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Experimental Study on Radial Turbine, with Special Reference to the Influence of the Number of Impeller Blades on Performance Characteristics

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Abstract

Experiments were conducted to ascertain the influence of the number of impeller blades on the performance characteristics of a radial turbine. Thus, impellers fitted with 4, 8, 16, 24 and 32 blades respectively were employed, and the radial turbine was driven by compressed air delivered from Roots blowers. Besides usual measurements of pressures and temperatures, the flow directions as well as the magnitudes of the absolute velocities at the entrance of the impeller were measured by a yaw meter.

The following results were obtained. (1) The optimum number of impeller blades of a radial turbine is not less than that of a centrifugal compressor. (2) In regions of maximum overall adiabatic efficiencies, the relative velocities at the entrance of the impeller decline rotationwise by the angles 20～30 degrees.

I. Introduction

Radial turbines are widely used in every field, for instance, as small gas turbines[1] for fire pump drive as well as the exhaust turbines for superchargers.[2] Literatures concerning the performance characteristics of a radial turbine are already published by several authors, i.e. W.T. Nuell,[3] L.R. Wosika,[4] P.F. Martinuzzi,[5] while in Japan literatures by Yasuo Mori,[6] Nagao Mizumachi,[7] Masanobu Sekine[8] are available.

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1) A. D. Zakarian, R. R. Peterson; *Auto. Industries*, Vol. 104. No. 8 (1951-9-15) p. 34; *Mech. Engg.*, Vol. 74, No. 1 (1952), p. 26.
2) *Oil Engine and Gas Turbine*, Vol. 22, No. 260 (1955-2) p. 395; Vol. 23, No. 263 (1955-5), p. 32; Vol. 22, No. 262 (1953-4), p. 471.
3) W. T. Nuell: *Trans. ASME*, Vol. 74 (1952-5), p. 499.
4) L. R. Wosika: *Trans. ASME*, Vol. 74, (1952-11), p, 1337.
5) P. F. Martinuzzi: *Trans. ASME*, Vol. 74 (1952-7), p. 663
6) Y. Mori: *Trans. Japan Soc. Mech. Engrs.*, Vol. 20, No. 96 (1954), p. 552; Vol. 21. No. 109 (1955), p. 679.
7) N. Mizumachi: *Report Inst. Industrial Science, Tokyo Univ.*, Vol, 8, No. 1 (1958-12),
8) M. Sekine et al: *Preprint 33rd Annual Meeting*, J. S. M. E. (1956-4-3).

Although the problems concerning the number of impeller blades of a radial turbine are dealt with by H.R. Cox[9] to some extent, they are not yet completely solved, and so the present authors made an experimental investigation using an experimental radial turbine. It has been the general concept that smaller number of impeller blades may be available in the case of radial turbine than in the case of centrifugal compressor, because of the expansion flow process having less possibility to stall, and the present authors also expected the same results. Application of the formula for optimum number of impeller blades of a centrifugal compressor by L. R. Wosika $z=10+0.75D$, where D is the impeller diameter in inches, to the present case ($D=150$ mm) yields $z=14.44$. Therefore, we conducted experiments with impellers having 16, 8 and 4 blades and found the optimum number of impeller blades far exceeding the tested numbers. Thus the authors prepared two more impellers with 24 and 32 blades respectively, and the experiments were carried out over again for all the five impellers, using compressed air from Roots blowers.

The results of the present investigation were that the optimum number of impeller blades for radial turbine is not less than that for centrifugal compressor. Further, in these experiments, the absolute velocities as well as the flow directions of the air just entering the impeller were measured by a yaw meter, and some interesting results were obtained, which will be shown later.

II. Experimental Apparatus and the Method

The skeleton drawing of the experimental apparatus is shown in Fig. 1. The air delivered by two Roots blowers, A and B, enters first into the pressure tank,

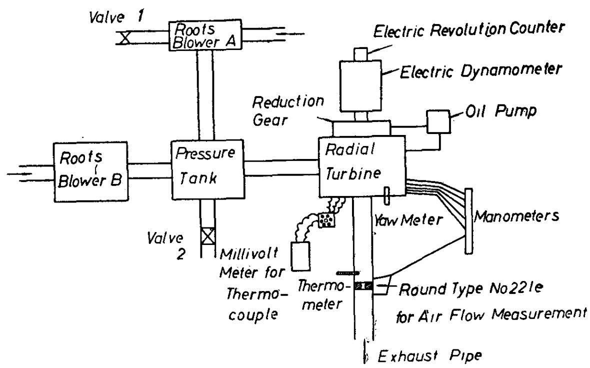

Fig. 1. Skeleton Drawing of the Experimental Apparatus

and then is delivered to the experimental radial turbine. Flowing through the nozzle and impeller passage, the air is exhausted finally into the atmosphere through the exhaust pipe. Roots blower A is driven by a 12 PS induction motor, while Roots blower B is driven by a 15 PS induction motor. The weight flow as well

