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Developmental Study of Impulse Type Oil Turbine*

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To take out mechanical power from high pressure oil, an impulse type oil turbine of a construction similar to the Terry steam turbine was developed on the basis of the fundamental experiment and the study of the performance of a test turbine.

The turbine mainly consists of a nozzle, a turbine wheel with many deflection grooves and several fixed deflection boxes. Comparing with a positive-displacement type oil motor, the oil turbine has such features as feasibility of high rotational speed, low idling resistance, simple construction etc.

As the fundamental experiment, a static test of two-dimensional deflectors was carried out and design data for a deflector with minimum losses were obtained.

Based on the data obtained from the above, a test turbine was built, and its performance was measured at different rotational speeds and also at different jet velocities. From this study the limit of performance of an impulse type oil turbine was clarified, and the necessary data for designing an efficient oil turbine were obtained.

1. Introduction

To take out mechanical power from a high pressure oil source of relatively small quantity, a positive-displacement type oil motor is widely used today, owing to its remarkable development in recent years. However, we may also utilize an impulse type oil turbine for the same purpose as the oil motor.

This paper intends to make clear the characteristics of such impulse oil turbine on the basis of a fundamental experiment and of the study on the performance of a test turbine.

Comparing with a positive-displacement type oil motor, the following features can be expected from an oil turbine:

- 1) Feasibility of high rotational speed,
 - 2) Small energy losses in the case when the output shaft rotates idly without a supply of working oil,
 - 3) Simple construction and easy manufacture.
- These characteristics, especially 1) and 2) of the above, are the inherent property of oil turbine and can not be expected from an oil motor.

The quality of an oil turbine is judged mainly through the values of stall torque and maximum efficiency. Since the improvements of stall torque

and maximum efficiency are inconsistent with each other, as we know from the experience on torque converter, we are forced to put emphasis on one of these two factors. In this paper the priority of development is put on the improvement of stall torque, supposing the application such as an auxiliary prime mover used only for helping start.

The investigated impulse oil turbine has a construction similar to the Terry steam turbine, which is utilized usually for a small auxiliary steam turbine. As illustrated in Fig. 1 a Terry turbine consists, in principle, of a nozzle, a turbine wheel with many deflection grooves on its outer periphery (rotor deflector) and several deflection boxes fixed to the casing (stator deflector).

The oil jet from the nozzle goes and returns several times between rotor and stator deflectors.

Fig. 1 Construction of Terry type impulse turbine

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The reaction force acting on rotor deflectors, caused by the momentum change of the oil jet, generates the output torque of the turbine wheel.

Since the Terry steam turbine has been usually used only for a small auxiliary steam turbine, it has not been studied intensively. We can find only some qualitative descriptions⁽¹⁾ about the operation and inner flow, while there have been no quantitative informations necessary for the design of an oil turbine.

2. Stationary test of two-dimensional model deflectors

2.1 Purpose of test

The output torque of a Terry type impulse oil turbine is generated by the momentum change of high speed oil jet while deflecting 180° in rotor deflectors. The oil jet which has once generated a reaction force in a certain rotor deflector, reenters the second rotor deflector after making 180° deflection in the stator deflector, and it gives an additional reaction force. Therefore, the form of a stator deflector should be so determined that the second deflection may take place in the rotor deflector adjacent to the one, in which the first deflection has taken place. Thus the mixing of oil jets with different velocities can be avoided.

If we take the case where the turbine wheel stands still and no energy losses occur while deflecting, the oil jet retains its initial velocity in spite of deflections, and the output torque increases as the repetition of deflection in a rotor deflector. We may even imagine an infinitely large stall torque if infinitely many stator deflectors are provided.

From the above consideration the greatest emphasis should be put on the development of efficient deflectors for realizing a large stall torque as well as high efficiency of the oil turbine.

