Model Based Tuning of a Variable-Speed Governor for a Distributor Fuel-Injection Pump

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Rather obsolete, mechanical governors are still used in an enormous number of existing and even some new diesel engine injection systems. Change of a governor characteristic and adapting to a specific engine application is possible through adjustments of governor's elements. In order to achieve fast and accurate evaluation of adjustable parameters, mathematical model of a governor is introduced. This paper deals with the impact of adjustable parameters on static and dynamic characteristics of the mechanical variable-speed governor in DPA distributor type fuel injection pump. Dynamic testing of a closed loop Diesel engine – governor – injection system is conducted within comprehensive 1D multi-domain simulation environment.

Keywords: *diesel engine, speed governor, mathematical modeling, multi-domain simulation.*

1. INTRODUCTION

Basic Diesel engine control is firmly related to its speed governing. The actual engine working point is determined by equilibrium of, two mainly opposing torques, i.e. external load and effective engine torque. There are stable and unstable areas of engine behaviour within its working range, but in neither of these, it is impossible to maintain engine speed satisfactory without an appropriate engine speed governing system.

Diesel engines, with their moderate torque gradient, are particularly demanding in the engine speed governing. When they are exposed to frequently varying load, engine speed is subjected to wide fluctuation in the engine's stable part of working range. In the unstable part of the working range, it is almost impossible to maintain engine speed, especially in the area of low engine speed and load.

A purely mechanical engine speed governing system balances fuel metering device position in order to maintain engine speed, regardless of the engine load variation which causes the engine speed disturbance. There are several ways for the governor to sense engine speed disturbance and either of approaches is based on the engine crankshaft speed, acceleration or engine load variation detection. According to the selected measured variable(s), it is possible to design a variety of indirect or direct-acting speed governing systems [1].

Modern diesel engines are usually equipped with a sophisticated electronic control system (*Electronic Diesel Control – EDC*) based on a microcontroller coupled with sensors and electromechanical actuators as a typical mechatronic approach solution [2]. EDC concept has a number of advantages and definitely becomes common because of its capability to deal with a fast wider spectra of problems in diesel engine control than speed governing itself.

On the other side, the fact is that there is a huge amount of diesel engines equipped with classical mechanical speed governors. Majority of them is already in exploitation, but lots of them are still coming out from production lines. Compared to EDC, mechanical speed governing systems are far more limited in their ability to adapt to different engines and applications. A given type of mechanical governing system has a limited number of adjustable parameters which, optimized, can be adapted to specific engine and its demands.

Adjustment of these parameters is, unfortunately and very often in practice, realised through engineer's or technician's experience, skill and "feeling" – not with the engineering determinacy as it should be. The result of this type of approach is often an unsatisfactory functioning speed governing system. A wrong setup of the governor is sometimes manifested not so obviously. It can cause higher fuel consumption, for example.

The basic idea presented in this paper is to create a simple but effective mathematical model of a mechanical speed governing system. Solved numerically, the model can give exact values of adjustable parameters. Furthermore, the model can be used as an engineering tool for investigating the influences of various parameters of the governor's design on its functioning.

2. THE MATHEMATICAL MODEL OF THE DPA TYPE MECHANICAL SPEED GOVERNOR

This paper deals with the mechanical governor implemented in the DPA (*CAV–Lucas*) type high pressure fuel injection pumps which are, under license, produced by IPM (*Industrija Precizne Mehanike, Belgrade*), (Fig. 1).

This type of governor is direct acting, and its main input variable is the pump's driveshaft rotational speed. The governor senses, actually, a half of engine's speed

Received: December 2012, Accepted March 2013
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doi:10.5937/fmet1401040M

since the driveshaft (1) is directly coupled with the engine's crankshaft with the ratio 1:2. Governor's sensing elements are flyweights (2) which can swing in the plane containing the driveshaft. Flyweights' centrifugal force is acting upon the sliding sleeve (3) and the control arm (4) which is balanced by the tension force of the control spring system consisting of a main (7) and idle spring (8). Fuel metering device is coupled with the control arm, so the swinging of the flyweights reflects on the amount of fuel metered to the engine [3]. The Governing control loop has a negative feedback and therefore, maintains a chosen engine speed, regardless of engine load variation.

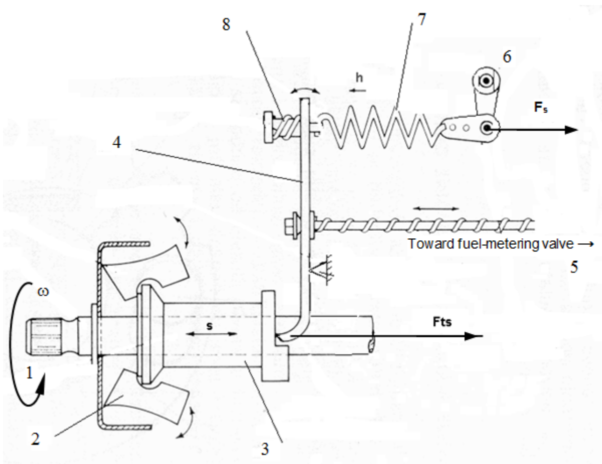

Figure 1. The variable speed governor of DPA type rotary pump - main variables of the governor's mathematical model.

