Development of Diesel Engine Simulator for Use with Self-Propulsion Ship Model*

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Growing interest in the numerical simulation of total ship propulsion plant demands an easy to handle diesel engine simulation tool in a field which is as extensive as there are applications – every design process or research project has its own fit-for-purpose model. Moreover, engine simulation is a trade-off between model simplicity, execution time, and accuracy. In this paper attention is focused on the development of a diesel engine dynamics simulation model which combines fast-simulation time, sufficient accuracy, and limited engine data requirement. The cycle mean value modeling methodology is adopted in this paper and the derived results were validated against experimental data of prototype engine.

1. Introduction

Energy consumption is increasing rapidly due to growth of world fleet thus the exhaust gas emission increases and contributes to global warming. The annual CO₂ emission, from the diesel engines used in shipping industry accounts for 3% of total global emissions. The International Maritime Organization (IMO), responsible for international regulations for shipping, adopted in July 2012 the industry's first regulations toward CO₂ emission reduction. Furthermore, strict emission control for environmental pollutants such as NO_x and SO_x is also a major challenge, and some regulations have already been introduced. Hence technological improvement in ship design and operation, to solve global environmental problems is one of today's major international issues.

In order to measure the technological improvement of a new ship towards energy efficiency and air pollutants emission, Marine Environmental Protection Committee (MEPC) has introduced the Energy Efficiency Design Index (EEDI) through a new Chapter 4 to MARPOL Annex VI entered into force on 1 Jan 2013. The evaluation of EEDI is based upon not only design characteristics of a ship but also her performance in actual sea condition. In that way it is indispensable to assess overall ship efficiency in actual sea condition at design stage, whereas it is commonly evaluated in the calm sea due to the absence of practical experimental

methodology. In recent time, however, this problem has received much research attention resulting in developing various numerical modeling approaches of overall ship propulsion plant. Among them the most interesting and promising one is a self-propulsion ship model driven by a diesel engine simulator [1] which combines physical ship model with numerical model of propulsion engine and allows to realize dynamic response of ship propulsion system in real sea condition. Beyond all doubt provision of detailed model of marine diesel engine is important for successful development of this experimental methodology.

The objective of this paper is to present a comprehensive but easily handled marine diesel engine simulator, suitable for use in the framework of self-propulsion model of a ship

2. Formulation of Propulsion Plant Dynamics

A generic ship propulsion plant can be considered to be made up of three main components: the engine, propeller and ship hull, which coupled dynamics, in general case, include equation of forces balance acting on a hull:

$$M \frac{dv}{dt} = T_p - (R_c + R_{aw}) \quad (1)$$

and equation of torques balance applied on a propeller shaft:

$$2\pi I_e \frac{dn_e}{dt} = Q_e - Q_p, \quad (2)$$

where M is the mass of ship, v is the advancing speed of ship, T_p is the propeller thrust and $R_c + R_{aw}$ is the total resistance acting on a hull in actual seas, I_e is the moment of inertia of whole shaft line, n_e is the engine rotating speed, Q_e and Q_p are the engine and propeller torques,

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respectively and dt denotes the time derivative.

The diesel engine produces an output of torque from input of injected amount of fuel with its speed determined by the point of balance between engine torque and propeller torque. Nevertheless if the transients being simulated are severe enough, the air amount supplied for fuel combustion delays from fuel supply and affect torque generation, and this phenomenon must be taken into account in the engine model. In that respect not only air flow through the engine must be simulated but the turbocharger, which is an integral part of the engine, must be modeled. This fact results in addition of turbocharger dynamics with respect to its rotational speed n_{tc} , expressed as follows:

$$2\pi I_{tc} \frac{dn_{tc}}{dt} = Q_t - Q_c, \quad (3)$$

where I_{tc} is the moment of inertia of turbocharger, Q_t and Q_c are the turbine and compressor torques, respectively. Last but not least essential part of any modern propulsion engine is a governor which aims at keeping engine speed constant by controlling the produced torque via the amount of fuel being injected. Therefore equation of the speed governor dynamics must be added. One may use a generic PID governor [2] or a conventional hydro-mechanical governor [3], however the detailed consideration of governor performance is out of scope of this paper. Simulation of ship propulsion plant dynamics requires an explicit mathematical formulation of every component constituent of the above mentioned equations, however since this study deals solely with engine modeling, the focus will be put on formulation and evaluation of the engine, turbine and compressor torques

3. Prerequisites for Engine Modeling

Nonetheless of varying complexity of the engine simulation methodologies and of absence one universal mathematical model of a diesel engine, all models root in a common ancestry coming from fundamental principles of internal combustion engines operation.