The forms of actual rotor and stator deflectors can not be determined from only a hydrodynamic

view point, because they are restricted also by several constructional and manufactural conditions. The actual forms of both deflectors, therefore, can not be so simple and become inevitably three-dimensional; especially the form of a stator deflector is complicated for the reason as stated before.

In this research the actual forms of deflectors have been simplified by two-dimensional model deflectors having parallel side walls, and the performance test has been made on these model deflectors instead of on the actual deflectors, in order to obtain fundamental data for designing efficient deflectors.

2.2 Performance test on model deflectors

Twenty-two model deflectors made of brass plate of uniform thickness t (Fig. 2) are prepared for the test, and they have different groove forms as listed in Table 1. Seventeen of them (No. 1~No. 17) have a simple groove form, that is, a half-circle with radius r ; four of them (No. 18~No. 21) have a groove form composed of two quarter-circles with different radii, r_{inlet} and r_{outlet} ; the last one (No. 22) has a parabolic groove form. Regarding the seventeen deflectors with a simple groove form, radius r of bottom circle, length h of parallel part and thickness t are so combined that their optimal values may be independently determined from the test result. The oil jet shot out from the nozzle has a diameter of $d_j = 4.57 \text{ mm}$ and enters the test deflector with the relative position as illustrated in Fig. 2.

The center of inlet jet lies on the middle plane in thickness direction, except the case to be mentioned later on. The side wall of a rotor deflector installed in an actual Terry steam turbine

Fig. 2 Two-dimensional model deflector

Table 1 Dimensions of two-dimensional model deflectors

No.	t mm	h mm	r mm	No.	t mm	h mm	r mm
1	5	0	10	14	5	0	20
2	7	0	10	15	7	0	20
3	5	5	10	16	5	10	20
4	5	0	15	17	7	10	20
5	6	0	15	No.	t mm	r_{inlet} mm	r_{outlet} mm
6	7	0	15	18	6	11	19
7	10	0	15	19	6	19	11
8	5	5	15	20	6	13	17
9	6	5	15	21	6	17	13
10	7	5	15	No.	t mm	Form	
11	10	5	15	22	6	Parabola	
12	5	10	15				
13	7	10	15				

has a circular notch, which may probably serve to prevent an overdeflection of oil jet caused by the vacuum space enclosed with oil jet and side walls⁽²⁾. In order to ascertain this effect, various kinds of side walls are prepared as listed in Table 2. These walls are made of a transparent acrylic glass for the visual observation of inner flow. A combination of certain deflector and side wall is indicated by number and symbol in a series, namely the number of deflector (Table 1) and the symbol of side wall (Table 2) such as 10-A.

The performance test of deflectors has been carried out by using the equipment illustrated in Fig. 3. The working oil (cf. Fig. 12) pressurized up to 10~50 kg/cm² by a vane type oil pump is shot out from the nozzle with jet velocity $v_j = 40 \sim 90$ m/sec and deflects in the test deflector. The reaction force F acting on the deflector is measured by strain gauges stuck on a thin duralumin cylinder which supports the deflector. Four strain gauges for axial strain and four dummy strain gauges for tangential strain are combined to compose a bridge, which gives an indication only proportional to the axial force, compensating for the effect of bending force and temperature difference.

The quality of deflector is indicated by the reaction coefficient f defined by

$$f = F/2F_0 \quad (1)$$

Where,

F : reaction force of deflector,

F_0 : reaction force of sufficiently large flat plate, in the case when oil jet runs perpendicularly against the plate,

G : weight flow of working oil.

In an ideal case that oil jet deflects accurately 180° without any energy losses, the reaction coefficient

becomes unity.

