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RESEARCH ARTICLE

The Aero-Thermodynamic Possibilities of Increasing Turbocharging Efficiency

Endre Pásztor

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Abstract

Turbocharging is a basic possibility to increase the utilization of reciprocating internal combustion engines. It can be stated that reciprocating engines of more than 100kW nominal power are oftenequipped with turbochargers and in most of the cases they also feature an intercooler. In order to be able to investigate the possibility of turbocharger development, one has to thoroughly discover the methods for improvement.

In the present article an investigation is carried out to obtain the dependence of efficiency of turbocharging on the aero-thermodynamic properties of the turbocharger. Present paper covers partially – as an overview – the previous results of turbocharger development while its aim is to provide guidelines for the possible upper limits of turbocharging in the automotive industry. The research does not include special fields of application (e.g. racing engines with extremely high acceleration) with especially highly supercharged reciprocating engines.

In order to reduce the complexity of the relationships under investigation only those parameters are considered as variables that have a significant effect on the efficiency of turbocharging.

Keywords

increase of indicated mean pressure · pressure ratio of turbocharger compressor · losses of compressor and turbine

Endre Pásztor
Department of Aeronautics, Naval Architecture and Railway Vehicles, Budapest University of Technology and Economics
e-mail: pasztor@vrht.bme.hu

List of symbols (in the order of first appearance)

Symbol	Unit	Description
π_c	–	Compressor pressure ratio (p_2 / p_1)
p_{i0}	N/m^2	Indicated mean pressure of the basic engine without turbocharging
η_{mm}	–	Mechanical efficiency of turbocharged engine
p_{icht}	N/m^2	Indicated mean pressure of turbocharged engine without intercooler
p_i	N/m^2	General indicated mean pressure
V	m^3	piston displacement
n	1/s	rotation speed of engine
W_{eff}	J	effective work
η_{eff}	–	effective efficiency
Q_{in}	J	heat added during the process
B_{in}	kg	fuel added to the process
C_f	J/kg	calorific value of the fuel
m_a	kg	mass of air charge
α	–	air-fuel ratio
L_a	m_a/B_{in}	stoichiometric ratio
ρ	kg/m^3	density
η_t	–	thermodynamic efficiency
η_i	–	indicated efficiency
p_{ich}	N/m^2	indicated mean pressure of basic work cycle
n_c	–	polytropic exponent of compression
η_{polc}	–	polytropic efficiency of compression
σ_{01}	–	pressure loss factor of intake duct upstream of turbocharger compressor ($\sigma_{01} = p_1 / p_0 = 0.98$)
p_0	N/m^2	ambient pressure ($p_0 = 10^5$N/m^2 = 1bar)
T_0	K	ambient temperature (T_0 = 298K)
T	K	general temperature
κ_a	–	isentropic exponent of air
$\eta_{is\,C}$	–	isentropic efficiency of compressor
ρ_2*	kg/m^3	density of engine intake air using intercooler
T_2*	K	engine intake air temperature using intercooler
ϕ	–	effectiveness of intercooler (efficiency)

R	J/kgK	specific gas constant of working medium
p_{iex}	N/m^2	indicated mean pressure of scavenging and back pressure of engine without intercooler
p_{iex}*	N/m^2	same as p_{iex}, but with using intercooler
σ_{22*}	–	pressure loss factor of intercooler ($\sigma_{22*} = p_2* / p_2 = 0.99$)
p_{icht}*	N/m^2	final indicated mean pressure of turbocharged engine with intercooler
W_c	J	work requirement of compressor
W_t	J	delivered work of turbine
c_{pg}	J/kgK	isobaric specific heat of gas entering the turbine (c_{pg} = 1200 J/kgK)
c_{pa}	J/kgK	isobaric specific heat of air entering the compressor (c_{pa} = 1005 J/kgK)
m_g	kg	mass of exhaust gas entering the turbine
η_m	–	mechanical efficiency of turbocharger
π_T	–	turbine pressure ratio ($\pi_T = p_3 / p_4$)
$\eta_{is\ T}$	–	isentropic efficiency of turbine
κ_g	–	isentropic exponent of combustion gases ($\kappa_g = 1.3$)
η_{istot}	–	total efficiency of turbocharger ($\eta_{is\ tot} = \eta_{is\ C} \cdot \eta_{is\ T} \cdot \eta_m$)
σ_{45}	–	pressure loss factor of exhaust system ($\sigma_{45} = p_5 / p_4 = 0.98$)
P	–	p^*_{icht} / p_{icht}, notation on Fig. 7

1 Introduction, posing the problem basic considerations

In order to be able to determine the possibilities for the efficiency increase of the turbocharged internal combustion reciprocating engines (henceforth turbocharged engines), it is important to investigate what is the extent to which present parameters of turbocharging allow further development.