9) H. R. Cox: *Gas Turbine Principles and Practice* (1955), p. 9.

as the pressure of the air entering the radial turbine may be controlled by the two valves 1 and 2. Round type nozzle for air flow measurement is fitted to the exhaust pipe as shown. The radial turbine is coupled to the 2 PS electric dynamometer through a reduction gear, the reduction ratio of which is 1/3.4. Thus, the electric dynamometer indicates the horsepower difference between the shaft horsepower of the turbine and the lost horsepower in the reduction gear. Lubricating oil is fed to the turbine and to the reduction gear by a gear pump by a 1/20 PS electric motor. A yaw meter of 4 mm diameter is installed at the exit of the nozzle (just upstream to the entrance of the impeller) as shown in Fig. 4, and the absolute velocities as well as the flow directions of the air entering the impeller were measured.

The experimental radial turbine used is shown in Fig. 2. Five impellers fitted with 32, 24, 16, 8 and 4 blades were machined from aluminium casting, and these impellers were of 150 mm outside diameters and 90 mm inside diameters as shown in Fig. 3. Common exducer having 8 blades was employed for these five impellers, the exit angle of the exducer being 45 degrees corresponding to the design weight flow $G=0.15$ kg/s and impeller revolutions $n=6000$ r. p. m.. The inlet flow area for the turbine and the exit flow area from the turbine amount to 0.003 m^2 and 0.00636 m^2 respectively, and the nozzle used is, as shown in Fig. 2, of parallel wall type without guide vanes, the outside diameter and the inside diameter of which are 220 mm and 160 mm respectively.

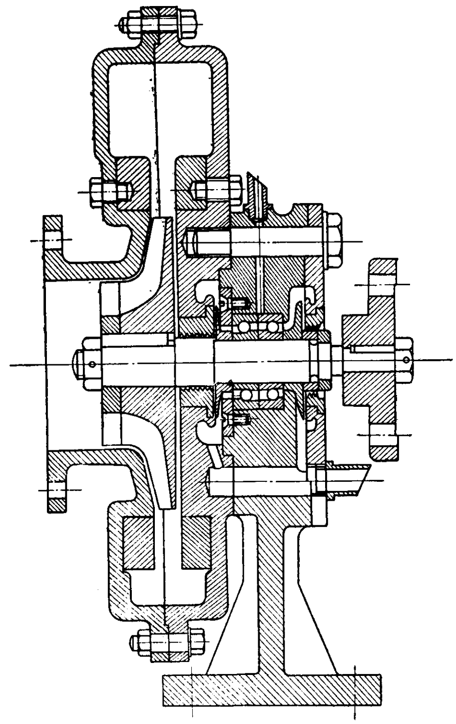

Fig. 2. Experimental Radial Turbine

Observations of pressures and temperatures were made at the sites shown in Fig. 4. Thus, pressures $p_1 \sim p_5$ and temperatures $t_1 \sim t_5$ within the scroll and pressures $p_6 \sim p_9$ and temperatures $t_6 \sim t_9$ along the impeller passage were measured, the latter pressures and temperatures being observed from the casing by means of water manometers and thermocouples. p_6 and t_6 correspond to the pressure and and temperature at the inlet to the impeller respectively, while p_9 and t_9 denote those just at the exit end of the exducer blades. The location at which the yaw meter is fitted is also shown in the figure. In the exhaust pipe, the pressure and temperature upstream to the nozzle p_{10}, t_{10} as well as the down stream pressure

p_{11} were measured. Further, the atmospheric pressure p_a and temperature t_a were read in each observation. In the above-mentioned observations, pressures were measured by mercury or water manometers, while temperature observations were made by means of thermocouples and thermometers.

