AIR COOLING INFLUENCE ON THE PERFORMANCE AND CHARACTERISTICS OF TURBOCHARGED DIRECT INJECTION GASOLINE ENGINE

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Abstract: *The paper presents numerical analysis of air after turbocharger cooling process and its influences on the gasoline engine operating parameters. Analysis was performed with numerical 0D (zero-dimensional) simulation model by using two sets of simulations – for gasoline turbocharged engine without air cooler and for the same engine with air cooler included. Between engine rotational speeds of 1000 rpm and 5000 rpm, air cooling process increases analyzed engine developed power and engine torque up to 20.67%, increases cylinder maximum pressure up to 17.03%, increases engine volumetric efficiency up to 23.65% and reduces brake specific fuel consumption up to 1.32% when compared with the same engine without air cooler. At the highest engine rotational speeds (between 5000 rpm and 6000 rpm) it was observed that selected air cooler does not offer the required and expected performance, so air cooling process in this engine operating area can and should be improved.*

KEYWORDS: AIR COOLER, GASOLINE ENGINE, TURBOCHARGER, ENGINE PERFORMANCE

1. Introduction

Internal combustion spark ignition gasoline engines are commonly used in different spheres of human life, especially in automotive applications [1]. Compared to diesel engines, many engineers and experts see gasoline engines as an essential component for the future, especially because of the significantly lower emissions [2].

Various phenomena related to gasoline engines are currently analyzed and discussed. A gasoline engine exhaust gas analysis based on engine speed is investigated and explained in [3]. Spark timing influence on the performance of a gasoline engine analyzed authors in [4]. Importance, effects and influences of turbocharging process on the gasoline engine operating parameters and characteristics were described and investigated in [5] and [6] with all advantages and disadvantages of this process.

Alternative fuels for gasoline engines are investigated by many authors. Gasoline engine performance and emission characteristics while using solketal-gasoline fuel blends were investigated in [7]. Optimized ethanol-gasoline blends for turbocharged engines were analyzed in [8].

In this paper is presented influence of air cooling process after turbocharger on the main gasoline engine operating parameters. Analysis was performed by using numerical 0D (zero-dimensional) simulation model. A set of numerical model simulations was firstly used to obtain required operating parameters of turbocharged gasoline engine without air cooler. After first set of simulations, numerical model is upgraded with air cooler sub-model which is included in calculation of engine operating parameters. Second set of simulations of the same turbocharged gasoline engine was performed with included air cooler. Comparison of obtained results showed that air cooling process significantly improves engine operating parameters. At the highest engine rotational speed (6000 rpm) it was observed that selected air cooler does not offer the best performance, so air cooling process in this engine operating area can and should be improved.

2. 0D numerical model basic equations

In this analysis, used numerical model for simulations is 0D (zero-dimensional) model presented in [9]. Numerical model is basically developed for simulation of diesel engines. After a while, this model is upgraded on QD (quasi-dimensional) numerical model presented in [10], [11] and [12].

In order to provide a simulation of a gasoline engine with this 0D model, the model is modified in all important parts which present differences in operating characteristics between gasoline and diesel engines. Modified 0D model is tested on several gasoline engines which measurements were obtained from the manufacturers. For all analyzed gasoline engines and its main operating parameters were obtained deviations between measurements and 0D numerical model results in the range of ± 3%.

The essential 0D model equations are related to the temperature and pressure change in each engine control volume. Temperature change in each engine control volume is defined by an equation:

$$\left(\frac{\mathrm{d}T}{\mathrm{d}\varphi}\right)_\mathrm{j} = \frac{\frac{1}{m_\mathrm{j}}\left[\left(\frac{\mathrm{d}Q}{\mathrm{d}\varphi}\right)_\mathrm{j} - p_\mathrm{j}\left(\frac{\mathrm{d}V}{\mathrm{d}\varphi}\right)_\mathrm{j} - u_\mathrm{j}\left(\frac{\mathrm{d}m}{\mathrm{d}\varphi}\right)_\mathrm{j} - m_\mathrm{j}\left(\frac{\partial u}{\partial \lambda}\right)_\mathrm{j}\left(\frac{\mathrm{d}\lambda}{\mathrm{d}\varphi}\right)_\mathrm{j}\right] - E_\mathrm{j}}{\left(\frac{\partial u}{\partial T}\right)_\mathrm{j} + \frac{Z_\mathrm{j}}{K_\mathrm{j}}\frac{p_\mathrm{j}}{T_\mathrm{j}}\left(\frac{\partial u}{\partial p}\right)_\mathrm{j}} \tag{1}$$

