane thickness, and that N_s and η also decrease with the increasing thickness except for the small thickness.

The change in ϕ at the specific values of the head coefficient ($\psi=0.1, 0.2, 0.3$, and ψ_m) by the passage area ratio d/d_0 is shown in Fig. 5 except in the backflow range. As will be seen from this figure the solid line which represents the line of $\phi/\phi_0=d/d_0$ is in good agreement with the test data, so it may be seen that the change in ϕ - ψ characteristic by the thickness may be obtained actually by dealing with that reducing ratio of the capacity which is proportional to the reducing ratio of the flow passage area with an increase of the vane thickness while the total head remains constant.

It is desirable to know which is more harmful on the performance, the decrease of the input head or the increase of the hydraulic loss in the impeller. Then the input head coefficient $\bar{\psi}_{th}$ ($=gH_{th}/r_{2m}^2\omega^2$, H_{th} is the input head) and the hydraulic efficiency $\eta_h(=\psi/\bar{\psi}_{th})$ are shown in Fig. 6, and hydraulic loss coefficient in the impeller $\theta_i(=gh_i/r_{2m}^2\omega^2$, h_i is the loss of head in the impeller) and the impeller stage efficiency $\eta_i[(\bar{\psi}_{th}-\theta_i)/\bar{\psi}_{th}]$ are shown in Fig. 7.

As will be seen from these figures, $\bar{\psi}_{th}$ decreases and θ_i increases by increasing the thickness, so

Fig. 3 Overall performances (effects of vane thickness)

Fig. 4 Optimum performance (effects of vane thickness)

Fig. 5 Change in capacity at same heads (effects of vane thickness)

η_h and η_i decrease too, and the reduction of $\bar{\psi}_{th}$ due to the increase of the thickness has a

slightly more harmful effect on the optimum performance than the increase of θ_i .

Fig. 6 Input head and hydraulic efficiency (effects of vane thickness)

Fig. 7 Impeller efficiency and hydraulic loss (effects of vane thickness)

Fig. 8 Deviation angle (effects of vane thickness)

Fig. 9 Overall performance (effects of vane roughness)

The change in the deviation angle due to the thickness is shown in Fig. 8, where R is the outlet radius ratio ($=r_2/r_{2m}$, r_2 is the radius at the impeller outlet), and $\Delta\beta_2$ the deviation angle of the flow at the impeller discharge. Although the difference of $\Delta\beta_2$ is very slight near the open edge, $\Delta\beta_2$ increases with an increasing thickness at smaller radius because of larger vane solidity⁽³⁾. For this reason $\bar{\psi}_{th}$ decreases with an increasing thickness as mentioned above.

4. Effects of whole surface finishes

4.1 Experimental results

Fig. 9 shows the performances with the impellers having various

roughness. It will be seen that the roughness exerts a little effect in the lower capacity range where the back flow arises, though it has damaging effects on the performance in the higher capacity range. However the effects of roughness can not be known from this figure because of variation in vane thickness. Then converting all the performances into the ones having the same thickness as the smooth impeller (3s), the impeller torque coefficient $\tau_i = (T_b - T_{mech}) / (\gamma A r_{2m}^3 \omega^2 / g)$, where T_{mech} is the mechanical loss torque in the packing and the bearing] and the hydraulic efficiency η_h for the same thickness impellers are shown in Fig.10. Although τ_i and consequently $\bar{\phi}_{th} (= \tau_i / \phi)$ change slightly by the roughness, ϕ and η_h decrease considerably with an increasing roughness because of the marked increase of hydraulic loss in the impeller as described later. It is also seen from Fig.10 that the roughness exerts a most harmful influence on the performance especially at the best efficiency point and also at higher capacity range, where the friction

Fig. 10 Comparison of performances (effects of vane roughness)

Fig. 11 Optimum performances (effects of vane roughness)

loss is considerably larger than other losses because of larger relative velocity.

