TEKA Kom. Mot. Energ. Roln. – OL PAN, 2010, 10, 39–46

SIMULATION ASSESSMENT OF OPTIMISATION POSSIBILITIES OF COOPERATION OF THE TRACTION ENGINE AND THE TURBOCHARGER

Krzysztof Danilecki

Faculty of Automotive Vehicles Operation, The West Pomeranian University of Technology in Szczecin

Summary. The paper presents a selection method of the turbocharger to the engine, which allows for obtaining the maximum efficiency of the supercharging system for the selected engine operation point on the external characteristics. The values of the characteristics of the compressor and the turbine have been determined on the basis of simulation studies using own numerical program. The calculations have been carried out for the SW 680 engine with the factory B4A turbocharger.

Key words: combustion engine, modelling, turbocharging

INTRODUCTION

One of the main requirements when matching a turbocharger to the engine is to ensure proper efficiency and stability in cooperation with the engine, while significantly reducing the temperature of the medium introduced into the cylinder and the work of the charge exchange and increasing the overall efficiency of the engine. This is also facilitated by proper shaping of the charge exchange system [Lisowski 2008 a, Lisowski 2008 b, Mysłowski 2002]. Optimisation of co-operation of the engine and the supercharging device is important both in terms of the fuel consumption and the dynamic and ecological properties of engines, and is particularly important in traction diesel engines operating in the conditions of high excess air. This is shown by the development trends of new designs of charge compression systems [Christmann *et al.* 2005, Mährle *et al.* 2007, Steinparzer *et al.* 2005, Syomin *et al.* 2010]. The crucial factor is the widespread use of the research methods based on reliable mathematical models of flow processes in the engine and the turbocharger [Sorenson *et al.* 2005, Winkler *et al.* 2007, Westin *et al.* 2005, Vavra *et al.* 2009].

The quality of the selection of the compressor is estimated on the basis of the location of points of the engine operation, plotted onto the flow characteristics of the compressor. Such characteristics present a dependence of compressor ratio $\pi_s - f\left(n_{w,cor}, \dot{m}_{s,cor}\right)$ and the compression efficiency $\eta_s - f\left(n_{w,cor}, \dot{m}_{s,cor}\right)$ as a function of air mass flow rate $\dot{m}_{s,cor}$ and the turbocharger speed $n_{w,cor}$ corrected to standard conditions [Wisłocki 1991, Moraal *et al.* 1999]. With the required values of the supercharging pressure and the air flow determined by the demand for air by the engine, the points of the engine operation should be in the characteristics area of the compressor with a high compression efficiency and within an appropriate distance from the pumping limit.

The mathematical description of the turbine [Macek *et al.* 2003, Moraal *et al.* 1999, Wisłocki 1991, Wanszejdt *et al.* 1977] is most often based on the experimentally obtained dimensionless characteristics that are presented in the form of the functional dependence of the scaled mass flow parameter $F_p = \dot{m}_t \cdot \sqrt{T_t^*} / p_t^* = f\left(n_w / \sqrt{T_t^*}, \pi_t^*\right)$ on the turbine speed parameter $n_w / \sqrt{T_t^*}$ $(u_t / \sqrt{T_t^*})$ and the expansion ratio π_t^* and the adiabatic efficiency dependence $\eta_{ad} \quad f\left(n_w / \sqrt{T_t^*}, u_t / c_{ad}\right)$ on the blade-speed parameter u_t / c_{ad} and the parameter of $n_w / \sqrt{T_t^*}$ $(u_t / \sqrt{T_t^*})$. The assessment of the turbine matching to the traction engine based on the available flow characteristics is, however, much more difficult. Such characteristics obtained on flow stands in the conditions of a steady flow of the stream of gas allow for an assessment of the degree of perfection of a particular design. They do not allow, however, for reliable assessment of the quality of their cooperation with the engine. In the case of traction engines, where there are used systems with pulsation supply one must take into account the impact of the pulsatility of the exhaust stream to the change in the efficiency and throughput of the turbine [Macek *et al.* 2005, Westin *et al.* 2005, Vavra *et al.* 2009].

