Characteristics of Radial Inward Turbines for Exhaust Gas Turbochargers under Nonsteady Flow Conditions* (Effects of Waveforms on Prediction of Turbine Performance)

Keiji KONISHI** and Haruo YOSHIKI***

This paper deals with effects of waveforms on predicting performance of a radial turbine for turbochargers under pulsating flows. The extensive unsteady tests were conducted with several pulse shapes and pressure amplitudes over a pulse frequency range from 10 to 70 Hz produced by a disk-type pulse generator rig. Five kinds of pulse shapes were simulated for exhaust gas flows from diesel engines. The time-mean method and the quasi-steady flow analysis were carried out to predict the turbine characteristics by using the steady flow turbine performance. As the results, the following points were clarified. For most conditions, the quasi-steady flow analysis of turbine performance is valid even if the turbine is affected by the waveform, amplitude and frequency. It is considered that the turbine under pulsating flow conditions works along the curves of steady flow performance except for windage states. The time-mean method predicts the power output fairly well, but grossly overestimates the mass flow rates for some waveforms.

Key Words: Supercharger, Unsteady Flow, Turbomachinery, Exhaust Radial Turbine, Quasi-Steady Analysis, Waveform

1. Introduction

Three factors of unsteady flow, the pulse frequency, pressure amplitude and waveform, are most important in predicting the performance of radial turbines driven by exhaust pulsating flow. In order to achieve the best match between engine and turbocharger, it is necessary to clarify how they effect the turbine performance.

Many studies have already been done experimentally and numerically with regard to the prediction of unsteady turbine performance^{(1)~(6),(9)~(12)}. However, all of them have paid attention only to the pulse frequency, and the effects of the amplitude and the waveform of pulse flow have not been clear. In previous papers^{(7),(8)}, the authors reported the effects of pulse frequency and velocity amplitude on turbine

mass flow rate and power output using the time-mean method and the quasi-steady analysis. Papers on the effect of waveform have been published only by Wallace and Blair⁽¹⁾ and Kosuge et al.⁽⁶⁾, because it is too difficult to change waveforms of unsteady flow under high pressure and with a large mass flow rate. Sufficient results have not yet been obtained since they have not described the influences of pressure amplitude and waveform of pulse flow, which does not correspond to the engine exhaust flow.

In the present investigations, the relationship between the characteristics of the turbine and the pulsating flow has been studied experimentally as basic research on the prediction of turbine performance under various pulsating flow conditions.

Furthermore, in the present paper, several waveforms have been produced by certain types of pulse generators, and their effects on turbine performance have been studied by the time-mean method and quasi-steady analysis on the basis of steady performance curves.

Nomenclature

f : pulse frequency Hz

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** Department of Mechanical System Engineering, Tokyo Metropolitan Institute of Technology, 6-6 Asahigaoka, Hino, Tokyo 191, Japan

*** Institute of Industrial Science, University of Tokyo, 22-1 Roppongi 7-chome, Minato-ku, Tokyo 106, Japan

- L : power output kW
 M : mass flow rate kg/s
 N : time-averaged turbine speed rpm
 N^* : corrected turbine speed $= N/\sqrt{T_{01}}$
 P : pressure Pa
 P_{01} : total pressure at turbine scroll inlet Pa
 R : expansion ratio $= P_{01}/P_a$
 T : temperature K
 T_{01} : total temperature at turbine scroll inlet K
 T_q : torque N · m
 V : velocity m/s
 ε_p : pressure waveform factor [Eq.(4)]
 θ : ratio of velocity amplitude⁽⁸⁾

Subscripts

- e : experimental value
 tm : characteristics by time-mean method
 q : characteristics by quasi-steady analysis
 m : time-averaged value
 $*$: corrected state
1: turbine scroll inlet
4: turbine rotor inlet
5: turbine rotor outlet
 a : atmospheric state

2. Experimental Apparatus and Procedures

The test rig is the same as that used in previous papers^{(7),(8)}, and it consists of a turbocharger, a pulse

Fig. 1 General arrangement of experimental apparatus

Fig. 2 Disk-type pulse generator

generator and an air source. The general arrangement of the test apparatus is shown in Fig. 1.