The changes in the optimum performance and the shut-off power ratio $t_{bs} (= T_{bs} / T_{bm}, T_{bs}$ and T_{bm} are the input torques at zero capacity and at the best efficiency point, respectively) by the roughness k_s are shown in Fig.11, where k_s^* will be described later. It may be seen from this figure that ϕ_m remains nearly constant, and ϕ_m, N_s and the maximum hydraulic efficiency η_{hm} decrease with an increasing roughness for the two kinds of test impellers as indicated in the preceding paper⁽⁴⁾. Besides, t_{bs} in Impeller M-39 Fo decreases with an increasing roughness, but the one in Impeller M-39 Co remains constant. The reason for this is the difference of the input power—capacity curves. At the best efficiency point, the value of input power or T_b in M-39 Co remains nearly constant but the one in M-39 Fo decreases with an increasing capacity⁽⁵⁾, hence the input power at the best efficiency point increases in M-39 Fo but remains unchanged in M-39 Co with an increasing roughness, since the capacity at the best efficiency point decreases by increasing the roughness as shown in Fig.11. On the other hand the shut-off power remains unchanged due to the roughness as shown in Fig.10, hence the tendency in t_{bs} curves differs between two impellers as mentioned above.

The changes in θ_i and η_i by the roughness are shown in Fig.12, where the pressure rise coefficient in the impeller at the best efficiency point, ΔP_{sm} , is also shown ($\Delta P_{sm} = g \Delta p_{sm} / \gamma r_{2m}^2 \omega^2$, Δp_{sm} is the pressure difference between the discharge pressure at the root mean square radius and the inlet wall pressure). It is natural that θ_i increases with an increasing roughness, hence η_i and ΔP_{sm} also decrease.

The velocity distributions at the impeller discharge are shown in Fig.13, where V_{2m} and V_{2u}

Fig. 12 Impeller characteristics (effects of vane roughness)

are the non-dimensional velocity components ($V_{2m} = v_{2m}/r_{2m}\omega$, $V_{2u} = v_{2u}/r_{2m}\omega$, v_{2m} and v_{2u} are the meridian and the peripheral velocity components, respectively, at the impeller discharge). There is a slight difference in meridian velocity distributions at the equal capacity between the smooth and the rough finishes, but the peripheral velocity component near the open edge considerably increases with an increasing roughness. The reason for this is probably that the boundary layer on the vane surface, in which flow has the same peripheral velocity as the impeller, and the thickness of layer increases with the roughness, shifts toward the open edge due to the centrifugal action, and then flows out in the impeller discharge⁽⁶⁾. Thus the larger the vane roughness, the larger the peripheral velocity component becomes near the open edge.

4.2 Consideration of experimental results

The value of the admissible roughness k_s^* , which is defined as the critical value of protrusion on the vane surface that has no effect on the through-flow in the impeller, is still unknown for such an impeller having three dimensional vanes as mixed-flow impellers. Then let us set up a rough standard of k_s^* in the test impellers by using the formula for k_s^* in the flat plate by H. Schlichting⁽⁷⁾ ($k_s^* = 100\nu/w$, where ν is the kinematic viscosity, and w the relative velocity).

Calculating by using the mean value of the relative velocities on the root mean square radii at the inlet and the outlet as w in the formula, the values of 0.023 mm for M-39 Fo and 0.020 mm for M-39 Co are obtained at the best efficiency point. So the impeller vane surface with the smallest protrusion (3s) can be regarded as the so-called hydraulically smooth surface, and the calculated k_s^* may be considered to be the admissible roughness in the tested impellers due to the decreasing tendency in maximum efficiency with an increasing roughness as shown in Fig. 11⁽⁸⁾. Then Fig. 14 shows the ratio of the optimum performance of various roughness impellers to that of smooth impeller with 3s finish impeller in M-39 Fo and the one at $k_s = 0.020$ mm obtained by extending the tendency of the experimental results for M-39 Co shown in Fig. 11. The ordinate k_s/l_m is the relative roughness, where l_m is the vane chord length in the development at the root mean square radius, and the subscript * refers to the performance with the smooth finish impeller. As will be seen from Fig. 14, the changes in both the capacity ratio ϕ_m/ϕ_m^* and the head ratio ψ_m/ψ_m^* are the same in two impellers, and the drop of ϕ_m/ϕ_m^* with an increasing roughness is considerably larger than that of ψ_m/ψ_m^* .

