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# TESTING OF FULL STAGE MIXED-FLOW TURBINE FOR AUTOMOTIVE APPLICATIONS

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**Abstract:** *This paper summarizes experimental results of an aerodynamic performance study carried out on mixed-flow full turbine stage of a exhaust gas turbocharger TCR12 produced by PBS Turbo. The investigation was carried out alternatively on two nozzle geometries. The stage was adapted for a VZLU test rig which is integrated into a cool flow closed-loop wind tunnel. The aim of the experiments was to acquire the stage overall performance in wide range of pressure ratio and rotational speed. The turbine stage outlet conical diffuser efficiency was also studied.*

**Keywords: Mixed-flow turbine, Full stage, Stage efficiency, Diffuser efficiency, Test rig.**

## 1. Introduction

TCR12 exhaust gas turbocharger produced by PBS Turbo is used in a very wide range for the charging of diesel and gas engines. It can be used for engines with constant and pulse pressure turbocharging with power output up to 760 kW. The main goal of the work was to verify data used for complete engine cycles design. An overall performance was measured in a wide range of design and off-design regimes with constant pressure operation. Another objective was a study of the flow field at the outlet of the rotor and in a conical diffuser downstream. Two nozzle rings geometries characterized by two vane stagger angles and the throat area difference of approximately 15 % were compared.

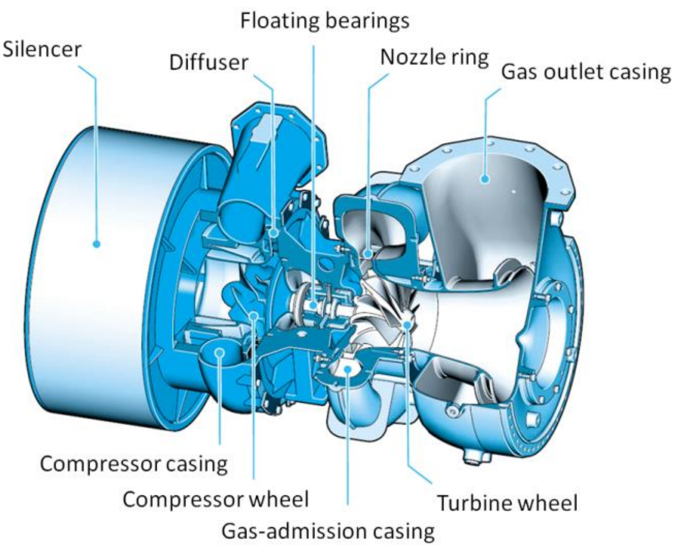


*Fig. 1: PBS Turbo TCR turbo-charger.*

## 2. Test rig configuration

A modular conception of the turbocharger enabled using original turbocharger parts for the test rig configuration. A compressor section was replaced by a dynamometer which loaded the turbine and the

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turbine was driven by VZLU closed-loop wind tunnel. This configuration allowed to set up independently pressure ratio and turbine rotational speed. The work medium was a dry air with inlet total pressure $p_{cin}$ up to 130 kPa and inlet total temperature $T_{cin}$ up to 60 °C.