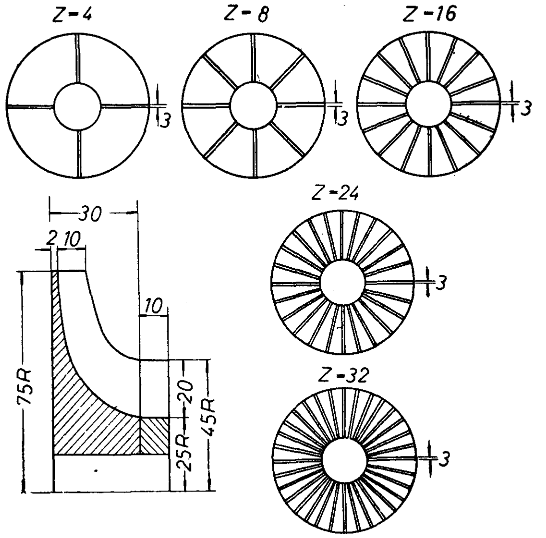

Fig. 3. Impellers used in the Experiment

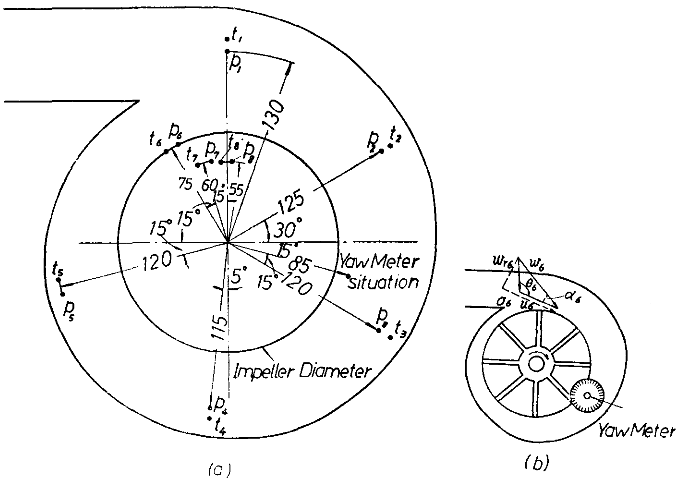

Fig. 4. Sites for Observations of Pressures, Temperatures and Flow Directions

As to the experimental method, the openings of the valves 1 and 2 were so adjusted that the weight flow of air through the radial turbine might remain constant, thus the impeller revolutions were varied from 4000 r.p.m. to 11000 r.p.m. by varying the load of the electric dynamometer to take observations at every 1000 r. p. m. interval.

III. Experimental Results and Considerations

The observed pressures were corrected to the standard atmospheric condition by the formula $p=p^0(760/p_a{}^0)$, where p^0 denotes the observed pressure when the barometric pressure corresponds to $p_a{}^0$, and p means the pressure when the barometric pressure is standard, i.e. 760 mm Hg. As to the temperatures, no correction was made.

The evaluation of overall adiabatic efficiency η_0 of the turbine was made by the following expression;

$$\eta_0=\frac{L}{\frac{1}{75}G\frac{\kappa}{\kappa-1}RT_1\left\{1-(p_9/p_1)^{\frac{\kappa-1}{\kappa}}\right\}+G\frac{1}{2g\cdot 75}(w_1{}^2-w_9{}^2)} \tag{1}$$

where L: turbine output PS, G: weight flow of air kg/s, T_1: absolute air temperature at nozzle inlet °K, p_1, p_9: static pressures at nozzle inlet and exducer exit respectively, w_1 : absolute velocity of air at nozzle inlet m/s, and w_9 : the theoretical absolute velocity of air at exducer exit calculated from the measured weight flow of air G assuming adiabatic isentropic expansion from p_1 to p_9.

The slip coefficient of impeller may be defined as follows,

$$\frac{u_6\sigma_6}{g}=\left[\frac{1}{2g}(w_6{}^2-w_9{}^2)+\frac{\kappa}{\kappa-1}RT_6\left\{1-\left(\frac{p_9}{p_6}\right)^{\frac{\kappa-1}{\kappa}}\right\}\right]\mu \tag{2}$$

where μ : slip coefficient of impeller, p_6, T_6 : static pressure and absolute temperature of air at the impeller inlet, w_6 : absolute velocity of air at the impeller inlet, σ_6 ; tangential component of absolute velocity of air at the impeller inlet, and u_6 : peripheral velocity at the impeller inlet (outside diameter of the impeller). In the above expression, w_6 may be read from the yaw meter observations, and values of the theoretical absolute velocities of air at exducer exit w_9 may be derived from the assumption that the expansion p_6 to p_9 is adiabatic and reversible.

