

EXPERIMENTAL STUDY ON PERFORMANCE OF CENTRIFUGAL PUMPS AT LOW REYNOLDS NUMBER

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ABSTRACT

Performance tests of single-stage, single-suction, centrifugal pumps with relatively low specific speed were done over a wide range of Reynolds number, 10^3 to 10^6 . The tests were carried out using seven different impellers combined with two volutes. Five impellers were of radial-vane type in order to reduce disk friction loss and to increase work input. Other two impellers were of open type with six logarithmic-spiral vanes.

The results showed that pump performance deteriorated completely independent of pump structure when Reynolds number decreased to 10^3 . However, choosing proper configuration parameters could materialize the improvement of pump efficiency at middle Reynolds number around 10^4 .

INTRODUCTION

Turbo-pumps have been widely used and are utilized even to transport crude oils with high viscosity. However, only few fundamental or experimental studies have been carried out concerning the performance of pumps when used for the purpose of handling highly viscous liquids. Therefore, even at present, centrifugal pumps for water usage are used for the transportation of highly viscous liquids such as crude oils and pulp slurry. However, there is the huge difference in the flow pattern based on the difference of Reynolds number between the liquids because their viscosity is 1000 times as much as water and unwanted hydraulic losses from the blade-angle mismatch at operation points occur and cause the increase of energy consumption that cannot be disregarded.

Up to now, the decrement in the efficiency when water pumps are being used to transport highly viscous liquids is expected as natural and the prediction on the performance are being emphasized. As a matter of fact, however, it is necessary to design original pumps to meet the needs for the transportation of highly viscous liquids (Tomita et al., 1975).

In the present study a series of model pumps were made and examined experimentally in order to develop pumps that were suitable for highly viscous liquids and also to propose pump designs for better efficiency at operation points in low Reynolds number region. Seven new model impellers were made and consisted of radial vane and logarithmic-spiral vane types. Both of these types were without shrouds for the purpose of reducing rotational disk friction losses. Two types of volute

casings were also made and combined with the impellers in the experiments.

The result of experiments showed that when the Reynolds number fell to 10^3 , the performance of these pumps deteriorated to a level far from practical use without reference to pump configuration. However, the losses could be reduced by considering pump structures around middle Reynolds number region, 10^4 . Also, comparisons were made with the performance of a closed type water pump using a volute casing with the same size outlet as well as the previous reports (Ippen, 1946, Toyokura et al., 1979a, Toyokura et al, 1980b, Okitsu et al., 1979, Aoki et al., 1980).

EXPERIMENTAL APPARATUS AND MEASUREMENT

Experimental Apparatus. Figure 1 shows the schematic of the experimental apparatus. Water of 0.7 m^3 as working liquid was reserved in tank 1 and was circulated by a single-end suction pumps 2, the suction pipe diameter of which was 50 mm. Including the impeller for a water pump there are altogether 8 types and all of them have impeller inlet diameter of 50mm. The outlet diameter of the impellers is classified into two types, 171mm and 140mm. Vane configurations and their outlet angles were also classified into two types, $\beta_2 = 90^\circ$ in radial vanes and $\beta_2 = 23^\circ$ in logarithmic-spiral vanes, respectively. Table 1 shows the data sheet of the applied impeller. Figure 2 shows the examples of cross sectional configurations of the impellers.

Impellers. Only the water pump is closed type with front and back shrouds. 5 types of the test impellers have no shroud and another 2 types have only back shroud. The inside flow of the highly viscous liquids is laminar and therefore a conventional closed impeller with shrouds is out of practical use due to increase of rotational disk friction losses. On the other hand, an open impeller has another friction loss on the casing wall facing blade tips instead of the disk friction loss. This type of friction loss is expected to be less than the disk friction loss. This is the reason why the shrouds of the impellers are removed.

Volute Casings. Table2 shows the configuration data of the applied volute casings. The volute casing No. 0 is for the water pump and used only with the impeller No. 0. The volute casings No.1 and No.2 have 171mm and 140mm in diameter of volute base circle, respectively. The gaps between the impellers and the volute casings are maintained constant by using spacers, excluding the impeller No. 0.

