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Primerjava brez- in enodimenzionalnih metod za simuliranje procesov v tlačno polnjenem dieselskem motorju

Comparison of Zero- and One-Dimensional Methods for Simulation of the Process in Turbocharged Diesel Engines

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V prispevku obravnavamo brezdimenzionalno metodo »polnjenje — praznjenje« in reševanje enodimenzionalnega pretoka v ceveh polnilnega in izpušnega sistema z dvokoračno metodo Lax-Wendroff. Metodi primerjamo z vidika porabe računskega časa in natančnosti rezultatov. Obravnavamo tlačno polnjeni dizelski motor in eksperimentalne rezultate (povprečne temperature in tlake v polnilnem in izpušnem sistemu, masni pretok, potek tlaka za šestim valjem) primerjamo z rezultati računalniških simuliranj. Primernost posamezne metode nato ocenimo na podlagi dosežene natančnosti in porabljenega računskega časa.

The zero-dimensional method »filling-emptying« and one-dimensional two-step Lax-Wendroff method are discussed in the paper. The two methods are compared with respect to accuracy of results and computation time. A turbo-charged Diesel engine was simulated using both methods. Experimental and computational results (mass flow, mean temperatures and pressures in intake and exhaust manifold, pressure-time histories behind the 6^{th} cylinder) were compared and the suitability of each simulation method was estimated.

0 UVOD

Računalniško simuliranje procesov v motorju z notranjim zgorevanjem lahko v veliki meri dopolni eksperimentalne raziskave. Predvsem s podatki, ki jih ne moremo ali jih le težko izmerimo, hkrati pa jasno pokaže šibka mesta v poznavanju posameznih procesov in s tem potrebo po podrobnejših primarnih raziskavah. Dandanes so za simuliranje termodinamičnih in tokovnih procesov v motorjih z notranjim zgorevanjem največkrat uporabljene brez- in enodimenzionalne metode. Kompleksnejše dvo- in trodimenzionalne metode so v rabi le za podrobnejše raziskave v posameznih delih sistema (cevni spoji, motorski ventili, polnilni in izpušni kanali itn.). Izbira najustreznejše simulirne metode v konkretnem primeru je odvisna od več dejavnikov. Na končno odločitev najpomembneje vplivata natančnost metode in poraba računalniškega časa. V nadaljevanju se bomo omejili na prikaz in primerjavo brez- in enodimenzionalne metode.

1 BREZDIMENZIONALNA METODA »POLNJENJE — PRAZNJENJE«

Makroskopsko gledano lahko tok plinov v motorju z notranjim zgorevanjem ponazorimo s pretokom plinov med opazovanimi prostorninami. To so polnilni in izpušni kolektorji motorja konstantne prostornine ter valji motorja spremenljivih delovnih prostornin (sl. 1). Uvedba teh

0 INTRODUCTION

Using simulation models, it is possible to analyse complex processes, and to obtain information which is difficult or even impossible to obtain experimentally. Furthermore, the use of process simulation for scientific investigation often highlights areas in which more experimental research is needed because of lack of fundamental knowledge. For engineering computations, zero- and one-dimensional models are still most frequently used in connection with simulation of an internal combustion engine thermodynamic cycle. However, more complex two- and three-dimensional methods prevail in detailed research on specific parts of a system (pipe junctions, valve ports, pulse converters, etc.). The selection of the most appropriate simulation method depends on several factors. The final decision depends first of all of the simulation method accuracy and computational time. Zero- and one-dimensional methods will be discussed in this paper.

1 ZERO-DIMENSIONAL »FILLING — EMPTYING« METHOD

From the macroscopic point of view, the gas flow in an internal combustion (IC) engine can be observed as a flow between control volumes, representing intake and exhaust manifold and engine cylinders (Fig. 1). The introduction of control volumes is the basic characteristic of prostornin je osnovna značilnost metode »polnjenje — praznjenje« [1]. Stanje plina v opazovani prostornini je v splošnem določeno z energijsko enačbo:

»filling–emptying« method [1]. The instantaneous thermodynamic state of a gas in the control volume is generally defined by energy:

$$\frac{dU}{d\theta}=\frac{dQ}{d\theta}-p\frac{dV}{d\theta}+\sum_i\left(h_{in,i}\frac{dm_{in,i}}{d\theta}\right)-\sum_j\left(h\frac{dm_{out,j}}{d\theta}\right) \tag{1}$$

in kontinuitetno enačbo:

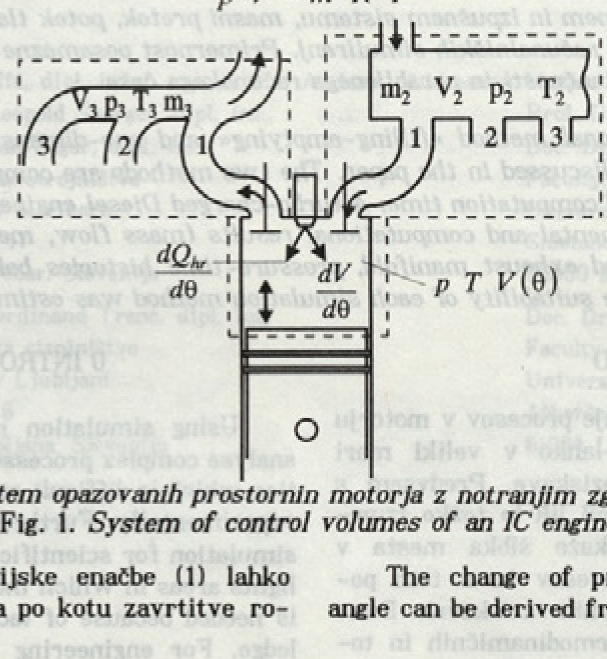
and mass conservation equation:

$$\frac{dm}{d\theta}=\sum_i\frac{dm_{in,i}}{d\theta}-\sum_j\frac{dm_{out,j}}{d\theta}+\frac{dm_{f,b}}{d\theta} \tag{2}$$

ter enačbo stanja:

and by equation of state:

$$p\cdot V=m\cdot R\cdot T \tag{3}$$



Sl. 1. *Sistem opazovanih prostornin motorja z notranjim zgorevanjem*

Fig. 1. *System of control volumes of an IC engine*

Po preureditvi energijske enačbe (1) lahko zapišemo spremembo tlaka po kotu zavrtitve ročične gredi:

The change of pressure during one crank angle can be derived from the energy equation (1):

$$\frac{dp}{d\theta}=\frac{\frac{dQ_{hr}}{d\theta}-\frac{dQ_{ht}}{d\theta}-p\frac{dV}{d\theta}-u\frac{dm}{d\theta}+\sum_i\left(h_{in,i}\frac{dm_{in,i}}{d\theta}\right)-\sum_j\left(h\frac{dm_{out,j}}{d\theta}\right)-\frac{\partial u}{\partial T}\,\frac{p\frac{dV}{d\theta}-TR\frac{dm}{d\theta}-mT\frac{\partial R}{\partial\lambda}\frac{d\lambda}{d\theta}}{R+\frac{\partial R}{\partial T}T}-m\frac{\partial u}{\partial\lambda}\frac{d\lambda}{d\theta}}{\frac{\partial u}{\partial T}\,\frac{V-mT\frac{\partial R}{\partial p}}{R+\frac{\partial R}{\partial T}T}+\frac{\partial u}{\partial p}m} \tag{4}$$

Izraz (4) je splošna enačba brezdimenzionalnega enoconskega modela za spremembo tlaka v opazovni prostornini. Upošteva tudi spremembo prostornine in proces zgorevanja. Zato jo v tej obliki uporabljamo le za opis stanja v valjih motorja, v primeru kolektorja pa upoštevamo:

Equation (4) is the general equation of the pressure change in a control volume by the zero–dimensional one–zone model approach. Both the combustion process and the volume change are taken into account. This general form is therefore used only in connection with in–cylinder processes, while the following simplifications are introduced with a simple collector:

$$\frac{dV}{d\theta}=0 \quad and \quad \frac{dQ_{kr}}{d\theta}=0 \tag{5}$$

Da bi določili stanje v posameznih opazovnih prostorninah motorja, moramo zapisati in rešiti sistem enačb celotnega sklopa opazovanih prostornin. Pri tem sta diferencialni enačbi vsake posamezne opazovane prostornine v splošnem podani z izrazoma (2) in (4). Dvojice enačb (2) in (4) v sistem navadnih nelinearnih diferencialnih enačb 1. reda povežejo robni pogoji. Ti določajo pretok mase in energije med posameznimi opazovanimi prostorninami.

Equations of the whole system of control volumes (Fig. 1) have to be solved, to determine the instantaneous thermodynamic state in a particular control volume. Differential equations of each control volume are generally given by equations (2) and (4). These are connected into a system of ordinary, non-linear differential equations of 1st order by boundary conditions, determining the flow of mass and energy between control volumes.

1.1 Robni pogoji

Robni pogoji določajo pretok mase in energije skozi priključke opazovanih prostornin. Glede na lego opazovane prostornine v sistemu lahko robni pogoji določajo vtok ali iztok v okolico ali v pripojeno opazovano prostornino. Ta pretok mase in energije je zapisan analogno pretoku v konvergentni šobi. Za primer s slike 2 torej velja:

1.1 Boundary conditions

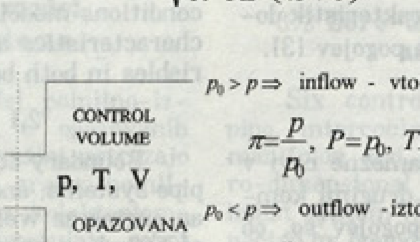
The flow of mass and energy into and out of a control volume are determined by the boundary conditions. Depending on the position of the control volume in the system, the inflow or outflow conditions are influenced by the ambient conditions or by the thermodynamic state in the adjoining control volume. The mass flow is defined by the analogy of convergent nozzle flow. For the case in the Fig. 2 they are:

— podzvočni pretok $\pi \geq \left(\frac{2\kappa}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}$: — subsonic flow

$$m = A_g \mu P \sqrt{\frac{2\kappa}{\kappa-1}\frac{1}{R\cdot TE}\left[\pi^{\frac{2}{\kappa}}-\pi^{\frac{\kappa+1}{\kappa-1}}\right]} \tag{6}$$

— dušeni pretok $\pi < \left(\frac{2\kappa}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}$: — choked flow

$$m = A_g \mu P \sqrt{\frac{\kappa}{R\cdot TE}\left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}} \tag{7}$$



Sl. 2. *Robni pogoji za metodo »polnjenje — praznjenje«*
Fig. 2. *Boundary conditions for »filling — emptying« method*

Geometrijski prerez A_g je lahko konstanten ali pa se spreminja v odvisnosti od obremenitve (uplinjač) ali zavrtitve ročične gredi (ventilski prerez). Podobne ugotovitve veljajo tudi za pretočno število μ. Po določitvi masnega pretoka določimo še energijski tok. Ta je enak produktu masnega pretoka in specifične entalpije plina v prostoru, iz katerega izteka plin.

