

Primerjava izvedb 4-valjnega tlačno polnjenega motorja z dvojnimi in enojnim natokom turbine

Comparison of Application of Single and Double Entry Turbine for Pulse System Turbocharging of 4-Cylinder Engine

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Turbina z dvema ali več vtoki se pogosto uporablja v izvedbah turbokompresorjev za tlačno polnjene motorje z notranjim zgorevanjem. Prednosti, ki jih takšen neenakomeren natok turbine prinaša, so povezane z učinkovitejšo izmenjavo delovne snovi, saj uporaba ločenih vtokov v turbino dovoljuje povezavo valjev motorja v podsisteme, v katerih je medsebojni vpliv zaporednih izpuhov ugodnejši. Kolikšne so dejanske prednosti takega sistema je odvisno od izvedbe motorja. V prispevku smo poskusili poiskati prednosti in morebitne pomanjkljivosti, ki jih ima izvedba z dvonatočno turbino pred izvedbo s turbino z enim samim natokom, v primeru 4-valjnega dizel motorja za gospodarska vozila. Pri tem smo uporabili rezultate enodimenzionalnega računalniškega simuliranja, ki smo jih za izvedbo z dvonatočno turbino preverili tudi z rezultati meritev. Ugotovljene razlike med sistemoma so precej manjše od pričakovanih in dopuščajo sklepe, ki so tudi v prid izvedbe z enakomernim natokom turbine.

Ključne besede: motorji z notranjim zgorevanjem, motorji tlačno polnjeni, natok turbine enojni, natok turbine dvojni, primerjave

Double entry turbines are usually used in turbochargers for internal combustion engines. Selected cylinders are linked together in subsystems and the turbine is partially admitted in order to minimize the interaction between exhaust intervals and to increase the efficiency of the gas exchange process. The concrete advantages of this partial admission system depend first of all on the characteristics of a selected engine. A turbocharged 4-cylinder vehicular Diesel engine has been studied, by means of computer simulation, to determine the disadvantages and possible advantages of the application of a single entry turbine instead of the commonly used double entry turbine. The results of the one-dimensional simulation method agree well with experimental results obtained with a double entry turbine system and enable comparison of both systems to be made. The differences between both systems are unexpectedly small, and some conclusions - confirming the advantages of the single entry turbine - may be made.

Keywords: internal combustion engine, turbochargers, single entry turbines, double entry turbines, comparison

0 UVOD

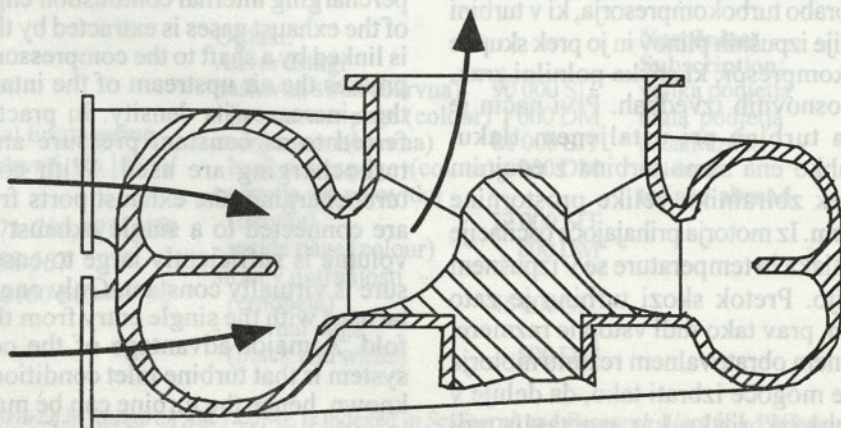
Tlačno polnjenje motorjev z notranjim zgorevanjem z uporabo turbokompresorja, ki v turbini izkorišča del energije izpušnih plinov in jo prek skupne gredi prenaša na kompresor, ki stiska polnilni zrak, se izvaja v dveh osnovnih izvedbah. Prvi način je sistem napajanja turbine pri ustaljenem tlaku. Uporabljena je lahko ena sama turbina z enojnim vtokom, ki je prek zbiralnika velike prostornine povezana z motorjem. Iz motorja prihajajoče oscilacije masnega pretoka, tlaka in temperature se v izpušnem zbiralniku zadržijo. Pretok skozi turbino je zato navidezno ustaljen, prav tako tudi vstopne razmere. Zato so ti pri izbranem obratovalnem režimu motorja znani in turbino je mogoče izbrati tako, da deluje v področju največjega izkoristka, kar zagotavlja tudi ustrezno visoke izkoristke turbokompresorja. To je osnovna prednost sistema napajanja turbine pri ustaljenem tlaku.