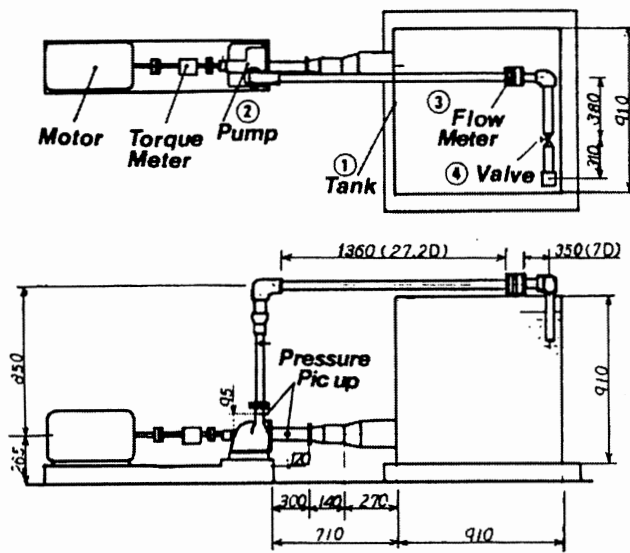


Fig. 1 Schematic of experimental apparatus

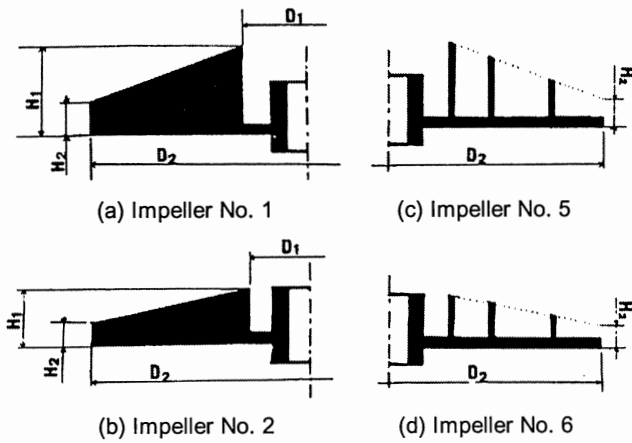


Fig. 2 Examples of impeller cross section

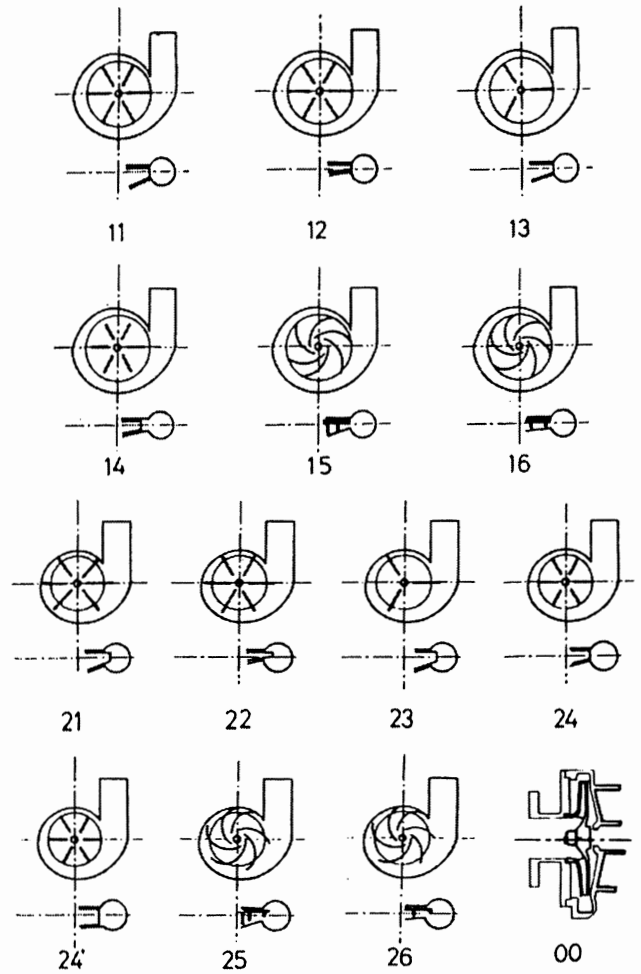


Fig. 3 Pump number and each schematic figure

Table 1 Specifications of test impellers

Impeller number	0	1	2	3	4	4'	5	6
Type of blade (L:Logarithmic,R:Radial)	L	R	R	R	R	R	L	L
Exit angle β_2 (°)	23	90	90	90	90	90	23	23
Existence of front shroud		non	non	non	non	non	non	non
Existence of back shroud		non	non	non	non	non		
Inner diameter of impeller D_1 (mm)	50	50	50	50	50	50	50	50
Outer diameter of impeller D_2 (mm)	171	171	171	171	140	140	171	171
Blade height in impeller inlet H_1 (mm)	9	35	20	35	35	35	35	20
Blade height in impeller exit H_2 (mm)	4	13	11	13	13	31	13	11
Number of blades Z	6	6	6	3	6	6	6	6

