

Numerična analiza zaporednega tlačnega polnjenja pri hitrotekočih dizelskih motorjih

Numerical Analysis of Sequential Turbocharging for High Speed Diesel Engines

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Tlačno polnjeni dizelski motorji za pogon cestnih in zunajcestnih vozil delujejo v širokem področju obremenitev in vrtilnih frekvenc, zato je izredno težko zadostiti potrebam motorja po zgorevalnem zraku samo z enim turbopuhalom. Poznamo različne načine tlačne polnitve, s katerimi lahko ta problem omejimo. V tem članku je analiziran način tlačnega polnjenja motorja z dvema (po potrebi) zaporedno vezanima turbopuhalom. Ti rezultati so primerjani z rezultati, dobljenimi na istem motorju, vendar enkrat z enim, običajnim turbopuhalom, drugič pa s turbopuhalom, ki ga poganja plinska turbina s spremenljivo geometrijsko obliko vodilnika. Izvedena je tudi primerjava karakteristik motorja z vsemi tremi izvedbami pri ustaljenem in v prehodnem načinu delovanja. Rezultati kažejo določene prednosti zaporednega dvojnega tlačnega polnjenja za ustaljeno delovanje in delovanje motorja v prehodnih režimih.

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(Ključne besede: motorji dieselski, polnjenje tlačno, modeliranje numerično, karakteristike)

Turbocharged diesel engines for road-going and off-road vehicles operate over a wide range of engine speeds and loads. It is very difficult to cover effectively the whole operational range using only one turbocharger, so many new strategies in the supercharging of such engines are available. In this paper, a Diesel engine using a sequential turbocharging system, comprising two turbochargers is analyzed. Results are compared with the same engine using a single turbocharger, with fixed turbine geometry, matched to the engine's operational range, and to the same engine equipped with a turbocharger with variable turbine geometry. Steady state and transient operational characteristics are compared. The results show some advantages of the sequential turbocharging system in steady state and transient engine operation.

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(Keywords: Diesel engine, turbocharging, numerical modelling, characteristics)

0 UVOD

Tlačno polnjenje tovornih vozil so uvedli že leta 1957 [2]. Uporaba in učinek tlačne polnitve sta preseгла vse meje, ki jih omogoča način polnjenja valjev pri danih okoliščinah. Prednosti tlačno polnjenih motorjev se kažejo v visokih vrednostih srednjega efektivnega tlaka in majhni emisiji škodljivih snovi v izpušnih plinih.

Pomanjkljivosti tlačno polnjenih motorjev so povezane z ožjim uporabnim področjem vrtilnih frekvenc motorja (zaradi česar včasih potrebujemo menjalnike z več prestavami), nižjim vrtilnim momentom motorja pri nizkih vrtilnih frekvencah in počasnejšim odzivom motorja pri spremembi obremenitve.

Tlačno polnjeni dizelski motorji za pogon gospodarskih cestnih vozil delujejo v širokem razponu obremenitev in vrtilnih frekvenc. Pri razvoju sodobnih tlačno polnjenih motorjev zasledimo

0 INTRODUCTION

Turbocharging was introduced for Diesel truck engines in 1957 [2]. Its usage has grown to the point where it is no longer possible to find naturally aspirated Diesel engines in this type of application. The advantages of turbocharged Diesel engines are in high-medium effective pressures and low emissions.

The disadvantages of turbocharged Diesel engines (with a single turbocharger) are: a narrow useful engine speed range (so a multi-speed gearbox has to be used), low torque at low speeds and a very slow transient response.

Turbocharged Diesel engines for road-going and off-road vehicles operate over a wide range of engine speeds and loads. Recent developments in turbocharging systems have provided results which

tehnične rešitve, ki prej omenjene probleme uspešno omejujejo.

Pomembno pomanjkljivost pri pospeševanju in obremenjevanju tlačno polnjenih dizelskih motorjev pomeni primanjkljaj zraka za zgorevanje v valjih. Pri nizkih vrtilnih frekvencah in majhni obremenitvi motorja je tlak polnitve nizek. Zaradi posledično majhne mase zraka v valju ne more kakovostno zgoreti povečana količina goriva. Potrebni višji tlak zraka za zgorevanje lahko omogoči le puhalo, ki se vrti hitreje. Pospešiti puhalo na visoke vrtilne frekvence v kratkem času pa ni lahka naloga. Poznamo različne načine, kako čimprej doseči višje vrtilne frekvence rotorja puhala: impulzni način tlačnega polnjenja, zmanjšanje masnega vztrajnostnega momenta rotorja puhala, uporaba plinske turbine s časovno spremenljivo geometrijsko obliko vodilnika (ČSG), prilagajanje (zmanjševanje) vbrizgane količine goriva dejanski količini zraka v valjih, pomožne naprave za pospeševanje rotorjev z zunanjim virom energije ipd.

