Effect of different working fluids on shell and tube heat exchanger to recover heat from exhaust of an automotive diesel engine

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Abstract: In this research, experiments were conducted to measure the exhaust waste heat available from a 60 kW automobile engine. The performance of an available shell and tube heat exchanger using water as the working fluid was conducted. With the available data, computer simulation was carried out to improve the design of the heat exchanger. Two heat exchangers were used: one to generate saturated and the other to generate super heated vapours. These two heat exchangers can be arranged in parallel or series. In series arrangement, the exhaust gas was first passed through superheated heat exchanger and then through the saturated heat exchanger. Whereas, in parallel arrangement, the exhaust gas was divided to pass through saturated and superheated heat exchangers. In both cases, working fluid was passed first through saturated heat exchanger and then through superheated heat exchanger. Computer simulation was carried out to investigate the effectiveness of the proposed heat exchanger for different working fluid like water, ammonia, and HFC-134a. It is found that with the exhaust heat available from the diesel engine additional 15%, 13% and 8% power can be achieved by using water, HFC-134a and ammonia as working fluid respectively.

Keywords: *Waste heat recovery, Organic Rankine Cycle, Diesel engine*

1. Introduction

Diesel engines represent a major kind of Internal Combustion Engine (ICE). These diesel engines have a wide field of applications and as energy converters they are characterized by their high efficiency. Trucks and road engines usually use high speed diesel engines with 220 kW output or more. Earth moving machineries use engines with an output of up to 520 kW or even higher up to 740 kW. Diesel engines are also used in small electrical generating units or as standby units for medium capacity power stations. However, Small air-cooled diesel engines of up to 35 kW output are used for irrigational purposes, small agricultural tractors and construction machines whereas large farms employ tractors of up to 150 kW output.

In general, diesel engines have an efficiency of about 35% and thus the rest of the input energy is wasted. Despite recent improvements of diesel engine efficiency, a considerable amount of energy is still expelled to the ambient with the exhaust gas. In a water-cooled engine about 35 and 30-40% [1] of the input energy is wasted in the coolant and exhaust gases respectively. The amount of such loss, recoverable at least partly, greatly depends on the engine load. Johnson [2] found that for a typical 3.0 l engine with a maximum output power of 115 kW, the total waste heat dissipated can vary from 20 kW to as much as 40 kW across the range of usual engine operation. It is suggested that for a typical and representative driving cycle, the average heating power available from waste heat is about 23 kW.

Since the wasted energy represents about two-thirds of the input energy and for the sake of a better fuel economy, exhaust gas from diesel engines can provide an important heat source that may be used in a number of ways to provide additional power and improve overall engine efficiency. These technical possibilities are currently under investigation by research institutes and engine manufacturers. For the heavy duty automotive diesel engines, one of the most promising technical solutions for exhaust gas waste heat utilization appears to be the use of a ‘‘Bottoming Rankine Cycle”. A Rankine cycle using water as working fluid is not enough

efficient to recover waste heat below 640 K [3]. The Organic Rankine Cycle (ORC) is a promising process to recover the heat from the exhaust of an engine and generate electricity from it [4, 5]. The ORC works like a simple Rankine steam power cycle but uses an organic working fluid instead of water. A certain challenge is to choose a suitable organic working fluid for the ORC. The working fluid should fulfil safety criteria; it should be environmentally friendly, and inexpensive. Another important aspect for the choice of the working fluid is the temperature of the available heat source. A question, which also has to be considered for using ORC, is whether an organic substance is really better than water as working fluid for a given task.

