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Hydraulic System of Fuel Oil Supply to the Electronically Controlled Main Engine

Abstract

Hydraulic system of the fuel oil supply to the electronically controlled main engine resolve the problem of the optimized control of the engine which act beneficially to the specific fuel oil consumption and a reduction of the NOx emission. The assumed numerical example is giving insight to the power consumption for the hydraulic control of the fuel oil injection and exhaust valve operation for the two stroke marine engine. The calculated power is higher for the fuel oil pressure booster operation compared to the exhaust valve actuator for about 88%, which is resulted due to the higher operating pressure of the fuel injection to the cylinder, versus lower operating pressure required for the opening of the exhaust valve.

Key words: two stroke marine engine, hydraulic control, fuel oil power booster, exhaust valve power.

1. Introduction

The computer tunings and modeling of the marine two stroke diesel engines were not fully possible without changing of the fuel injection and exhaust gas valve timing adjustment. The changing of those parameters were difficult to achieve with the classical camshaft drive as the cams are in the fixed geometry and position of the camshaft. The cam geometry is not the only thing that affects optimal consumption of the two stroke marine engine, but it is one of significance [1, 2]. Comprehensive research also takes into account other parameters, some of which are listed here, which relate to the optimal fuel consumption and reduction of NOx emissions, such as fuel injector nozzle performance [3, 4]. The mathematical spray models of the fuel oil injector nozzle parameters and their validation studied in the literature under [5, 6], were concluded that implemented models showed very good agreement with the experimental results with respect to their vapor penetration and the spray shape what is related to the NOx emissions. The quasidimensional models of the liquid fuel evaporation and combustion in the diesel engine studied under [7, 8], consists of many experimentally obtained coefficients which influences chemical reaction dynamics of the liquid fuel evaporation and combustion inside the cylinder of a diesel engine. The improved numerical model is given in the literature under [9], without numerical iteration calculation. That study was going deeper to developed software for the numerical model of the evaporation and fuel oil combustion process. The ignition delay and its effect to the numerical model studied under [10, 11]. The proper definition and calculation of the same can properly predict the change in the cylinder pressure and cylinder heat release of any diesel engine, what are again related to the optimal fuel consumption and lower NOx emissions of the marine diesel engines. The above process may be optimized by multilayer perceptron, with achieving lowest specific fuel consumption and minimization of the cylinder process highest temperature for the reducing NOx emission [12, 13]. The optimization processes may vary by applied techniques which may include genetic algorithm approach [14-16], neural network application [17-20], program solution with programming languages, such are Modelica [21, 22]. However the comprehensive optimization of the fuel oil consumption and the reduction of the NOx emission is not achievable without electronically controlled fuel pressure and injection timing at the wide range operating what is briefly described in the below text.

2. The system of the fuel oil injection MAN ME marine two stroke engine

The concept of the electronically controlled engines replaces cam shaft drive of the high- pressure fuel pump injection and exhaust valve control by the hydraulically operated FIVA control valve, which is controlled by the electronic control system in the hydraulic power unit [23]. The FIVA (Fuel Injection-Valve actuation), is operating hydraulic fuel, oil pressure booster which is now acting as a fuel oil pump. The crank shaft angle is detected by the pickup sensor and tachometer which sends the signal to the ECU unit, (Engine Control Unit). The ECU control function is related to the plant control system, the safety system, supervision and alarm system, and is directly connected to sensors and actuators [24]. The next in the control line is CCU (Cyilnder Control Unit) which is connected to the components to be controlled and activated. The Control hierarchy is given in the Figure 1.