Vrtilni moment motorja pri nizkih vrtilnih frekvencah lahko povečamo z zmanjšanjem pretočnega prereza vodilnika turbine, oziroma vsega spiralnega vodilnika turbine – okrova plinske turbine. Večje pretočne prereze skozi turbino, ki jih potrebujemo pri višjih obremenitvah in vrtilnih frekvencah motorja, omogoča krmiljeni obtočni vod za izpušne pline mimo turbine, ki obenem krmili tudi tlak polnilnega zraka za puhalom. Boljšo tehnično rešitev pomeni vzporedno tlačno polnjenje z uporabo dveh ali več vzporedno vezanih turbopuhal, ali pa uporaba plinske turbine s spremenljivo geometrijsko obliko vodilnika.

Pri vzporednem tlačnem polnjenju uporabljamo dve vzporedno priključeni turbopuhali, ki pokrivata potrebe po zraku tudi pri polni obremenitvi in visokih vrtilnih frekvencah motorja. Glede na potrebe po zgorevalnem zraku, lahko tako vključimo vzporedno dve turbopuhali (sl. 1). S pnevmatskim krmilnikom dovajamo plinski(ma) turbini(ama) več ali manj izpušnih plinov.

V novejšem času pogosto srečujemo tudi načine tlačne polnitve s turbino, ki ima spremenljivo (nastavljivo) geometrijsko obliko vodilnika. Vodilne

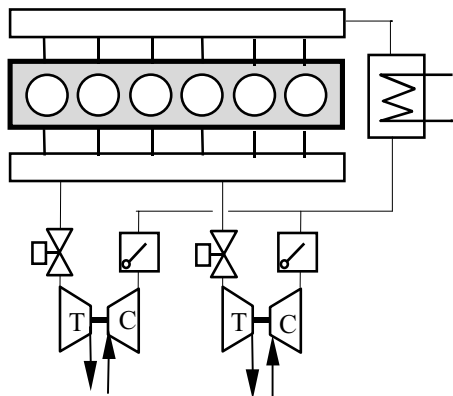
eliminate the disadvantages in former Diesel engines.

One of the main drawbacks in transient load acceptance or vehicle acceleration with turbocharged Diesel engines is the lack of sufficient air in the engine's cylinders. The supercharging pressure is low at low engine speeds and loads and there is not enough air to combust the increased quantities of injected fuel. To reach the required air mass in the engine's cylinder, the supercharging pressure must be increased, and this is achieved by higher turbocharger speeds. It is necessary that the turbocharger rotor very quickly reaches the elevated speed; this is not an easy task. To obtain the faster turbocharger rotor speeds there are many strategies: a pulse charging system, lower turbocharger rotor inertia, use of variable turbine geometry (VTG), fuel injection limitation according to the supercharging air pressure, speed assistance from other energy sources, etc.

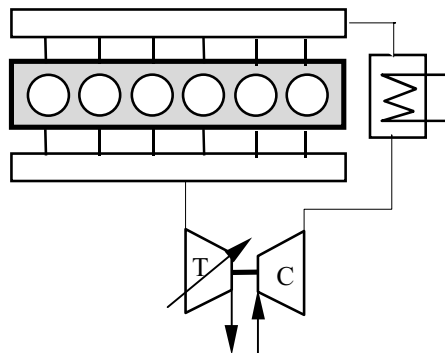
To increase the turbocharged engine torque at lower engine speeds it is possible to reduce the turbine flow area (or to select a smaller turbine housing), which would increase the supercharging pressure at lower engine speeds and loads. To enable higher flow areas at elevated engine speeds there is the possibility of introducing a waste gate valve, which also limits the supercharging pressure. However, better solutions are the use of sequential turbocharging or turbocharger with variable turbine geometry.

Sequential turbocharging makes use of two or more turbochargers to meet the engine's air requirements at full load. Turbochargers are switched on to operate in parallel, depending on the engine load and speed, (Fig. 1). Switching is achieved on the turbine access by pneumatic devices.

A more recently introduced solution is the use of variable turbine geometry (Fig. 2). This is achieved by using turbine guide vanes which can



Sl. 1. Tlačno polnjenje z dvema vzporedno priključenima turbopuhalom
Fig. 1. Sequential turbocharging



Sl. 2. Tlačno polnjenje s turbino s spremenljivo geometrijsko obliko statorja
Fig. 2. Turbocharging by variable turbine geometry

lopatice so pri tej izvedbi vrtljive in spreminjajo pretočni prerez, ali pa pretočni prerez spreminjamo z uporabo premične bočne stene vodilnika turbine. Obe izvedbi spreminjata pretočni prerez turbine in ga prilagajata trenutnim optimalnim razmeram delovanja motorja.

V tem članku je prikazana primerjava delovanja motorja z vzporednim načinom vezave dveh puhal in polnjenja z eno – običajno plinsko turbino, obenem pa je primerjava opravljena tudi za primer uporabe turbine s spremenljivo geometrijsko obliko vodilnika. Vzporedni način vklopjanja in izklopjanja turbopuhala obravnava uporabo dveh enako velikih turbopuhal (pri uporabi različno velikih turbopuhal bi se srečali s problemom vpliva različnih tlačno – pretočnih karakteristik turbopuhal v različnih razmerah delovanja motorja). Pri nižjih vrtilnih frekvencah in obremenitvah motorja je vklopljeno le eno turbopuhalo, pri delovanju motorja v zgornjem delovnem področju vrtilnih frekvenc in obremenitev pa sta vzporedno priklopljeni obe turbopuhali.

