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THERMODYNAMIC OPTIMISATION OF DIESEL-ENGINE TURBOCHARGING SYSTEM PART I

The authors present an optimisation method, based on the thermodynamic consideration and applied to the inlet and exhaust systems of turbocharged engine. The goal function in this method is defined as a sum of exergy irreversible losses – occurring in the whole flow path. The decision variables, optimisation parameters and, also, the constraint conditions in the discussed method are defined and determined. The validation results of specially written and unique programmes, used for flow simulations in the analysed systems, are also presented. The optimisation results, based on the discussed method and related to a specific turbocharged engine are discussed in part II.

NOMENCLATURE

$\dot{B}$	–	exergy flux,
$\Delta\dot{B}$	–	external loss of exergy flux,
$\delta\dot{B}$	–	internal loss of exergy flux,
$\Sigma\delta\dot{B}_i$	–	a total irreversible exergy loss,
c_p	–	specific heat at constant pressure,
c_v	–	specific heat at constant volume,
D	–	pipe diameter,
F	–	surface area,
$f_F = \delta(\ln F)/\delta x$	–	a term characterising pipe area change,
$G = \lambda_f \rho u \lvert u \rvert / 2D$	–	friction force term,

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L	–	pipe length,
$\dot{L}$	–	work flux, power,
m	–	mass,
$\dot{m}$	–	mass flux,
p	–	pressure,
$\dot{Q}$	–	heat flux,
$q = \dot{Q}/m$	–	heat flux per unit mass,
R	–	gas constant,
s	–	specific entropy,
S	–	entropy,
$\Delta\dot{S}$	–	increment of entropy flux,
T	–	temperature,
t	–	time,
u	–	flow velocity,
V	–	volume,
x	–	axis co-ordinate (m),
Y	–	criterion-goal function;

Greek symbols

$\Delta\alpha$	–	valve overlap angle,
η	–	efficiency,
κ	–	ratio of specific heats,
λ_f	–	friction number,
Λ	–	degree of reaction,
ω	–	angular velocity, weight,
π	–	pressure ratio function,
ρ	–	density.

1. Introduction

The recent inverstigation on Diesel engine turbocharging system improvements are conducted with the growing use of computers. Current models for computer simulation of problems in such a system could predict the engine working conditions with good accuracy. This facilitates the analysis of the influence of important design parameters on the turbocharging system functioning (see, for example, [1] to [4]). Attempts to optimise these systems, undertaken by researcher groups working in this field, have been based, however, on different – not universal – assumptions and procedures. Exergy balance method in a form presented, for example, in [5] through [8] gives an incomplete insight into the basic reasons for the system ineffective operation due to the fact that the irreversible exergy losses are derived from the balance equation for exergy and constitute the closing term for that equation.

In the method described in the work [9], a ratio of the pressures at the compressor outlet and turbine inlet was used as the basic optimisation criterion. Maximisation of this quantity led the researchers to obtaining optimum characteristics of the turbocharging system. However, such an approach makes it impossible to consider all characteristic system quantities, like, for example, air handling losses in the inlet manifold and in the inlet ports. Moreover, the above criterion gives no foundation for giving prognoses about the possibility of improving the operation of the important system nodes, as it contains no description of the losses.

The above mentioned weaknesses are not present in a thermodynamic optimisation routine known as entropy generation minimisation method, EGM [10]. Concentrating on finding out the origins of internal irreversibilities that occur in constructions, the entropy generation minimisation method makes it possible to find out such a configuration of various systems and to chose design parameters, which ensure the minimum irreversible losses and thus the maximum possible system efficiency [2]. Therefore, the EGM method is an extension to the exergy method in which the minimum entropy generation is equivalent to the minimum irreversible losses. A review of the literature shows that this method is very useful in studying processes and finding out optimum solutions for the system design or process arrangement, which has been shown, for example, in Refs. [11], [12].

In the current study, a methodology is formulated for thermodynamic optimisation of the charge exchange system (the inlet and exhaust subsystems) of turbocharged engines by using the EGM method. A goal-function is implemented, defined as the sum of all irreversible energy losses arising in the considered system. A set of decision variables and optimisation parameters is determined, which describe the turbocharging system operation in a unique way. Constraint conditions, essential for the system operation and resulting from its interaction with the surroundings, are also presented.