Table 2 Specifications of volute casings

Volute number	0	1	2
Sectional configuration	Trapezoid	Circle	Circle
Diameter of base circle D_3 (mm)	171	171	140
Diameter in suction side D_s (mm)	50	50	50
Diameter in delivery side D_d (mm)	40	40	60

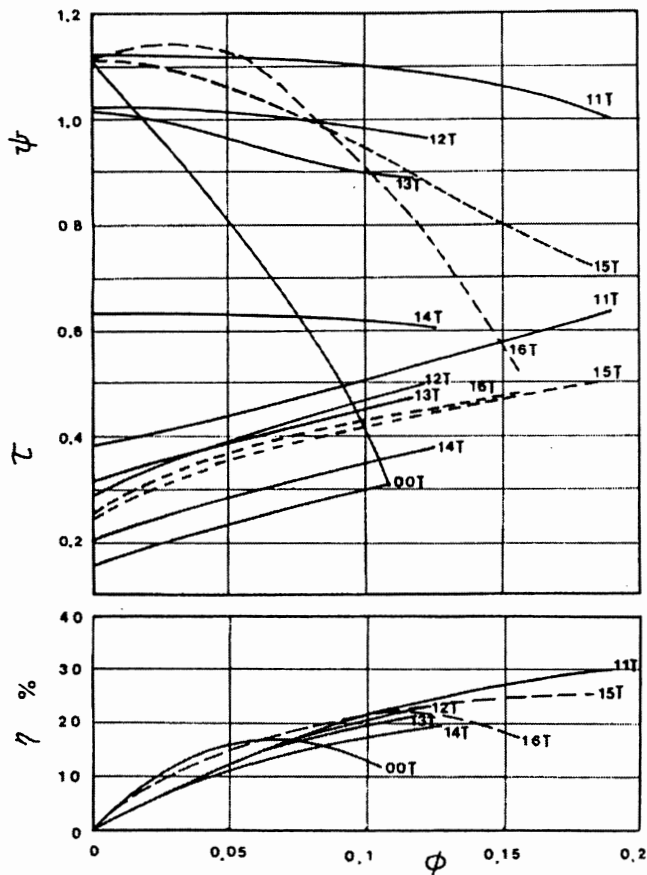


Fig. 4 Characteristics curves of test pumps at $Re = 10^4$ (1)

Pump Types. The respective test pump has two numerals, the first of which denotes the volute casing number and the second the impeller number. Figure 3 shows the sectional view of the combinations and the circle in the figure indicates the volute base circle. Except for the pump type No. 14, the diameter of impeller outlet coincides with that of the volute base circle from the types No. 11 to No. 16, which are called as the free vortex type. On the other hand, the impeller outer part of impeller reaches inside in the volute from the types No. 21 to No. 26 except for the pump type No. 14 because the impeller diameter is larger than the volute base circle. They are called as the forced vortex type. In the pump type No. 24 the impeller diameter coincides with the volute base circle.

Measurement. Experiments have been carried out by keeping rotational speed constant without the occurrence of babbles in the inlet of the pumps. It is also important to keep the oil temperature constant because the oil viscosity is largely dependent on the oil temperature. Before the measurement, we waited for a few hours with the pump operation on until the heat balance was realized between the heat generation by power losses and an electric heater and the heat transfer to the air and the oil temperature reached constant.

The flow control valve played a role of changing the flow rate and total head was measured by pressure difference between discharge and suction. The shaft power P was measured by a torque meter of slip-ring type. The discharge Q was measured by