Prikazana primerjava obsega rezultate numerične obravnave sistema motorja in turbopuhala.

1 RAČUNSKI MODEL

Tlačno polnjeni dizelski motor lahko razdelimo na nadzorne prostornine (sl.3) s homogeno sestavo snovi (stanjem), ki se s časom spreminja. Ta ničdimenzijski model je opisan z energijsko in kontinuitetno enačbo (3) ter začetnimi in robnimi pogoji. Nekatere komponente, npr. plinska turbina, puhalo, sistem za vbrizgavanje goriva itn., so opisani s karakterističnimi polji. Delovna snov – zmes plinov je obravnavana kot zmes idealnih plinov. Spremembo vrtilne frekvence motorja in turbopuhala določimo iz ravnotežja momentov in ohranitve kinetične energije.

Celotni sistem motorja je opisan z vektorji opisnih indeksov nadzornih prostornin (sl.3). Povezave med posameznimi nadzornimi prostorninami so opisane z matriko indeksov povezav (sl.3). Takšen način opisa motorja omogoča, da se program za numerično simuliranje prilagaja različnim izvedbam motorja in celovito pomaga pri uspešnem reševanju sistema enačb za različne kombinacije opreme motorjev.

Kontinuitetna enačba za nadzorne prostornine je:

$$\left(\frac{dm}{d\varphi}\right)_i = \sum_j \left(\frac{dm}{d\varphi}\right)_{ij} + \left(\frac{dm_f}{d\varphi}\right)_i \quad (1).$$

Energijsko enačbo nadzorne prostornine lahko zapišemo v obliki:

$$\left(\frac{dQ}{d\varphi}\right)_i = \sum_j \left(h_k \frac{dm}{d\varphi}\right)_{ij} + \left(\frac{dQ_f}{d\varphi}\right)_i + \left(\frac{dQ_w}{d\varphi}\right)_i \quad (2).$$

Z uporabo prvega glavnega zakona termodinamike in zakona o ohranitvi mase lahko

be adjusted to various angles or by using a fixed turbine nozzle ring and a moving sidewall. Both solutions enable a decreasing or increasing turbine flow area and a good match with the engine's needs.

In this paper a comparison of sequential turbocharging with standard turbocharging, using a single turbocharger, and with variable turbine geometry turbocharging, is presented. The sequential turbocharging is performed using two turbochargers of equal size (which is necessary to avoid problems with flow patterns at various pressure ratios in turbochargers of different size). At low engine loads and speeds only one turbocharger is in operation. At higher loads, both turbochargers are in operation.

The comparison is performed using a numerical simulation of the complete turbocharged Diesel engine system.

1 NUMERICAL MODEL

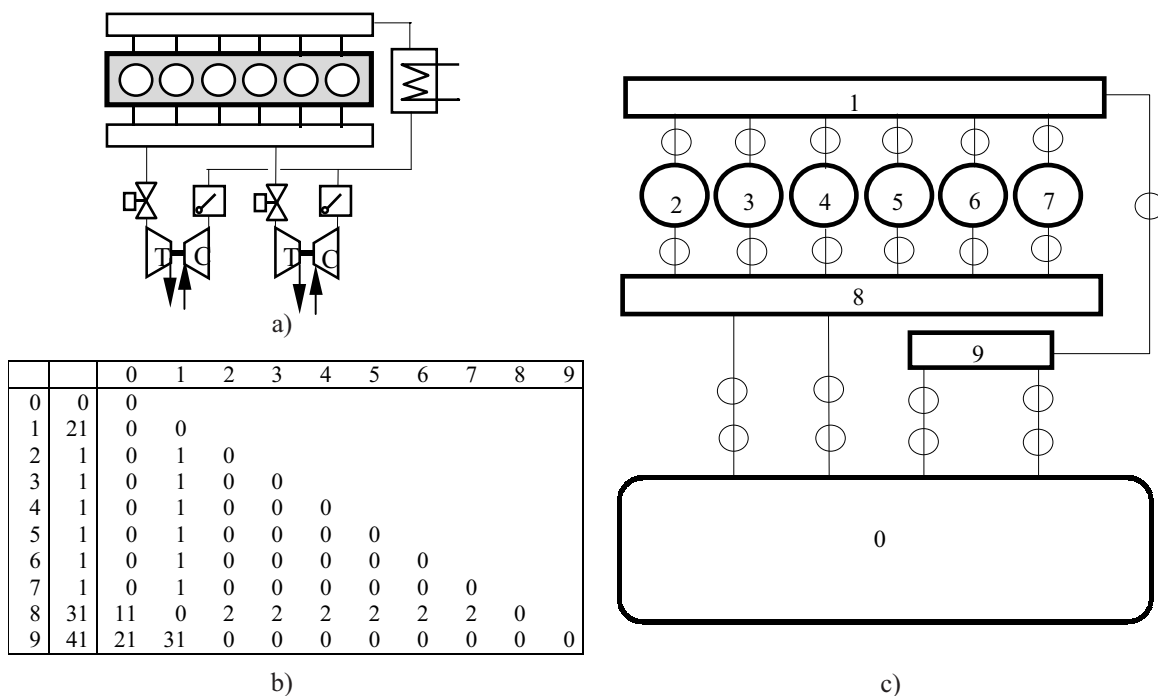
A turbocharged Diesel engine system can be divided into control volumes (Fig. 3) with space homogeneity and time dependent changes. This zero-dimensional model is described by two governing equations of mass and energy conservation [3], along with initial and boundary conditions. Some components, such as: exhaust turbine, charger, injection equipment, etc. are represented by a steady-state field of characteristics. The gas properties are considered to be the same as for an ideal gas mixture. Engine and turbocharger speed changes are calculated from the torque balance and kinetic energy conservation.