A systematic approach towards using an installation based on the Rankine Cycle in truck applications dates back to the early 1970s where a research program funded by the US Department of Energy (DOE) was conducted by Mack Trucks and Thermo Electron Corporation [6-8]. Under this program, an ORC system was installed on a Mack Truck diesel engine and the lab test results revealed an improvement of bsfc of 10–12%, which was verified by highway tests. During the following years similar research programs were performed by other research institutes and vehicle manufacturers. Aly [9] was able to produce 16% additional power from the exhaust of a Mercedes-Benz OM422A diesel engine by using R-12 as working fluid for the ORC. ORC systems with capacities from 750 to 1500 kWe were examined by Koebbman [10]. Recently, the solution of Rankine Cycle Systems has increased its potential competitiveness in the market even more [11, 12]. This is a result of technical advancements in a series of critical components for the operation of such an installation (heat exchanger, condenser and expander) but also stems from the highly increased fuel prices. Nowadays, the installation of a Rankine Cycle is not only considered as a feasible solution for efficiency improvement in heavy duty diesel engines for trucks [13, 14] but also for smaller application such as passenger cars [15].

In this project, experiments were conducted to measure the exhaust heat available from a 60 kW automobile engine at different speeds and loads. A shell and tube heat exchanger was purchased and installed into the engine. The performance of the heat exchanger using water as the working fluid was then conducted. With the available data, computer simulation was carried out to improve the design of the heat exchanger. The optimized model of the heat exchanger was then simulated to generate super heated vapour. Ammonia and HFC-134a is used as working fluids. Water is used as reference for comparison. The thermo physical properties of working fluids are compared and presented in Table 1. It is apparent that dry and isentropic organic fluids generally have much lower relative enthalpy drops during expansion than the water-steam mixture. Unlike water, most organic fluids suffer chemical decomposition and deterioration at high temperature and pressure. Therefore, an ORC system must be operated well below the temperature and pressure at which the fluids are chemically unstable. Most organic fluids have relatively low critical pressures and are therefore usually operated under low pressures and with much smaller heat capacities than water-vapour cycles. A suitable organic fluid must have a relatively high boiling point. Based on these features ammonia and HFC-134a are selected for the current study. Finally, power output from the turbine is calculated considering isentropic efficiency of real turbine[16, 17].

Table 1. Thermophysical properties of working fluids.

Parameter	H_2O	NH_3	HFC-134a
Molecular weight	18	17	102
Slope of the saturation vapour line	Negative	Negative	Isentropic
Enthalpy drop across the turbine (kJ/kg)	1570~900	725~70	55~22
Max. Stability Temperature (K)	None	750	450
Critical point (K)	647	405.3	374.15
Boiling point at 1 atm (K)	373	239.7	248
Latent heat at 1 atm (kJ/kg)	2256.6	1347	215.52

2. Experimental setup

The engine used in the current study is a four cylinder Toyota 13B diesel engine which is coupled with a water dynamometer. The specification of the engine is given in the Table 2. The schematic of the experimental setup is shown in Fig. 1. The engine run at different loads with variable speeds and exhaust temperatures were recorded to calculate available heat energy from the exhaust. Then the exhaust of the engine was connected to a shell and tube heat exchanger to study the performance of the heat exchanger and those data were used to improve the design of the heat exchanger by computer simulation.

Table 2. Engine specification.

Engine model	13B
Make	Toyota
Type of engine	4 cylinder charged water cooled diesel engine
Bore	102 mm
Stroke	105 mm
Compression ratio	17.6:1
Torque	217 N.m @ 2200 rpm

3. Heat Exchanger design

The data found from the experiment are used to optimize the design of shell and tube heat exchanger by computer simulation. Effect of important parameter of heat exchanger like radius of the shell, no of tubes, length of the heat exchanger, pressure drop is investigated and final model of the heat exchanger is proposed. The specification of the model of the proposed shell and tube heat exchanger is shown in the Table 3. Two heat exchangers are used: one heat exchanger is used to generate saturated vapor from the liquid working fluid and the second heat exchanger is used to generate super heated vapor from that saturated vapor. These two heat exchangers can be arranged into two configurations, parallel and series as shown in the Fig. 2.

Table 3. Heat exchanger specification.

