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Simulation of the Behaviour Dynamics of a Diesel Engine Under Load

Ass. Prof .Ph. sc.Ratko Bozic
(Univesity of Zadar 23000, Croatia)
Zana Bozic Brkic, mag.ing.
(Open University POU Bozic, Split 21000, Croatia)
Msc. Bozic Sandro
(Ship Managment Ltd Split 21000, Croatia)

ABSTRACT
The purpose of this paper is to present the efficiency of the application of the system dynamics simulation modelling in investigating the behaviour dynamics of the diesel engine complex system. The marine diesel engine is defined by a set of non-linear differential equations, i.e. by a continuous simulation model of a higher order and the so-called equations of state. At the same time, the simulation model is discrete as it strictly satisfies the chosen value of the fundamental integration time step DT. The paper presents a mathematical model of the system consisting of a marine diesel engine model, which forms the basis for designing a system dynamic qualitative model (mental-verbal, structural and schematic model) and a quantitative model (system dynamic mathematical and information simulation model). A scenario of mixed propulsion states has been presented. Parameters obtained with the aid of the hardware simulator Kongsberg ERS-L11 MAN B&W-5L90MC–VLCC Version MC90-IV have been used in conducting the simulation.

Keywords - Simulation, System Dynamics, Modelling, Diesel Engine, Simulator, Heuristic Optimisation.

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I. INTRODUCTION

As there are a large number of various diesel engines in use, it appears that designing a universal model applicable to each individual type of engine would be very useful and effective, particularly in the case of designing complex mathematical models of diesel engines that include all essential sub-processes:

- description of the working medium (fuel, air),
- mechanical system dynamics,
- cooling system,
- lubrication system,

as well as their mutual interactions.

However, such a complex model is not suitable for the computer-supported simulation modelling of a diesel engine. Other sub-processes of the universal model are indirectly involved through the values of the various parameters that have been defined for given stationary states.

Some of the features of the diesel engine operation process have a discrete quality (injection process, ignition, burning and combustion of the air and fuel mixture) so that a discrete model of the engine is undoubtedly more exact. Since in high-speed diesel engines the time of discretisation (injection period) is much shorter than the dominant time constants of the very process, the discrete model of the engine closely approaches the continuous one, so that only the latter will be analysed in this paper.

Figure 1. shows a generic scheme of a marine turbocharged diesel engine. The following basic functional units can be noticed:

- engine mechanism (M),
- exhaust gas receiver (KIP),
- turbine (T) and the compressor (K) of the turbocharger,
- cavenge air receiver (KZ),
- high-pressure fuel pump (VTSg).

Each of the above functional units has been allocated input and output variables, variables of states and disorders, which are relevant for the model. They connect the units into a unique multivariable system.

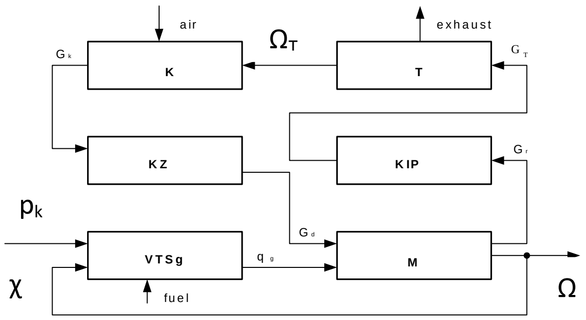

Figure 1. *Generic scheme of a marine diesel engine*

Where:

M – engine,
KIP – exhaust gas receiver,
T – turbine,
K – compressor,
KZ – scavenge air receiver,
VTSg – high-pressure fuel pump,
Ω – angular speed of the engine crankshaft (KV),
G_r – amount of the exhaust gas from the engine, entering the KIP,
p_r – exhaust gas pressure in the KIP,
ρ_r – exhaust gas density,
G_T – amount of the exhaust gas flowing through the turbine,
Ω_T – angular speed of the turbocharger,
G_k – amount of air that the turbocharger supplies to the KZ,
p_k – air pressure in the receiver,
ρ_k – specific air mass in the KZ,
χ – regulator action,
q_g – amount of fuel supplied to the engine by VTS.

II. EQUATION OF STATE OF A MARINE TURBOCHARGED DIESEL ENGINE

Dynamic features of a diesel turbocharged engine are determined by the dynamic features of the components (Figure 1.), therefore the differential equations of these units have to be considered together. However, in some cases at low volumes in the inlet and exhaust pipelines, and when there is no gas-dynamic supercharging, the impact of the pipeline volume on the engine dynamic features is negligible. This enables the simplification of its equations. As a result, it may be assumed that $V_B \approx V_r \approx 0$ and, accordingly, $T_B = T_r = T_h = 0$. Taking this condition into consideration, the system of the engine unit equations takes the following form:

$$d_D(p)_\varphi = \chi + \Theta_k \rho - \Theta_D \alpha_D$$

$$d_T(p)_{\varphi T} = \xi + \Theta_{T1}\chi - \Theta_{T2}\rho \qquad (1)$$

$$k_{B\rho} = \varphi_T - \Theta_B \varphi$$

$$k_r \xi = \varphi + \Theta_r \rho - k_h \chi$$

Since the paper discusses the system dynamics modelling from the aspect of revolution speed, it is necessary to analyse changes in angular speed of the diesel engine crankshaft. That is why the resulting equation can be presented in the following form:

$$\varphi = \Delta\varphi$$

where the determinants of the equation system (1) take the form:

$$\varphi = \begin{vmatrix} d_D(p) & 0 & -\Theta_k & 0 \\ 0 & d_T(p) & \Theta_{T2} & -1 \\ \Theta_B & -1 & k_B & 0 \\ -1 & 0 & -\Theta_r & k_r \end{vmatrix}$$

$$\Delta\varphi = \begin{vmatrix} \chi - \Theta_D \alpha_D & 0 & -\Theta_k & 0 \\ \Theta_1 \chi & d_T(p) & \Theta_{Tr} & -1 \\ 0 & -1 & k_B & 0 \\ -k_h \chi & 0 & -\Theta_r & k_r \end{vmatrix}$$

By solving these determinants and taking these expressions into account:

$$d(p) = T_D p + k_D \text{ and}$$

$$d_T(p) = T_T p + k_T$$

we obtain the possibility to present the equation of a turbocharged diesel engine, as follows:

$$d_{DH}(p)_\varphi = S(p)_\chi - u(p)\alpha_D \qquad (2)$$

where:

$d_{DH} = T_H^2 p^2 + T_{DH} p + k_{DH}$ - related operator of the turbocharged engine

$S(p) = T_s p + k_s$ - operator of action from the aspect of fuel supply control

$u(p) = T_u p + k_u$ - operator of action from the aspect of consumer adjusting

This is the form taken by the dependence of the coefficients of the above expressions on the parameters of the elements that are comprised within the engine system:

$$T_H^2 = T_D T_T k_s k_r$$

$$T_{DH} = [T_D(k_B k_T + \Theta_{T2}) + T_T(k_D k_B + \Theta_B \Theta_k)]k_r - \Theta_r T_D$$

$$k_{DH} = k_D k_r (k_B k_T + \Theta_{T2}) - \Theta_k (1 - k_T k_r \Theta_B) - \Theta_r k_D$$

$$T_s = T_T \Theta_D k_D k_r$$

$$k_s = k_r (k_B k_T + \Theta_k \Theta_{T1} + \Theta_{T2}) - \Theta_k k_h - \Theta_r$$

$$k_u = \Theta_D k_r (k_B k_T + \Theta_{T2}) - \Theta_r \Theta_D$$

The equation (2), in a differential form, can be formulated:

$$T_H^2 \frac{d^2\varphi}{dt^2} + T_{DH}\frac{d\varphi}{dt} + \varphi k_{DH} = T_s \frac{d\chi}{dt} + \chi k_s - T_u \frac{d\alpha_D}{dt} - k_u \alpha_D$$

(3)

If all elements of the equation (2) are divided by the related operator, this results in:

$$\varphi = Y_{DH}^{\chi}(p)_{\chi} + Y_{DH}^{\alpha}(p)\alpha_D$$

where the transfer functions:

$$Y_{DH}^{\chi}(p) = \frac{S(p)}{d_{DH}(p)} = \frac{T_s p + k_s}{T_H^2 p^2 + T_{DH} p + k_{DH}}$$

$$Y_{DH}^{\alpha}(p) = \frac{u(p)}{d_{DH}(p)} = \frac{T_u p + k_u}{T_H^2 p^2 + T_{DH} p + k_{DH}}$$

allow for designing a turbocharged engine scheme which is more suitable than the one in Figure 2.

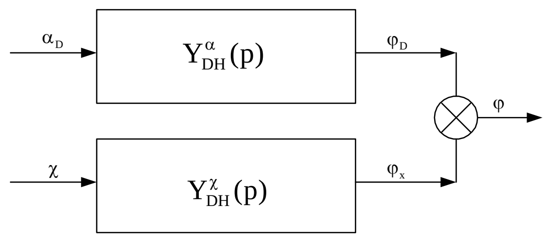

***Figure 2.** Simplified structural scheme of a turbocharged diesel engine*

III. COMPUTER SIMULATION MODEL FOR A MARINE DIESEL ENGINE

System dynamic mathematical model for a marine diesel engine

The system dynamic mathematical model for a turbocharged diesel engine can be defined by the expression (the explicit form of the equation 3):

$$\frac{d^2\varphi}{dt^2} = -\frac{d\varphi}{dt}\frac{T_{DH}}{T_H^2} - \varphi\frac{k_{DH}}{T_H^2} + \frac{d\chi}{dt}\frac{T_S}{T_H^2} + \chi\frac{k_S}{T_H^2} - \frac{d\alpha_D}{dt}\frac{T_u}{T_H^2} - \alpha_D\frac{k_u}{T_H^2}$$

(4)

where:

$d^2\varphi$= D2FI- acceleration of the angular speed of the engine crankshaft (KV)

d φ= D1F1- angular speed change rate of the engine crankshaft (KV) (speed gradient)

T_{DH}= TDH- time characterising the sluggishness of the engine as a regulated object

T_H^2= TH2- square of the time characterising the sluggishness of the engine as a regulated object

T_S= TS -time characterising the engine sluggishness

d χ =DKAPA- speed of moving the *rack lever* high – fuel pressure pump (VTS)

χ = KAPA- relative movement of the rack lever of the fuel pressure pump (VTS)

k_S= KS- coefficient of the engine gain

d α_D= DALFAD - change rate of the relative consumer load

α_D= ALFAD- relative change of the consumer load

T_u= TU -time characterising the generator sluggishness

k_u= KU - coefficient of the gain of the external engine load

φ= FI - relative change of the angular speed of the KV

k_{DH}= KDH - factor depending on the self-equalisation coefficient and engine gain coefficient

ALFAD - external engine load

SLOPE - subroutine of the first derivation (KAPA and ALFAD)

On the basis of the given mathematical model for a marine diesel engine, it is possible to design system-dynamic simulation models, i.e. mental-verbal, structural, flowchart and computer model for the observed reality of a marine diesel engine.

System-dynamic mental-verbal model for a marine diesel engine

Based on the described mental-verbal model, it is possible to make a structural diagram as shown in Figure 3.

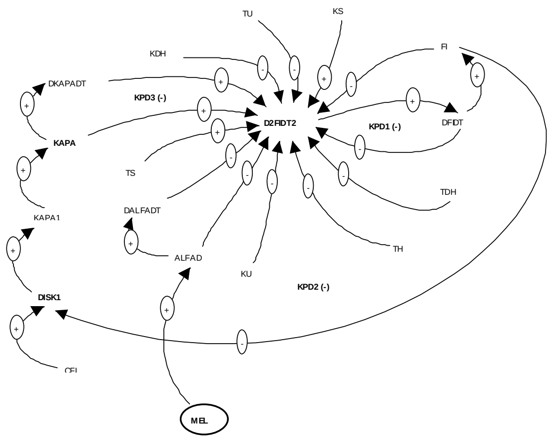

Figure 3. *Structural model for a marine diesel engine*

There are several feedback loops (KPD) in the observed system:
1. KPD1(-):D2FIDT2(+)=>DFIDT(+)=> FI=>D2FIDT2
2. KPD2(-):FI(-)=>DISK1(+)=>KAPA1(+)=> KAPA(+)=>D2FIDT2(+)=>DFIDT(+)=>FI
3. KPD3(-):FI(-)=>DISK1(+)=>KAPA1(+)=> KAPA(+)=>DKAPADT(+)=>D2FIDT2(+)=> DFIDT(+)=>FI

System-dynamic flowchart diagram of a marine diesel engine

The flowchart diagram has been created in line with the previously made mental-verbal and structural models, as shown in Figure 4.