The system is described using control volumes description indices, which are arranged in vector (Fig. 3). The connections between various control volumes are described by connection matrix indices (Fig. 3). This method of description enables the numerical simulation program to be adapted to any given engine system and to be governed in an efficient way, without the necessity to adapt the program to a particular engine system.

The mass conservation equation for control volumes is:

The energy conservation equation for control volumes is:

Combining the first law of thermodynamics and the mass conservation equation, the basic



Sl. 3. Shematski prikaz sistema tlačne polnitve dizelskega motorja z vzporedno vezanima turbopuhalom
 a) funkcionalna shema motorja b) vektor opisnih indeksov prostornin in matrika indeksov povezav
 c) diskretizacija sistema

Fig. 3. System description for a Diesel engine with sequential turbocharging
 a) System layout, b) Description indices in vector of volumes and connection matrix,
 c) System discretization

dobimo osnovno enačbo za spremembo temperature medija v nadzorni prostornini za idealni plin:

equation for temperature change in the control volume for ideal gas properties can be derived as:

$$\left(\frac{dT}{d\varphi}\right)_i = \frac{\left(\frac{dQ}{d\varphi}\right)_i - p_i\left(\frac{dV}{d\varphi}\right)_i - u_i\left(\frac{dm}{d\varphi}\right)_i - m_i\left(\frac{\partial u}{\partial \lambda}\right)_i\left(\frac{d\lambda}{d\varphi}\right)_i}{m_i\left(\frac{\partial u}{\partial T}\right)_i} \quad (3).$$

Enačbo stanja za zmes idealnih plinov zapišemo v obliki:

The state equation for an ideal gas mixture is:

$$p_i = \frac{m_i R_i T_i}{V_i} \quad (4).$$

Med nadzornimi prostorninami prihaja do izmenjave mase (z masnim tokom) in energije (z entalpijo snovi). Skupaj z masnim tokom plinov potujejo tudi delci rezultatov zgorevanja, zato spremembo sestave izpušnih plinov spremljamo s sprotnim določanjem razmernika zraka.

Between the control volumes, mass is interchanged through mass flow, and energy through enthalpy flow. Together with the mass flow, species produced by the combustion process are transferred, so it is necessary to model the change of air excess ratio due to the mass transfer.

Robni pogoji, ki opisujejo zgradbo – komponente motorja, so vgrajeni v matematični model. Sem prištevamo obremenitev motorja na sklopki, razvito moč motorja, vrtilno frekvenco motorja, vrtilno frekvenco turbopuhala, lego ročice za gorivo, maso vbrizganega goriva itn.

The boundary conditions relating to the system are integrated in to the mathematical model. They are: the engine load on coupling, engine power, engine rotation speed, turbocharger speed, fuel rack position, injected fuel mass, etc.

Model je dopolnjen s karakteristikami plinske turbine, puhala, karakteristikami opreme za vbrizgavanje goriva, modelom krmilnika motorja ter modeli dinamike motorja in turbopuhala.

The complete model is equipped with: turbine characteristics, charger characteristics, fuel injection equipment characteristics, engine controller model, engine dynamics and the turbocharger dynamics.

Dinamiko rotorja turbopuhala lahko opišemo z enačbo:

The turbocharger rotor dynamic is described by the equation:

$$J_{TC} \frac{d\omega_{TC}}{dt} = M_T + M_C \quad (5).$$

Dinamiko motorja opisuje enačba:

The engine dynamics are described by the equation:

$$(J_{DE} + J_V) \frac{d\omega}{dt} = M_{DE} + M_V \quad (6).$$

Potrebna moč motorja za pogon vozila pri nespremenjeni hitrosti vozila je:

The power to drive the truck with constant speed is:

$$P_V = (f_r m_V g + \rho c_w A \frac{v^2}{2} + m_V g \sin \alpha) v \quad (7).$$

Spremembo hitrosti vozila opišemo z enačbo:

The change of the truck's speed is given by:

$$\frac{dv}{dt} = \frac{(P_{DE} - P_V)}{m v} \quad (8).$$

Sistem nelinearnih diferencialnih enačb rešujemo s postopkom Runge-Kutta-Fehlberg s samodejnim preverjanjem časovnega koraka integracije.