Heat exchanger type	Shell and tube counter flow, hot fluid in tubes and cold fluid in the shell
Shell inside radius	35.4 mm
No of tube	18
Tube inside diameter	10 mm
Length of the heat exchanger	2 m

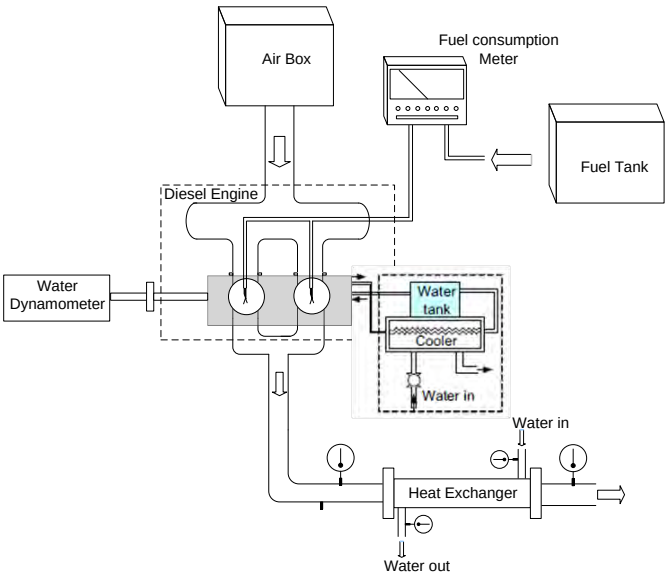

Fig. 1: Schematic diagram of experimental setup.

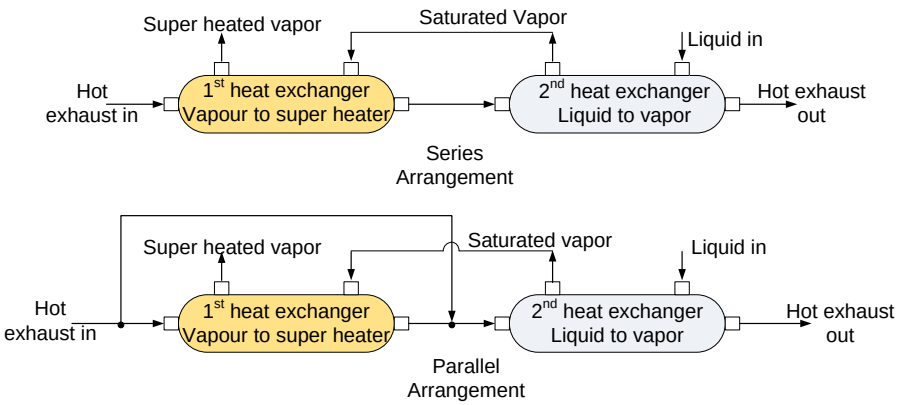

Fig. 2: Heat exchanger arrangement.

4. CFD Model

The optimized design of the shell and tube heat exchanger is modeled for heat transfer between hot and cold fluid in Flow Simulation which is CFD simulation module of Solidworks 2009. The computational mesh used to solve the model heat exchanger contained 109,992 cells. The cold fluid was considered to be liquid phase at 323K with corresponding saturation pressure for the second heat exchanger (Fig. 2) and saturated vapor at working pressure for the first heat exchanger (Fig. 2). The hot fluid considered as air with mass flow rate of 0.10215 kgs^{-1} and temperature of 938 K. The operating pressure of hot fluid is set to101.325 kPa and the cold fluid supply pressure and mass flow rate are varied. Steady and incompressible flow was assumed in all models. The Standard k- ε , a two-equation Reynolds-Averaged Navier-Stokes (RANS) model that is currently the most widely used for calculating flow problems has been used in this model.