0 INTRODUCTION

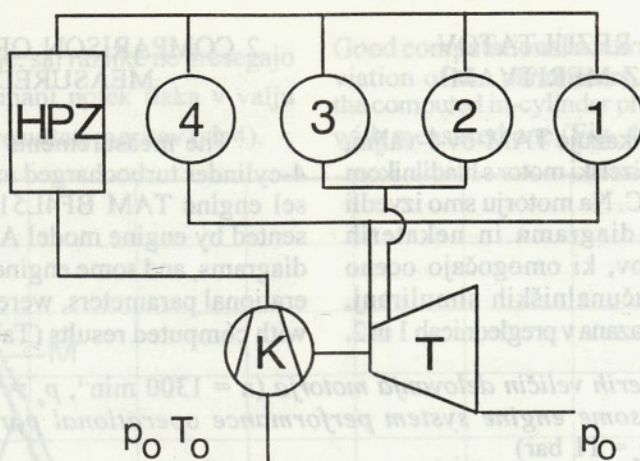
Turbochargers are commonly applied for supercharging internal combustion engines. The power of the exhaust gases is extracted by the turbine, which is linked by a shaft to the compressor. The latter compresses the air upstream of the intake manifold and thus increases its density. In practice, systems referred to as constant pressure and pulse system turbocharging are used. With constant pressure turbocharging, the exhaust ports from all cylinders are connected to a single exhaust manifold whose volume is sufficiently large to ensure that its pressure is virtually constant. Only one turbine needs to be used with the single entry from the exhaust manifold. A major advantage of the constant pressure system is that turbine inlet conditions are steady and known, hence the turbine can be matched to operate at optimum efficiency at specific engine conditions. The main disadvantage is that the available energy entering the turbine is low, since full advantage has not been taken of the pulse energy.

Pomanjkljivost tega sistema pa je slaba izkoriščenost kinetične energije izpušnih plinov, ki se zaradi trenja, ki spremlja mešanje v izpušnem zbiralniku, v veliki meri spremeni v toploto in le manjši del v potencialno energijo. Ugodnejši prenos kinetične energije od motorskih valjev k turbini zagotavlja sunkovit način polnjenja. Izpušne cevi so kratke in ozke, kar zagotavlja, da se ohrani večji del kinetične energije. Izkoristek energije izpušnih plinov je tako praviloma nekoliko višji, kljub večjim izgubam v turbini, ki zaradi izrazito neustaljenega natoka deluje z nižjim izkoristkom. Povezava izpušnih cevi valjev v en sam sistem, ki napaja turbino z enojnim vtokom, lahko zaradi vpliva med valji precej poslabša razmere med izmenjavo delovne snovi. Poveča se delo, potrebno za izmenjavo delovne snovi in pojavi se povratni tok v obdobju prekrivanja ventilov. Zato je treba povezati valje v podsisteme treh ali dveh valjev, ki polnijo vsak svojo turbino ali vsak svoj natok dvonatočne turbine, t. im. dvojček turbine (sl. 1). Dolžina obdobja med izpuhi medsebojno povezanih valjev, je izbrana tako, da se izognemo nezaželenemu prekrivanju izpušnih sunkov nizkega in visokega tlaka in znaša 240° ročične gredi in več [1]. Zaradi tega sta v primeru 4-valjnega tlačno polnjenega motorja z zaporedjem vžiga 1-3-4-2 povezana 1. in 4. ter 2. in 3. valj motorja (sl. 2). Tako je razmik med izpuhoma povezanih valjev 1-4 in 2-3 kar 360° ročične gredi. Natok je zato izrazito neustaljen, kar negativno vpliva na izentropni izkoristek turbine in postavi se vprašanje, ali ima uporaba sunkovitega sistema polnjenja z dvema vtokoma v turbino še vedno prednost pred alternativnim sunkovitim sistemom z enojnim natokom turbine. Na to vprašanje smo poskusili odgovoriti z uporabo računalniškega simuliranja tlačno polnjenega dizel motorja, nekaj rezultatov in končne ugotovitve pa predstavljamo v nadaljevanju tega besedila.

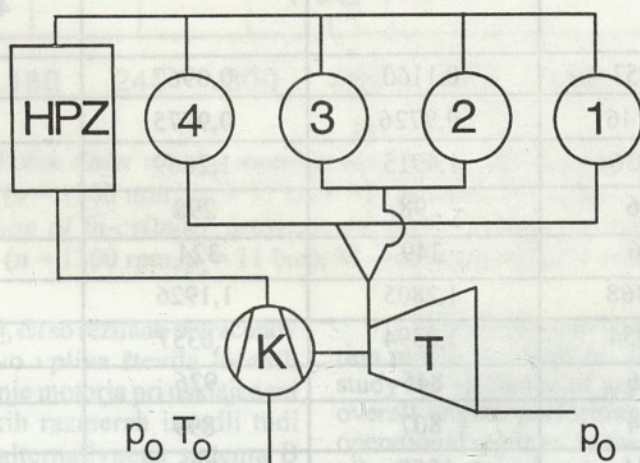
The pulse system, on the other hand, is a system that endeavours to make some use of available pulse energy. The objective is to make the maximum use of the high pressure and temperature which exists in the cylinder when the exhaust valve opens, even at the expense of creating a highly unsteady flow through the turbine. In most cases, the benefit from increasing the available energy will more than offset the loss in turbine efficiency due to unsteady flow. The pulse system requires the exhaust manifold to be as small as reasonably possible. However, it is often disadvantageous to connect all engine cylinders to a single turbine entry. The pressure waves from each cylinder may significantly interfere with the exhaust processes of each cylinder during the valve overlap and exhaust stroke, thereby increasing piston pumping work. This difficulty can be avoided by simply connecting the cylinders, in groups of three or less, to two turbines or to separate entries of a single turbine (Fig. 1). If the correct cylinders are grouped together, then the pressure wave interference may be avoided. The general rule is to connect cylinders whose firing sequences are separated by at least a 240° crank angle (in the case of a four stroke engine) to a turbine inlet [1]. Thus, for a four cylinder engine, shown in figure 2 (firing order 1-3-4-2), pairs of cylinders 1-4 and 2-3 would be connected to a double entry turbine. The phase shift between the individual exhaust into subdivided exhaust systems would be 360° crank angle, and the turbine would operate entirely under partial admission and highly unsteady flow, with considerably lower isentropic efficiency. Thus, the question arises as to whether the application of a double entry turbine is advantageous over the single entry turbine when considering the pulse turbocharging system of a four cylinder engine. To answer this question, the turbocharged, 4-cylinder, 4-stroke, Diesel engine was studied by means of one-dimensional computer simulation. Some of the results are presented in this paper.