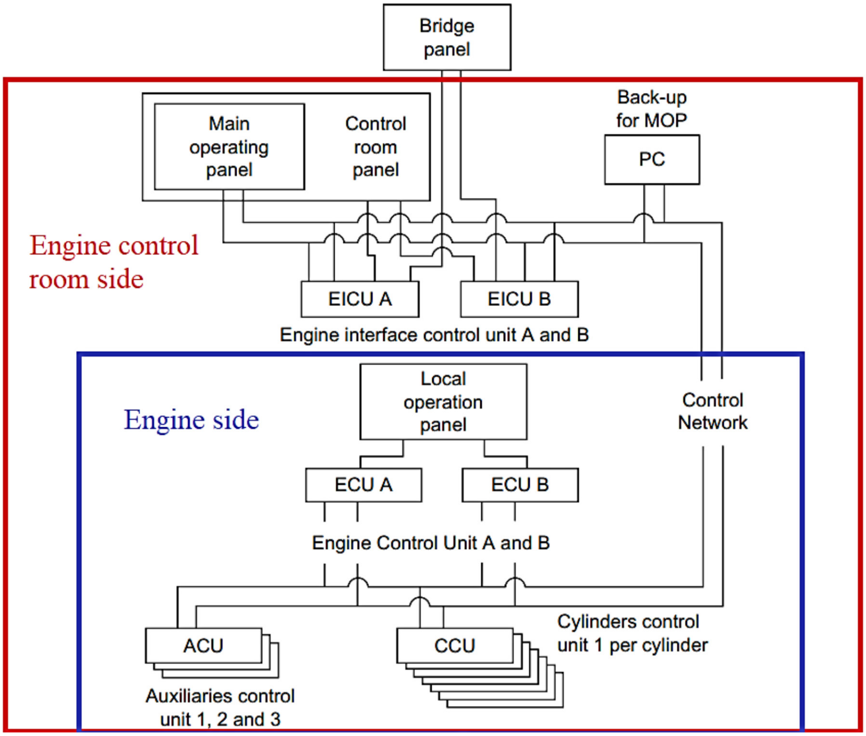

Figure 1. The control hierarchy of the ME engines.

The fuel oil pressure booster consists of a plunger powered by a hydraulic piston which is activated by the oil pressure from the lower side of the unit. The booster has bigger diameter compared to the plunger what enables higher operation fuel oil pressure, with the relatively lower control oil pressure. As per the Figure 2, the control oil pressure is controlled by FIVA valve which is an electronically controlled proportional valve. The exhaust valve is controlled by the same FIVA valve which operates the exhaust valve actuator. The shape of the exhaust gas valve actuator is different from those which is related to the fuel oil pressure booster as it is not required so high control lube oil operating pressure for the opening of the exhaust valve. The outlet control lube oil pressure to the exhaust valve is lower then the operating pressure from the FIVA valve from the lower side of the exhaust valve actuator.

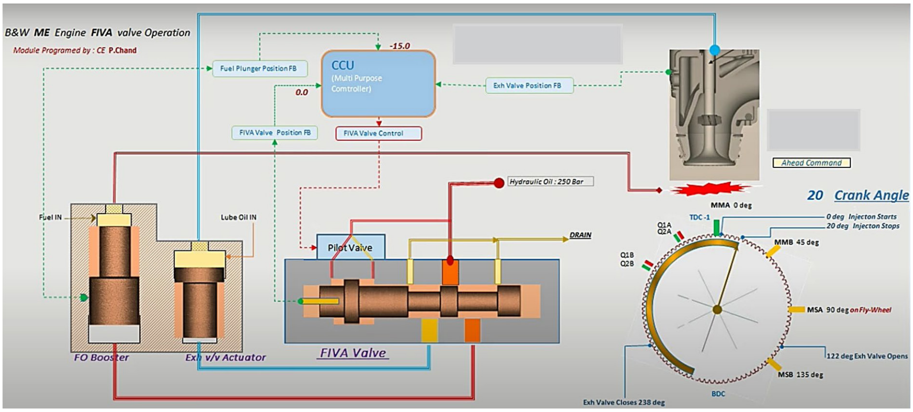

Figure 2. MAN FIVA valve operation, [25].