The reaction force of flat plate F_0 can be determined also by using the relation $F_0 = (G/g)v_j = G^2/g\gamma_j A_j$ instead of by direct measurement, where γ_j ; specific weight of working oil at nozzle exit, $A_j = (\pi/4)d_j^2$; nozzle area. The weight flow G of the above formula can be measured with sufficient accuracy by using a weighing tank, while the specific weight γ_j at nozzle exit can not be determined accurately because of the following reason:

While an oil jet deflects in a groove, a small part of oil jet spreads near the deflector forming tiny oil drops (atomization). Thus, the working oil mixes strongly with air and takes in air bubbles, so that the oil becomes emulsive immediately after the beginning of oil circulation. The specific weight of the oil decreases from its initial value until the absorption and separation of air bubbles get balanced. Since the emulsive working oil is, to some extent, compressible, its specific weight varies with the pressure. Analysis of preliminary experiment shows that the specific weight at nozzle exit is still larger than the value at atmospheric condition, and that oil jet can not have expanded thoroughly to the atmospheric pressure during the short acceleration time in the nozzle.

As explained above, the specific weight at nozzle exit does not agree with the value at atmospheric condition, and can not be estimated accurately. Accordingly, the reaction force acting on a flat plate has been directly measured and used as the basis of reaction coefficient. The working oil circulates from the oil pump to the test section, to the oil tank with a cooler, and then back to the oil pump. The oil temperature before nozzle

Table 2 Dimensions of notch on side wall

Symb.	h mm	r mm
A	no notch	
B	0	4
C	5	4
D	0	6
E	5	6
F	0	9
G	5	9

Fig. 3 Test equipment

Fig. 4 Performance of two-dimensional model deflectors

has been set between 23° and 30°C, corresponding to the kinematic viscosity $\nu = 8.5 \sim 5.5 \times 10^{-5} \text{ m}^2/\text{sec}$ (mean value; $\bar{\nu} = 7 \times 10^{-5} \text{ m}^2/\text{sec}$).

2.3 Result of experiment

In Fig. 4 the reaction coefficient of four representative model deflectors is plotted against the jet velocity on the abscissa. The reaction coefficient varies slightly as the jet velocity changes, although there is no general relationship between them. In the range of jet velocity $v_j = 40 \sim 90 \text{ m/sec}$, the Reynolds number $Re = v_j d_j / \bar{\nu}$ falls within a relatively narrow range of $2.5 \sim 6 \times 10^3$.

If we assume that the flow in a deflector is influenced only by the viscous effect, the reaction coefficient must vary only with the Reynolds number, and thus there must exist some common tendency between them.

From the above comparison it becomes clear that there is another factor besides the viscous effect giving influence on the flow in deflector, that is, the effect of atomization of oil jet. This atomization causes naturally a decrease of reaction coefficient. The intensity of atomization is affected mainly by the surface tension of oil and by the intensity of the secondary flow of oil jet which depends upon the form of deflection groove. Therefore, the effect of atomization appears in different ways for each deflector. The above circumstance explains the reason why there is no common tendency between reaction coefficient and jet velocity.

It should be added that the result obtained from the present experiment, strictly speaking, can not be applied to the flow of other working fluid than oil, even if the Reynolds number of both flows coincides.

Performance tests have been done on the

Table 3 Reaction coefficient f of two-dimensional model deflectors

Type	f	Type	f	Type	f
1-A	0.80	3-B	0.855	13-E	0.865
4-A	0.85	5-B	0.825	15-E	0.87
5-A	0.82			18-E	0.88
6-A	0.86	5-C	0.835	19-E	0.88
9-A	0.845	9-C	0.835	20-E	0.88
10-A	0.875			21-E	0.88
12-A	0.875	5-D	0.86	22-E	0.89
13-A	0.86	6-D	0.86		
15-A	0.875	7-D	0.85	5-F	0.83
17-A	0.84	9-D	0.875	9-F	0.885
18-A	0.88	15-D	0.87	14-F	0.875
19-A	0.885			15-F	0.86
20-A	0.87	5-E	0.825		
21-A	0.88	8-E	0.85	9-G	0.88
22-A	0.88	9-E	0.88	15-G	0.87
		10-E	0.875	16-G	0.85
		11-E	0.85	17-G	0.835

combinations of deflector and side wall listed in Table 3, in which the maximum reaction coefficients obtained in the test range are also given.

