Waste Heat Recovery for a Diesel Tractor Engine: A Comparison Study

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Abstract. One of the major concerns of nowadays is related to the Waste Heat Recovery (WHR) from the thermal engines, especially regarding the high rating power engines. For Diesel tractor engines, long time operated at high loads, the amount of the saved energy is linked to a significant fuel economy. The proposed study is based on the AMESIM simulation code and performs a comparison between two methods of heat recovery: the one using a supplementary turbine (turbo-compound method) and that using an Organic Rankine Cycle (ORC). The tested engine is an IVECO Cursor 10 Diesel engine type, equipping CASE tractors designed for agricultural activities. The energy savings provided by the engine exhaust gas heat recovery could be further used in supplementing the engine traction, either to be delivered to the auxiliary systems, leading in both cases to a decrease of the fuel consumption.

Key words: Diesel tractor engine, Waste Heat Recovery, Turbo-compound, Organic Rankine Cycle, fuel consumption decrease.

1 Introduction

By nowadays the society development has reached unprecedented limits comparing to the last decades, but its actual features could not be replaced by the efforts to reduce the consumption of the resources. Therefore human kind is concerned to increase the efficiency of all the processes, to reduce fuel consumption, to recycle materials etc. On this issue, the energy recovery from the internal combustion engines, especially from the heavy ones, with high costs is a dominant preoccupation in the present.

The waste heat recovery (WHR) from the vehicles engines could be classified in two categories: engine-power train-applied or engine-bottoming technologies, depending if they are directly applied or retrofitted to the engine-power train system, or if they recover wasted engine energy. These methods will allow the transfer of the recovered energy to the propulsion system or to the auxiliary systems of the engine. For heavy-duty Diesel engines turbocharging is a necessity to increase the engine power. Thus, any modification of the exhaust system would affect the operation of this group, as of the entire engine assembly.

Speaking about the multiplicity of the WHR methods, two of them play a significant role: the insert of a supplementary power gas turbine along with the one motioning the turbocharging group (the turbocompund method) and the use of a new energy circuit to transfer this energy to another work agent applying a Rankine, or an Organic Rankine Cycle (RC or ORC).

Heavy engines represent a key issue in the research studies regarding the energy recovery. The accent in [1] is put on the HDDE (Heavy-Duty Diesel Engines), on both automotive and stationary applications. Analyzing studies developed by the greatest companies, like Cummins, BMW or Honda reveal a multitude of results without a conclusion suggesting a certain beneficiary system. There are work papers proposing even more sophisticated energy recovery methods, such as the use of the Kalina Cycle [2], which uses a blend of two miscible fluids, ammonia and water for instance, participating in various fractions in order to ensure a better overtake of the waste energy.

Once the technical aspects are simpler in case of using the turbocompund method (designing a new turbine with the natural result of influencing the engine operation), more problems are to occur by applying the Rankine Cycle release. Authors of paper [3] identify no less than 6 different ways to implement such an energy recovery system. Additionally, they are analyzing 8 different types of cycle working fluids. Paper [4] expresses more than 30 sorts of fluid agents. Other authors also investigate several types of agents and how to get to the best choosing mode of them [5]. Their results are inconclusive referring to all the involving criteria and similar comments are given also by [6]. Authors of paper [7] went even far, by searching among 9 cycle agents from the thermodynamic results and the presumed costs point of view. As expected, after using a multi-objective optimization process, the conclusions showed that a comprehensive result remains highly difficult to be obtained.

The energy recovery for the engines with low temperature combustion (LTC) is presented in paper [8]. In case of an engine fueled with lean mixtures of natural gas and by split-injection of Diesel fuel, using also various fractions of exhaust gas recirculation (EGR) and various values of injection timings, an increase of 18% of the rating power was obtained, together with the drop of the emissions.

Another category of studies refers to the control of an ORC system [9,10,11]. These papers highlight numerical models regarding the possibilities to manage these control options in order to increase the engine performances. Paper [12] reveals a theoretical and a comparative study between using the two above-mentioned energy recovery methods, trying to identify also the optimal adjustments (corresponding to the injection timing, to the combustion period, to the designing elements of the turbine) for the improvement of the engine operation. The main conclusions were related to the fact that is no possible to increase the engine efficiency for all the analyzed regimes (the break specific fuel consumption – BSFC has opposite variation from one to another of the regimes), meanwhile highlighting a slight benefit when using the turbocompund method.

2. Iveco Cursor 10 Diesel engine simulation using AMESIM numerical code

Iveco engines are worldwide spread into the mechanical agriculture industry, being appreciated for their technical qualities and reliability. The group of 6-cylinders versions is formed by the following types: Cursor 9, 8.71 l swept volume, rating between 200 and 305 kW, Cursor 10, 10.3 l swept volume, rating between 265 and 315 kW, Cursor 13, 12.9 swept volume, rating between 325 and 427 kW and Cursor 16, 15.9 swept volume, rating between 480 and 570 kW. This engines family easily covers a wide range of agricultural machinery size.

The proposed engine to be analyzed is Cursor 10 -400 with characteristics presented in Table 1. and different versions of this engine are used also for commercial trucks.

Tuble I Englide Cursor To Too specifications				
Maximum power [kW/hp]	294/400			
Rated speed [rpm]	2100			
Number of cylinder	6			
Compression ratio	16.5:1			
Bore [mm]	125			
Stroke [mm]	140			
Cubic capacity [1]	10.38			
Fuel system	Direct injection mechanical			
Admission	turbocharged			

 Table 1 Engine Cursor 10 -400 specifications

One of the basic requests of these machines is their operation at high load of the engine. Due to the fact that the total weight of the vehicle is not an important issue and for certain reasons being brought at a minimum value, the design of a supplementary system for the energy recovery should not become a difficult task. In paper [1] is mentioning that an agricultural tractor engine should operate for 80% of its period at 80 to 100% load and covering a speed within 80 and 100% of its nominal value. With the elements listed in Table 1, the AMESIM model is presented in Fig. 1.



Fig. 1. AMESIM model for Cursor 10 Diesel type engine

The model is formed by several elements describing the different parts of the engine. The cylinder, as an important component of the engine is detailed in Fig. 2.



Fig. 2. Piston-cylinder system

The processes developing inside the cylinder are defining the specificity of Diesel engine. Therefore, the gas exchange part of the procedures, given by the cams and the valves motion, the fueling system formed by an injector delivering the fuel dose upon a characteristic well-established by the user and the crankshaft curve with the afferent connecting-rod, transforming the translation motion into rotation, all these elements have to be added into the model. Inside the cylinder the gas exchange processes are defined by the finite volumes hypothesis and the air-fuel mixture formation, the autoignition delay and the combustion are those proposed by Chmela [14] and implemented using the AMESIM numerical code.

Related to the turbocharging processes the models proposed by AMESIM are used, by applying its operating maps. Initially, an engine model has been performed so that the chosen values for this paper could be obtained. At 2100 rpm the engine rated a power of 294 kW, corresponding to an injected fuel cycle dose of 151 mg. This is further related to a fuel consumption of 56.5 kg/h and consequently to a BSFC of 192 g/kWh, a value confirmed by [13]. In-cylinder pressure and temperature variations are highlighted in Fig. 3, with the maximum corresponding values of 18 MPa and 2500 K.



Fig. 3. In-cylinder pressure and temperature variations

Among the other characteristic variables corresponding to the top rating power, the turbocharging pressure of 0.25 MPa, the turbocharging group speed of 86400 rpm, the exhaust gas flow of 384 g/s and the exhaust gas temperature of 905 K, obtained by the

charge combustion under a relative air-fuel ratio of 1.54, all these are also mentioned in the model.

The engine operation at full load for an excessive long period is less probably, therefore an 80% load coefficient regime has been adopted, for a lower than 1800 rpm speed. At this speed the rated power is 240 kW, thus a 200kW power will be used for the chosen engine operating regime. This power is obtained using a cycle fuel dose of 118 mg, respectively a fuel consumption of 38 kg/h and a BSFC value of 190 g/kWh. The turbocharging group speed is 79000 rpm, the exhaust gas reaching a temperature of 809 K, while the gas flow measures 308 g/s and the relative air-fuel ratio is positioned at 1.86. In this scenario the top firing pressure becomes 17.5 MPa and the in-cylinder temperature is reaching its top value of 2580 K (see Fig. 4.).



Fig. 4. In-cylinder pressure and temperature variations for the operating regime

3. WHR using the power turbine (the turbocompound method)

The system modeling in this case is performed by adding supplementary elements to the initial version of the model, corresponding to the power turbine issues (see Fig. 5.)



Fig. 5. The power turbine model

The supplementary power turbine is fueled by the exhaust gas flow in parallel with the turbocharging group turbine through a flap system adjusting the gas flow value. Downstream the main turbine a control volume is positioned and the power turbine is connected to a constant speed generator. The analysis of the system operation will be followed as a result of varying the flap flowing section and the power turbine speed.

The obtained results

For the given load regime, the simulation for the energy recovery used a parametric computation of the throttle opening degree toward the supplementary turbine (Table 2).