The geometric flow area A_g is constant or it changes with engine load (carburettor) or crank angle (valve). The same statement can be used also for the discharge coefficient μ. Once the mass flow is computed, the energy flow is determined. It is equal to the product of mass flow and specific enthalpy in the upstream control volume.

2 ENODIMENZIONALNA METODA

Poglavitna pomanjkljivost brezdimenzionalnih metod je prostorsko povprečenje termodinamičnega stanja v posameznih opazovanih prostorninah. Tako ni mogoče spremljati lokalnih sprememb v sistemu, ki pomembno vplivajo na procese pretoka mase in energije. To pomanjkljivost v veliki meri odpravijo enodimenzionalne metode. Z njimi popišemo navidezni enodimenzionalni, neustaljeni, neizentropni tok stisljivega fluida s sistemom treh parcialnih diferencialnih enačb hiperboličnega tipa, ki ga sestavljajo [2]:

- kontinuitetna enačba:

$$\frac{\partial \rho}{\partial t}+\rho\frac{\partial u}{\partial x}+u\frac{\partial \rho}{\partial x}+\frac{\rho u}{A}\frac{dA}{dx}=0 \qquad (8),$$

— enačba ohranitve gibalne količine:

$$\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+\frac{1}{\rho}\frac{\partial p}{\partial x}+G=0 \qquad (9),$$

pri čemer člen G pomeni vpliv trenja in je za okroglo cev: $G = 2\,f\,u\,|u|\,D^{-1}$

— energijska enačba:

$$\dot{q}\,\rho A dx=\frac{\partial}{\partial t}\left[(\rho A dx)\left(c_v T+\frac{u^2}{2}\right)\right]+\frac{\partial}{\partial x}\left[(\rho u A)\left(c_v T+\frac{u^2}{2}+\frac{p}{\rho}\right)\right]dx \qquad (10).$$

Člen q je specifični toplotni tok, ki se s konvekcijo prenaša s stene cevi na fluid.

Sistem enačb rešujemo numerično. Pri tem uporabljamo kombinacijo metode karakteristik in dvokoračne metode Lax-Wendroff [2]. Cev razdelimo v želeno število enakih odsekov. Z metodo Lax-Wendroff določimo vrednosti v notranjih točkah cevi, robne pogoje, tj. vrednosti v začetni in končni točki pa z uporabo metode karakteristik določimo z ustreznim modelom robnih pogojev [3].

2.1 **Robni pogoji**

Z robnimi pogoji povežemo posamezne cevi v cevni sistem, sistem pa z okolico in drugimi komponentami motorja. Enačbe robnih pogojev so, ob predpostavki navidezne ustaljenosti procesov, izpeljane iz izrazov za ohranitev mase, energije in gibalne količine na robu cevi. Pri tem je povezava robne in notranje točke cevi izvedena z metodo karakteristik. Enačbe robnega pogoja sestavljajo sistem nelinearnih algebrajskih enačb, t.im. matematično fizikalni model robnega pogoja, ki ga je treba rešiti v vsakem računskem koraku. R.Benson [3] je za metodo karakteristik razvil vrsto uspešnih matematično fizikalnih modelov robnih pogojev, ki smo jih uporabili v računalniškem paketu. Dodali smo še modele za popis robnih pogojev na priključkih dvonatočne turbine in kompresorja [4].

2 ONE-DIMENSIONAL METHOD

The main deficiency of the zero-dimensional method is a spatial average of the thermodynamic state in the control volume. The local conditions, influencing the mass and energy transfer significantly, can thus not be adequately observed. Introducing one-dimensional methods, this problem can be successfully overcome. One-dimensional, non-steady, non-homentropic, compressible fluid flow is defined by three partial differential equations of hyperbolic type [2]:

- continuity equation:

- momentum equation:

where G represents the influence of friction, which, for a circular pipe is: $G = 2\,f\,u\,|u|\,D^{-1}$

- energy equation:

Here q is the rate of heat transfer between pipe and fluid per unit time per unit mass.

This system of three partial differential equations is solved numerically, using a combination of the method of characteristics and the two-step method Lax-Wendroff [2]. Each pipe is subdivided into a number of meshes. The Lax-Wendroff method is used to determine the values of variables in inner mesh points, and boundary conditions models [3] incorporating the method of characteristics are introduced to evaluate the variables in both boundary mesh points.