As the exhaust valve is closed by the 'air spring', for the proper operation of the exhaust valve it is required the control air pressure as well. The control hydraulic oil is taken from the engine system oil, which is filtered and pressurised by the hydraulic pumps. These pumps are driven by the main engine and for the starting purposes are provided two electrically driven pumps. The typical two stroke engine marine lube oil is given in the Table 1.

Table 1. Typical two stroke marine lube oil, [26].

	Methods	Units	2005	3005	4005
S.A.E. Grade			20	30	40
Density at 15 ºC	ISO 3675	kg/m^3	890	890	890
Kinematic viscosity at 40 ºC	ISO 3104	mm^2/s	70	105	150
Kinematic viscosity at 100 ºC	ISO 3104	mm^2/s	8.8	11.5	14.7
Flash point (COC)	ASTM D 92	mm^2/s	➢ or = 220	➢220	➢230
Pour Point	ISO 3016	ºC	-6	-9	-9
BN	ASTM D 2896	mgKOH/g	6	6	6

The system is very flexible in the control of the combustion process what was previously not achievable with the fixed camshaft angle. The flexibility of the fuel combustion process is mainly outlined by the following features which is a different running mode of the main engine. The different running mode of the main engine meets criteria such as; 'low fuel consumption' or 'limited exhaust gas emissions'. The flexibility of the system is given in the fuel consumption optimization at various loads, compared to the conventional cam shaft drive which were optimized performance in the range from 80-85% of MCR, [27]. The optimization of the large two stroke engines by electro-hydraulic control takes place in the varying of the P max which is achieved by the latter closing of the exhaust valve, what acts beneficial to the specific fuel oil consumption. In this type of engines the control oil pressure also varies with load variation, what is also beneficiary acting to the specific fuel oil consumption, what is shown in the Figure 3. There are three options of the engine tuning which includes high load tuning, partial load tuning and low load tuning. The optimal parameters were moved from the before mentioned area of 80 to 85% to the lower zones in the range from about the 55 to 85%. This is advantageous as nowadays charterers and shipowners do not prefer anymore higher speed of the ships due to very high fuel oil cost, but rather include one or two more vessels in line rotation which will make up delivered goods with the lowest specific fuel oil consumption.

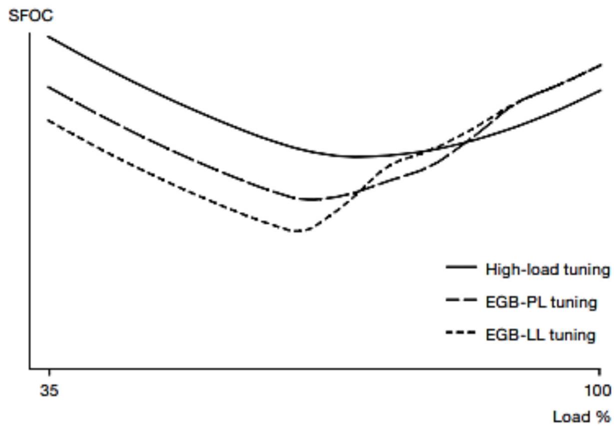

Figure 3. Influence of the engine tuning to the specific fuel consumption [23].

3. The preliminary lube oil power calculation for the two stroke engine required for the opening of the exhaust valve and fuel oil booster with given assumptions

The marine two stroke engine specific fuel oil consumption at 105 min^{-1} is 171 g/kW. The engine is equipped with the 8 cylinders and the total power is 19040 kW. The opening of the exhaust valve and fuel oil injection is according to the Figure 4, where ones per each stroke or one full revolution of the crankshaft exhaust valve is opened and fuel oil injected. The movement of the FIVA valve to the right implies that the control hydraulic oil is entering to the fuel oil booster via channel ‘’P’’ and moves it forward what overtakes the spring force of the fuel oil injector where after fuel oil injection into the cylinder takes place. While the FIVA valve is in the fully right position, the exhaust valve activator is connected to the sump tank by the channel ‘’T’’. Movement of the FIVA valve to the left position, causing that hydraulic control oil enters to the exhaust valve activator via channel ‘’P’’ what push exhaust valve downside to the fully opened position when the air force is overcome. At that moment the FIVA valve drain channel is connected with the fuel oil booster with the tank by the side’’.