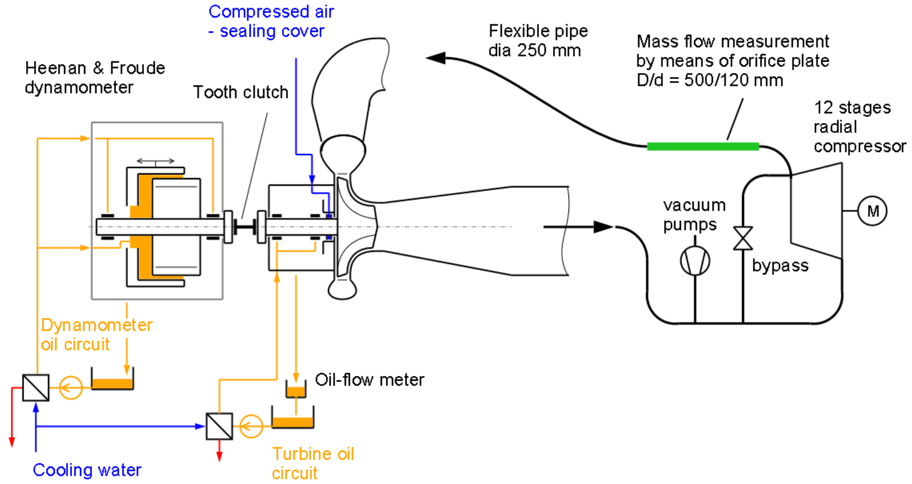


*Fig. 2: The test rig and the wind tunnel scheme.*

A mass flow through the turbine was measured by means of an orifice plate placed far before the turbine inlet. The torque was determined by a force transducer HBM C9B placed on the arm of the high-speed dynamometer Heenan & Froude V-375 – oil as a working fluid. The turbine inlet volute, nozzle and diffuser were fitted with static pressure taps (100 measured positions) and thermocouples (24 positions). Radial distribution of the flow field was measured by a five-hole pneumatic probe at the inlet of the turbine, at the outlet of the rotor and at the outlet of the diffuser.

A bearing oil flow rate, required for bearing losses estimation, was obtained by measurement of the oil level in the tank with a sharp-edged hole in the bottom of the tank.

## 3. Turbine overall performance

The overall performance measurement matrix was defined by hot pressure ratios and hot reduced speeds. It covered range of pressure ratio $\pi$ up to 4.6 and range of speed ratio $u/c_{is}$ from 0.35 to 0.9. Regimes adjustment was done as accurately as possible, thus it wasn't necessary to use data interpolation in a post processing procedures. The accuracy was kept under 0.05 in the case of the pressure ratio and around 50 $min^{-1}$ $K^{-0.5}$ for reduced speed. A comparison of the two nozzle rings geometries results is shown in graphs (Figs. 3 and 4) with mass flow capacity and efficiency characteristics.

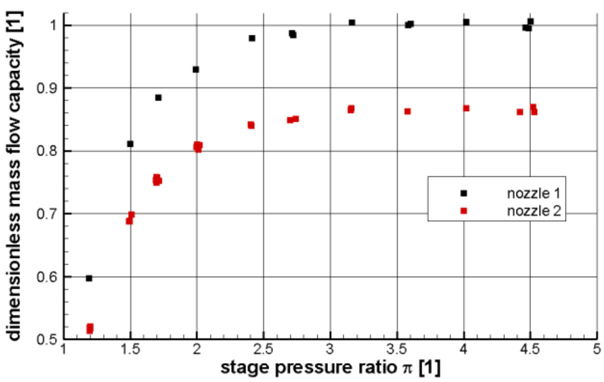


*Fig. 3: Turbine mass flow capacity.*

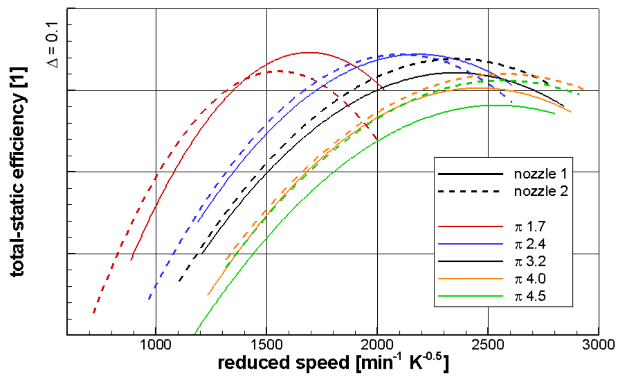


*Fig. 4: Turbine efficiency changes.*

## 4. Hot parameters equation

The measurement was performed under a cool flow conditions with the air, but hot parameters (H subscript) performances with exhaust gas were required. Therefore, correction to hot conditions had to be applied. Following equations were found for the hot parameters calculation. These equations were obtained with a presumption of the equalities of Mach numbers and speed ratios of the cool and hot flow.