In the test the center of inlet jet is set on the middle plane in thickness direction. The relative position between inlet jet and rotor deflectors of actual turbine, however, changes every moment with the rotation of wheel. In order to make clear the effect of relative position on reaction coefficient, further experiment has been carried out by using deflectors 8-A, 9-A, 10-A, 11-A (thickness $t = 5, 6, 7, 10 \text{ mm}$), and the result is shown in Fig. 5. So long as the whole part of inlet jet enters the groove, in other words the inlet jet does not touch the side wall directly, the reaction coefficient remains almost constant.

2.4 Analysis of result

The dimensions of model deflectors have been so chosen that the optimal radius of bottom circle, the optimal length of parallel part, the optimal thickness, and the optimal arrangement of side wall can be all determined from the test result. However, it seems to be almost impossible to make such clear and quantitative conclusion from the result shown in Table 3, since the reaction coefficient of deflectors with somewhat satisfactory quality, falls within a narrow range comparable to the accuracy of experiment.

With the initial intention given up, some remarks for designing efficient deflectors are given hereafter based on the common characteristics of comparatively efficient deflectors such as 18-A, 19-A, 21-A, 22-A, 9-E, 18-E, 19-E, 20-E, 21-E, 22-E, 9-F and 9-G:

Fig. 5 Effect of relative position of inlet jet and deflector on reaction coefficient

1) Radius of bottom circle should be $r=(3\sim 4)d_J$. Too small radius causes an increase of secondary flow losses, while too large radius causes an increase of frictional losses.

2) Parallel part in entrance and exit region is effective. The optimal length is about $h=d_J$. This parallel part not only improves the reaction coefficient of rotor deflector itself, but suppresses the atomization of the outlet flow and keeps the inlet flow to the following stator deflector in good condition.

3) Thickness of two-dimensional deflector should be $t=(1.5\sim 2)d_J$.

4) Notch form of side wall gives no remarkable influence on the reaction coefficient. However, the photographic observation has demonstrated that the oil jet slightly overdeflects when there is no notch on the side wall (side wall A)—as to be expected. When the notch is too large, some of the oil overflows outside, resulting naturally in a decrease of reaction coefficient.

5) Groove form composed of two quarter-circles (No.18~21) and of parabola (No.22) has achieved no noticeable improvement of reaction coefficient, comparing with adequate half-circle.

6) By sharpening the entrance edge of side wall, the undesirable interaction between inlet oil jet and side wall can be essentially avoided. Therefore, the reaction coefficient of an adequately designed rotor deflector is expectedly very close to the value obtained from a model deflector.

2.5 Analysis of losses

While deflecting 180° in a groove there arise energy losses corresponding to the head loss of $h_t=\zeta(v_J^2/2g)$. If we neglect simply the energy losses caused by the atomization of oil jet, the losses should be equal to the difference of kinetic energy of inlet and outlet jet, thus

$$\frac{v_J^2 - v_J'^2}{2g} = \zeta \frac{v_J^2}{2g} \quad \dots\dots\dots (2)$$

where v_J' indicates the mean velocity of outlet jet.

Under the assumption that oil jet deflects accurately 180° , the reaction coefficient f can be

written

$$f = \frac{G(v_J + v_J')/g}{2Gv_J/g} = \frac{v_J + v_J'}{2v_J} = \frac{1}{2}(1 + \sqrt{1 - \zeta}) \quad \dots\dots\dots (3)$$

Since the flow phenomena in a deflector are quite complicated, it is scarcely possible to calculate the accompanying losses with sufficient accuracy. Here, an approximate value of losses will be estimated on the basis of losses in bent pipes, on which many investigations have been made.

