621.438-253.018

Experimental Study of a Radial Turbine with Special Reference to the Influence of Intermediate Blades*

By Ichiro WATANABE** and Toshimichi SAKAI***

Experiments were conducted to find out the influence of intermediate blades within the impeller channel on the performance characteristics of a radial turbine.

In the first experiments, one or two intermediate blades were installed within each impeller channel respectively, and the influence of the number of intermediate blades upon the performance was examined. The results indicate that the installation of intermediate blades improves the performance considerably. The second experiments concern the investigation of the effect of the length of an intermediate blade upon the performance for the case of a single intermediate blade in each blade channel. The most favorable length of the intermediate blade obtained from the experiments is about 0.4 or 0.5 times the whole blade length.

1. Introduction

Because of their simple mechanism and structural superiority in comparison with axial flow turbines, radial turbines are used for small-sized superchargers and power gas turbines. Research papers about the performance characteristics of a radial turbine are given by several authors, such as W. T. Nuell⁽¹⁾, L. R. Wosika⁽²⁾, P. F. Martinuzzi⁽³⁾, F. J. Wallace⁽⁴⁾, E. M. Knoernschild⁽⁵⁾, Y. Mori⁽⁶⁾, N. Mizumachi⁽⁷⁾ and T. Watanabe⁽⁸⁾. By them such aspects about radial turbine performances have been studied, as losses in various sections of the turbine, pressure distributions on the blade surface, flow behavior at the turbine impeller inlet and exit, and so on. As shown in the previous paper⁽⁹⁾, the present authors studied problems concerning the number of blades and a better performance has been obtained for a comparatively large number of impeller blades. But the more blades the impeller has, the more the flow area around exit becomes limited and the more difficult the fabrication of the impeller blade becomes. To overcome these inconveniences, a valid method is supposed to be use of intermediate blades in the entrance of the blade channels. But concerning the intermediate blades there are many unknown factors, for example the most suitable number and length of intermediate blades to optimize the

performance, the characteristics of the impeller having intermediate blades compared with those of the ordinary impeller. In this report, the results of the following investigations about the radial turbine having an intermediate blade at the impeller inlet are presented.

(1) Effect of the number of intermediate blades: The performance characteristics of a radial turbine with the impeller having no intermediate blades were compared for two cases; i.e. one intermediate blade for each channel and two for each.

(2) Influence of the length of intermediate blades: As this effect was supposed to be considerable, four kinds of impellers with intermediate blades having different lengths were made and examined.

2. Apparatus and procedures

The schematic outline of experimental apparatus is shown in Fig. 1. The air compressed by Roots blower, after passing the pressure tank 1,

Fig. 1 Experimental arrangement

* Received 30th August, 1963.

** Professor, Faculty of Engineering, Keio University.

*** Assistant, Faculty of Engineering, Keio University, Koganei.

the air flow meter of circular nozzle type and the pressure tank 2, drives the turbine and then is exhausted into the atmosphere. Total pressure just before the turbine is measured at the pressure tank 2. The cross sectional view of the turbine is shown in Fig. 2 and positions for observation of pressures, temperatures, air flow directions and velocities are shown in Fig. 3. The turbine nozzle used is a parallel walled vaneless type. The impeller having 16 blades is designed to get constant relative velocity distributions along the channel passage at pressure ratio 1.18. Common exducer having 8 blades and an exit angle of 45 degrees is employed for all impellers. The length of intermediate blade is indicated as the ratio of intermediate blade length L to the full length of blade L_0 . Dimensions of the impeller are shown in Fig. 4 and Table 1. As to the experimental procedures, the turbine expansion ratio was fixed, and the rotating speed of the impeller was changed by varying the

load of the electric dynamometer. In the efficiency calculations, the reduction gear losses are not considered.

3. Experimental results and considerations

Symbols in this paper are as follows;

P : pressure kg/cm^2

T : absolute temperature $^{\circ}\text{K}$

G : weight flow of air kg/sec

L : turbine output kg m/kg

η_{ad} : overall adiabatic efficiency

$$\eta_{ad} = L / \left[\frac{\kappa}{\kappa-1} RT_0 \left\{ 1 - \left(\frac{P_2}{P_0} \right)^{\frac{\kappa-1}{\kappa}} \right\} + \frac{c_0^2 - c_2^2}{2g} \right]$$

u : peripheral velocity of impeller m/sec

c : absolute velocity of air

α_1 : inlet angle of absolute velocity

θ_1 : inlet angle of relative velocity

r : reaction grade

$$r = \left[1 - \left(\frac{P_1}{P_0} \right)^{\frac{\kappa-1}{\kappa}} \right] / \left[1 - \left(\frac{P_2}{P_0} \right)^{\frac{\kappa-1}{\kappa}} \right]$$