Sl.1. Prerez dvonatočne, t. im. dvojček turbine
Fig. 1. Cross-section of double entry turbine (so called twin-turbine)



Sl. 2. Model izvedbe A
Fig. 2. Simulation model A



Sl. 3. Model izvedbe B
Fig. 3. Simulation model B

1 RAČUNALNIŠKO SIMULIRANJE TLAČNO POLNJENEGA MOTORJA

1 COMPUTER SIMULATION OF TURBOCHARGED ENGINE

Uporabili smo računalniški paket za enodimenzionalno simuliranje tokovnih in termodinamičnih procesov v polnilno-izpušnem sistemu in valjih motorja [2]. Model izvedbe tlačno polnjenega motorja z dvonatočno turbino (v nadaljevanju model A) prikazuje slika 2, alternativni model z enojnim vtokom v turbino (v nadaljevanju model B) pa je na sliki 3. Edina razlika med modeloma je na vtočni strani turbine. V modelu A sta natoka ločena, uporabljena je dvojček turbina, ki jo napajata podsistema valjev 1-4 in 2-3. V alternativnem modelu B pa se cevi obeh podsistemov pred turbino združita, uporabljena turbina pa je enonatočna. Simuliranje smo izvedli pri ustaljenih delovnih režimih motorja (stalna hitrost in moment motorja), ki smo ju pri ustrezno izbranih začetnih pogojih in pri dopustnem 0,2% odstopanju vrtiljajev turbokompresorja dveh zaporednih motorjskih procesov (1 krog 4-taktnega motorja je 720° zasuka ročične gredi), dosegli po desetih do petnajstih preračunanih krogih.

A one-dimensional method was used to simulate the flows and thermodynamic processes in intake and exhaust manifolds, and a zero-dimensional, one-zone model was applied in connection with in-cylinder processes [2]. The classic model of the four cylinder engine using double entry turbine (model A) is shown in figure 2, while the alternative one, applying single entry turbine (model B) is presented in figure 3. The only difference between these models is at the inflow side of turbine. The separated exhaust subsystems of cylinder groups 1-4 and 2-3 are joined upstream of the single entry turbine in model B, while both groups are separately connected to a double entry turbine in model A. The simulations were carried out at steady state engine operational regimes (at constant engine speed and torque). Step by step turbocharging matching was used, and approximately 10 to 15 successive cycles (720° of the crank angular displacement) were required, until steady state was reached. Allowance was made for a 0.2% deviation of the turbocharger speed after two successive cycles.

2 PRIMERJAVA REZULTATOV PRERAČUNOV Z MERITVAMI

Model A (slika 2) prikazuje TAM-ov 4-valjni, tlačno polnjeni, prototipni, dizelski motor s hladilnikom polnilnega zraka BF4L515C. Na motorju smo izvedli meritve indikatorskega diagrama in nekaterih osnovnih makroparametrov, ki omogočajo oceno primernosti rezultatov računalniških simuliranj. Primerjava rezultatov je prikazana v preglednicah 1 in 2.

2 COMPARISON OF COMPUTED AND MEASURED RESULTS

The measurements were carried out on the 4-cylinder turbocharged aftercooled prototype Diesel engine TAM BF4L515C schematically represented by engine model A (Fig. 2). The in-cylinder diagrams, and some engine system performance operational parameters, were measured and compared with computed results (Table 1 and 2).

Pregl. 1. *Primerjava nekaterih veličin delovanja motorja ($n = 1300 \text{ min}^{-1}$, $p_e = 11 \text{ bar}$)*

Table 1. *Comparison of some engine system performance operational parameters ($n = 1300 \text{ rpm}$, $p_e = 11 \text{ bar}$)*

veličina parameter	meritev measurement	izvedba A model A	izvedba B model B	izvedba B 40% zmanjšanje A_s model B 40% reduction of A_s
m kg/s	0,1153	0,1160	0,0967	0,1087
p_1 bar	0,9746	0,9726	0,9775	0,9739
p_2 bar	1,5036	1,4915	1,2669	1,4268
T_1 K	296	298	298	298
T_2 K	346	349	324	344
p_3 bar	1,1868	1,2805	1,1926	1,2400
p_4 bar	0,9954	1,0394	1,0357	1,0413
T_3 K	836	845	926	863
T_4 K	794	807	899	829
n_{TK} s^{-1}	1074	1050	844	994