Thus engine brake torque is torque produced at the cylinder as a result of fuel combustion, less the torque absorbed by friction and torque of pumping stroke (typical only for four-stroke engine). The engine brake torque is therefore represented by:

$$Q_e = Q_i - Q_{fr} - Q_p. \quad (4)$$

The engine indicated torque Q_i is directly proportional to an indicated mean effective pressure (imep) developed in the engine cylinder:

$$Q_i = \frac{z i V_s}{2 \pi} P_i, \quad (5)$$

where z is the number of engine cylinders, i stands for number of cycles per crankshaft revolution, V_s is the engine cylinder displacement volume, P_i is the imep.

The friction torque Q_{fr} is assumed to be a function of engine speed [7]:

$$Q_{fr} = \frac{z i V_s}{2 \pi} \left[k_{fr2} \left(\frac{n_e}{N_{e0}} \right)^2 + k_{fr1} \left(\frac{n_e}{N_{e0}} \right) + k_{fr0} \right], \quad (6)$$

where n_e is the engine speed, N_{e0} stands for engine speed at maximum continuous rating (MCR) of an engine, constants k_{fr2} , k_{fr1} and k_{fr0} are provided either by engine manufacturer or by available operational data of engine.

Henceforth a subscript zero denotes the value of parameter at MCR of the prototype engine.

The pumping torque Q_p is modeled using the charge air P_s and exhaust gas P_e manifold pressures:

$$Q_p = \frac{z i V_s}{2 \pi} (P_s - P_e). \quad (7)$$

The quasi-steady operation of turbocharger is characterized by the isentropic work of gas expansion in turbine and isentropic work required for air pressure increase in compressor. These works appropriately modified by the efficiencies η_{ic} and η_{it} , yield the expressions for compressor and turbine torques, respectively:

$$Q_c = \frac{C_{pa} T_a \dot{G}_a}{2 \pi n_{tc} \eta_{ic}} \left(\left[\frac{P_s}{P_a} \right]^{\frac{k_a-1}{k_a}} - 1 \right), \quad (8)$$

$$Q_T = \frac{C_{pe} T_e \dot{G}_e \eta_{it}}{2 \pi n_{tc}} \left(1 - \left[\frac{P_a}{P_e} \right]^{\frac{k_e-1}{k_e}} \right), \quad (9)$$

where C_{pa} , C_{pe} are the specific heats at constant pressure of air and exhaust gas, respectively; k_a , k_e are the specific heat ratios for air and exhaust gas, respectively; $\dot{G}_a$, $\dot{G}_e$ are the flows of air and exhaust gas through engine and turbine, respectively; T_e is the exhaust gas temperature before turbine; T_a , P_a are the temperature and pressure of ambient air. Here it should be noted that owing to advance design of modern turbochargers the friction losses is neglected in this model.

The solution of torque equations requires evaluation of certain thermodynamic variables this, in turn demands consideration of engine as an open thermodynamic system where the main components of engine are represented by the control volumes in which time-dependent pressures, temperatures and flows can be modeled. From this point a

qualitative difference arises between various simulation approaches [4]. One may use empirical equations derived by regression analysis from the performance data furnished by the engine manufacturer. On the other hand one-dimensional and three-dimensional (CFD) models are used for detailed simulation of the thermodynamic and flow dynamic processes inside the engine components. The representation of the real engine processes is enhanced as the complexity of the used simulation tool is increased, but at the same time, greater amount of input data is required, the model execution time significantly increases and the model usage become laborious. In that respect a cycle mean value (CMV) model stands between simple empirical models and sophisticated filling-emptying models, owing to its simplicity and ability to represent the engine components behavior with sufficient accuracy. Moreover it is the most suitable for the self-propulsion model of a ship due to fast execution time.

Fig.1 Outline of engine model components

The physical principle, employed in CMV model is thermodynamic equivalence of the air and exhaust gas masses flow through engine and equivalent orifices as shown in Fig.1. In that respect, modeling of the air mass flow through the engine cylinders is considered equivalent to that through an orifice with its effective area remaining constant over time. The turbocharger turbine is also

considered as an orifice through which the exhaust gas flows. The turbocharger compressor is considered as a pump delivering pressurized air to the engine. The model is completed by the temperature/mass increase due to fuel injection and combustion.

