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A. Notă de constatare şi confirmare a indiciilor de plagiat se bazează pe fişa suspiciunii inclusă în decizie.

Fişa suspiciunii de plagiat

Opera suspicionată (OS) Suspicious work	Opera autentică (OA) Authentic work

OS	ZICHIL Valentin; PINTILIE ,Gheorghe; SAVIN Carmen, and JUDELE, Adrian. Numerical method using finite element method for forces and acceleration of the piston-crank mechanism at the rotary valves D.I. diesel engine. Modelare şi Optimizare în Construcţia de Maşini (MOCM). **9**(1). p.117-123. 2003
OA	ZWEIRI Y.H.; WHIDBORNE J.F. and SENEVIRATNE L.D. Dynamic simulation of a single-cylinder diesel engine including dynamometer modelling and friction. *Proc.Instn.Mech.Engrs.* vol 213 part D. pag 391-402. 1999.

Incidenţa minimă a informaţiei preluate /Minimal incidence of taken over information	
p.117:26 – p.117:33	p.392:47d – p.393.01s; p.393:05s – p.393:08s; p.393:37s – p.393:40s
p.118:01 – p.118:06	p.393:03d – p.393:07d
p.118: Figure 1	p.393s: Fig.1
p.118:07 – p.118:16	p.393:31s – p.393:36s; p.393:08d - p.394:03s

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> *„…este un element de comunicare prezentat în formă scrisă, ca text, imagine sau combinat, care posedă un subiect, o organizare sau o construcţie logică şi de argumentare care presupune nişte premise, un raţionament şi o concluzie. Piesa de creaţie presupune în mod necesar o formă de exprimare specifică unei persoane. Piesa de creaţie se poate asocia cu întreaga operă autentică sau cu o parte a acesteia…"*

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[1] Legea nr. 206/2004 privind buna conduită în cercetarea ştiinţifică, dezvoltarea tehnologică şi inovare, publicată în Monitorul Oficial al României, Partea I, nr. 505 din 4 iunie 2004
[2] ISOC, D. *Ghid de acţiune împotriva plagiatului: bună-conduită, prevenire, combatere*. Cluj-Napoca: Ecou Transilvan, 2012.
[3] ISOC, D. *Prevenitor de plagiat*. Cluj-Napoca: Ecou Transilvan, 2014.

Dynamic simulation of a single-cylinder diesel engine including dynamometer modelling and friction

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Abstract: The recent drive to reduce emissions and improve efficiency means that there is a real need for improved models for the simulation of direct-injection diesel engines. With a view to improving the transient fuel control, a model of the non-linear transient dynamics of a generic direct-injection single-cylinder diesel engine is developed in order to predict the instantaneous engine speed and torque. The instantaneous crankshaft speed is determined from the solution of the engine–external load dynamics equation, where the engine torque is tracked on a crank angle basis. The model is based on an analysis of all the major forces internal to the engine and dynamometer. The friction components of the piston assembly, the bearings, the valve train, the pumping and the pumps are also included. The model is implemented in MATLAB/SIMULINK, and simulation results of both transient and steady state dynamics are presented. The simulation results of the instantaneous engine speed are compared with measured data, and it is seen that there is excellent agreement between them.

Keywords: diesel engine, simulation, dynamic modelling, transient response, engine friction

NOTATION

A	piston area (m^2)
B	width of the ring in the direction of motion (slider width) (m)
c	compression ratio
d	cylinder bore diameter (m)
D	damping coefficient [N m/(rad/s)]
D_b	bearing diameter (m)
D_{iv}	intake valve diameter (m)
F_{cr}	connecting rod force (N)
F_f	friction force (N)
$F_{f_{RLi}}$	friction force between piston ring and liner (N)
$F_{f_{SL}}$	friction force between piston skirt and liner (N)
F_{st}	side thrust force (N)
$G(\theta)$	geometric function
h	total oil-film thickness (m)
h_l	maximum oil-film thickness (m)
h_m	minimum oil-film thickness (m)
i	subscript to identify each piston ring
j	subscript to identify each load at different times
J	moment of inertia of the crankshaft, flywheel, main gear and rotating part of the connecting rod (kg m^2)
J_1	moment of inertia of the dynamometer rotating parts (kg m^2)
k	subscript to identify each friction torque of the engine
l	circumferential length of the ring (m)
L	connecting rod length (m)
M	mass of the piston, rings, pin and small end of the connecting rod (kg)
n_c	number of cylinders
n_{iv}	number of intake valves
p_{atm}	atmospheric pressure (101 kPa)
p_i	indicated pressure (Pa)
r	crank radius (m)
S	stiffness (N m/rad)
t	time (s)
T_b	brake torque (N m)
T_D	damping torque (N m)
T_{f1}	piston ring assembly friction torque (N m)
T_{f2}	crankshaft bearing friction torque (N m)
T_{f3}	valve train friction torque (N m)
T_{f4}	pumping losses torque (N m)
T_{f5}	pump friction torque (N m)
T_i	indicated torque (N m)
T_{Lj}	engine load torque (N m)

