EXPERIMENTAL EXAMINATION OF EXHAUST GAS TURBINE FOR TURBO-CHARGER IN UNSTEADY (PULSATING) OPERATION

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Received April 28, 1983.

Summary

In this paper the impact losses on the inward part of the rotary bladed wheel of the turbine in turbo-chargers and the subsequent decrease in the turbine efficiency were examined by the authors. The essence of their experiments was to establish an unsteady (pulsating) state of operation with the no-load turbo-charger by an electric engine operated rotary-valve installed immediately ahead of the turbine in the inlet duct. During the experiments the amplitude and frequency of the pressure change in the pulsating gas flow could be altered. Certain parameters of the pulsating gas flow and the correlation between the flow and the impaired efficiency of the turbine in pulsating state of operation were examined.

Authors were led to the conclusion that the turbine efficiency is impaired due to the pulse of the gas flow. This impaired turbine efficiency was most considerably affected by the amplitude of the pulsating gas flow while frequency of pulse. and a hardly measurable change in efficiency was brought about due to the change in turbine speed.

During the experiments the maximum decrease in turbine efficiency was 4% when the rate of pulse of the gas flow entering the turbine was:

$$\frac{\Delta p}{p_3} = \frac{(p_{\max} - p_{\min}) \cdot 2}{(p_{\max} + p_{\min})} = 0.5$$

During our experiments this was the maximum rate of pulse implemented in the gas flow.

Introduction

Turbo-chargers have recently gained more and more importance. Turbo-chargers in co-operation with internal combustion engines operate under conditions entirely different from those of a jet-propulsion unit of an aircraft or a thermal gas power-plant. The main difference lies in the fact that they have to operate suitably under operating conditions considerably differing from each other.

The operating conditions for the turbine of a turbo-charger are characterized by extremely unsteady processes. Examination of these processes is important, among others, with respect to the progress of the change of charge. But it also has a decisive significance as to co-operation between the compressor and the turbine of the turbo-charger since efficiency of both units may be impaired to a non negligible measure due to unsteady (pulsating) operation.

This paper we hope contribute to the examination of this problem and so the efficiency decrease of the exhaust gas turbine in unsteady (pulsating) operation will be made clearer. The theoretical examination of this problem will be published in a subsequent paper.

Emergence and significance of the problem and the present state of examination

The examination of the exhaust-manifold and supercharging system in internal combustion engines (Fig. 1) is mostly carried out by means of unsteady gas-dynamics [1].

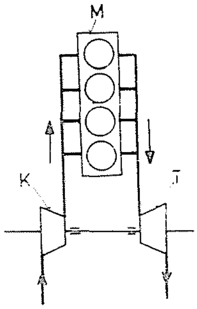

Fig. 1. Schematic diagram of internal combustion engines (M) and the turbo-charging system K: compressor, T: turbine

In determining the power of the turbine and its efficiency, resp., the unsteady (pulsating) operating conditions should be reckoned with. The exhaust gases flow out through the stationary bladed wheel of the turbine at an absolute velocity c_1 of uniform direction ($\alpha_1 \approx$ const.) but changing in magnitude due to the pulse. Consequently, the relative velocity w_1 of the gases entering the rotary bladed wheel will be changed not only in magnitude but in direction (β_1), too due to the constant peripheral velocity u_1 of the rotary wheel. *As shown in Fig. 2, some impact loss is caused on the entering side and consequently on the whole rotary bladed wheel owing to the changing angle β_1. Thus the efficiency of the turbine is impaired by this impact loss. The examination of this fact is aimed at in this experimental test-program.*

It follows from the above that when determining the exact turbine power and/or efficiency, the unsteady state of operation should be reckoned

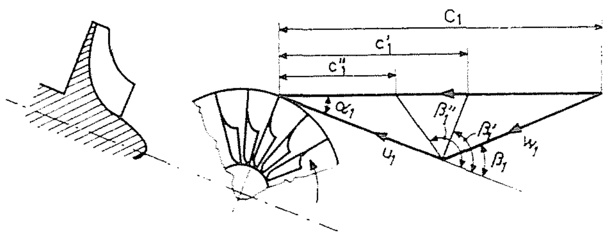

Fig. 2. Flow conditions of the bladed wheel of radial flow turbines: c_1: absolute velocity: W_1: relative velocity; u_1: peripheral speed; α_1: inlet angle of absolute velocity; β_1: inlet angle of relative velocity

with. In such cases the values c_1, W_1 and β_1 cannot be regarded as constant because neglecting the cahanges would result in a grave error when determining the charcteristics for exhaust gas turbines.