5. Results

To design an effective heat exchanger for heat recovery from the exhaust of an engine, it is required to know how much energy is available in the exhaust. So some base line tests are performed. The exhaust gas temperature at various speed and engine power is presented in the Fig. 3. It is found from the figure that engine power and the temperature of the exhaust gases

for all three engine speeds show an approximately linear relationship. Exhaust gas temperature increases with increase of power output and speed of the engine. This indicates that heat recovery will be more viable for higher powers.

In the relationship between power and temperature there is a definite relationship between engine power and the amount of recoverable energy present in the exhaust gases. The relationship this time is not linear but there is still a general upward trend, revealing that, as the engine power increases, so does the amount of recoverable energy. This is clearly seen in Fig. 4. This finding is highly significant section in terms of the focus of this research project.

In particular, the potential applications which formed the original thinking behind this project are given credibility, in that the amount of energy which may be tapped is of an order that

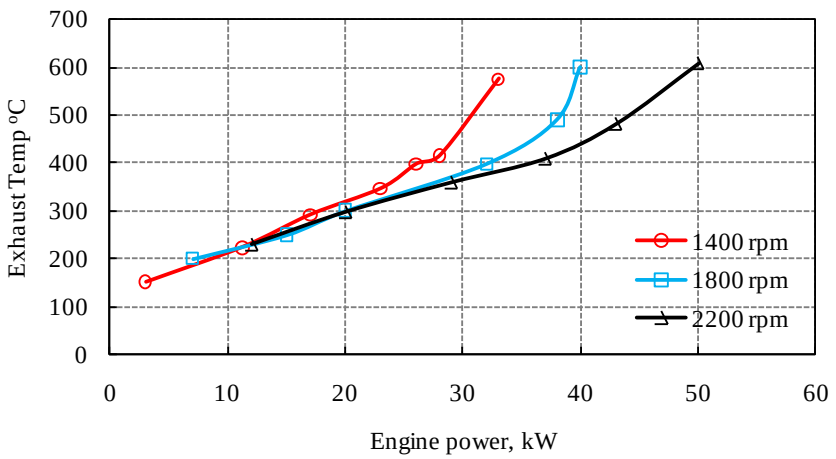

Fig. 3: Exhaust gas temperature variation with engine power from experiment.

justifies the attempt to capture and exploit it. For example, even if the results of just the lowest speed (1400 rpm) are considered, the potential to capture and use what is currently wasted energy, is extremely significant - the maximum recoverable energy for this speed is approximately 17 kW from the exhaust gas with the engine running at 33 kW (which is half the engine's power). Similarly, at 1800 rpm, a maximum value of approximately 21 kW was obtained from the exhaust gases, with the engine running at approximately 39 kW. At 2200 rpm the results show a maximum recoverable potential of approximately 23 kW when running at 45 kW. These results indicate that some 50% of the engine's running load is currently wasted but could be recoverable and converted to a usable form. All the above calculations were based on the abilities of a heat exchanger to be able to reduce the initial exhaust temperature at any particular speed and load to 50°C.

Based on the available data from the experiment, the heat exchanger design was optimized by computer simulation. Fig. 5 shows that the effectiveness of the heat exchanger decreases with the larger shell diameter for all three working fluids. Rubaiyat and Bari [18] found that there is no significant effect of working pressure on heat exchanger effectiveness. They also found that average pressure drop for different parameter of heat exchanger was about 250 Pa[18]. Effectiveness is higher for smaller diameter of the shell because of turbulent flow which facilitates the heat transfer. Heat exchanger effectiveness increases with the length of the heat exchanger as presented in the Fig. 6. It is found from the figure that after 1.6 m length the effectiveness increase is not very significant.

