THE DYNAMIC BEHAVIOR OF A DIESEL ENGINE.

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ABSTRACT

In modeling the dynamic behavior of a pump/pipeline system, a subject of previous papers at the WEDA/TAMU conferences. The diesel engine is modeled very simple as a first order system. The time constant of the first order system determines the time delay between changing a set point for the revolutions of the diesel and the change of the real revolutions of the diesel engine. This modeling is too simple to describe the real behavior of the diesel engine. More complex models exist, but in general they are too complex, describing the full thermodynamic behavior of diesels. Those models are used for the simulation of engine rooms.

So there is a need for a model that is more advanced then a first order system and less advanced then the very complex models. Such a model has been derived, based on the Seiliger (thermodynamic) process. The results of the model show that the diesel engine behaves like a second order system when operating in the governor area and more like a first order system in the constant torque (overload) area. This model will be incorporated in the pump/pipeline model. The paper will give a detailed description of this new model, including the MATLAB model and give examples of the response of the model.

Keywords: Dredging, Diesel Engine, Dynamics

INTRODUCTION

The simulation model of a diesel engine can be regarded as an explanation of the real engine operation, which combines the mathematical relationship between the relative components and can be used to simulate the dynamic process of the diesel engine. A clear overview of engine operations is helpful to understand the modeling of the real diesel engine.
As introduced by Heywood, 1988 [1], table 1.3, the normal application in huge load marine vessels is the diesel engine. In the dredging field, the same level of output work is needed and so normally diesel engines are used as the main power supply.

Intake system

In an internal combustion engine, air is induced into the cylinders. The airflow first passes through an air filter to get the qualified fresh air. Then it flows into the compressor, during which the air pressure is increased to be higher than the atmospheric pressure. The charge air then flows through an inter cooler to decrease the intake air temperature. Hence the air density is increased again prior to the cylinder. Finally, it flows through a manifold and inlet valve into the cylinder.

Cylinder operation cycle

Four-stroke cycle (Otto cycle):

1. An intake stroke, which draws fresh mixture into cylinder. To increase the mass inducted, the inlet valve opens shortly before the stroke starts and closes after it ends.
2. A compression stroke, when both valves are closed and the mixture inside the cylinder is compressed to a small fraction of its initial volume. Toward the end of the compression stroke, combustion is initiated and the cylinder pressure rises more rapidly.
3. A power stroke, or expansion stroke, which starts with the piston at top-center (TC) and ends at bottom-center (BC) as the high-temperature, high-pressure gases push the piston down and force the crank to rotate. As the piston approaches BC the exhaust valve opens to initiate the exhaust process and drop the cylinder pressure to close to the exhaust pressure.

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4. An exhaust stroke, where the remaining burned gas exits the cylinder: first, because the cylinder pressure may be substantially higher than the exhaust pressure, and as they are swept out by the pistons it moves toward the TC. As the piston approaches the TC the inlet valve opens. Just after the TC the exhaust valve closes and the cycle starts again. The four-stroke cycle requires two crankshaft revolutions for each power stroke.

Two-stroke cycle:
1. A compression stroke, which starts by closing the inlet and exhaust ports, and then compresses the cylinder contents and draws fresh charge air into the crankcase. As the piston approaches the TC, combustion is initiated.
2. A power or expansion stroke, similar to that in the four-stroke cycle until the piston approaches the BC, when first the exhaust ports and then the intake ports are uncovered.

Most of the burnt gases exit the cylinder in an exhaust blow-down process. When the inlet ports are uncovered, the fresh charge, which has been compressed in the crankcase, flows into the cylinder. The piston and the ports are generally shaped to detect the incoming charge from flowing directly into the exhaust ports and to achieve effective scavenging of the residual gases.
Each engine cycle with one power stroke is completed in one crankshaft revolution. It can obtain a higher power from a given engine size.

In this paper the four-stroke process is modeled.

The exhaust and turbocharger system
The exhaust flows through the outlet valve and manifold ant then flows into turbine. It drives the turbine, which powers the compressor. The turbocharger, a compressor-turbine combination, uses the energy available in the engine exhaust stream to achieve compression of the intake flow. The function of the turbine and the compressor is to increase the maximum power that can be obtained from a given displacement engine. The work transfer to the piston per cycle, in each cylinder, which controls the power the engine can deliver, depends on the amount of fuel burned per cylinder per cycle. This depends on the amount of fresh air that is induced each cycle. The airflow at a given engine speed is essentially unchanged. But for the turbocharged engine, the inlet air is compressed by this exhaust-driven turbine-compressor combination. With the compression function of the compressor, the intake air pressure is increased, and hence the air density. Increasing the air density prior to entry into the engine thus makes the increasing in fuel flow allowed. Then it increases the maximum power that an engine can offer.

Fuel injection system
In an internal combustion engine, the fuel is injected directly into the engine cylinder, just before the combustion process is required to start. Load control is achieved by varying the amount of fuel injected each cycle.
In a large size engine, direct-injection systems are used. The diesel fuel-injection system consists of an injection pump, delivery pipes and fuel injector nozzles, the governor and a timing device. The injection pump generates the pressure required for fuel injection. The fuel under pressure is forced through the high-pressure fuel-injection tubing to the injection nozzle, which then injects it into the combustion chamber.
The amount of fuel injected is determined by the injection pump cam design and the position of the helical groove. As the pump plunger arrives at the bottom dead center (BDC), the pump-barrel inlet ports are open. Through them, the fuel, which is under supply-pump pressure, flows from the pump's fuel gallery into the high-pressure chamber of the plunger and barrel assembly. Then in the following pre-stroke process, retraction stroke (only if a constant-volume valve is used), the fuel pressure increases even higher. During the followed effective stroke, the fuel is forced through the high-pressure line to the nozzle. The effective stroke is terminated as soon as the plunger's helix opens the spill port. Changing the plunger's effective stroke varies the injected fuel quantity. To do so, the control rack turns the pump plunger in the barrel so that helix, which runs diagonally around the plunger circumference, can open the inlet port sooner or later and in doing so change the end-of-delivery point and thus the injected fuel quantity.
The plunger speed, and therefore the duration of injection, depends upon the plunger actuating cam's lift relative to the angle of cam rotation. This is why a wide variety of different cam contours are required for everyday operations.
The mathematical modeling of the cam contour and helix groove is up to specific components used in real engine work. Besides, the fuel spay condition is difficult to model either. Here the fuel injection system is assumed as a linear system with the signal input from the governor, which is up to the load condition.

Explanation of the process in a diagram
A diagram is introduced in [3] to explain the operation principles of the diesel engine. It is modified to fit our requirement. The dynamic behavior of the turbine and compressor system is taken into account.

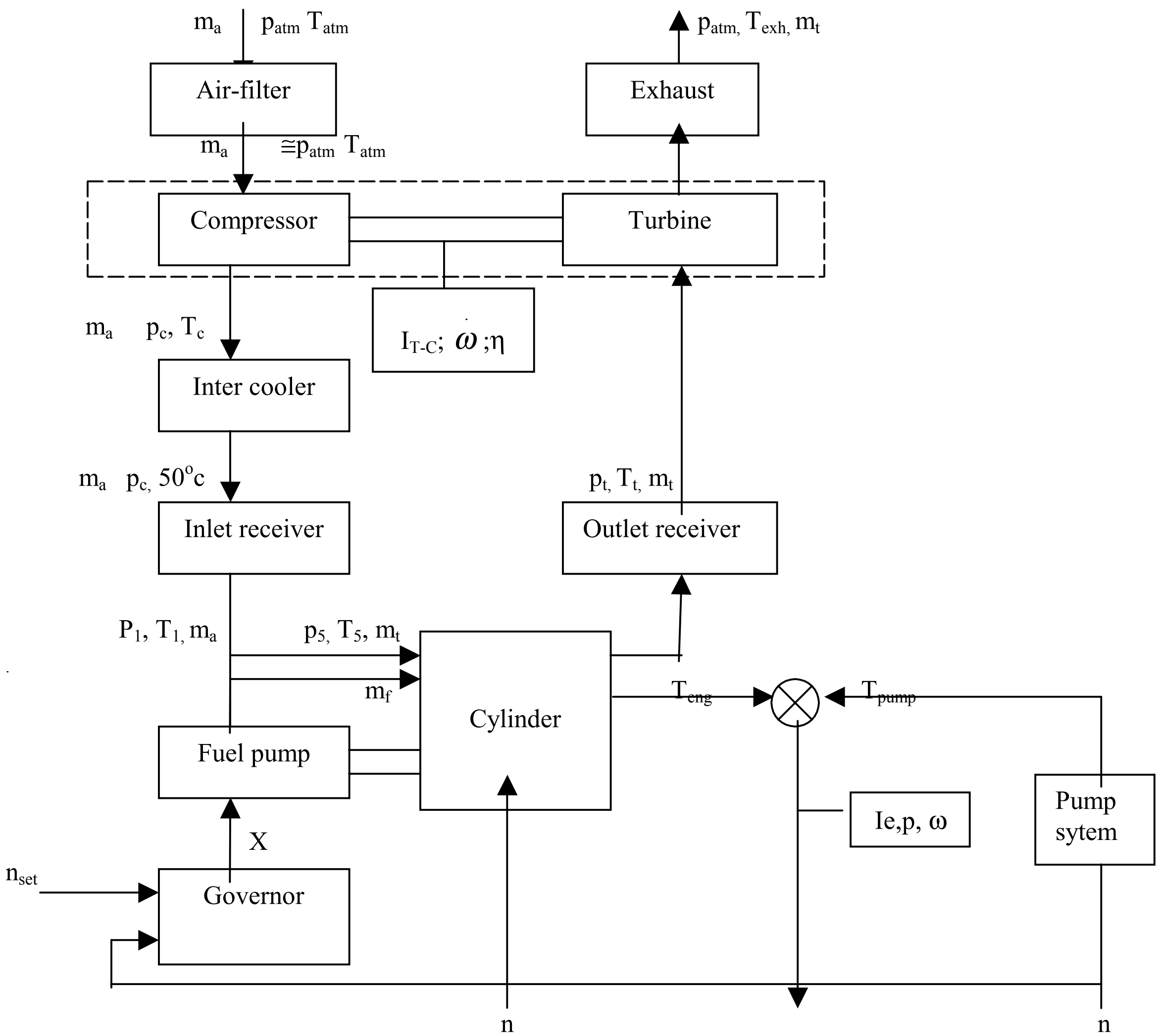

Figure 1: Diagram of a detailed diesel engine model

MODELING THE ENGINE

In this paragraph, the principles of modeling the system and different parts is introduced. All works are based on these principles. This part is defined with steps in the time domain, as the final model is simulated per cycle, which is more convenient. With this introduction it is easier to understand the modeling process, because as we model it in per cycle, the transfer due to units and steps makes the calculation more complicated.

Diesel engine system model
With the availability of detailed technical data, the former model is very ideal. The classic dynamic behavior includes inertia of the diesel engine and turbine-compressor system as the main parts. For other subsystems the dynamic behavior is mainly based on the thermodynamic behavior, which is rather complicated. Fortunately, the charge air condition is shown in the manual of a specific engine. With these parameters it is possible to ignore the sub-systems like, air-filter, inter-cooler, inter-receiver and outlet-receiver in a simple model. The cylinder model is the most important part. It explains the combustion process inside the cylinder and calculates important operating

parameters such as the output work, mean effective pressure, break torque, revolution speed, etc. To model the combustion process the subsystem used to explain the heat release is necessary. There are several assumptions about parameters and processes in this subsystem with which the simulation results are nearby the realistic situation. The dynamic behavior of the turbine and the compressor systems are very complicated. The performance map of the turbine and the compressor system is a common tool used in the modeling. Though the performance map is available from the manufacturer, unfortunately, for different engines such a performance map may be different. It brings the disadvantage that there is no general accurate model applied in a wide field. A general assumption is used to explain its dynamic action. Load control is achieved by varying the amount of fuel injected each cycle while the airflow at a given engine speed is essentially unchanged. So the governor and the fuel pump system are a necessary part. Following is a diagram to explain the principle of the engine model.

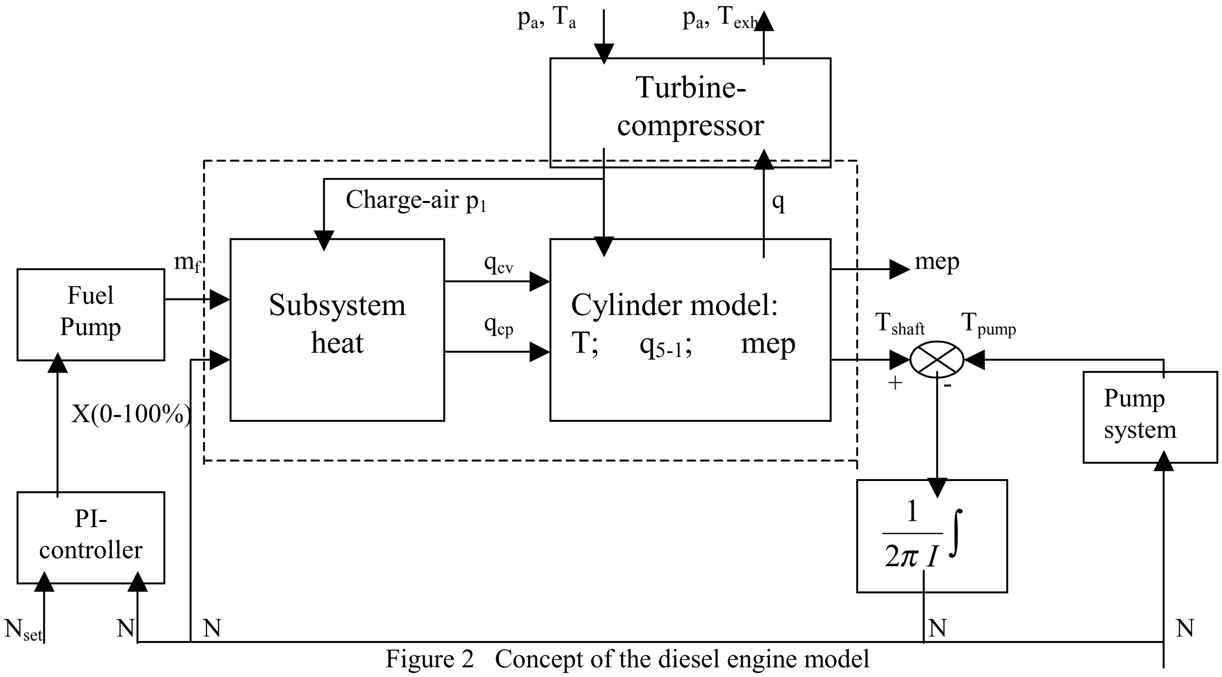

Figure 2 Concept of the diesel engine model

Typical Seiliger diagram

The well-known typical 6-point-Seiliger process has six predefined points and five predefined processes to simulate the internal-combustion process:

Stage 1-2: Compression stroke (polytropic process)
Stage 2-3: Constant volume (or premixed) combustion
Stage 3-4: Constant pressure combustion (and expansion)
Stage 4-5: Constant temperature combustion (and expansion)
Stage 5-6: Expansion stroke (polytropic process).

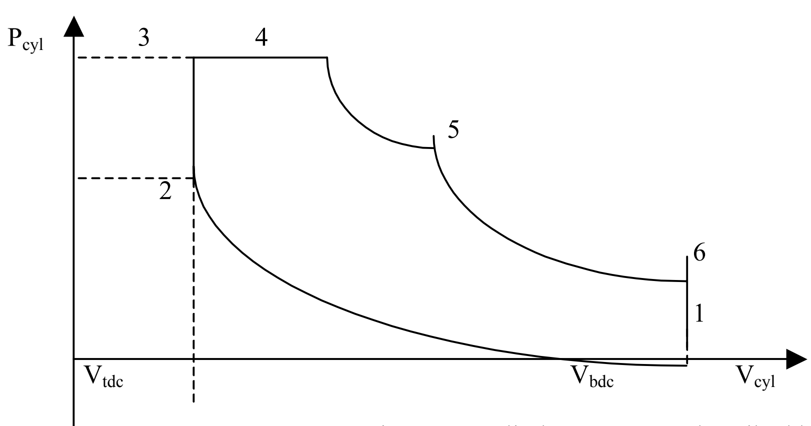

Figure 3 Cylinder process as described by Seiliger

Simplified Seiliger diagram

The cycle inside of cylinder is approximated due to typical Seiliger process to simplify the derivation. The simplification ignores the constant temperature combustion process. Stage 1-2, is divided into two periods. During the first period, gas may gain heat from cylinder wall. But, during the second period, as the gas temperature increases due to compression, gas may release heat via cylinder wall. So this stage is assumed to be an isentropic process in the simplified model.

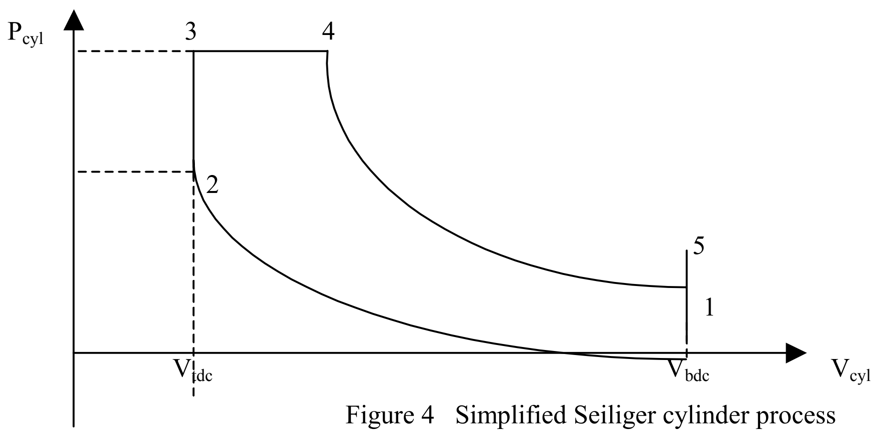

Figure 4 Simplified Seiliger cylinder process

Once the charged air condition and fuel consumption are known, the Seiliger diagram is calculated. This gives the mean effective pressure and work output and heat released to the exhaust system.

There are two characteristic variables in the simplified Seiliger diagram that determine the shape of the combustion part:

$$a = \frac{T_3}{T_2} = \frac{p_3}{p_2}; \quad b = \frac{T_4}{T_3} = \frac{V_4}{V_3} \tag{1}$$

The theoretical derivation of variables a and b is introduced by Grimmelius, 1999 [2].

$$a = 1 + \frac{X_a \cdot \dfrac{\tau_{id}}{\tau_{id,nom}} \cdot q_{f,23}}{c_v \cdot T_2} \tag{2}$$

With:

$$\begin{aligned}
&X_a = f(IT) \cdot (K_{1a} + k_{2a} \cdot \frac{N}{N_{nom}}) \\
&\tau_{id} = f(p_2, T_2, fuelproperties) && (ignition\ delay) \\
&q_{f,23} = \frac{\Phi_{m,f} \cdot H_0 \cdot \eta_{cb}}{\Phi_{m,l}} && for \quad \Phi_{m,f} \le \Phi_{m,f.\max} \\
&q_{f,23} = \frac{\Phi_{m,f,\max} \cdot H_0 \cdot \eta_{cb}}{\Phi_{m,l}} && for \quad \Phi_{m,f} > \Phi_{m,f,\max}
\end{aligned} \tag{3}$$

Variable b is not calculated directly. Instead, the intermitted parameter bb is calculated:

$$bb = a \cdot b = 1 + \frac{X_b \cdot q_f}{c_p \cdot T_2}; \tag{4}$$

With:

$$X_b = k_{1b} + k_{2b} \frac{N}{N_{nom}} : \qquad q_f = \frac{\Phi_{m,f} \cdot H_0 \cdot \eta_{cb}}{\Phi_{m,l}} \tag{5}$$

The variables a and b are determined by the condition in point 2, the engine speed and the fuel consumption. But in the real process of modeling the cylinder cycle, as more technical data are available, these variables can be derived in simpler way. However, the principle is same.

Calculation of Seiliger diagram
Stage 1-2: isentropic process:
V_1 is the maximum cylinder volume while V_2 is the minimum cylinder volume.

$$T_2 = T_1 \left(\frac{V_1}{V_2} \right)^{k-1}; \qquad p_2 = p_1 \left(\frac{V_1}{V_2} \right)^{k}; \qquad W_{1-2} = m \int_{V_1}^{V_2} p dv = m \cdot \frac{p_1}{1-k} (V_1^{\,k} V_2^{\,1-k} - V_1). \tag{6}$$

T_1 and p_1 are parameters from the charged air condition. W_{1-2} is the work required to compress mixed gas inside the cylinder. So it is negative output work.
Stage 2-3: constant volume process:

$$V_3 = V_2; \qquad p_3 = a \cdot p_2; \qquad T_3 = a \cdot T_2. \tag{7}$$

Stage 3-4: constant pressure process:

$$p_4 = p_3; \qquad T_4 = a \cdot T_3; \qquad V_4 = b \cdot V_3; \qquad W_{3-4} = m \cdot p_3 \cdot (V_4 - V_3) = m \cdot p_3 \cdot V_3 \cdot (b-1). \tag{8}$$

Stage 4-5: polytropic process:

$$V_5 = V_1; \qquad p_5 = p_4 \cdot \left(\frac{V_4}{V_5} \right)^{n}; \qquad T_5 = T_4 \cdot \left(\frac{V_4}{V_5} \right)^{n-1}; \qquad W_{4-5} = m \cdot \int_{V_4}^{V_5} p dv = m \frac{p_4}{1-n} (V_4^{\,n} V_5^{\,1-n} - V_4). \tag{9}$$

n is an assumed a polytropic exponent, which should match the output of the real engine.

$$q_{5-1} = m \cdot C_v \cdot (T_5 - T_1) \tag{10}$$

q_{5-1} is the heat released from the cylinder through the outlet receiver to the turbine.

Subsystem heat calculation
This subsystem is used to define the different processes of the internal combustion, such as the constant volume process and the constant pressure process. All the heat is assumed to be produced by injected fuel. Specific fuel consumption data is available in engine operation manuals. The heat value is a characteristic of the fuel in use. Of course not all produced heat can be taken into account, an assumed efficiency is taken into account.

$$Q_{eff} = \eta_q \cdot Q_{tot} = \eta_q \cdot bsfc \cdot H_0 \tag{11}$$

The heat loss can be divided into two parts. The first part is produced by the constant volume process. The second part is from the constant pressure process. We assume a parameter X_a here, to show this division.

$$Q_{cv} = Q_{eff} \cdot X_a; \qquad Q_{cp} = Q_{eff} \cdot (1 - X_a). \tag{12}$$

With these heat values the parameters a and b in the simplified cylinder model can be calculated. Then, the whole process of the simplified Seiliger process is made down. The X_a is shown as:

$$X_a = k_{1a} + k_{2a} \frac{N}{N_{nom}}. \tag{13}$$

k_{1a} and k_{2a} are matched by modeling and adjusting to different engine types.

TURBINE AND COMPRESSOR SYSTEM

Ideal first-order dynamic model of a turbine-compressor system
The turbine and compressor are assumed to be of the centrifugal type. The power used to drive the turbine comes from heat transfer released from the cylinder, which is the so called exhaust-driven turbine. This power is transfered to the compressor to compress the intake air. With a certain value of power input, the air can be conditioned to a certain density level and velocity. This system can be modeled as a first order classic dynamic system.

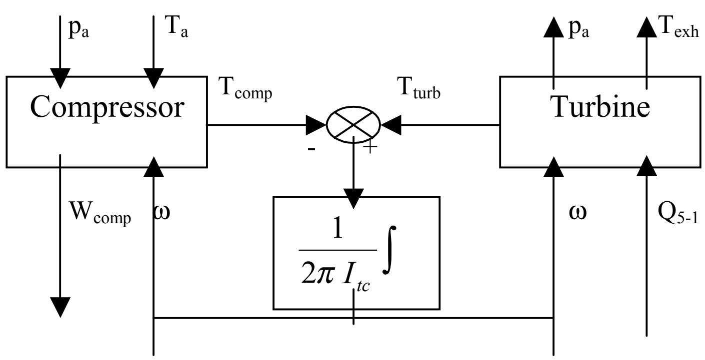

Figure 5 Ideal first-order model of turbine-compressor system

In Heywood, 1988 [1] there are detailed calculations about the relation between work, torque, revolutions of the turbine and compressor components. The work-transfer rate of power required to drive the compressor is according to (6.42) and (6.58) in [1]:

$$-\dot{W}_C = \frac{\dot{m}_C \cdot c_{p,i} \cdot T_a}{\eta_C} \left[\left(\frac{p_{out}}{p_a} \right)^{(\gamma-1)/\gamma} - 1 \right] = T \cdot \omega \tag{14}$$

The work offered to the turbine by the heat from the cylinder, can be calculated as follows:

$$\dot{W}_T = \eta_q \cdot \dot{Q}_{5-1} \tag{15}$$

With the same principles as used for the compressor, the thermodynamic process of the turbine can be explained as:

$$\dot{W}_T = \frac{\dot{m}_T \, c_{p,T} T_{i,T}}{\eta_T} \left[1 - \left(\frac{p_a}{p_{i,T}} \right)^{(\gamma-1)/\gamma} \right] \tag{16}$$

Part of the work from the turbine is transferred to drive the compressor according to (6.49) in Heywood [1]:

$$-\dot{W}_C = \eta_m \cdot \dot{W}_T \tag{17}$$

To solve this model, the initial value of the revolutions and the inertia are required. But with the technical data supplied, the inertia data are always kept in secret by the manufacturer. For the different types of diesel engines, different types of turbines and compressor systems may be applied. So for different diesel engines this model has to be modified to match the real situation. There are several assumptions about parameters made for this model. These parameters may be modified to match the simulation results with the real operating curves. The most important parameter is the time constant for this first-order system. As introduced by van Duyn 1998 [4], the time constant for turbine and compressor system normally range from 1 to 2 seconds. For engines with large output power, as those used in the dredging field, a choice of 2 seconds is a rough assumption. If the detailed technical data are available, the first model is more ideal.

Simplified turbine-compressor model

Without enough data the model is simplified to be a combination of a static system model and a first order transfer function.

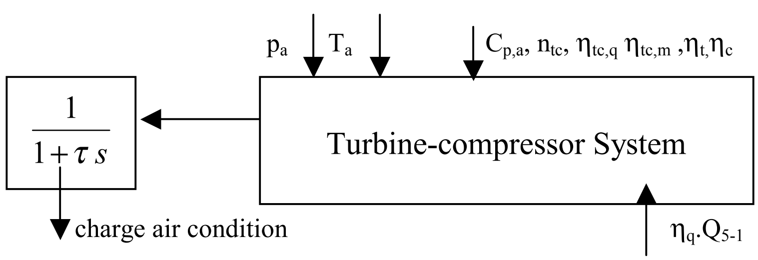

Figure 6 Diagram of simplified turbine-compressor system

The power offered to the turbine and compressor system is Q_{5-1} which is calculated in cylinder model. For the efficiencies reasonable values can be assumed. Then in one cycle with the unit working fluid, the static equation is:

$$Q_{5-1} = m_t \cdot c_{p,c} \cdot Ta \cdot \left[\left(\frac{p_{e,c}}{p_a} \right)^{(n_{tc}-1)/n_{tc}} - 1 \right] / \eta \tag{18}$$

η is the product of all efficiencies of the exhaust system, intake system and turbine-compressor system.

Fuel management subsystem
The amount of fuel injected is determined by the load of the system. Since it is difficult to connect the load condition directly to the fuel management system in this model, it is carried out in a different way. As the load condition or the torque offered by engine system changes, it must lead to a change of the revolutions. The difference between the real revolutions and the set point of the revolutions can be taken as the input of the governor of the fuel management system. Besides, during the modeling process, a saturation box is added into the system to prevent too much or too less fuel injection.

Mathematical model matching
During the simulation all the processes should be known mathematicly. Based on the principles introduced in the former chapter, with the mathematical derivations known, it is more convenient to understand the process. Besides, by comparing the output data of the model with the technical data offered in the operation manual, some parameters may be adjusted to match the model to a realistic situation.
Following, all the processes are derived per cycle per cylinder, with the working fluid as unit mass.

Calculation in subsystem heat
The output of this system is used in the cylinder model to derive parameters a and b which are used to define the internal combustion process. In one cycle, the heat produced by heat is:

$$Q_{eff,cyc} = \eta_q \cdot m_{f,cyc} \cdot H_0 \tag{19}$$

Here the amount of fuel is measured in one cycle, so that it can match with the fuel management model easily.
Then we transfer it into unit per cycle per kilo mass flow.

$$m_{a,cy}c = \frac{p_1 \cdot V_1}{R_a \cdot T_1}; \tag{20}$$

Then:

$$q_{eff,cyc,kg} = \frac{Q_{eff,cyc}}{m_{a,cyc} + m_{f,cyc}} \tag{21}$$

Calculation inside the cylinder model
How to define each point of the simplified Seiliger diagram has been introduced before. Here, what is left is how to derive a and b. It is difficult to make assumptions of the value of X_a directly. But since a and b are constant parameters, they should have certain value at any time. Normally, air condition characteristics at the nominal point are offered with operation manual. These parameters come from engine tests of manufactures which also can be applied to real dredging work. Among these parameters the charged air condition and the maximum cylinder pressure are available. After the assumed isentropic compression the charged air pressure transfers to pressure after compression p_2.

$$a = \frac{p_{\max}}{p_2} \tag{22}$$

Special heat c_v has no great change even when the temperature change is around several hundreds degrees. The variation is less than 5%. So in stage 2-3, the constant-volume combustion, roughly the heat release is:

$$q_{cv} = c_v \cdot (T_3 - T_2) = c_v \cdot T_2 \cdot (1 + a) \tag{23}$$

It is similar to parameter b:

$$q_{cp} = c_p \cdot T_3 \cdot (1+b) \quad \text{and} \quad q_{cp} = q_{eff} - q_{cv} \tag{24}$$

Output work per cycle with unit mass flow is the result of negative work in stage 1-2 and positive work in stage 3-4 and 4-5.

$$W_{cyc,kg,cyl} = \frac{p_1}{1-k}(V_1{}^k V_2{}^{1-k} - V_1) + p_3 V_3 (b-1) + \frac{p_4}{1-n}(V_4{}^n V_1{}^{1-n} - V_4) \tag{25}$$

Roughly the heat release is:

$$q_{5-1,cyc,kg,cyl} = c_v (T_5 - T_1) \tag{26}$$

The mean effective pressure is:

$$mep = \frac{W_{cyc,kg,cyl}(m_{a,cyc,cyl} + m_{f,cyc,cyl})}{V_d} \tag{27}$$

The output torque of the shaft is used to compare with the required torque from the pump system. The difference will lead to the dynamic action of the engine rotation parts and pump, as well.

$$T_{shaft} = \frac{W_{cyc,kg,cyl}(m_{a,cyc,cyl} + m_{f,cyc,cyl})\eta_m}{4\pi} \tag{28}$$

What is interesting, it has no relation with revolution speed.
For heat efficiency in the subsystem heat and mechanic efficiency, it is assumed these are a function of parameters of the charged air condition, thus it changes together with different charge pressures. This matches the real engine operation.

$$\eta_q = 1 - \frac{c_q}{p_1}; \qquad \eta_m = 1 - \frac{c_m}{p_1}. \tag{29}$$

During which c_q and c_m are constants, which will be matched in the simulations.

Calculation in turbine-compressor system
For the ideal model, more detailed technical data are required, so this paper is limited to the simplified model. Because the intake and exhaust system are all ignored in the simplified model, some important parameters have to be transferred into the turbine-compressor system. One of them is efficiency. All efficiencies such as, the intake system, the exhaust system, the turbine and the compressor are combined together into one value. It is possible to match this total efficiency by simulation.

$$p_{1,cyc,kg,cyl} = (1 + \frac{1}{\tau s}) \cdot p_a \cdot (\frac{\eta_{com} q_{5-1,cyc,kg,cyl}}{c_{p,a} T_a} + 1)^{\frac{n_{tc}}{n_{tc}-1}} \tag{30}$$

The combined efficiency also can transfer back to all separated values. By comparing the combined efficiency value from the simulation with the products of all included efficiencies, it can be seen that the modeling process is nearby a realistic situation. The efficiency of the heat transfer in the turbine can be calculated with the value of the heat released to environment and the heat offered from cylinder (q_{5-1}). The efficiency of the intake system is roughly based on the water cooling-system. The operation efficiency of the turbine and the compressor may be an assumption based on the development of the manufacturer of those systems.

CONCLUSIONS

In appendix I several output results of the simulations are shown in graphical form. The results are based on changing the pump load. T_{pump} is set as a step function: from 0 to 25 seconds, it equals to nominal value of shaft torque; at the time of 25th seconds it steps to 1.2 times the shaft torque.
From the graphs, we may see that at the start point there is no output value of shaft torque, and thus the required torque is higher that torque offered by engine. The engine is decelerated and engine speed decreases. In the mean time the engine will ask for more fuel. The signal output from PI control is increased which increases the fuel injection. With more fuel, the torque of the shaft is increasing. After certain steps, the engine and pump system reach the balance point. All parameters are in stable point at that time. From the graph we may find the process is not simply a first-order action system, but it acts as a higher order system, due to the complicated thermodynamic processes and classic dynamic behavior of engine and turbine-compressor system and PI controller.
At 25 seconds, as the required torque suddenly increases to 1.2 times the initial value, the engine will ask for more power again. Then the fuel injection, the shaft torque and the charging air pressure are increasing from stationary point. But this time as the required torque are too high, which leads to overload, not enough fuel is available, due to the limitation of the fuel management system. As the torque reaches the constant value of what is required by the pump, the engine speed decreases quickly.
These graphs roughly show the behavior of a diesel engine with increased load. This behavior are nearby realistic condition. Later, if we apply this model with parameters from real facilities, such as real fuel pump and turbine and compressor etc, the graph will show the real situation more accurate.

After the model is completed, it has been applied with data of a real M.A.N-L58/64 engine to check the reliability of work. With the manual of the M.A.N engine, not all the data required are available. So some estimations have to be made. Hence, the result may not show us the real operation the engine, but it shows the rough behavior of some important parameter in the dynamic process.

SYMBOLS

a	Constant specified for constant-volume process	[-]
b	Constant specified for constant-pressure process	[-]
bsfc	Specific fuel consumption	[g/kwh]
c	Specific heat	[kJ/kg.K]
c_p	Specific heat at constant pressure	[kJ/kg.K]
c_v	Specific heat at constant volume	[kJ/kg.K]
H_0	Heat value	[kJ/kg]
I	Moment of inertial	[kg.m^2]
k	Specific heat ratio	[-]
m	Mass	[kg]
$\dot{m}$	Mass flow rate	[kg/s]
n	Polytropic exponent	[-]
N	Revolution speed in rpm	[rpm]
p	Pressure	[pa]
q	Heat transfer per unit mass fluid	[kJ]
Q	Heat transfer	[kJ]
$\dot{Q}$	Heat transfer rate	[kJ/s]
R	Gas constant	[J/kg.k]
r	Compression ratio	[-]
T	Time	[s]
T	Temperature	[K]
T	Torque	[N.m]
V	Volume	[m^3]
V_d	Displaced cylinder volume	[m^3]
W	Work transfer	[J]
X	Signal output of governor	
X_a	Parameter for division of internal combustion process	
η	Efficiency	
τ	Characteristic time	
τ_{id}	Ignition delay	
ω	Revolution speed in Hz	
Φ	Mass or energy flow	

SUBSCRIPTS

a	Air
atm	Atmosphere
c,comp	Compressor
cp	Constant pressure process
cv	Constant volume process
cyc	Per cycle
cyl	Per cylinder
eff	Effective value
eng	Engine
exh	Exhaust
f	Fuel
i	Intake system
kg	Unit mass work fluid
m	mechanic
Nom	Nominal point characteristics
pump	Pump system
q	heat
set	Set point
shaft	Shaft characteristic
T,turb	Turbine
tc	Turbine-compressor system
1,2,3,4,5,6	Stage in cylinder process cycle

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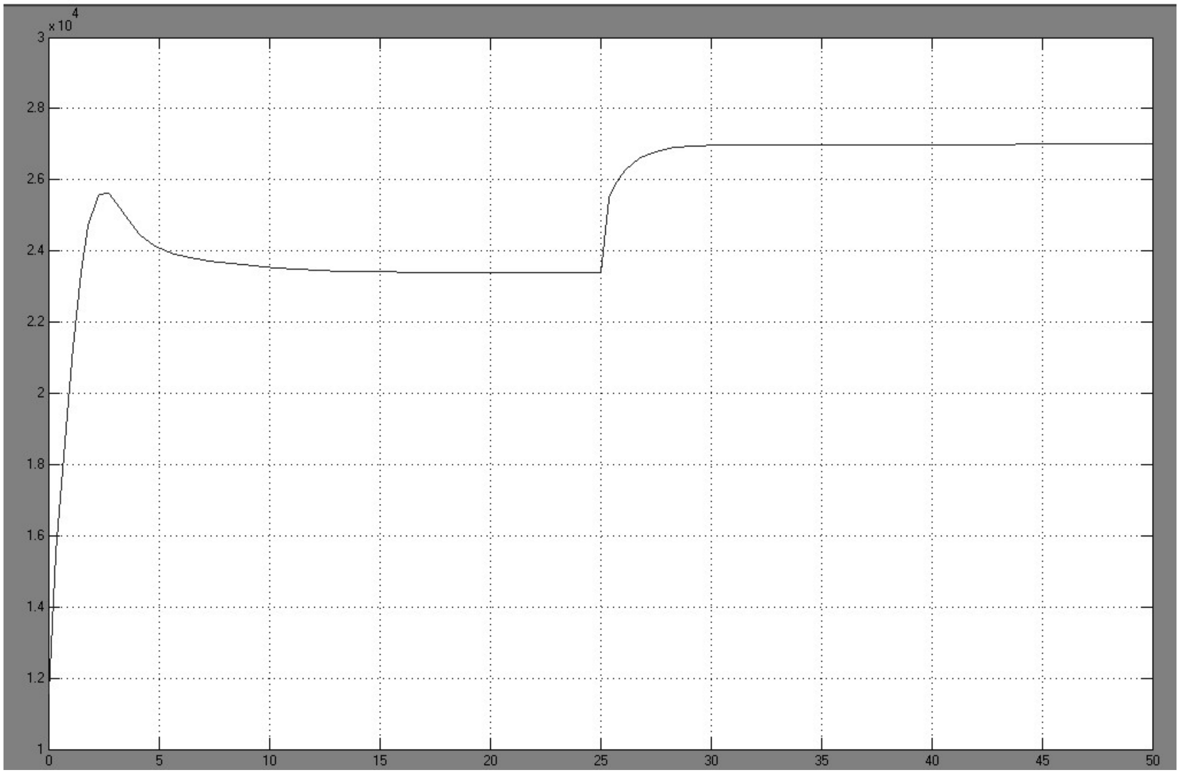

Figure 7: Shaft torque (Y-axis T_{shaft} in [N.m]; X-axis time in [s])

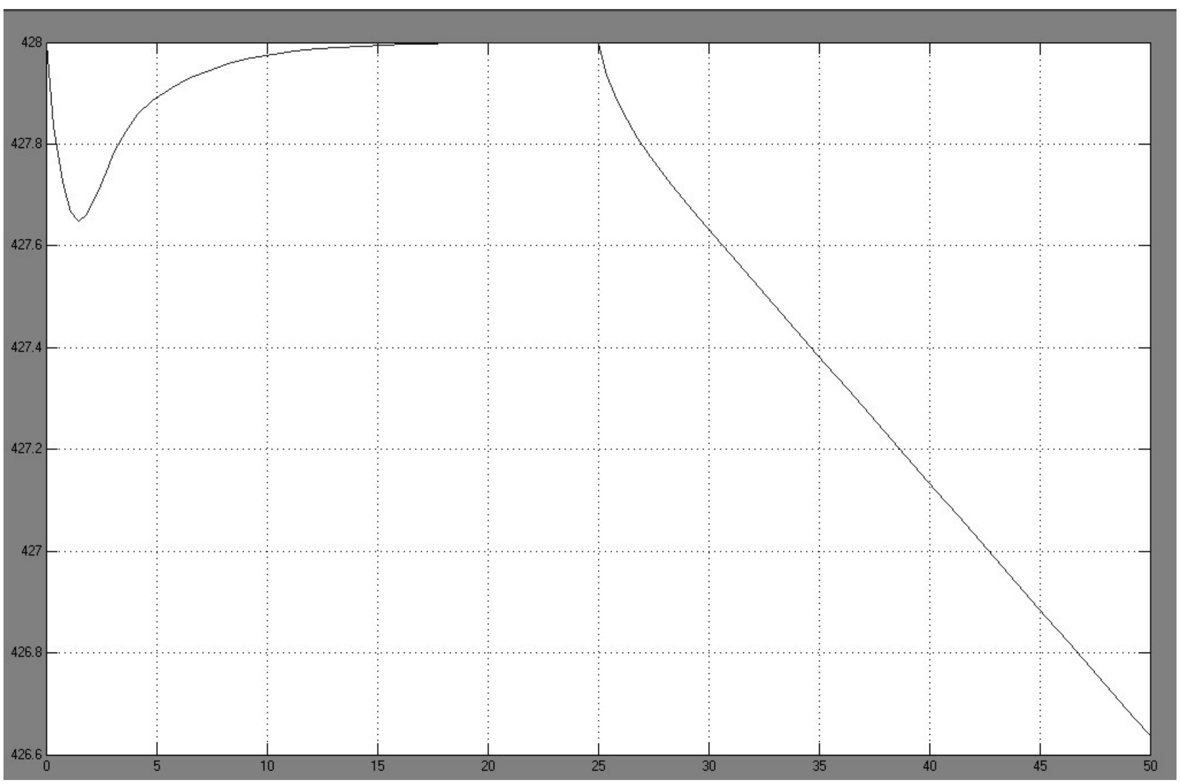

Figure 8: Revolution speed (Y-axis: revolution speed in [rpm]; X-axis: time in [s])

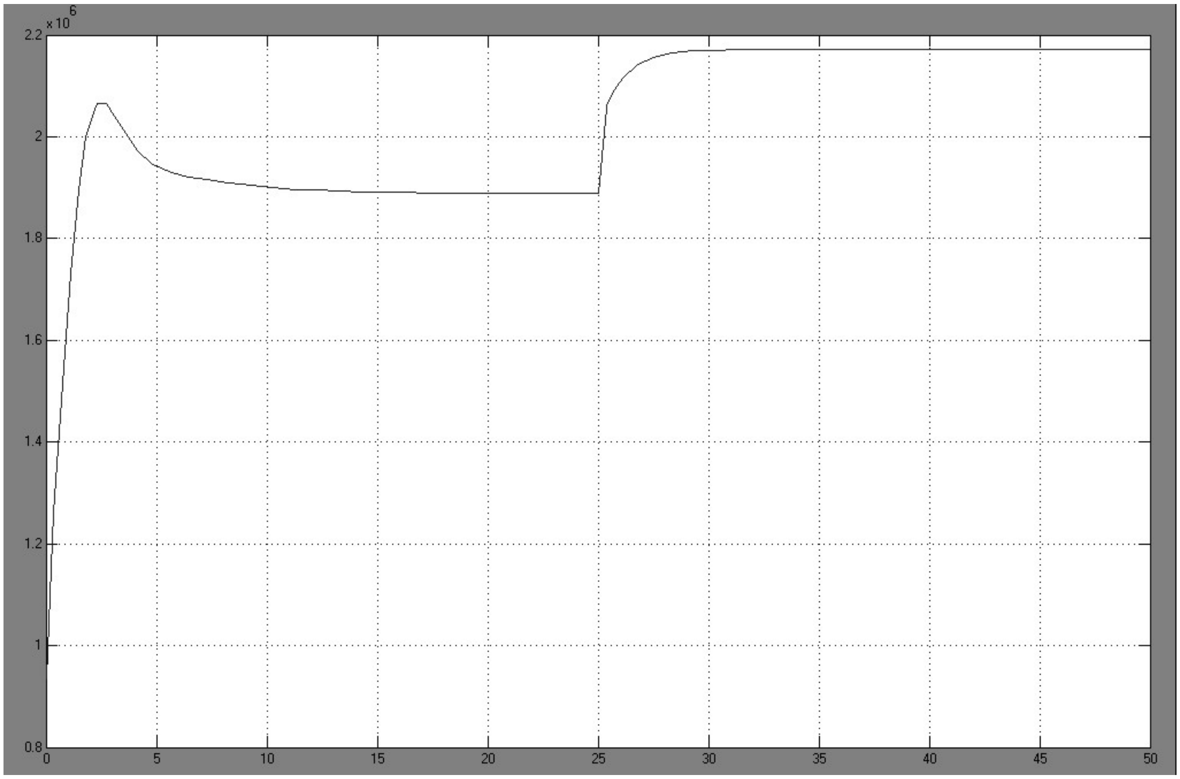

Figure 9: Mean effective pressure (Y-axis: mep in [pa]; X-axis: time in [s])

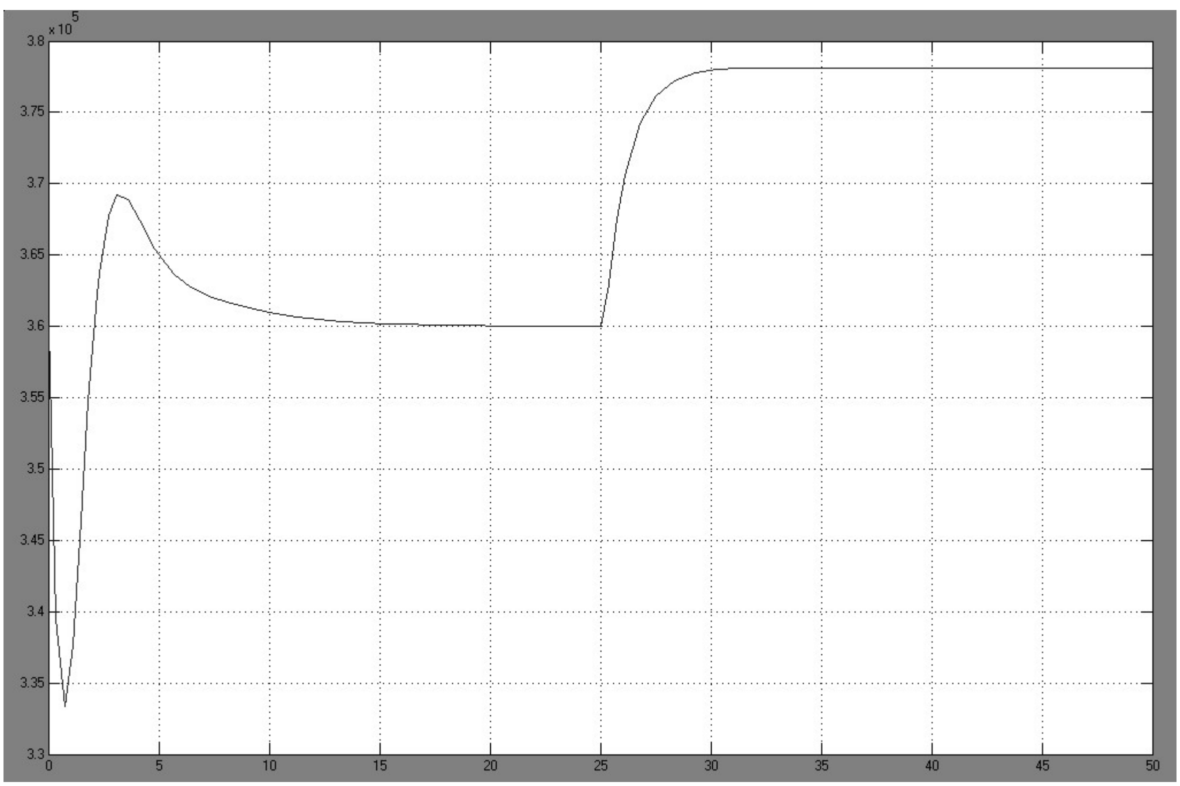

Figure 10: Charge air pressure (Y-axis: pressure [pa]; X-axis: time [s])

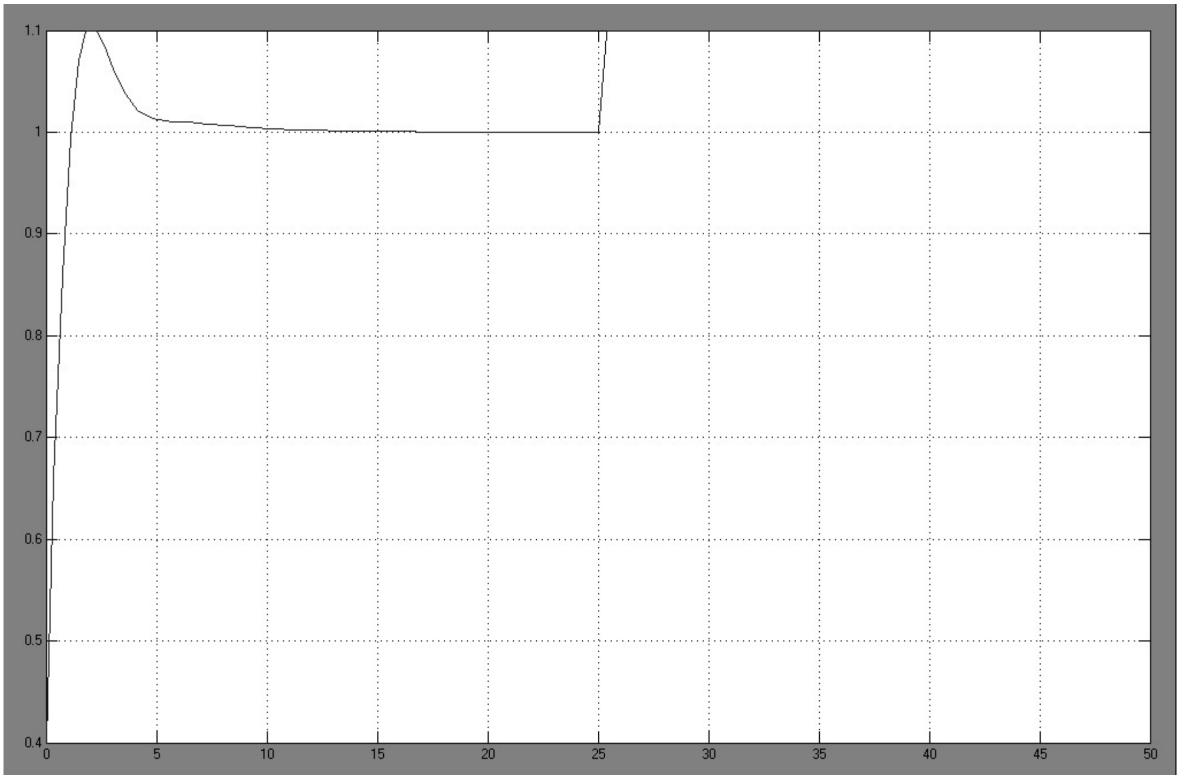

Figure 11: Signal output of governor (Y-axis: x [%]; X-axis: time [s])

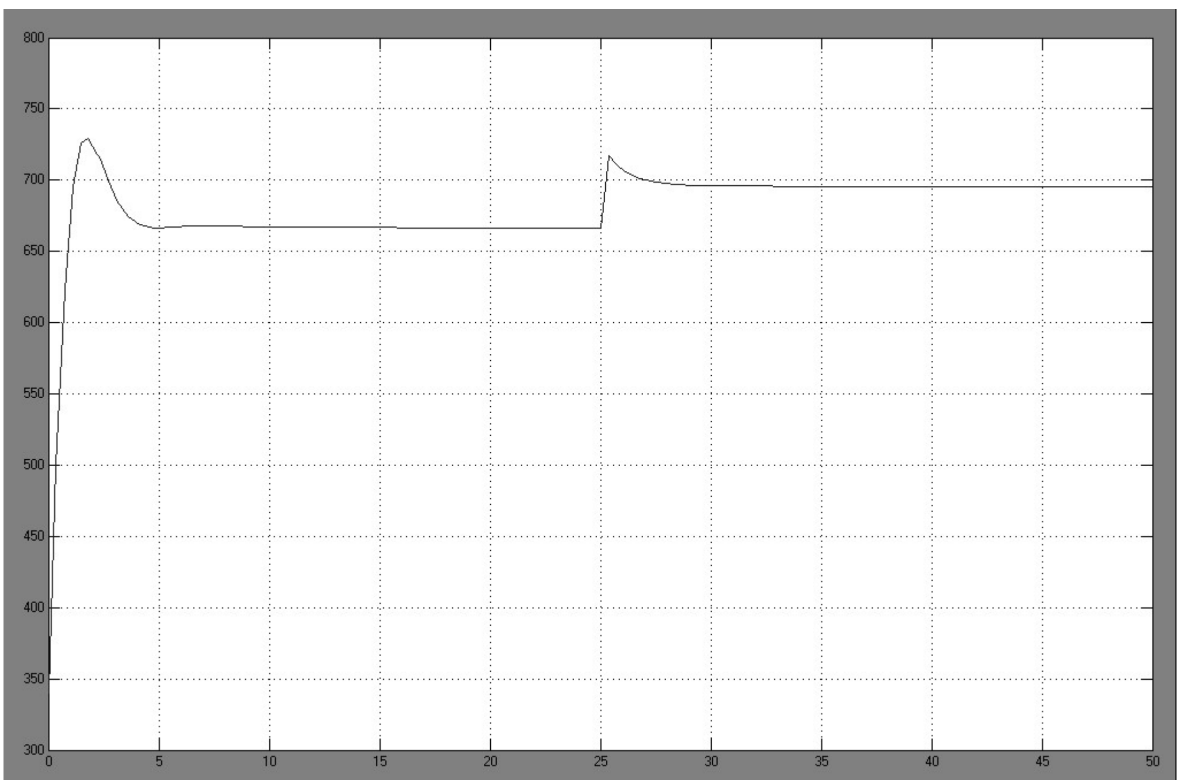

Figure 12: Heat released from cylinder (Y-axis: heat [kJ]; X-axis: time [s])

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