2. Irreversible losses of exergy in the turbocharging system

The assumptions underlying the exergy model, as well as the turbocharging system model itself, are presented in the Ref. [13] and in the thesis [14]. The exergy balance equation for the considered model of Diesel engine turbocharging system is as follows:

$$\dot{B}_{inl} = \Delta\dot{B}_{ext} + \dot{B}_{out} + \dot{L} + \dot{\pi} \cdot T_0, \tag{1}$$

where: $\dot{B}_{inl}$ – flux of exergy at inflow to the system,

$\Delta\dot{B}_{ext}$ – external loss of the exergy flux,

$\dot{B}_{out}$ – exergy flux at outlet from the system,

$\dot{L}$ – mechanical work flux,

$\dot{\pi}$ – the sum of internal entropy irreversible increments.

According to the adopted methodology, only the $\dot{\pi} T_0$ and $\Delta\dot{B}_{ext}$ terms are considered in the following study.

The external exergy loss, $\Delta\dot{B}_{\ \ t}$, comprises the losses associated with the heat transferred out to the surroundings. These processes run in the cylinder head ducts, exhaust manifold, turbocharger, intercooler and are also connected with the exhaust waste heat loss.

The sum of internal entropy irreversible increments, $\dot{\pi}T_0$, comprises the losses arising in the air filter (indexed with AF), compressor impeller (CR), compressor diffuser (CD), intercooler (IC), inlet manifold (IM), inlet ports (IP), exhaust ports (EP), exhaust manifold (EM), turbine guide vanes (TV), turbine rotor (TR) and in the wastegate (WG).

The above losses appear at the right hand side of the following equation:

$$\dot{\pi}T_0 = \sum_i \delta\dot{B}_i = \delta\dot{B}_{AF} + \delta\dot{B}_{CR} + \delta\dot{B}_{CD} + \delta\dot{B}_{IC} + \delta\dot{B}_{IM} + \delta\dot{B}_{IP} + \\ + \delta\dot{B}_{EP} + \delta\dot{B}_{EM} + \delta\dot{B}_{TV} + \delta\dot{B}_{TR} + \delta\dot{B}_{WG}. \tag{2}$$

In practice, the losses $\delta\dot{B}_{AF}, \delta\dot{B}_{CR}$ and $\delta\dot{B}_{CD}, \delta\dot{B}_{IM}$ and $\delta\dot{B}_{IP}$, $\delta\dot{B}_{EP}$ and $\delta\dot{B}_{EM}$, as well as $\delta\dot{B}_{WG}, \delta\dot{B}_{TV}$ and $\delta\dot{B}_{TR}$ are grouped into four elements (it is also well known that the $\delta\dot{B}_{IM}$ and $\delta\dot{B}_{EM}$ losses do not exceed the $\delta\dot{B}_{EP}$ and $\delta\dot{B}_{IP}$ losses by more than 10%). Therefore, relation (2) is simplified and takes the form:

$$\dot{\pi}T_0 = \sum_i \delta\dot{B}_i = \delta\dot{B}_C + \delta\dot{B}_{IC} + \delta\dot{B}_{IN} + \delta\dot{B}_{EX} + \delta\dot{B}_T, \tag{3}$$

where the consecutive indexes denote the cumulative losses arising in:

C – compressor,

IC – intercooler,

IN – inlet manifold and ports,

EX – exhaust manifold and ports and

T – turbine.

The results of an exemplary exergy balance are depicted in Fig. 1 (a) and (b). This is a balance for the inlet and exhaust system in a turbocharged Diesel engine. The irreversible losses that account for about 1/4 of the total balance (a) are divided in (b) into parts. The main irreversible losses arise in the compressor, turbine, inlet and exhaust ports and in the manifolds. It can be observed that the intercooler loss is less significant.

The internal losses that arise in each subsystem are located in a hypothetical source-nodes for the purpose of performing calculations. These nodes are placed at the outlet boundaries of each subsystem.

All the mentioned above losses are described by the Gouy-Stodola theorem that takes the following form for the elements of the considered system:

$$\delta\dot{B}_N = T_0 \cdot \Delta\dot{S}_N, \tag{4}$$

where N designates the respective term in Eq. (3) and $\Delta\dot{S}_N$ is the irreversible entropy increment flux.