2.1 **Boundary conditions**

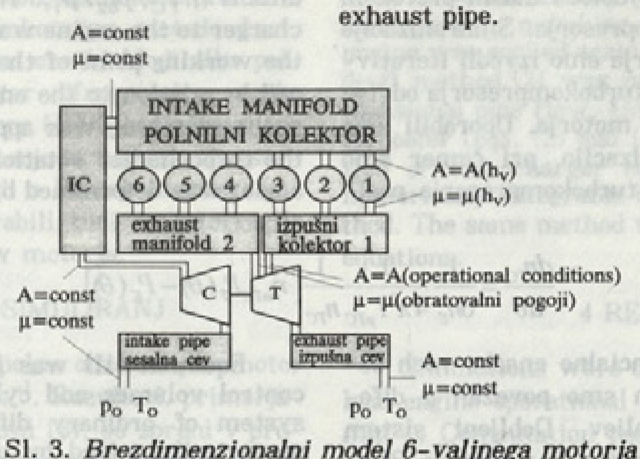
Boundary conditions connect single pipes into pipe systems, and these with other components of an engine as well as with the ambient. Equations of boundary conditions are derived from conservation equations of mass, momentum and energy under quasi-steady assumption at the boundary. The boundary mesh point is connected with the inner mesh point by the method of characteristics. Equations of particular boundary conditions are generally constituted by a system of non-linear algebraic equations, the so-called »mathematical model of boundary conditions«, which has to be solved in each computational time step. Several successful mathematical models of boundary conditions, developed by Benson [3], have been incorporated in our computer program and our own model was used to simulate processes within the radial twin turbine [4].

3 RAČUNALNIŠKA SIMULIRANJA

Obravnavali smo tlačno polnjeni 6-valjni dieselski motor. Motor je shematsko prikazan na sl. 3. Polnilni sistem sestavljajo sesalna cev, kompresor, hladilnik polnilnega zraka, polnilni kolektor in cevni priključki valjev motorja. Izpušni sistem sestavljata dva ločena cevna sistema, ki sta spojena z dvonatočno turbino, iz katere iztekajo plini v izpušno cev.

3 COMPUTER SIMULATIONS

A six-cylinder, turbocharged, vehicular Diesel engine was simulated. The engine is schematically shown in Fig. 3. The intake system is constituted by the intake pipe, compressor intercooler, intake manifold and intake valve ports. There are two separate exhaust manifolds, connected by a twin-turbine (pulse system turbocharging) and exhaust pipe.



Sl. 3. *Brezdimenzionalni model 6-valjnega motorja*

Fig. 3. *Zero-dimensional model of a 6-cylinder engine*

3.1 **Model termodinamičnih procesov v valju**

Procese v valjih motorja smo simulirali z enoconskim brezdimenzionalnim modelom, tako da smo z metodo Runge-Kutta 4 numerično integrirali kontinuitetno (2) in energijsko enačbo (4) delovnega prostora. Sprostitev toplote smo simuliralis funkcijo Vibe [5]. Oba parametra funkcije Vibe, trajanje in intenzivnost zgorevanja, pa smo izbrali z računalniško obdelavo posnetih indikatorskih diagramov.

3.2 **Brezdimenzionalni model polnilno-izpušnega sistema**

Brezdimenzionalni (OD) model polnilno-izpušnega sistema (sl.3) sestavlja 6 opazovanih prostornin. Te po prostornini in površini ustrezajo sesalni cevi, hladilniku polnilnega zraka, polnilnemu kolektorju, obema izpušnima kolektorjema in izpušni cevi. Med seboj jih povezujejo robni pogoji. Z izjemo kompresorja so robni pogoji zasnovani na analogiji konvergentne šobe (poglavje 1.1). Pri tem so pretočni prerezi in pretočna števila na vstopu v sesalno cev in polnilni kolektor konstantni. Enako velja tudi za iztok iz izpušne cevi. Iztočni prerezi polnilnega kolektorja in vtočni prerezi izpušnih kolektorjev (povezava z valji) so spremenljivi in odvisni od dviga ventila, kar velja tudi za pretočna števila ventilov, ki so izmerjena. Oba iztočna prereza izpušnih kolektorjev sta prav tako spremenljiva in določena s pretočnimi števili dvonatočne turbine, ki so odvisna od trenutnih obratovalnih pogojev turbine (n_{TC}, T_{031}, p_{031}, T_{032}, p_{032}, p_4).

3.1 **Model of thermodynamic processes in cylinder**

Thermodynamic processes in the cylinder were simulated by using a one-zone, zero-dimensional model. The two conservation equations, continuity (2) and energy (4), were numerically integrated by means of the Runge-Kutta 4 method. The rate of heat release was modelled via a simple Vibe function [5], where parameters of the Vibe function, intensity and duration of combustion, were numerically obtained, using measured in-cylinder pressure diagrams.

3.2 **Zero-dimensional model of intake and exhaust system**

Six control volumes, representing suction pipe, intercooler, intake manifold, both exhaust manifolds and exhaust pipe, constitute the zero-dimensional model (OD) of the intake and exhaust system (Fig. 3). Control volumes are connected by boundary conditions. All of these, except the compressor, are based on the convergent nozzle analogy (see 1.1). Geometrical flow area and discharge coefficients of suction pipe and intake manifold inflow boundary are constant, while variable values are used with other boundaries. The geometrical valve flow area (intake manifold outflow boundaries and exhaust manifold inflow boundaries) varies with the crank angle, while the valve discharge coefficient depends on the valve lift. This dependence was obtained experimentally. Turbine discharge coefficients (both exhaust manifolds outflow boundaries) are a function of instantaneous turbine operation conditions (n_{TC}, T_{031}, p_{031}, T_{032}, p_{032}, p_4) and were computed