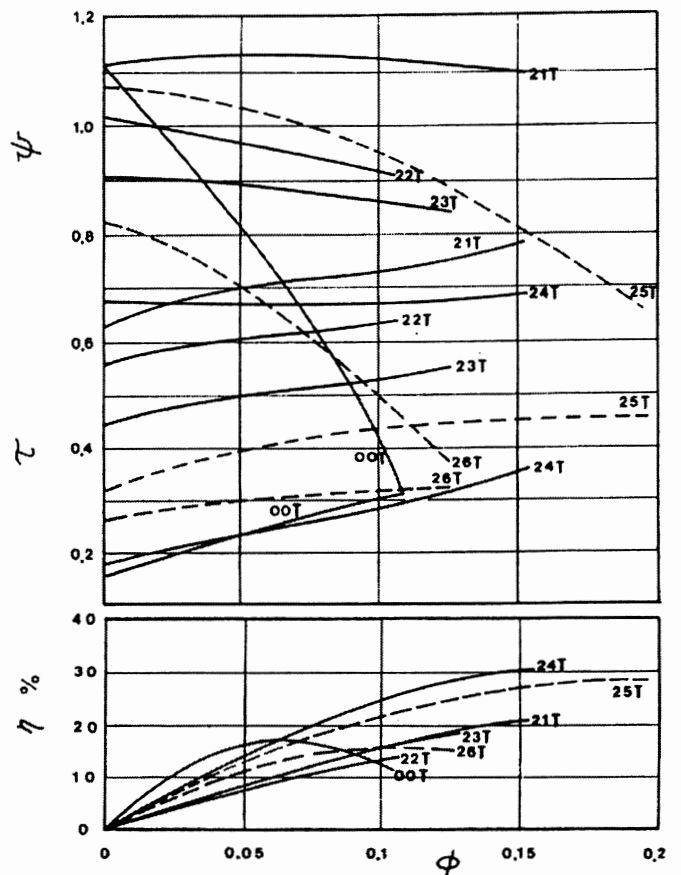


Fig. 5 Characteristic curves of test pumps at $Re = 10^4$ (2)

an orifice for low Reynolds number flow, which was calibrated beforehand.

Clean water, turbine-oil 140 and bright stock oil were used as operative liquids. The experiments were carried out under the condition of pump rotational speed from 1000 to 1400 rpm and the viscosity of the liquids from 1×10^{-4} to 1×10^{-6} m^2/s .

Results of the experiments were summarized in terms of the following non-dimensional parameters.

Flow coefficient:	$\Phi = Q/(A_2 u_2)$
Head coefficient:	$\Psi = H/(u_2^2/2g)$
Input power coefficient:	$\tau = P/(\rho A_2 u_2^3/2)$
Efficiency:	$\eta = \Phi\Psi/\tau$
Reynolds number:	$Re = r_2^2 u_2/\nu$

A_2 : outlet area of the impeller
 g : gravitational acceleration
 r_2 : radius of impeller outlet
 u_2 : peripheral velocity at the impeller outlet
 ν : kinematic viscosity of working liquid
 ρ : density of working liquid

Three letters express the experiment number. They consist of two numerals shown in Fig. 3 and an alphabet, B or T of which denotes the bright stock oil or turbine oil, respectively. For example, No. 24T denotes the data acquired in the experiment with pump No. 24 and turbine oil.

PERFORMANCE TEST RESULTS AROUND $Re = 10^4$

Figures 4 and 5 show characteristic curves of test pumps for turbine oil with the volute casing No. 1 and 2, respectively. The closed water pump No. 00, made of cast-iron and of small size, has 161 (rpm, m^3/min , m) of specific speed at the best efficiency point and the best efficiency is approximately 50% when used in water. Table 3 shows the front and back clearance between the impeller and casing.

Concerning to the total head coefficient in the pump type with the volute No. 1 shown in Fig. 4, the followings are clarified.

Table 3 Gaps between impellers and volute casings at $Re = 10^4$ (Gaps at the impeller outlet in 00T)

Exp. number	11T	12T	13T	14T	15T
Front clearance (mm)	1.6	2.0	1.7	1.3	1.0
Back clearance (mm)	1.6	1.9	1.3	1.6	1.2

16T	21T	22T	23T	24T	25T	26T	00T
1.4	2.0	1.8	2.1	1.6	1.9	2.2	3.8
1.4	2.3	1.9	1.9	1.6	1.8	1.9	11.8

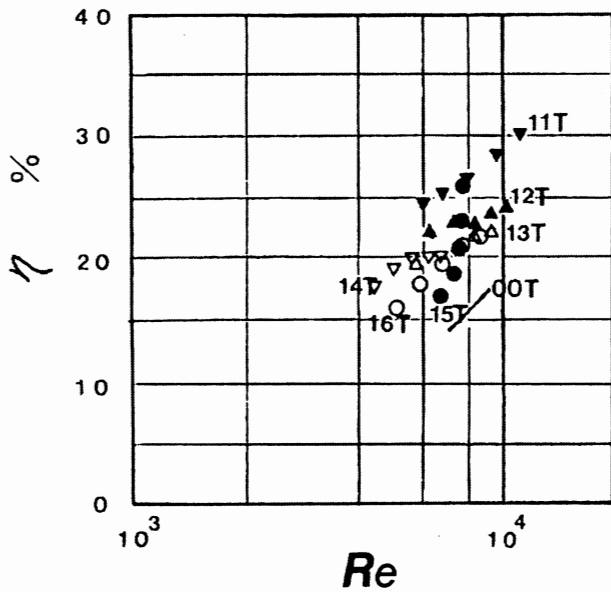


Fig. 6 Influence of Re on η_{max} at $Re = 10^4$ (1)

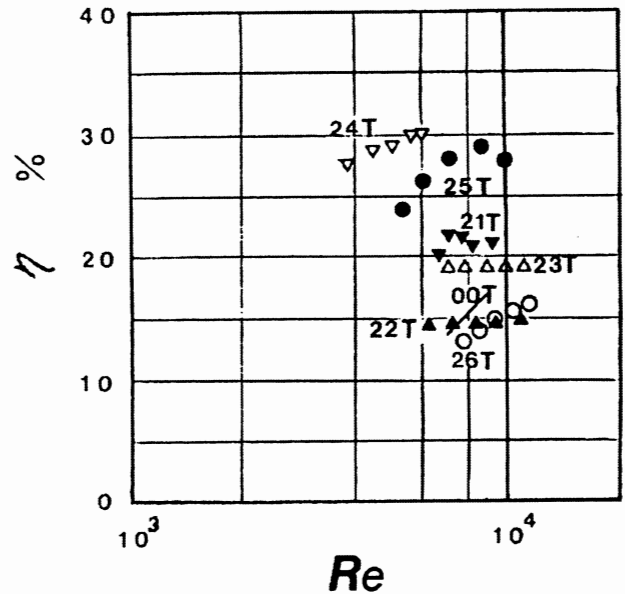


Fig. 7 Influence of Re on η_{max} at $Re = 10^4$ (2)

Table 4 Estimate of the best efficiency with various shape parameters in test pumps at $Re = 10^4$

Exp. number	η_{max}	Blade height (H,L)		Number of blades		Blade type (L:Logarithmic, R:Radial)		Type of volute casing
11T	29.7	⊙	H	○	6	○	R	Free vortex type except 14T
12T	23.2	△	L	×			R	
13T	21.9	△			3	×		
14T	20.0	△						
15T	26.0	○	H	○			L	
16T	21.8	△	L	×			L	
21T	21.2	△	H	○	6	○	R	Forced vortex type except 24T
22T	15.0	×	L	×			R	
23T	19.2	×			3	×		
24T	29.8	⊙						
25T	29.1	○	H	○			L	
26T	16.1	×	L	×			L	
00T	17.0	×						

Table 5 Estimate of the gradient of the best efficiency for Re in test pumps at $Re = 10^4$

Exp. number	$\eta \propto Ro^X$ value of X	Blade height (H,L)		Number of blades		Blade type (L:Logarithmic, R:Radial)		Type of volute casing
11T	0.62	×	H	×	6	×	R	Free vortex type except 14T
12T	0.16	○	L	○			R	
13T	0.28	△			3	○		
14T	0.54	×						
15T	2.05	×	H	×			L	
16T	0.80	×	L	○			L	
21T	0.14	○	H	×	6	×	R	Forced vortex type except 24T
22T	0.07	⊙	L	○			R	
23T	0.01	⊙			3	○		
24T	0.12	○						
25T	0.94	×	H	×			L	
26T	0.63	×	L	○			L	
00T	0.84	×						

Table 6 Gaps between impellers and volute casings at $Re = 10^3$ (Gaps at the impeller outlet in 00T)

Exp. number	11B	24B	24'B	24B-4	00B
Front clearance (mm)	1.6	1.6	0.8	1.2	3.8
Back clearance (mm)	1.6	1.6	1.6	3.0	11.8

Influence of the Blade Configuration. The pump types written by broken lines, which are No. 15T and 16T with logarithmic-spiral vanes, have steeper slope of the characteristic curve than the other pump types, written by solid lines, with radial vanes.

Influence of the Blade Width. The pump types No. 11T and 12T have the radial vanes with different blade width and similarly the types No. 15T and 16T have the logarithmic-spiral vanes with different blade width. Compared the total head among them, the types No. 11T and 15T with larger blade width show higher head than the counterparts No. 12T and 16T, respectively.

Influence of the number of vanes. The pump types No. 11T and 13T have the different number of vanes with the same configuration. The type No. 11T with six vanes shows higher head coefficient than No. 13T with three vanes. These features are also found in the pump type with the volute No. 2 shown in Fig. 5.

Input Power Coefficient. Concerning to the coefficient of input power shown in Figs. 4 and 5, there are little change in increase tendency among these impellers.

Best Efficiency. The best efficiency of the test impellers is more than 18 %, which the conventional closed type pump gives. The flow coefficient at the best efficiency point moves to the region with larger flow coefficient. Especially the pump types No. 11 and 24 obtain about as much as 30% of the best efficiency, which is improved from the conventional closed type pump.

MAXIMUM EFFICIENCY AROUND $RE = 10^4$

Figures 6 and 7 show the maximum efficiency of all pump types when turbine oil was used. An electric heater was used to change the oil temperature and its viscosity and the Reynolds number was changed around 10^4 . In order to estimate the influence of shape parameters on the maximum efficiency, the maximum efficiency and the estimation results are shown in Table 4. There, the estimation results are shown by terms of double circle, circle, triangle and X, which mean excellent, good, not good and poor, respectively. The table also shows the comparison of shape parameters. Which is better for the efficiency in comparison between higher (denoted as H) and lower (denoted as L) blade height, between 6 and 3 in the number of blades, between logarithmic-spiral and radial type in the blade configuration and between free vortex and forced vortex in the volute type.

Maximum Efficiency. The maximum efficiency is realized in the pump type No. 11T and No. 24T. The second best pump type is No. 15T and 25T. The maximum efficiency of the pump

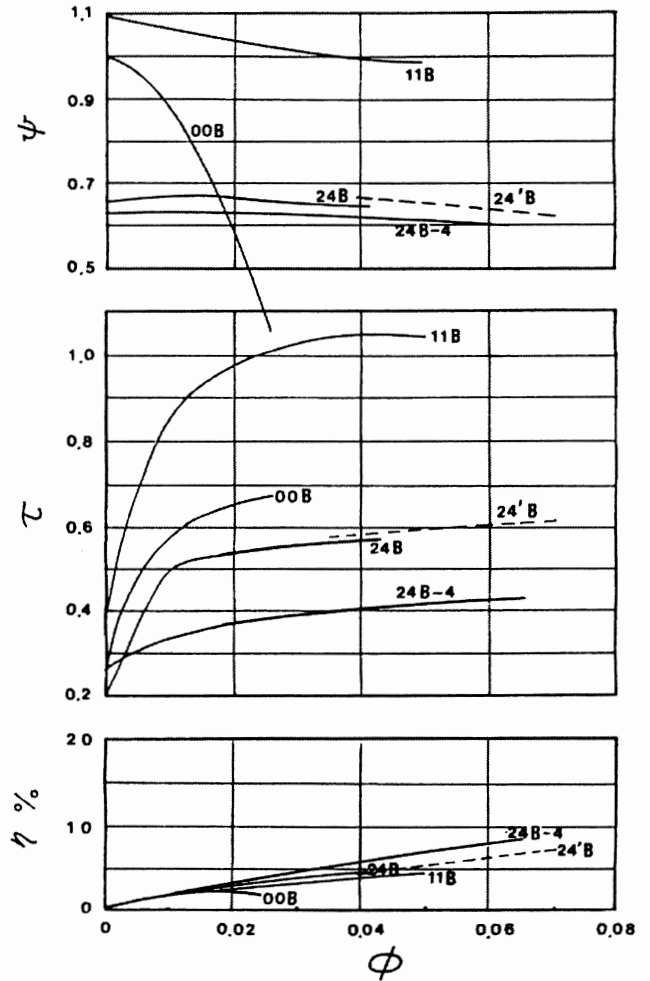


Fig. 8 Characteristic curves of test pumps at $Re = 10^3$

type No. 22T, 23T and 26T is low. Compared between free vortex and forced vortex type, the higher efficiency is realized relatively in free vortex type.

Influence of blade height. In free vortex type, the pump type No. 11T and 12T have the same impeller diameter and the similar radial vane. The pump type with higher blade height No. 11T has higher efficiency than No. 12T. Similarly, between No. 15T and 16T with the similar logarithmic-spiral vane the pump type with higher blade height No. 15T has higher efficiency. Also compared between No. 21T and 22T, or between No. 25T and 26T, the pump type with higher blade height has higher efficiency even in forced vortex type.

Influence of Blade number. Compared between No. 11T and 13T, or between No. 21T and 23T, the pump type with 6 blades has higher efficiency than that with 3 blades not only in free vortex type but also forced vortex type.

Influence of Blade type. Compared between No. 11T and 15T, or between No. 12T and 16T in free vortex type, the pump type with radial vane has higher efficiency than that with logarithmic-spiral vane. On the other hand, it is interesting that forced vortex type has the quite inverse tendency.