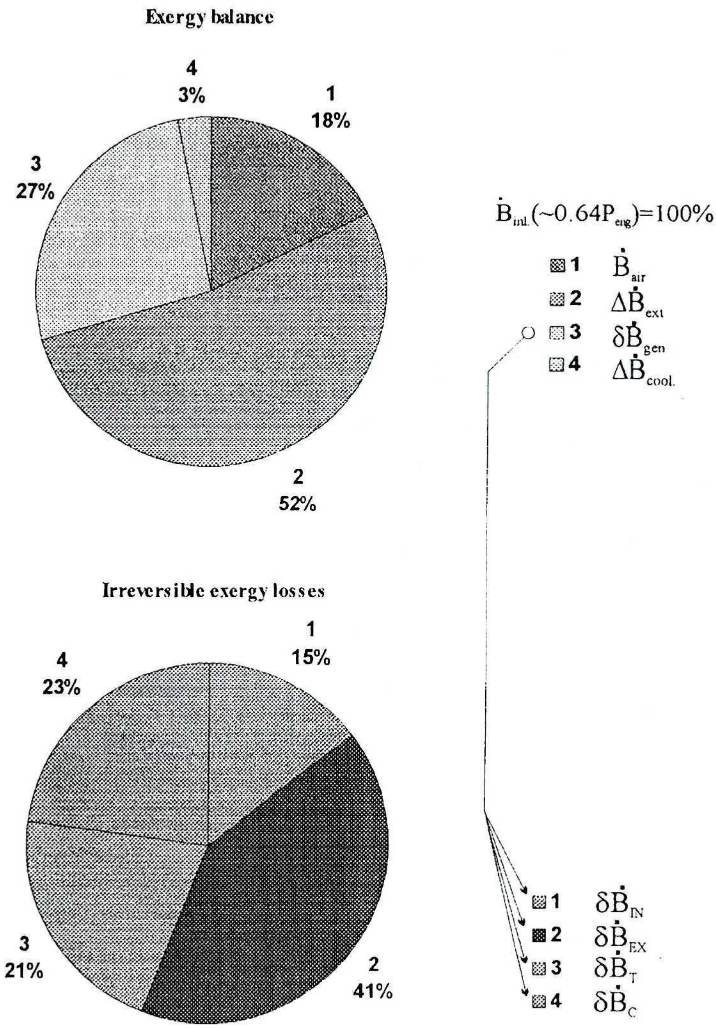

Fig. 1. Exergy balance and irreversible exergy losses arising in the inlet and exhaust system of a turbocharged Diesel engine (according to Ref. [15]):

Exergy fluxes:

$\dot{B}_{inl}$	– inlet to the system,	$\delta\dot{B}_{gen}$	– internal irreversible losses attributable,
$\dot{B}_{air}$	– charging air,	$\delta\dot{B}_{IN}$	– air manifold and inlet ports,
$\Delta\dot{B}_{ext}$	– waste heat loss,	$\delta\dot{B}_{EX}$	– exhaust manifold and ports,
$\Delta\dot{B}_{cool}$	– heat transfer losses,	$\delta\dot{B}_T$	– turbine,
P_{eg}	– engine power,	$\delta\dot{B}_C$	– compressor.

3. Definition of quantities essential in turbocharging system optimisation

3.1. Criterion-goal function

According to the Gouy-Stodola theorem, Eq. (4), a minimum entropy generation in the system is accompanied by minimum internal irreversible losses of exergy. Thus, optimisation of the inlet and exhaust system in a turbocharged Diesel engine, considered in this paper, can be performed equivalently to the EGM method by minimising the irreversible exergy losses. A validity of such an approach in case of heat and flow systems is proved in many papers [10], [11]. In the case of the considered turbocharging system, like in the case of other heat and flow systems, minimisation of the internal entropy generation can lead to improvement of its functional characteristics. The system efficiency should rise and, in general, a greater mass of air can be awaited to flow into the engine cylinders, which is the goal of turbocharging the engine. Moreover, the efficiency of the entire turbocharged engine system can be expected to rise.

The main criterion of the mentioned method of finding an optimum solution to the system design and operating conditions is apparent in its name. This is entropy production or the sum of all irreversible exergy losses, denoted as $\dot{\pi}$ or $\sum_i \delta\dot{B}_i$.

The criterion-goal function is defined as the total irreversible exergy loss arising in the entire system, averaged over the single working cycle of the engine, evaluated for specific engine torque and rotational speed. The function is represented by the following formula:

$$Y_k = \frac{1}{t_{c,k}} \int_0^{t_{c,k}} \left(\sum \delta\dot{B}_i \right) dt, \qquad (5)$$

where $\delta\dot{B}_i$ – the internal irreversible exergy loss produced by the i-th source described above,

$t_{c,k}$ – the time period of a single engine working cycle at the engine speed n_k.

It is assumed that the nominal engine speed is within the range $<n_M, n_N>$, where n_M is the peak torque speed and n_N is the engine nominal power speed. Computations of the engine system performance are done at several engine rotational speeds. Then, the goal function is given by the following weighted-mean value

$$Y = \sum_k \omega_k Y_k, \qquad (6)$$

where ω_k are the weights related to the instantaneous values of engine rotational speed and torque.

3.2. Decision variables and parameters of the turbocharging system optimisation

In the optimisation problem formulated above, system operation is described by the set of non-linear partial differential equations, referred to as gas dynamic equations, which according to Ref. [16] are written as follows:

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u)}{\partial x}=-\rho u f_F, \tag{7}$$

$$\frac{\partial}{\partial t}(\rho u)+\frac{\partial}{\partial x}(\rho u^2+p)=-\rho u^2 f_F-G, \tag{8}$$

$$\frac{\partial}{\partial t}\left\{(\rho F dx)\left[c_V T+\frac{u^2}{2}\right]\right\}+\frac{\partial}{\partial x}\left\{(\rho u F)\left[c_V T+\frac{u^2}{2}+\frac{p}{\rho}\right]\right\}dx=\rho q F dx, \tag{9}$$

where: t – time,

x – axis co-ordinate,

ρ – density,

u – flow velocity,

p – pressure,

c_V – specific heat at constant volume,

D – pipe diameter,

F – surface area,

$f_F = d(\ln F)/dx$ - a term characterising pipe area change,

$G=\lambda_f \rho u|u| 2D$ – friction force term,

λ_f – friction number,

$\dot{Q}$ – heat flux,

$q=\dot{Q}/m$ – heat flux per unit mass.

Some of the characteristic design and operational parameters of the system are given explicitly in these equations. These are: thermodynamic parameters, duct geometry, or friction factor. There are also variables, determining the structure and functioning of the charge exchange system in a turbocharged Diesel engine that are given in an implicit way. This is the case of compressor and turbine, which are included in the calculations after converting their flow data into useful formulae by using the method of characteristics. A full functional description of these flow machines can be found, for example, in the monograph [16].

Following the analysis of the system design and operation, quantities adopted for the so called decision variables are listed below:

- time diagram of the exhaust or inlet valve,
- valve overlap period,
- turbine equivalent area,
- turbine A/R ratio, i.e. the ratio of the cross-sectional area of the casing and the distance of the centroid of the area from the turbine axis,
- maximum cross-sectional area of the waste gate valve,
- inlet manifold volume,

- compressor equivalent area,
- inlet system pipe length,
- temperature decrease in intercooler.

The following optimisation parameters are considered in this work:

- engine rotational speed,
- thermodynamic parameters, i.e. temperature, pressure and density, of exhaust gas contained in the engine cylinder at the moment of exhaust valve opening, equivalent to adopting the specific value of engine torque,
- thermodynamic parameters of the surrounding air.

3.3. Constraint conditions of the turbocharging system optimisation

Three constraint conditions have been found for the analysed system, which cannot be overstepped in the optimisation procedure. They follow from the necessity to maintain the unchanged quality of the working process of the turbocharged engine with altered design, as compared to the quality of the same engine with original design arrangement.

The first condition results from the need of maintaining the turbocharged engine torque at unchanged level in given operating conditions. This means that the engine cylinders have to be fed with the amount of air, which is not less than the cylinder charge of original design arrangement. A limit value for the air mass, following from the above, is

$$m_c \geq m_{c0}, \tag{10}$$

where m_{c0} – a total mass of air charged into all engine cylinders during a single engine working cycle.

The second condition pertains to the in-cylinder work done during the charge exchange period. This work is performed by the piston during the exhaust process and by the air compressed in compressor during the charging process. It follows from the above that a constraint must be put on the cylinder charge-exchange work to ensure that the total cycle work is not decreased, when the design arrangement has been altered:

$$L_{tl} \leq L_{tl0}, \tag{11}$$

where:

$L_{p0} = \sum_{i=1}^{nc} \left(\int_{V_{evo}}^{V_{ivo}} p_{ci} dV_i \right)$ – designates the portion of the cylinder charging work done in the original system design arrangement,

n_c – No. of engine cylinders,

V_{evo} – cylinder volume at the moment of exhaust valve opening,

V_{ivo} – cylinder volume at the moment of inlet valve closure,

V_i – instantaneous volume of the i-th cylinder, positive when increasing and negative when decreasing,

p_{ci} – instantaneous cylinder pressure.

The third condition follows from the exergy analysis of the system and accounts for the impact of external exergy losses on the effectiveness of the charge exchange system in a turbocharged Diesel engine. The exhaust waste-heat renders the main exergy loss and this cannot be neglected in the system optimisation considerations. Thus, it is necessary to put a constraint on that loss, which is dependent on the mass flowrate and thermal parameters of the exhaust gas, and design variables of the system. The condition can be written as

$$\Delta\dot{B}_{ext} \leq \Delta\dot{B}^{0}_{ext}, \tag{12}$$

where $\Delta\dot{B}^{0}_{ext}$ – the exhaust waste-heat loss.

4. Control tests and validation of programmes used in the optimisation analysis

Computer simulations of gas flow described by Eqs (7)÷(9) were realised by using a numerical code, specially adopted for this purpose, based on the finite element method. The main features of the program are outlined in the papers [17], [18]. The code uses some procedures of the method of characteristics reported in the works [16], [19]. They facilitate proper treatment of boundary conditions and also modelling of the gas flow inside a pipe. The program is divided into modules and the calculations can be performed with the use of the FE method or, alternatively, the method of characteristics, but in both cases the boundary conditions are solved by using the latter approach.

The code had been verified in a number of control tests before it was used for the numerical experiment. The results generated by the developed numerical method have been compared to results obtained from other methods known from the literature, and experimental measurements. Only the most significant parameters are presented in this paper. The comparison of the Author's own simulations with the literature data pertains to the variations in course of chosen quantities with time and is shown in Figs. 2 and 3.

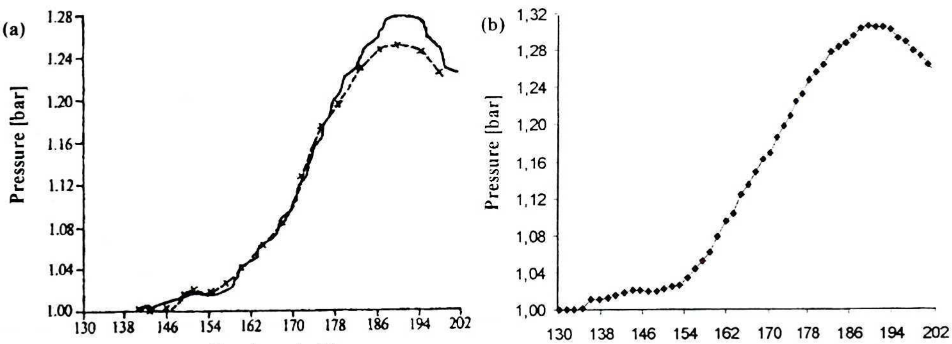

Fig. 2. Pressures at the nozzle in the exhaust pipe of one-cylinder engine calculated by using: (a) the graphical and numerical method of characteristics [16], (b) the developed code

Figure 2 shows comparison of the pressures in the system comprising cylinder, exhaust valve, pipe and nozzle. The ratio of the effective cross-sectional area of nozzle and pipe is 0.25. The test data were adopted from the Ref. [16], p. 171.

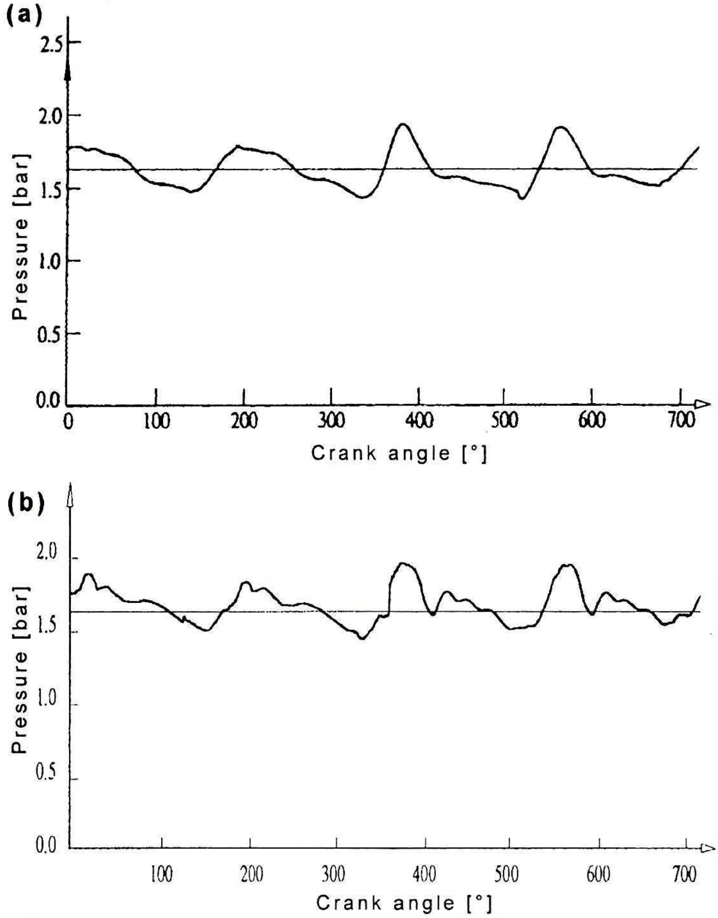

Fig. 3. Comparison of pressures in exhaust pipe of the 4-cylinder, turbocharged Diesel engine calculated by using: (a) the method of characteristics [19], (b) the developed code

Figure 3 illustrates the variations in course of the pressure in the exhaust system of a 4-cylinder turbocharged Diesel engine. The results have been obtained using the system data reported in the Ref. [19], sec. 19.4.2, p. 1092.

Apart from the comparisons with other methods, the developed code has been validated with the use of experimental data from the engine test bed measurements. The comparison of computation results with the measurements reported in Ref. [20] is illustrated in Fig. 4 and 5. The courses of temporal variations of pressures and temperatures are related to average values of the respective test bed results. The data shown in Fig. 4 refer to the turbine entry in a 4-cylinder, turbocharged Diesel, STAR T370 engine. The basic engine data

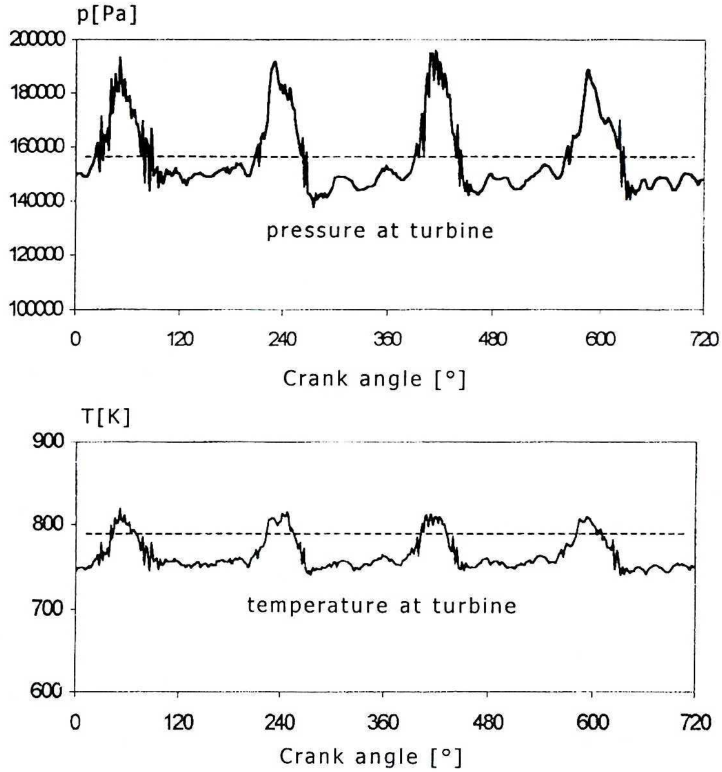

Fig. 4. Pressure and temperature at turbine entry in T370 engine:
-------- test-bed measurement, ——— calculations

are specified in Table 1. The continuous lines represent the computation results while the measurement averages are drawn by using the straight dash lines in all the figures. The results of numerical experiment are presented in the same way in Fig. 5, for the intercooler outlet boundary.