In the variable-speed type governor, the desired speed regime is set up by positioning the control lever (6). This position determines the tension force level in the control spring, and therefore the level of centrifugal force (i.e. driveshaft speed) needed to activate the governor.

2.1 Governor's differential equations

Mathematical model is formed as a system of differential equations linking governor's input values (driveshaft speed and control lever position) with its output – fuel metering device position. Since all governor's elements are linked, kinematical and all linkages, except springs, are assumed to be indefinitely stiff, so it is more convenient to introduce some new, auxiliary variables as model inputs/outputs. Instead of control lever position, the control spring deformation h is used, and fuel metering device position is replaced with the position of the control sleeve s.

The first step in forming differential equations is a derivation of reduced inertial properties of governor's moving elements. The appropriate reduction point common for all elements could be some point on the control sleeve since it has a simple linear motion, which should be also the solution - output of the differential equation system.

When reduced, the force in the governor spring takes the following form:

$$F_{ss}(s,h) = c \cdot \left(u_p \cdot h + s \cdot c_{red}\right). \tag{1}$$

where c and c_{red}, respectively, are stiffness and reduced stiffness of governor spring, and u_p is lever ratio of the control arm.

Applying the energy conservation law, the flyweight centrifugal force, reduced to the sliding sleeve, takes the following form:

$$F_{fs}(\omega,s) = \frac{1}{2} \cdot i \cdot \omega^2 \cdot \frac{dJ}{ds}. \tag{2}$$

where i is the number of flyweights and J is the displaced moment of inertia (to the axis of rotation – driveshaft axis).

Governor's elements are also influenced by friction forces. The resulting friction force f_f is derived by summing dry f_{df} and viscous f_{vf} friction. According to D'Alembert's principle, it is possible to derive the differential equation of motion of the governor's elements:

$$m_R \cdot \frac{d^2 s}{dt^2} + \nu \cdot \frac{ds}{dt} + F_{ss}\left(s,h,t\right) - \\ - F_{fs}\left(\omega,s\right) \pm f_{df} = 0. \tag{3}$$

where m_R is the total, reduced mass of all movable governors' elements and ν is the viscous friction coefficient.

At the stationary regime, forces, acting upon governor's elements, are balanced and therefore, (3) takes the following form [1]:

$$F_{ss}\left(s,h,t\right) - F_{fs}\left(\omega,s,t\right) \pm f_{df} = 0. \tag{4}$$

The forces F_{ss} and F_{fs} are mainly influenced by variables s, h and ω. The forces of hydraulic friction are caused by friction between moving elements of the governor (dominantly flyweights) and the diesel fuel which surrounds them, since the governor's housing is completely filled with the fuel. Assuming that the fluid (diesel fuel) within the pump is static, viscous friction force can be described as follows:

$$f_{vf} = \nu \cdot \frac{ds}{dt}. \tag{5}$$

where differential ds/dt is the velocity of the sliding sleeve. Equation (5) correlates the viscous friction force and sliding sleeve velocity strictly linearly with ν as a coefficient of proportion, which can be experimentally determined. With this approach, fuel within the pump is introduced in the mathematical model as a damping element. Dry friction forces can be represented by a single force as:

$$f_{df} = F_{fs} \cdot \mu. \tag{6}$$

where μ is a friction coefficient which correlates the force level, within the governor mechanism, with the single, general dry friction force.

Summation of masses of all moving governor elements results in total reduced mass of governor:

$$m_R = \sum m_{Ri}. \tag{7}$$

Elements, whose mass is contained in this sum, are governor flyweights, control arm, fuel metering valve, governor spring(s), control arm and linkage hook with its spring. The reduction of masses is accomplished by applying the law of conservation of energy:

For linear motion:

$$\frac{1}{2} \cdot m_{Ri} \cdot v_s^2 = \frac{1}{2} \cdot m_i \cdot v_i^2 . \quad (8)$$

For rotational motion:

$$\frac{1}{2} \cdot m_{Ri} \cdot v_s^2 = \frac{1}{2} \cdot J_{oi} \cdot \omega_{oi}^2 = \frac{1}{2} \cdot J_{oi} \cdot \left(\frac{v_s}{l_i} \right)^2 . \quad (9)$$

Variables used in (8) and (9) are:

m_{Ri} reduced mass of i-th element,
m_i mass of i-th element,
v_s sliding sleeve velocity,
v_i i-th element linear velocity,
J_{oi} i-th element moment of inertia (related to the axis of rotation),
l_i the minimal distance between the axis of rotation of the i-th element and a point with the sliding sleeve's velocity.

Inertial and mass properties of governor flyweights are derived from 3D solid CAD model. During reduction of the spring mass, its kinetic energy is also taken into account. It is assumed that the velocity of spring segments changes linearly from the still through the opposite movable end of the spring.