Analyzing the obtained data it appears that the supplementary turbine did not significantly influence the engine performances, because the relative air-fuel ratio is very high (1.8) related to the original engine design. By reducing the relative air-fuel ratio down to 1.36, the power delivered by the turbine did not decrease. The throttle opening over 60% did not lead to a gain of power for the supplementary turbine, moreover, a working limit of the gas flow corresponding to this speed level being reached. The obtained results concerning other speeds did not conduct to significant differences comparing to the references either.

Throttle	Turbocompressor	Engine	Relative	Exhaust gas	Turbine power
opening	speed [rpm]	power	air-fuel	temperature [K]	[kW]
[%]		[kW]	ratio [-]	_	
20	71771	202	1.6	861	0.48
40	62914	200	1.36	920	3.32
60	62341	199.7	1.34	926	3.69
80	62244	199.7	1.34	927	3.72
100	62244	199.7	1.34	927	3.73

Table 2 Turbocompound results at 80000 rpm speed for the power turbine

4. WHR using a system based on ORC

This energy recovery method uses the residual heat contained by the exhaust gas downstream the turbocharging group turbine, transferring it to the Rankine installation fluid so that it could be heated and vaporized. The entire system is presented in Fig. 6.



Fig. 6. The model for the ORC system

The fluid agent is vaporized, overheated and than is expended into a turbine. The system needs also a condenser to bring back the agent into the liquid phase. In what concerns the heating and the vaporizing part of the installation, the following elements were used (see Fig. 7).

In order to simulate the exhaust gas flow, a boiler has been used, to transform three signals in specific parameters for the exhaust gas, the mass gas flow, its temperature and composition, as given in the previous model. The exhaust gas will pass through a heat exchange system formed by 5 consecutive elements, which in fact simulate a single exchange system, but due to the need of numerical computation convergence it appears preferable to be divided. Concerning the organic fluid agent, a pump for the liquid phase is insert, to push the agent in the other side of the heat exchange system, on a reversed sense comparing to the exhaust gas. The organic fluid will be heated and vaporized while flowing through the heat exchange pipes.



Fig. 7. The boiler model

The gain of the mechanical work was possible using a turbine-type element, connected to an electric generator (as seen in Fig. 8.).



Fig. 8. The model for the power turbine

The connection of the boiler together with the turbine is made using two throttles, working simultaneously oppositely, meaning that when the overheating temperature increases, the first one opens the circuit towards the turbine and the second one is closing

the flowing circuit from the boiler. For the condenser a model has been proposed, very similar to the one used for the boiler (see Fig. 9). The condensation part is completed by a liquid-vapors separator and an expansion vessel.



Fig. 9. The model for the condenser

The obtained results

The heating and vaporizing processes are very unsteady from modeling point of view, because of the density and the heat transfer coefficient rapid variations which occur during this period. Thus, in the vaporizing initial period (see Fig. 10), the temperature is very much varying and very fast in correlation with the pressure until they stabilize inside the system.



Fig. 10. Agent temperature and pressure variations at pipe no.5 (detail)

After a great number of integration iterations the system is stabilized, the pressure reaching about 1.4 MPa, corresponding to a temperature of 268°C, inside the domain of the overheated vapors for ethanol (Fig. 11).



Fig. 11. Agent temperature and pressure variations at pipe no.5

For the turbine producing the mechanical work it has been usefully considered the use of a variable speed (during the computation period), as a function of the flap opening the receiving gas flow circuit, the final value of the speed reaching 6000 rpm. The power extracted by the organic agent from the exhaust gas is reflected by the exhaust gas temperature at the level of the boiler's 5 pipes (see Fig. 11). The heat transfer toward the organic fluid is relatively reduced, with a gain of power of about 5 kW. The power benefit of the turbine turns to an almost neglected value of 60 W, therefore the system efficiency is barely reduced.



Fig. 12. Exhaust gas temperature variation

Conclusions

To add a supplementary turbine in order to produce mechanical work to supply the vehicle traction does not represent a difficult technical task for the heavy-duty engines manufacturers. The problems occur when this turbine has to be connected to the propulsion system of the engine and when optimal operating regimes must be identified. Using the proposed model, a power extra gain of about 3.7 kW should reveal itself as a supplement for the engine 200 kW maximum rating power, without affecting its complementary performances.

The energy recovery applying a system based on the Organic Rankine Cycle consists in a highly difficult design project. The exhaust gas temperature could become a major constrain especially when choosing the working fluid agent. Correlations between the exhaust gas flow, its enthalpy and the geometrical dimensions of the turbocharging turbine, the functional characteristics of the agent pump and those of the organic agent expansion turbine must be established.

The proposed model has not been adjusted yet from all these points of view, therefore until now the results are not sufficiently satisfactory.

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