Fig. 5 represents the relation between overall adiabatic efficiency η_0 and impeller revolutions per minute n for the case $G=0.1080$ kg/s, the number of impeller blades being taken as parameters. This belongs to the smaller weight flow regions, and as is clear from the figure, the highest value of η_0 is obtained for the case $z=32$ at higher revolutions, and η_0 becomes worse as z decreases. In the regions $n=7000$ $\sim$8000 r.p.m., it is seen that the cases $z=16$, 24 and 32 are favourable. Figs. 6 and 7 represent the relations between η_0 and n in the case $G=0.1251$ kg/s and $G=0.1457$ kg/s respectively, i. e. the cases of medium and larger weight flows. In these cases, η_0 becomes better at $z=8$, 16 and 24, while η_0 decreases at $z=32$. It is conceivable that this is due to the increment of frictional losses within the impeller

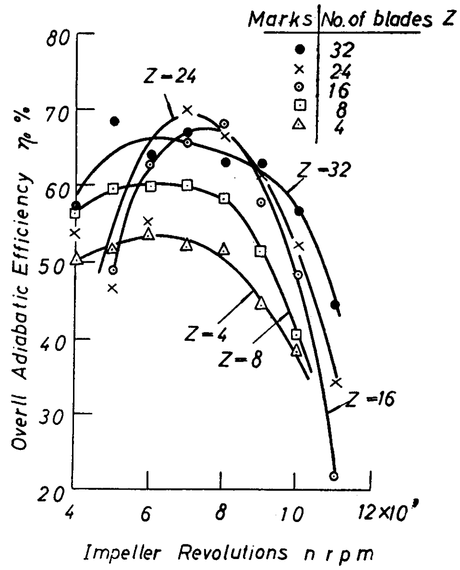

Fig. 5. Relation between η_0 and n (G=0.1080 kg/s)

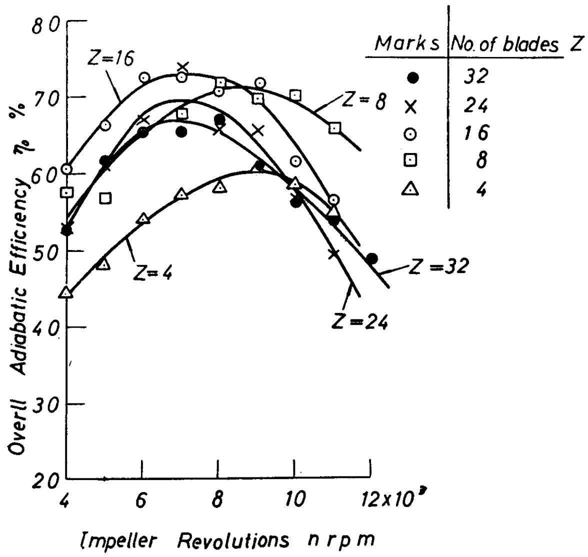

Fig. 6. Relation between η_0 and n (G=0.1251 kg/s)

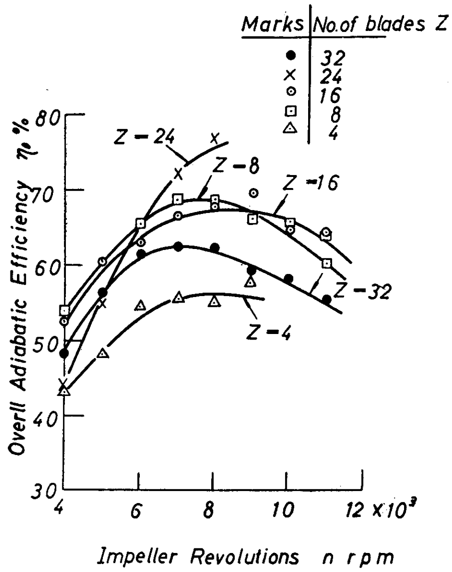

Fig. 7. Relation between η_0 and n (G=0.1457 kg/s)

passage. The value η_0 is somewhat low in every case, which may be attributed to the losses in the reduction gear.

It has been considered generally that the optimum number of impeller blades will be less than that for centrifugal compressor on account of expansion flow process. Dr. B. Eckert, whom one of the authors had met in his journey, said to him that the optimum number of impeller blade for the present case would be around $z=17$, and Dr. W. T. Nuell in Garrett Corp. showed him that the optimum number z_{opt} might be determined frcm the criterion that $z_{opt}=8$ for $D_2/D_1=2$ and $z_{opt}=12$ for $D_2/D_1=1.5$. For the present case, in which $D_2/D_1=150/90=1.67$, we have $z_{opt}=10$ according to Nuell's criterion, but the results already given in Figs. 5, 6 and 7 show that z_{opt} is greater than the criterion. As stated previously, the optimum number of impeller blades for centrifugal compressor $z_{opt}=14.44$, and so we find it necessary to adopt a larger value of z_{opt} for the radial turbine than for the centrifugal compressor when the weight flow of air is relatively small, and further to adopt z_{opt} for the radial turbine at least equal to z_{opt} for the centrifugal compressor for a large weight flow.