where the coefficients Z_j, K_j and E_j are defined as:

$$Z_\mathrm{j} = 1 + \frac{T_\mathrm{j}}{R_\mathrm{j}}\left(\frac{\partial R}{\partial T}\right)_\mathrm{j} \tag{1a}$$

$$K_\mathrm{j} = 1 - \frac{p_\mathrm{j}}{R_\mathrm{j}}\left(\frac{\partial R}{\partial p}\right)_\mathrm{j} \tag{1b}$$

$$E_\mathrm{j} = \frac{p_\mathrm{j}}{B_\mathrm{j}}\left(\frac{\partial u}{\partial p}\right)_\mathrm{j}\left[\frac{1}{m_\mathrm{j}}\left(\frac{\mathrm{d}m}{\mathrm{d}\varphi}\right)_\mathrm{j} - \frac{1}{V_\mathrm{j}}\left(\frac{\mathrm{d}V}{\mathrm{d}\varphi}\right)_\mathrm{j} + \frac{1}{R_\mathrm{j}}\left(\frac{\partial R}{\partial \lambda}\right)_\mathrm{j}\left(\frac{\mathrm{d}\lambda}{\mathrm{d}\varphi}\right)_\mathrm{j}\right] \tag{1c}$$

Pressure change in each engine control volume is calculated by using ideal gas state equation:

$$p_\mathrm{j} = \frac{m_\mathrm{j}\,R_\mathrm{j}\,T_\mathrm{j}}{V_\mathrm{j}} \tag{2}$$

The above equation for temperature and pressure change in the control volume assumes that the heat exchange over system boundaries has the members in the heat of fuel combustion Q_f and heat transferred to chamber walls Q_w:

$$\sum\frac{\mathrm{d}Q}{\mathrm{d}\varphi} = \frac{\mathrm{d}Q_\mathrm{f}}{\mathrm{d}\varphi} + \frac{\mathrm{d}Q_\mathrm{w}}{\mathrm{d}\varphi} \tag{3}$$

The mass balance equation for the control volume is determined by mass exchange over system boundaries. The mass balance equation has members in the mass exchange on the inlet ports, exhaust (outlet) ports, special additional ports (for example the starting air valve) and through the seals:

$$\sum\frac{\mathrm{d}m}{\mathrm{d}\varphi} = \left(\frac{\mathrm{d}m}{\mathrm{d}\varphi}\right)_\mathrm{inl} + \left(\frac{\mathrm{d}m}{\mathrm{d}\varphi}\right)_\mathrm{exh} + \left(\frac{\mathrm{d}m}{\mathrm{d}\varphi}\right)_\mathrm{f} + \left(\frac{\mathrm{d}m}{\mathrm{d}\varphi}\right)_\mathrm{aA} + \left(\frac{\mathrm{d}m}{\mathrm{d}\varphi}\right)_\mathrm{leak} \tag{4}$$

The heat exchange over system boundaries is performed partially by sensitive heat of mass flow over system boundaries:

$$\sum h\frac{\mathrm{d}m}{\mathrm{d}\varphi} = \left(h\frac{\mathrm{d}m}{\mathrm{d}\varphi}\right)_\mathrm{inl} + \left(h\frac{\mathrm{d}m}{\mathrm{d}\varphi}\right)_\mathrm{exh} + \left(h\frac{\mathrm{d}m}{\mathrm{d}\varphi}\right)_\mathrm{f} + \left(h\frac{\mathrm{d}m}{\mathrm{d}\varphi}\right)_\mathrm{aA} + \left(h\frac{\mathrm{d}m}{\mathrm{d}\varphi}\right)_\mathrm{leak} \tag{5}$$

In all of the above equations used symbols are: T = operating medium temperature, φ = engine crankshaft angle, m = operating medium mass, Q = heat amount, p = operating medium pressure, V = operating area volume, u = operating medium specific internal energy, λ = excess air ratio, R = operating medium gas constant, h = specific enthalpy of operating medium, j = index for any engine control volume, f = fuel, w = control volume walls, inl = inlet, exh = exhaust (outlet), aA = additional air, leak = leakage.