The pulse generator shown in Fig. 2 has a disk-type shutter valve and a bypass valve. It generates pulsating flow by intermitting the steady flow with a rotating disk valve which is located between two stator disks. The shutter disk has two entry holes which are located at intervals of 180 degrees for twin-entry admission systems; the referential diameter of entry holes is 60 mm, and the outer diameter of the disk is 312 mm. Accordingly, it produces two pulses of flow per revolution of the disk. Each pulse flow has the same phase and drives the turbine. Waveforms are changed by using several kinds of stator and rotor disks with various sectional configurations. The typical sectional configurations of the shutter disk are shown in Fig. 3. The rotating disk is driven by a variable-speed AC motor which is controlled by an inverter system, and the pulse frequency is changed by the shutter disk speed. The setting error of pulse frequency is less than 0.3% in the condition of pulse frequency of 10 Hz. The velocity amplitude is varied continuously by changing the bypass valve opening, and the inlet flow to the turbine is varied from nearly steady flow to nearly reverse flow. Hence, desirable pulsating flows which have various kinds of pulse frequencies and amplitudes as well as waveforms are obtained. Leakage losses of working gas to atmosphere are negligible from the seal of the driving pulley. Although the leakage of working gas between the stator and rotor disks affects the averaged mass flow rate through the turbine, it is negligibly small at the maximum pressure amplitude.

The performance tests were carried out only at the corrected turbine speed of $N^*=800$ in both cases of steady and unsteady flow. The steady flow tests were performed by fixing the disk valve and by a typical procedure in which the turbine load was controlled, as described in Ref.(7). In the present study, the steady flow test was also conducted each time before and after the unsteady one to calibrate the hot-wire anemometer, temperature unit, pressure transducer and others.

The unsteady flow tests were carried out only in

Fig. 3 Typical disk configurations

the full-load condition which was obtained by fully opening the outlet valve of the compressor, and the time-averaged expansion ratio was held almost constant throughout the experiments. Five kinds of waveforms simulating exhaust gas flow from the actual engine manifolds were used in the experiments, and the sinusoidal waveform used in Refs. (7) and (8) was considered as the basic waveform. Furthermore, the unsteady flow tests were performed over a pulse frequency range from 10 to 70 Hz for each waveform. The velocity amplitude was chosen at 10 to 13 points for each frequency. After the amplitude was set by adjusting the bypass valve of the pulse generator, the corrected turbine speed was arranged to $N^* = 800$ by controlling the mass flow rate through the turbine.

3. Typical Pulsating Flow

3.1 Definition of parameters of nonsteady flow

The pressure waveform factor ε_p is defined as a new parameter to describe pulsating flow, since the ratios of velocity and pressure amplitude used in Ref. (8) cannot be used in consideration of the difference of pulse shapes. The concept of this parameter has been suggested by Benson⁽⁴⁾ and it has the same concept as the form factor used in electrical engineering. Generally, it is defined as the root mean square (rms) of pressure amplitude divided by mean pressure, as shown in Eq. (1). Thus, the value of rms inherently includes a DC component and the same concept of mean effective pressure, which is used in the area of internal combustion engines, is adapted for it. That is, the work done by one cycle of pulsating flow is regarded as equivalent work by the steady flow.

$$\varepsilon_p = \frac{\text{value of rms for pressure waveform}}{\text{value of time-averaged pressure}} \quad (1)$$

The waveform of total pressure measured at the inlet of the turbine scroll is approximated by Fourier series to evaluate the pressure waveform factor ε_p .

$$P_{01}(t) = P_{01m} + \sum_{n=1}^{\infty} (a_n \cos nt + b_n \sin nt) \quad (2)$$

Then the value of $P_{01}(t)_{\text{rms}}$ becomes

$$P_{01}(t)_{\text{rms}} = \left[(P_{01m})^2 + \frac{1}{2} \sum_{n=1}^{\infty} (a_n^2 + b_n^2) \right]^{1/2} \quad (3)$$

It is, however, difficult for Eq. (1) to express the tendency of change since the values of ε_p are very small. Furthermore, considering the energy of work done by exhaust gas expansion to the atmospheric state, the factor is redefined by using the mean pressure subtracting atmospheric pressure as below.

$$\varepsilon_p = \frac{\left[(P_{01m} - P_a)^2 + \frac{1}{2} \sum_{n=1}^k (a_n^2 + b_n^2) \right]^{1/2}}{(P_{01m} - P_a)} \quad (4)$$

Here, k is chosen as 4 because the factors of a_k and b_k become negligibly small if k is greater than 5.

Although the velocity waveform factor based on velocity fluctuations may be defined in a similar manner, only the pressure waveform factor is used in the present paper for convenience.