The construction of such an unsteady-state (pulsating) experimental model was provoked by the consideration that the changing magnitude of velocity c_1 involves an impaired efficiency of the exhaust gas turbine. We are interested to know what the value or reduction in efficiency might be.

In order to change velocity c_1 and produce the pulse, resp. a rotary-valve operated by an electric motor (Fig. 3 N° 12 and Fig. 4 and 5) was installed ahead of the turbine. In this way the unsteady state of operation was established resulting in an efficiency reduction in the rotary bladed wheel of the turbine. From the experimental results the value of efficiency reduction in the turbine was determined.

This subject was dealt with by many researchers in scientific investigations. The experiments and computations by Mitrohin [2] led to the conclusions that the entrance angle β_1 of the relative velocity W_1 should not be regarded as constant because the neglect of this change would lead to an unaccurate calculation of the turbine power and efficiency.

Bródszky [3] came to the conclusion that the efficiency of the turbine is the worse the greater the measure of the pulse and furthermore, that the knowledge of the change in pressure, temperature and velocity against time is required to the correct determination of ideal and real work of the unsteady inward flow turbine.

According to Zinner [4] the turbine power calculated from steady flow turbine characteristics is always higher than that measured in pulsating operation. This is an affirmation to the fact that the energy used up in a steady inward flow turbine is utilized much better than in a pulsating inward flow turbine.

As a result of experiments at the Research Institute in Harkov [5] concerning the characteristic-curve of co-operation between the supercharged Diesel-engine and the supercharging compressor, it has been stated that the values of pressure ratio in the compressor determined from the results of co-operation with the turbo-charged engine (i.e. in pulsating state of operation) deviate from those received in experiments under steady flow conditions. According to these examinations the rate of deviation depends on the loading conditions of the engine, the intensity of pulse of air entering the compressor, the design properties of the compressor as well as the value of the supercharging pressure.

The medium-speed Diesel engines used in shipping were examined by Peter Boy [6]. In his opinion with an increasing charge some of the manoeuvring properties become impaired to a considerable measure due to the unsteady state of operation. Consequently, he thinks it necessary that the supercharged engines should be examined also under unsteady operating conditions with a view to improve the accelerating and the manoeuvring ability.

As a result of experiments, Wallace and Blair [7] came to the following conclusions: the gas absorbing ability of the turbine was worse with unsteady flow, the decrease in absorbing ability was much more obvious under lower pressure conditions than under higher ones.

As the experimental results of Srimshaw [8] show, the turbine power measured in pulsating operation proved to be reduced more considerably than that received from the calculations respecting the steady flow.

The aim of the present paper is to develop further — though on a small scale — the above results outlined roughly.

Description of the experimental apparatus; the experimental method used in the present examination and the experimental program

Description of the experimental apparatus

Experiments were carried on with the help of a turbo-charger type "Jafi *I*" transformed into a no-load gas-turbine. The experimental apparatus was constructed in the laboratory of the Automotive Engineering Institute of the Technical University, Budapest. (Fig. 3) (Authors would like to express their acknowledgement for the aid offered by the Institute during the experi-

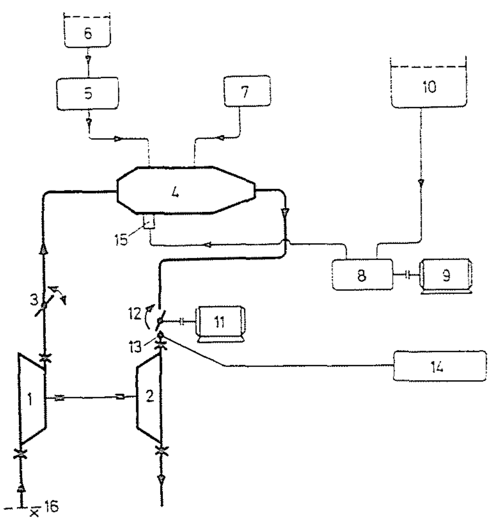

Fig. 3. Scheme of operation of the apparatus. 1. Compressor, 2. Turbine, 3. Throttle-valve. 4. Combustion chamber, 5. Starter fuel-pump, 6. Starter fuel-tank, 7. Devices supplying firing spark, 8. Main fuel-pump, 9. Driving motor for fuel-pump, 10. Fuel tank, 11. Driving motor for rotary-valve, 12. Rotary-valve, 13. Piezo-electric sensor (signalling device), 14. Pressure-impulse recorder device, 15. Spray nozzle, * points for measuring pressure and temperature

mental work.) Since the construction of the apparatus can be clearly seen from the diagram shown in Fig. 3, a detailed explanation is omitted here.