PROBLEM DESCRIPTION

Ensuring of the best working conditions of the turbine operation requires keeping the optimum value of the ratio of the peripheral speed u_t of the turbine rotor to the gas flow velocity through the turbine c_{ad} at the adiabatic expansion $u_t / c_{ad} \quad (u_t / c_{ad})_{opt}$, that should correspond to the maximum turbine efficiency η_t. In the systems with pulsation turbine supply, for assessment of the actual conditions of the turbine operation it is necessary to determine the flow velocity of the exhaust gases stream c_{ad} with taking into account the energy of the pressure pulses supplying the turbine. In such conditions, the average velocity c_{ad-av} significantly differs from the values obtained in a steady flow. The criterion for optimum matching of the characteristics of the turbine with pulsed supply to the engine at the velocity of c_{ad-av} may be presented in the form of the dependence:

$$u_t / c_{ad} \quad (u_t / c_{ad})_{opt}. \tag{1}$$

With a known stream $\dot{m}_t$ and the curve of the variation of the pressure and the temperature of exhaust gases, the velocity c_{ad-av} (for any point of the engine operation determined by the speed parameter value $n_w / \sqrt{T_t^*}$) can be determined on the basis of the averaged, within one working cycle of the engine (720 degrees of crankshaft rotation) value of the stream of the exhaust gas enthalpy [Grodzijewskij 1986, Korczewski 2004]:

$$c_{ad\ av} \quad \frac{2 \cdot R_{sp} \cdot \int_{\alpha\ 0}^{\alpha\ 720} \dot{m}_t \cdot T_t^* \cdot \frac{\kappa_t}{\kappa_t \quad 1} \cdot \left[1 \quad \left(\frac{1}{\pi_t^*}\right)^{\frac{\kappa_t \quad 1}{\kappa_t}}\right] d\alpha}{\int_{\alpha\ 0}^{\alpha\ 720} \dot{m}_t d\alpha}. \tag{2}$$

The impact of the pulsatility of the exhaust gases stream on the parameters of the turbine operation can be presented by the so-called pulsatility coefficients. The values of these coefficients can be determined based on the results of identification research of a particular engine [Kowalczyk *et al.* 1989] or can be taken from the literature [Wanszejdt *et al.* 1977]. The coefficients presented there have been arranged according to the criterion of similarity of the exhaust gases flow in the exhaust pipes of engines with similar design to with specified number of cylinders connected to one outlet pipe.

The coordinates of each point of the engine operation in the characteristics of the compressor correspond to the specified rotational speed of the turbocharger and the value of the turbine speed parameter $n_w/\sqrt{T_t^*}$ $(u_t/\sqrt{T_t^*})$, depending on which the values of the characteristics of the turbine are determined. Thus, the value of the parameter $n_w/\sqrt{T_t^*}$ $(u_t/\sqrt{T_t^*})$, that determines the required conditions of the turbine operation is to be taken as set, depending on the point of the compressor selection. In this case, for the calculated value of the flow parameter F_p the criterion of the optimum matching of the turbine is reduced to compliance with the condition:

$$u_t/c_{ad\ av} \quad (u_t/c_{ad})_{opt}. \tag{3}$$

The appropriate value of the speed u_t can be achieved by changing of the turbine blade diameter D_T, while c_{ad-av} – as a result of the selection of effective turbine flow area A_T. In the case of a single-stage supercharging without regulation, a significant restriction for the optimisation of the parameter $u_t/c_{ad} - c_{av}$ is the condition of not exceeding the permissible value of the supercharging pressure at the rated engine operation point. This condition determines the need to keep a determined value of the effective turbine flow area A_T. Depending on the selection of the turbocharger, compliance with the condition (3) can be obtained at the speed corresponding to the maximum torque, or for the rated engine operation conditions, or at any other point of its characteristics.

