The concept of determining the control function of turbochargers in sequential turbo-charging

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Abstract. The sequential turbo-charging consists in using at least two turbochargers that work individually or in a parallel system depending on the changes in engine speed and load. With the right selection of the turbochargers for the engine, each turbocharger can work with greater efficiency in a much wider range of speed and load variations than in an engine using a single turbocharger. However, the benefits of the sequential turbo-charging requires solving the problem of the optimal selection of two different turbochargers and - additional - ensuring the proper conditions for the cooperation between both of them. This is particularly important during the switching over phase of the turbocharger, and thus also for control issues. The paper presents the concept of optimization of the function controlling the switching over of turbochargers in the whole field of engine operation. The operating ranges of the turbochargers were determined on the basis of the economic criterion of the engine operation. For this purpose, the load characteristics of the brake-specific fuel consumption obtained in the mode of operation with one and two simultaneously operating turbochargers were used. The points of intersection for these characteristics determined the work areas of each turbocharger depending on the engine speed and load. The control function determined in this way was used to develop the models and the algorithm of the numerical control system for turbochargers. The correctness of the system operation was verified during tests on the engine dynamometer.

1. Introduction
Currently, there is no doubt that the turbocharging is the most effective method of increasing the concentration of the power (from the unit of the displacement volume) for the compression ignition engines [[5]]. The significant improvement of the engine's characteristics is ensured by the sequential turbo-charging. The sequential turbo-charging is based on the use of at least two turbochargers, connected parallel to each other, which operate in different ranges of engine speed. The most frequently studied are systems with two turbochargers [[9]]. In the first range, one turbocharger operates at low engine speeds. In the second range, at high engine speeds, the second turbocharger is switched on [[6], [8]]. There are also some solutions known in which two turbochargers can operate in three ranges [[12]]. As a result, after the correct selection of the turbochargers, an increase in maximum power can be achieved as well as a clear improvement in the curve of the maximum torque at low and medium values of engine speed. The sequential turbo-charging has been known and used in the cars since the mid-1980s [Błąd! Nie można odnaleźć źródła odwołania.], but in recent years there has been a clear increase in interest in its further development possibilities. This is indicated by the researches and the latest designs of BMW, Opel, now also Volkswagen / Audi [Błąd! Nie można odnaleźć źródła odwołania., Błąd! Nie można odnaleźć źródła odwołania., Błąd! Nie można odnaleźć źródła odwołania.]. This article presents a two-phase sequential turbocharging system with two unequal-size
turbochargers working in parallel. For such a system, by means of the numerical simulations, a strategy for determining of the engine work ranges with one and two turbochargers is presented.

2. Description of the problem

The scheme of the tested supercharging system is shown in Fig. 1. In the first range, only one large turbocharger TC works at the low engine speeds. Using only one of the turbochargers to boost the engine allows to increase significantly the supercharging pressure and engine torque at low engine-speed conditions, where the exhaust energy is lower.

Thanks to the full usage of the engine’s exhaust flow and lower inertia of such a turbocharger, an improvement in the transient response is obtained.

The second small turbocharger TC is used at higher engine-speed conditions when the exhaust energy is sufficiently large. Two pilot valves are used to activate the small TC: a compressor start valve (VSC) and a turbine switching valve (VST).

The inclusion of a small TC causes the separation of the exhaust gas flow from the engine between two turbochargers working in parallel, which is accompanied by a rapid reduction of the supercharging pressure. This is an unfavourable property of this system occurring during the change of the turbocharger operation mode. The effect of such switching of the second turbocharger is significant oscillation of the engine's torque.

![Figure 1. Schematic diagram of the sequential turbocharging system with two unequal-size turbochargers.](image)

Such unfavourable behaviour of the system during the change of the turbocharger's operating phase causes that the correct choice of the turbocharger size is the main problem. A study by Volvo [[1]] shows that the use of the identical turbochargers allows a significant increase in torque at low speed. However, the flow starts to become choked on the small turbocharger, and the activation of the second turbocharger is accompanied by very large oscillations of the supercharging pressure and torque. The brake-specific fuel consumption also increases. When different turbochargers are used, an increase in the torque has been achieved with more stable engine operation. The second smaller turbocharger can be switched on in the conditions of the higher engine speed, and the torque oscillations are much smaller.