In order to simplify the numerical solving of the differential equation (3), it is transformed into the system of two, first-order differential equations. The unknown variable s is replaced with the variable α which traces the angular movement of the governor flyweight (in the plane containing the axis of the pump shaft). This introduces new, auxiliary variables $y_1=\alpha$ and $y_2=\dot{\alpha}$.

After the transform, the system of equations takes the following form [4]:

$$\begin{aligned} y_1' &= y_2 \\ y_2' &= {y_2}^2 \cdot \tan y_1 - \frac{v}{m_R} \cdot y_2 - \\ &-\left(F_{ss}(s,h,t) - F_{fs}(\omega,s,t) \pm f_{df}\right) \cdot \frac{1}{m_R \cdot R \cdot \cos y_1} \end{aligned} \quad (10)$$

Variable R represents the length of the flyweight's lever arm (between its fulcrum and sliding sleeve).

Since (3), i.e. the system (10), is describing the movement of elements within the governor independently of the engine, it is required to supply the system with the engine speed vector as an input. The second input required is the position of the control lever, i.e. the deformation h of the control spring. Since the model output is actually the metering-valve position, it is possible to incorporate this model, as a contribution, to much more sophisticated and more complex models dealing with the engine cycle simulation and the simulation of the fuel-injection system hydrodynamics.

The system of equations (10) is solved by the Runge–Kutta method. Efficiency in solving the system (10) is dependent on the chosen solving options and, more dependent on nature of input variables (h, ω), i.e. by introducing more dynamics in inputs the whole system behaves more as a stiff one.

3. IDENTIFICATION OF UNKNOWN PARAMETERS OF GOVERNOR'S MATHEMATICAL MODEL

In order to identify unknown parameters and to verify mathematical model of the governor, test bench has been developed, capable for dynamic testing of a speed governor.

Input variables of the model are numerous governor design parameters, tension level of the governor spring and engine speed, whereas the output variable is the position of the sliding sleeve. According to the nature of the model variables, test test bench (Fig. 2) is equipped so that it can be capable to:

- Drive the high pressure pump / simulate the IC engine driveshaft
- Measure instantaneous drive shaft speed
- Measure position of the sliding sleeve (which is directly coupled with fuel metering-valve)
- Measure tension level of the governor spring

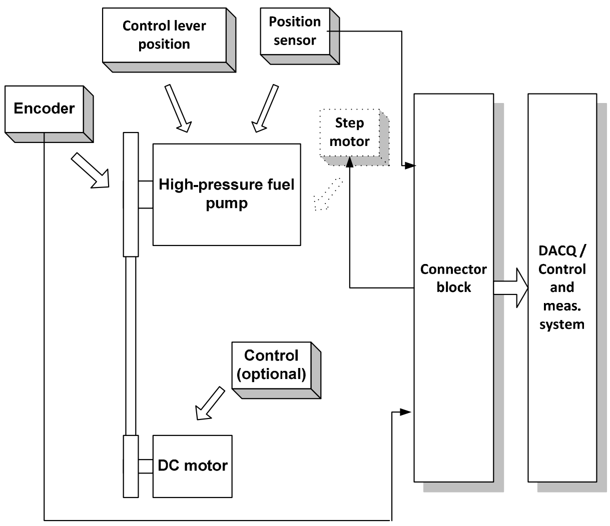

Figure 2. Schematic of the test bench installation for the dynamical testing of the speed governor

The measurement of the slide sleeve position has been realized by means of the Hall-effect linear position sensor. The sleeve position measurement system is calibrated dynamically by custom-made apparatus based on a high resolution, step motor drive used as a slide sleeve actuator during calibration. Calibration curve is derived as an average from 20 series of measurements with 1700 discrete positions within each of them. Furthermore, each discrete position within series is measured successively 500 times and then averaged.

Angular speed of the drive shaft is measured by 3-channel incremental optical encoder (1024 CPR). Data acquisition and synchronous control of the step motor during position measurement system calibration are achieved by using a modular PC based acquisition system ED 2000 (actually *Intelligent Instrumentation* 20001C series system). Since that acquisition system used, had no DMA controller, maximum acquisition performance was achievable only through real time OS. QNX Neutrino RTOS has been selected for that purpose, and acquisition software (including low level hardware drivers [5], [4]) has been developed.

Identification of the hydraulic damping coefficient ν is achieved by the analysis of forced oscillations of sliding sleeve on a governor's stationary speed regime, as suggested in [1]. Measurements and data analysis showed that the damping coefficient has a value of $\nu = 27$ Ns/m. By employing the Levenberg-Marquardt algorithm based optimisation technique, the reduced total dry friction coefficient has been also identified, as $\mu = 0.07$.

Figure 3 shows a comparison between simulated and measured governor characteristics by control spring tension level of h=4 mm and varied governor's shaft speed ω_p. Very good agreement results are also achieved for other governor operating regimes (other levels of control spring tension) [4].

