Review of organic Rankine cycles for internal combustion engine exhaust waste heat recovery

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Abstract: - Escalating fuel prices and future carbon dioxide emission limits are creating a renewed interest in methods to increase the thermal efficiency of engines beyond the limit of in-cylinder techniques. One promising mechanism that accomplishes both objectives is the conversion of engine waste heat to a more useful form of energy, either mechanical or electrical. This paper reviews the use of internal combustion engine exhaust waste heat recovery focusing on Organic Rankine Cycles since this thermodynamic cycle works well with the medium-grade energy of the exhaust. Selection of the cycle expander and working fluid are the primary focus of the review, since they are regarded as having the largest impact on system performance. Results demonstrate a potential fuel economy improvement around 10% with modern refrigerants and advancements in expander technology.

Keywords: Refrigerant progress, Internal Combustion Engine, Waste Heat Recover.

I. INTRODUCTION

Energy conservation in the world is becoming very important in recent years, especially the use of low grade temperature and small-scale heat sources. Energy extraction from industrial waste heat, biomass energy, solar energy, and turbine exhaust heat is becoming more popular. Organic Rankine Cycle is an effective way to convert these heat sources into electrical power. Organic Rankine Cycle offers the ability to deal with low temperature heat to generate power. The traditional Rankine Cycle which uses water as the working fluid needs much higher temperature heat source while Organic Rankine Cycle ORC can generate power at a much lower temperature. The heat source temperature can vary from 50 to over 250°C. In recent years a lot of research has been conducted around the world and many ORC systems have been successfully installed in different countries, especially in USA, Canada, Germany and Italy.

Organic Rankine Cycles offer power production from renewable, waste heat and law-grade heat sources like, geothermal energy, biomass, solar energy and waste heat from industry and thermal power plants. Furthermore, Organic Rankine Cycle can be used to recover energy from exhaust gases from power trains, improving the fuel consumption and reducing their impact on climate changes.

Organic Rankine Cycle and working fluids have been widely studied in different scientific articles. Some papers widely studied the usage of Organic Rankine Cycle ORC in different applications like waste heat recovery, geothermal power plants, biomass power plants and solar thermal power plants.

According to Roadmap 2050 from the European Climate Foundation 2010, the greenhouse emissions can be reduced by 80% in 2050. This target can be achieved through the modification of the current energy system and the following modifications should be accomplished by 2050:

• Increase effectiveness and reduce energy intensity of buildings by 950 TWh/year and of Energy industry by 450 TWh/year.

• Use electricity instead of fossil fuels for transportation and space heating.

• Shift to renewable energies and clean power generation (Wind energy 25%, PV1 19%, CSP2 5%, Biomass 12%, Geothermal 2% and Large hydro 12%).

• Increase the grid capacity and reinforce the inter-regional transmission lines.

The objective of this research paper is to explain the operating principles of the components used in waste heat recovery systems based on the Organic Rankine cycle from internal combustion engines.

There hasn't been a final conclusion published, which fluid and expansion device would be the favourable choice in a Rankine cycle for vehicular application. However, previous studies pointed out that these components are among the most important key factors for the system's performance.

II. METHODOLOGY

Considered configurations of Organic Rankine cycle in the review

Using an adaptable configuration corresponding to the heat sources is one of the methods to improve the performance of co-generation system. Various configurations have been presented for heat recovery using ORC. Some of them are simple, preheat, and regenerated Rankine cycle. The amount of recovered heat and consequently cycle output power for these configurations depend on their characteristics. In this study two different configurations are proposed for simultaneous coolant and exhaust gas heat recovery: Preheat and two-stage configurations.

2.1Preheat configuration.

The first configuration presented in this paper is conventional preheat Rankine cycle. As shown in fig.1, the working fluid pressure has increased through process 1-2, after that a preheater was used for waste heat recovery from the coolant. The working fluid was preheated in this heat exchanger (process 2-3) then, entered the evaporator and was converted to saturated vapour by absorbing heat from the exhaust gas (process 3-4). After that it entered the expander and generated mechanical power by expanding (process 4-5), this configuration is unable to absorb the total heat released by the coolant. Thus, an air-cooled heat exchanger should be used to reduce the temperature of coolant before returning the engine.

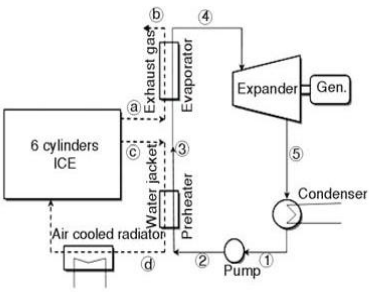

Fig.1: Conventional preheat Rankine cycle

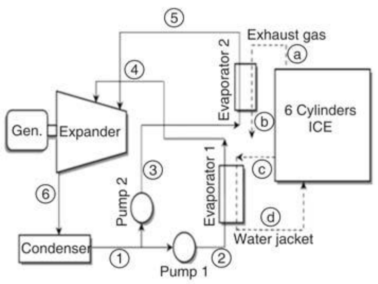

Fig.2: Two-stage configuration Rankine cycle

2.2 Two-stage configuration.

The second configuration is shown in fig. it is the system rarely used for heat recovery from ICEs until now. Two pumps and one expander with two inlets and one outlet, usually called dual expander, were used in this system. As shown in this figure 2, in this configuration, the working fluid flows in two different stages with different pressures after leaving the condenser (process 1-2 and 1-3). Low pressure stage relates to heat recovery from the coolant (process 2-4) and another one relates to heat recovery from exhaust gas (process 3-5). Finally, both low and high pressure flows expanded through a dual expander (process 4 and 5-6). Using this configuration, total wasted heat of coolant can be recovered and there is no need to air-cooled heat exchanger for coolant. Using two-stage configuration for Rankin cycle named as “dual pressure steam cycle” was recommended by Wang et al in 2009 for heat recovery from a cement production factory. They used a dual steam turbine for compression and power generation stage.