3.2 Typical waveforms of pulsating flow

Generally, the exhaust gas flow from engine manifolds with multicylinders has multi-peaks in waveforms at the turbine inlet in one cycle. In this case, it seems that the unsteadiness of exhaust gas flow is weakened since the pressure amplitude is smoothed, although the pulse frequency becomes higher. Further, it has already been made clear in Ref. (8) that the amplitude of pulse flow most strongly influences the turbine characteristics. Then the waveforms of pulse flow were produced in order to simulate the flow with large pressure amplitude from a single cylinder engine.

Typical waveforms are shown in Figs. 4-8. They show the relationship between the disk valve angle and the total pressure P_{01} , total temperature T_{01} , velocity V_1 at the turbine scroll inlet, total pressures at the rotor inlet and outlet P_{04} , P_{05} , and turbine shaft torque T_q measured by a torque meter, respectively. In the figures, the ratio of the valve opening area A_v is also represented in reference to the cross-sectional area of entry pipe A_o .

The experimental conditions for those figures are as follows: the corrected turbine speed N^* is 800, and the pulse frequency f is 10 Hz at the maximum pressure amplitude which is obtained by almost fully closing the bypass valve of the pulse generator. However, the ratio of maximum amplitude of each waveform is not constant because of the limitation of the air source capacity.

3.2.1 Typical waveform of No.1 Typical waveform No.1 is shown in Fig. 4. It is a fundamental waveform which is produced by the shutter valve consisting of two stator and one rotor disks with circular holes. The curve of the opening area to disk valve angle becomes a triangle and the pressure waveform thus obtained resembles the waveform produced by the rotary valve. Its compression wave rapidly rises and its expansion wave smoothly decreases.

All of the following waveforms are produced to have a rapid pressure rise and a velocity growth in exhaust pipes in order to simulate the blowdown from the engine manifolds. Thus, the configuration of the stator disk hole is cut circular, and the rotor disk hole is cut similar to that shown in Fig. 3. Various kinds of waveforms are similarly obtained by changing the configurations of the disk holes.

3.2.2 Typical waveform of No.2 As shown in Fig. 5, waveform No. 2 simulates a triangular wave

which has a long period in a low-pressure state after rapid rise and gradual reduction. The wave is relatively similar to the actual engine exhaust flow, although the maximum expansion ratio becomes a little smaller than wave No. 1 due to the shorter opening time of the shutter valve.

3.2.3 Typical waveform of No. 3 Waveform No. 3 simulates a trapezoid or square wave which has a long high pressure period, as shown in Fig. 6.

3.2.4 Typical waveform of No. 4 To induce rapid changes of compression and expansion, wave-

form No. 4 shown in Fig. 7 is the acutest and has a maximum amplitude wave like a pulse, which is obtained by the shortest opening time of the disk valve.

3.2.5 Typical waveform of No. 5 Waveform No. 5 shown in Fig. 8 simulates the actual waveform of engine exhaust by taking into account the results of the other valves.

4. Experimental Results and Discussion

The steady flow performance curves are the same

Fig. 4 Typical waveform No. 1 ($f=10 \text{ Hz}$, $\theta=0.966$, $\epsilon_p=1.376$)

Fig. 6 Typical waveform No. 3 ($f=10 \text{ Hz}$, $\theta=0.669$, $\epsilon_p=1.209$)

Fig. 5 Typical waveform No. 2 ($f=10 \text{ Hz}$, $\theta=0.801$, $\epsilon_p=1.285$)

Fig. 7 Typical waveform No. 4 ($f=10 \text{ Hz}$, $\theta=0.830$, $\epsilon_p=1.372$)

as in the previous papers^{(7),(8)}. The effect of waveforms on the validity of prediction are studied by the time-mean method and the quasi-steady analysis based on steady flow performance curves. In the present paper, the validity of the two methods is estimated by comparing the calculation with the experiment. The values of mass flow rate and turbine output have been treated in time-averaged corrected forms.

Figure 9 shows the typical relationship between the ratio of velocity amplitude θ , which is defined by the velocity amplitude and time-averaged velocity, and the pressure waveform factor ϵ_p by Eq.(4). Generally, such a relation does not hold a functional connection since it is changed by the flow resistance through the turbine and by the pulse frequency for the same turbine. When the values of ϵ_p are compared with each other in the case of $\theta=0.67$, they equal 1.343 for wave No. 4, 1.293 for No. 1, 1.250 for No. 2, 1.209 for No. 3, and 1.206 for No. 5.