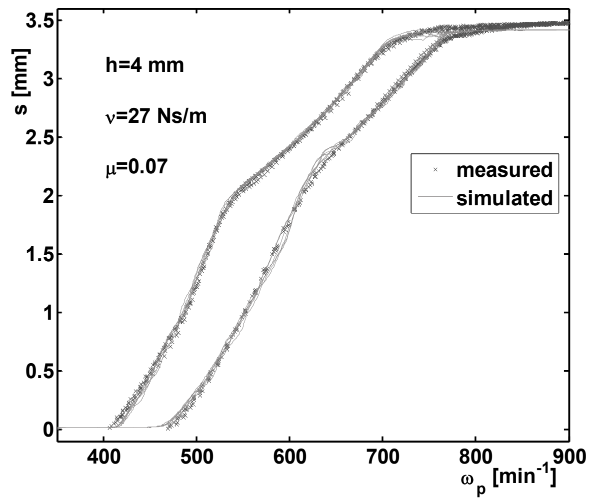

Figure 3. Measured and simulated characteristic of the governor – a comparison.

4. STATIC CHARACTERISTICS OF THE GOVERNOR

Variable speed governor incorporated into the DPA type fuel injection pump has several adjustable parameters:

- 3 possible anchoring points of the main control spring on the control arm (parameter u_p – top, middle and bottom spring anchoring points)
- 3 possible anchoring points of the main control spring on the control lever side (parameter h_0)
- Replaceable main and idle control spring (stiffness values of c_{main} and c_{idle})
- Various possible number of flyweights (parameter i_{flyw}=2,4 or 6)

This basically gives, 27 possible parameter combinations (with already chosen idle and main control spring). Evaluation of governor's variable parameters influence on its characteristics can be achieved by the analysis of two additional, mathematically derived, parameters. This is a very convenient way of describing the quality of governor functioning from the aspect of its stability and capability to maintain the engine speed within the predefined limits. These two parameters are defined as the factor of stability C_p and the factor of speed inequality δ [1]:

$$C_p = \frac{\partial F_{ss}(s,h)}{\partial s} - \frac{\partial F_{fs}(\omega,s)}{\partial s} . \tag{11}$$

$$\delta = \frac{\omega_{max} - \omega_{nom}}{\omega_{mean}} . \tag{12}$$

The factor of stability C_p is numerically defined as a difference of gradients of the two dominating forces opposing each other within the governor: the flyweights' centrifugal force F_{fs} (ω, s) and the control spring tension force F_{ss} (s, h).

A positive value of C_p, within a wide range of the governors working parameters (rotational speed of the shaft and flyweights' angular position) implies that the stationary regimes of the governor are highly stable.

Factor δ describes the amount of the shaft speed increase (in percent) needed, to cause the swing of the flyweights from their minimal (nominal) to their maximum position [1].

Figure 4 (top) shows how the parameter δ changes and depends on the position of the control lever (i.e. tension force in the control spring expressed via its deformation h). Each level of tension force F_{ss} (s, h) in the control spring gives a new pair of driveshaft speeds: $\omega_{\min}$ and $\omega_{\max}$. The governor engages by reaching driveshaft speed $\omega_{\min}$. With the speed increase, flyweights are swinging outwards, and they reach their maximum position at $\omega_{\max}$ (Fig. 4, bottom). The driveshaft speed ω_{mean} is an average of these two extremes.

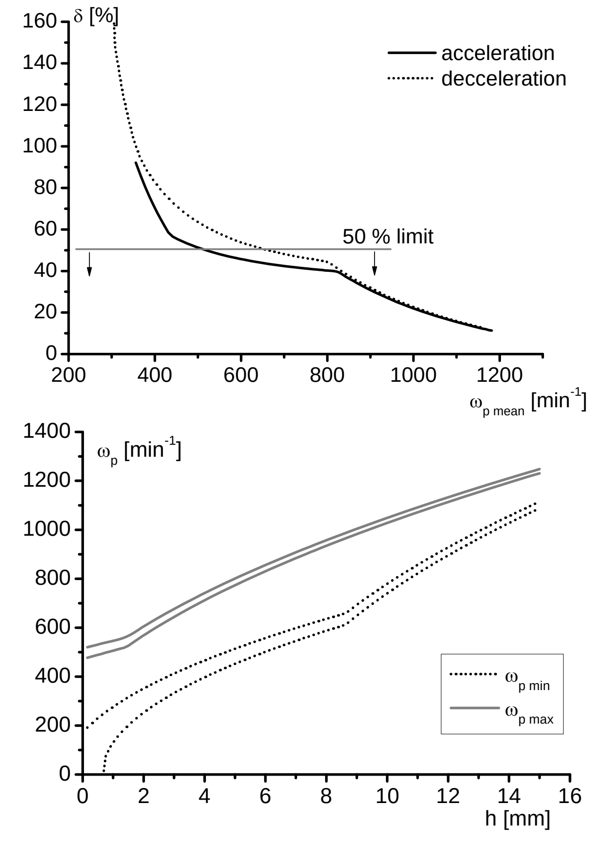

Figure 4. Driveshaft speed limits (bottom); Inequality factor δ (top); Hysteresis as an effect of friction losses.