During the experiments especially great attention was paid to the correct and accurate measuring of thermal characteristics of the compressor and the turbine. At each measuring point two or three thermo-elements were spaced (at an angle of 180°—120° from one another) and the values measured in this way were averaged. The inward and exhaust pressure-values both of the turbine and compressor were also measured at two or three points again spaced from each other at an angle of 180°—120°. Mercury and water manometers were applied to measure the pressures. The lub oil temperature of the turbo-charger was kept at 100 C° in order to maintain the mechanical efficiency at a constant and favourable value [9].

The inward air was taken from the surrounding air in the laboratory, of nearly uniform temperature. The speed of revolution of the turbo-charger was measured by an electronic revolution counter. The measuring signals were produced by magnetizing the set-screw of the compressor and voltage was induced in a coil placed at a corresponding distance from it, by the rotation of the compressor. This voltage was transformed into measurable signals by an electronic amplifier. The turbo-charger was started with the help of a standby compressor.

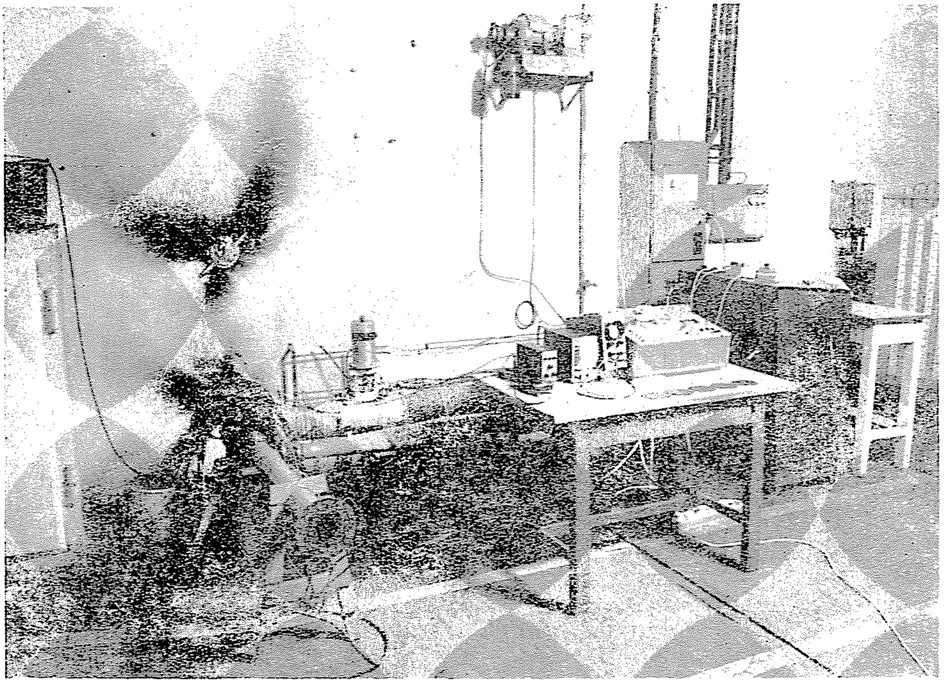

Fig. 4. Experimental apparatus integrated with measuring gauges

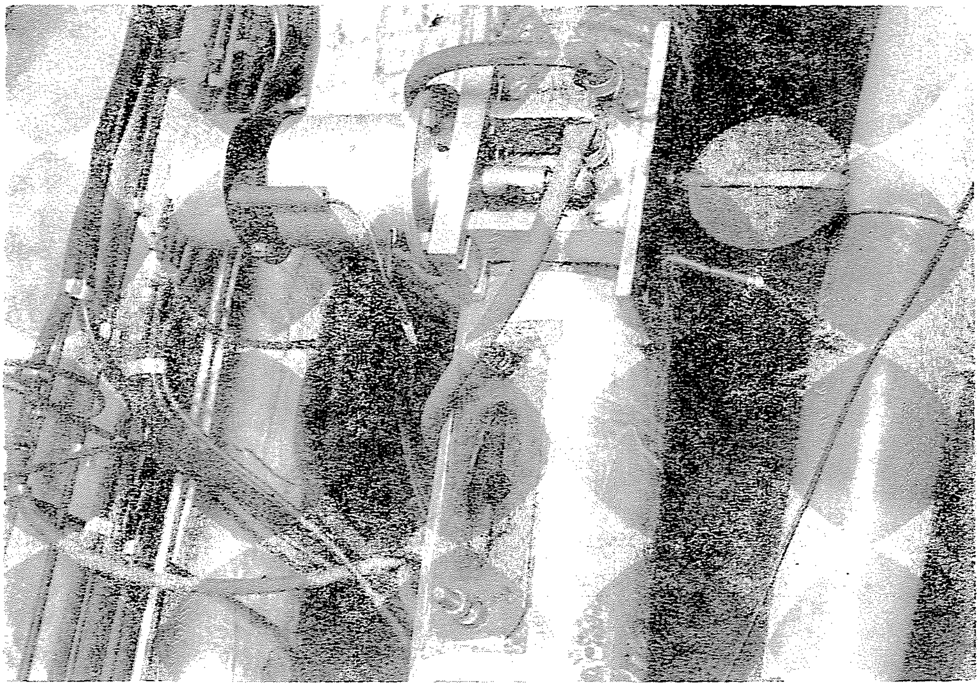

Fig. 5. The removed rotary-valve with its electric motor and a cutout in the pipe ahead of the turbine for installing the rotary-valve

The operating conditions of the turbo-charger were examined at 30.000—55.000 rpm. The steady states of operation at different loading conditions representing the basis for measuring were established by adequately adjusting the throttle-valve behind the compressor. (Fig. 3 N⁰ 3.). The unsteady (pulsating) state of operation was established by inserting a valve ahead of the turbine (12). The valve was rotated with the help of an electric motor assembled there (11). The perspective drawing of the experimental apparatus is shown in Fig. 4, while the photo of the removed rotary-valve can be seen in Fig. 5. The inlet duct ahead of the turbine, into which the rotary-valve was built in (Fig. 5) had a diameter of 120 mm, the diameters of the rotary-valves were in turn 110, 113, 115, 118 and 119 mm, resp. with an aim to establish pulses of different amplitudes. The frequencies of the pulsating gas flow produced by a rotary-valve of variable revolution number were: 25, 40 and 60 Hz, resp.

The pressure change (pulsating) in time was measured by a piezo-electric pressure indicator gauge (13). The electric signals were recorded on a special tape-recorder (14).