Odvisnost pretočnih števil in izkoristka turbine od pogojev obratovanja smo določili z modelom dvonatočne turbine [4] in konvencionalnih karakteristik turbine. Robnih pogojev kompresorja zaradi toka fluida od nižjega k višjemu tlaku ne moremo popisati z analogijo konvergentne šobe. Zato smo na podlagi kompresorskih karakteristik izdelali preprost model kompresorja, ki za trenutne obratovalne razmere (n_{TC}, p_{02}/p_{01})določa masni pretok in izentropski izkoristek kompresorja. Sinhronizacijo motorja in turbokompresorja smo izvedli iterativno, tako da je obratovanje turbokompresorja odvisno od razmer obratovanja motorja. Uporabili smo t.im. neustaljeno sinhronizacijo, pri čemer smo dinamiko gonilnie dvojice turbokompresorja popisali z izrazom:

using a radial twin-turbine flow model [4], based on conventional turbine characteristics. A convergent nozzle analogy cannot be applied for compressor boundary conditions, due to characteristic fluid flow from low to high pressure. A simple compressor model based on compressor map was thus introduced to calculate mass flow and isentropic efficiency for instantaneous operation conditions (n_{TC}, p_{02}/p_1). The matching of the turbocharger to the engine was done iteratively, so that the working point of the turbocharger was obtained in relation to the engine conditions. Instantaneous matching was applied and the variation of the turbocharger rotational speed with the crank angle, was determined by the equation:

$$\frac{dn_{TC}}{d\theta}=\frac{1}{6n_E}\,\frac{1}{4\pi^2 I_{pTC}}\,\frac{1}{n_{TC}}\left[\eta_{mTC}P_T(\theta)-P_K(\theta)\right] \qquad (11).$$

Enačbo (11) in diferencialne enačbe vseh šestih opazovnih prostornin smo povezali z diferencialnimi enačbami valjev. Dobljeni sistem enačb pa smo numerično reševali z metodo Runge-Kutta 4.

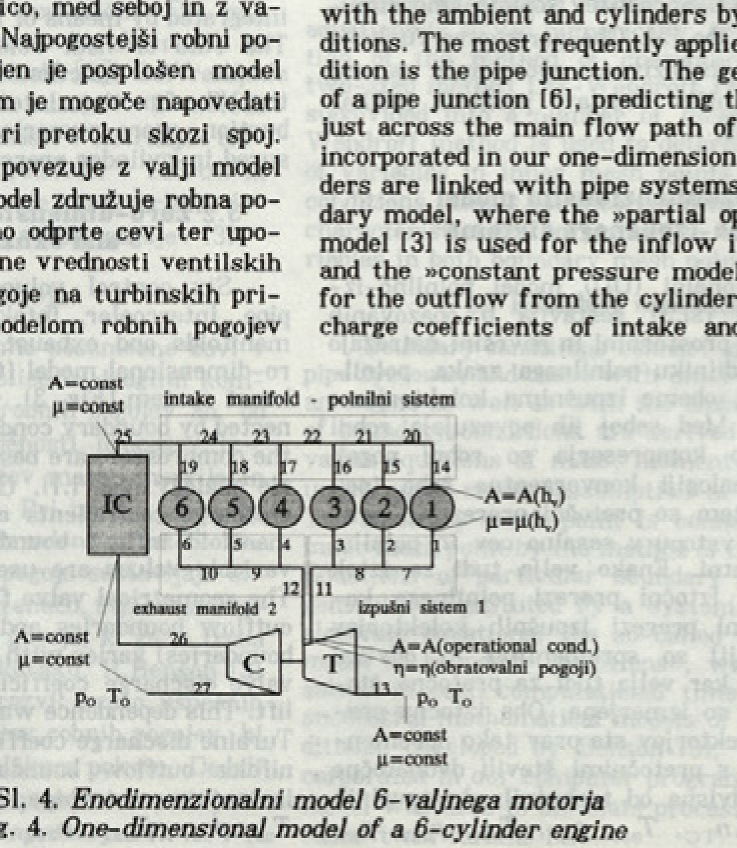
Equation (11) was linked with equations of control volumes and cylinders together and this system of ordinary differential equations was numerically solved by using the Runge-Kutta 4 method.

3.3 Enodimenzionalni model polnilno izpušnega sistema

Enodimenzionalni (1D) model polnilno izpušnega sistema prikazuje slika 4. Sestavlja ga 27 cevi in kolektor, ki jih z okolico, med seboj in z valji, povezujejo robni pogoji. Najpogostejši robni pogoj je cevni spoj. Uporabljen je posplošen model cevnega spoja [6], s katerim je mogoče napovedati padec tlaka zaradi izgub pri pretoku skozi spoj. Polnilni in izpušni sistem povezuje z valji model pretoka skozi ventil [3]. Model združuje robna pogoja vtoka in iztoka iz delno odprte cevi ter uporablja izmerjene ekvivalentne vrednosti ventilskih pretočnih števil. Robne pogoje na turbinskih priključkih smo določili z modelom robnih pogojev