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T_r	reciprocating inertia torque (N m)
T_S	torsional stiffness torque (N m)
U	velocity of the slider (m/s)
V_d	displacement volume (m^3)
W	load on the slider (N/m)
x	Cartesian coordinates in the x direction
y	piston displacement (m)
α	weighting coefficient
β	connecting rod angle (rad)
δ	piston pin offset (m)
Δt	time step (s)
θ	crankshaft angular position (rad)
θ_1	dynamometer angular position (rad)
μ	dynamic viscosity of the oil (N s/m^2)
ξ	characteristic and inclination angle of the upper of a tilted ring profile (rad)
ρ	oil density (kg/m^3)
ϕ	connecting rod angle when the piston is at TDC (rad)

1 INTRODUCTION

The direct-injection (DI) diesel engine model has long been established as an effective tool for studying engine performance and contributing to evaluation and new developments. Most of the work done in this area has concentrated on steady state models for the purpose of modifying engine design parameters in order to minimize emissions and maximize power and fuel economy of the engine. However, recent regulations have imposed stringent emission and fuel economy standards that cannot be addressed by a steady state analysis of the engine. Simulation of transient engine response is needed to predict performance and fuel economy of diesel engines that frequently experience rapid changes in speed and load. Hence, to contribute towards solving this problem, the current research work is conducted with the aim of developing a non-linear dynamic model for direct-injection single-cylinder diesel engines that can simulate the engine performance under transient and steady state operating conditions.

Previous efforts in the area of engine dynamic modelling can be grouped into two major categories: steady state non-linear and transient non-linear models. Examples of steady state non-linear models can be found in references [1] to [4], which simulate real spark-ignition engines in order to estimate engine torque and cylinder pressure.

Some examples of transient non-linear dynamic models can be found in reference [5], where a model composed of thermodynamic and dynamic constitutive elements for a transient, multicylinder diesel engine simulation is developed. This model utilizes a quasi-steady thermodynamic process. A comparison of predicted and measured pressure traces during the transient response was satisfactory overall, but also indicated some limitations of the quasi-steady process submodels, and so Filipi and Assanis [6] have extended the steady state diesel engine simulation to include the prediction of instantaneous engine speed and torque during transients.

Important aspects of engine dynamic operation are the instantaneous torque and the cyclic nature of the gas pressure force and the slider–crank kinematics. Therefore, the objective of this work is to develop a non-linear single-cylinder diesel engine model with full transient capability explaining the relationship between the net engine torque and the angular speed of the crankshaft. Another aspect that plays an important role in engine transient modelling is the evaluation of frictional losses, especially piston assembly friction because it is a factor that strongly affects the economy, performance and durability of the reciprocating internal combustion engines. The model takes this point into consideration in dealing with the detailed analysis of engine friction components. Another advantage of this model is that full dynamic dynamometer modelling with step loading (to avoid the chatter effect) is included. In addition, the piston pin offset has been taken into consideration during the transient analysis, and motoring analysis capability could also be implemented. This paper presents the salient features of the developed model, along with a brief description of a SIMULINK [7] implementation. The dynamic engine operation is illustrated by simulation results, and the predicted engine response is validated through comparison with measured data from two different engines.

The paper is arranged as follows. Firstly, the engine and dynamometer model, which is composed of the engine and dynamometer dynamic model, the instantaneous single-cylinder engine torque model and the friction torque model, is formulated on a crank angle basis. Next is a description of the implementation, followed by some simulation results to show the model behaviour and validation. Finally, there is a discussion and some conclusions are drawn.

2 ENGINE MODELLING

2.1 Engine dynamic model

Figure 1 shows a model of the engine coupled to a dynamometer. The following two equations describe the dynamic system:

$$T_i - \sum_{k=1}^{5} T_{fk} - T_r = J\ddot{\theta} + T_S + T_D \tag{1}$$

$$T_D + T_S = J_1\ddot{\theta}_1 + \sum_{j=1}^{n} T_{Lj} \tag{2}$$

The above equations simply state Newton's second law for a rotational body. The variables used for these and all the other equations are defined in the Notation. The indicated engine torque, T_i, is generated by the conversion of chemical to thermal to mechanical energy during the combustion process. The reciprocating torque, T_r, is produced by the motion of the piston assembly and the small end of the connecting rod. The reciprocating torque and the friction torque terms, $\Sigma_{k=1}^{5} T_{fk}$, are subtracted from the instantaneous indicated torque value to produce the brake torque at the shaft. The resistance torque, $\Sigma_{j=1}^{n} T_{Lj}$, which is the result of external loading imposed on the engine by the dynamometer, is in addition to the dynamometer inertia. Owing to rapid changes in the cylinder pressure and consequent changes in the forces acting on the crank during a cycle, the instantaneous rotational speed of the crankshaft is unsteady during any engine cycle, even if the mean speed is constant. The torsional stiffness torque, T_S, and damping torque, T_D, which depend on the stiffness and damping in the coupling between the engine and dynamometer, are given by the linear relationships