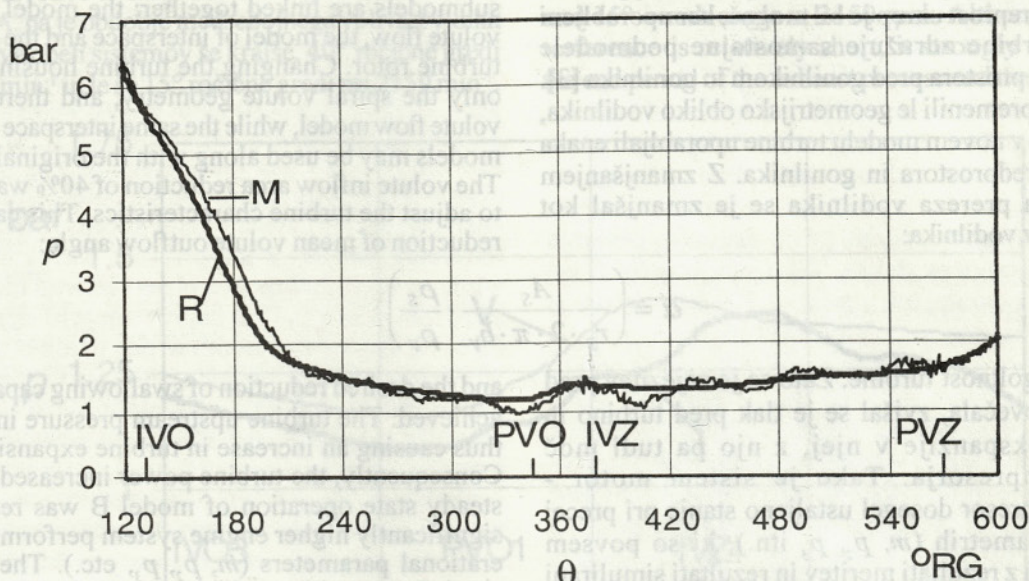
Pregl. 2. *Primerjava nekaterih veličin delovanja motorja ($n = 2150 \text{ min}^{-1}$, $p_e = 9,2 \text{ bar}$)*

Table 2. *Comparison of some engine system performance operational parameters ($n = 2150 \text{ rpm}$, $p_e = 9.2 \text{ bar}$)*

veličina parameter	meritev measurement	izvedba A model A	izvedba B model B	izvedba B 40% zmanjšanje A_s model B 40% reduction of A_s
m kg/s	0,2314	0,2280	0,1845	0,2267
p_1 bar	0,9608	1,0260	0,9642	0,9544
p_2 bar	1,9451	1,9124	1,5344	1,8671
T_1 K	294	296	297	296
T_2 K	379	383	353	380
p_3 bar	1,5477	1,6238	1,4105	1,5949
p_4 bar	1,0015	1,0260	1,0146	1,0242
T_3 K	780	805	878	792
T_4 K	720	732	827	728
n_{TK} s^{-1}	1471	1467	1195	1449

Doseženo je dobro ujemanje, saj razlike ne presegajo 3 odstotkov in tudi izračunani potek tlaka v valju motorja se dobro ujema z rezultati meritev (sl. 4).

Good computational accuracy was achieved. The deviation of the parameters does not exceed 3%, and the computed in-cylinder pressure diagram agrees well with measured one (Fig. 4).



Sl. 4. Potek tlaka v valju motorja med izmenjavo delovne snovi

($n = 1300 \text{ min}^{-1}$, $p_e = 11 \text{ bar}$); M - meritev, R - račun

Fig. 4. Comparison of in-cylinder pressure diagrams during gas exchange process

($n = 1300 \text{ rpm}$, $p_e = 11 \text{ bar}$); M - measurement, R - computation

Sklepamo lahko torej, da so rezultati preračunov dobra osnova za raziskavo vpliva števila ločenih vtokov v turbino na delovanje motorja pri usklajenem režimu. Zato smo v enakih razmerah izvedli tudi simuliranja obratovanja alternativnega sistema B (sl. 3). Rezultati so prikazani v preglednicah 1 in 2. Odstopanja od meritev so neprimerno večja, kar pa je bilo pričakovati. Uporabili smo namreč povsem enako turbino (dimenzije, dva natoka) kakor pri izvedbi A, vendar s to razliko, da je turbina ves čas v polnem natoku, saj se cevna sistema združita že pred njo. Tako so razmere pretoka v spiralnem vodilniku turbine povsem različne. Ker je pretok v izpušnem sistemu 4-valjnega motorja izrazito ciklični, izpuhi v trajanju 250° ročične gredi si sledijo vsakih 180° ročične gredi, požirata ločena natoka turbine v sistemu A izmenjuje vsakih 360° ročične gredi skoraj celoten pretok skozi turbino, v izvedbi B pa se pretok enakomerno porazdeli med oba natoka, zato je v posamezni spirali za polovico manjši. Zaradi tega je zajezitev pretoka v spirali manjša, torej nižji tlak pred turbino. Zato pade stopnja ekspanzije v turbini, z njo moč turbokompresorja in tlak polnjenja motorja p_2 . Sistem motor - turbokompresor pa doseže ustaljeno stanje pri pomembno nižjih parametrih (m , p_2 , p_3 itn.). Primerjava izvedb A in B je v teh razmerah brez pomena, saj je sistem B dejansko opremljen s povsem neprimerno turbino. Ena od možnosti zamenjave turbine je preprosta zamenjava okrova turbine, kar storijo tudi izdelovalci turbin, ki ponujajo enak gonilni par turbokompresorja v različnih okrovi. S tem

Therefore it can be concluded that the simulation model is adequate, and that it may be used to study the influence of separated turbine inflows on overall engine performance at steady state engine operational regimes. In the next step, the calculations were carried out for alternative model B under equal conditions. The results are shown in tables 1 and 2. As expected, the deviations are much bigger than those of engine model A. The same turbine, originally selected for operation under partial admission conditions of model A, was used in both simulation models. When applied to model A, the turbine operates under partial admission conditions. The exhaust duration is 250° crank angle shifted by 180° crank angle for each cylinder. Thus, the exhaust valve overlap is small and each turbine inflow swallows almost the whole mass flow. However, while operating in model B, the mass flow is equally distributed into both turbine inflows. This results in the reduction of turbine flow restriction and lowers the pressure upstream of turbine. The turbine pressure ratio is therefore decreased, causing the reduction of turbine power and consequently the reduction of turbocharger speed and boost pressure p_2 . The steady state operation of the turbocharger-engine system is achieved at significantly lower engine parameters (m , p_2 , p_3 , etc.). It can be concluded that the system B is not equipped with the appropriate turbine, and that the comparison of models A and B is pointless under these circumstances. One of the possibilities for adjusting the turbine to the alternative system B is to change the turbine housing. Using different turbine housings (different volute flow areas) with the same turbine rotor