3.3 One-dimensional model of intake and exhaust system

A one-dimensional (1D) model of the intake and exhaust system is schematically shown in Fig. 4. It is composed of 27 pipes and of one collector, linked in an appropriate system and connected with the ambient and cylinders by boundary conditions. The most frequently applied boundary condition is the pipe junction. The generalised model of a pipe junction [6], predicting the pressure loss just across the main flow path of the junction, is incorporated in our one-dimensional model. Cylinders are linked with pipe systems by valve boundary model, where the »partial opened end« flow model [3] is used for the inflow into the cylinder and the »constant pressure model« [3] is applied for the outflow from the cylinder. Measured discharge coefficients of intake and exhaust ports



Sl. 4. *Enodimenzionalni model 6-valjnega motorja*
Fig. 4. *One-dimensional model of a 6-cylinder engine*

dvonatočne turbine [7], v katerem uporabimo spremenljiva pretočna števila in izentropski izkoristek turbine, ki jih poprej izračunamo z modelom pretoka v dvonatočni turbini [4]. Model kompresorja smo povzeli po Bensonu [3] in temelji na posnetih karakteristikah kompresorja. Hladilnik polnilnega zraka je obravnavan kar kot opazovana prostornina, tako da je stanje v njem popisano z diferencialnima enačbama (2) in (4). Enodimenzionalni pretok v ceveh smo reševali z dvokoračno metodo Lax—Wendroff [2]. Diferencialni enačbi hladilnika polnilnega zraka (2) in (4) ter diferencialno enačbo dinamike gonilniškega sklopa (11) (neustaljena iterativna sinhronizacija motorja in turbokompresorja) smo numerično reševali z Eulerjevo integracijo. Isto metodo smo uporabili tudi za integracijo diferencialnih enačb valjev motorja.

4 REZULTATI SIMULIRANJ

Simuliranja smo za polno obremenjen motor izvedli pri 1100 in 2150 min^{-1}. Rezultati, primerjani z izmerjenimi vrednostmi [8], so zbrani v preglednicah 1 in 2. V obeh primerih opazimo nesorazmerno veliko razliko med izmerjenim in napovedanim številom vrtljajev turbokompresorja, ki močno presega druge odstopke. Delno ju zato lahko pripišemo pogrešku meritve. To trditev potrjuje tudi dejstvo, da so vrtljaji turbokompresorja (n^*_{TC}), ki slede za izmerjene parametre pretoka v kompresorju iz kompresorskih karakteristik, prav tako znatno višji od izmerjenih. Pri manjših vrtljajih motorja (1100 min^{-1}) je ujemanje povprečnih parametrov izmenjave delovne snovi razmeroma dobro. Med rezultati modela 0D izraziteje odstopa predvsem masni pretok, ki je kljub nižjemu polnilnemu tlaku p_2 tudi do 5 odstotkov (preglednica 1) večji od izmerjenega. Odstopek je posledica tlačnih izgub v polnilnem sistemu, ki jih model 0D ne upošteva.

were again used. Gas turbine boundary conditions are determined by a »twin-nozzle« model [7], where by instantaneous turbine discharge coefficients and isentropic efficiency, computed by means of radial twin-turbine flow model [4] are applied. The compressor boundary conditions model is based on a compressor map and it is adopted from Benson [3]. The intercooler is simulated by a control volume analogy, so the thermodynamic state is determined by equations (2) and (4). The instantaneous matching of the turbocharger to the engine was applied again. The two-step Lax-Wendroff method [2] was used in relation to one-dimensional pipe flow. Differential equations of intercooler (eqs. (2) and (4)) and differential equation of turbocharger rotational speed (11) were numerically integrated by means of Euler's method. The same method was used to solve cylinder equations.

4 RESULTS

Simulations were carried out under two full load engine operational regimes at 1100 and 2150 min^{-1}. Computation results (mean values) compared with measured data [8] are given in Tables 1 and 2. Higher deviations of computed turbocharger rotational speed can be observed in both examples but they may originate from a measurement error. Considerably higher turbocharger rotational speeds (n^*_{TC}), were obtained from the compressor map. Satisfactory agreement is shown between the experiment and the calculation at the lower engine speed (1100 min^{-1}). However, the mass flow rate is markedy overestimated by the 0D model. The difference in p_2 is as high as +5% (Tab. 1) in spite of lower charge pressure, and it is probably caused by neglecting pressure losses within the intake system. The results obtained with the 1D model are much better, because the friction and junction losses are taken into account.

Preglednica 1: *Primerjava izračunanih in izmerjenih povprečnih parametrov izmenjave delovne snovi* (n_E = 1100 min^{-1})

Table 1: *Comparison of predicted and measured mean parameters of a gas exchange process* (n_E = 1100 min^{-1})

T 1	meritev measured	0D	Δ %	1D	Δ%
p_1 bar	0,9816	0,9827	+0,1	0,9813	-0,0
p_2 bar	1,3853	1,3771	-0,6	1,4084	+1,7
T_1 K	302,0	300,6	-0,5	300,6	-0,5
T_2 K	346,0	340,0	-1,7	343,8	-0,6
T_{ic} K	300,0	300,2	+0,1	300,1	+0,0
m_1 kg/s	0,1324	0,1391	+5,0	0,1354	+2,3
p_3 bar	1,2090	1,2065	-0,2	1,2183	+0,8
p_4 bar	0,9998	0,9938	-0,6	1,0038	+0,4
T_3 K	888,0	867,1	-2,3	880,6	-0,8
T_4 K	841,0	834,3	-0,8	846,9	+0,7
n_{TC} s^{-1}	832,7	877,0	+5,3	893,0	+7,2
CPU s		35,0		670,0	