As a whole, free vortex type pumps exceed in the best

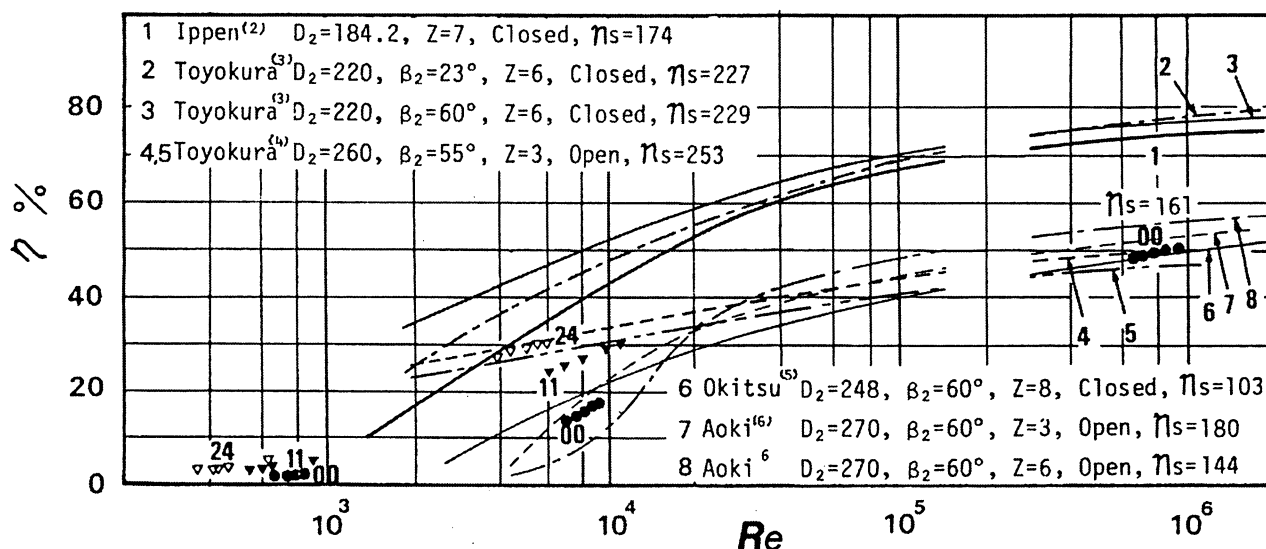


Fig. 9 Influence of Reynolds number on the best efficiency in various pumps

efficiency than forced vortex type pumps. Pumps with higher blade width show better efficiency than that with lower blade width and pumps with six blades have better efficiency than that with three blades. Integrating these results, making input power from an impeller larger is good for getting better efficiency in designing a pump operating around $Re = 10^4$.

GRADIENT OF MAXIMUM EFFICIENCY AROUND $Re = 10^4$

Table 5 shows another estimation on the deterioration rate of the best efficiency accompanied by decrement in Reynolds number. When the gradient of the best efficiency is supposed to be proportional to Re , the exponent is given from Figs. 6 and 7. In order to estimate the influence of shape parameters on the gradient of the maximum efficiency, the estimation results are shown by the same way as Table 4.

Gradient of Maximum Efficiency. Because the gradient of the maximum efficiency means the deterioration rate with the decrement of Re , the lower gradient indicates the better characteristics. Compared between free vortex and forced vortex type, the smaller exponent is realized relatively in forced vortex type.

Influence of blade height. In both free vortex and forced vortex type, the pump type with lower blade height has lower gradient.

Influence of Blade number. In both free vortex and forced vortex type, the pump type with 3 blades has lower gradient than that with 6 blades.

Influence of Blade type. In both free vortex and forced vortex type, the pump type with radial vane has lower gradient than that with logarithmic-spiral vane

As a whole, pumps with impeller of lower blade width, three blades or radial blades show less efficiency drops independent of volute casing configurations and types of pumps in the decrease of Reynolds number. Among the impellers with

radial vane, forced vortex type pumps have slight decrease in pump head and little influence of Reynolds number on efficiency.

The reasons are considered as follows. Generally, the averaged flow velocity in volute casings depends on the flow rate of oil. As Reynolds number decreases, the flow rate at the best efficiency point also decreases, leading to decrease of the averaged flow velocity in volute casings. On the other hand, as the oil goes out of the impeller with relatively very high absolute velocity the difference of velocity between them causes the mixing losses, which increases with decrease of the Reynolds number. In forced vortex type pumps, the tip of impeller blade pushes oils in the volute casing in direction of rotation to minimize the above difference in velocity, leading to reducing the mixing loss. Moreover, in free vortex type volute casing a secondary flow is induced in the surface perpendicular to the averaged flow direction, although forced vortex type pumps restrains the secondary flow due to the effect of mixture near the tip of the blade.

PERFORMANCE TEST RESULTS AROUND $Re = 10^3$

Free vortex type pumps No. 00, No. 11, No. 24 and No. 24' are subjected to the experiments of bright stock oil. Pumps No. 11 and No. 24 have better efficiency in the experiments of turbine oil around $Re = 10^4$. Pump No. 24' has the blade width changed from the pump No. 24. Table 6 shows the gaps between impellers and casing walls. Figure 8 shows the characteristic curves. Although all of the test pumps attain higher efficiency than the conventional closed type pump in the larger flow operation, those are out of practical use due to less than 10% of efficiency.