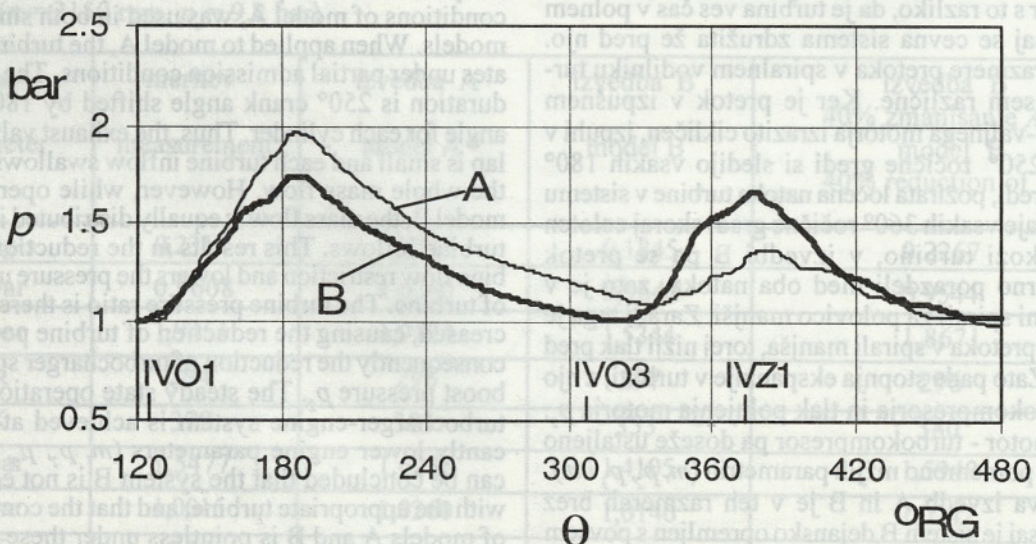
spreminjajo prerez spiralnega vodilnika, rezultat pa je večja ali manjša zajezitev pretoka, t. im. goltnost turbine. Podobno smo postopali tudi v našem primeru. Vstopni prerez spirale vodilnika smo zmanjšali za 40%. Ta preprost ukrep je bil mogoč, ker uporabljeni model turbine združuje samostojne podmodele vodilnika, prostora pred gonilnikom in gonilnika [3]. Ker smo spremenili le geometrijsko obliko vodilnika, smo lahko v novem modelu turbine uporabljali enaka modela predprostora in gonilnika. Z zmanjšanjem vstopnega prereza vodilnika se je zmanjšal kot iztekanja iz vodilnika:

$$\alpha = \left(\frac{A_s}{r_s \cdot 2 \cdot \pi \cdot b_v} \cdot \frac{\rho_s}{\rho_v} \right) \quad (1)$$

in z njim goltnost turbine. Zato se je zajezitev pred turbino povečala, zvišal se je tlak pred turbino in stopnja ekspanzije v njej, z njo pa tudi moč turbokompresorja. Tako je sistem motor - turbokompresor dosegel ustaljeno stanje pri precej višjih parametrih (m , p_2 , p_3 itn.), ki so povsem primerljivi z rezultati meritev in rezultati simuliranj sistema A (preglednici 1 in 2) ter dopuščajo primerjavo izvedb A in B.

3 PRIMERJAVA IZVEDB A IN B

Že prej smo zapisali, da je osnovni razlog za uporabo dvonatočne turbine ločitev zaporednih izpušnih intervalov in zmanjšanje njihovega medsebojnega vpliva. V obravnavanem 4-valjnem motorju trajajo intervali medsebojnega vpliva 70° ročične gredi (sl. 5; $\theta_{IVO3} - \theta_{IVZ1}$). Ker pa se tlačni valovi širijo s hitrostjo zvoka, je njihov vpliv krajši za čas, ki je potreben, da prepotujejo razdaljo med dvema valjema (3 in 1 na sl. 5).



Sl. 5. Primerjava tlaka v izpušnem sistemu motorja za prvim valjem ($n = 1300 \text{ min}^{-1}$, $p_e = 11 \text{ bar}$); A - izvedba A, B - izvedba B

Fig. 5. Comparison of pressure-time histories in exhaust manifold behind the first cylinder ($n = 1300 \text{ rpm}$, $p_e = 11 \text{ bar}$); A - model A, B - model B

is a commonly used design practice to vary turbine swallowing capacity. The same method has been used here. The radial turbine model used in the performed computation is presented in detail in [3]. Three submodels are linked together: the model of spiral volute flow, the model of interspace and the model of turbine rotor. Changing the turbine housing affects only the spiral volute geometry, and therefore the volute flow model, while the same interspace and rotor models may be used along with the original turbine. The volute inflow area reduction of 40% was chosen to adjust the turbine characteristics. This caused the reduction of mean volute outflow angle:

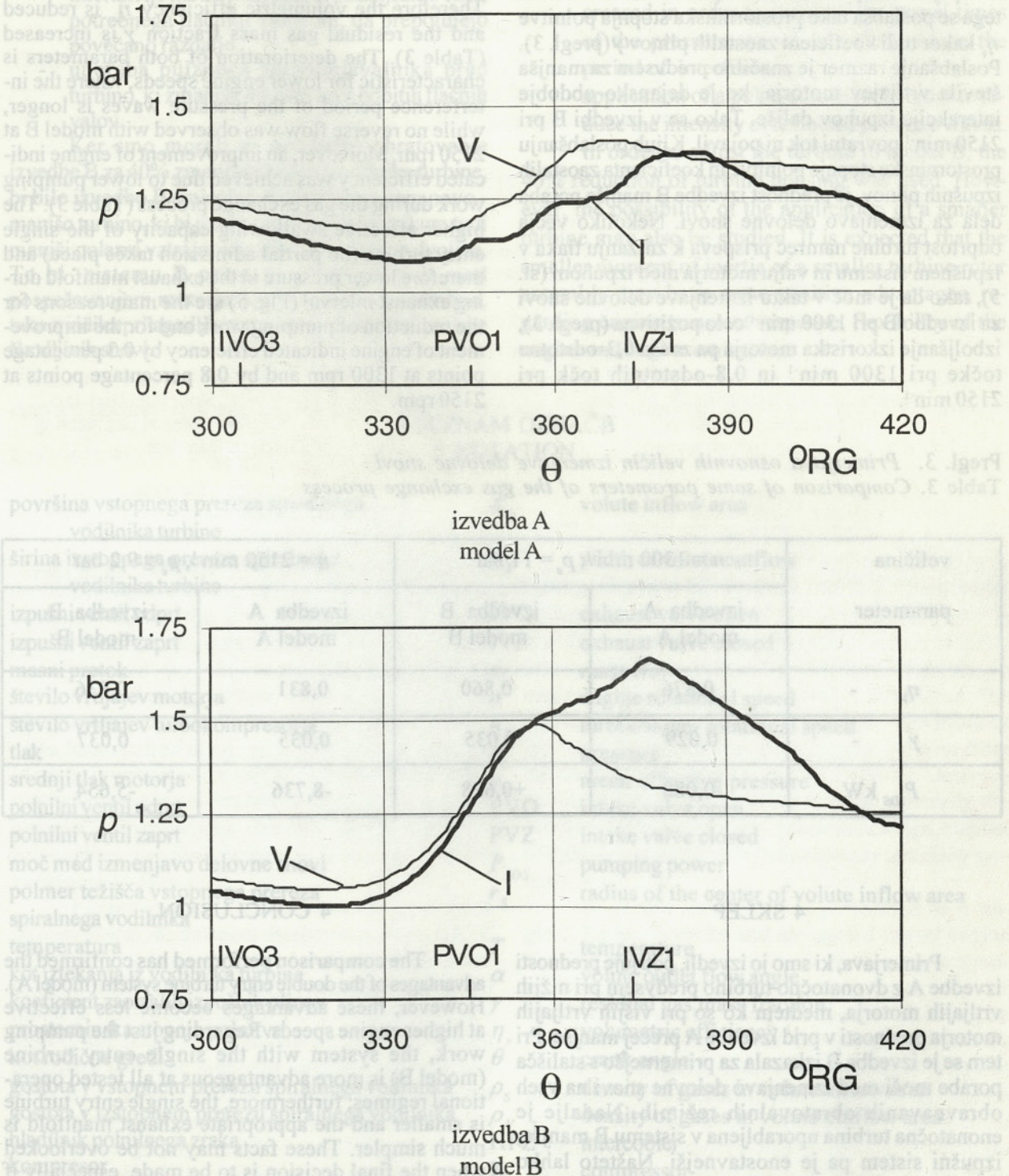
and the desired reduction of swallowing capacity was achieved. The turbine upstream pressure increased, thus causing an increase in turbine expansion ratio. Consequently, the turbine power increased, and the steady state operation of model B was reached at significantly higher engine system performance operational parameters (m , p_2 , p_3 , etc.). These agree much better with model A simulation results and measurements (Tables 1 and 2). The comparison of both models A and B may now be performed.

3 COMPARISON OF MODELS A AND B

As already stated in the introduction, the double entry turbine is used to avoid the interference of pressure waves in the exhaust system during valve overlap. The valve overlap for the engine considered is a 70° crank angle (Fig. 5; $\theta_{EVO3} - \theta_{EVC1}$). However, the actual duration of interference is shorter, as determined by the time necessary for the pressure waves to travel, at the speed of sound, over the distance between the exhaust valves 3 and 1 in figure 5.

Tako je interval interakcije v sistemu B pri 1300 min⁻¹ 50° ročične gredi, pri 2150 min⁻¹ pa le še 30° ročične gredi. Če pa upoštevamo še to, da je tlak v izpušnem sistemu izvedbe B nižji kakor v izpušnem sistemu izvedbe A, pa je obdobje medsebojnega vpliva z vidika primerjave obeh sistemov še krajše, 40° ročične gredi pri 1300 min⁻¹ in le še 15° ročične gredi pri 2150 min⁻¹.