The set of nonlinear differential equations is solved using the Runge-Kutta-Fehlberg method with an automatic control of the integration time step.

2 PRIMERJAVA ČASOVNO USTALJENIH KARAKTERISTIK MOTORJA

2 COMPARISON OF STEADY STATE ENGINE CHARACTERISTICS

Uporaba zgoraj opisanega modela za računsko simuliranje je izvedena na sodobnem 6-valjnem motorju za pogon cestnih vozil z močjo 205 kW/ 2500 min⁻¹ za prej omenjene tri različne izvedbe tlačnega polnjenja istega motorja. V prvi različici je obravnavana običajna, tovarniška izvedba motorja z enim turbopuhalom in sunkovitim sistemom natoka plinov v turbino. Karakteristični pretočni prerez turbine znaša 9,3 cm². Pri drugi izvedbi uporablja motor turbino z zvezno spremenljivo geometrijsko obliko (prerezom) vodilnika (ČSG, sl. 2). Analiza rezultatov je pokazala, da je bil pri delovanju motorja s turbino ČSG izrabilen le del možnega območja krmiljenja pretoka (spremembe prereza turbine), in sicer od 7,44 cm² (pri vrtilni frekvenci motorja 1100 min⁻¹) do največ 9,58 cm² (pri 2500 min⁻¹).

The described numerical simulation model is performed on a contemporary six cylinder in-line truck engine with a nominal power of 205 kW at 2500 rpm for the three different turbocharging systems and the same engine. First is the normal production engine, fitted with a single turbocharger with pulse charging, matched to the engine. The turbine flow area was 9.3 cm². Second is the same Diesel engine turbocharged by the turbocharger with variable turbine geometry (VTG, Fig. 2). From the analysis it was evident that only moderate changes of turbine flow area are adequate for this engine. The turbine flow area of the original turbocharger was changed linearly with engine speed from 7.44 cm² at 1100 rpm to 9.58 cm² at 2500 rpm.

Pri tretji različici obravnavamo sistem dveh enako velikih turbopuhal, ki se vklapljata in izklapljata glede na trenutne potrebe motorja (sl.1). Za obe turbopuhali velja, da sta hidravlično podobni puhalu osnovne različice: pri enakih specifičnih pretokih in enakih tlakih delujeta tudi z enakim izkoristkom. Skupni masni tok obeh pihal je znašal pri enakem tlačnem razmerju nekoliko manj kakor izmerjeni masni tok skozi osnovno izvedbo (enojno puhalo) turbopuhala. Obe puhali sta poganjali turbini s polovičnim imenskim karakterističnim prerezom spirale vodilnika: vsak posamezni tipični prerez vodilnika turbine je znašal 4,65 cm².

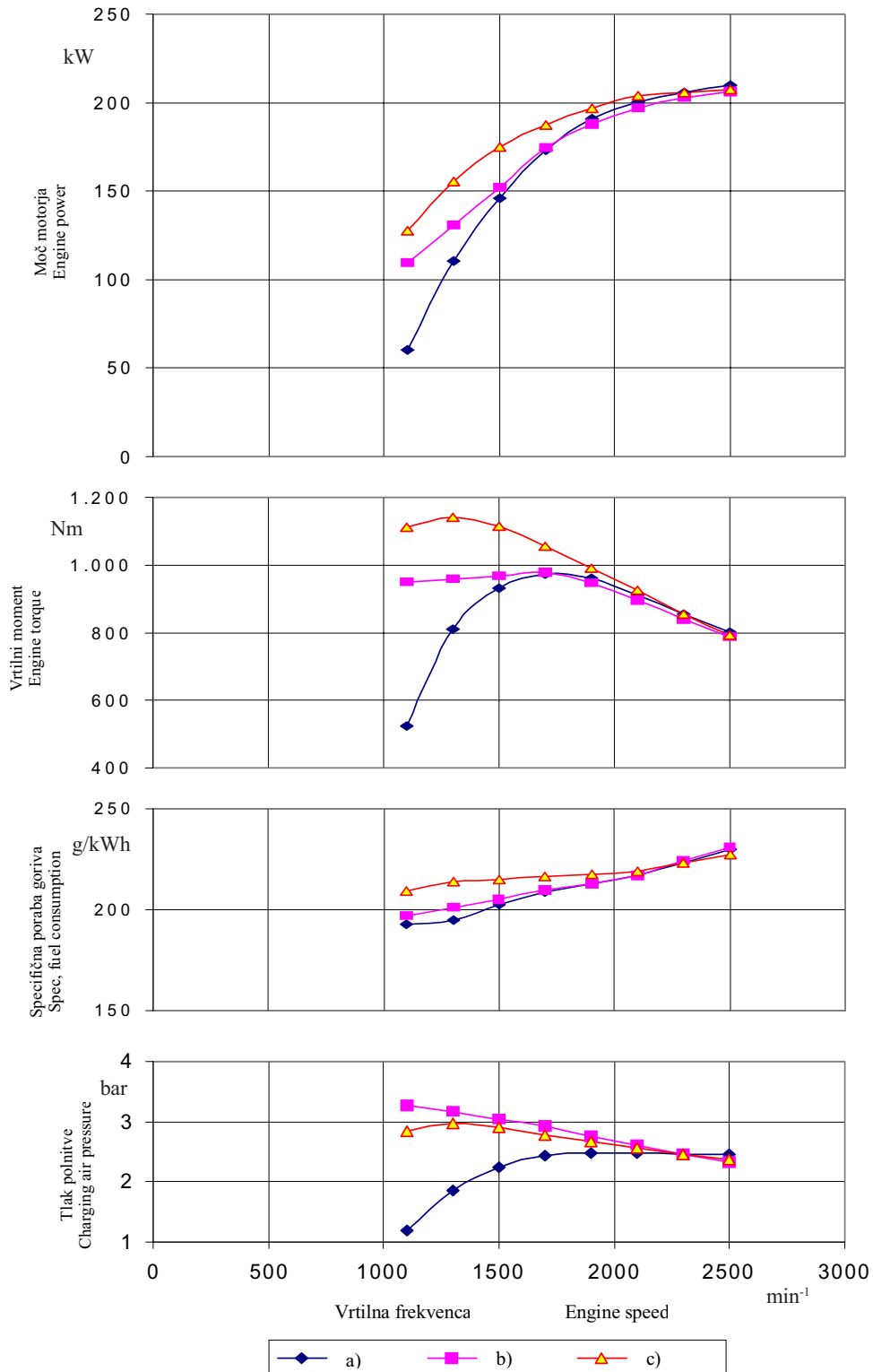
The third turbocharging system is based on the same Diesel engine fitted with two turbochargers of equal size, these are brought into operation depending on engine demand (Fig. 1). Both turbochargers were selected as hydraulically similar with the same pressure and flow coefficients, and the same efficiencies. The resulting cumulative air flow at the same pressure ratio was a little bit lower than with the original turbocharger. Both turbochargers have radial turbines with a constant flow area of 4.65 cm².