From this fact, the acutest wave, No. 4, is considered a flow with strong unsteadiness since the pressure waveform factor takes a large value. On the other hand, wave No. 3, simulating the trapezoid or square wave, is considered a smoothed wave. Therefore, ϵ_p is considered a good parameter to denote the distortion in addition to the amplitude of pulsating flows.

4.1 Prediction of turbine characteristics by time-mean method

4.1.1 Characteristics of mass flow rate The results of prediction of turbine mass flow rate by the time-mean method under pulsating flow conditions are shown in Figs. 10-12, which represent the relation-

Fig. 8 Typical waveform No. 5 ($f=10$ Hz, $\theta=0.744$, $\epsilon_p=1.217$)

ship between the pressure waveform factor ϵ_p and the ratio of mass flow rate M_{tm}^*/M_e^* using the waveform as a parameter. Figure 10 shows results for pulse frequency of $f=10$ Hz, Fig. 11 for $f=30$ Hz and Fig. 12 for $f=50$ Hz. In the figures the broken line shows the result of Ref.(7) under the condition of sinusoidal waveform, and the solid line shows the tendency of the present results by the method of least squares. The experiments for sinusoidal waveforms were formerly conducted in order to clarify the basic performance of a radial turbine driven pulsating flows.

Fig. 9 Relationship between θ and ϵ_p ($f=10$ Hz)

Fig. 10 Mass flow rate predicted by time-mean method ($f=10$ Hz)

Fig. 11 Mass flow rate predicted by time-mean method ($f=30$ Hz)

The effects of waveform on prediction of the mass flow rate are recognized clearly as a difference in waveform at the pulse frequency of $f=10$ Hz. But the difference diminishes gradually in accordance with the growth of pulse frequency and is barely existent at $f=50$ Hz. As general trends, the ratio of mass flow rate for each waveform is in proportion to the square of ε_p , and the proportional constant for each waveform becomes large to the same order as that obtained in the relation between ε_p and θ . In Fig. 10, which clearly illustrates the effects of waveform, ε_p values for the same error of prediction are compared with each other. For example, at 10% error, ε_p is the greatest for wave No. 4 and becomes smaller in the order of waves No. 2, No. 3 and No. 5 because of the reduction in unsteadiness. The results of sinusoidal wave produced by the rotary pulse generator locates between waves No. 3 and No. 2.

Moreover, it might be considered that the ratio of mass flow rate for the same value of ε_p indicates the influence of θ on the unsteadiness. As shown in Fig. 7, wave No. 4 has a large amplitude of fluctuation, but θ of wave No. 4 becomes small since the time-averaged velocity is also large. The predicted value M_{tm}^* takes almost the same value for any waveform, because a similar value of the time-averaged expansion ratio has been set at each experiment. Wave No. 4 has a large time-averaged velocity and takes a large M_e^* since the velocity cannot change sharply because of the inertial force. Therefore, wave No. 4 indicates a smaller ratio of mass flow rate than the others for the same value of ε_p in spite of its strong unsteadiness.

The dependence of waveform on prediction of mass flow rate decreases slightly for the pulse frequency of $f=30$ Hz, as shown in Fig. 11. Furthermore in Fig. 12, the dependence becomes negligible at $f=50$ Hz and finally converges into the result of wave No. 3.

On the other hand, throughout the entire range of the experiments, the pulse frequency does not remark-

ably affect the prediction of mass flow rate for any waveform. This result is the same as that in Ref. (7). With increasing pulse frequency, the accuracy of prediction is improved. The main reason for this is that the accuracy of prediction is affected by the velocity amplitude of pulsating flow. Hence, as the pulse frequency increases, the flow becomes steady as if the small fluctuation were superimposed on the mean flow. Finally, all of the predictions approach the result of tranquil wave No. 3 since the difference in waveforms is diminished in the high-frequency range.