Therefore, the interval of interference is a 50° crank angle at 1300 rpm and 30° crank angle at 2150 rpm, and it is actually even shorter if both pressure traces are compared (Fig. 5): 40° crank angle at 1300 rpm and 15° crank angle at 2150 rpm. Although this interference is relatively short, it strongly influences the end period of the exhaust process and the beginning



Sl. 6. Primerjava tlaka v izpušnem sistemu - I in valju motorja - V; valj 1 ($n = 1300 \text{ min}^{-1}$, $p_e = 11 \text{ bar}$)
Fig. 6. Comparison of in-cylinder pressure-time histories - V and pressure-time histories in exhaust ports - I; cylinder 1 ($n = 1300 \text{ rpm}$, $p_e = 11 \text{ bar}$);

Navkljub razmeroma kratkemu intervalu delovanja ima tlačni val, ki se širi od sosednjega valja močan negativen vpliv na končno fazo izpuha in pričetek polnjenja valja. Zaradi naglega zviševanja tlaka v izpušnem sistemu, se poveča tlak v valju (sl. 6-B) in povzroči povratni tok izpušnih plinov v polnilni kanal, nato pa se pri nadaljnjem zviševanju tlaka v izpušni cevi pojavi tudi povratni tok v izpušnem kanalu. Zaradi tega se poslabša tako prostorninska stopnja polnitve η_v kakor tudi koeficient zaostalih plinov γ (pregl. 3). Poslabšanje razmer je značilno predvsem za manjša števila vrtljajev motorja, ko je dejansko obdobje interakcije izpuhov daljše. Tako se v izvedbi B pri 2150 min⁻¹ povratni tok ni pojavil. Kljub poslabšanju prostorninske stopnje polnitve in koeficienta zaostalih izpušnih plinov, je prednost izvedbe B manjša poraba dela za izmenjavo delovne snovi. Nekoliko večja odprtost turbine namreč prispeva k znižanju tlaka v izpušnem sistemu in valju motorja med izpuhom (sl. 5), tako da je moč v taktu izmenjave delovne snovi za izvedbo B pri 1300 min⁻¹ celo pozitivna (pregl. 3), izboljšanje izkoristka motorja pa znaša 0,3-odstotne točke pri 1300 min⁻¹ in 0,8-odstotnih točk pri 2150 min⁻¹.

of the intake process. Increased pressure in the exhaust manifold raises the in-cylinder pressure (Fig. 6-B). This causes the reverse flow of exhaust gases from the cylinder into the intake manifold at the beginning of the intake process, and as the pressure in the exhaust manifold increases further, the reverse flow from the exhaust manifold into the cylinder is observed at the end of the exhaust period. Therefore the volumetric efficiency η_v is reduced and the residual gas mass fraction γ is increased (Table 3). The deterioration of both parameters is characteristic for lower engine speeds, where the interference period of the pressure waves is longer, while no reverse flow was observed with model B at 2150 rpm. Moreover, an improvement of engine indicated efficiency was achieved due to lower pumping work during the gas exchange process (Table 3). The higher effective swallowing capacity of the single entry turbine (no partial admission takes place) and therefore lower pressure in the exhaust manifold during exhaust interval (Fig. 5) are the main reasons for the reduction of pumping work and for the improvement of engine indicated efficiency by 0.3 percentage points at 1300 rpm and by 0.8 percentage points at 2150 rpm.

Pregl. 3. Primerjava osnovnih veličin izmenjave delovne snovi

Table 3. Comparison of some parameters of the gas exchange process

veličina parameter	$n = 1300 \text{ min}^{-1}, p_e = 11 \text{ bar}$		$n = 2150 \text{ min}^{-1}, p_e = 9,2 \text{ bar}$	
	izvedba A model A	izvedba B model B	izvedba A model A	izvedba B model B
η_v -	0,876	0,860	0,831	0,836
γ -	0,029	0,035	0,035	0,037
P_{IDS} kW	-0,088	+0,628	-8,736	-5,654

4 SKLEP

Primerjava, ki smo jo izvedli, potrjuje prednosti izvedbe A z dvonatočno turbino predvsem pri nižjih vrtljajih motorja, medtem ko so pri višjih vrtljajih motorja prednosti v prid izvedbe A precej manjše. Pri tem se je izvedba B izkazala za primernejšo s stališča porabe moči med izmenjavo delovne snovi na vseh obravnavanih obratovalnih režimih. Nadalje je enonatočna turbina uporabljena v sistemu B manjša, izpušni sistem pa je enostavnejši. Našteto lahko prinese izvedbi B pomembno prednost, še posebej, če z ustreznimi posegi na motorju skrajšamo obdobje interakcije izpušnih tlačnih valov. Kot možne rešitve se ponujajo:

4 CONCLUSION

The comparison performed has confirmed the advantages of the double entry turbine system (model A). However, these advantages become less effective at higher engine speeds. Regarding just the pumping work, the system with the single entry turbine (model B) is more advantageous at all tested operational regimes; furthermore, the single entry turbine is smaller and the appropriate exhaust manifold is much simpler. These facts may not be overlooked when the final decision is to be made, especially if appropriate modifications to the engine exhaust system are made in order to shorten the period of exhaust waves interference. The following modifications may be suggested:

- sprememba krmilnih časov polnilnega in izpušnega ventila, s čimer bi preprečili povratni tok v polnilnem in izpušnem kanalu pri nižjih vrtljajih motorja,
- sprememba geometrijske oblike izpušnega sistema, tj. zvečanje razdalj med valji z zaporednim vžigom in s tem skrajšanje interakcije med njimi, zaradi podaljšanja časa, potrebnega tlačnim valovom, da prepotujejo povečano razdaljo,
- uporaba pulznega spreminjevalnika pred turbino, ki zmanjša intenzivnost odbitih tlačnih valov.