Na sliki 4 je prikazana primerjava izračunanih moči, vrtilnih momentov, specifične porabe goriva in tlaka polnitve, za vse tri izvedbe, ter polno obremenitev motorja. Vbrizgana količina goriva je bila omejena s tlakom delovnega zraka oziroma z najmanjšim potrebnim

The Fig. 4 shows a comparison of: calculated engine power, engine torque, specific fuel consumption and charging air pressure for the three different turbocharging systems under full load. The injected fuel was limited with respect to the charging air pressure to an air excess ratio of $\lambda_{min} = 1.5$.

razmernikom zraka $\lambda_{\min} = 1,5$. Pri vzporednem načinu tlačne polnitve sta hkrati delovali obe turbopuhali.

For the sequential turbocharging system, both turbochargers were in operation.



Sl. 4. Primerjava izračunanih parametrov motorja za različne izvedbe tlačne polnitve na istem motorju
 a) z enim turbopuhalom, b) z dvema vzporedno vezanima turbopuhalom,
 c) s turbopuhalom in turbino s spremenljivo geometrijsko obliko statorja
 Fig. 4. Comparison of calculated engine parameters for various turbocharging systems:
 a) standard, b) sequential, c) variable turbine geometry

Iz diagramov je razvidno, da obstaja očitna razlika med pomembnimi parametri motorja za vse tri opazovane izvedbe. Vidno najmanjšo moč motorja in vrtilni moment dobimo z osnovno opremo motorja.

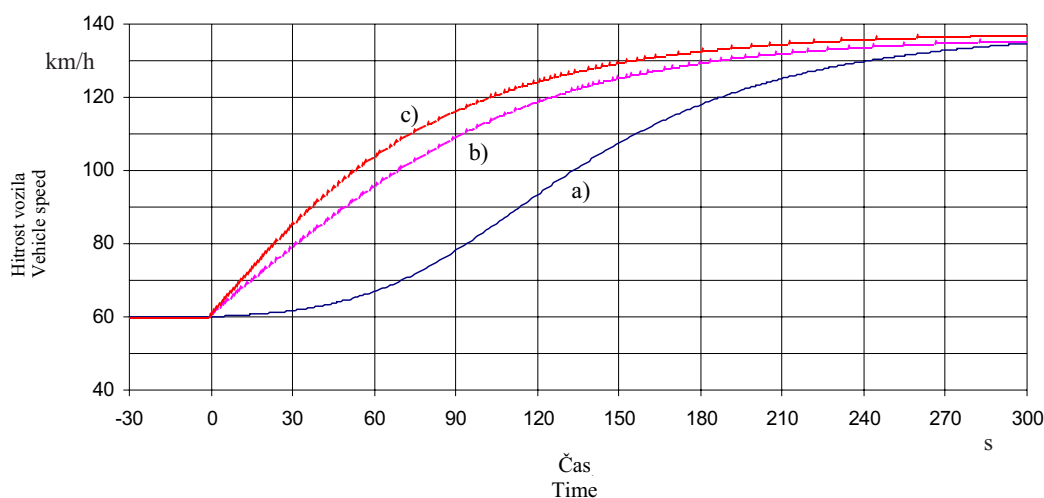
Tlačna polnitev s pomočjo turbine s ČSG zagotavlja zelo velik vrtilni moment in moč motorja pri nizkih vrtilnih frekvencah, specifična poraba goriva pa je v tem primeru visoka. V primerjavi z osnovno izvedbo motorja, sta se moment in moč povečala za 112 odstotkov pri 1100 min^{-1} . Pri isti vrtilni frekvenci se je poraba goriva relativno povečala za 8,5 odstotkov, tlak polnitve pa za 138 odstotkov.