$$T_S = S(\theta - \theta_1) \tag{3}$$

and

$$T_D = D(\dot{\theta} - \dot{\theta}_1) \tag{4}$$

2.2 Instantaneous engine torque model

Figure 2 shows the piston–crank mechanism with approximate kinetically equivalent point masses replacing the connecting rod. The model includes the piston pin offset. Important geometrical parameters are the crankshaft angular position, θ, the angle of the connecting rod, β, the crank radius, r, which is equal to half of the stroke, the connecting rod length, L, the piston pin offset, δ, and the connecting rod angle when the piston is at top dead centre (TDC), ϕ.

The relationship between the indicated gas pressure, P_i, and the indicated torque, T_i, is deterministic and is a function of engine geometry. This relationship is expressed as

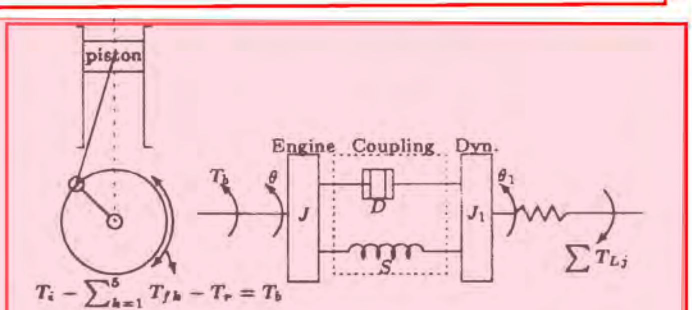

Fig. 1 Engine and dynamometer model

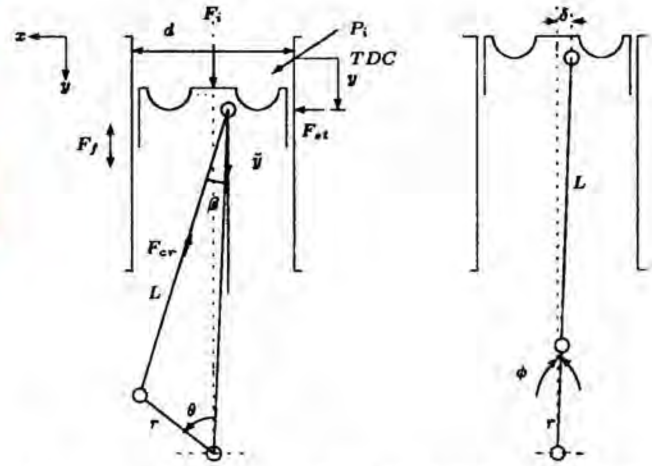

Fig. 2 Forces and acceleration of the piston–crank mechanism

$$T_i = (p_i - p_{atm})ArG(\theta) \tag{5}$$

where

$$G(\theta) = \frac{\sin(\theta + \beta)}{\cos\beta} = \sin\theta + \sqrt{\frac{1-\lambda}{\lambda}}\cos\theta \tag{6}$$

and

$$\lambda = 1 - \left[\frac{\delta + r\sin(\theta - \phi)}{L}\right]^2 \tag{7}$$

From the piston–crank geometry, the piston displacement, y, is given by

$$y = \sqrt{(r+L)^2 - \delta^2} - [L\cos\beta + r\cos(\theta - \phi)] \tag{8}$$

where angles ϕ and β can be expressed as

$$\phi = \sin^{-1}\frac{\delta}{r+L} \qquad \text{and}$$

$$\beta = \sin^{-1}\frac{\delta + r\sin(\theta - \phi)}{L} \tag{9}$$

2.2.1 *Reciprocating torque, T_r*

This term is the torque produced by the motion of the engine reciprocating components and is given as

$$T_r = MrG(\theta)\ddot{y} = MrG(\theta)[G_1(\theta)\dot{\theta}^2 + G_2(\theta)\ddot{\theta}] \tag{10}$$

where geometrical functions $G_1(\theta)$ and $G_2(\theta)$ are

$$G_1(\theta) = r\left\{\cos(\theta - \phi)\left[1 + \frac{(r/L)\cos(\theta - \phi)}{\lambda^{3/2}}\right] - \sqrt{\frac{1-\lambda}{\lambda}}\sin(\theta - \phi)\right\} \tag{11}$$