SIMULATION RESEARCH

For assessment of the conditions for cooperation of the engine and the turbocharger an own computational model has been used, based on mathematical models of particular components of the system: diesel engine, radial compressor, radial turbine [Danilecki 2007]. The identification of the model coefficients has been carried out on the basis of the results of measurements carried out for the SW 680 engine. The design of the model allows for determination of the average values of the circulation of the engine with a selected turbocharger set for the given load value and the speed. In this model, changes in the turbine power and throughput with pulse exhaust gases flow are taken into account by means of the pulsatility coefficients k_N and k_f [Wanszejdt *et al.* 1977]. The power balance equation for the compressor N_s and the turbine N_t with taking into account the pulsatility coefficient k_N is defined by the dependence:

$$N_s = k_N \cdot N_t. \tag{4}$$

The values of the pulsatility coefficient k_N are determined from the empirical functional dependence $k_N = f(\pi_s, \pi_t)$ from the compression of the compressor π_s and the expansion ratio of the turbine π_t.

Changes of the stream of the exhaust gases mass $\dot{m}_t$ with pulsating flow through the turbine, which are taken into account by the coefficient k_f are determined by the expression:

$$F_p \cdot k_f \quad \frac{\dot{m}_t \cdot \sqrt{T_t^*}}{p_t^*}. \tag{5}$$

The coefficient k_f is calculated from the formula:

$$k_f \quad 1/\sqrt{k_N}. \tag{6}$$

The average velocity of the stream c_{ad-av} is determined from the dependence:

$$c_{ad\ av} = \sqrt{2 \cdot H_t^{ad}}\ [\mathrm{m/s}]. \tag{7}$$

The enthalpy of the exhaust gases at the expansion ratio determined with taking into account the pulsatility coefficient k_T (5) is calculated from the formula:

$$H_t^{ad} = \frac{\kappa_t}{\kappa_t - 1} R_{sp} \cdot T_t^* \cdot \left[1 - \left(\frac{1}{\pi_t^*}\right)^{\frac{\kappa_t - 1}{\kappa_t}}\right] [\mathrm{J/kg}]. \tag{8}$$

To determine the expansion ratio in the turbine $\pi_t = p_t/p_o$ with the calculated stream of exhaust gases (5) the turbine flow characteristics is used $F_p = \dot{m}_t \cdot \sqrt{T_t^*} / p_t^*$. The values F_p are determined depending on the expansion ratio π_t and the turbine speed parameter $u_w / \sqrt{T_t^*}$, while the adiabatic efficiency η_{ad} from the dependence simplified to the form of $\eta_{ad} = f(u_t/c_{ad})$.

Using the developed computational model that allows for determination of the course of the engine and turbocharger characteristics, relevant re-calculations have been carried out for different values of the blade diameter D_T and the flow area A_T of the turbine. For the value of the flow parameter F_p determined from the turbine characteristics, such values D_T and A_T of the parameters have been sought, at which the conditions of the engine supercharging are kept at the point of the compressor selection, and where the optimisation criterion is met (3).

Comparative calculations have been carried out for the SW 680 engine, which is equipped at the factory with the B4A turbocharger, with the compressor rotor with denotation of 270 and the turbine with the blade diameter of D_T = 94 mm, and a effective turbine flow area A_T of 21 cm^2. The co-operation of the engine and the turbocharger has been analysed in the conditions of the external characteristics and the load characteristics. Calculations in the conditions of the external characteristics have been carried out within the range of the rotational speed of 1200 rpm at p_e = 0.974 MPa up to the speed of 2200 rpm at p_e = 0.749 MPa, which correspond to the rated power of the engine. The load characteristics have been determined at the speed of 1400 rpm within the range of variability of the average effective pressure from p_e = 0.55 MPa up to p_e = 0.985 MPa, corresponding to the maximum torque value.

DISCUSSION OF RESULTS

It results from the obtained calculation results (Fig. 1) that in all the considered points of the engine operation, the turbine operates at the values of the parameter u_t/c_{ad-av} lower than the optimum that for the most of small-scale turbochargers designed for traction engines is virtually constant, and is approximately 0.7 [Moraal *et al.* 1999, Mysłowski 2002]. In such conditions, also the efficiency of the turbine η_t is lower than its maximum value. The most

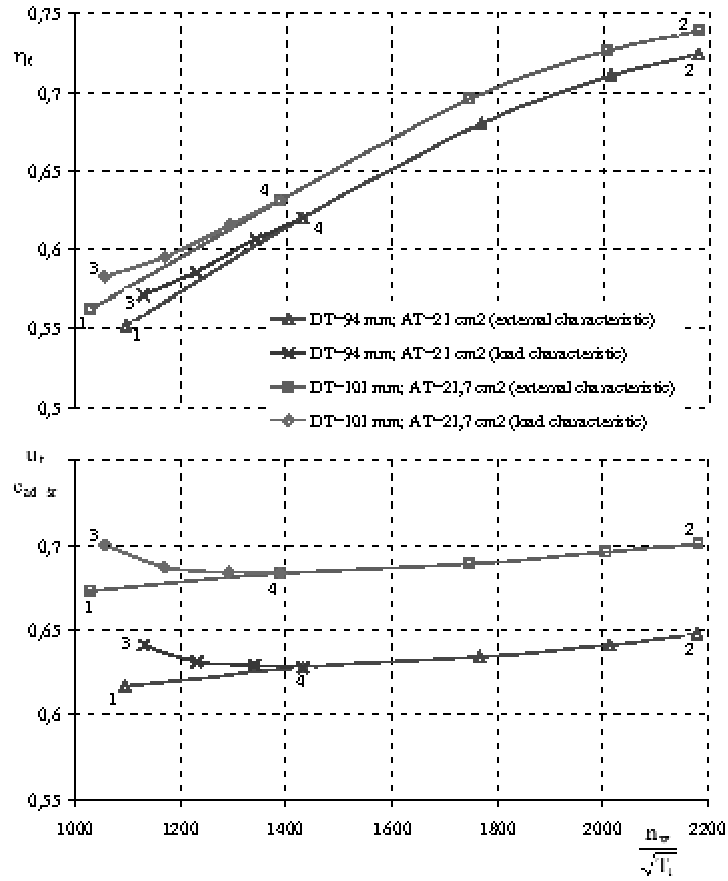

Fig. 1. Changes in the parameter u_t/c_{ad-tr} depending on the blade diameter D_T and the flow area A_T of the turbine of the SW 680 engine in the conditions of the external characteristics and the load characteristics; (1–at n = 1200 rpm at p_e = 0.974 MPa, 2–n = 2200 rpm at p_e = 0.749 MPa, 3–n = 1400 rpm at p_e = 0.55 MPa, 4–n = 1400 rpm at p_e = 0.985 MPa)

favourable conditions of the turbine operation are close to the rated operation point of the engine and in the conditions of the load characteristics (points 2 and 3, Figure 1), where u_t/c_{ad-tr} reaches the highest values. To ensure optimum operation of the turbine both at the rated power and at the partial loads of the engine, one should increase the value of the parameter u_t/c_{ad-tr} from the value of 0.647 up to 0.7. Simple re-calculations show that this can be achieved by increasing the diameter D_T of the turbine rotor from 94 to approximately 102 mm. Such change leads to the desired changes in the peripheral speed of the rotor u_t, while allowing for keeping a constant rotational speed of the rotor and the required value of the parameter $n_w/\sqrt{T_t^*}$. Please note that following the increase in the diameter D_T also the maximum value of the efficiency of the turbine increases, which results in a certain change in the speed value c_{ad-tr}. As a result, this involves a certain correction of the effective turbine flow area A_T. The calculated optimum values of these parameters are: D_T =101 mm, A_T=21.7 cm^2. At the same time, the efficiency of the turbine has increased from 0.724 up to 0.739, while maintaining the supercharging conditions at the rated engine operation point (point 2, Figure

2). As a result of the increase in the diameter D_T and the flow area A_T one can observe shifting of the engine operation points toward the lower values of the compression and the air flow within the range of low and medium speeds and under partial loads (points 1, 3, 4, Figure 2), leading to slight deterioration of supercharging at the maximum torque.

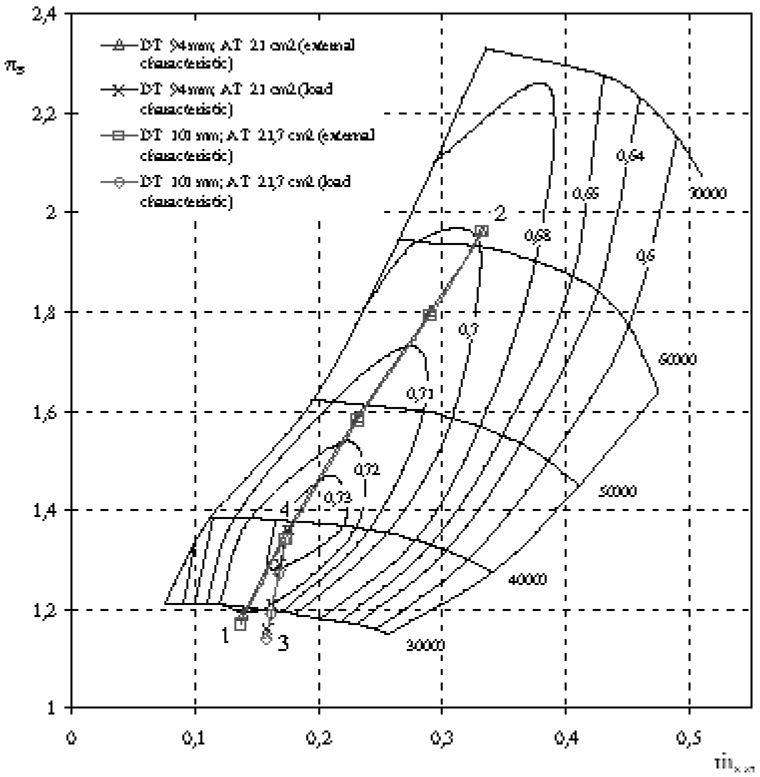

Fig. 2. Changes in the position of the points of the SW 680 engine in the characteristics of the compressor B4A 270 depending on the blade diameter D_T and the flow area A_T of the engine turbine in the conditions of the external and load characteristics, (1–n = 1200 rpm, 2–n = 2200 rpm, 3–n = 1400 rpm at p_e = 0.55 MPa, 4–n = 1400 rpm at p_e = 0.985 MPa)

The performed calculations have shown that the optimisation of the conditions of the turbine operation at the maximum torque of the engine would require an increase in the rotor diameter from 94 up to 103 mm, while maintaining the flow area $A_T = 21$ cm^2. With so selected turbine parameters in the rated conditions, shifting of the engine operation point would take place in the compressor characteristics (point 2, Figure 2) towards higher values of compression, exceeding the imposed limits (increase in the value of the parameter $n_w/\sqrt{T_c^*}$).

CONCLUSIONS

The presented method allows for assessment of the quality of matching the experimentally determined characteristics of the turbine to the flow characteristics of the traction engine. It allows for selection of the turbine design parameters, at which optimum conditions for its cooperation with the engine in the conditions of a pulsating exhaust gases flow will be ensured. Application of this method may be significant both at the optimisation of the existing engines, where the curves of the variations of the exhaust gases pressure are known in the exhaust system, and at the development of new designs, for which the relevant calculations can be performed based on the pulsatility coefficients available in the literature.

REFERENCES

Christmann R., Schmalzl H.-P., Schmitt F., Schwarz A. 2005.: Zweistufig geregelte Aufladung für Pkw- und Nfz-Motoren. Motortechnische Zeitschrift MTZ, nr 1.

Danilecki K. 2007.: Model of turbo-charging system of traction diesel engine. Polish Scientific Society of Combustion Engines. Combustion Engines, nr 3.

Grodzijewskij W. I. 1986.: Metodyka optimalnoj nastrojki turbokompresora dla sowmiestnoj raboty s dwigatieljem. Dwigatjeli wnutrienniego sgoranija, nr 7.

Kowalczyk M., Kozak W., Wisłocki K., Jaskuła A. 1989.: Symulacyjne przewidywanie osiągów silnika wysokoprężnego przy doborze turbosprężarki. Sympozjum Motorcomput '89. Metody obliczeniowe w projektowaniu, badaniu i eksploatacji silników spalinowych. Jadwisin, 24-26.04.1989.