Table 1.

T370 engine basic data

Engine type	STAR T370, turbocharged, Diesel, intercooled, air to water[1)]
Bore/ stroke diameters	110mm/120mm
No of cylinders/ swept volume	4/ 4.561 dm^3
Compression ratio	17
Nominal power/ speed	78 KW at 2600 rev/min
Max. torque/ speed	378 Nm at 1300-1400 rev/min
Inlet/ exhaust valve diameters	48 mm/44 mm
Inlet valve open/closed	23° before TDC[2)]/ 50° after BDC[3)]
Exhaust valve open/ closed	50° before BDC[3)]/ 23° after TDC[2)]
Turbocharger	B-65 WSK PZL Rzeszów

Geometry of the system			
System element	Diameter [mm]	Length [mm]	Notes
Filter inlet	100	900	
Compressor inlet	100 – 55	600	
Compressor outlet	35 – 80	1600	35/300 [mm]
Manifold inlet	70	900	
Air manifold	70	650	
Exh. manifold sec. I	25-45	320	
Exh. manifold sec. II	25-45	580	

[1)] in experimental arrangement; [2)] top-dead-centre; [3)] bottom-dead-centre

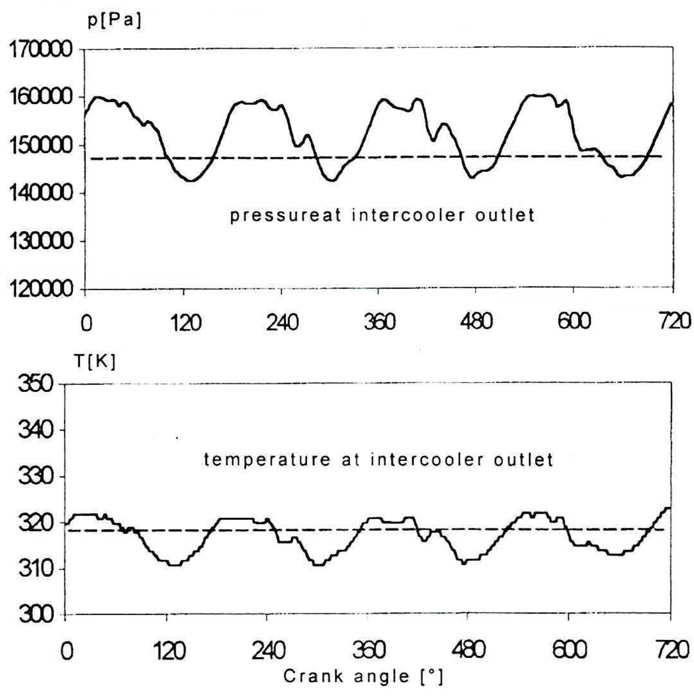

Fig. 5. Pressure and temperature at intercooler outlet in T370 engine:
------- test-bed measurement, ——— calculations

5. Conclusions

The reported work has focused on defining the basic formulae of the optimisation procedure for the turbocharging system of a Diesel engine. Following the analysis of exergy losses, reversible and irreversible, a criterion-goal function has been defined together with other quantities necessary for performing the optimisation.

The flow model of the charge exchange system has been verified in computer simulations performed with the use of the program developed by the Author. The code rendered information about the distribution of gas flow-field variables in the entire system. The analysis of results presented in section 3 induces a conclusion that the adopted method and computer program both make it possible to model the actual turbocharging system behaviour in an acceptable way. This relates to predicting the courses of temperature and pressure variations in time and their average values. The control test, performed against the results of experimental measurements of the STAR T370 engine, preceded applying of the entropy generation minimisation method to the charge exchange system studied, which is described later in part II.

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Termodynamiczna optymalizacja układu turbodoładowania silnika wysokoprężnego

Część I

Streszczenie

W pracy przedstawiono metodykę termodynamicznej optymalizacji układu dolotowo-wylotowego wysokoprężnego silnika z turbodoładowaniem. Funkcję celu zdefiniowano w postaci sumy strat nieodwracalnych egzergii jakie występują w rozpatrywanym układzie. Określono zmienne decyzyjne i parametry oraz warunki ograniczające procedury optymalizacyjnej. Ponadto, zaprezentowano krótką charakterystykę oraz wyniki weryfikacji modelu obliczeniowego przepływu.