Fig. 13 Velocity distributions at outlet (effects of vane roughness)

Fig. 14 Change in optimum performances (effects of vane roughness)

There is a slight difference in the drop of the maximum hydraulic efficiency η_{hm}/η_{hm}^* between the two impellers. For reference, both the experimental data on the centrifugal pump by M. Shirakura⁽¹⁾ and those on the axial-flow fan by T. Ikui^{(1)†} are also plotted in the figure. It can be seen that the vane roughness in the mixed-flow impeller has a more harmful effect than in the axial-flow impeller. However, it can be expected that the difference in the η_{hm}/η_{hm}^* between the mixed- and the radial-flow impellers will become smaller because both the front and the back shroud have the same finishes as on the vane surface in radial-flow impeller, thus the roughness of the vane only will have less effect than shown in the figure.

The hydraulic losses in an open impeller may be composed of the friction losses on the vane surface and on the annular wall, the shock loss in the impeller inlet, and the tip-clearance loss. Because it may be considered that all the shock loss, the tip-clearance loss and the annular wall friction loss remain

† The overall efficiency is plotted instead of the hydraulic efficiency which has not been given in Ikui's paper.

constant at the same capacity with an increasing roughness, in this case the difference of the total loss will depend on the difference of the friction loss due to the change in roughness in the impeller. Then let us compare the impeller friction loss obtained as mentioned above to the friction loss in the flat plate.

The drag of a flat plate per unit width, D_f , may be given as the following equation,

$$D_f = c_f \gamma l w^2 / 2g \quad (1)$$

where l is the length of the plate and c_f the drag coefficient per one side of the plate.

If the plate having a smooth surface has a round leading edge and is covered with the turbulent boundary layer, c_f can be given by the following equation⁽⁹⁾,

$$c_f = 0.074 R_e^{-0.2} \quad (2)$$

where R_e is the Reynolds number ($=wl/\nu$). Then in the rough plate having the protrusion k_s , the following equation is given for c_f in a complete turbulence region⁽¹⁰⁾.

$$c_f = \{1.89 + 1.62 \log_{10} (l/k_s)\}^{-2.5} \quad (3)$$

Using w_m and l_m instead of w and l , respectively, in Eq. (1) as in the calculation for k_s^* , the friction loss in the impeller h_i may be given by the following equation,

$$h_i = 2z D_f w_m / \gamma Q = c_f (z l_m B w_m^3 / g Q) \quad (4)$$

where z is the number of vanes, B the width of

flow passage through the impeller, and Q the capacity. So the change in h_i may be given by the following equation,

$$\Delta h_i = (c_f - c_{f0}) (z l_m B w_m^3 / g Q) \quad (5)$$

where $\Delta h_i = h_i - h_{i0}$, and the subscript 0 refers to the smooth surface. Thus, if the change in the impeller friction loss by the roughness is the same as in the flat plate having the same dimension, Δh_i obtained by the experimental results may be proportional to $(c_f - c_{f0})$.

Fig. 15 shows the relation between $(c_f - c_{f0})$ and $\Delta \zeta_i$, where $\Delta \zeta_i$ is the difference of the loss coefficient ζ_i by the change in roughness, and $h_i = \zeta_i (w_m^2 / 2g)$. It may be seen from this figure that $\Delta \zeta_i$ is practically proportional to $(c_f - c_{f0})$, and this relation in the capacity higher than the best efficiency point does not change with the capacity.