The efficiency of turbocharging depends appreciably on the various losses arising in the compressor and exhaust turbine (generally, in the supercharging system), on the $\pi_c = p_2 / p_1$ pressure ratio of the compressor, on the t_3 exhaust gas temperature, and on the parameters of the increasingly applied intercooler. The analysis covers the investigation of the possibility of cross scavenging, where the maximum attainable efficiency is approached with cutting edge technology including rotating heat exchangers.To increase the understandability of the study the reciprocating engine is investigated with simplified conditions in some extent. According to this, the development of oscillatory phenomena in the supercharged engine is not taken into consideration. This approximation does not influence the ascertainments; however, it results in a clearer article.

The parameters, which affect the turbocharging in a large extent, are varied within the present realistic ranges; nevertheless, it is also shown how far the efficiency of turbocharging may be increased with the anticipatory future peak values (Shiwale et. al, 2013). The factors, whose effect is less or negligible, are taken into account as their realistic, constant value. These considerations are clearly represented in the calculations as well as in the list of symbols.

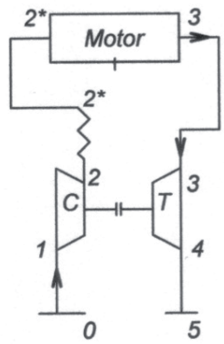

Fig. 1. Operational schematic of turbocharged engine
Thermodynamic stations: 0 – upstream of intake duct; 1 – upstream of compressor; 2 – compressor discharge; 2* – intercooler exit; 3 – downstream of engine (turbine inlet); 4 – turbine discharge; 5 – downstream of exhaust duct

The schematic of the turbocharged engine and the substantial thermodynamic stations are shown on Fig. 1. The investigations are henceforth carried out according to the notations of Fig. 1. In this figure the outline for intercooler is also represented. Without intercooler the stations 2 and 2* are implicitly the same.

In the present paper we do not make a distinction between gasoline and diesel engines, as there is no difference in thermodynamic aspects between the two types of engines. Our investigation is carried out supposing a real, maximal thermally loaded condition.

Our analysis is started with a basic engine without supercharging having an indicated mean pressure of p_{i0} and the goal is to establish the correlation between increasing indicated mean pressure of the turbocharged engine and the various conditions of turbocharging. We do not include the survey for the change of effective mean pressure as the η_{mm} mechanical efficiency can be upgraded by 2 or 3 percent under optimal conditions due to supercharging. The conformation of η_{mm} is particularly important at part power settings (Pásztor, 1970) but it is not taken into account.

In the figure representing the results of the equation system the ratio of p_{icht} / p_{i0} is indicated.

In our study first the theoretical background of turbocharging is discussed, and the equations for computation are determined, respectively; finally the results are demonstrated. In the present paper investigations about instable operational modes are not covered, as it is included in various domestic and foreign literatures (Beneda, 2013; Boyce, 1993; Kalabic et. al., 2011).

2 Equation system

2.1 Increase of indicated mean pressure of the main cycle

Our considerations are started with an equation that is applicable to engines regardless of the presence or absence of supercharging (Pásztor and Szoboszlay, 1967). Effective work of the engine can be calculated as:

$$W_{eff} = p_i \cdot V \cdot \eta_{mm} \tag{1}$$

The effective work W_{eff} is also expressed as the product of effective efficiency and introduced heat:

$$W_{eff} = \eta_{eff} \cdot Q_{in} = \eta_{eff} \cdot B_{in} \cdot C_f \tag{2}$$

The amount of supplied fuel can be obtained as:

$$B_{in} = \frac{m_a}{\alpha \cdot L_a} \tag{3}$$

The indicated mean pressure of the main thermodynamic cycle can be represented from Equations (1), (2) and (3), including the definition of density (ρ):

$$p_i = \rho \cdot \frac{\eta_{eff} \cdot C_f}{\alpha \cdot \eta_{mm} \cdot L_a} \tag{4}$$

Examining Equation (4) it is evident that the indicated mean pressure of the main cycle, or its growth respectively, is only depending on the density of the intake air. From the other factors, η_{eff} has reached its practical limits already; the α fuel-air ratio cannot be decreased under the unity neither in the aspects of combustion engineering or without the risk of sudden damage to the engine; L_a and C_f is adjustable only in minimal extent supposing liquid or gaseous fuels.