$$G_2(\theta) = r\left[\sin(\theta - \phi) + \sqrt{\frac{1-\lambda}{\lambda}}\cos(\theta - \phi)\right] \tag{12}$$

where M is the mass of the piston, rings, pin and small end of the connecting rod, and $\ddot{y}$ is the acceleration of the reciprocating components. The connecting rod is treated as an equivalent mass system, the first concentric mass is assumed to be connected to the crankpin as a big end while the second concentric mass is attached to the piston assembly as a small end. The forces acting on the connecting rod, the inertia forces and the bearing forces act at the ends of the rod. It is assumed that the big end of the connecting rod may be placed at the crankpin centre rather than at the correct point. Thus, there are no transverse components of the force between the ends of the rod to bend or shear the link, and therefore the member is in axial tension or compression.

Implementation of the instantaneous torque model obviously requires accurate masses of the reciprocating components in addition to the detailed engine geometry. The piston pin is slightly offset in order to reduce engine noise and wear during the change in direction of the normal force on the piston at the end of compression.

2.3 Friction torque model

2.3.1 *Piston ring assembly friction torque, T_{f1}*

The literature **[8–10]** suggests that the piston ring assembly may be responsible for 50–75 per cent of the entire engine friction. The components that contribute to friction are: compression rings, oil control ring, piston skirt and piston pin. The forces acting on the piston assembly include static ring tension, the gas pressure force and the inertia force. The piston assembly friction is dominated by the ring friction components [**11**]. This model takes into account only the hydrodynamic lubrication, since the friction torque is identically zero at the top and the bottom dead centre position. The piston assembly friction torque is expressed as

$$T_{f1} = F_{f1} r G(\theta) \tag{13}$$

where

$$F_{f1} = \mathrm{sgn}(\dot{y})\left[\sum F_{f_{RLi}} + F_{f_{SL}}\right] \tag{14}$$

where $\mathrm{sgn}(\dot{y})$ is the signum function (i.e. the sign of friction force is the same as the sign of piston velocity) defined as

$$\mathrm{sgn}(\dot{y}) = \begin{cases} 1, & \dot{y} > 0 \\ 0, & \dot{y} = 0 \\ -1, & \dot{y} < 0 \end{cases} \tag{15}$$

The present approach is based on calculating the piston assembly friction using a simplified model [**12, 13**] that is based on hydrodynamic lubrication. The lubrication is considered to be one-dimensional as both ring and bore are assumed to be perfectly circular with the same centre, in which case the clearance in the circumferential direction is constant, the ring is considered to be infinitely long and there is no gap effect. In this case the Reynolds equation becomes

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right) = -6U\frac{\partial h}{\partial x} + 12\frac{\partial h}{\partial t} \tag{16}$$

The load equation is

$$W = \int_0^B p \, \mathrm{d}x \tag{17}$$

and the friction force is

$$F_f = \int_0^B \left(-\frac{h}{2}\frac{\partial p}{\partial x} + \frac{\mu U}{h}\right)\mathrm{d}x \tag{18}$$

By integrating the Reynolds equation twice with boundary conditions $x = 0$, $p = p_1(t)$ and $x = B$, $p = p_2(t)$, the oil-film pressure is expressed as

$$p = \frac{6\{U - 2(h_l - h_m)/[\tan\xi(\Delta t)]\}\mu B}{h_m^2 K} \times\left[\frac{1}{h_2} - \frac{K+1}{h_2^2(K+2)} - \frac{1}{K+2}\right] + p_1 + (p_1 - p_2)\frac{(K+1)^2(h_2^2 - 1)}{[(K+1)^2 - 1]h_2^2} \tag{19}$$

where

$$h_2 = h/h_m \quad \text{and} \quad K = \frac{B\tan\xi}{h_m} \tag{20}$$

Finally, from equation (18) the friction force per circumferential unit length is

$$\frac{F_f}{l} = \frac{h_m}{2}[p_1 - p_2(K+1)] + \frac{1}{2}\left(\frac{W}{l}\right)\tan\xi + \frac{\mu UB}{h_m K}\ln(K+1) \tag{21}$$

where W is obtained from equation (17).

2.3.2 *Bearings friction torque, T_{f2}*

Bearings friction contributions come from the journal bearings and their associated seals. Journal bearings are usually designed to provide a minimum film thickness of about 2 μm. The journal bearings operate under hydrodynamic lubrication, which means a large load can be carried by the journal bearing with low energy losses under normal operating conditions. Following work done by Rezeka and Henein [**14**], the friction torque, T_{f2}, in the bearing is expressed as

$$T_{f2} = \alpha A \frac{D_b}{2}(p_i - p_{atm})\frac{|\cos\theta|}{\sqrt{\theta}} \tag{22}$$

addition, the model is being extended for multicylinder engines. Work is also ongoing to include modelling of operation from cold start.

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