The change in gradient, visible in Figure 4, indicates a sudden change of the spring system stiffness which appears when idle spring reaches its full compression.

A relation *ω – δ,* shown in Figure 4, is simulated with the following set of the governor's parameters: u_p=2.08 (control arm top hole)*;* c_{main}=1.95 N/mm; c_{idle}=1.14 N/mm; Δ_{idle}=5.45 mm; i_{flyw}=6;

As a rule of thumb, factor δ should not be greater than 50% at the lowest engine speed [1], since this limit ensures governing quality of the chosen speed regime. It is obvious that, for the set of chosen governor's parameters, this limit is exceeded already at a driveshaft speed of 590 rpm. Having in mind the engine's crankshaft – pump driveshaft speed ratio, the conclusion is that the governor with this setup has satisfactory regime regulation only above engine's 1180 rpm. When friction forces are taken into account curves, discussed above, take awider form (Fig. 4) with visible hysteresis included. With this, more realistic approach, δ_{50} limit is moved above engine's 1310 rpm. The governor's setup, used in this example, is taken from the *IPM* production line pump, used on *Perkins* 4.203 type tractor engines (a.k.a. DM34 produced by domestic *IMR*) with the working range between 1000 and 2450 rpm.

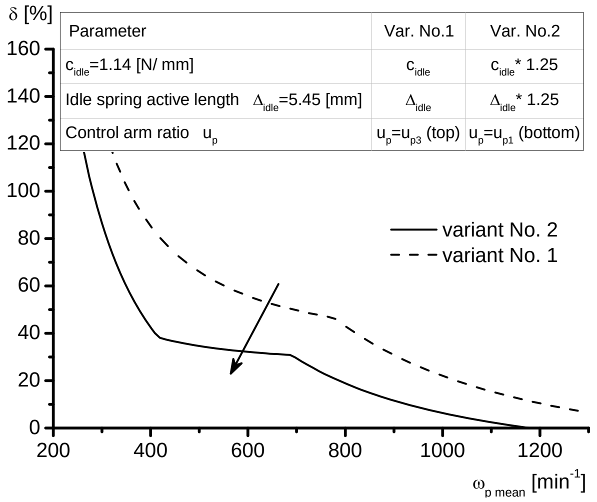

Figure 5. Comparison of two different governor's setups; goal: lowering the δ over entire engine speed range

Each of the governor's adjustable parameter influences its behaviour and, consequently, the factor *δ*. Variations of idle spring stiffness c_{idle} , idle spring active length Δ_{idle} and control arm ratio u_p can influence the factor *δ* considerably, which is shown in detail in [4]. Furthermore, carefully chosen combined adjustment of these parameters can improve overall engine speed governing capabilities.

When the operating engine speed limits are already known, the model-based analysis can be very helpful in choosing the governor's parameter towards the optimum ones. Figure 5 shows the comparison of two different governor's setups. The goal of this variation was to lower the speed inequality factor δ over the entire engine speed range. The comparison shows that it is possible to significantly lower the factor δ only by changing the idle spring and its anchor point on the control arm. Low value of the factor δ does not necessary mean that the governor's behaviour is also improved. The example, shown in Figure 5 demonstrates a drop of the factor δ, at higher speeds, so low that the governor is almost facing with the astatic limit which can have negative consequences in governing the engine's higher speed regimes. Being astatic simply means that governor lose its capability to react to engine speed change. In order to improve δ only in the lower speed range it is more than enough, in this example, to change the idle speed spring stiffness and active length only.

A study on the governor's behaviour through a model based analysis of its features, like C_p or δ factors, treats the engine speed governor as an open-loop control system. A control loop on the real engine is much more complicated and includes the described governor, high pressure fuel-injection system and diesel engine itself. In order to deepen the analysis and emphasize the fact that conclusions derived can already be applied on the real engine system, a simulation of the whole system is needed.

The output of the governor system is directly applied to the actuation of the fuel metering device. Through complex non-linear processes in the high-pressure fuel-injection system a fuel metering device position is translated to the amount of fuel delivered to the engine. On the other hand, engine speed is influenced not only by injected fuel amount but also with many other factors, dominated by the engine load conditions. The dynamics of the engine, specially turbocharged ones, plays a significant role in the overall behaviour of the closed-loop governor-fuel injection- engine system.

5. COUPLED CLOSED-LOOP SIMULATION

The closed-loop simulation of the diesel engine and speed governing system, in this paper, is based on the co-simulation of several complex mathematical models gathered in Simulink™ environment:

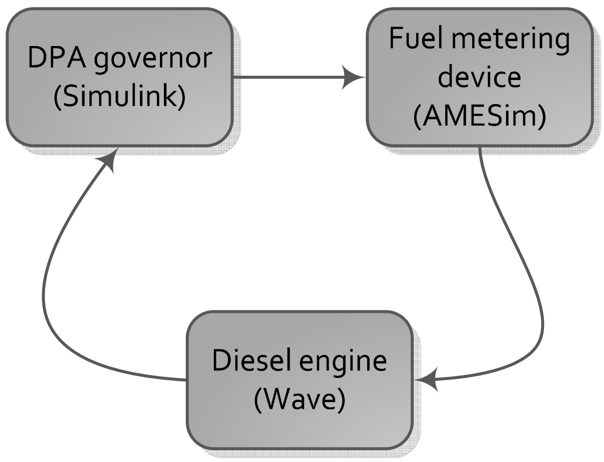

Figure 6. Multiple model co-simulation concept for testing a complex multi-domain physical systems.