The p_{ich} indicated mean pressure of the supercharged cycle can be expressed in accordance with the notations of Figure 1:

$$p_i = \rho \cdot \frac{\eta_{eff} \cdot C_f}{\alpha \cdot \eta_{mm} \cdot L_a} \tag{4/a}$$

2.2 Determination of ρ_2 intake air density

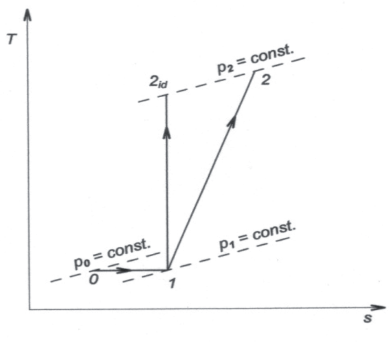

Fig. 2. T-s diagram of compression process

In view of compressor inlet parameters (p_0; T_0; ρ_0) the ρ_2 real density (Figure 2) of the air supplied to the supercharged engine without intercooler can be obtained taking n_c and η_{polc} factors into account as follows:

$$\rho_2 = \rho_1 \cdot (\pi_c)^{\frac{1}{n_c}} = \frac{\sigma_{01} \cdot p_0}{R \cdot T_0} \cdot (\pi_c)^{1-\left(\frac{\kappa_a - 1}{\kappa_a}\right)\cdot\frac{1}{\eta_{pol\,c}}} \tag{5}$$

Using the isentropic efficiency of the compression process one gets:

$$\rho_2 = \rho_1 \cdot \pi_c \cdot \frac{T_1}{T_2} = \rho_1 \cdot \pi_c \cdot \frac{1}{1 + \left[(\pi_c)^{\frac{\kappa_a - 1}{\kappa_a}} - 1\right] \cdot \frac{1}{\eta_{is\,c}}} \tag{6}$$

The relationship between n_c; $\eta_{pol\,c}$ and $\eta_{is\,c}$ can be expressed as:

$$\eta_{is\,c} = \frac{(\pi_c)^{\frac{\kappa_a - 1}{\kappa_a}} - 1}{(\pi_c)^{\frac{\kappa_a - 1}{\kappa_a}\cdot\frac{1}{\eta_{pol\,c}}} - 1} \tag{7}$$

It depends on the task to be resolved, which of the three loss factors should be utilized. With large ($\pi_c > 5$) pressure ratios the usage of n_c and $\eta_{pol\,c}$ is practical as they are independent of the π_c pressure ratio (Bosnjakovic, 1972).

Taking into account that the realized pressure ratio in turbocharger compressors is moderate ($\pi_c \approx$ 2–4) where $\eta_{is\,c} \approx \eta_{pol\,c}$, and considering that compressor maps are often showing values of $\eta_{is\,c}$ (Mayer, 1996a), henceforth the isentropic efficiency is used.

2.3 Determination of the density of the intercooled compressed air

With the cooling of the compressed air the indicated mean pressure can be significantly augmented through the increase in density. In turbocharged engines equipped with intercooler the ρ_2* density of intake air is the following according to Figure 1:

$$\rho_2 = \frac{\sigma_{01} \cdot p_0}{R \cdot T_2^*} \cdot \pi_c \tag{8}$$

where $T_2^* < T_2$ depending on the φ efficacy of the intercooler.

Using the definition of intercooler efficiency we can obtain the value for T_2*, supposing ambient air as cooling medium.

$$\phi = \frac{T_2 - T_2^*}{T_2 - T_0}; T_2^* = T_2 \cdot (1 - \phi) + \phi \cdot T_0 \tag{9}$$

The ρ_2*density downstream of the intercooler can be expressed using Equations (6); (8) and (9):

$$\rho_2^* = \frac{p_2^*}{R \cdot T_2^*} = \frac{\sigma_{01} \cdot p_0 \cdot \pi_c \cdot \sigma_{22^*}}{R \cdot \left[T_2 \cdot (1-\phi) + \phi \cdot T_0\right]}$$

$$= \frac{\sigma_{01} \cdot p_0 \cdot \pi_c}{R} \cdot \frac{\sigma_{22^*}}{\left\{T_0 \cdot \left[\frac{\pi_c^{\frac{\kappa_a - 1}{\kappa_a}} - 1}{\eta_{is\,c}} + 1\right]\right\} \cdot (1-\phi) + \phi \cdot T_0} \tag{10}$$

2.4 The change of indicated mean pressure of the supercharged engine due to the positive area of scavenging in p-V diagramand back pressur of exhaust gases

The increase of indicated mean pressure of the supercharged engine is definitively depending on the ρ_2 density. The value of p_{ich} is influenced in a significantly less extent by the $(p_2 - p_3)$ pressure differential assuming $p_2 > p_3$.