Calorific gas properties (u, $\partial u/\partial\lambda$, $\partial u/\partial T$, $\partial u/\partial p$, $\partial R/\partial\lambda$, $\partial R/\partial T$, $\partial R/\partial p$) are modeled from the analytical expressions relating the temperature and gas composition, [13] and [14].

Main numerical model assumption says that in each engine cylinder was happen the same change of pressure and temperature (phase-shifted). Such assumption makes the numerical model as fast as possible.

3. Engine, turbocharger and air cooler characteristics

Analyzed engine is a high speed, turbocharged, four stroke gasoline engine with direct fuel injection. The engine is designed for application in passenger road vehicles. Main engine operating parameters and specifications are presented in Table 1.

Table 1. *Analyzed gasoline engine main operating parameters*

Number of cylinders	4
Cylinder bore	84 mm
Stroke	86 mm
Connecting rod length	129.8 mm
Cylinder clearance volume	0.0477 dm^3
Cylinder coolant	Water
Intake manifold volume	2.0 dm^3
Exhaust manifold volume	2.5 dm^3
Ignition order	1-3-4-2
Compression ratio	11
Fuel lower calorific value	43000 kJ/kg
Fuel density	750 kg/m^3

The engine uses one turbocharger for air pressure increase (turbocharger model KKK 30.60/13.21 [15]) with the main geometry characteristics presented in Table 2.

Table 2. *Turbocharger KKK 30.60/13.21 main geometry characteristics [15]*

Geometry parameter	Dimension
Charger inlet diameter	0.0457 m
Charger outlet diameter	0.0762 m
Inlet turbine flow surface	0.0013 m^2

First simulations with presented 0D numerical model were provided for turbocharged gasoline engine which characteristics and specifications were presented in Table 1 and Table 2. After obtaining required simulation results, numerical model is upgraded with air cooler after charger.

The main purpose of air cooling after the charger is to reduce air temperature and thus increase air density. In such way, more air can be delivered to engine cylinders (air pressure remains approximately the same as without cooler, it is reduced only for a small amount of pressure losses in the cooler).

For this analysis it is selected a standard automotive air cooler which is, in general, air-air heat exchanger. In numerical model, air cooler is described with proper equations for cooling medium (air) mass flow, pressure loss and overall cooler efficiency according to producer specifications [16]. A change in air cooler efficiency and pressure loss in relation to cooling air mass flow is shown in Fig. 1.

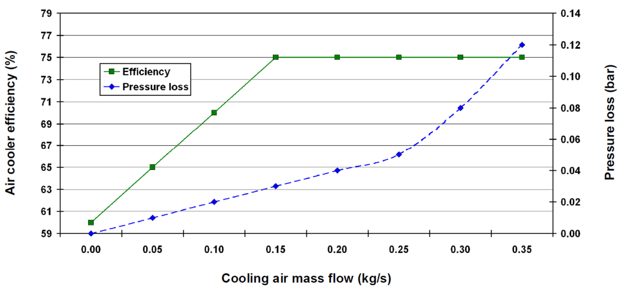

Fig. 1. *Air cooler efficiency and pressure loss in relation to cooling air mass flow*

After proper inclusion of air cooler sub-model into the entire 0D numerical model, it was performed second numerical simulations and proper required simulation results were obtained.

4. Numerical model results and discussion

Numerical model results were presented in this chapter in relation to engine rotational speed. For each engine parameter was presented a comparison between analyzed gasoline turbocharged engine and the same engine with air cooler included.

Developed power of turbocharged gasoline engine without air cooler continuously increases with an increase in the engine rotational speed from 18.50 kW at 1000 rpm to 176.99 kW at 6000 rpm, Fig. 2. Inclusion of air cooler into turbocharged engine process resulted with an increase in engine power for all observed rotational speeds, with the exception of the highest engine rotational speed.