Process of measuring, experimental program

a. *Steady-state (non-pulsating) operation*

After checking each element of the apparatus — for the sake of undisturbed and exact measuring — the characteristics of the no-load turbo-charger were measured (Fig. 3 N° 3.) at three throttle-valve positions (0, 1, 2) with a stationary rotary-valve (N° 12). Measuring started with a fully opened throttle-valve (0). The greatest throttle still allowing the operation of the system was provided by the throttle-valve in position (2).

The revolution number of the turbo-charger was adjusted in the interval between 30.000—55.000 rpm, increased with 5000 rpm, and the thermal-flow characteristics of the no-load turbo-charger were measured at such an rpm.

b. *Unsteady-state (pulsating) operation*

As a control, the characteristics of the turbo-charger were recorded at a steady (non-pulsating) throttle, with a stationary rotary-valve and at a set revolution number of the turbo-charger. Afterwards, the electric motor was put into operation rotating a rotary-valve of given diameter and producing a pulse of 25, 40 and 60 Hz*t* resp. All the measurable characteristics were recorded successively as were their variations. Subsequently, the same characteristics were measured at a greater rpm of the turbine. The revolution number of the turbo-charger was also increased by 5000 rpm until 55.000 rpm was reached. Then a major diameter rotary-valve was installed ahead of the turbine and the whole measuring program was repeated, while the turbine speed and the frequency of the pulse were again changed.

The independent variables of the apparatus in the program were the following:

— the speed of the turbo-charger (n 30.000—55.000 rpm)
— the diameter of the rotary-valve (110—119 mm)
— the frequency of the pulse (25—60 Hz)

This combination required about 140 measurements in all, including the basic measurements, i.e. that many points were reached in the course of measuring.

Prior to the evaluation of the measurement results, the results measured were carefully analysed and plotted in a diagram of very large scale, the measurement deviations were compensated, so that only the latter results were processed in the computation program.

Processing and evaluation of the experimental results

The procedure and basic results of the averaging evaluation

The thermal flow characteristics were measure by averaging, except for the frequency and amplitude of the pulse. As a result, the procedure of the basic evaluation was the same both in steady and unsteady state of operation.