Without using intercooler the p_{iex} indicated mean pressure arising from back pressure $(p_2 - p_3)$ and scavenging:

$$p_{iex} = (p_2 - p_3) - (p_3 - p_0) \tag{11}$$

Using an intercooler the above equation takes the form of:

$$p_{iex}* = (\sigma_{22*} \cdot p_2 - p_3) - (p_3 - p_0) \tag{12}$$

The p_{iex}* indicated mean pressure is slightly reduced (0.4-0.6%) due to the pressure loss of the intercooler.

As a consequence of the back pressure of the exhaust turbine the indicated mean pressure of the turbocharged engine is decreased with the value of $(p_3 - p_0)$, however, the power requirement of the compressor should not be subtracted as it should be done with the mechanically driven superchargers. The pressure difference $(p_3 - p_0)$ reduces the indicated mean pressure by 4-5%; nevertheless, this decrease is significantly less compared to the power requirement of the compressor. This is one of the advantages of turbocharging.

The total cycle indicated mean pressure can be calculated as:

$$p_{icht} = p_{ich} + p_{iex}; \qquad p_{icht}^* = p_{ich} + p_{iex}^* \tag{13}$$

Besides the increment of indicated mean pressure by some percent the effect of more perfect scavenging is of a higher importance due to valve overlap. It has a substantial relevancy on the reducing thermal loading of the engine and improving its service life (Fülöp, 1990).

The detailed description of the above mentioned effects would reach beyond the limits of the present work; however, the determination of the p_3 pressure is included.

Under steady state conditions the turbocharger operates in a thermodynamic balance condition, i.e. $W_C = W_T$ that can be written as follows, regarding the notation of Figures 2 and 3:

$$c_{pg} \cdot m_g \cdot (T_3 - T_4) = c_{pa} \cdot m_a \cdot (T_2 - T_1) \tag{14}$$

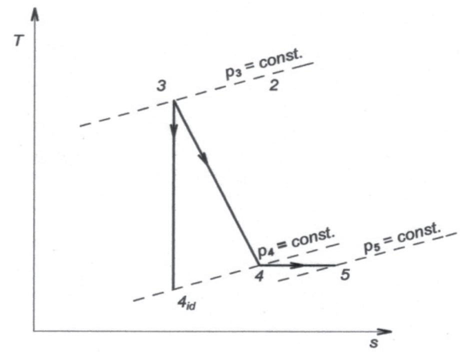

Fig. 3. T-s diagram of expansion process

Taking into account the isentropic efficiencies for the compressor and turbine and the η_m mechanical efficiency of the turbocharger, including the compressor pressure ratio $\pi_C = p_2 / p_1$ and turbine pressure ratio $\pi_T = p_3 / p_4$, and raising the known temperatures of T_1 and T_3, one can obtain:

$$c_{pg} \cdot 1{,}02 \cdot m_a \cdot T_3 \cdot \eta_{is\,t} \cdot \eta_m \cdot \left(1 - \frac{1}{\pi_T^{\frac{\kappa_g - 1}{\kappa_g}}}\right) = \frac{c_{pa} \cdot m_a \cdot T_1}{\eta_{is\,C}} \cdot \left[(\pi_C)^{\frac{\kappa_a - 1}{\kappa_a}} - 1\right] \tag{15}$$

The amount of air, what has been delivered by the compressor, is reducing by some 1% due to sealing losses between the piston and its sleeve. On the contrary, the gas mass through the exhaust turbine is increasing the previous value by 2% because of the fuel supply; the consequence is the approximation of $m_g \approx 1.02 \cdot m_a$ relationship which strongly depends on the engine health.

Introducing the simplified annotation of $\eta_{is\ C} \cdot \eta_{is\ T} \cdot \eta_m = \eta_{is\ tot}$ and taking into account that $p_4 = p_5 / \sigma_{45}$; $p_1 = p_0 \cdot \sigma_{01}$, the value for p_3 can be expressed.

When investigating the Equation (15) one can see the advantage of the introduction of $\eta_{is\ tot}$, which is combining the three individual factors to a single "total" efficiency, while using polytropic exponents or efficiency similar possibility is not given. The p_3 pressure is influenced by the product of the three factors, independently of their components.