4.1.2 Characteristics of power output Figure 13 shows prediction results of power output by the time-mean method. It shows the relationship between pressure waveform factor ε_p and the ratio of power output L_{tm}^*/L_e^* by using the waveform as a parameter in the manner of the mass flow rate. The ratio of power output is evaluated by the value of predicted L_{tm}^* to experimental L_e^* which is obtained by adding the bearing losses to the measurement from the torque meter. Although not shown in a figure, the same tendency is obtained for another pulse frequency. Marked influence of pressure waveform factor and amplitude of flow is rarely observed in the results of Ref. (7) for sinusoidal waveform. This leads to the conclusion that the results of prediction of power output by the time-mean method are not affected by the pulse frequency or amplitude as well as waveform. Therefore, the time-averaged power output L_e^* under unsteady flow conditions is the same as the value of the steady one if the turbine works under the same condition of expansion ratio and corrected turbine speed.

That is, the mass flow rate of the turbine under pulsating flow decreases even if the power output of the turbine is kept constant. The reason for this is considered to be that the turbine operated under nonsteady flow conditions does the much work in the higher-pressure region, then allows the mass flow rate

Fig. 12 Mass flow rate predicted by time-mean method ($f=50$ Hz)

Fig. 13 Power output predicted by time-mean method ($f=10$ Hz)

to decrease.

4.2 Prediction of turbine performance by quasi-steady analysis

4.2.1 Characteristics of mass flow rate

Results of prediction of mass flow rate by the quasi-steady approach under pulsating flow conditions are shown in Figs. 14-17. They show the relationships between pressure waveform factor ε_p (or θ) and ratio of mass flow rate M_q^*/M_e^* using the waveform as a parameter in the manner of the time-mean method.

The quasi-steady values can estimate the measured value with good accuracy until the value of ε_p reaches 1.2. With an increase in ε_p over 1.2, the ratio of mass flow rate becomes slightly underestimated. Since the accuracy of prediction is almost the same for each waveform, it is difficult to discuss the effect of waveform definitively on the basis of the figure.

Using the ratio of velocity amplitude θ , Fig. 17 indicates the same results as shown in Fig. 14 at the condition of $f=10$ Hz. Comparing the ratio of mass flow rate at the same value of $\theta=0.7$, for example, it seems that the ratio has a slight dependence on ε_p , namely, a kind of effect of waveforms. Therefore, wave No. 4, with large distortion, causes a large error

in prediction, and waves No. 1, No. 2, No. 3 and No. 5 follow in order according to the unsteadiness obtained from Fig. 9.

In quasi-steady analysis the dependence of pulse frequency is only slightly observed. The reason why the accuracy of prediction is improved with the increase of f is considered to be due to the reduction of the amplitude of flow.

Throughout a large number of experiments, the prediction error remains lower than 5 or 6%. This value is a little larger than the previous results⁽⁶⁾ for sinusoidal waveforms. For the pulse frequency of $f=10$ Hz, the effect of the ratio of velocity amplitude θ on the flow rate has not been recognized in the case of sinusoidal waveform⁽⁶⁾, but the effect of ε_p is slightly recognized for each waveform in Fig. 14. The conclusive reason for this is unknown, but this is considered to be partially caused by the following facts: the maximum value of the waveform factor of the sinusoidal wave is less than that of the pulse wave and is about 1.28, the waveform has a slight effect on the prediction of the mass flow rate as shown in Fig. 17, and the region of windage state has some influence on the prediction.

Fig. 14 Mass flow rate predicted by quasi-steady analysis ($f=10$ Hz)

Fig. 16 Mass flow rate predicted by quasi-steady analysis ($f=50$ Hz)

Fig. 15 Mass flow rate predicted by quasi-steady analysis ($f=30$ Hz)

Fig. 17 Relationship between θ and mass flow rate predicted by quasi-steady analysis ($f=10$ Hz)

Consequently, it may be said that the influence of waveform on the prediction of turbine mass flow rate is negligible. It is concluded, therefore, that the hypothesis of quasi-steady flow on the prediction of the turbine mass flow rate is valid through the whole range of turbine expansion ratios. In other words, it can be considered that the characteristics of the turbine mass flow rate under nonsteady flow conditions exist instantaneously on the steady performance curves except for the windage state.

4.2.2 Characteristics of power output The prediction results of power output are shown in Figs. 18-20, which represent the relationship between pressure waveform factor ε_p and ratio of power output L_q^*/L_e^* using waveform as a parameter at the pulse frequencies of $f=10, 30$ and 50 Hz, respectively. It seems that the accuracy of measurement of power output L_e^* due to bearing losses is the main factor in the scattering of results.

The effects of pulse frequency and waveform are not remarkable, although the prediction error becomes smaller with increasing pulse frequency for each waveform because of the reduction in the amplitude of flow. The ratio of power output tends to be

underestimated as the pressure waveform factor ε_p increases, but the maximum error is less than 10%. This value is a little larger than the value of the maximum error of the mass flow rate. The reason for this is considered to be that the expansion ratio of steady flow power output at the windage state ($R_L=1.086$) is larger than that of the mass flow rate ($R_M=1.037$). In the present study, the turbine power output and mass flow rate are set to zero at the instant that the expansion ratio is lower than that at the windage state, and then the prediction error of power output becomes large under the conditions of larger pressure waveform factor.

As mentioned above, the turbine characteristics of power output under nonsteady flow conditions also exist instantaneously on the steady flow performance curves except for the windage state. Judging from these results of mass flow rate and power output, it may be said that the turbine having good characteristics under steady flow conditions shows good performance under nonsteady flow conditions, too. This conclusion provides a fundamental idea for the design of an exhaust radial turbine.

5. Limit of Application of Prediction

Applying two methods of analysis to the prediction of turbine performance, the influence of pulse frequency can be neglected in both analyses. In analysis by the time-mean method, the effect of waveform on the mass flow rate appears in the low-frequency range. But the effect of the amplitude of flow does not present a serious problem since the pressure amplitude is treated as the time-averaged value and the time-mean expansion ratio exists in the range of the steady flow performance curve. On the other hand, in quasi-steady analysis, the influence of waveform can be neglected, but it is necessary to take note of the amplitude of pulse flow (fluctuation range of expansion ratio). Namely, in the case of quasi-steady analysis,

Fig. 18 Power output predicted by quasi-steady analysis ($f=10$ Hz)

Fig. 19 Power output predicted by quasi-steady analysis ($f=30$ Hz)

Fig. 20 Power output predicted by quasi-steady analysis ($f=50$ Hz)

sis, the expansion ratio is instantaneously calculated by the fluctuation of pressure and it may extend beyond the range of the steady flow curve. If this phenomenon occurs, the prediction accuracy deteriorates. In particular, it becomes difficult to predict the performance of a turbine operating under the lower expansion ratio.

Namely, turbomachinery has a peculiar working region called the windage state where both the mass flow rate and power output are considered to be zero. The steady flow performance curves have not yet been obtained in the region of expansion ratio lower than the windage state. Turbine behavior in this region has not yet been made clear. Accordingly, it is best not to apply the assumption of quasi-steady flow to this range for prediction of turbine performance at the present time.

Therefore, it is necessary to take care when the quasi-steady analysis is adopted for engine exhaust flow which has a long time period with expansion ratio lower than the windage state. In particular, the prediction of power output is significantly affected because the expansion ratio of windage for power output is higher than that for the mass flow rate.

However, such conditions as that the turbine works under the windage state do not occur except in the case that an engine operates under a full load and at low speed. Actually the exhaust turbine usually operates in the high-speed range of the engine, and it is driven by pulse flow from a multicylinder engine. Therefore, the pulse flow is smoothed and comes to have large averaged flow components: then the gas flow from an engine becomes more gentle than under the present experimental conditions. Consequently, the effect of the windage state does not play an important role in actual engines.

6. Conclusions

In the present investigation of the effect of pulsating waveforms on turbine performance by the time-mean method and quasi-steady analysis, the following results have been obtained.

(1) The prediction of the turbine performance of the mass flow rate and the power output under the quasi-steady assumption is valid regardless of the flow conditions of waveforms, amplitudes and pulse frequencies. Therefore, it may be considered that the turbine under the nonsteady flow condition works along the steady flow performance curve.

(2) The turbine having good characteristics under steady flow conditions realizes good performance under unsteady flows, too. This result provides a fundamental idea for the design of exhaust radial turbines.

(3) The windage state, which is the exceptional working region on steady flow performance curves, has a deleterious effect on the prediction of turbine performance by quasi-steady analysis.

(4) The effect of the waveform appears strongly in the prediction of the mass flow rate by the time-mean method for low pulse frequencies, but it can be neglected in other cases.

(5) The amplitude of pulse flow acts as the most important influence factor in both methods of prediction.

(6) The effect of the pulse frequency is not remarkable in either method and is negligible.

(7) The characteristics under nonsteady flow conditions can be predicted with high accuracy by means of quasi-steady analysis for mass flow rate and by the time-averaged method for power output.

(8) The pressure waveform factor can describe the unsteadiness of flow by taking account of the distortion of the wave as well as the amplitude.

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