The isentropic efficiency of the compressor:

$$\eta_{isk} = \frac{T_1\left(\pi_k^{\frac{\varkappa_l - 1}{\varkappa_l}} - 1\right)}{T_2 - T_1} \tag{1}$$

where: T_1 is the inward air temperature of the compressor [K]

T_2 is the exhaust air temperature of the compressor [K]

$\pi_k = \frac{p_2}{p_1}$: the pressure ratio of the compressor [—]

$\varkappa_l$ is the isentropic exponent of the air flowing through the compressor [—]

The equation for the energy-balance state of the system is:

$$\dot{m}_k \cdot C_{pl} \cdot T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{\varkappa_l - 1}{\varkappa_l}} - 1\right] \frac{1}{\eta_{isk}} = \dot{m}_T \cdot Cp_g \cdot T_3 \left[1 - \left(\frac{p_4}{p_3}\right)^{\frac{\varkappa_g - 1}{\varkappa_g}}\right] \eta_{iST}\, \eta_m \tag{2}$$

where: $\dot{m}_k$, $\dot{m}_T$ is the mass flow of the air and gas, resp. [kg/sec.]

$\pi_T = \frac{p_3}{p_4}$: the pressure ratio of the turbine

c_{pl}, c_{pg}: are the specific heat of the air and gas, resp. at a constant pressure $[J/\text{kg} \cdot K]$

T_3 = the inward gas temperature of the turbine $[K]$

η_{iST} = the isentropic efficiency of the turbine [—]

η_m = the mechanical efficiency of the turbo-charger [—]

In the first approximation the negligible value of η_m was reckoned with in the isentropic efficiency of the turbine and then omitted henceforth.

$\varkappa_g$ = the isentropic exponent of the gas flowing through the turbine: calculated according [9] in literature:

$$\varkappa_g = \frac{0.228 + 0.53 \cdot 10^{-4} \cdot T_3'}{0.158 + 0.53 \cdot 10^{-4} \cdot T_3'} [-] \tag{3}$$

where: T_3' = the average temperature of the gas expanding down the turbine [K]

Since the rotary-valve was installed right ahead of the turbine and was at a sufficient distance from the compressor (Fig. 3), during measurements and their evaluation in the compressor no pulse of a measure was experienced that would have resulted in impaired efficiency. The turbo-charger characteristics and characteristic-curve of operation, resp. are shown in Fig. 6 at different steady throttles and various rotary-valve diameters. During the variation of the amplitude and frequency of the pulse there no change of measurable value was experienced in the characteristics of the compressor due to the conditions mentioned above. So the steady and unsteady state of operation were represented jointly in this diagram. In the diagram the efficiency values (η_{isk}) at different points were also indicated, which equally apply both in steady and unsteady state of operation.

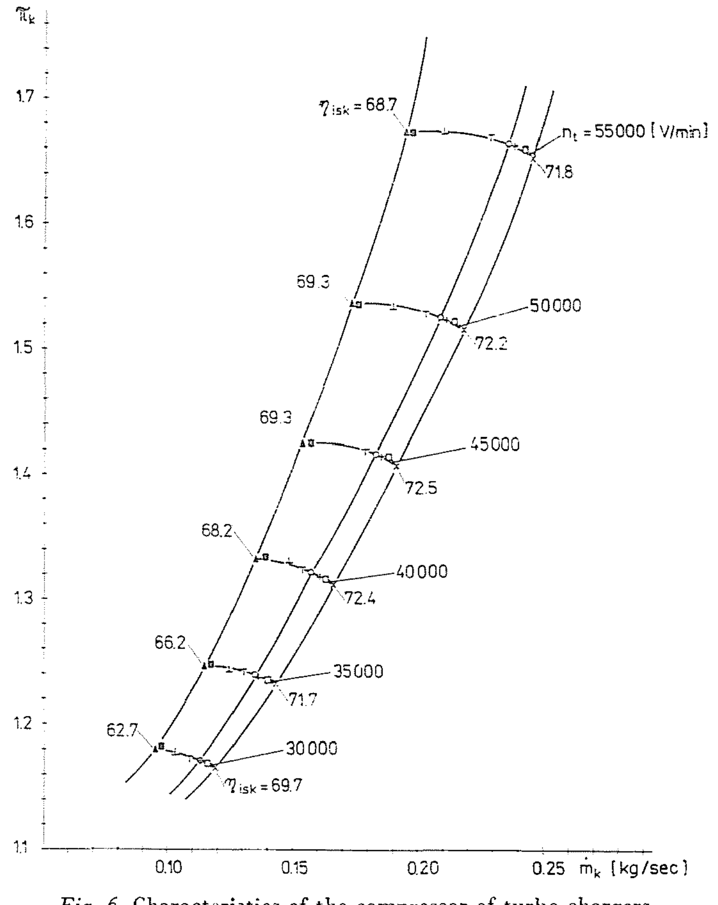

Fig. 6. Characteristics of the compressor of turbo-chargers.

x(0) throttle	□ valve Ø 110 mm.	⊥ valve Ø 118 mm.
0(1) throttle	+ valve Ø 113 mm.	▩ valve Ø 119 mm
⊿(2) throttle	⊤ valve Ø 115 mm.	