3 Results of the investigation and their evaluation

3.1 Initial data of the investigation and their correlations

The increase of p_{icht} total indicated mean pressure of the engine is determined by Equations (4/a); (11) and (13). In the investigation the following parameters have been taken into consideration.

a) Isentropic efficiency of the compressor ($\eta_{is\ C}$). Its value has been changed in the range of 0.67-0.78. The $\eta_{isC}< 0.65$ results in a rather poor effect of supercharging; values over 0.78 can be achieved only in turbochargers of relatively large engines (e.g. maritime engines). The utilization of axial compressors is practically impossible.

b) Isentropic efficiency of the turbine ($\eta_{is\ T}$), which has some 2% larger value in contrast to the compressor. The aerodynamic details of this effect cannot be discussed here.

c) The mechanical efficiency of the turbocharger η_m has been used as a constant parameter with a value of $\eta_m = 0.98$. According to the previously mentioned thoughts, the $\eta_{is\,tot}$ total efficiency is containing the following products. The value $\eta_{istot} = 0.6$ can be regarded as the maximum achievable limit, which could not be exceeded in the future due to the inevitable friction and incidence losses.

$\eta_{is\ C}$	0.671	0.707	0.742	0.775
$\eta_{is\ T}$	0.685	0.721	0.757	0.791
η_m	0.98	0.98	0.98	0.98
η_{istot}	0.45	0.5	0.55	0.6

d) The compressor pressure ratio η_c has been varied in the range of 1.5 and 3.5. Pressure ratios $\eta_c< 1.5$ can be supposed as resulting unsatisfactory supercharging conditions while $\eta_c> 3.5$ is rather high and is not widely utilized in commercial engines produced in large numbers.

e) The turbine inlet temperature t_3 (see Figures 1 and 3) is an essential factor of turbocharging efficiency. The t_3 temperature cannot be taken arbitrarily; it depends primarily on the turbocharged engine thermal loads and its construction (Mayer, 1996a). Based on other investigations, the extremely low speed maritime engines including water cooled exhaust duct can reach a minimum of $t_3 \approx 450°C$, while high speed engines of road transport vehicles can operate with $t_3 \approx 800\text{-}850°C$. Excluding the two extreme cases we have taken into account t_3 values of 500, 600 and 700°C in our calculations.

f) Those pressure loss factors, which are influencing the efficiency in a moderate extent, have been assumed having a constant, realistic value. According to Figure 1 (see definitions in the list of symbols):

$$\sigma_{01} = \frac{p_1}{p_0} = 0,98;$$

$$\sigma_{22^*} = \frac{p_2^*}{p_2} = 0,99;$$

$$\sigma_{45} = \frac{p_5}{p_4} = 0,98;$$

g) During the evaluation of our results we have used the p_{icht} / p_{i0} ratio instead of the total indicated mean pressure p_{icht} of the supercharged engine. As of our considerations, this ratio presents the tendency of changes better than the p_{icht} pressure itself. The base engine of our investigation has a p_{i0} indicated mean pressure of $1.1{\cdot}10^6 N/m^2 = 11 bar$. This corresponds to the category of an engine of common automobiles.

3.2 The increase of total indicated mean pressure of the supercharged engine without intercooler

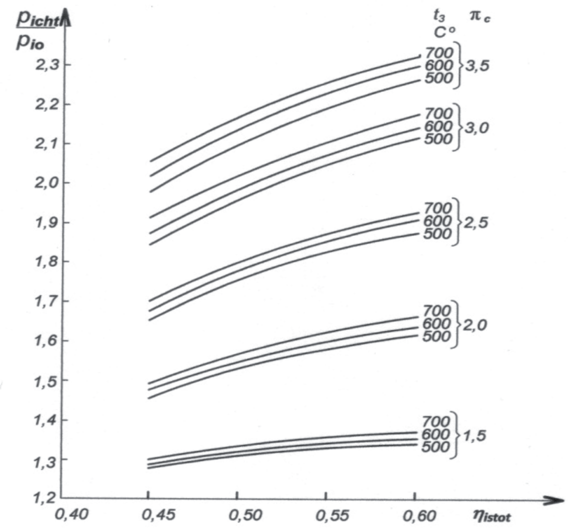

Fig. 4. Effectiveness of supercharging (ratio of indicated mean pressures of supercharged and basic engines) as a function of total efficiency $\eta_{is\,tot}$ at different π_c compressor pressure ratios and t_3 turbine inlet temperatures without intercooling

On Figure 4 one can see the p_{icht} / p_{i0} ratio for a supercharged engine without intercooler as the function of η_{istot} with various π_c pressure ratios and t_3 turbine inlet temperatures. The rise of each of the three parameters cause a significant increase in the degree of supercharging, which shows an intensive growth when η_{istot} is improved at high pressure ratio, but its tendency is diminishing, consequently the favorablevalue of η_{istot} is extremely important under such circumstances. The increase of t_3 turbine inlet temperature clearly raises the degree of supercharging, but the ratios are not modified substantially.

The approximate change of

$$\Delta\left(\frac{p_{icht}}{p_{i0}}\right)\Big/\Delta\eta_{is\,tot}$$

gradient is shown as a function of π_c on Figure 5. It is only an approximation due to the omitted minimal effect of change in t_3.