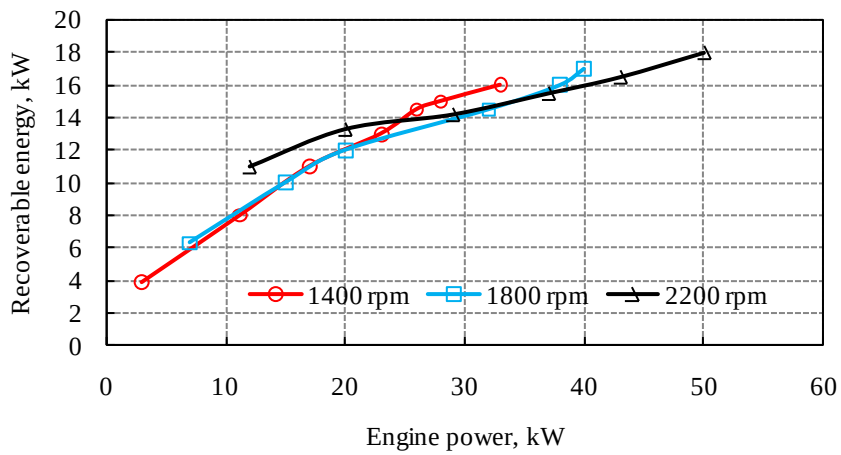

Fig. 4: Recoverable energy variation with engine power from experiment.

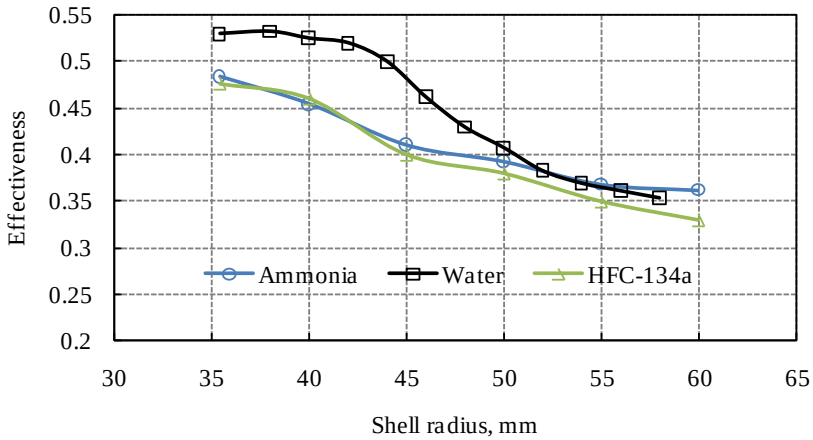

Fig. 5: Heat exchanger effectiveness vs. shell radius from CFD simulation.

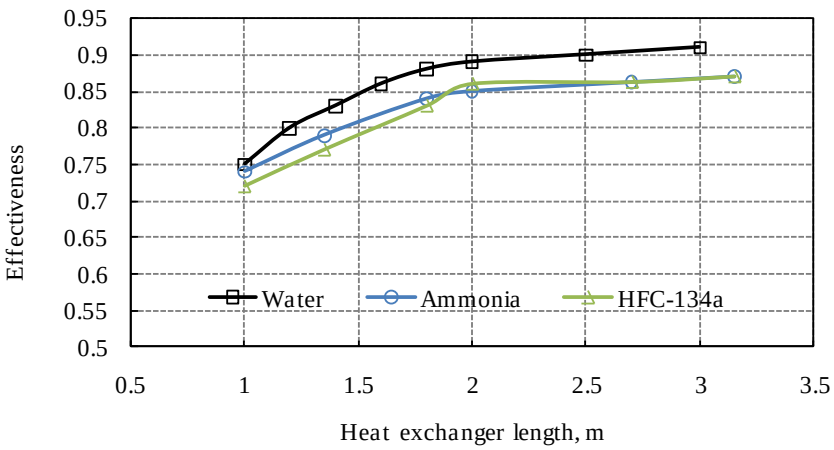

Fig. 6: Heat exchanger effectiveness vs. heat exchanger length from CFD simulation.