For this purpose the loss formula presented by H. Ito⁽³⁾ has been directly applied, and the following formula is given:

$$\zeta = 0.00241\alpha\theta R_e^{-0.17} \left(\frac{R}{a} \right)^{0.84} + \lambda \frac{h}{d_{eq}} \quad \left\{ \dots\dots (4) \right.$$

$$R_e(a/R)^2 > 91$$

The above contains an additional term $\lambda(h/d_{eq})$, which is to take account of losses on the parallel part in exit region. The constant α of the above formula is calculated by $\alpha = 1 + 116(R/a)^{-4.52}$ in the case when the deflection angle θ equals to 180° . Moreover, $R_e = (v_J d_{eq}/\nu)$, $d_{eq} = \pi d_J^2 / (t + 2H)$, $H = \pi d_J^2 / 4t$, $R = r - H/2$ and $a = d_{eq}/2$. Friction coefficient λ of parallel part is assumed to be $\lambda = 0.04$ based on the Reynolds number.

Regarding the five representative model deflectors, theoretical values of reaction coefficient have been calculated using Eqs. (3) and (4), and are compared with the corresponding test result (side wall A) as shown in Table 4. Although the agreement of theory with experiment is not so satisfactory, the theoretical result suggests that the experimental result such as $f=0.89$ by 22-E, $f=0.885$ by 19-A and 9-F, corresponds almost to the upper limit of efficiency, and that further noticeable improvement of reaction coefficient is

Table 4 Comparison of theoretical and experimental results

No. of deflector	f : theory	f : experiment (side wall A)
1	0.876	0.80
4	0.890	0.85
9	0.878	0.845
15	0.859	0.85
17	0.834	0.84

$v_J \approx 90$ m/sec, $\nu \approx 7 \times 10^{-5}$ m²/sec

Fig. 6 Rotor deflector

rather hopeless.

3. Performance test of oil turbine

3.1 Test rig

Based on the result obtained from the stationary test of model deflectors, a research oil turbine has been built for the study of performance. The rotor deflector of the turbine has a form similar to that of No.9 model deflector as illustrated in Fig. 6. The edge of the side wall on entrance side is cut off sharply, so that the undesirable interaction between inlet jet and side wall can be avoided as far as possible. Distance between the nozzle exit and the outer periphery of turbine wheel is set as short as possible, in order to let the inlet oil jet enter grooves retaining its original circular cross section.

The form of stator deflector is illustrated in Fig. 7; it has been designed so that the most part of deflected flow from a rotor deflector can be collected by the stator deflector, and that the collected flow can reenter the next rotor deflector possibly as a concentrated uniform jet. Number of stator deflectors fixed to the casing is easily adjustable.

The casing has a sufficiently large space compared with the turbine wheel itself, so that the resistance torque due to the interaction between floating oil drops and turbine wheel is reduced as far as possible.

The arrangement of test rig is illustrated in Fig. 8. The output torque generated by the turbine wheel is measured by a strain gauge type torque

indicator. The output power is absorbed finally by a Prony dynamometer. Between turbine shaft and Prony dynamometer a continuous speed change gear is installed to enlarge the measurable speed range of the test turbine.

In the experiment output torque T , the weight flow G of working oil, oil temperature before nozzle, specific weight of working oil sampled from the return pipe, etc. have been measured at different rotational speeds and at different oil pressures before nozzle.

3.2 Test result

Characteristics of oil turbine have been indicated by non-dimensional parameters, such as velocity coefficient φ , torque coefficient τ , and efficiency η as follows:

$$\varphi = \frac{u \cos \theta}{v_j} = \frac{\omega D \cos \theta}{2v_j} \quad \dots\dots\dots (5)$$

$$\tau = \frac{T}{G v_j D \cos \theta / g} \quad \dots\dots\dots (6)$$

$$\eta = \frac{\omega T}{G v_j^2 / 2g} = 4\varphi\tau \quad \dots\dots\dots (7)$$

(compare Fig. 6 regarding symbols)