The results of these tests show that the transition between the different phases of operation represents one of the most crucial challenges of sequential turbocharging. Therefore, it is necessary to choose the
right strategy for determining the operating ranges of the turbochargers depending on the engine's operating state. [Blad! Nie można odnaleźć źródła odwołania.]

Two strategy selection criteria can be adopted. The first criterion is important at high engine flows. Then the second turbocharger is turned on when the flow through one turbocharger begins to block, or when the supercharging pressure increases dangerously. This is to protect the motor and the turbocharger against excessive load. At low engine loads, this protection is not necessary, as there is no risk of exceeding the maximum flow and supercharging pressure. In this case, while determining the operating range of the turbochargers, it is advisable to adopt the criterion of the maximum efficiency of the engine [[4]].

3. The concept for optimizing of the turbochargers’ control

In order to select optimal operating ranges for the turbochargers, simulation tests were carried out using proprietary numerical program [[3]]. This program is based on an empirical engine model and a mathematical description of the turbochargers’ characteristics. It allows to simulate the engine cooperation with the turbochargers of various sizes working individually and in parallel. The calculations were made for the self-ignition engine type SW-680, with the engine displacement of 11.1 dm³, intended for the drive of large-capacity trucks. The mathematical models of WSK Rzeszów turbochargers were used in the calculations. A large B3C turbocharger with a turbine inlet cross-section \( A_{T(I)} \) of 16.8 cm² was used. In a small B65 turbocharger, the turbine section \( A_{T(I)} \) was 5.65 cm².

The quality assessment of the turbochargers’ cooperation with the engine according to the adopted criteria (Section 2) was made on the basis of the characteristics of the supercharging pressure (\( p_{ba} \)) and the brake-specific fuel consumptions (\( b \)) depending on the engine load (torque). The \( p_{ba} \) and \( b \) variations were tested for different engine speeds in the working phase with one and two turbochargers. Figure 2 presents the results of the calculation of the supercharging pressure (\( p_{ba} \)) obtained at different engine speeds with a large turbocharger TC. It was assumed that the pressure (\( p_{ba} \)) cannot exceed the maximum value of 0.18 MPa of the base engine with conventional charging. This limitation is illustrated by the horizontal line \( p_{ba}(\text{max}) \).

**Figure 2.** The load characteristics of the supercharging pressure (\( p_{ba} \)) of the engine SW 680 with a large turbocharger at different engine speeds with marked line of the maximum supercharging pressure \( p_{ba}(\text{max}) \).

On the basis of these characteristics, for each engine speed, it is possible to determine the torque values (\( T_{tq} \)) at the intersection point of the (\( p_{ba} \)) pressure curves with the line of maximum pressure \( p_{ba} \).
(max). The coordinates \((n, T_{\text{tq}})\) of these points were plotted in the area of engine work limited by the curve of maximum torque, as illustrated in Fig. 3.

**Figure 3.** The course of the optimal control characteristic defining the operating ranges of the turbochargers in the field of the SW 680 engine

The course of these points determines the limit of the working range of a large TC turbocharger due to the criterion of allowable engine load. Above this line, it is necessary to start a small turbocharger TC.

In order to optimize the operating range of the turbochargers according to the maximum load efficiency of the engine, the load characteristics of the brake-specific fuel consumptions \((b)\) and the supercharging pressure \((p_{\text{ba}})\) were compared during the work phase with one and two turbochargers, which were determined at different engine speeds. The results of these calculations are shown in Figure 4.
Figure 4. The load characteristics of the brake-specific fuel consumptions $b$ and of the supercharging pressure ($p_{ba}$) of the engine SW 680 with one large turbocharger (black line) and with two turbochargers (red line) at different engine speeds:

a) - $n = 1000$ r / min,
b) - $n = 1200$ r / min,
c) - $n = 1,400$ r / min,
d) - $n = 1,600$ r / min

On the basis of the obtained characteristics for each engine speed, the values of the torque ($T_{tq}$) at the intersection point of the curves of brake-specific fuel consumption ($b$) of the engine with one large and with two working turbochargers (point 3) can be determined. The ($T_{tq}$) values determined in this way are marked in the Figure 5. The coordinates ($n$, $T_{tq}$) of these points in the engine's field limited by the maximum torque curve determine the operating ranges of the turbochargers at changes of the engine speed and load. To keep the maximum engine efficiency, both turbochargers should be connected in the area below the designated line. Above this line, the working area of one large turbocharger is available.