$n^*_{TC}=860 s^{-1}$

Preglednica 2: *Primerjava izračunanih in izmerjenih povprečnih parametrov izmenjave delovne snovi* (n_E = 2150 min^{-1})

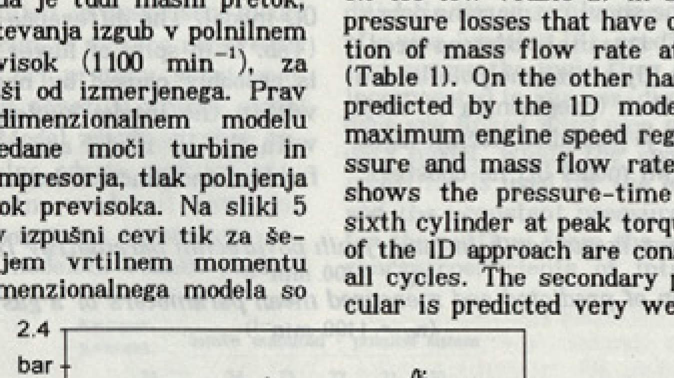
Table 2: *Comparison of predicted and measured mean parameters of a gas exchange process* (n_E = 2150 min^{-1})

T 2	meritev measured	0D	Δ %	1D	Δ%
p_1 bar	0,9541	0,9628	+0,9	0,9446	-1,0
p_2 bar	2,3987	2,1341	-11,0	2,4357	+1,5
T_1 K	304,0	300,0	-1,3	299,2	-1,6
T_2 K	422,0	403,0	-4,5	428,3	+1,5
T_{ic} K	328,0	328,0	0,0	328,2	+0,1
m_1 kg/s	0,3974	0,3714	-6,5	0,4187	+5,4
p_3 bar	2,1329	2,0766	-2,6	2,2157	+3,9
p_4 bar	1,0501	1,0339	-1,5	1,0472	-0,3
T_3 K	853,5	869,0	+1,8	835,9	-2,1
T_4 K	745,0	781,0	+4,8	717,6	-3,7
n_{TC} s^{-1}	1476,0	1477,5	+0,1	1599,5	+8,4
CPU s		58,0		392,0	

n^*_{TC}=1612s^{-1}

Rezultati enodimenzionalnega modela, ki obravnava trenje v ceveh in izgube v spojih, so namreč neprimerno boljši. Pri največjem številu vrtljajev (preglednica 2) so razlike med rezultati simuliranj in meritev večje. Rezultati 0D modela so praviloma nižji od izmerjenih, zaradi neustreznega popisa prenosa energije od valjev k turbini. Velika kinetična energija, ki jo imajo izpušni plini ob iztekanju iz valjev, se namreč v opazovani prostornini zaduši in ne prenese k turbini. Zato se zmanjša moč turbine in število vrtljajev turbokompresorja. Slednje povzroči tudi padec polnilnega tlaka p_2. Ta učinek je še posebej izrazit pri večjih vrtljajih motorja, ko so večje tudi hitrosti (kinetična energija) plinov. Tlak polnjenja pade tako močno (11%), da je tudi masni pretok, ki je sicer zaradi neupoštevanja izgub v polnilnem sistemu praviloma previsok (1100 min^{-1}), za dobrih 6 odstotkov manjši od izmerjenega. Prav nasprotno sta v enodimenzionalnem modelu zaradi previsoko napovedane moči turbine in števila vrtljajev turbokompresorja, tlak polnjenja in predvsem masni pretok previsoka. Na sliki 5 je prikazan potek tlaka v izpušni cevi tik za šestim valjem pri največjem vrtilnem momentu motorja. Rezultati enodimenzionalnega modela so

Higher discrepancy between experimental and computational results is observed at the maximum rated power regime (Table 2). Results of the 0D model are considerably lower than measured values. This is caused to a large extent by inadequate prediction of energy transfer from the cylinders to the turbine by 0D model. The high kinetic energy of the gas leaving the exhaust port is damped in the control volume and it is not transmitted to the turbine. Turbine power and consecutive turbocharger rotational speed are therefore underestimated and the boost pressure p_2 is reduced (see Tables 1 and 2). This effect is especially marked at higher engine speeds, when the flow velocities (kinetic energy) in the exhaust manifold are high. The boost pressure drops drastically (11%) and the predicted mass flow rate is 6% too low (Table 2) in spite of neglecting the pressure losses that have caused the overestimation of mass flow rate at lower engine speeds (Table 1). On the other hand, the turbine power, predicted by the 1D model, is overestimated at maximum engine speed regime and the boost pressure and mass flow rate are increased. Fig. 5 shows the pressure-time histories behind the sixth cylinder at peak torque regime. The results of the 1D approach are considerably better during all cycles. The secondary pressure rise in particular is predicted very well and it is confirmed



Sl. 5. *Primerjava potekov tlaka za šestim valjem (režim največjega vrtilnega momenta)*
Fig. 5. *Comparison of pressure time histories behind the sixth cylinder (peak torque regime)*

izrazito boljši v vseh fazah, tako med izpuhom iz šestega valja kakor tudi v intervalu, ko je izpušni ventil šestega valja zaprt. Še posebej dobro je napovedan ponoven dvig tlaka v drugi polovici izpuha. Ta nastane zaradi vpliva sosednje izpušne veje in je za model 0D precej manj izrazit. Slabše rezultate modela 0D gre v tem primeru pripisati predvsem velikemu vplivu, ki ga ima na procese v kompaktnih izpušnih sistemih za impulzno tlačno polnjenje širjenje tlačnih valov in ga model 0D popolnoma zanemari.