Next, as we are acquainted with the magnitude and direction of the absolute velocity w_6 at the inlet to the impeller, we are now able to check from Figs. 5, 6 and 7 whether or not the relative velocity at the impeller inlet w_{r6} enters radially inward along the blades at points of maximum overall adiabatic efficiencies η_0. As is seen from Fig. 4 (b), the magnitudes and directions of w_6 were measured at the region where the influence of the scroll entrance is considered negligible. From the observed magnitudes of w_6, angle α_6 and peripheral velocity u_6, we are able to find values of w_{r6} and θ_6 from the velocity diagram as shown in Fig. 4 (b). Thus it is possible to check whether θ_6 amounts to 90 degrees (radially inward) or not. The results for $z=4$, 8, 16, 24 and 32 are shown in Figs. 8, 9, 10, 11 and 12 respectively. From these results we have : $\theta_6=104°\sim122°$ for $z=4$ (Fig. 8), $\theta_6=108°\sim120°$ for $z=8$ (Fig, 9), $\theta_6=108°\sim121°$ for $z=16$ (Fig. 10), $\theta_6=112°\sim122°$ for $z=24$ (Fig.11) and $\theta_6=110°\sim128°$ for $z=32$ (Fig. 12). Then we find that θ_6 is larger than 90°, and we are to choose the value of θ_6 around $110°\sim120°$ in the design of a radial turbine. Also we see that the actual values of θ_6 do not vary largely as the number of impeller blades z changes, though θ_6 for $z=32$ is slightly larger than in the other cases. The reason why the values of θ_6 for maximum η_0 amount to $110°\sim120°$ instead of 90° is not clear, but is is conceivable that the problem is intimately related to the flow pattern, especially the separation of flow within the impeller passage.

The relation between the shaft horsepower of the radial turbine and the impeller revolutions n is shown in Figs. 13, 14, 15, 16 and 17 for the case $z=4, 8, 16, 24$ and 32, respectively, the weight flow G being selected as parameter. From these diagrams, it is seen that the optimum n for maximum shaft horsepower lies somewhat higher than that for maximum η_0.

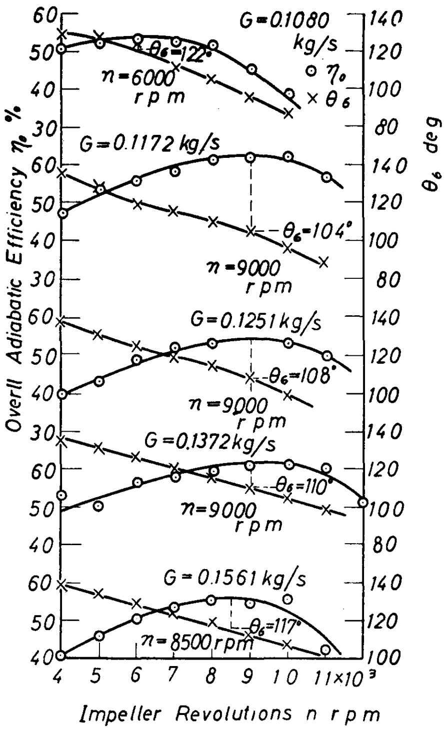

Fig. 8. Relation between η_0 and θ_6 ($z=4$)

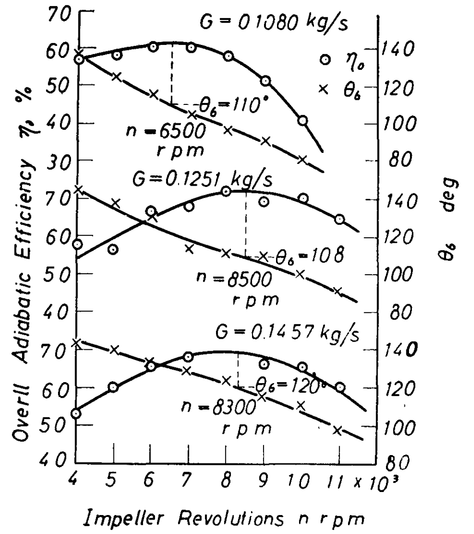

Fig. 9. Relation between η_0 and θ_6 ($z=8$)

