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EFFECT OF DESIGN-PARAMETERS OF HEAT EXCHANGER ON RECOVERING HEAT FROM EXHAUST OF DIESEL ENGINE USING ORGANIC RANKINE CYCLE

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ABSTRACT
The heat from exhaust gas of diesel engines can be an important heat source to provide additional power and improve overall engine efficiency. The sizings of heat exchangers that can handle the heat load with reasonable size, weight and pressure drop are important factors for designing the heat exchanger for this type of application. In this project, experiments were conducted to measure the exhaust heat available from a 60 kW automobile engine at different speeds and loads. With the available data, computer simulation was carried out to design a shell and tube heat exchanger for two organic working fluids: Ammonia and HFC-134a. Geometric variables including length, number of tubes, and baffle design are all investigated separately. Upon investigating how these parameters influence heat exchanger effectiveness, an optimum design of a shell & tube heat exchanger is proposed. It is found that with the exhaust heat available from the diesel engine additional 13% and 8% power can be achieved by using ammonia and HFC-134a as working fluid respectively considering 70% isentropic efficiency of the turbine.

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

1. INTRODUCTION
Today's modern life relies heavily on internal combustion engines. Despite the fact that some degree of new technology has been introduced in recent years, the majority of vehicles are still powered by either spark ignition (SI) or compression ignition (CI) engines. CI engines also known as diesel engines have a wide field of applications and as energy converters they are characterized by their high efficiency. 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. Water or air-cooled engines are used for a range of 35-150 kW and unless strictly air cooled engine is required, water-cooled engines are preferred for higher power ranges. Earth moving machineries uses engines with an output of up to 520 kW or even higher, up to 740 kW. Marine applications usually employ engines with an output range of 150 kW or more. Trucks and road engines usually use high speed diesel engines with 220 kW output or more. Diesel engines are used in small electrical generating units or as standby units for medium capacity power stations.

In general, diesel engines have an efficiency of about 35% and thus the rest input energy is wasted. Despite recent improvements of diesel engine efficiency, a considerable amount of energy is still expelled to the ambient from 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 500°C [3]. Typical heavy duty diesel engine vehicle exhaust temperature ranges from 500 to 600°C [4] and they can vary depending on the size, speed and load of the engine. The Organic Rankine Cycle (ORC) is a promising process to recover the heat from the exhaust of an engine and to generate electricity from it [5, 6]. The ORC works like a simple Rankine steam power cycle but uses an organic working fluid instead of water. 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. Organic fluids are to be preferred to water when the required power is limited and the heat source temperature is low, as these fluids often have lower heat of vaporization. Furthermore, turbines for ORC can provide higher efficiencies at part loads as well and are usually less complex (1 or 2 stages, for an axial turbine) due to the lower enthalpy drop of the fluid [7]. There are certain criteria to choose a suitable organic working fluid for the optimum performance of 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 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 [8-10]. 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 [7] 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 [11]. Recently, the solution of Rankine Cycle Systems has increased its potential competitiveness in the market even more [12, 13]. 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 [14, 15] but also for smaller application such as passenger cars [16].

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 to the engine exhaust system. Experiment was conducted to validate the concept of extracting heat from the exhaust with water. Then with the available data, computer simulation was carried out to design a heat exchanger for ORC using two organic fluids namely ammonia and HFC-134a. The optimized model of the heat exchanger was then simulated to generate super-heated vapour. Ammonia and HFC-134a is used as working fluids. The thermo physical properties of working fluids are compared and presented in Table 1. The saturation vapour curve is the most crucial characteristic of a working fluid in an ORC. There are generally three types of vapour saturation curves in the temperature-entropy (T-S) diagram: a dry fluid with positive slopes (dT/dS), a wet fluid with negative slopes, and an isentropic fluid with nearly infinitely large slopes. 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 turbines [17, 18].

Table1:Thermo-physical properties of working fluids.