This concept enables a fast and simple initial testing of complex systems with processes spreading over multiple physical domains. In this particular example, a complete Diesel IC engine closed-loop controlled model is described by sub-models covering combustion thermodynamics, heat transfer and gas flow phenomena, complex hydraulics and mechanical assembly's dynamics.

5.1 IC Engine model

Simulation of the internal combustion engine working cycle was performed using Ricardo WAVE™ simulation software package. The simulation model used within the WAVE™ simulation software is a 1D model which includes the sub-models of various phenomena occurring in real engine working cycle [6]. The model used, represents the flow through the engine as a flow network of quasi-one-dimensional compressible flow equations. The flow network is meshed into the individual volumes which are connected via energy, momentum and mass conservation equations and boundary conditions. The combustion model used is based on the well-known simple Wiebe heat release function within a single-zone combustion chamber. The heat transfer sub-model used is based on the Woschni model with simplified approach assuming a uniform heat flow coefficient on all cylinder wall, cylinder head and piston surfaces. The same heat transfer model setup is applied to all cylinders. Estimation of the mechanical friction losses is achieved by means of modified equations of an empirical Chen-Flynn model which calculated the overall friction losses by calculation of terms dependent on peak in-cylinder pressure, mean piston velocity and term, which takes into account the driving torque of the auxiliaries.

Inertia and dynamics of the turbocharger shaft and the engine mechanism are modelled within the main Simulink modelling environment. In order to couple the engine thermodynamic process model with the rest of the modelled system, the component called "wiring connector" is used, which provides the capabilities for transferring outgoing parameters like engine and turbocharger torques and accepting engine speed and fuel flow controlling parameters as inputs from the Simulink™ environment.

The overall capabilities of the WAVE™ simulation software provided a realistic simulation of a general turbo charged diesel engine processes – since the specific engine data are irrelevant for this type of testing. The schematic of the WAVE™ engine model is shown in Figure 7 (left).

5.2 Fuel metering valve model

The amount of fuel injected into the cylinder of a diesel engine depends on many factors, and it is influenced by various complex processes in the fuel-injection system. In a DPA type fuel injection pump, fuel is metered in the low-pressure part of the system, and therefore the focus is put on this part of the fuel-injection system model, with a very crude assumption that the fuel metered is completely transferred through the high pressure subsystem to the engine cylinder.

The fuel metering device, used in the DPA type fuel injection pump, is actually a rotational spool valve with a slot orifice. In order to take into account the nonlinearities introduced by the flow phenomena on this element, and to properly estimate the relation between the position of the metering device and the fuel flow through it, a hydraulic assembly mathematical model is created using LMS AMESim™ multi-domain system modelling environment. This 1D model contains the low pressure, regulated, transfer fuel supply pump which is feeding the metering device and control valve itself.

The model of the valve hydraulic element calculates the effective flow area and the basic flow rate of the diesel fuel through the valve orifice slot. The flow type is dependent on the valve opening and the switch is monitored by integrated function, which ensures the continuity in the flow through the valve [7]. The schematic of the AMESim™ based model is shown in Figure 7 (right).

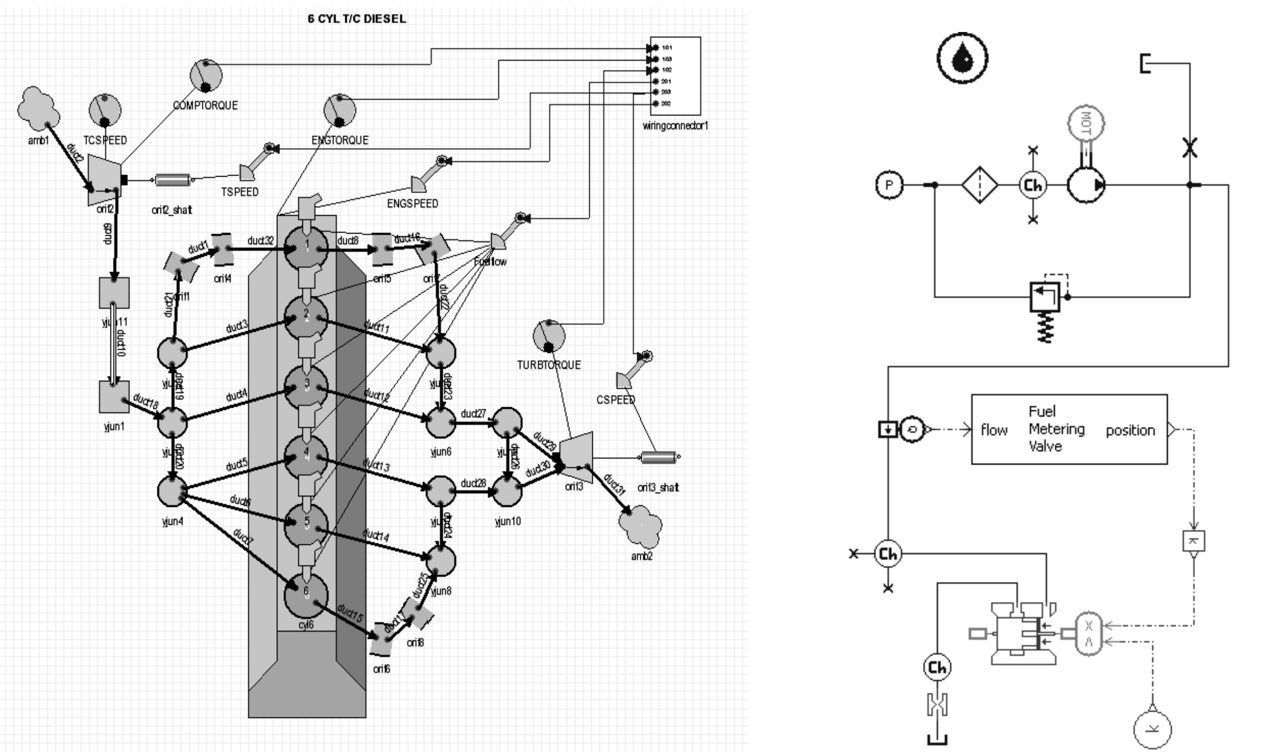

Figure 7. Schematics of the Diesel engine submodel in WAVE™ (left) and fuel metering device in AMESim™ (right)

5.3 Simulation results

The simulation of a closed-loop diesel engine system equipped with the described mechanical type governor is conducted through the a series of numerical experiments. The initial torque demand, from the engine, is varied in a predefined sequence which is shown on the top part of Figure 8. A variation in the load torque causes the engine speed variation and consequent reaction of the speed governor. The governor, tested in the simulation, had two setups: basic and optimised version (with respect to parameter δ), as explained in Figure 5.

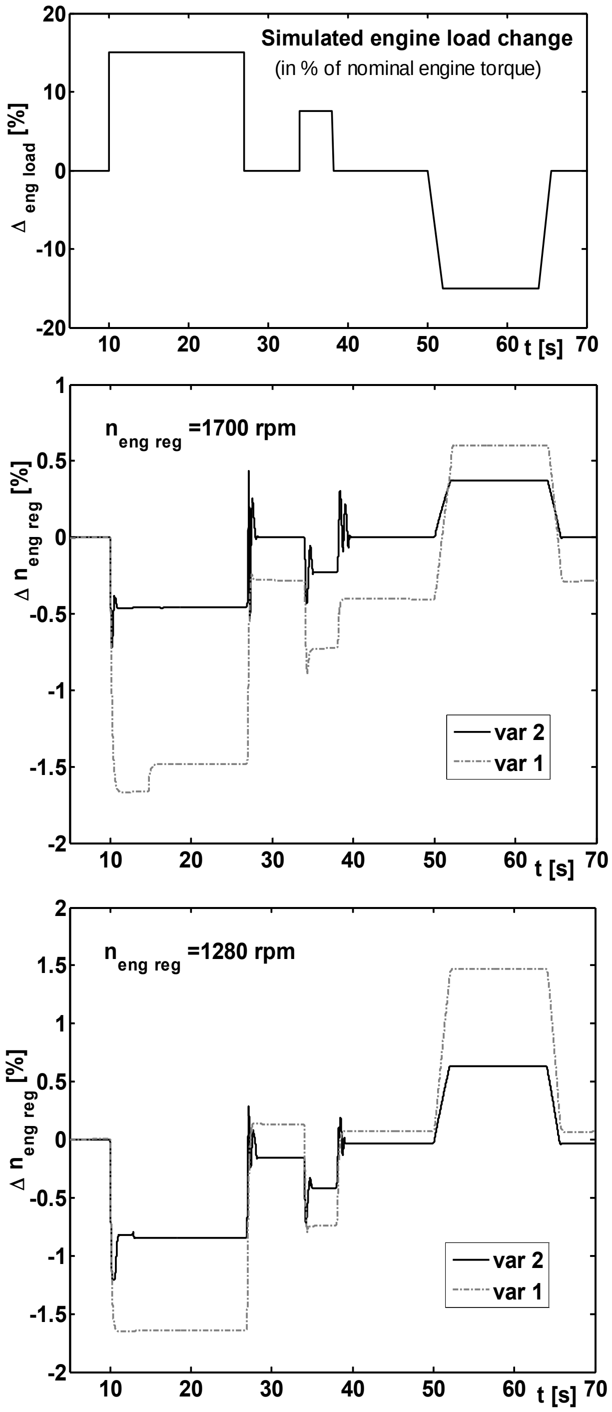

Figure 8. Comparison of two different governor's setups (refer to Fig. 5); Governed engine speed – 1700 rpm (middle) i.e. 1280 rpm (bottom); Engine load torque variation sequence (top).