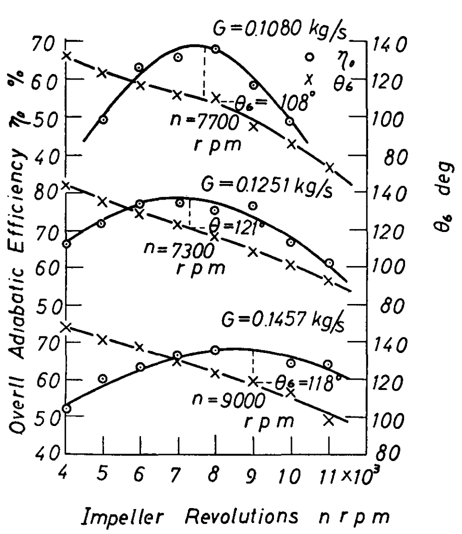

Fig. 10. Relation between η_0 and θ_6 ($z=16$)

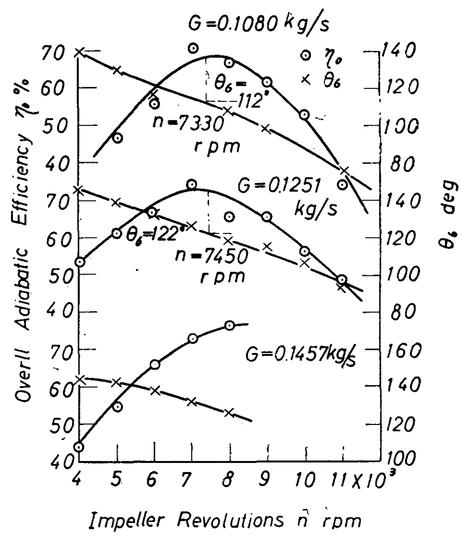

Fig. 11. Relation between η_0 and θ_6 ($z=24$)