Ker smo morali za normalno obratovanje izvedbe B za 40% zmanjšati prerez vodilnika turbine, bi bilo izvedbo B smiselno testirati s povsem novo - manjšo turbino, ki bi imela z manjšim gonilnikom tudi manjši polarni vztrajnostni moment gonilne dvojice. To bi sistemu B prineslo prednosti tudi med pospeševanjem motorja, kar je še kako pomembno, tako z vidika prilagodljivosti motorja kakor tudi emisije škodljivih snovi.

- the change of the exhaust and inlet valve timings, in order to minimize the reverse flows from the cylinder into the intake manifold and from the exhaust manifold into the cylinder,
- the adjustment of exhaust manifold geometry. The length of the pipes, connecting the cylinders with the successive firing, should be increased in order to increase the travel times of the pressure waves and thus reduce the period of interference,
- application of a pulse-converter in order to reduce the intensity of reflected pressure waves.

In order to adjust the turbine to model B, the 40% reduction of turbine housing was used. However, the possibility of the application of a smaller turbine may also be studied. It is expected that the smaller moment of inertia of a smaller turbine rotor assembly may have some positive advantages, regarding engine transient response, flexibility of the engine and engine emissions.

5 SEZNAM OZNAČB 5 NOTATION

površina vstopnega prereza spiralnega vodilnika turbine	A_s	volute inflow area
širina izstopnega prereza spiralnega vodilnika turbine	b_v	width of volute outflow
izpušni ventil odprt	IVO	exhaust valve open
izpušni ventil zaprt	IVZ	exhaust valve closed
masni pretok	m	mass flow
število vrtljajev motorja	n	engine rotational speed
število vrtljajev turbokompresorja	n_{TK}	turbocharger rotational speed
tlak	p	pressure
srednji tlak motorja	p_e	mean effective pressure
polnilni ventil odprt	PVO	intake valve open
polnilni ventil zaprt	PVZ	intake valve closed
moč med izmenjavo delovne snovi	P_{IDS}	pumping power
polmer težišča vstopnega prereza spiralnega vodilnika	r_s	radius of the center of volute inflow area
temperatura	T	temperature
kot iztekanja iz vodilnika turbine	α	volute outlet flow angle
koeficient zaostalih izpušnih plinov	γ	residual gas mass fraction
prostorninska stopnja polnitve	η_v	volumetric efficiency
kot ročične gredi	θ	crank angle
gostota v vstopnem prerezu spiralnega vodilnika	ρ_s	density of gases in volute inflow area
gostota v izstopnem prerezu spiralnega vodilnika	ρ_v	density of gases in volute outflow area
hladilnik polnilnega zraka	HPZ	intercooler
kompresor	K	compressor
turbina	T	turbine
ročična gred	RG	crank angle

INDEKSI
SUFFIXES

stanje okolice	0	ambient conditions
pred kompresorjem	1	upstream of compressor
za kompresorjem	2	downstream of compressor
pred turbino	3	upstream of turbine
za turbino	4	downstream of turbine

6 LITERATURA
6 REFERENCES

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parameter	exhaust valve closed	exhaust valve open	mean effective pressure	turbocharger rotational speed	engine rotational speed	mass flow	exhaust valve closed	exhaust valve open
mean effective pressure	828,0	828,0	828,0	530,0	530,0	0,0620	0,0620	0,0620
turbocharger rotational speed	530,0	530,0	530,0	530,0	530,0	0,0620	0,0620	0,0620
engine rotational speed	530,0	530,0	530,0	530,0	530,0	0,0620	0,0620	0,0620
mass flow	0,0620	0,0620	0,0620	0,0620	0,0620	0,0620	0,0620	0,0620
exhaust valve closed	PVZ	PVO	828,0	530,0	530,0	0,0620	0,0620	0,0620
exhaust valve open	PVO	PVZ	828,0	530,0	530,0	0,0620	0,0620	0,0620
mean effective pressure	828,0	828,0	828,0	530,0	530,0	0,0620	0,0620	0,0620
turbocharger rotational speed	530,0	530,0	530,0	530,0	530,0	0,0620	0,0620	0,0620
engine rotational speed	530,0	530,0	530,0	530,0	530,0	0,0620	0,0620	0,0620
mass flow	0,0620	0,0620	0,0620	0,0620	0,0620	0,0620	0,0620	0,0620

Primerjava, ki jo izvaja nič- in eno-dimenzionalni model, kaže prednosti izvedbe A z dvonozastno turbino pred izvedbo B z enonozastno turbino pri nižjih vrtiljajih motorja, medtem ko pri višjih vrtiljajih motorja prednosti v prid izvedbi B. Pri tem se je izvedba B izkazala za bolj primerno zaradi manjših izgub zaradi uporabe enonozaste turbine. V enonozasti turbini je enostavnejša konstrukcija izpušnega sistema, kar pomeni manjšo težo in stroške izdelave. Pomembna prednost izvedbe B je tudi možnost posega na motorju, kar omogoča prilagoditev izpušnih tlačnih valov in zmanjšanje njihovega vpliva na delovanje motorja.

The comparison performed has confirmed the advantages of the double nozzle turbine system. However, these advantages are more pronounced at higher engine speeds. In this respect, the system with the single nozzle turbine (model B) is more appropriate. The construction of the exhaust system is simpler and the cost of manufacturing is smaller. These facts may be taken into account when the final decision is to be made. Appropriate modifications to the engine exhaust system are made in order to shorten the exhaust waves interference. The following modifications may be suggested: