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Simuliranje prehodnih režimov delovanja motorja z notranjim zgorevanjem

Simulation of the Transient Operation of an Internal Combustion Engine

Aleš Hribernik

V prispevku sta predstavljena nelinearni model enovaljnega dizelskega motorja za preračun prehodnih delovnih režimov motorja in napoved spremenljivega momenta in vrtilne frekvence motorja. Model je razvit iz modela za simuliranje termodinamičnega krožnega procesa motorja v ustaljenih razmerah, razširjen z modelom dinamike ročičnega mehanizma predstavlja motor kot nelinearni dinamični sistem. Spremenljiva vrtilna frekvenca motorja je rezultat reševanja enačbe dinamike zveze motor - breme, pri čemer je spremenljivi moment motorja določen v odvisnosti od lege (kota) ročične gredi. Model smo preverili z rezultati meritev enovaljnega dizelskega motorja. Primerjava računskih in izmerjenih vrtilnih frekvenc je pokazala zelo dobro ujemanje ponavljajočih se sprememb vrtilne frekvence in odziva motorja med naglim pospeševanjem.

(Ključne besede: motorji z notranjim zgorevanjem, modeli termodinamični, dinamika motorjev, simuliranje)

A non-linear, transient, single-cylinder Diesel engine simulation has been developed to predict the instantaneous engine speed and torque. The basis of the model is a thermodynamic, steady state diesel engine cycle simulation. The transient extension of the original model represents the Diesel engine as a non-linear, dynamic system. The instantaneous crankshaft speed is determined from the solution of the engine-external load dynamic equation, where the engine torque is tracked on the crank-angle basis. Validation of the transient model during rapid engine acceleration shows that both the cyclic fluctuations in the instantaneous engine speed line and the overall engine response are in good agreement with experimental measurements.

(Keywords: internal combustion engines, thermodynamic models, engine-dynamic, simulation)

0 UVOD

Motorji z notranjim zgorevanjem so v osnovi nelinearni mehanski sistemi. Zato je mogoče pričakovati natančne rezultate simuliranj le, kadar so fizikalni modeli zasnovani kot povsem nelinearni sistemi. Simulirni modeli termodinamičnih krožnih procesov motorjev z notranjim zgorevanjem se z izpopolnje-vanjem pojavnega popisa procesov vse bolj izpopolnjujejo in so zelo uspešni in dandanes že neizogibni za napoved karakteristik delovanja motorja v ustaljenih razmerah. Zato je povsem naravna pot razvoja nelinearnega modela motorja z notranjim zgorevanjem razširitev in dograditev modela termodinamičnega krožnega procesa motorja z upoštevanjem dinamike motorja. To je bilo tudi naše vodilo pri razvijanju nelinearnega modela motorja. Uporabili smo preskušeni programski paket za simuliranje termodinamičnih in tokovnih procesov v motorju z notranjim zgorevanjem [1] in ga dogradili z modelom dinamike motorja.

0 INTRODUCTION

Since the internal combustion engine is a highly non-linear mechanical system, truly predictive capabilities can only be attained if purely non-linear simulations are developed. Thermodynamic engine simulations have been refined with phenomenological process descriptions over the years and have been used very successfully for the prediction of steady-state engine operation. Consequently, the natural approach to the development of a non-linear engine simulation is by building on and extending such steady-state thermodynamic models to consider the engine dynamics. The development of our non-linear engine model was undertaken by taking into account these factors. The foundation for the transient model was the successfully tested thermodynamic simulation of an internal combustion engine [1] which was upgraded by the engine dynamics model.

1 TERMODINAMIČNI MODEL MOTORJA

Uporabili smo brezdimenzijski, termodinamični model za simuliranje dizelskega motorja v ustaljenih razmerah, ki je zasnovan na metodi “polnjenje-praznjenje”. Model smo v zadnjih desetih letih razvijali in testirali na Fakulteti za strojništvo v Mariboru, ga obširno predstavili [1] in z njegovo uporabo izvedli številne raziskave delovnih karakteristik sesalnih in tlačno polnjenih motorjev [2], [3]. Osnovne značilnosti modela so na kratko povzete v nadaljevanju.

Pretok plinov v motorju je v modelu ponazorjen s pretokom snovi med nadzornimi prostorninami. Te predstavljajo polnilne in izpušne zbiralnike ter valje motorja. Pretok mase in energije med nadzornimi prostorninami določajo robni pogoji, ki so zapisani z izrazi pretoka skozi zaslonko. Predpostavljeno je, da je stanje plina v nadzorni prostornini enotno, zato ga določata že diferencialni enačbi o ohranitvi mase in energije. Z robnimi pogoji povežemo dvojice diferencialnih enačb posameznih nadzornih prostornin v sistem navadnih nelinearnih diferencialnih enačb prvega reda. Rezultat reševanja tega sistema diferencialnih enačb v času (kotu zavrtitve ročične gredi) pa so potek tlaka, temperature in sestave delovne snovi v posamezni nadzorni prostornini.

2 DINAMIKA ROČIČNEGA MEHANIZMA MOTORJA

Slika 1 prikazuje poenostavljen dinamični sistem enovaljnega motorja. Lega ročice za dobavo goriva in z njim količina vbrizganega goriva ter stanje okolice sta osnovna podatka za simuliranje motorskega krožnega procesa. Rezultat te je indicirani moment motorja, iz katerega sledi po odbitju momenta trenja v motorju dejanski moment na motorski gredi. Razlika med dejanskim momentom in momentom zunanjega bremena je rezultirajoči moment, katerega posledica

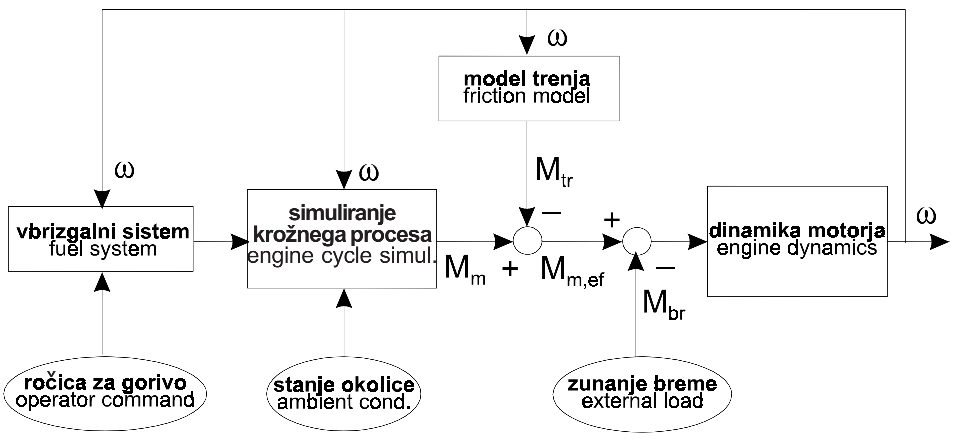
1 THERMODYNAMICS MODEL OF THE ENGINE

The thermodynamic, zero-dimensional, steady state Diesel engine simulation based on the “Filling-Emptying” method was used. The model has been developed and tested at the Faculty of Mechanical Engineering in Maribor over the last ten years. The parent simulation has been fully presented [1] and used to study operational characteristics of several naturally aspirated and turbocharged Diesel engines [2], [3]. The basic characteristics of the model will be briefly presented.

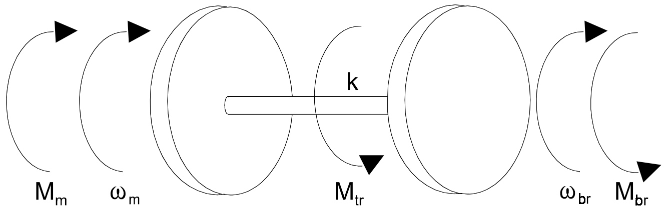
The flow of gases within the engine is presented by the flow of mass and energy between the control volumes. These represent intake and exhaust manifolds and the cylinders of the engine. The flow of mass and energy into and out of the control volume are determined by boundary conditions based on the analogy of convergent nozzle flow. It is assumed, that the thermodynamic state within the control volume is uniform. As a result, it may be predicted by solving the mass and energy conservation equations. Differential equations of each control volume are connected into a system of ordinary, non-linear differential equations of the first order by boundary conditions. The solution of this system yields pressure, temperature and gas composition variation with time (crank angle) within each control volume.

2 ENGINE DYNAMICS

Figure 1 shows a schematic representation of the single-cylinder engine dynamic system. At a given instant in time, the operator command, i.e. fuelling rate, and the ambient conditions are key inputs to the Diesel engine cycle simulation. The indicated engine torque is the output required for the engine dynamics equation. The friction torque is subtracted from the indicated torque to produce brake torque at the shaft. Subsequently, the resistance torque



Sl. 1. *Blokovni diagram enovaljnega motorja*
Fig. 1. *Block diagram for the single-cylinder engine*



Sl. 2. *Model dinamike sistema motor-zunanje breme*
Fig. 2. *Equivalent system for engine-external load dynamics*

je zvečanje ali zmanjšanje vrtilne frekvence motorja. Zaradi ponavljajočega se delovanja motorja je vrtilna frekvenca motorja neustaljena, tudi kadar je srednje število vrtljajev motorja konstantno, kar je posledica hitrih sprememb tlaka v valju in z njimi sil, ki delujejo na ročični mehanizem.

Kakor prikazuje slika 2, lahko sistem motor - zunanje breme ponazorimo s sistemom dveh diskov, povezanih z elastično gredjo. Pri tem je polarni vztrajnostni moment prvega diska ekvivalent polarnega vztrajnostnega momenta rotirajočih delov motorja. Podobno je vztrajnostni polarni moment drugega diska ekvivalent vztrajnostnega polarnega momenta zunanjega bremena, ki je lahko vozilo, delovni stroj ali zavora v preskušališču motorjev. V primeru elastične gredi popišeta dinamiko sistema s slike 2 naslednji enačbi:

is subtracted from the brake torque and the net value calculated. If the net value is positive the engine will accelerate and vice versa. The instantaneous rotational speed of the crankshaft is unsteady during any engine cycle, even if the mean engine speed is constant, due to rapid changes in cylinder pressure and the consequent changes in forces acting on the crank during the cycle.

As shown in Fig. 2, the engine/load system can be modelled as two disks connected by a weightless shaft, one disk having the equivalent polar moment of inertia of the engine moving parts, and the other having the equivalent polar moment of inertia of the load (vehicle or dynamometer in the test cell). The following two equations describe the dynamic system in the case of an elastic shaft:

$$I_m \dot{\omega}_m + M_{tr} + k(\theta_m - \theta_{br}) = M_m \qquad (1)$$

$$I_{br} \dot{\omega}_{br} + M_{br} = k(\theta_m - \theta_{br}) \qquad (2).$$

Z dovolj veliko natančnostjo lahko predpostavimo, da je gred toga in da lahko zasuk (θ_m - θ_{br}) med diskoma zanemarimo. Iz tega torej izhaja, da sta kotni hitrosti motorja in zunanjega bremena enaki:

The power transmission shaft can usually be considered sufficiently rigid, and consequently the twist (θ_m - θ_{br}) between the two disks can be neglected. Hence, the rotational velocity of engine is equal to one of the external load:

$$\omega_m = \omega_{br}$$

In ker zato velja:

Thus there is no phase shift:

$$\theta_m = \theta_{br}$$

lahko sistem enačb zapišemo kot:

and the system dynamic equation becomes:

$$\dot{\omega}_m = \frac{M_m - M_{tr} - M_{br}}{I_m + I_{br}} \qquad (3).$$

Enačbo (3) smo uvrstili v sistem diferencialnih enačb prvega reda, s katerim smo popisali termodinamični model motorja in razširjeni sistem enačb numerično integrirali po metodi Runge-Kutta 4+. Ker se integracija izvaja po kotu zavrtitve ročične gredi in ne po času, je treba v sistem diferencialnih enačb vključiti še enačbo za spremembo časa po kotu:

Equation (3) was incorporated into a system of differential equations of the first order that form the engine thermodynamic cycle simulation. This extended system was numerically integrated by the Runge-Kutta 4+ method. The engine cycle simulation uses the angular position of the crank shaft in degrees crank-angle and not explicitly the time. Hence, the following equation:

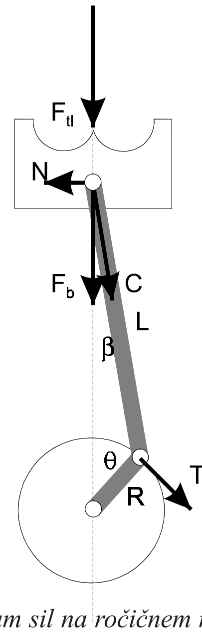
$$\frac{dt}{d\theta} = \frac{\pi}{180\omega} \qquad (4),$$

saj zaradi spremenljive kotne hitrosti ročične gredi časovni korak ni več sorazmeren integracijskemu koraku, podanim s kotom ročične gredi.

was added to the system of differential equations in order to determine the crank-angle time interdependence, since the time step is no longer proportional to the crank-angle integration step when the rotational engine speed is not constant.

2.1 Določitev trenutnega indiciranega momenta in polarnega vztrajnostnega momenta motorja

2.1 Instantaneous indicated engine torque and polar moment of inertia of the engine



Sl. 3. *Diagram sil na ročičnem mehanizmu*
Fig. 3. *Forces on the slider-crank mechanism*

Ročični mehanizem prikazuje slika 3. Trenutna lega bata in razmerje sil določata kot zavrtitve ročične gredi in geometrijske oblike ročičnega mehanizma: dolžina ojnice L in ročica gredi R. Poenostavljeno delujeta na bat dve osnovni sili. To sta tlačna sila:

A schematic of the piston (slider) – crank mechanism is shown in Fig. 3. The instantaneous piston position and the relationship between the acting forces are determined by the crank shaft angular position and the geometrical parameters of the slider-crank mechanism: connecting rod length L and crank radius R. There are two forces acting on the piston, i.e. the pressure force:

$$\vec{F}_{tl} = \left(p_v - p_{s,k}\right) \frac{\pi D_b^2}{4} \qquad (5),$$

ki je posledica razlike tlakov v valju in karterju motorja ter vztrajnostna sila:

which is the result of the pressure difference in the cylinder and in the crank case, and the inertial force:

$$\vec{F}_{vztr} = -\left(m_b + m_{ojn}'\right) a_b \qquad (6)$$

Pri tem je m_{ojn}' masa dela ojnice, katerega gibanje lahko približno označimo s translatornim gibanjem bata (običajno 1/3 m_{ojn} [4]), a_b pa je linearni pospešek bata, ki ga natančno določa izraz:

where m_{ojn}' is the reciprocating part of the connecting rod mass (usually 1/3 m_{ojn} [4]), and a_b is the linear acceleration of the reciprocating parts. The acceleration can be calculated using the following exact expression:

$$a_b = R\omega^2\left[\cos\theta + \frac{\lambda\left(\cos 2\theta + \lambda^2 \sin^4\theta\right)}{\left(1-\lambda^2\sin^2\theta\right)^{3/2}}\right] + R\dot{\omega}\sin\theta \cdot \left[1 + \frac{\lambda\cos\theta}{\left(1-\lambda^2\sin^2\theta\right)^{1/2}}\right] \qquad (7),$$

pri čemer je λ definiran z razmerjem:

where λ is defined by the ratio:

$$\lambda = \frac{R}{L} \quad (8).$$

Rezultirajoča sila, ki deluje na bat, je torej:

The resultant force on the piston is therefore:

$$\vec{F}_b = \vec{F}_{tl} + \vec{F}_{vztr} \quad (9).$$

Njena komponenta v smeri osi ojnice je:

The component force in the direction of the connecting rod axis is:

$$\vec{C} = \vec{F}_b \frac{1}{\cos\beta} \quad (10)$$

Ta deluje na ležaj ojnice in ročične gredi, pri čemer je njena komponenta v tangentni smeri:

and the component of the force on the crank journal in the tangential direction is:

$$\vec{T} = \vec{C}\sin(\theta+\beta) = \vec{F}_b \frac{\sin(\theta+\beta)}{\cos\beta} \quad (11).$$

Moment, ki se z ročičnega mehanizma prenaša na motorsko gred, je torej:

The torque on the shaft can be calculated therefore by:

$$M_m = RT \quad (12).$$

Pri nižjih vrtilnih frekvencah motorja, ki so značilne za večino dizelskih motorjev, lahko z dovolj veliko natančnostjo predpostavimo, da je polarni vztrajnostni moment motorja konstanten. Sestavljajo ga prispevki vztrajnika in rotirajočih delov ročičnega mehanizma, to je motorske gredi z ročico in dela ojnice, katerega gibanje lahko označimo za rotacijo (običajno 2/3 m_{ojn} [4]). Pri velikih vrtilnih frekvencah, ki so značilne predvsem za bencinske motorje in manjše hitro tekoče dizelske motorje pa je vpliv spreminjajočega polarnega vztrajnostnega momenta že tako velik, da ga je v dinamičnem modelu motorja treba upoštevati [5].

Under low and medium engine speeds, characteristic for most Diesel engines, it can be assumed that the polar moment of inertia of the engine is constant. It consists of the moment of inertia of the flywheel and of the rotating parts of the slider-crank mechanism, i.e. the engine crankshaft and the rotating part of the connecting rod (usually 2/3 m_{ojn} [4]). However, under high engine speeds, characteristic for gasoline engines and high speed Diesel engines, the influence of the fluctuating part of the polar moment of inertia becomes very high, and it has to be considered in the engine dynamics model [5].

2.2 Navor trenja in zunanjega bremena

2.2 Friction torque and external load torque

Notranje trenje v motorju običajno določamo na podlagi meritev delovnih karakteristik gnanega motorja. Značilni izraz za določitev izgub srednjega indiciranega tlaka dizelskega motorja je oblike [6]:

Engine friction is usually estimated from experiments by using motoring tests. A typical correlation for the frictional losses of the mean indicated pressure of a Diesel engine has the following form [6]:

$$p_{tr} = C_1 + \frac{48n}{1000} + 0{,}4A_b^2 \quad (13),$$

pri čemer je C_1 koeficient, ki ga dobimo na podlagi meritev. Srednji moment trenja pa izhaja iz izraza:

where C_1 is a coefficient that can be determined by experiment. The friction torque can be calculated from:

$$M_{tr} = \frac{p_{tr}V_h}{2\pi n}1000 \quad (14).$$

Moment zunanjega bremena kakor tudi njegov polarni vztrajnostni moment pa sta odvisna od tipa gnanega stroja. Tako ju za primer motornih

The torque of an external load and its moment of inertia depend on the engine application. If the model is to be used to simulate engine transients vozil določimo z ustreznim dinamičnim modelom, ki popiše prenos momenta od motorske gredi, prek sklopke, menjalnika in diferenciala na pogonska kolesa vozila. Pri tem je moment obremenitve določen z upori vožnje, polarni vztrajnostni moment pa s prispevki rotirajočih delov, ki sodelujejo pri prenosu momenta od motorja in prek pogonskih koles na cestišče. Precej preprostejši pa je primer motorja na preskušališču, kjer moment bremena uravnava zavora, katere polarni vztrajnostni moment je tudi vztrajnostni moment zunanjega bremena.

within the vehicle, an additional model of vehicle dynamics that represents the transmission of torque from the engine shaft, over a clutch, gear box, and differential onto the driving wheels is necessary. The load torque is then determined by the driving resistance, while the moment of inertia consists of contributions from all of the rotating parts that help to transmit the torque from the engine to the road. However, if the model is intended to simulate an engine on a transient test bed, the load is applied and controlled by the dynamometer and the polar moment of inertia of the load is simply equal to the dynamometer inertia.

3 REZULTATI SIMULACIJ IN PRIMERJAVE Z MERITVAMI

Model smo uporabili za simuliranje razmer med pospeševanjem enovaljnega, zračno hlajenega, 4-taktnega dizelskega motorja z direktnim vbrizgom goriva. Osnovni podatki o motorju so zbrani v preglednici 1.

Na sliki 4 je prikazan spremenljivi moment obravnavanega enovaljnega motorja, izračunan po enačbi (12) in pri ustaljenih obratovalnih razmerah, tj. n_{sr}= konst. Kljub ustaljenim delovnim razmeram so spremembe navora v toku motorskega ciklusa zelo velike. V začetni fazi (zgorevanje – 0 do 180^{0}RG) in končni fazi (kompresija – 540 do 720 ^{0}RG) prevladuje

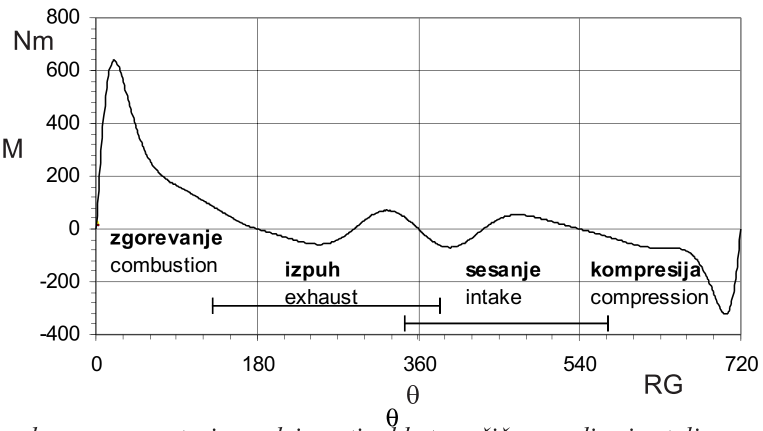
3 RESULTS OF SIMULATION AND EXPERIMENTAL VALIDATION

A model has been used for the simulation of transient operation of an air-cooled, direct-injection, 4-stroke, single-cylinder Diesel engine. Basic engine data are specified in Table 1.

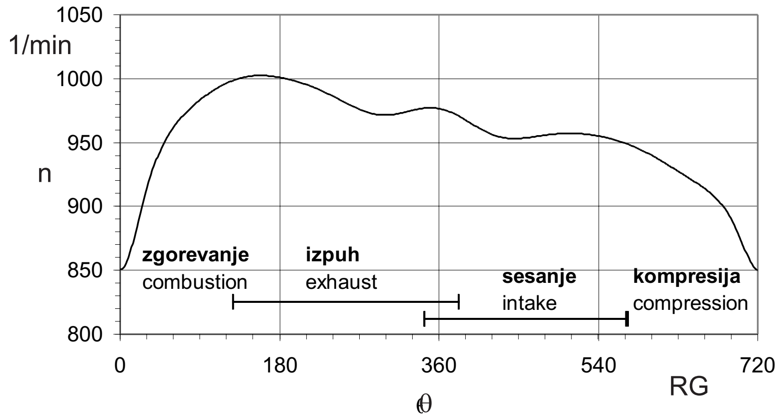
The instantaneous engine torque of a single cylinder engine, calculated using Eq. (12) under steady-state operation, i.e. n_{sr}= const., is shown in Fig. 4. Despite steady-state engine operation huge fluctuations of engine torque are observed. In the 1st period (combustion – 0 to 180 ^{0}CA) and in the 4th one (compression – 540 to 720 ^{0}CA) dominates the pressure force, while the inertial force dominates

Preglednica 1. *Podatki o motorju Deutz MAG F1L 210 D*
Table 1. *Deutz MAG F1L 210 D engine specification*

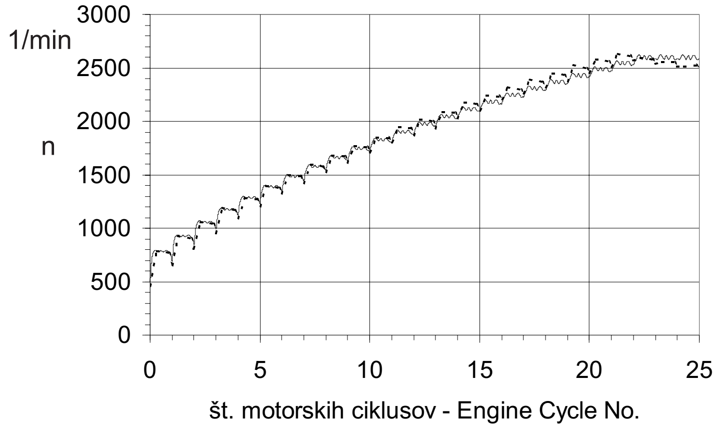
premer valja bore	m	0,095
gib stroke	m	0,095
dolžina ojnice connecting rod length	m	0,160
kompresijsko razmerje compression ratio	-	17
največja dobava goriva maximum fuel per cycle	mm^3	40



Sl. 4. *Sprememba navora motorja v odvisnosti od kota ročične gredi pri ustaljenem režimu n_{sr}=960 min^{-1}*
Fig. 4. *Fluctuations of the engine torque with crank angle under steady state operation n_{sr}=960 rpm*



Sl. 5. *Sprememba vrtilne frekvence motorja v odvisnosti od kota ročične gredi pri ustaljenem režimu n_{sr}=960 min^{-1}*
Fig. 5. *Fluctuations of the engine speed with crank angle under steady state operation n_{sr}=960 rpm*



Sl. 6. *Primerjava izmerjenega (·······) in izračunanega (——) števila vrtljajev motorja v odvisnosti od števila krožnih procesov motorja med pospeševanjem (M_{br}=0 Nm, 40 mm^3/proces)*
Fig. 6. *Comparison of predicted (——) and measured (·······) instantaneous engine speeed during acceleration (M_{br}=0 Nm, 40 mm^3/cycle)*

vpliv tlačne sile, medtem ko je v toku izmenjave delovne snovi (180 do 540 ^{0}RG) prevladujoč vpliv vztrajnostnih sil, prispevek tlačnih sil pa je zaradi majhnih razlik tlaka v prostoru nad batom in pod njim neznaten. Posledica tako velikih sprememb momenta je nihanje vrtilne frekvence motorja (slika 5). To je še posebej izrazito pri enovaljnem motorju. V obravnavanem primeru dosegajo razlike v vrtilni frekvenci motorja do 150 min^{-1}, oziroma prek 15% srednje vrednosti.

Na sliki 6 je prikazana primerjava izračunanih in izmerjenih vrednosti vrtilnih frekvenc motorja v prehodnem obratovalnem režimu. Razbremenjen motor pri največji dobavi goriva pospešuje od prostega teka do mejne vrtilne frekvence, pri katerem krmilnik ustavi dobavo goriva in prepreči nadaljnje pospeševanje. Pri tem preide motor skozi celoten spekter vrtljajev. Značilno zanj je, da se amplitude variacij vrtilne frekvence zmanjšujejo z večanjem hitrosti motorja. Ugotovimo lahko, da je ujemanje izmerjenih in izračunanih vrednosti vse do 2000 min^{-1} zelo dobro, nato pa začne

during the gas exchange process (180 to 540 ^{0}CA) when the pressure force is insignificant, because of the small difference of pressures above and below the piston. As a consequence of the huge fluctuations in engine torque, the variations in the instantaneous engine speed are also very pronounced (Fig. 5). This is especially characteristic for a single cylinder engine. In the presented example these variations exceed 150 rpm, i.e. more than 15% of the nominal (mean) engine speed.