Results, shown in the Figure 8, present the governor response for two distinctive engine speed regimes: 1280 and 1700 rpm. The load variation, used in load excitation sequence in the Figure 8, is exemplary and its amplitude is not crucial in deriving the conclusions on the governor setup behaviour.

As presumed, from the δ parameter analysis, governor setup designated as "var. 2" has a better performance in terms of engine speed fluctuation on both engine speed regimes. In general, performance gain (in scope of δ) is achieved by elevating the level of equilibrium forces within the governor mechanism. This setup provides also smaller hysteresis in maintaining the predefined engine speed, and this is clearly noticeable on the higher engine speed simulation (Fig. 8, middle): after load torque change, back to the initial value, the engine speed is settled on the almost identical, initially set level.

With the basic setup ("var. 1"), the engine speed is always "near" but visible distanced from the set level, despite the load torque change to its initial value. At lower engine speeds, this hysteresis decrease virtue of the "var. 2" setup vanishes. It is also noticeable that the speed governance performance, at lower engine speed, is of inferior quality because of the parameter δ rise with the engine speed drop, in general (Fig. 4). However, the advantages of the parameter δ lowering, at this speed range, are also evident.

It is also noticeable that higher level of equilibrium forces, within the governor mechanism, causes under-damped response to step load torque transient.

6. CONCLUSIONS

In the era of electronically controlled diesel engines, mechanical speed governors seem obsolete, but there are still a large number of engines in exploitation equipped with this kind of devices. Some of them are still coming out from production lines. The high-pressure diesel fuel injection pumps equipped with direct acting mechanical speed governors have some advantages and are still dominating on some markets in the world.

Adjustment of the governor's variable parameters manually and its adaptation to the specific engine type is a time-consuming process. Mathematical model, presented in this paper and created simulation software has shown its usability as a valuable tool for shortening this process and, furthermore, as a tool which can be used for deriving a lot of conclusions in the process of a governor redesign or diagnostics.

Open-loop simulation of the mechanical governor, described in this paper, gives valuable and usable results. Despite this simplification, this approach enables comprehensive analysis of variable parameters influence on governor's behaviour, which is demonstrated on a complex closed-loop governed diesel engine simulation.

In order to conduct the dynamical testing of the mechanical governor, an appropriate test bench has been designed. Measurements were conducted using of a digital acquisition system under QNX Neutrino Real time OS with in-house developed acquisition software and DAQ hardware drivers.

Simulation results of the calibrated governor's mathematical model lead to the following conclusions:

- The model is capable of providing accurate predictions of governor's variable parameters influence on its performance and its characteristics.
- Simulation software, based on the described mathematical model, can be a valuable tune-up and failure diagnosis tool.

Variable speed governor implemented in the DPA high-pressure fuel pump belongs to the type of the simplest direct-acting mechanical governors. Despite its simple design, the performance of this governor can be still satisfactory, and it has a very wide adaptation capability to a variety of engines. Relatively large hysteresis can be improved by raising the balanced forces, within the governor, to a higher level [4].

The presented model of a DPA type governor also explains why this type of governors cannot provide even performance over the entire engine speed range thus emphasizing the benefits of mechatronic approach of engine speed governing, implemented through EDC systems.

ACKNOWLEDGMENT

Results presented in this paper are parts of the activities conducted on the project "Research and development of mechanically and electronically controlled high-pressure fuel injection systems for diesel engines" granted by Ministry of Science and Technological Development of Republic of Serbia (MIS. 3.06.0160.B/1).

This paper is also written in the memory of late Professor Dr. Miloš Cvetić whose valuable ideas and support vastly contributed to the abovementioned project and its results.

Real-time data acquisition would not have been possible without the assistance from QNX Software Systems and their granted license of QNX Neutrino RTOS with QNX Momentics development system.

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МОДЕЛСКИ ЗАСНОВАНО ПОДЕШАВАЊЕ СВЕРЕЖИМСКОГ РЕГУЛАТОРА ДИСТРИБУТОР ПУМПЕ ЗА УБРИЗГАВАЊЕ ГОРИВА

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Иако превазиђени, механички регулатори су још увек у великој мери заступљени као део постојећих, па чак и нових система за убризгавање горива код дизел-мотора. Механички регулатори се намени прилагођавају помоћу подесивих елемената којима се дискретно мења карактеристика регулатора. Ради што једноставнијег подешавања и свеобухватног сагледавања утицаја појединих параметара на регулаторску карактеристику, формиран је и представљен математички модел регулатора. Рад се бави анализом утицаја подесивих параметара на статичке и динамичке карактеристике свережимског регулатора дистрибутор пумпе типа DPA. Анализа динамичког понашања регулатора спрегнутог са мотором, у затвореној петљи, вршена је у оквиру свеобухватне једнодимензионалне вишедоменске симулације.