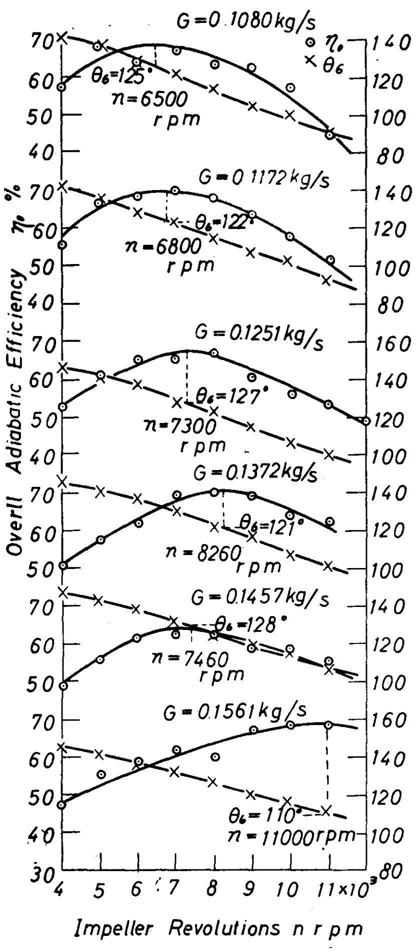

Fig. 12. Relation between η_0 and θ_6 $(z=32)$

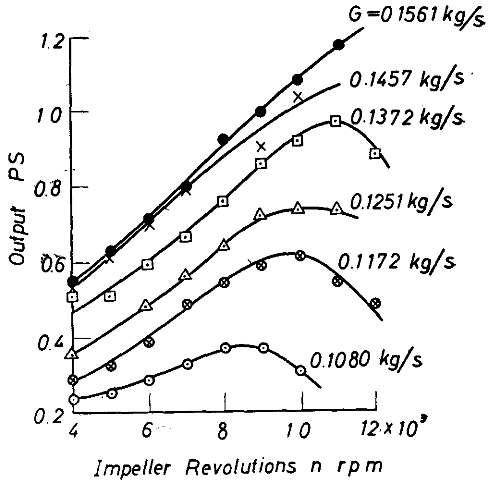

Fig. 13. Relation between output and n $(z=4)$

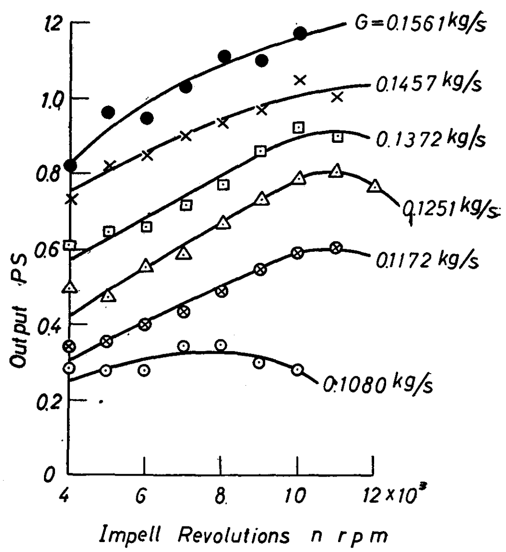

Fig. 14. Relation between output and n $(z=8)$

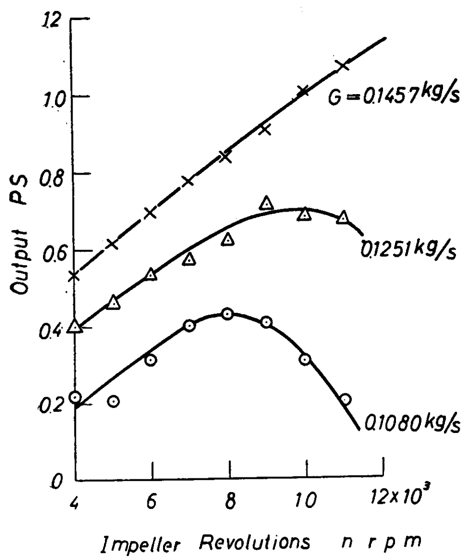

Fig. 15. Relation between output and n ($z=16$)

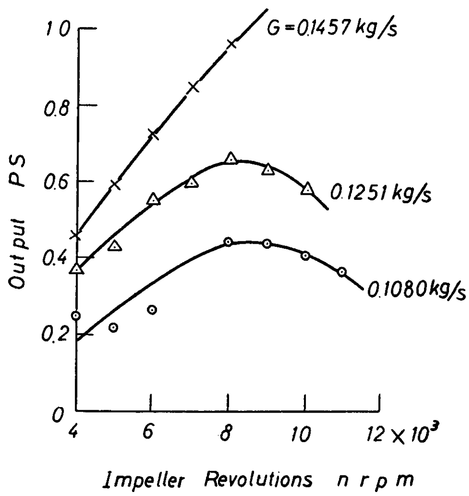

Fig. 16. Relation between output and n ($z=24$)

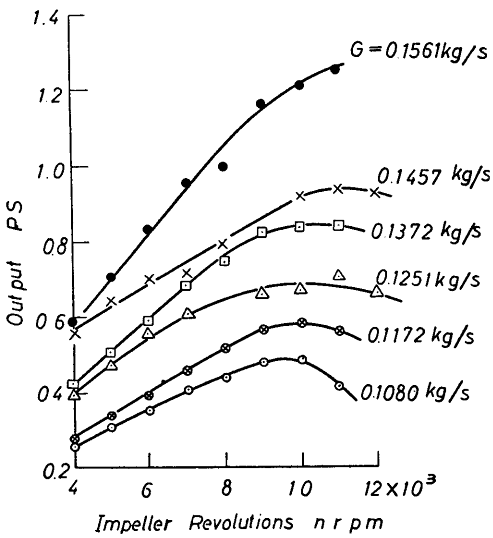

Fig. 17. Relation between output and n ($z=32$)

From the pressure distributions within the scroll as well as the impeller passage, the case of which for $z=32$ and $G=0.1251$ kg/s is shown in Fig. 18, we learn that the slope of the expansion curve within the impeller passage becomes steeper as n becomes larger. Further, we are able to evaluate the reaction grade from the pressure distributions. We took the pressure p_2 as the representative pressure in the scroll, because we are now considering the flow near the location of yaw meter. Thus, for example, as to the case $z=32$ and $G=0.1251$ kg/s (Fig. 18), we have the pressure drop within the nozzle $\Delta p_n=p_2-p_6=10850-10560=290$ mm Aq and pressure drop within the impeller $\Delta p_i=p_6-p_9=10560-10420=140$ mm Aq for $n=4000$ r.p.m. to find the reaction grade $r=\Delta p_i/(\Delta p_n+\Delta p_i)=140/430=0.326$. Similar evaluations yield $r=140/465=0.301$ for $n=5000$ r.p.m., $r=190/500=0.380$ for $n=6000$ r.p.m., $r=250/555=0.450$ for $n=7000$ r.p.m., $r=295/615=0.480$ for $n=8000$ r.p.m. and $r=330/690=0.478$ for $n=9000$ r.p.m.. Examination of the case for $z=8$, $G=0.1251$ kg/s reveals that $r=290/710=0.408_5$ for $n=7000$ r.p.m., $r=320/710=0.451$ for $n=8000$ r.p.m. and $r=385/900=0.428$ for $n=9000$ r.p.m.. Thus we see that the reaction grade for maximum η_0 in the case of $z=32$ and $G=0.1251$ kg/s (Fig. 18) ($n=7300$ r.p.m. from Fig. 12) amounts to $r=0.450\sim0.480$, while that for maximum η_0 in the case of $z=8$, $G=0.1251$ kg/s ($n=8500$ r. p. m.) lies within the range $r=0.451\sim0.428$. Finally, we learn from these evaluations that the reaction grade which makes the overall adiabatic efficiency η_0 maximum lies in the region $0.43\sim0.48$.