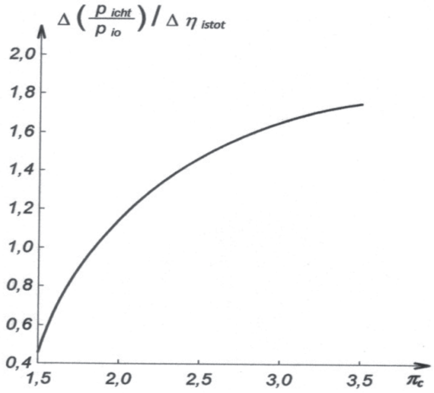

Fig. 5. Approximate change of gradient $\Delta\left(\frac{p_{icht}}{p_{i0}}\right)\Big/\Delta\eta_{istot}$ as a function of pressure ratio π_c without intercooling

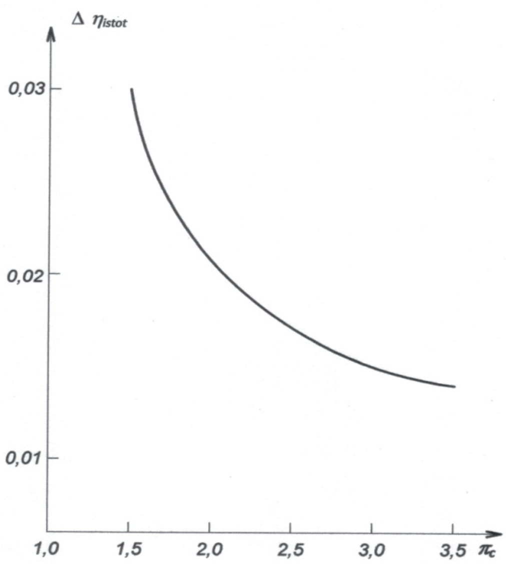

Fig. 6. Possible reduction of total efficiency $\Delta\eta_{istot}$ due to turbine inlet temperature increase Δt_3 = 100°C as a function of pressure ratio π_c

According to the consequences drawn evaluating Figure 4, taking into account a constant $\Delta\eta_{istot}$ change in the efficiency, the effectiveness of the supercharging rises monotonously as the function of π_c, but the tendency of growth is declining. As it can be stated by evaluating Figures 4 and 5, it is not economic to increase π_c above 3.5-4 without intercooler, due to the relative drop in the efficacy. This negative effect is caused by the strong increase in T_2 compressor discharge temperature with the rising π_c, if one omits the intercooler, resulting in the relative decrease in the density of the compressed air.

According to Figure 4, the reduction in η_{istot} can be compensated by the rise of turbine inlet temperature t_3. The same conclusion can be investigated on Fig. 6 with the help of other parameters, where the drop of $\Delta\eta_{istot}$ can be seen as the function of π_c caused by a temperature increase of Δt_3 = 100°C. As stated by Fig. 6, in the intermediate range of π_c = 2–2.5 the reducing efficiency of $\Delta\eta_{istot} \approx$ 0.02–0.017 can be compensated with the above mentioned increase in turbine inlet temperature t_3. The diagram indicates average changes, although the conditions are hardly depending on the selection of the value of $\Delta\eta_{istot}$ from the available η_{istot} ranges, according to our investigation. In another viewpoint, the $\Delta\eta_{istot}$ shows a decreasing tendency with increasing π_c to offset the effect of turbine inlet temperature increase of Δt_3 = 100°C. This is a result of the rise of $p_{icht}\big/p_{i0} = f\left(\eta_{is\ tot}\right)$ gradient, as shown on Fig. 4.

The largest problem is that the turbine inlet temperature t_3 cannot be modified arbitrarily; it is a function of many parameters of the turbocharger and the supercharged engine, which are frequently inconsistent with each other, but the t_3 temperature depends on the thermal load of the supercharged engine. This problem is so complicated that its detailed discussion cannot be covered herein. We return to this subject in terms of scavenging (Dezsenyi et. al., 1990).

3.3 Using an intercooler downstream of the compressor

The positive effect of cooling the airflow, which is supplied by the turbocharger compressor at a temperature of t_2, is clear. The basic diagram of the intercooled instance is shown on Fig. 7, which is strongly similar in comparison with Fig. 4; the only important difference is that the process takes place at an elevated level. In order to accentuate the positive changes, Fig. 7 does not show the absolute value of p^*_{icht} for the intercooled turbocharged engine having an intercooler efficiency factor of φ, but the ratio of change in contrast to the case without intercooler (φ = 0). According to this, on the ordinate axis of Fig. 7 the measure is $\left(p^*_{icht}\big/p_{icht}\right) = P$. On Fig. 7, this is denoted as P. In order to increase perspicuity of Fig. 7 it contains only the case for t_3 = 600°C, as P is hardly influenced by t_3.