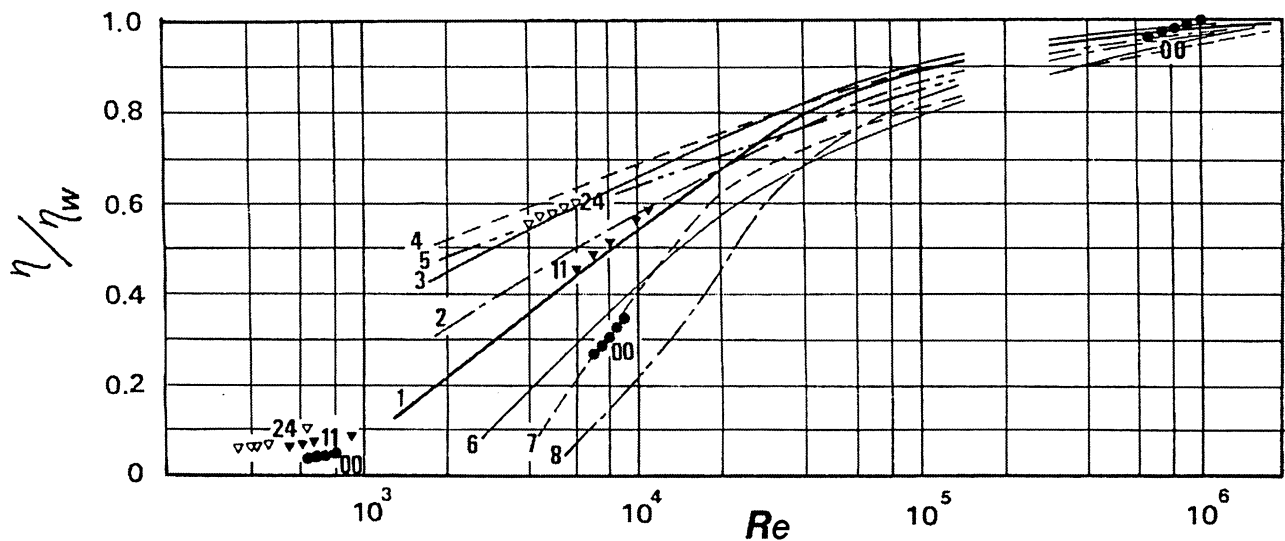


Fig. 10 Influence of Reynolds number on deterioration ratio of the best efficiency

INFLUENCE OF REYNOLDS NUMBER ON BEST EFFICIENCY

Figure 9 shows the influence of Reynolds number on the best efficiency for pumps No. 00, No. 11 and No. 24 as well as the pumps reported in the references. In the figure, the value of Ns denotes the specific speed when used in water. The ordinary water pumps No. 00, which belongs to the category of pumps with lowest specific speed, has only as much as 50 % of efficiency in water around $Re = 10^6$. The pump No. 11 and No. 24 has 53 % and 50 % of efficiency in water, respectively. The pump No. 24 shows improvement in efficiency at the region less than $Re = 10^4$.

As the specific speed of pumps increases at low Reynolds number region, the deterioration rate of efficiency decreases because the influence of the boundary layer decreases relatively in the flow passage of the impeller. These facts indicate that the efficiency is a function of Reynolds number and specific speed of pumps.

Figure 10 shows the deterioration rate of the best efficiency at each Reynolds number against the best efficiency in water. Because the disk friction loss is much dependent on the impeller diameter, the pump No. 24 with smaller impeller diameter brings less deterioration rate of efficiency, which shows clearly the improvement in efficiency.

CONCLUSION

According to the experimental study on the performance of pumps when used in water, turbine oil and bright stock oil as test liquids. Around $Re = 10^4$, impellers with radial vanes have less deterioration in efficiency than impellers with logarithmic-spiral vanes with decrease of Reynolds number. Increasing impellers blade width has higher best efficiency at higher flow rate. The impellers with six vanes show higher efficiency than impellers with three blades. The free vortex type pumps show higher

efficiency than the forced vortex type pumps. As a whole, compared with closed impellers, open impellers can reduce losses.

Around $Re = 10^3$, although the test impellers show higher efficiency than the ordinary closed impeller, all types of the impellers have no more than 10 % of efficiency. In the range of $Re = 1000 \sim 2000$, the efficiency decreases drastically. In this region the centrifugal pumps dealing with highly viscous liquids show the limitation of improvement in performance.

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