Non-dimensional characteristics of oil turbine change with the number Z of stator deflectors. The test has been undertaken in case of $Z=0, 1, 2, 3, 5$ and 7 . In order to clarify the effect of jet velocity on non-dimensional characteristics, the oil pressure before nozzle has been set at 40, 30, 20 kg/cm², corresponding to the jet velocity $v_j=80, 68, 55$ m/sec. Since the stall Torque T_s varies slightly with the relative position between inlet jet and side wall of rotor deflector, the maximum value obtained in the test has been adopted as

Fig. 7 Stator deflector (inside dimension)

Fig. 8 Arrangement of test rig

representative.

Non-dimensional characteristics determined from the test are shown in Fig. 9(a) for $Z=0$, (b) for $Z=1$, (c) for $Z=2$, (d) for $Z=3$, and (e) for $Z=7$. As the result at $Z=5$ agrees almost with that at $Z=7$, the corresponding figure is omitted in the above.

3.3 Analysis of test result

At first the result without stator deflector ($Z=0$) is analysed. The mean value of stall torque coefficient $\tau_s (=gT_s/Gv_J D \cos \theta)$ obtained from the test is 0.86, and it agrees nearly with the deflection coefficient f of corresponding model deflector, namely, $f=0.845$ with 9-A, 0.835 with 9-C, 0.875 with 9-D, 0.88 with 9-E, 0.885 with 9-F, 0.88 with 9-G.

Theoretical hydraulic torque coefficient $(\tau_h)_{th}$ is calculated from $(\tau_h)_{th}=f[(w_J)_{eff}/v_J]$ based on the assumption that the component of relative jet velocity parallel to the side wall, $(w_J)_{eff}$, is effective for the generation of reaction force (cf. Fig. 10). This theoretical hydraulic torque coefficient and the measured hydraulic torque coefficient $(\tau_h)_{exp}(=gT_h/Gv_J D \cos \theta)$ are plotted in Fig. 10 against velocity coefficient ϕ on the abscissa, where

T_h indicates the net output torque of turbine wheel, that is, the output torque at shaft end plus resistance torque. The resistance torque, which contains both mechanical friction torque of bearings and windage torque of wheel, has been calculated on the basis of measured attenuation of rotational speed, in the case when the wheel rotates freely without the supply of working oil.

Judging from the good agreement between theoretical and experimental results in the range of low rotational speed, as seen in Fig. 10, it can be

Fig. 9 Nondimensional characteristics of test oil turbine

concluded that the reaction coefficient is scarcely affected by the rotation of wheel.

As the rotational speed increases, the interference between wheel and floating oil drops gets stronger, and the resistance torque increases, so that the agreement between theory and experiment becomes worse. Especially when the weight flow of working oil exceeds a certain limit, the wheel begins to scrape out directly the foamy drain oil accumulated in the bottom of casing, and the resistance torque increases remarkably.

The influence of jet velocity on the hydraulic performance of turbine seems to be very small like in the case of a two-dimensional model deflector.

In the case of $Z=1$ and 2, however, the difference of characteristics due to jet velocity is clearly observed. For both stall torque coefficient and maximum efficiency, the jet velocity $v_j=68$ m/sec has given the best result.

As the number of stator deflectors increases, there appears a bend on the characteristic curve, and this is the very evidence of the effectiveness of stator deflectors.

Fig.11 shows how the stall torque coefficient τ_s , maximum efficiency $\eta_{\max}$, and maximum hydraulic efficiency $\eta_{h\max}=(2g\omega T_h/Gv_j^2)_{\max}$ change with the number Z of stator deflectors. For stall torque coefficient the case of $Z=2$ is optimal, and has realized $\tau_s=1.08$ at $v_j=68$ m/sec, $\tau_s=1.03$ as the mean value in test range. Regarding the maximum efficiency and maximum hydraulic efficiency, only one stator deflector is most effective. In this case ($Z=1$) $\eta_{\max}=0.73$ and $\eta_{h\max}=0.84$ have been obtained as the mean value in test range.