The following assumption where taken for the control lube oil power calculations:
- Mass fuel oil consumption, $\dot{m}$ 171 g/kWh,
- Fuel oil and control lube oil density, ρ 1000 kg/m^3,
- Two stroke engine power 19040 kW,
- The cylinder number, i 8, and
- The fuel oil booster stroke at the engine load 0,04 m.

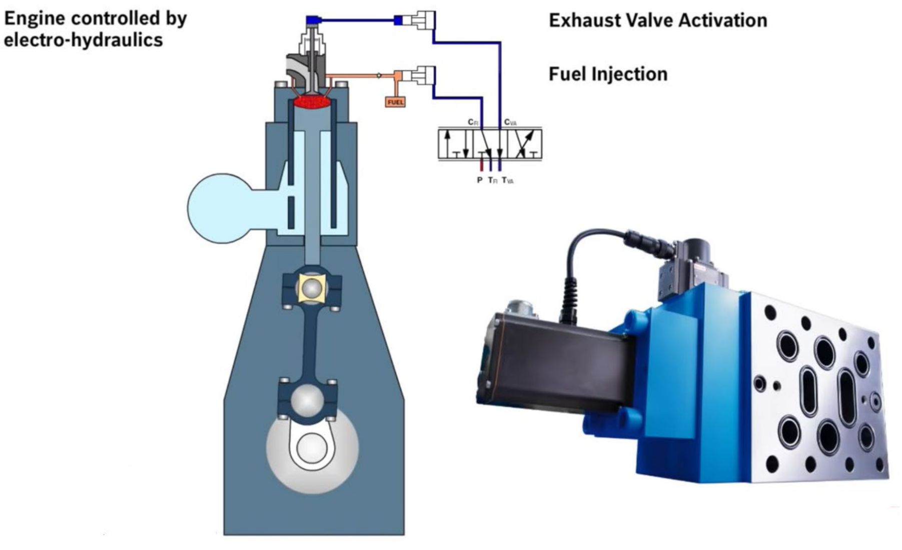

Figure 4. The lube oil distribution of the fuel oil injection and opening of the exhaust valve, [28].

3.1. The fuel oil booster diameter

The fuel oil consumption

$$\dot{m} = \frac{\dot{m} \cdot P}{1000}, \left[\frac{kg}{h}\right] \tag{1}$$

The fuel oil consumption per one cylinder

$$\dot{m} = \frac{\dot{m}}{i}, \left[\frac{kg}{h}\right] \tag{2}$$

Mass fuel oil consumption per cylinder

$$\dot{m} = \frac{\dot{m}}{3600}, \left[\frac{kg}{s}\right] \tag{3}$$

Mass lube oil rate to the fuel oil pressure booster

$$\dot{m} = \frac{d^2 \cdot \pi}{4} \cdot h \cdot \rho, \left[\frac{kg}{s}\right] \tag{4}$$

3.2. The mass flow rate of the control oil to the exhaust valve

The exhaust valve stroke 0,04 m while the exhaust valve actuator diameter is 0,02 m.

Mass lube oil rate to exhaust valve actuator

$$\dot{m} = \frac{d^2 \cdot \pi}{4} \cdot h \cdot \rho, \left[\frac{kg}{s}\right] \tag{5}$$

3.3. The control oil pump power per cylinder unit

The efficiency of the hydraulic pump with inclined plane is 0,9. The net delivery high of the same is 1800 m. The main engine speed 105 min^{-1}

The ideal power of the hydraulic pump is [29]:

$$P = \frac{q \cdot \rho \cdot g \cdot z \cdot n}{1000 \cdot 60 \cdot \eta}, [kW] \tag{6}$$