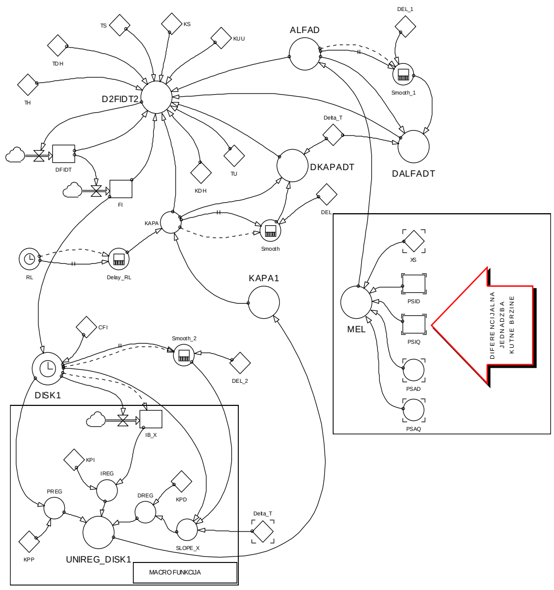

Figure 4. Flowchart diagram of a marine diesel engine with UNIREG regulator

IV. SIMULATION SCENARIOS OF THE MARINE DIESEL ENGINE MODEL

Simulation scenario of the start-up of an idling marine diesel engine - starting

The marine diesel engine is started at TIME=0 at zero load.

The diesel engine system is fitted with an electronic universal (UNIREG_DISK1) regulator. After a series of scenarios, we have obtained coefficients of the electronic universal PID regulator for the start-up of the idling marine diesel engine. The coefficients have been obtained with the aid of the heuristic optimisation. The latter implies the use of all familiar methods, including the ones that cannot be expressed in a mathematically exact way, so that expertise in modelling is necessary. Heuristic optimisation in system dynamics involves the "retry and error" method, a method that could be described as trying to obtain results gradually, i.e. "step by step".

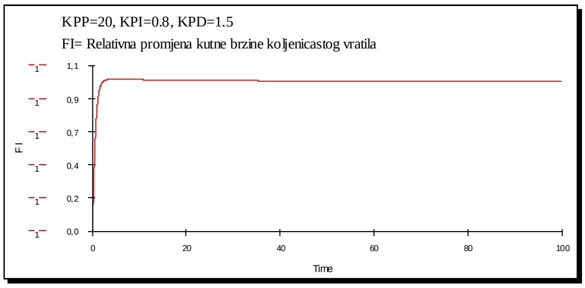

Figure 5. KPP=20, KPI=0.8, KPD=1.5

Relative change of the crankshaft angular speed

Comments on the obtained results from the Scenario I:

The behaviour dynamics of an idling marine diesel engine is entirely in accordance with its model, i.e. with the second order differential equation. The revolution speed regulation has been performed by the electronic universal PID regulator whose elements' coefficients have been changing. The heuristic optimisation of the PID regulator parameters has been carried out, resulting in the following combination of coefficients:

KPP=20, KPI=0.8, KPD=1.5

This combination entirely meets the criteria of the revolution speed of a marine diesel engine. The same results have been obtained in the Kongsberg hardware simulator at the University of Split.

Figure 6. Full mission engine room simulator Kongsberg ERS-L11 MAN B&W-5L90MC–VLCC Version MC90-IV

V. CONCLUSION

The presentation and the results of the simulation of the system-dynamic models for a marine diesel engine lead to the conclusion that applying System Dynamics, a modern scientific discipline, in the research of behaviour dynamics of complex ship systems is an exceptionally effective, rational, and prospective methodology. It is extraordinarily useful in education of both engineering faculty students and graduate engineers of all profiles as it provides a cost-effective, fast and accurate method of acquiring new insights and skills in the field of engineering systems and processes. This brief presentation gives to an expert all the necessary data and the opportunity to collect further information on the same system using a fast and scientific method of investigation of a complex system.

Which means:

Do not simulate the behaviour dynamics of complex systems using the research method of the "black box", because the education and designing practice of complex systems has confirmed that it is much better to simulate using the research approach of the "white box", i.e. the System Dynamics methodology.

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