5 SKLEPI

Na podlagi primerjave rezultatov obeh simulirnih metod z meritvami lahko sklenemo naslednje:

— rezultati enodimenzionalnega modela so natančnejši tako z vidika povprečenih parametrov pretoka, kakor tudi lokalnega poteka stanja (tlaka) v sistemu;

— poglavitna pomanjkljivost modela 0D je v konkretnem primeru prevelik masni pretok, ki je posledica neupoštevaja lokalnih izgub v polnilnem sistemu;

— nadaljnja pomanjkljivost modela 0D je neustrezen popis procesov v ceveh. Ta je izrazita predvsem pri simuliranju motorjev z impulznim sistemom tlačnega polnjenja, za katere je značilno, da se energija izpušnih plinov v obliki tlačnih valov prenaša (s hitrostjo zvoka) od valjev k turbini. Posledica tega je premajhna moč turbine in predvsem pri večjih vrtljajih motorja izrazit padec polnilnega tlaka;

— naštetih pomanjkljivosti v enodimenzionalnem modelu ne opazimo. Masni pretok je sicer večji od izmerjenega v vseh treh primerih, vendar predvsem zaradi višjih tlakov polnjenja, zaradi prevelike moči turbine in vrtljajev turbokompresorja;

— računski časi za dosego ustaljenega stanja so za model 0D do 19-krat krajši (preglednici 1 in 2). Zato je za parametrske študije, pri katerih nas zanimajo predvsem globalne razmere v motorju, primernejša uporaba modela 0D, ki smo ga poprej dobro »uglasili« z meritvami (masni pretok, moč turbine). Za podrobnejše spremljanje razmer v polnilnem predvsem pa v izpušnem sistemu impulzno tlačno polnjenega motorja je vsekakor ustreznejši enodimenzionalni model.

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with experimental results. This secondary pressure rise is caused by the influence of processes in the adjoining branch pipe. The lower accuracy of pressure-time histories predicted by the 0D model is ascribed to the large influence of pressure wave distribution on the flow and on the thermodynamic processes in the small compact exhaust manifolds characteristic of pulse turbocharging. Zero-dimensional methods neglect this important feature.

5 CONCLUSION

Comparing the predictions of the two described computational methods with experimental results, the following conclusions can be noted:

— better agreement of exhaust pressure histories and better accuracy of the computed mean parameters of the gas exchange process was achieved using the 1D model;

— one of the basic deficiencies of the 0D model is overestimation of mass flow rate, caused by neglecting pressure losses within the intake manifold;

— neglecting pressure waves distribution is a further weakness of the 0D model. This is especially marked in the simulation of a turbocharged engine with pulse system turbocharging, where the kinetic energy of the gas leaving the exhaust port is partially transmitted to the turbine (with sonic velocity) in the form of a pressure wave. Consequently, too low turbine power is predicted, significanly reducing the charge pressure;

— both problems may be avoided by applying the 1D model. There is some inaccuracy in relation to the mass flow rate in all three examples, however this is a consequence of overestimated turbine power, increasing the charge pressure;

— convergence was achieved almost 19 times faster by the 0D model (Tables 1 and 2). This implies that a well »tuned« 0D model may be used for parametric studies, when only the global engine operation conditions are of interest. However, the 1D model is necessary for more detailed studies of thermodynamic and flow conditions within the intake and, especially, the exhaust system, and the design of a pulse turbocharched engine.

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6 OZNAČBE — NOTATION

A — površina
area
c_v — izohorna specifična toplota
specific heat at constant volume
D — premer
diameter
f — koeficient trenja
friction factor
I_p — polarni vztrajnostni moment
moment of inertia
h — entalpija
enthalpy
m — masa
mass
n — število vrtljajev
rotational speed
p — tlak
pressure
P — moč
power
Q — toplota
heat
R — plinska konstanta
gas constant
t — čas
time
T — temperatura
temperature
u — specifična notranja energija, hitrost
specific internal energy, velocity
U — notranja energija
internal energy
V — prostornina
volume
η — izkoristek
efficiency
Θ — kot ročične gredi
crank angle
$\varkappa$ — razmerje specifičnih toplot
specific heat ratio
λ — ekvivalentni razmernik zraka
equivalent air fuel ratio
μ — pretočno število
discharge coefficient
ρ — gostota
density

INDEKSI — SUBSCRIPTS

b — zgorelo
burned
C — kompresor
compressor
E — motor
engine
f — gorivo
fuel
g — geometrijski
geometric
hr — sprostitev toplote
heat release
ht — prenos toplote
heat transfer
IC — hladilnik polnilnega zraka
intercooler
in — vtok
inflow
m — mehanski
mechanic
out — iztok
outflow
T — turbina
turbine
TC — turbokompresor
turbocharger
0 — zajezni pogoji, okolica
stagnation conditions, ambient conditions
1 — pred kompresorjem
upstream of compressor
2 — za kompresorjem
downstream of compressor
3 — pred turbino
upstream of turbine
4 — za turbino
downstream of turbine

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