Thus, from these results it may be understood that the difference in the performance is equal to that of the friction loss, therefore the hydraulic loss can be separated into the friction and the other losses.

It is, however, questionable that the relation between $\Delta \zeta_i$ and $(c_f - c_{f0})$ in Fig. 15 does not pass the zero point. The reason for this is probably that the values of c_f obtained by Eqs. (2) and (3) become larger than the real ones because of the admissible roughness being left out of consideration. In this connection, using $(k_s - k_s^*)$ instead of k_s in Eq. (3) and $k_s^* = 0.023$ mm, the linear relation passing the zero point can be obtained.

5. Effects of local surface finishes

5.1 Experimental results

The overall performances obtained by using the impeller cloth-coated either on the pressure face or on the suction face of the impeller are shown in Fig. 16. As will be seen from this figure, it is evident that there is little difference in the input torque coefficient τ_b between the two impellers in the whole capacity range, but both the head coefficient ϕ and the overall efficiency η for the impeller with the rough suction face of the vane are somewhat lower than for the one with the rough pressure face especially in the best efficiency point. Since there is no difference in low flow range where the backflow arises, the results in low capacity range are omitted.

For reference, the performances

Fig. 15 Coefficient of friction loss

Fig. 16 Effects of local roughness (pressure face and suction face)

obtained by using the impeller cloth-coated on the whole surface and the impeller with smooth vanes (30s) without the piece of cloth are also illustrated by the broken and the dash-dot lines, respectively, in Fig. 16. It is natural that the performances obtained by using the impeller having rough surface on its one side lie midway between the broken and dash-dot lines. However these performances can not be compared qualitatively with each other because of variation in vane thickness, so these will be described in the following paragraph.

Fig. 17 shows the performance obtained by using the impeller having cloth-coated vanes either on the tip side or on the hub side of vanes. Although τ_b does not change, ϕ and η_h obtained by using impeller having the rough tip side are considerably lower than those of the rough hub side impeller except in low capacity range, so it is evident that the protrusion on the vane surface near the open edge exerts a most harmful influence on the pump performance.

Fig. 17 Effects of local roughness (tip side and hub side)

Fig. 18 Effects of local roughness (inlet side and outlet side)

Fig. 18 shows the performance obtained by using the impeller cloth-coated either on the leading or on the trailing half of vanes. Although there is little difference in τ_b between the two impellers tested as well, both ϕ and η_h for the impeller having the rough leading half (rough inlet side) vanes are considerably lower than the ones for the rough latter half (rough outlet side) vanes in somewhat higher capacity than the best efficiency point. This difference is so slight in the maximum capacity ($\phi \approx 0$) as to be negligible, and the reason for this is probably that the separating point of the flow along the vane moves forward due to an increase in the negative attack angle of the relative flow to the vane.

5.2 Consideration of experimental results

Fig. 19 shows all the efficiency-capacity characteristics of the impeller having the same thickness as the fully cloth-coated one, which are obtained from the performance as mentioned in previous paragraph by using the results in Chap. 3

so as to be able to compare each performance qualitatively. It is evident from this figure that the protrusions on the tip side and on the inlet side of the vane surface exert a most harmful influence upon the pump performance, and, on the other hand, the protrusion on the hub side and the outlet side has a little effect.

The reason for the great difference between the performance of the tip side and that of the hub side roughness is probably due to difference of each relative velocity. In this connection, calculating the mean value of the relative velocities at inlet and outlet of each blade row on the conical developments which divide respectively into two equal areas in each rough surface, it becomes known that the mean relative velocity on the tip side surface is about two times larger than the one on the hub side surface.

The difference in the effect of roughness at the inlet and the outlet side surface is probably due to the difference of each relative velocity (w_1/w_2 at the mean radius is about 1.73) and also due to the difference of the boundary layer thickness at each surface.