Korczewski Z. 2004.: Zastosowanie metod analizy czasowej i widmowej do identyfikacji procesów gazodynamicznych w układzie spalin wylotowych silnika okrętowego z turbodoładowaniem. Diagnostyka, nr 30 t.I.

Lisowski M. 2008 a.: Analysis of dynamic supercharge influence on economical engine work indexes. Teka Komisji Motoryzacji I Energetyki Rolnictwa PAN oddział w Lublinie, Tom VIII, Lublin.

Lisowski M. 2008 b.: An analysis of fuel consumption by a lorry with the compression ignition dynamic charged engine. Teka Komisji Motoryzacji I Energetyki Rolnictwa PAN oddział w Lublinie, Tom VIIIA, Lublin.

Macek J., Vavra J., Safarik P., Hyhlik T. 2003.: Modelling of a Turbocharger Centripetal Radial Turbine. Journal of Middle European Construction and Design of Cars, 2/2003.

Macek, J., Vitek, O. 2008: Simulation of Pulsating Flow Usteady Operation of a Turbocharger Radial Turbine, SAE Paper No. 2008-01-0295.

Moraal P., Kolmanovsky I. 1999.: Turbocharger Modeling for Automotive Control Applications. SAE Paper 1999-01-0908.

Mährle W., Krauss M., Luttermann Ch., Klauer N. 2007.: High Precision Injection in Verbindung mit Aufladung am neuen BMW Twin Turbo Ottomotor. Motortechnische Zeitschrift MTZ nr 4.

Mysłowski J. 2002.: Doładowanie silników. WKiŁ, Warszawa.

Syomin D., Rogovoy A. 2010.: Power characteristics of superchargers With vortex work chamber. Teka Komisji Motoryzacji I Energetyki Rolnictwa PAN oddział w Lublinie, Tom XB, Lublin.

Steinparzer F., Kratochwill H., Mattes W., Stütz W. 2005.: Der neue BMW Sechszylinder-Dieselmotor mit Stufenaufladung . Motortechnische Zeitschrift MTZ, nr 5.

Sorenson S. C., Hendricks E., Magnusson S., Bertelsen A. 2005.: Compact and Accurate Turbocharger Modelling for Engine Control. SAE Paper 2005-01-1942

Winkler N., Angstrom H. E. 2007.: Study of Measured and Model Based Generated Turbine Performance Maps within a 1D Model of a Heavy-Duty Diesel Engine Operated During Transient Conditions. SAE Paper 2007-01-0491

Westin F., Angstrom H. E. 2005.: Calculation Accuracy of Pulsating Flow Through the Turbine of SI-Engine Turbochargers – Part 1: Calculations for Choice of Turbines with Different Flow Characteristics. SAE Paper 2005-01-0222

Vavra J., Macek J., Vitek O., Takats M. 2009.: Investigation of Radial Turbocharger Turbine Characteristics under Real Conditions. SAE Paper 2009-01-0311

Wanszejdt W.A., Iwaczenko N. P., Kollierowa Ł. K. 1977.: Dizeli. Sprawocznik. Maszinostrojenije, Moskwa.

Wisłocki K. 1991.: Systemy doładowania szybkoobrotowych silników spalinowych. WKŁ. Warszawa.

SYMULACYJNA OCENA MOŻLIWOŚCI OPTYMALIZACJI WSPÓŁPRACY SILNIKA TRAKCYJNEGO I TURBOSPRĘŻARKI

Streszczenie. W pracy przedstawiono sposób doboru turbosprężarki do silnika, który pozwala na uzyskanie maksymalnej sprawności układu doładowania dla wybranego punktu pracy silnika na charakterystyce zewnętrznej. Wartości charakterystyk sprężarki oraz turbiny wyznaczono na podstawie badań symulacyjnych przy wykorzystaniu własnego programu numerycznego. Obliczenia przeprowadzono dla silnika SW 680 z fabryczną turbosprężarką B4A.

Słowa kluczowe: silnik spalinowy, modelowanie, turbodoładowanie