4. Mean Value Model Description

4.1 Steady-state air/exhaust gas flow model

Hereinafter, the detailed description the core part of CMV model, namely flow of air and exhaust gas through an engine, is given and exemplified with a four-stroke turbocharged engine Niigata 6L19HX rated at 750 kW, 1000 rpm. Many of the equations constituting flow model are not dependent on the two or four-stroke aspect of cylinder cycle. There are some important changes though, which will be pointed out in the text.

In spite of intermittent flow of air through an engine cylinder, in the CMV model it is considered constant through an orifice with effective average area $\bar{A}_v$ and therefore quasi one-dimensional equation of flow through an orifice can be used as follows:

$$\dot{G}_{scv} = \bar{A}_v \frac{P_s}{\sqrt{R_a T_s}} \left\{ \frac{2 k_a}{k_a - 1} \left[\left(\frac{P_e}{P_s} \right)^{\frac{2}{k_a}} - \left(\frac{P_e}{P_s} \right)^{\frac{k_a + 1}{k_a}} \right] \right\}^{\frac{1}{2}} \quad (10)$$

In case of four-stroke engine the scavenging air flow should be supplemented with the air amount trapped in cylinder during pumping stroke:

$$\dot{G}_{tr} = \frac{z i V_s P_s n_e \eta_v}{T_s R_a} \quad (11)$$

The exhaust gas mass flow through the turbine is obtained similarly, assuming compressible flow through an orifice:

$$\dot{G}_e = a_t A_t \frac{P_e}{\sqrt{R_e T_e}} \left\{ \frac{2 k_e}{k_e - 1} \left[\left(\frac{P_a}{P_e} \right)^{\frac{2}{k_e}} - \left(\frac{P_a}{P_e} \right)^{\frac{k_e + 1}{k_e}} \right] \right\}^{\frac{1}{2}} \quad (12)$$

The energy and mass balances applied on the air and exhaust gas flows through the engine result in following equations, respectively:

$$(\dot{G}_a C_{p_a} T_s + \dot{G}_f H_u \xi_a) = \dot{G}_e C_{p_e} T_e, \quad (13)$$

$$\dot{G}_e = \dot{G}_a + \dot{G}_f, \quad \therefore \dot{G}_a = \dot{G}_{scv} + \dot{G}_{tr}, \quad (14)$$

where the fuel mass flow $\dot{G}_f$ is considered to be a proportion of nominal fuel mass g_{f_0} being injected at MCR and is governed by a fuel pump index h_p as follows:

$$\dot{G}_f = z i g_{f_0} h_p n_e. \quad (15)$$

Equations (10) through (15) comprise the bulk of steady-state flow model and solutions provide steady-state values of exhaust pressure P_e , exhaust temperature T_e (before turbine) and air mass flow rate $\dot{G}_a$. But a prior requirement is that the model be supplied with values for the constants and be supplemented with additional equations describing compressor performance and combustion process in terms of charge air pressure ratio $\left(\frac{P_c}{P_a}\right)$, charge air temperature T_s and indicated mean effective pressure P_i , respectively. Following chapters outline a procedure for finding all of the required constants and functions either.

4.2 Turbocharger compressor

Since the pressurized air is supplied by the compressor connected to the turbine, the flow equations should be supplemented with the model of compressor performance providing that equilibrium air/exhaust flow is held.

The compressor, in common approach, is modeled using its performance map but in the most cases it is laborious to get access to this map, thus the alternative way of performance representation is necessary. Since in marine propulsion plant systems, the engine is loaded according to the propeller law, the compressor pressure ratio can be modeled as a function of the turbocharger shaft speed based on following consideration.