Parameter	*NH_3*	*HFC-134a*
Molecular weight	17	102
Slope of the saturation vapour line	Negative	Isentropic
Enthalpy drop across the turbine (kJ/kg)	725~70	55~22
Max. Stability Temperature (K)	750	450
Critical point (K)	405.3	374.15
Boiling point at 1 atm (K)	239.7	248
Latent heat at 1 atm (kJ/kg)	1347	215.52

2. EXPERIMENTAL SETUP

The engine used in the current study is a four cylinder Toyota 13B diesel engine for pick-up van. The engine 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 was tested 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

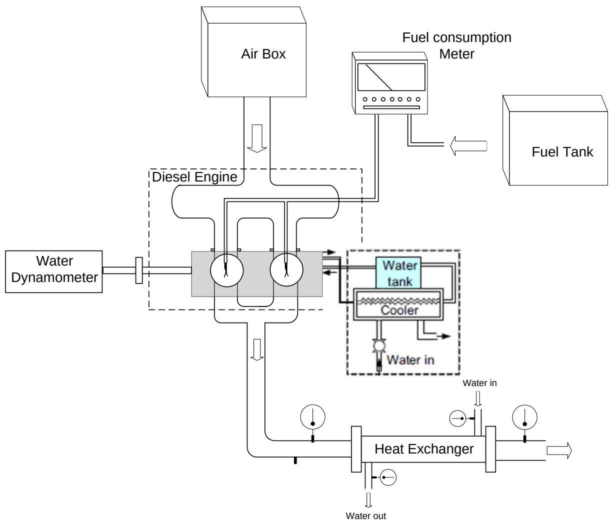

Fig 1. Schematic diagram of experimental setup.

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. Two heat exchangers are used: one heat exchanger is used to generate saturated vapor from the liquid working fluid namely vapor generator and the second heat exchanger is used to generate super-heated vapor from that saturated vapor namely super heater.

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 and saturated vapour at working pressure for the first heat exchanger. 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 pressure (from 20 to 50 bars) 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. 2. 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 is almost linear up to 30 kW. After 30 kW, the increase of recoverable energy with power increase is not that much significant because of higher losses associated with larger temperature gradient. 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. 3. 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 justify the attempt to capture

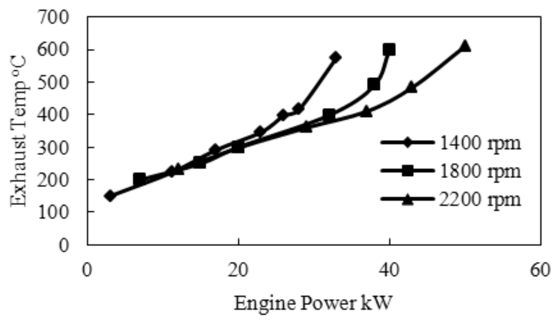

Fig 2. Exhaust gas temperature variation with engine load from experiment.

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. 4 shows that the effectiveness of the heat exchanger decreases with the larger shell radius for both working fluids. Effectiveness is higher for smaller radius of the shell because of turbulent flow which facilitates the heat transfer. Rubaiyat and Bari [19] found that there is no significant effect of working pressure on heat exchanger effectiveness for different parameters of the

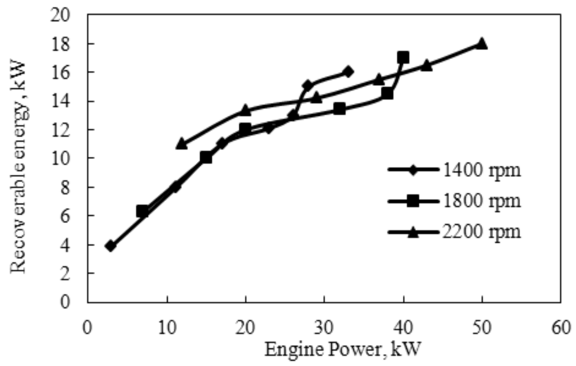

Fig 3. Recoverable energy variation with engine load from experiment.

heat exchanger. They also found that average pressure drop for different parameter of heat exchanger was about 250 Pa. Heat exchanger effectiveness increases with the length of the heat exchanger as presented in the Fig. 5. It is found from the figure that after 1.6 m length the effectiveness increase is not very significant. Fig. 6 represents the effect of no of tubes on