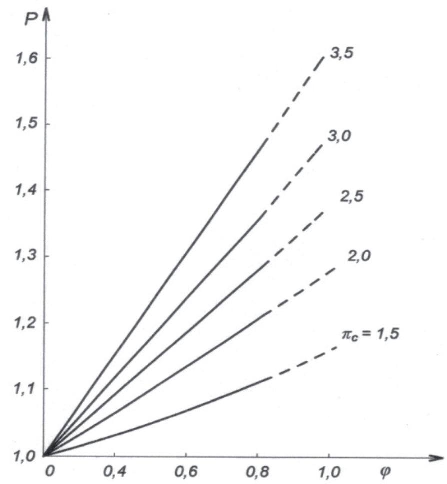

Fig. 7. Change of ratio $P = \left(p^*_{icht}\big/p_{icht}\right)$ as a function of φ heat exchange factor at t_3 = 600°C and various π_c compressor pressure ratios

The intercooler heat exchange factor φ has a maximal value of approximately $\varphi \approx 0.8$. Values above this level cannot be realized, even with rotating (regenerative) heat exchangers (Fülöp, 1975), except that the installation of heat exchangers in automobiles is always a problem. On Fig. 7, one can clearly see the extremely beneficial effect of intercooling and the fact, that intercooling is worthy only with relatively large values of π_c. Assuming an average heat exchange factor ($\varphi \approx 0.5$–0.6) and a compressor pressure ratio of $\pi_c = 3.5$, the achievable increase in engine power is approximately 26–28%.

The heat exchange factor $\varphi > 0.8$ might be realized with one type of air cycle machines (Pásztor, 1977; Pásztor, 1981). Although these devices are very useful, they are rather heavy and their dimensions are also excessive. As the effective efficiency of the supercharged engine is rather diminishing, than increasing using these units, their spread is quite limited. The deterioration of effective efficiency results from the power requirement of the negative cooling cycle.

The favorable effect of intercooling manifests itself in the increasing efficacy of the scavenging (internal cooling of the engine) besides the rise of effective power. This effect cannot be easily represented numerically, but this leads to a significant growth in the service life of the supercharged engine, especially when large level of supercharging is used. There is a less important impact of intercooling, namely the decrease in the temperature of the incoming air leads to a proportionally diminishing power requirement for the compression (in the engine cycle) while the indicated mean pressure increases by some percent. The ideal process is depicted by Fig. 8. It is independent of the sign of the area of scavenging in the p-V diagram, but its effect is damped by the warming of the air inside the engine.

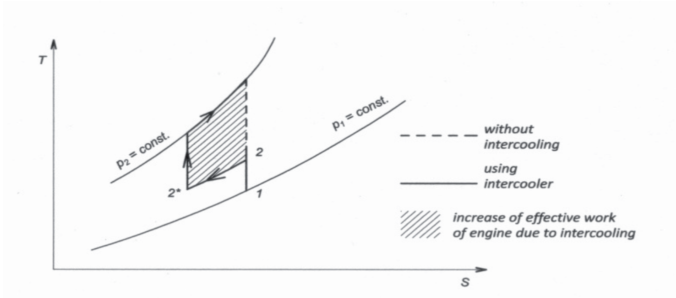

Fig. 8. Effect of intercooling on the compression process of the turbocharged engine (dashed – without intercooling; solid – using intercooler; hatched area – increase of effective work of engine due to intercooling)

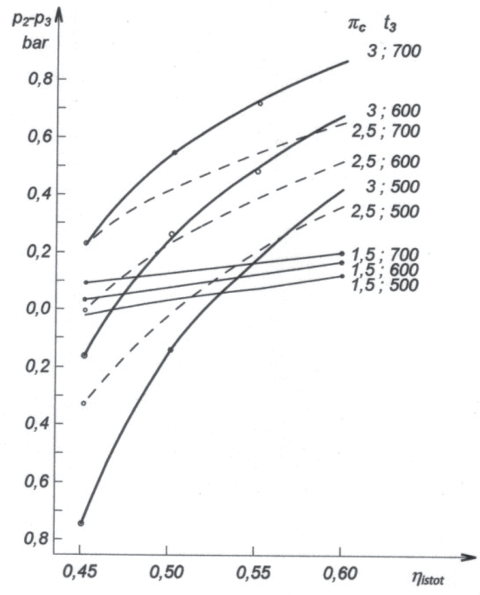

Fig. 9. Change of ($p_2 - p_3$) scavenging pressure difference as a function of η_{istot} at different π_c pressure ratios and t_3 turbine inlet temperatures

3.4 Area of scavenging in p-V diagram as a function of turbocharging parameters

The area of scavenging in the p-V diagram strongly influences the service life of the engine due to internal cooling as well as the removal of residual gases from the previous combustion process. According to these ascertainments, a slightly more detailed investigation follows.