$$Q_{tH} = Q_t \cdot \sqrt{\frac{\kappa_H \cdot r \cdot \left(1+\frac{\kappa_H-1}{2}\cdot M_{isS}^2\right)^{\frac{\kappa_H+1}{1-\kappa_H}}}{\kappa \cdot r_H \cdot \left(1+\frac{\kappa-1}{2}\cdot M_{isS}^2\right)^{\frac{\kappa+1}{1-\kappa}}}} \tag{1}$$

$$n_{redH} = n_{red} \cdot \sqrt{\frac{\frac{\kappa_H}{\kappa_H-1}\cdot r_H \cdot \left(1-\pi_H^{\frac{1-\kappa_H}{\kappa_H}}\right)}{\frac{\kappa}{\kappa-1}\cdot r \cdot \left(1-\pi^{\frac{1-\kappa}{\kappa}}\right)}} \tag{2}$$

$$\pi_H = \left(1+\frac{\kappa_H-1}{\kappa-1}\cdot\left(\pi^{\frac{\kappa-1}{\kappa}}-1\right)\right)^{\frac{\kappa_H}{k_H-1}} \tag{3}$$

Nomenclature in equations: $c_{is}$ – stage outlet isentropic velocity, $M_{isS}$ – nozzle outlet isentropic Mach number, $n_{red}$ – reduced rotational speed, $Q_t$ – mass flow capacity, r – specific gas constant, u – rotor tip velocity, κ – specific heats ratio, π – stage pressure ratio

## 5. Diffuser performance

It is a simple conical diffuser which geometry is specified by non-dimensional length $L/d_{in}$ ~ 3.1 and by area ratio $A_{out}/A_{in}$ ~ 2.9. A pressure recovery efficiency ~ 0.63 for such diffuser with uniform inlet flow can be expected.

The diffuser performance is noticeably affected by non-uniform swirling inlet flow. It is demonstrated by measured diffuser efficiency (Fig. 5). The efficiency is much higher than expected in wide range of velocity ratio u/c. It is caused by a positive effect of swirling inlet flow kinetic energy. On the other hand, the strong negative effect of diffuser inlet conditions occurs for extremely overloaded and unloaded turbine. The centrifugal force of the swirling flow causes a huge backflow region in the center of the diffuser in these regimes (see radial distribution of reduced mass flow in Fig. 7).

An increase of the turbine pressure ratio leads to a better diffuser performance, namely in optimal turbine regimes (u/c ~ 0.7). It is shown in Fig. 6 with pressure recovery coefficient distribution along the diffuser length.

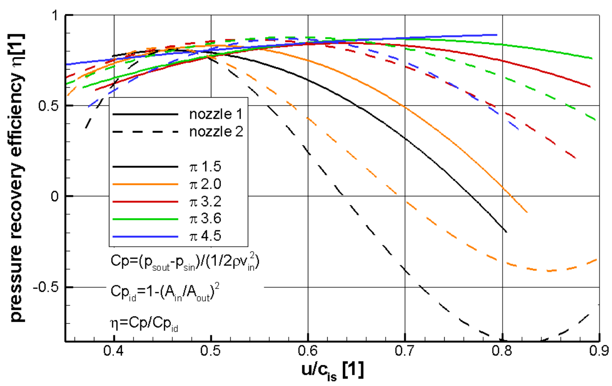


*Fig. 5: The diffuser efficiency characteristic.*

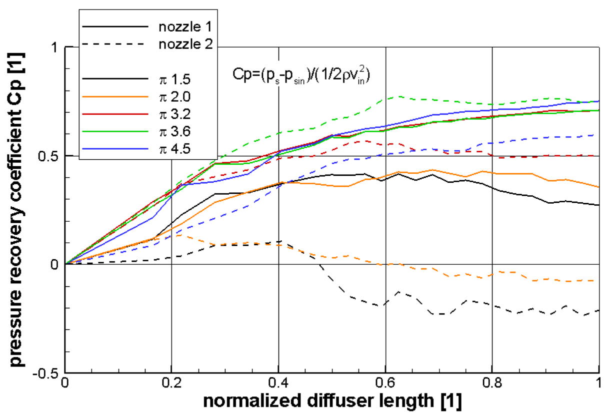


*Fig. 6: Diffuser pressure recovery coefficient distribution – pressure ratio influence.*