In Fig. 7 the characteristics and the characteristic-curve of co-operation, resp. for the turbine in the turbo-charger are shown at different steady throttles and various rotary-valve diameters. Here, too, the values of efficiency at different points (η_{iST}) were indicated. Of course, these values apply only for a steady state of operation.

The turbine characteristics — except efficiency — were affected only in such a slight measure by the frequency of the pulse that it was not plotted in this Figure in order to keep the diagram easy to survey.

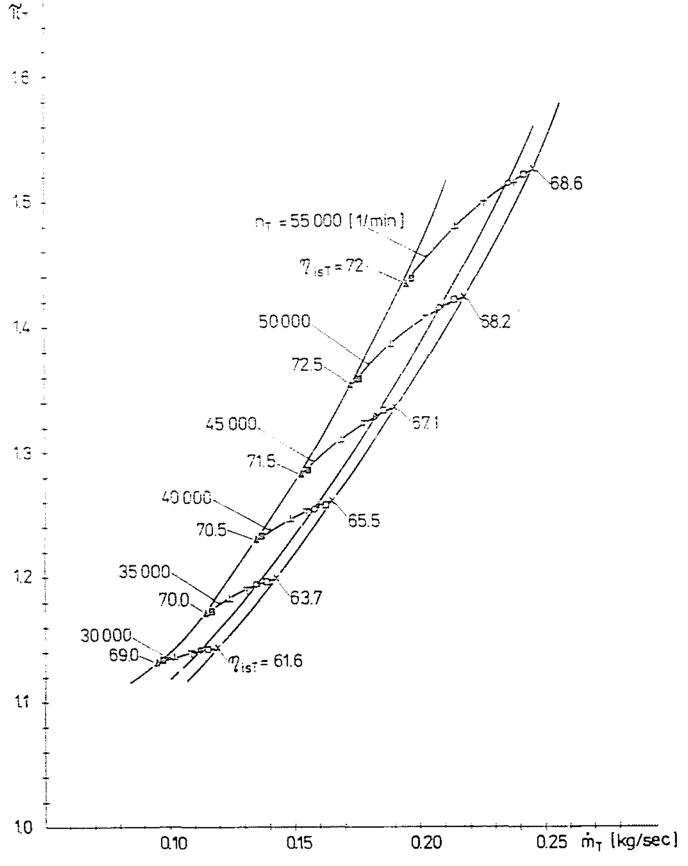

Fig. 7. Characteristics of the turbine of turbo-chargers. Notations used here are the same as those in Fig. 6

Evaluation and fundamental results of the turbine in pulsating state of operation

The procedure to evaluate the experimental results in unsteady state of operation was as follows:

Each point of pulsating operation was determined and the point in the steady state of operation was found which coincided with the point in unsteady state of operation regarding the throttle (Fig. 7). The value of isentropic turbine efficiency interpreted as steady (η_{iST}) was determined for each point. From this the value of isentropic turbine efficiency (η_{instT}) calculated from particular measurement results in unsteady (pulsating) state of operation was subtracted and the remainder was designated by ($\Delta\eta$):

$$\text{i.e.} \quad \Delta\eta = \eta_{iST} - \eta_{instT}$$

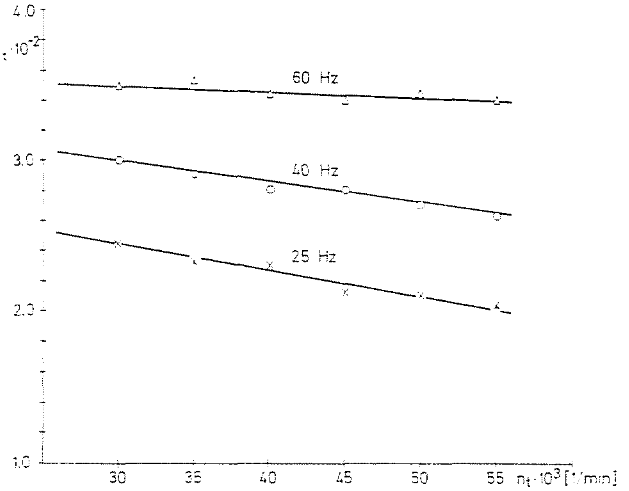

Fig. 8. The change of $\Delta\eta/\eta_{iST}$ as a function of turbine speed at a diameter of 118 mm of the rotary-valve and at avarious frequencies of the pulse

Then the ratio η/η_{iST} was calculated and was represented as a function of turbine speed at each rotary-valve diameter and the various frequencies of the pulse. In Fig. 8. for example. the ratio $\Delta\eta/\eta_{iST}$ is shown as a function of turbine speed at a rotary-valve diameter of 118 mm. The Figure shows that decrease in efficiency was brought about by the increase in frequency but the change in turbine speed did practically not involve a relative change in efficiency.