Subscripts;

0: turbine inlet, 1: impeller inlet, 2: impeller exit

t : total condition, s : static condition

Table 1 Dimensions of the impeller

Number of blades	Number of intermediate blades	Blade thickness d mm	Intermediate blade length L/L_0	Notation
8	0	3		8_0
	8	3	0.3	$8_8-0.3$
			0.4	$8_8-0.4$
			0.5	$8_8-0.5$
			0.65	$8_8-0.65$
16	0	3		16_0
8	16	2	0.65	$8_{16}-0.65$
24	0	2		24_0

Fig. 2 Experimental arrangement

Fig. 3 Positions for observation of pressures, temperatures and air flow directions

Fig. 4 Turbine blade

Weight flow of air and impeller peripheral velocities are corrected by the dimensionless factors as follows.

$$G\sqrt{T_{t0}/P_{s2}} \quad u_1/\sqrt{T_{t0}}$$

3.1 Effect of the number of intermediate blades Comparisons of results were made about impellers having 16 and 24 blades, impellers having no intermediate blades, and 8-blade impellers with and without intermediate blades. Fig. 5 represents the relations between overall adiabatic efficiencies for these impellers and corrected impeller periph-

eral velocities.

The turbine expansion ratios are used as parameters.

Fig. 6 (a) shows comparison of overall adiabatic efficiencies for these five impellers, using the turbine expansion ratio as parameter. By this figure, an 8-blade impeller shows a considerably low efficiencies compared with those of the other impellers. Impellers with intermediate blades show slightly high efficiencies compared with those of impellers without intermediate blades. It is conceivable that this effect is due to the fact that a large number of blades near the exducer causes an increase of frictional loss within the channel due to smaller distance between adjacent blades.

Fig. 6 (b) shows the relations between optimum overall adiabatic efficiencies and turbine expansion ratios for these five impellers. From this figure it is clear that the impellers which have a larger number of blades at the inlet, such as $8_{18}-0.65$ and 24_0 impellers show higher efficiencies at the low expansion ratio. From this result it is conceivable that the impellers which have a larger number of blades at the inlet give better part-load performances.

Fig. 5 Relation between η_{ad} and $u_1/\sqrt{T_{t0}}$

Fig. 6 (c) shows the relation between the corrected air flow rate $G\sqrt{T_{t0}/P_{s2}}$ and the corrected impeller peripheral velocity $u_1/\sqrt{T_{t0}}$. In this case, turbine expansion ratio is taken as a parameter as in the former case. Impellers having a smaller number of blades at the inlet show less flow rate, in spite of larger sectional area of impeller inlet.

In the region of small impeller peripheral velocities, this tendency becomes remarkable. Theoretically⁽¹⁰⁾⁽¹¹⁾ and experimentally⁽¹²⁾ it is known that the relative flow patterns within the impeller channel show deviations from the radial directions. It seems that the impellers which have a smaller number of blades at the impeller inlet show larger deviations and less favorable flow condition. Impellers having 24 blades at the inlet, such as $8_{16}-0.65$ and 24_0 impellers, show larger flow rates compared with those having 16 blades at the inlet, such as $8_8-0.65$ and 16_0 impellers, but the adiabatic efficiencies are of a comparable order. It is conceivable that the impellers which have 24 blades at the inlet have larger skin friction in the impeller blade channels. Thus in spite of the improvements of the flow condition, adiabatic efficiencies are not so much improved.

3.2 The effect of the length of the intermediate blades To investigate this effect, a single intermediate blade was installed in each blade channel. Experiments were carried out with four kinds of impellers having intermediate blades of different lengths, dimensions of which are shown in Table 1. In order to compare these four impellers concerning flow rates and overall adiabatic

Fig. 6

efficiencies, performance curves for these are shown in Fig. 7. This figure represents the relation between corrected impeller peripheral velocity and

Fig. 7

corrected flow rate. Solid lines show curves of constant pressure ratios and broken lines show curves of constant overall adiabatic efficiencies.

From these figures the following trends are found out. Turbine overall adiabatic efficiency increases with an increasing length of intermediate blades from $L/L_0=0.3$ to $L/L_0=0.4$ and it attains optimum value at $L/L_0=0.5$ and in case of $L/L_0 > 0.5$ the overall adiabatic efficiency decrease rapidly. The

Fig. 8 (1)

reason why the above results were obtained will be explained as follows. Namely, when the intermediate blade length is too short, its effect is not sufficient and when the length is too long, on account of the previous reason, frictional losses within the impeller channels near the exducer are increased.