Sistem tlačne polnitve z dvema vzporedno vezanima puhaloma omogoča (v primerjavi z osnovno izvedbo motorja) večji vrtilni moment in tlak polnitve pri nižjih vrtilnih frekvencah brez bistvenega povečanja specifične porabe goriva. Pri tej primerjavi sta se moč in vrtilni moment sorazmerno povečala za 81,2 odstotkov pri 1100 min^{-1} . Specifična poraba goriva se je povečala za 2,2 odstotkov, tlak polnitve pa za 173 odstotkov.

Prikazani rezultati jasno nakazujejo prednost uporabe vzporednega delovanja dveh turbopuhal za doseganje velikega vrtilnega momenta motorja pri nižjih vrtilnih frekvencah in zmerni specifični porabi goriva.

3 PRIMERJAVA DINAMIČNEGA ODZIVA MOTORJA

Primerjava dinamičnega odziva dizelskega motorja z različnimi načini tlačnega polnjenja je opravljena s poskusom pospeševanja tovornega vozila s skupno maso 20 t v šesti prestavi na ravni cesti s hitrosti 60 km/h na končno hitrost 130 km/h. Na sliki 5 so prikazani rezultati izračuna spremembe hitrosti vozila v odvisnosti od časa.



Sl. 5. Pospeševanje vozila s skupno maso 20t s hitrosti 60 km/h na hitrost 130 km/h za različne izvedbe tlačne polnitve: a) z enim turbopuhalom, b) z dvema vzporedno vezanima turbopuhalom, c) s turbopuhalom in turbino s spremenljivo geometrijsko obliko statorja

Fig. 5. Acceleration of the 20 t truck from 60 to 130 km/h
a) standard, b) sequential, c) variable turbine geometry

As can be seen, there is a visible difference in the engine parameters for the various turbocharging systems. It is evident that the engine power and the torque at lower engine speeds are minimal for the standard engine.

Turbocharging by variable turbine geometry has advantages at higher levels of engine power and torque at lower engine speeds, but the specific fuel consumption is higher. The engine power and torque increase is 112 % when compared to the standard engine at the engine speed of 1100 rpm. At the same speed, specific fuel consumption is increased by 8.5% and the charging air pressure is higher by 138%.

Sequential turbocharging offers elevated torque and charging air pressure at lower engine speeds, without a significant increase in the specific fuel consumption, when compared with the standard engine. The engine power and torque increase is 81.2 % at an engine speed of 1100 rpm. At the same speed, the specific fuel consumption increase is 2.2 % and charging air pressure is higher by 173 %.

The presented results clearly indicate the possibilities of sequential turbocharging at lower engine speeds to maintain high engine torque with a moderate specific fuel consumption.

3 COMPARISON OF TRANSIENT RESPONSE OF THE ENGINE

The transient response of Diesel engines, using various turbocharging systems, is compared in a trial to increase the speed of a 20 t truck in 6th gear from constant speed of 60 km/h to 130 km/h on a horizontal straight track. Fig. 5 presents the calculated results for truck speed change during this transient.

Potreben čas za pospešitev vozila z motorjem, ki je imel običajno turbopuhalo, je znašal 242 s. Motor s puhalom in turbino VGT je omogočil doseganje večje moči in zato najkrajši potreben čas pospeševanja vozila. Motor z dvema vzporednima turbopuhalom je potreboval za pospeševanje 20 odstotkov več časa, osnovna izvedba motorja pa kar 54 odstotkov daljši čas v primerjavi z motorjem, ki je imel turbopuhalo s turbino s spremenljivo geometrijsko obliko vodilnika. Motor z dvema turbopuhalom je porabil za celoten proces pospeševanja najmanj goriva. Motor s ČSG je porabil za isti proces 26,3 odstotkov več goriva, motor z običajnim – enojnim turbopuhalom pa kar 43,3 odstotkov več goriva.

Preglednica 1 prikazuje rezultate računskega simuliranja: čas, ki je potreben, da se tovorno vozilo pospeši s hitrosti 60 km/h na hitrost 130 km/h, ter maso porabljenega goriva za omenjeno pospeševanje in vse tri načine tlačnega polnjenja istega motorja. Pri izračunu je bilo upoštevano, da je bila začetna moč motorja prilagojena ustaljeni vožnji vozila s hitrostjo 60 km/h. V času $t = 0$ s je bila ročica za gorivo v trenutku prestavljena v lego največje dobave goriva. Ker obravnavamo hitrotekoči dizelski motor z majhnim, časovno zelo odzivnim turbopuhalom, lahko predpostavimo, da pri pospeševanju turbopuhala ni bilo primanjkljaja energije, čas pospeševanja turbopuhala je zelo kratek v primerjavi s časom pospeševanja vozila. Čas pospeševanja obeh manjših turbopuhala pri vzporednem načinu vklapljanja je krajši od časa pospeševanja turbine s spremenljivo geometrijsko obliko vodilnika, čeprav doseže ta turbina tudi pri pospeševanju zelo velikih izkoristkov.

The time to increase the speed of the truck from 60 to 130 km/h for the standard engine is 242 s. The turbocharged Diesel engine with variable turbine geometry exhibits the fastest increase in speed. The sequentially turbocharged Diesel engine reached the same speed in a 20 % longer time. The standard engine required 56 % more time than the engine with the VTG turbocharger. When comparing the fuel consumed to speed-up the truck, the sequentially turbocharged engine has the lowest consumption. The VTG turbocharged diesel engine uses 26.3 %, and the standard engine 43.3 % more fuel to do the same job.

The table 1 presents the results of calculations: time spent to accelerate the truck from 60 to 130 km/h and the amount of fuel consumed to perform this task. In the simulation it is assumed that the engine power was in balance with the necessary power to drive the vehicle at a speed of 60 km/h. At the time $t = 0$, the fuel rack was instantaneously set to the position of maximum fuel delivery. As the Diesel engine (DE) is at high speed and the turbochargers are small, there is no evident lack of energy due to the turbocharger transient. The transient characteristic times of this turbocharged Diesel engines are relatively small, when compared with the duration of the vehicle's acceleration transient. The transient time of the smaller turbochargers, used in sequential turbocharging, is shorter than that of the VGT turbocharger, despite very good use of the exhaust gas energy in this turbocharger.

Preglednica 1. *Potreben čas in potrebna količina goriva za pospeševanje tovornega vozila z maso 20 t*
Table 1. *Time and fuel consumption to accelerate the 20 t truck*

	Osnovna izvedba Standard Diesel engine	Izvedba z dvema puhaloma Sequential turbocharged DE	Izvedba s turbino ČSG VGT turbocharged DE
Čas pospeševanja Time for acceleration s	242	186	155
Poraba goriva Consumed fuel mass kg	2,021	1,409	1,781

4 SKLEP

Rezultati numeričnega modeliranja tlačno polnjenega motorja na ustaljenem in prehodnih režimih delovanja jasno prikazujejo določeno prednost uporabe tlačnega polnjenja z dvema vzporedno priključenima turbopuhalom. Takšen način tlačne polnitve z dvema manjšima turbopuhalom manjše velikosti nakazuje možnost doseganja velikih moči motorja in manjšo porabo goriva v primerjavi z motorjem, ki ga poganja turbina s spremenljivo geometrijsko obliko statorskih

4 CONCLUSION

The results of the presented numerical simulation of steady state and transient characteristics of a sequential turbocharged Diesel engine for heavy road vehicles clearly demonstrates some advantages of this type of turbocharging. Sequential turbocharging, using two equal turbochargers of the same type and size, has evident advantages in high engine power and lower fuel consumption when compared to variable guide vanes geometry

lopatic. Razlike so očitne pri nižjih vrtilnih frekvencah in obremenitvah motorja. Pri prehodnih režimih delovanja motorja (pospeševanje vozila ali vožnja v gostem prometu) se omenjena prednost kaže v nižji porabi goriva.

turbocharging, especially at lower engine speeds and loads. In engine transients, this results in lower fuel consumption to accelerate the vehicle or when driving in heavy traffic conditions.

5 OZNAKE
5 SYMBOLS

čelna površina vozila	A	m^2	vehicle front surface
plinska konstanta	R	$J/kg \cdot K$	gas constant
koeficient aerodinamičnega upora	c_w		drag coefficient
temperatura	T	K	temperature
koeficient upora pri kotaljenju	f_r		rolling friction coefficient
specifična notranja energija	u	J/kg	specific internal energy
pospešek prostega pada	g	m/s^2	acceleration due to gravity
prostornina	V	m^3	volume
specifična entalpija	h	J/kg	specific enthalpy
kot strmine (ceste)	α		road inclination angle
masni vztrajnostni moment	J	$kg \cdot m^2$	mass moment of inertia
kot zavrtitve ročične gredi	φ		crank rotation angle
masa	m	kg	mass
gostota	ρ	kg/m^3	density
vrtilni moment	M	Nm	torque
presežek zraka	λ		air excess ratio
tlak	p	Pa	pressure
kotna hitrost	ω	s^{-1}	angular speed
toplota	Q	J	heat energy
Indeksi:			Indices:
puhalo	c		charger
indeks odvisen o smeri toka	k		index depending on flow direction
dizelski motor	DE		Diesel engine
turbina	T		turbine
indeks opazovane nadzorne prostornine	i		index of the considered volume
turbopuhalo	TC		turbocharger
indeks povezanih prostornin	j		index of connected volumes
vozilo	v		vehicle

6 LITERATURA
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