At low engine rotational speeds (up to 3000 rpm), increase in engine developed power caused by air cooling process is not significant. In comparison with the engine without air cooler, air cooling process increases engine power for 0.32 kW at 1000 rpm, 1.25 kW at 2000 rpm and 4.90 kW at 3000 rpm. A significant increase in engine power when air cooling process is applied can be seen at 4000 rpm (22.07 kW) and at 5000 rpm (14.47 kW).

From 5000 rpm to the highest observed engine rotational speed of 6000 rpm, the engine developed power unexpectedly decreases during the usage of air cooling process, Fig. 2. At 6000 rpm, engine power developed with air cooler is even lower than engine power developed without air cooler (the difference is 2.23 kW).

Obtained results lead to a conclusion that the selected air cooler has satisfying characteristics at all engine rotational speeds except the highest one. Further analysis and air cooler optimization surely can improve the effects of air cooling process even at highest engine rotational speed.

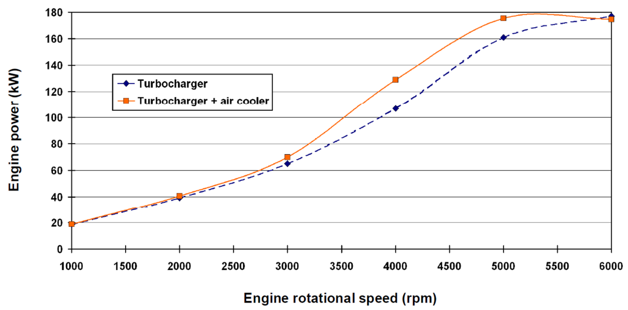

Fig. 2. *Change in power of a turbocharged engine with and without air cooler*

Change in analyzed engine torque with and without air cooler is presented in Fig. 3. For the engine without air cooler can be seen that torque continuously increases with an increase in rotational speed right until 5000 rpm. From 5000 rpm to 6000 rpm, engine torque decreases because on the highest engine rotational speeds, there is no need for a high torque to ensure smooth engine operation. Engine without air cooler develops maximum torque at 5000 rpm, and that torque amounts 307.45 Nm.

The air cooling process increases engine torque in each engine rotational speed except the highest one, when compared to engine without air cooler. The difference in engine torque of air cooled, in comparison with only turbocharged engine, increase during the increase in the engine rotational speed up to 5000 rpm. At 1000 rpm the difference in torque between those two engines amounts 3.07 Nm, at 2000 rpm it amounts 5.99 Nm and at 3000 rpm it amounts 15.58 Nm. The difference of engine torque is the highest, Fig. 3, at 4000 rpm (amounts 52.69 Nm) and at 5000 rpm (amounts 27.63 Nm).

For air cooled engine, as for the engine without air cooling, torque decreases from 5000 rpm to 6000 rpm. The decrease in engine torque of air cooled engine after 5000 rpm is too sharp, so at the highest engine rotational speed of 6000 rpm, engine torque is lower for air cooled engine in comparison with the only turbocharged engine. This trend of engine torque, at the highest engine rotational speeds, resulted with a decrease in engine power, Fig. 2, for air cooled engine. Engine torque is another parameter from which can be seen that operation of selected air cooler at the highest rotational speeds is not optimal.

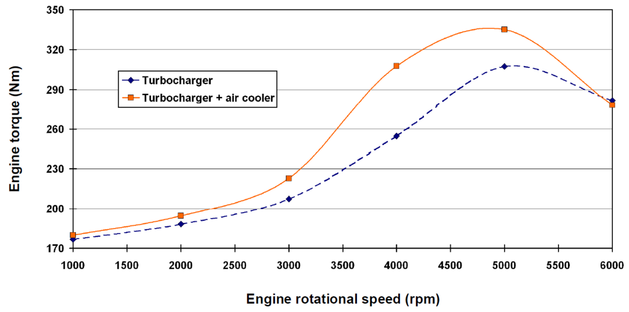

Fig. 3. *Change in torque of a turbocharged engine with and without air cooler*

In the scientific and professional literature widely can be found that air cooling process reduces engine brake specific fuel consumption. The same is obtained in this analysis, Fig. 4.

Analyzed turbocharged engine with air cooler has lower brake specific fuel consumption at each engine rotational speed when compared to the same engine without air cooler. Only exception is operating point at 5000 rpm, where the air cooled engine has slightly higher brake specific fuel consumption.

Air cooling process in this analysis reduces engine brake specific fuel consumption from 0.72 g/kWh at 1000 rpm up to the maximum value of 3.04 g/kWh at 4000 rpm in comparison with the same engine without air cooler.

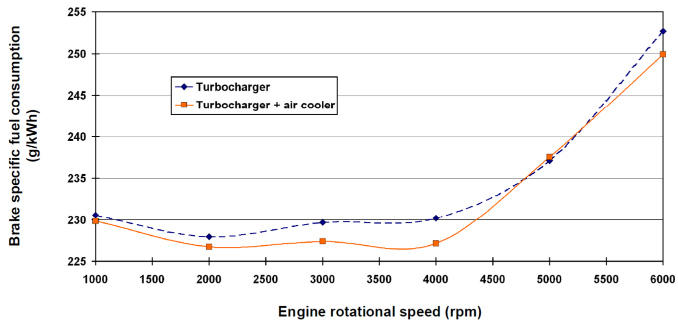

Fig. 4. *Change in brake specific fuel consumption of a turbocharged engine with and without air cooler*

The air cooling process gives higher maximum pressures in the engine cylinder when compared to engine without air cooling, Fig. 5. Only exception is again the highest engine rotational speed where the only turbocharged engine gives higher maximum cylinder pressure when compared to turbocharged engine with air cooling. As concluded before, air cooling operation at the highest analyzed engine rotational speed is not optimal, which can again be seen in the too sharp decrease in cylinder maximum pressure from 5000 rpm to 6000 rpm for the engine with air cooling.

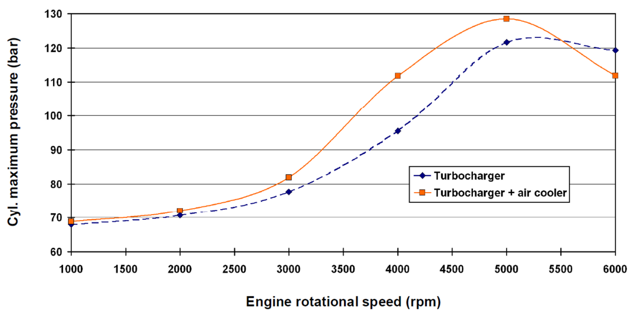

Fig. 5. *Change in cylinder maximum pressure of a turbocharged engine with and without air cooler*

Both analyzed engines maximum cylinder pressures reaches at 5000 rpm. For engine only with the turbocharger, maximum cylinder pressure at 5000 rpm amounts 121.70 bars, while at the same rotational speed, maximum cylinder pressure for engine with turbocharger and air cooler amounts 128.54 bars. For both engine variants the critical maximum cylinder pressure has not reached, which for such engines amounts approximately 170 bars.

Engine volumetric efficiency is a ratio of air mass brought to engine cylinders and air mass which can be brought to engine cylinders at the environment state. For both engines analyzed in this study, with and without air cooler, volumetric efficiency is higher then 100% from 3000 rpm up to 6000 rpm, as expected for turbocharged engines, Fig. 6.

At each engine rotational speed, except the highest one (6000 rpm), engine with air cooler has higher volumetric efficiency when compared to the engine only with a turbocharger. The highest differences in volumetric efficiency between two analyzed engines can be seen at 4000 rpm (124.35% for engine without air cooling in comparison with 148.00% for engine with air cooling) and at 5000 rpm (154.20% for engine without air cooling in comparison with 168.27% for engine with air cooling), Fig. 6.

Between engine rotational speeds of 5000 rpm and 6000 rpm, volumetric efficiency decreases for both engines. It should be noted that for engine with air cooler, between observed rotational speeds, decrease in volumetric efficiency is too sharp, so at 6000 rpm engine with air cooler has lower volumetric efficiency in comparison with an engine with turbocharger only.

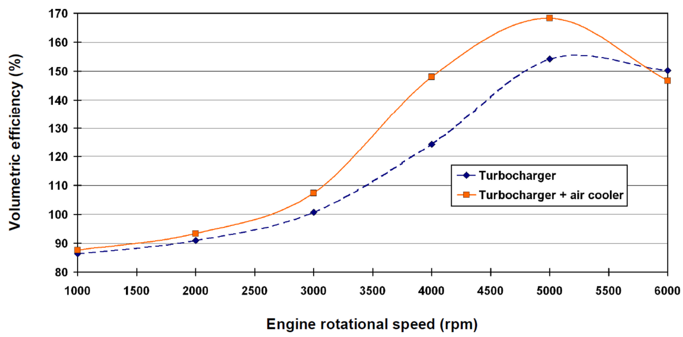

Fig. 6. *Change in volumetric efficiency of a turbocharged engine with and without air cooler*

For both analyzed engines, change in brake mean effective pressure and indicated mean effective pressure were presented in Fig. 7 and Fig. 8. Both mean effective pressures represent engine load, on the engine cylinder basis (indicated mean effective pressure) and on the basis of engine crankshaft outlet (brake mean effective pressure). When compared mean effective pressures for each observed engine, indicated mean effective pressure is higher than brake mean effective pressure at each rotational speed for engine mechanical losses between engine cylinders and crankshaft outlet.

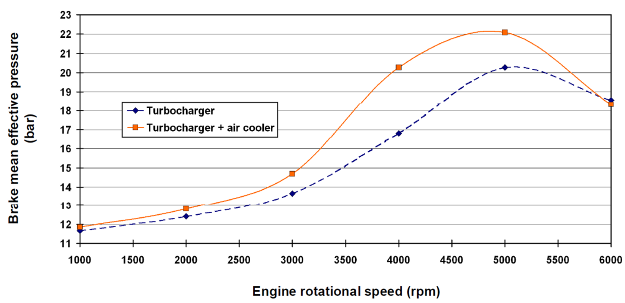

Fig. 7. *Change in brake mean effective pressure of a turbocharged engine with and without air cooler*

At each engine rotational speed, except the highest one, air cooled engine has higher brake mean effective pressure and indicated mean effective pressure in comparison with the engine without air cooler. From Fig. 7 and Fig. 8 can be seen that trends in brake mean effective pressure and indicated mean effective pressure, for both observed engines are the same.

Both mean effective pressures are the highest at 5000 rpm, for both observed engines. For engine without air cooler, maximum brake mean effective pressure amounts 20.27 bars, while maximum indicated mean effective pressure amounts 21.88 bars. Engine with air cooler has maximum brake mean effective pressure of 22.09 bar and maximum indicated mean effective pressure equal to 23.70 bar.

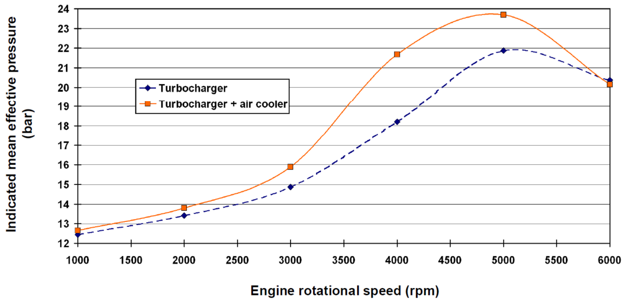

Fig. 8. *Change in indicated mean effective pressure of a turbocharged engine with and without air cooler*

5. Conclusions

This paper presents numerical analysis of air cooling process influences on the gasoline turbocharged engine operating parameters. Presented analysis was performed with numerical 0D (zero-dimensional) simulation model by using two sets of simulations. A first set of numerical model simulations was used to obtain required operating parameters of turbocharged gasoline engine without air cooler. Numerical model is then upgraded with air cooler sub-model. A second set of numerical model simulations of the same turbocharged gasoline engine was performed with air cooler included in calculation of engine operating parameters. Air cooling process influence on the analyzed engine operating parameters should be observed in two different fields of engine rotational speeds.