3.3 Flow measurement at the impeller inlet

The flow directions and the magnitudes of the

(c) Flow condition at the impeller inlet

(d) Flow condition at the impeller inlet

Fig. 8 (2)

absolute velocities of the flowing air are measured by a cylindrical yaw meter at the entrance of the impeller. Fig. 8 indicates the inlet angle of absolute velocity and relative velocity for the cases of 8-blade impeller (8_0), intermediate blades impellers ($8_8-0.4$, $8_8-0.65$) and 16-blade impeller (16_0). In these cases turbine expansion ratio is 1.18. Also in this figure turbine reaction grades and overall adiabatic efficiencies are shown. The velocity triangle in this figure represents the condition corresponding to the optimum overall adiabatic

Fig. 9 Flow conditions and performances of the impellers

efficiency. From the figure of the inlet angle of the relative velocity θ_1 , it is clear that the θ_1 is not exactly radial but larger than 90° at the optimum point. To show the effect of intermediate blades more clearly, in Fig. 9, the relation between the intermediate blade length and the optimum value of overall adiabatic efficiency is shown. Also in this figure, corrected impeller peripheral velocity, reaction grade, inlet angle of relative and absolute velocities for the optimum overall adiabatic efficiency are shown. About the direction of relative velocity θ_1 , $s_8=0.4$ impeller shows the closest value to the radial direction ($\theta_1=90^\circ$) at the optimum point.

4. Conclusions

From the above-mentioned results, the following conclusions were obtained.

(1) The number of blades at the impeller inlet has significant influence on the performance of the radial turbine, and by setting the intermediate blades at the channel inlets, a better performance than with impellers without intermediate blades can be attained.

(2) Turbine flow rate is largely affected by the number of blades at the impeller inlet. Compared at the same turbine expansion ratios, the impeller which has a larger number of blades at the impeller inlet has a larger flow rate.

(3) The impeller which has a larger number of blades at the inlet shows better part-load performances. So it can be used for a wide range of turbine loading.

(4) If the intermediate blade is too short, the effect of adjusting the streaming is insufficient, and if it is too long, the skin friction in flow channels increases. Namely, existence of a suitable length of intermediate blade to give the optimum performance is conceivable, and in these experiments, the most favorable length of intermediate blade was about 0.4 to 0.5 times the full length blade.

(5) The direction of relative velocity to give the optimum overall adiabatic efficiency is greater than 90° , which coincides with the results of the previous paper⁽⁹⁾. But in the case of the impeller having intermediate blades of suitable length, this angle approaches to 90° . From this result it is conceivable that the impeller having intermediate blades of suitable length have a favorable condition of stream at the blade channel.

(6) As for the fabrication, the impeller having intermediate blades is easier to work because of less number of blades near the exducer, and is found very suitable for radial turbines.

The present authors are much indebted to Professors K. Komotori, T. Sato, T. Ando of Keio Univ. and Dr. K. Fujie of the Hitachi Co. Ltd. for their useful comments and Assistant T. Mashimo for his devoted cooperation in experiments.

References

- (1) W.T. Nuell: *Trans. ASME*, Vol. 74 (1952-5), p. 499.
- (2) L. R. Wosika: *Trans. ASME*, Vol. 74 (1952-11), p. 1337.
- (3) P.F. Martinuzzi: *Trans. ASME*, Vol. 174 (1952-7), p. 633.
- (4) F.J. Wallace: *Proc. Inst. Mech. Engrs.*, Vol. 172, No. 33 (1958), p. 931.
- (5) E.M. Knoernschild: *ASME Publ.*, Paper No. 59-A-294.
- (6) Y. Mori: *Trans. Japan Soc. Mech. Engrs.*, Vol. 20, No. 96 (1954), p. 552; Vol. 21, No. 109 (1955), p. 679, Y. Mori and others: *Bulletin of JSME*, Vol. 6, No. 24 (1963), p. 757.
- (7) N. Mizumachi: *Report Inst. Industrial Science, Univ. Tokyo*, Vol. 8, No. 1 (1958-12).
- (8) T. Watanabe: *Trans. Japan Soc. Mech. Engrs.*, Vol. 27, No. 180 (1961), p. 1256.
- (9) I. Watanabe and T. Ando: *Trans. Japan Soc. Mech. Engrs.*, Vol. 25, No. 149 (1959), p. 4.
- (10) J.D. Stanitz: *NACA Rep.* 935.
- (11) K. Fujie: *Trans. Japan Soc. Mech. Engrs.*, Vol. 23, No. 125 (1957), p. 3.
- (12) I. Watanabe and others: *Preprint of the 40th Annual Meeting of Japan Soc. Mech. Engrs.*, No. 87 (1963-4), p. 35.