A comparison of measured and predicted values of the instantaneous engine speed during an engine transient is presented in Fig. 6. The engine accelerates from idle under full load, defined as a 100% fuelling rate, and goes through the whole speed range until it reaches its maximum speed. From that point on, the controller cuts off the fuel in order to prevent the engine from over speeding. The amplitudes of cyclic speed fluctuations tend to decrease as the mean engine speed increases. The agreement between calculated and measured values below 2000 rpm is very good. After that point the izračunana hitrost vse bolj zaostajati, tako da je računsko potreben celotni motorski krožni proces več, da motor doseže največje vrtilne frekvence pri 2500 min^{-1}. Vzrokov za to je lahko več. Med osnovnimi pa so gotovo pomanjkanje podatkov o spremembi temperature mazalnega olja in njenem vplivu na trenje motorja, kakor tudi natančnejši podatki o dobavi goriva, ki je odvisna od vrtilne frekvence motorja.

4 SKLEP

Ustaljeni, fenomenološki, brezdimenzijski model je bil uporabljen kot podlaga za razvoj modela za popis prehodnih delovnih režimov enovaljnega dizelskega motorja. Razširitev modela z zasledovanjem spremenljivega momenta motorja po kotu ročične gredi in z reševanjem enačb dinamike motorja je omogočila popis neustaljenih razmer in prehodnih pojavov. Natančnost modela smo preverili na podlagi primerjave računskih rezultatov z meritvami. Napovedi fluktuacije vrtilne frekvence med krožnim procesom se zelo dobro ujemajo z izmerjenimi vrednostmi. Tudi pospeševanje razbremenjenega motorja pri največji dobavi goriva je pravilno napovedano. Pri manjših hitrostih motorja so variacije števila vrtljajev znotraj procesa tudi do 200 min^{-1}, medtem ko se zaradi povečanja vztrajnostnih sil ročičnega mehanizma vpliv plinskih sil med zgorevanjem nekoliko omili in so zato variacije vrtilne frekvence precej manjše, delovanje motorja pa precej bolj stabilno.

model underpredicts the engine speed and it takes the model one extra cycle to reach the maximum engine speed at 2500 rpm. The minor discrepancies at very high speed can be attributed to insufficient model inputs that are difficult to measure during a transient, such as the change of oil temperature and friction, and the effect of engine speed on the cyclic amount of fuel injected.

4 CONCLUSION

A steady-state, phenomenological zero-dimensional model has been used as the basis for the development of a transient, single-cylinder, Diesel engine model. Its extension has involved an instantaneous engine torque and engine dynamics model. Subsequently, the transient simulation has been validated against experimental results from a single-cylinder engine. Simulation predictions of the cyclic fluctuations in the instantaneous crankshaft speed characteristic are in good agreement with the experiment. The overall engine response during free acceleration with the maximum amount of fuel injected also agrees well with the measured data. The amplitude of the fluctuation in the rotational speed of the crankshaft can be as high as 200 rpm, at low speed. However, increased inertia forces on the reciprocating components at higher speeds causes a reduction of the peak instantaneous pressure force during combustion. Hence, instantaneous fluctuations in the crankshaft angular velocity are significantly reduced at higher speeds, where the engine operation is much more uniform.

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Uporabljeni simboli
Nomenclature

površina	A	area
pospešek	a	acceleration
komponenta sile v smeri osi ojnice	C	component force in the direction of the connecting rod axis
premer	D	diameter
sila	F	force
polarni vztrajnostni moment	I	polar moment of inertia
togost gredi	k	shaft rigidity
dolžina ojnice	L	connection rod length
moment	M	torque
masa	m	mass
vrtilna frekvenca motorja	n	engine speed
tlak	p	pressure
komponenta sile v smeri tangente na ročico	T	component of the force on the crank journal in the tangential direction
čas	t	time
gibna prostornina	V_h	swept volume
kot nihanja ojnice okoli osi bata	β	angle of swing of the connecting rod from the cylinder axis
razmerje med ročico in dolžino ojnice	λ	ratio of crank radius and connecting rod length
kot zavrtitve ročične gredi	θ	angular position of crankshaft in degrees crank-angle
kotna hitrost	ω	angular velocity

Indeksi:		**Subscripts:**
bat	b	piston
breme	br	load
dejansko	ef	effective
motor	m	engine
sesalni zbiralnik	s,k	intake manifold
srednji	sr	mean
tlačni	tl	pressure
trenje	tr	friction
valj	v	cylinder
vztrajnost	vztr	inertia

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