The first field of engine rotational speeds is between 1000 rpm and 5000 rpm. In this field of engine rotational speeds, air cooling process increases analyzed engine developed power and engine torque up to 20.67% and reduces brake specific fuel consumption up to 1.32% when compared with the engine without air cooling. In the same field of engine rotational speeds when compared with the engine without air cooler, air cooling process increases cylinder maximum pressure up to 17.03% (without reaching critical maximum cylinder pressure), increases engine volumetric efficiency up to 23.65% and increases engine brake and indicated mean effective pressures up to approximately 20%.

At the highest engine rotational speeds (between 5000 rpm and 6000 rpm) it was observed that selected air cooler, according to producer specifications, does not offer the required and expected performance, so air cooling process in this engine operating area can and should be improved.

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7. References

[1] Stone, R.: *Introduction to Internal Combustion Engines*, Fourth edition, Palgrave Macmillan, 2012.

[2] Yinhui, W., Rong, Z., Yanhong, Q., Jianfei, P., Mengren, L., Jianrong, L., Yusheng, W., Min, H., Shijin, S.: *The impact of fuel compositions on the particulate emissions of direct injection gasoline engine*, Fuel 166, p. 543–552, 2016. (doi:10.1016/j.fuel.2015.11.019)

[3] Kilicarslan, A., Qatu, M.: *Exhaust gas analysis of an eight cylinder gasoline engine based on engine speed*, Energy Procedia 110, p. 459 – 464, 2017. (doi:10.1016/j.egypro.2017.03.169)

[4] Elsemary, I. M. M., Attia, A. A. A., Elnagar, K. H., Elsaleh, M. S.: *Spark timing effect on performance of gasoline engine fueled with mixture of hydrogen-gasoline*, International Journal of Hydrogen Energy 42 (52), p. 30813-30820, 2017. (doi:10.1016/j.ijhydene.2017.10.125)

[5] Garrett, T. K., Newton, K., Steeds, W.: *The Motor Vehicle*, Thirteenth edition, Butterworth-Heinemann, 2001.

[6] Mahmoudi, A. R., Khazaee, I., Ghazikhani, M.: *Simulating the effects of turbocharging on the emission levels of a gasoline engine*, Alexandria Engineering Journal 56 (4), p. 737-748, 2017. (doi:10.1016/j.aej.2017.03.005)

[7] Alptekin, E., Canakci, M.: *Performance and emission characteristics of solketal-gasoline fuel blend in a vehicle with spark ignition engine*, Applied Thermal Engineering 124, p. 504-509, 2017. (doi:10.1016/j.applthermaleng.2017.06.064)

[8] Zhang, B., Sarathy, M.: *Lifecycle optimized ethanol-gasoline blends for turbocharged engines*, Applied Energy 181, p. 38–53, 2016. (doi:10.1016/j.apenergy.2016.08.052)

[9] Medica, V.: *Simulation of turbocharged diesel engine driving electrical generator under dynamic working conditions*, Doctoral Thesis, University of Rijeka, Rijeka, 1988.

[10] Mrzljak, V., Medica, V., Bukovac, O.: *Volume agglomeration process in quasi-dimensional direct injection diesel engine numerical model*, Energy, 115, 2016, p. 658-667 (doi:10.1016/j.energy.2016.09.055)

[11] Mrzljak, V., Medica, V., Bukovac, O.: *Simulation of a Two-Stroke Slow Speed Diesel Engine Using a Quasi-Dimensional Model*, Transactions of Famena, 2, 2016, p. 35-44 (doi:10.21278/TOF.40203)

[12] Mrzljak, V., Medica, V., Bukovac, O.: *Quasi-Dimensional diesel engine model with direct calculation of cylinder temperature and pressure*, Technical Gazette 24(3), p. 681-686, 2017. (doi:10.17559/TV-20151116115801)

[13] Jankov, R.: *Mathematical modeling of flow, thermodynamic processes and engine operation characteristics*, Naučna knjiga, Beograd, Part 1 and Part 2, 1984.

[14] Mrzljak, V.: *Quasi-dimensional model for numerical simulations of marine two-stroke diesel engine*, Doctoral Thesis, Rijeka, University of Rijeka, 2015.

[15] Škifić, N.: *Influence analysis of engine equipment parameters on diesel engine characteristics*, Doctoral Thesis, Rijeka, University of Rijeka, 2003.

[16] http://www.mahle.com/en/ (accessed: 06.10.2018.)

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