It is striking that the intensive decrease in efficiency was measured with increasing frequency at a constant turbine speed. This fact cannot be explained by the change in frequency only. From the analysis of the pulse produced by the rotary-valve it turned out (see below) that the amplitude of the pressure-waves changed intensively with the change in frequency. The change in efficiency was brought about in a decisive measure by the change in amplitude as a function of frequency.

The pressure waves recorded on a special tape were re-played after suppressing the non-relevant noise and the frequency (Hz). the amplitude ($\Delta p = p_{\max} - p_{\min}$ of the waves and the ratio $\Delta p/p_3$ were determined $\left(p_3 = \dfrac{p_{\max}+p_{\min}}{2}\right)$

In Fig. 9, by way of example, the values of ratio $\Delta p/p_3$ are shown at a given rotary-valve diameter as a function of turbine speed with different frequencies. Since p_3 is fundamentally constant at a given turbine speed, consequently, its value is unambiguously increasing with increasing frequency though at a decreasing rate. This effect is first of all due to the fact that the time required for the process is decreasing with increasing frequency (i.e.

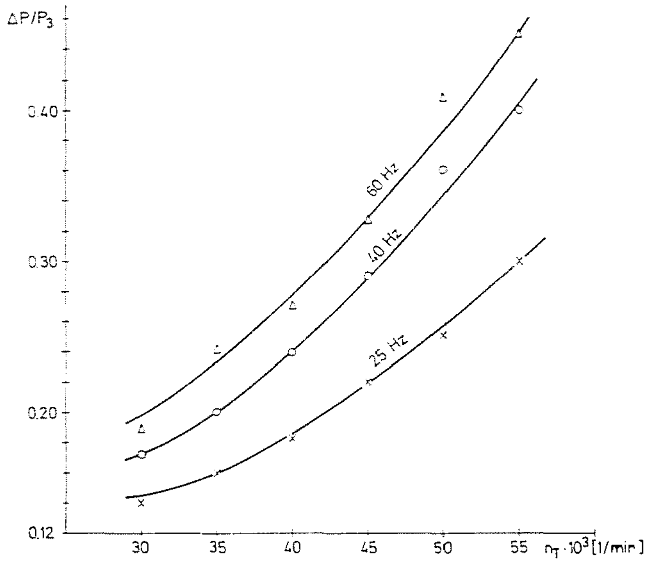

Fig. 9. The change of $\Delta p/p_3$ as a function of turbine speed at a diameter of 118 mm of the rotary-valve at various frequencies of the pulse

with the rotary-valve speed) and thus the acceleration and mass-force (force of inertia) of the air-gas mass and owing to this the developed amplitude (Δp) is increasing as well. The return-flow of the medium can also occur [10], [11], nonetheless the process is considerably influenced by the resistance of medium, too. The analysis of this was not aimed at in this paper, the fact was simply accepted. The purpose of our analysis was to demonstrate the change in turbine efficiency.

The rate of pulse is increasing with an increase of $\Delta p/p_3$; and so is the impact loss in the rotary bladed wheel (Fig. 2). So it is obvious that the efficiency of a pulsating turbine decreases with increasing frequency due to the increasing $\Delta p/p_3$ (Fig. 8).

The values of $\Delta p/p_3$ and $\Delta\eta/\eta_{iST}$ related to the same point of operation were determined. Then the conjugated functions $\Delta p/p_3 = f(\Delta\eta/\eta_{iST})$ were plotted with various turbine speeds and frequencies.

As an example, in Fig. 10 the change of the valves for $\Delta\eta/\eta_{iST}$ is shown as a function of $\Delta p/p_3$ with 50.000 rpm of the turbine and different frequencies.