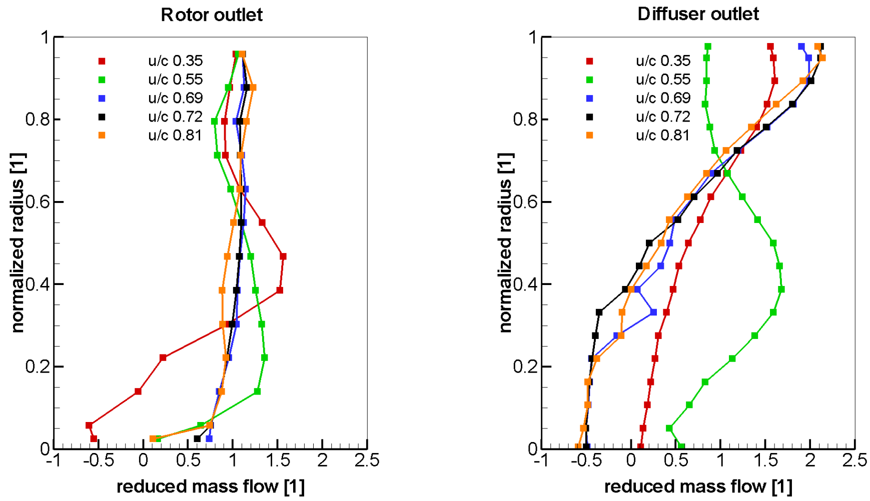


*Fig. 7: Radial distribution of mass flow at the rotor outlet and the diffuser outlet (turbine pressure ratio 2.0).*

## 6. Uncertainties

Uncertainty estimation was done with assumption of Gaussian distribution and confidence interval 95 %.

The most important parameters of the overall performance is the stage efficiency and the turbine mass flow capacity. These parameters are computed from conditions measured - pressure (± 0.1 % of reading), temperature (± 1 K), torque (± 0.13 Nm). As a result, uncertainty of the mass flow capacity is ± 0.7 % of reading in complete range of the measurement. An estimation of total-static efficiency uncertainty is shown in graph in Fig. 8. The efficiency uncertainty shown contains also turbine bearing losses determination uncertainty, which strongly affect the uncertainty level.

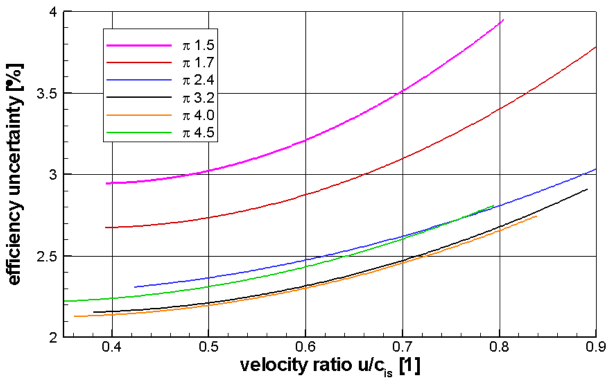


*Fig. 8: Estimated turbine efficiency uncertainty (% of reading).*

## 7. Conclusions

This paper has shown experimental results of the mixed-flow full stage turbine overall performance measurement. The measurement was performed in cool flow turbine test rig. The advantage of such experiment is, that the turbine stage can be equipped with standard instrumentation. Another advantage is independent flow source which ensures capturing of the turbine stage characteristics in wide range of operational conditions including off-design regimes.

The conical diffuser measurement results confirmed that efficiency of the diffuser preceded by turbine stage is completely different in comparison with a diffuser with uniform flow field at the inlet. An effect of the swirling flow on a stability of the diffuser flow field can be positive, but also strongly negative. The diffuser characteristic is strongly affected also by the absolute value of the mass flow (comparison of the nozzle 1 and 2 in Fig. 5 and 6).