The area of scavenging has a positive sign if $p_2 > p_3$. The value of pc has a clear impact on the magnitude of p_2, while p_3 comes from the thermodynamic equilibrium of the turbocharger rotor ($W_c = W_t$) as described by Eq. (15). The starting values are included in the list of symbols. The positive area of scavenging decreases if $\eta_{istot,}$, t_3 and π_c are reducing; and, as it is shown on Fig. 9, it can turn into a negative area. As of our investigation, assuming an average $t_3 = 600$°C and $\pi_c = 2.5$, a minimum of $\eta_{istot,} = 0.5$ is required to achieve a significantly positive area for scavenging. Taking into account a small turbocharger with relatively small air mass flow ($\eta_{is\ C} \approx 0.7$; $\eta_{is\ T} \approx 0.73$; $\eta_m \approx 0.98$) the above mentioned total isentropic efficiency is difficult to reach, here the utilization of so called "impulse" supercharging (Mayer, 1996b; Dezsényi et. al., 1990) is getting more significant, which helps to achieve slightly more favorable conditions. The detailed discussion of this method is not possible to fit within the limits of the present work. Returning to Fig. 9, one can see that in order to provide easy evaluation only $\pi_c = 1.5$; 2.5; and 3.5 are indicated, due to the intersection between the curves.

On Fig. 10 those π_c values have been collected, which result in a $(p_2 - p_3) = 0$ condition as the function of η_{istot}. The relatively large π_c requires very high t_3 turbine inlet temperature and favorable η_{istot} efficiency in order to maintain the difference $(p_2 - p_3)$ over zero. From the two latter figures one can draw the consequence that in the higher range of π_c (π_c>2.5 – 3), only the high η_{istot} efficiency allows favorable properties of turbocharging. In order to improve this process the increase in t3 would be necessary, but, as it has been stated before, it is a dependent parameter of the complete supercharging cycle and cannot be varied alone arbitrarily.

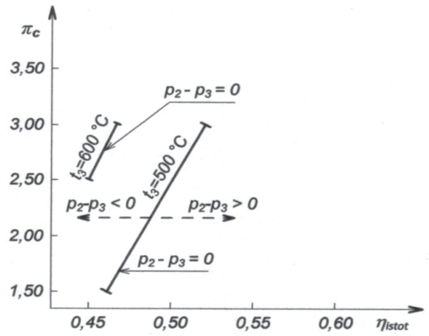

Fig. 10. Figure 10 Zero value ($p_2 - p_3$) scavenging pressure difference as a function of η_{istot} at t_3 = 500°C and 600°C. At t_3 = 700°C, there is no zero point

4 Conclusions

- Assuming medium η_{istot} and π_c the achievable increment in the power of turbocharged engine without intercooling falls in the range of 70-75% in contrast to the basic engine.
- Taking into consideration maximal values of η_{istot} and high π_c, the aforementioned increase can reach 100-120%. This is the ultimate limit without intercooler.
- Without intercooler the degree of supercharging increases significantly for a unit η_{istot} if the π_c pressure ratio is rising, but in tendency its measure is reducing.
- Assuming average π_c and efficiency increase $\Delta\eta_{istot}$ equal to unity, the degree of supercharging grows by 1.25 without intercooling.
- The increase in turbine inlet exhaust gas temperature t_3 can offset the effect of the reducing η_{istot}but the intensity of compensation is diminishing as π_c increases. Taking into account an average π_c, one can state that a reduction of $\Delta\eta_{istot}$= 0.2 can be compensated by the growth of t_3 of 100°C.
- The effectiveness of intercooling is strongly depending on π_c; a system with average π_c and φ the achievable additional degree of supercharging is approximately 20–25%. Assuming a very high π_c and favorable φ the additional degree of supercharging can reach a maximum of about 45%. The values over this limit can be realized only with additional cooling cycles.
- The ($p_2 - p_3$) pressure difference of scavenging depends extremely strongly on the three major parameters η_{istot}; π_c; t_3). Its important feature is that its sign changes rapidly to negative if π_c is high while only moderate t_3 is achievable.
- Supposing an average value for pc and a moderate t3, the minimum of η_{istot} = 0.5 is necessary to maintain the pressure difference ($p_2 - p_3$) in the positive range or at least at zero.

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