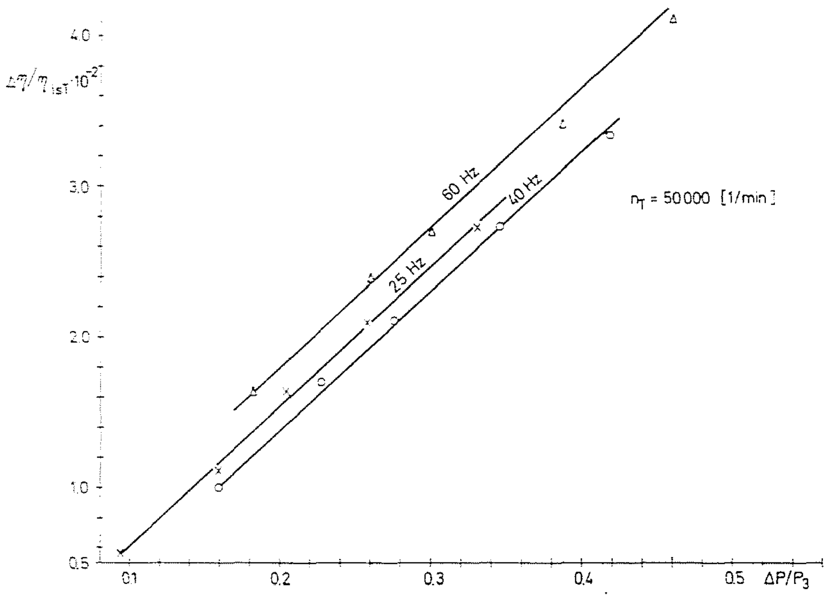

Fig. 10. The change of $\Delta\eta/\eta_{iST}$ as a function of $\Delta p/p_3$ at a given turbine speed and at various frequencies of the pulse

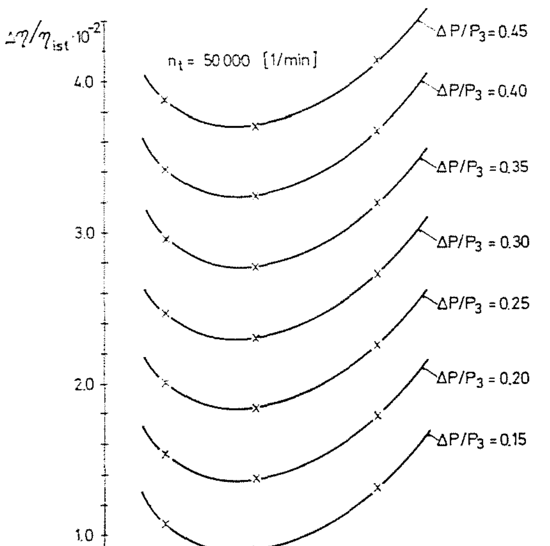

Fig. 11. The change of $\Delta\eta/\eta_{iST}$ as a function of frequency at a given turbine speed and at different $\Delta p/p_3$ values

It can be seen that the efficiency of the pulsating turbine decreases unequivocally at an identical speed, as a function of $\Delta p/p_3$. In this way our fundamental statement has been proved without any doubt. Figure 11 shows the values of $\Delta\eta/\eta_{iST}$ as a function of frequency with different $\Delta p/p_3$ values, as well as at a 50.000 rpm turbine speed. It is obvious that the ratio $\Delta\eta/\eta_{iST}$ is changing only at a relatively small rate at a constant $\Delta p/p_3$ value, as a function of frequency, while a very small minimum is produced. So it can be stated beyond any doubt that the efficiency of the pulsating turbine is not affected substantially by the change in frequency.

Within the frame of this paper all the experimental results cannot be presented but in the final conclusions the results of the whole set of measurements will be evaluated.

In the further part of the research work we will justify the results gained by measurements also theoretically.

Conclusions

The aim of our experimental program outlined in the foregoing was to determine the decrease in efficiency of the exhaust gas turbine in unsteady (pulsating) state of operation as well as to examine the correlation between certain parameters of the pulsating gas flow and the impaired efficiency of the turbine in unsteady state of operation. The experimental apparatus developed from a no-load turbo-charger, was of the type that the characteristics of the compressor were not affected by the pulse ahead of the turbine.

In the course of our examinations we were led to the conclusion that the turbine efficiency gets generally impaired due to the pulse. The decrease in efficiency is determined first of all by the value of $\Delta p/p_3$, i.e. the relative amplitude of the pulsating gas flow. In our experiments the value of $\Delta p/p_3$ produced by this apparatus was about 0.5 at the maximum and this resulted in a 4% decrease in the turbine efficiency. When evaluating this result we must consider that the pulse occurred immediately at the study of the turbine. The efficiency of the pulsating turbine is affected only in a negligible measure by the frequency of the pulse but is slightly impaired with an increasing frequency. The measurements show a max. decrease of 0.4% between 25—60 Hz but this could be seen only after the deviations in the directly measured data were compensated.

The values of $\Delta\eta/\eta_{iST}$ are practically not affected by the change in turbine speed. With increasing speed, the decrease in efficiency of the turbine was slightly diminished, the average value was between 0.3—0.5% along the entire speed range. This result could again be achieved only by a compensation of deviations in the measured data.

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