As seen from Fig.11, excessive stator deflectors get both stall torque coefficient and maximum

efficiency rather worse. However, the braking effect attributable to the excessive stator deflectors is not so essential. The visual observation of flow has demonstrated that the oil jet flows along the expected helical path between rotor and stator deflectors, only when the number of stator deflectors is less than two. If the number exceeds the limit, namely $Z \geq 3$, the third stator deflector is filled with stagnated oil and oil jet can not flow further, so that the excessive stator deflectors work only as resistance.

If we assume that both rotor and stator deflectors have the same reaction coefficient, $f_R=f_S=0.86$, the theoretical stall torque coefficient can be calculated by

$$(\tau_s)_{th}=f_R \sum_{n=0}^Z (2f_R-1)^n (2f_S-1)^n \dots \dots \dots (8)$$

The above theoretical result is plotted in Fig.11 as double dot chain line, and is essentially larger than the experimental result. This informs us that the effective reaction coefficient of stator deflector, namely (collection efficiency of stator deflector) $\times$ (reaction coefficient of collected flow in stator deflector), is considerably worse than that of rotor deflector.

If we assume $f_R=0.86$ for rotor deflector, $f_S=0.62$ for stator deflector, the theoretical result calculated by Eq. (8) (chain line in Fig. 11) agrees quantitatively with the test result. This agreement shows inversely that the effective reaction coefficient of stator deflector takes approximately the above assumed value, and that the collection efficiency of stator deflector is nearly 72% ($=f_S/f_R$).

Since we can not expect further improvement of reaction coefficient of rotor deflector, the key point for improving the performance of a Terry type impulse oil turbine is the development of

Fig.10 Comparison of theory and experiment

Fig.11 Change of performance as to number of stator deflectors

stator deflector.

The stator deflector used in the performance test (cf. Fig. 7) has a thickness which is nearly equal to the thickness of rotor deflector. In order to improve the collection efficiency of stator deflector, the performance test has been also done by adopting a larger stator deflector, whose thickness is almost twice that of rotor deflector. However, this modification has resulted only in a deterioration of performance.

When we consider only the increase of stall torque, the direction of side wall of rotor deflector should be set parallel to the absolute direction of inlet oil jet. Such rotor deflector, however, can not achieve a satisfactory performance near the point of maximum efficiency, because excessive losses occur due to the attack angle of the relative inlet jet to the side wall. Thus, the increase of stall torque coefficient and that of maximum efficiency become mutually inconsistent. In order to design an oil turbine for a specific purpose, the inclination angle of side wall of rotor deflector and also the number of stator deflectors must be determined corresponding to the intended purpose.

The working oil begins to take in air bubbles and to get emulsive immediately after the start of operation. The anti-emulsion agent contained in the working oil can not even prevent the emulsification against the strong mixing action of high speed oil jet. In order to realize design output torque, it is, therefore, necessary to take account of the decrease of specific weight caused by emulsification (10~15% at atmospheric condition, 3~8% at nozzle exit). Fig. 12 shows the property of working oil used in the experiment.

Only the turbine characteristics after the nozzle exit have been investigated in the present study. When we analyse a complete hydraulic transmission system composed of oil pump and oil turbine, we must further take account of pump and nozzle efficiency besides turbine efficiency. In order to increase the stall torque per unit input

Fig. 12 Property of working oil

power to oil pump, it is, of course, most effective to use working oil with large quantity and low pressure. In this case, however, the impulse oil turbine loses its principal benefits; and should be replaced by a turbine of other type.

4. Conclusion

An impulse oil turbine similar to the Terry type steam turbine has been investigated on the basis of fundamental experiment and performance study of a test turbine. From the result the characteristics and the performance limit of this kind of oil turbine have been made clear, and some design data for an efficient impulse oil turbine have been obtained.

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