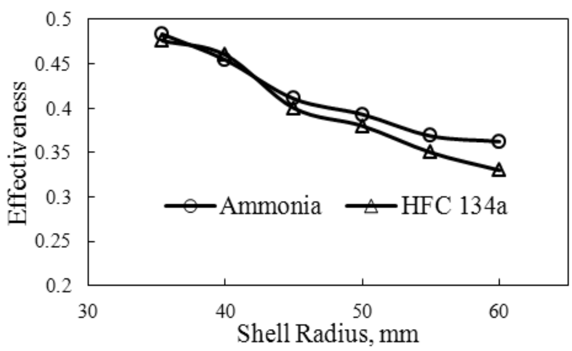

Fig 4. Heat exchanger effectiveness vs. shell radius from CFD simulation.

the effectiveness of the heat exchanger. It is found that for fixed radius as the number of tube increases, the effectiveness of the heat exchanger also increases. This is due to increase of heat transfer surface area. As the number of tube increases the effective area of the shell decreases and this causes an increase in velocity of shell side fluid. Thus increases the turbulence and facilitate heat transfer.

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 all two 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 ammonia can recover heat most efficiently from the exhaust of the engine than the other organic fluid, HFC-134a. This is because ammonia has very high enthalpy drop across the turbine (Table 1) compared to HFC-134a. The proposed shell and tube heat exchanger can recover maximum 13% and 8% additional power from the exhaust of the diesel engine using ammonia and HFC-134a as working fluid respectively considering 70% isentropic efficiency of the turbine [17, 18].

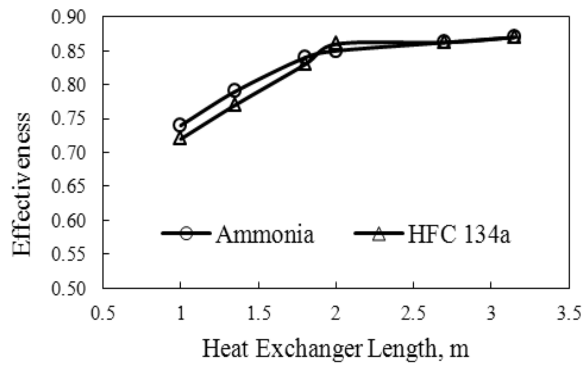

Fig 5. Heat exchanger effectiveness vs. heat exchanger length from CFD simulation.

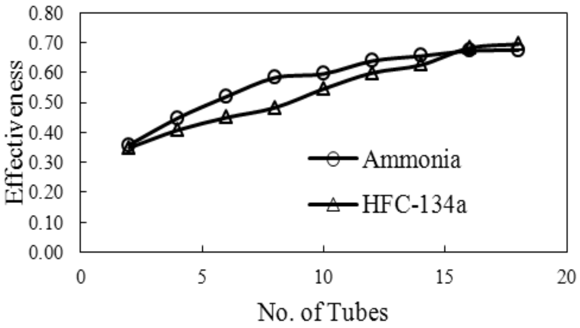

Fig 6. Heat exchanger effectiveness vs. heat exchanger tube no from CFD simulation.

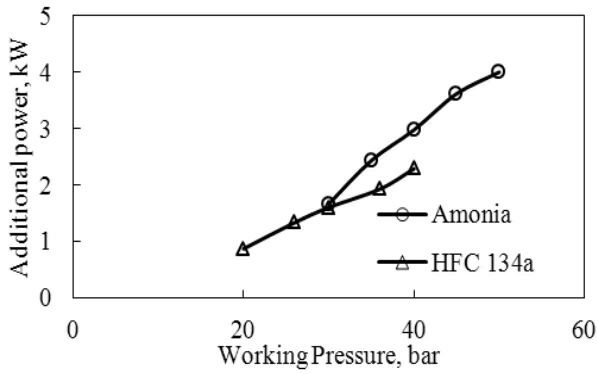

Fig 7. Additional power output variation for different working fluids with working pressure from CFD simulation.

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. A shell and tube heat exchanger is designed for this purpose by optimizing its different parameters. The specification of the proposed heat exchanger is presented in the Table3. From the simulation it is found that there is not much difference in the design of the two heat exchangers: super heater and vapor generator. So for the ease of manufacturing, the same design is used for both the heat exchangers. 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 13% and 8% power can be achieved with the proposed shell and tube heat exchanger by using ammonia and HFC-134a respectively.

Table 3: Heat exchanger Model specification

Heat exchanger type	Shell and tube counter flow, hot fluid in tubes and cold fluid in the shell
Shell inside radius	35 mm
No of tube	18
Tube inside radius	5 mm
Length of the heat exchanger	2 m

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