Extra power that can be recovered from the exhaust of the diesel engine with the proposed shell and tube heat exchanger model is presented in the Fig. 7. It is found that additional output power increases as the working pressure increases for both the parallel and series

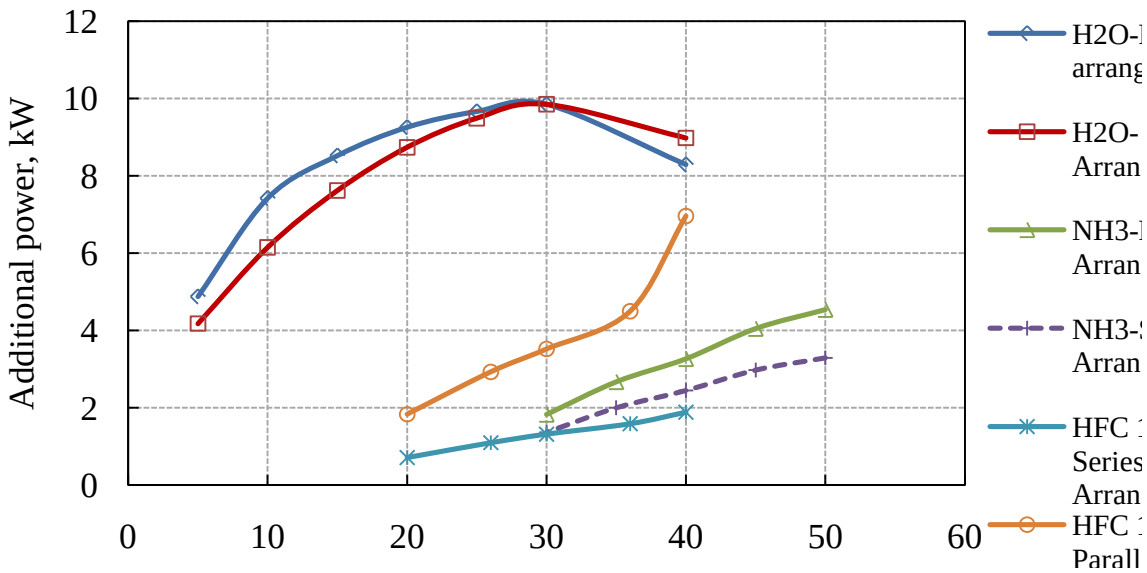

Fig. 7: Additional power output variation for different working fluids with working pressure from CFD simulation.

arrangement (Fig.2) of the heat exchangers for all three working fluids. This is because the condensing pressure was kept constant and as the working pressure increases the enthalpy drop across the turbine also increases. From the figure it is clear that water can recover heat most efficiently from the exhaust of the engine than the other organic fluids. This is because water has very high enthalpy drop across the turbine (Table 1) compared to other two organic fluids. Interestingly, it is found for water that higher power output can be achieved for parallel arrangement below 30 bar working pressure than the series arrangement whereas series arrangement can achieve higher power output above 30 bar working pressure than parallel arrangement. Maximum power output also achieved at the 30 bar working pressure. But other working fluids, ammonia and HFC-134a do not show any trend like that. For both ammonia and HFC-134a working fluid, parallel arrangement of the heat exchangers gives more additional power output. The proposed shell and tube heat exchanger can recover maximum 15%, 13% and 8% additional power from the exhaust of the diesel engine using water, HFC-134a and ammonia as working fluid respectively considering 70% isentropic efficiency of the turbine [16, 17].

6. Conclusion

The experimental and simulation results of the current project proved the concept of heat recovery from waste heat from the exhaust of diesel engines by using different working fluids. This research work shows that ORC can be a good option for waste recovery from diesel engines. This technique can increase the overall efficiency of diesel engine. Hence, this technology will reduce the fuel consumption and thereby will also reduce Green House Gases (GHG) and toxic emissions per kW of power produced. Additional 15%, 13% and 8% more power can be achieved with the proposed shell and tube heat exchanger by using water, HFC-134a and ammonia respectively.

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