The difference in the effect of roughness at pressure surface and at suction one is also explainable by the difference of each relative velocity which is directly related with the pressure difference between the pressure and the suction surface on the impeller vane. Since this pressure difference increases with a decreasing capacity, the effect of roughness at the suction face (where the relative velocity is somewhat larger than at the pressure face) becomes greater than the one at the pressure face as will be seen from Fig. 19, and there is little difference at the maximum discharge where the pressure difference is considerably small. Moreover, since the difference of the relative velocities between the pressure face and the suction face is smaller than the one between the tip side and the hub side of the vane, it may be comprehensible that the performance curves of the pressure or the suction face roughness impeller are situated between the curve of tip side roughness impeller and that of the hub side roughness ones.

From these experiments, then, it may be concluded that the protrusion near the inlet and moreover near the open edge of the impeller vane has a most harmful influence on the pump performance, and the more distant from this place, the less the effect of roughness becomes, thus there is a little effect of the protrusion near the outlet and moreover near the hub of the vane. These results apparently indicate that, in a surface finish work of the impeller vane, it is to be desired to make the surface smooth especially near the inlet of the tip part of a vane, and there is no necessity for making it smooth near the outlet

of the hub part.

6. Conclusion

In this paper, the effects of the impeller vane roughness and the vane thickness on the impeller characteristics, flow conditions and the overall performance of the mixed-flow propeller pump are presented. The effect of changing the local surface finish is also described. The main results obtained can be summarized as follows:

(1) Effects of impeller vane thickness:

(1) The head-capacity, input shaft power and overall efficiency fall with an increasing thickness especially in higher capacity range, but there is little effect of the vane thickness on the performance in lower capacity range where the backflow arises.

(2) The optimum efficiency, and the capacity and the input shaft power at the best efficiency point decrease linearly with an increasing thickness, but the head at this point remains nearly constant, consequently the specific speed also decreases with an increasing thickness.

(3) The reason for this reducing of the performance with an increasing thickness is both the reduction of the input head and the increase of losses in the impeller, and the former has slightly greater effect.

(4) The reduction of the input head with an increasing thickness is due to the increase of deviation angles of the relative discharge flow near the hub of the impeller.

(5) The head-capacity curves by various thickness impellers may be obtained actually by assuming that the capacity varies proportionally

Fig. 19 Effects of local roughness on overall efficiency

to the cross area of flow passage between vanes while the head remains constant.

(II) Effect of whole surface finishes:

(1) The head-capacity curve falls with an increasing roughness, but the input torque or the shaft power-capacity curve remains constant, thus the total efficiency decreases with an increasing roughness.

(2) The vane roughness has a harmful influence upon the characteristics at the best efficiency point and in the higher capacity range, but has little effect at low capacity where the backflow arises.

(3) The capacity and the efficiency at the optimum performance decrease with an increasing roughness, but the change in the head by roughness is comparatively slight, therefore the specific speed also decreases with an increasing roughness.

(4) The pressure rise in the impeller undergoes a considerable change with roughness, although the input head suffers a slight change. Thus the decrease in the head by the roughness is mainly due to the decrease in the pressure rise or the increase in the hydraulic losses in the impeller.

(5) The increase of the losses in the impeller with the roughness is equivalent to that in the flat plate.

(6) The admissible roughness on the vane surface may be roughly estimated by applying the Schlichting's formula concerning that of a flat plate to the blade row on the development of the root mean square radius of the vane.

(7) The meridian component of the absolute velocity at the impeller discharge remains nearly unchanged with the roughness, however its peripheral component suddenly increases with the roughness near the open edge. The reason for this is probably that the boundary layer on the vane surface shifts toward the open edge due to centrifugal action, and then flows out the impeller discharge.

(III) Effects of local surface finishes:

(1) The protrusion near the inlet and moreover near the open edge of the vane has a most

harmful influence upon the performance.

(2) There is little effect of the protrusion near the outlet and moreover near the hub of the impeller.

(3) The protrusion at the suction face of the vane has a somewhat harmful effect than the one at the pressure face.

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