The substitution equation (4) to (6) is the power which is related to the fuel oil booster per cylinder unit and exhaust valve activator

$$P = \frac{d^2 \cdot \pi \cdot h \cdot \rho \cdot g \cdot z \cdot n}{1000 \cdot 60 \cdot 4 \cdot \eta}, [kW] \tag{7}$$

The Figure 5 shows the power variation of the control oil pump related to the fuel oil delivery rate and exhaust valve opening rate of the main engine. As per given assumptions, fuel oil pressure booster consumes more power than the exhaust valve in the all observed ranges of the engine running point which were taken in the output percentage. The high power consumption is related to the higher pressure force which has to be delivered to the injector to overcome the spring pressure and to disperse the fuel oil into the fine spray inside the cylinder which is required for the proper combustion process inside the cylinder, [30] and NOx control, [31, 32]. The fuel injector nozzle geometry and change of Reynolds number by changing of fuel oil temperature has impact to the pressure of the hydraulic power pump unit but it is not relevant as such. The lower viscosity pulls higher power at the pump side, however the viscosity of the fuel oil to the nozzle is more related to the proper spraying inside the cylinder and effective combustion process what again implies efficient fuel oil consumption and NOx emissions.

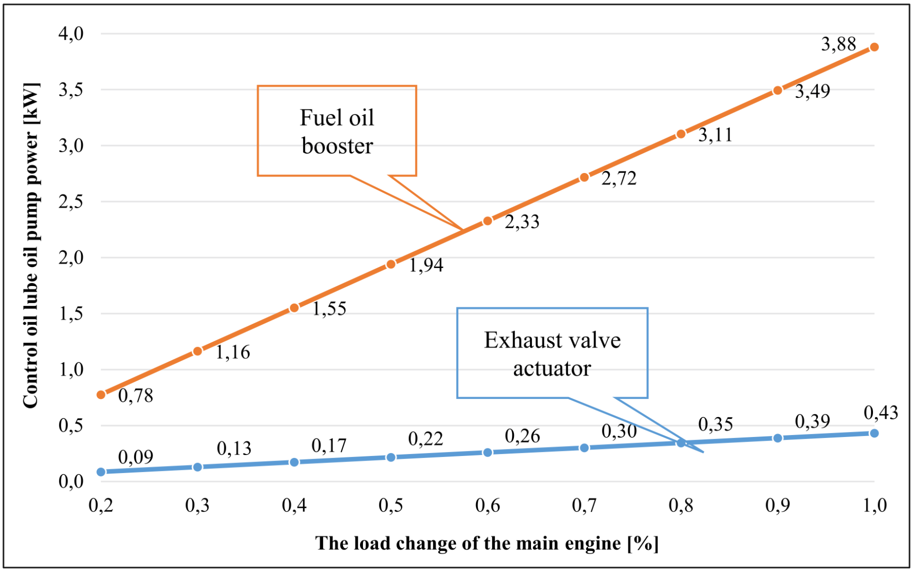

Figure 5. Control oil lube oil pump power versus main engine load change per cylinder unit.

4. Conclusion

The advantage of the hydraulically control two stroke marine engines are obvious in the sense of the optimized fuel consumption control in the wide running range and reduced NOx emission, what was not the case compared to the classical camshaft drive engines. The achievement of such control is carried out by the variable timing of fuel oil injection to the cylinder, variable fuel oil pressure control and with the adjustable angle of the exhaust valve open and close timing control. The classical camshaft drive engines had optimized engine consumption at the 80-85% of the MCR, while electronically controlled engines a have wider optimized range of fuel oil consumption and Nox emissions in the range from about 55 to 85%.

In this manuscript, it is roughly given, with the assumed parameter only variation in the power which is required for the operation of the fuel oil booster pump and exhaust valve. As per given result, it is obvious that power for the operation of the booster pump is higher about 88% compared to the power of the opening of the exhaust valve. This is caused due to lower operating pressure required for the exhaust valve operation compared to the fuel oil injection pressure of the fuel oil pressure booster.

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