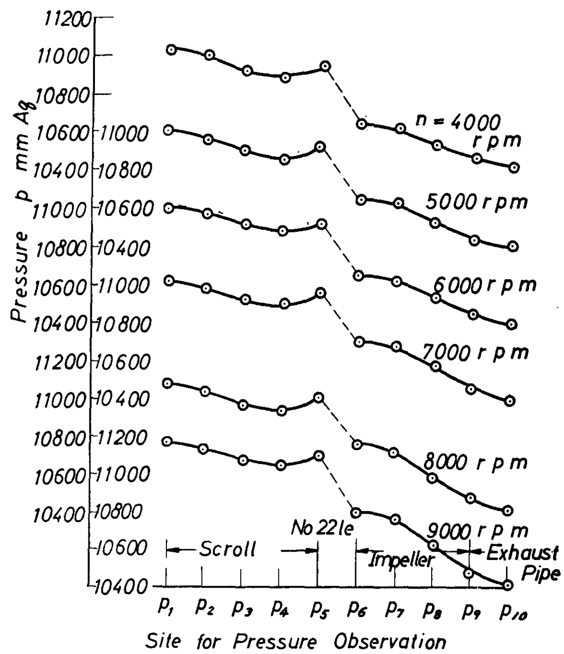

Fig. 18. Pressure distributions within the scroll and impeller passage ($z=32$, G$=0.1251$ kg/s)

Fig. 19 shows the relation between the slip coefficient μ evaluated by Eq. (2) and the impeller revolutions n indicating that μ decreases as n becomes larger. Fig. 19 corresponds to the case of Fig. 5 and from this figure we see also that μ for $z=16$ and 24 has a higher value than for the other cases.

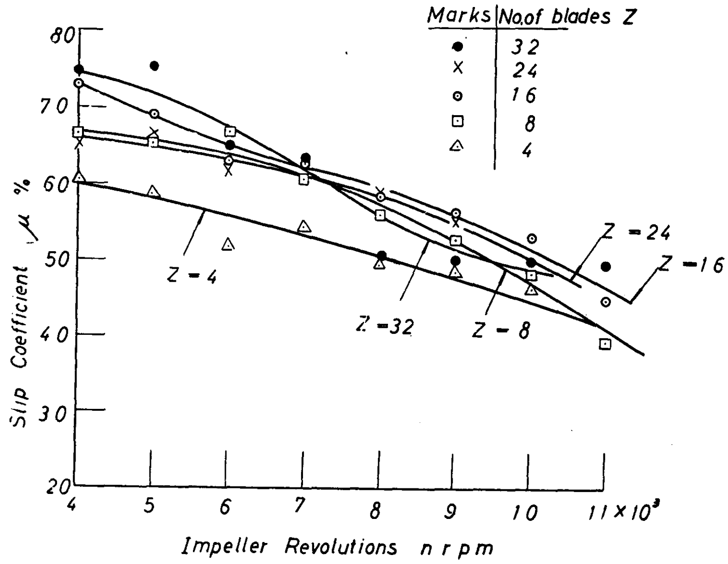

Fig. 19. Relation between μ and n ($G=0.1080$ kg/s)

IV. Conclusions

From the above-mentioned results and considerations, we reach the following conclusions.

(1) The optimum number of impeller blades for the radial turbine is not less than that for the centrifugal ccmpressor. The optimum number of impeller blades for the radial turbine is comparable with that for the centrifugal compressor for the larger weight flow, but it is preferable to select the former larger than the latter for the smaller weight flow.

(2) The inflow direction of the relative velocity at the impeller inlet θ_6 for the cases of maximum overall adiabatic efficiencies is not purely radial ($\theta_6=90°$), but θ_6 amounts to around $110°\sim120°$, i. e. the inflow direction decline rotationwise relative to the impeller at every number of impeller blades. Strictly speaking, θ_6 is somewhat larger in the case $z=32$, that is $\theta_6=121°\sim128°$.

(3) The impeller revolution which makes the shaft horsepower maximum is larger than that for maximum η_0 at every number of impeller blades and weight flow.

(4) The reaction grade corresponding to maximum overall adiabatic efficiency η_0 amounts to about $0.43\sim0.48$.

(5) The slip coefficient of impeller μ decreases as impeller revolutions increase for every number of impeller blades.

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