Isentropic work of compression is expressed as:

$$L_c = C_{p_a} T_a \left(\left(\frac{P_c}{P_a} \right)^{\frac{k_a-1}{k_a}} - 1 \right), \quad (16)$$

and the relative change of this work can be assumed equal to the square of relative change of turbocharger speed [5]:

$$L_c = L_{c_0} \left(\frac{N_{tc}}{N_{tc_0}} \right)^2. \quad (17)$$

Manipulation of eq.(16) and (17) with respect to pressure ratio yields:

$$\left(\frac{P_c}{P_a} \right) = \left\{ \left[\left(\frac{P_c}{P_a} \right)_0^{\frac{k_a-1}{k_a}} - 1 \right] \left(\frac{N_{tc}}{N_{tc_0}} \right)^2 + 1 \right\}^{\frac{k_a}{k_a-1}}. \quad (18)$$

As a result of compression the temperature of air increases following adiabatic process, including constant compressor efficiency η_{ic} :

$$T_c = T_a \left[\frac{1}{\eta_{ic}} \left(\left(\frac{P_c}{P_a} \right)^{\frac{k_a-1}{k_a}} - 1 \right) + 1 \right], \quad (19)$$

In order to keep density of charge air as high as possible an air cooler is installed between compressor output and

cylinders input which is modeled as follows [6]:

$$T_s = T_{sw} + k_c (T_c - T_{sw}) \left(\frac{\dot{G}_a}{\dot{G}_{a_0}} \right)^{\frac{1}{3}}. \quad (20)$$

The air cooler efficiency k_c is considered constant and can be evaluated with the parameters of engine nominal operating regime: $k_c = \frac{T_{s_0} - T_{sw_0}}{T_{c_0} - T_{sw_0}}$. Where T_{sw} denotes temperature of cooling water.

Here it should be noted, that air cooler introduces certain pressure drop, however it is too low to affect modeling accuracy and, therefore may be disregarded, hence $P_s = P_c$. The presented model of turbocharger compressor checks with the experimental data of prototype engine. The Fig.2 shows pressure ratio as a function of turbocharger speed and Fig.3 shows charge air temperature as a function of air flow. As can be seen the semi-empirical representation of compressor provide acceptable agreement

Fig.2 Compression ratio – turbocharger speed relation

Fig.3 Charge air cooler model

4.3 Evaluation of remaining constants

The remaining parameters can be related to the following groups: engine operational parameters, thermodynamic properties of gases and empirical parameters appearing in the flow model.

To the first group relate cylinder displacement volume V_s , moments of inertia of engine I_e and turbocharger I_{tc} , inlet/exhaust valves flow areas, low calorific value of fuel H_u , etc.; these parameters can be found in engine data sheets or provided by manufacturer.

The thermodynamic constants of air and exhaust gases such as specific heat capacities $C_{p,a,e}$, specific heat ratios $k_{a,e}$ and gas constants $R_{a,e}$ have well-known standard values which can be found in the literature on internal combustion engines [7].

The remaining parameters are the volumetric efficiency of the engine cylinder η_v , the effective scavenging flow area $\bar{A}_v$, the turbine flow area A_t , the turbine flow area correction a_t , the fraction of fuel energy in exhaust gases ξ_a and the turbine efficiency η_{iT} .

As presented the last group parameters enter into the equations as constants, however the accuracy of simulation can be improved by discarding this notion for some parameters and construct empirical relationships using engine performance data.

The sensitivity analysis performed in [4] showed that effective scavenging flow area and turbine flow area may retain constant, thus $\bar{A}_v$ can be estimated using inlet and exhaust valves timing [2]:

$$\bar{A}_v = \frac{z}{\varphi_c} \int_0^{\varphi_c} \frac{f_{v_{in}}(\varphi) f_{v_{exh}}(\varphi)}{\sqrt{f_{v_{in}}^2(\varphi) + f_{v_{exh}}^2(\varphi)}} d\varphi, \quad (21)$$

where φ_c is the duration of one cycle in degrees, $f_{v_{in}}(\varphi)$ and $f_{v_{exh}}(\varphi)$ are the engine configuration data given as inlet/exhaust valves area versus crankshaft angle.

The valves timing can be approximated trigonometrically [8], with the sufficient confidence as follows:

$$\frac{f_v(\varphi)}{f_{v_{max}}} = \begin{cases} 0.5 \left\{ 1 - \cos \left(\frac{\varphi - \varphi_1}{\delta_\varphi + \Delta\varphi} \pi \right) \right\}, & \varphi \leq \varphi_1 + \delta_\varphi \\ 0.5 \left\{ 1 - \cos \left(\frac{\varphi - \varphi_2}{\delta_\varphi + \Delta\varphi} \pi \right) \right\}, & \varphi > \varphi_2 - \delta_\varphi \end{cases}, \quad (22)$$

where φ_1 , φ_2 are the angles of valve opening begin and end of closing, δ_φ is the duration of valve opening/closing, $\Delta\varphi \approx 15^\circ$ is the empirical correction for opening/closing

duration as stated in [8].

Since the effective flow area of the turbine does not vary greatly during one engine cycle, no cycle-averaging is needed and it can be calculated from the geometrical configuration data of turbocharger as follows:

$$A_t = \frac{f_n f_r}{\sqrt{f_n^2 + f_r^2}}, \quad (23)$$

where f_n is the minimum flow area of turbine nozzle, f_r is the minimum flow area of the turbine wheel.

On the other hand the turbine configuration data normally is not furnished by manufacturer, therefore alternatively value of A_t can be estimated from eq. (12) substituting the engine operating data at MCR and assuming that $a_t = 1$.

The density of air trapped in the cylinders differs from that in the intake manifold. This is due to the heating of air by the surrounding metal and pressure drop across inlet valve. This effect of density difference is represented by the volumetric efficiency η_v . As argued in [4], consideration of the volumetric efficiency as function of engine speed significantly improve accuracy of the model, therefore the following empirical equation [5] is adopted:

$$\eta_v = \eta_{v_{in}} \left\{ 1 - \frac{0.014}{\eta_{v_{in}}} \left[\left(\frac{n_e}{N_{e_{in}}} \right)^2 - 1 \right] \right\}, \quad (24)$$

where $\eta_{v_{in}}$ denotes initial volumetric efficiency which is determined from engine parameters at low load assuming that air flow is composed only trapped air component, $N_{e_{in}}$ is the speed of engine at which $\eta_{v_{in}}$ was evaluated.

Data for turbine isentropic efficiency η_{iT} , taken from manufacturer's characteristics, can be represented by numerical tables or empirical functions. If only a value η_{iT_0} at MCR is known, an alternative is to use this value corrected by a square-law relationship between efficiency and the nozzle-wheel speed ratio, therefore following empirical form can be adopted [9]:

$$\eta_{iT} = \eta_{iT_0} \left[\left(1 - \frac{\bar{U}_t}{\bar{C}_s} \right) \frac{\bar{U}_t}{\bar{C}_s} + 1 \right], \quad (25)$$

where $\bar{U}_t = \frac{n_{tc}}{n_{tc0}}$ is the relative turbine speed,

$$\bar{C}_s = \sqrt{T_{e0} \left(1 - \left[\frac{P_a}{P_e} \right]_0^{\frac{k_e-1}{k_e}} \right) \left[T_e \left(1 - \left[\frac{P_a}{P_e} \right]^{\frac{k_e-1}{k_e}} \right) \right]^{-1}}$$

is the relative gas speed in the nozzle. This is taken as inverse ratio in order to satisfy a square-law relationship.

The proportion ξ_a of the fuel chemical energy contained in exhaust gas is considered linear function of the engine mean

effective pressure P_{mep} [2]:

$$\xi_a = k_{\xi_0} + k_{\xi_1} P_{mep}, \quad (26)$$

where $P_{mep} = \frac{Q_e 2 \pi}{z i V_s}$.

In preceding discussion the turbine flow area was considered to be a definite value, but on the other hand, the flow coefficient a_t that modifies actual area holds functional relation with turbine pressure ratio, i.e.:

$$a_t = k_{a_0} + k_{a_1} \left(\frac{P_a}{P_e} \right) + k_{a_2} \left(\frac{P_a}{P_e} \right)^2, \quad (27)$$

The coefficients constituting eq. (26) and eq. (27) can be found simultaneously from the steady-state flow model, eqs.(10) through (15) and also turbine efficiency can be found more precisely by satisfying steady-state balance between turbine torque eq.(9) and compressor torque eq.(8). The engine operational data, which are usually available in the form of characteristic curves, are used as input.

4.4 Engine indicated mean effective pressure

Before proceed to model validation and dynamic simulation one more variable should be properly defined that is imep which determines indicated torque of engine by eq. (5). Since the CMV model neglects the internal performance of combustion process keeping only average values during one cycle, the imep is determined by the fuel flow rate $\dot{h}_p$ as follows:

$$P_i = \eta_c P_{i_0} \dot{h}_p, \quad (28)$$

This linear dependence of imep on fuel flow rate is held if an ample amount of air for combustion being maintained in the engine cylinder. Nevertheless this linearity ceases to hold in case air/fuel ratio drop below certain limit; this situation may occur during fast and large transient owing to so called turbocharger lag. Such a phenomenon clearly must be accounted for in the dynamic engine model. Thus, following combustion efficiency is introduced to the model:

$$\eta_c = 1 - 1.1 e^{-(1.2 \alpha)^2}, \quad (29)$$

where $\alpha = \frac{\dot{G}_{tr}}{L_0 \dot{G}_f}$ is the air excess ratio, L_0 is the stoichiometric amount of air.

Since there is no one universal methodology describing engine torque degradation as a consequence of air depletion, the eq. (29) represents weighted mean among various approach presented in the past publications [2], [4], and [9].

5. Dynamic Simulation of Engine Model

The typical simulation algorithm performs numerical integration of differential equations (2) and (3) over time; at each time interval instantaneous fuel rack position, engine and turbocharger speeds serve as the input to calculation of an equilibrium solution of air/exhaust flows. Ability to calculate an accurate value of steady-state air and exhaust flows is in consequence an essential and large part of an accurate dynamic model. However it should be noted that steady-state flow model during transients involving sudden load acceptance gives unrealistic change in exhaust gas temperature; for transient operation, the following dynamic relation for the exhaust gas temperature is therefore should be added [12]:

$$\frac{dT_e}{dt} = (T_e - T_{e-s}) \frac{1}{\tau_{er}}, \quad (30)$$

where T_{e-s} is the steady-state value of exhaust temperature before transient, τ_{er} is the exhaust receiver time constant which can be estimated as follows [10]:

$$\tau_{er} = \frac{V_{er} P_{e0}}{k_e R_e T_{e0} \dot{G}_{e0}}, \quad (31)$$

where V_{er} is the volume of exhaust gas receiver.

At first the steady-state characteristics of engine simulation model was compared with that obtained experimentally for the most important engine operating parameters such as exhaust gas temperature, charge air/exhaust gas pressures, air/exhaust gas flows and engine torque at nominal propeller load. The results are presented in figures 4 through 7.

Fig. 4 Exhaust gas temperatures before/after turbine

Fig. 5 Charge air/exhaust gas pressures

Fig. 6 Charge air/exhaust gas flows

Fig. 7 Engine brake torque

In order to verify transient characteristics of simulation model, the engine acceleration test was performed. Figures 8 and 9 depict acceleration in engine speed as consequence of a fast up-ramp in fuel rack position, followed after some dwell by deceleration as consequence of a down-ramp in fuel rack. Engine speed and charge air pressure are shown, as are the calculated one for the same rack position ramps. Load on the engine is a water brake whose torque-speed characteristic is expressed by:

$$Q_{wb} = k_{wb0} + k_{wb1}n_e, \quad (32)$$

where k_{wb0} , k_{wb1} are the empirical gain coefficients.

The confirmation of engine transient dynamic can be claimed as good, however there is still room for improvement. It is obvious that neglecting the effect of air and exhaust gas receivers' inertia, results in faster response of the thermodynamic variables. On the other hand addition of the receivers' model requires additional data about engine and simulation time. This deficiency though, can be diminished by increasing the value of turbocharger inertia as shown in [11]. At the first stage the presented simplified engine model will be incorporated into the self-propelled ship model and coupled behavior will be examined. Further the engine model will be modified and checked again until the optimal configuration, in terms of model complexity and ability of real-time execution, is found.

6. Conclusion

Modeling of a diesel engine for the purpose of marine propulsion plant simulation is usually trade-off between the detailed representation of engine processes and simulation accuracy provided by the model. The presented model is accomplished through application of fundamental equations supplemented as necessary by empirical relationships. The transient simulation shows fairly well accuracy in the computation of engine speed and torque even though accuracy in the simulation of thermodynamic variables is to a certain extent compromised. In that respect the more engine data available the more accurate result can be achieved.

For the purpose of providing an engine simulation environment in the self-propulsion ship model, this approach is considered to be adequate and appropriate.

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Fig.8 Validation of model dynamics, initial engine power = 85% (marks denote measured data, lines denote calculated data)

Fig.9 Validation of model dynamics, initial engine power = 70% (marks denote measured data, lines denote calculated data)