Figure 5. The course of the optimal control characteristic defining the operating ranges of the turbochargers in the engine field of the SW 680 according to the criterion of maximum efficiency with marked changes in brake-specific fuel consumption

In Figure 5, for optimal control strategy at selected points of the engine's working field, differences in brake-specific fuel consumption ($b$) were indicated in comparison to the strategy shown in Figure 3. The comparison shows that the inclusion of a small turbocharger greatly improves engine performance.
in a large area of engine operation. A significant reduction of the brake-specific fuel consumption \( (b) \) can be achieved especially at low load and high engine speed. At 2000 rpm and a load of 348 Nm, the value of \( b \) is 5.1% lower than with one turbocharger working.

4. Design of the optimal control function

The determined limits for the optimal controlling strategy for the switching of the turbochargers were used to design the optimal control function. The intersection points (3) of the brake-specific fuel consumption curves \( (b) \) on the load characteristics (Figure 4) correspond to the specified supercharging pressure values. The changes in the supercharging pressure during changing of the operating ranges of the turbochargers can be used to determine the optimal control characteristics of the turbochargers. The function of the supercharging pressure \( (p_{ba}) \) is sought depending on the engine speed \( (n) \). It can be determined on the basis of the supercharging pressure in points 1, 2 on the load characteristics, which are summarized in Table 1.

Table 1. The summary of the supercharging pressure values determining the course of the optimal control function

| \( n \) [r/min] | 1-phase (large turbocharger) | 2-phase (two turbochargers) |
|----------------|-----------------------------|-----------------------------|
|                | \( T_{tq} \) [kN m] | \( p_{ba(1)} \) [MPa] | \( T_{tq} \) [kN m] | \( p_{ba(2)} \) [MPa] |
| 1000           | 0,455                      | 0,114                       | 0,455                      | 0,106                       |
| 1200           | 0,485                      | 0,124                       | 0,485                      | 0,111                       |
| 1400           | 0,560                      | 0,139                       | 0,560                      | 0,120                       |
| 1600           | 0,820                      | 0,180                       | 0,820                      | 0,144                       |

Figure 6. The course of control functions in the scope of operation of a large turbocharger (curve 1) and two turbochargers (curve 2)

In the case of control system designing, the mathematical model can be presented in the form of a polynomial approximating by the determined values of the supercharging pressure \( p_{ba} = f (n) \). The distribution of these values shown in Figure 6, which for the operating range of the large turbocharger were marked with circles, and for the operating range of two turbochargers – triangles, indicates the possibility of their approximation with the second-order polynomial. The normalization of the actual engine speed values \( (n) \) was carried out in such a way that the lower and upper values \( (n) \) were transformed to the standardized values \( \hat{x} \) from the interval \([-1, +1]\) according to the formula:

\[
\hat{x} = \frac{n - 1600}{600}
\]
The regression equations determined for the normalized engine speed values are:

- for the control function in the scope of operation of the large turbocharger (curve 1):
  \[ P_{ba(1)} = 0.1791 + 0.1369 \cdot x + 0.0731 \cdot x^2 \]  

- for the control function in the scope of operation of two turbochargers (curve 2)
  \[ P_{ba(2)} = 0.1435 + 0.0809 \cdot x + 0.0439 \cdot x^2 \]

5. Conclusion

The paper presents a two-phase sequential turbocharging system with two unequal-size turbochargers working in parallel. In such systems, one of the most important challenges is to optimize the transition between different phases of turbochargers. The test results, obtained in the steady state of the engine operation, clearly show that if the permissible engine loads are not exceeded, the optimal strategy for switching the turbochargers should be determined on the basis of the differences in brake-specific fuel consumption when the operating phase of the turbochargers changes. When the small turbocharger is switched on, the brake-specific fuel consumption \( b \) is considerably improved, especially at low load and high speed. The value of \( b \) decreases by 15.1 g/kWh at 2000 rpm and the load of 348 Nm and is by 5.1% lower than with one turbocharger working. The limits of the optimal turbocharger switching strategy have been used to create a controller that, depending on the operative conditions of the engine, changes the phase of the turbochargers to ensure maximum efficiency of the engine. The correct operation of the controller has been verified during the experimental tests of the engine.

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