2.3 Heat Recovery from Internal Combustion Engines

An Internal Combustion Engine only converts about one third of the fuel energy into mechanical power. For instance, for a typical 1.4 liter Spark Ignition ICE, with a thermal efficiency ranging from 15 to 32%, 1.7 to 45 kW are released through the radiator (at a temperature close to 80 - 100°C) and 4.6 to 120 kW through the exhaust gas (400 - 900°C). The heat recovery Rankine cycle system is an efficient means for recovering heat (in comparison with other technologies such as thermo-electricity and absorption cycle air-conditioning). The idea of associating a Rankine cycle to an ICE is not new and the first technical developments followed the 70’s energy crisis. For instance, Mack Trucks (Patel & Doyle, 1976) designed and built a prototype of such a system operating on the exhaust gas of a 288 HP truck engine. A 450 km on-road test demonstrated the technical feasibility of the system and its economical interest: an improvement of 12.5% of the fuel consumption was reported. Systems developed today differ from those of the 70’s because of the advances in the development of expansion devices and the broader choice of working fluids. However, at the present time, Rankine cycle systems are under development, but no commercial solution seems to be available yet. Most of the systems under development recover heat from the exhaust gases and from the cooling circuit (Freymann et al., 2008). By contrast, the system developed by (Oomori & Ogino 1993) only recovers heat from the cooling circuit. Different architectures can be proposed to recover engine waste heat: The heat recovery system can be a direct evaporation system or a heat transfer loop system. In the first case, the evaporator of the ORC is directly connected to the exhaust gases. The advantage of such a configuration is the high temperature of the heat recovery, allowing higher cycle efficiency. In the second case, thermal oil is used to recover heat on the exhaust gases and is then directed to the evaporator. This second system acts as buffer and reduces the transient character of the ORC heat source, which simplifies its control. It also shows the advantage of avoiding hot spots in the evaporator, which could damage the organic working fluid. The expander output can be mechanical or electrical. With a mechanical system, the expander shaft is directly connected to the engine drive belt, with a clutch to avoid power losses when the ORC cycle power output is too low. The main drawback of this configuration is the imposed expander speed: this speed is a fixed ratio of the engine speed and is not necessarily the optimal speed for maximizing cycle efficiency. In the case of electricity generation, the expander is coupled to an alternator, used to refill the batteries or supply auxiliary equipments such as the air conditioning. It should be noted that current vehicle alternators show a quite low efficiency (about 50 to 60%), which reduces the ORC output power. As for the expander, the pump can be directly connected to the drive belt, to the expander shaft, or to an electrical motor. In the latter case, the working fluid flow rate can be independently controlled, which makes the regulation of such a system much easier. The control of the system is particularly complex due to the (often) transient regime of the heat source. However, optimizing the control is crucial to improve the performance of the system. It is generally necessary to control both the pump speed and the expander speed to maintain the required conditions (temperature, pressure) at the expander inlet. Performance of the recently developed prototypes of Rankine cycles is promising. For instance, the system designed by Honda (Endo et al., 2007) showed a maximum cycle thermal efficiency of 13%. At 100 km/h, this yields a cycle output of 2.5 kW (for an engine output of 19.2 kW) and represents an increase of the engine thermal efficiency from 28.9% to 32.7%.

2.4 Alternative Where Methods and the Importance of ORC

It is also possible to achieve WHR using other unique thermodynamic cycles. One such method is the open Brayton cycle, which requires only three components. Another common option is the Stirling cycle engine that includes a closed system comprised of a regenerator and cylinder, which contains both a displacement and power piston. More recently, increased focus has been placed on the development of Kalina cycle systems, which parallels an ORC in configuration with the addition of an absorber and flash tank . This cycle uses a variable composition mixture of ammonia and water as the working fluid. Similarly, supercritical carbon-dioxide systems have drawn attention in various WHR applications. While most WHR research focuses on thermodynamic cycles, thermoelectric (TE) devices offer a unique alternative, since they directly convert thermal energy into electrical energy. As previously elucidated, an ORC is the focus of most small-scale WHR efforts due to its simplicity and ability to operate efficiently between small to moderate temperature differences. Another primary advantage of the ORC is the use of widely available and affordable components because of the similarities between ORC and refrigeration systems. While no single WHR method is superior for every system size and waste heat source, the authors believe ORC's provide an attractive combination of efficiency and affordability for engine exhaust WHR.

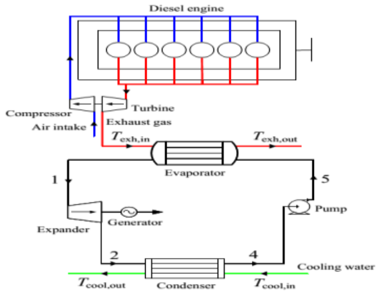

Fig.3: Construction of organic Rankine cycle

1.5 ORC System Improvement

Unlike Steam Rankine Cycle, the optimization of the Organic Rankine Cycle is quite limited, The limitations are mainly affected by the low heat source temperature. Using isentropic and dry fluids in ORC, the working fluid leaves the expander as superheated vapour and no attention is paid to the vapour quality at the end of the expander. The superheated vapour has a great advantage for turbo machine expanders which always get damages due to low vapour quality. The turbo machines expanders in ORC systems can have a much longer life spam than in steam cycles. To overcome the low vapour quality at the expander outlet, screw and scroll expanders can be used instead of turbo machinery expanders. Screw and scroll expanders have much better resistance for vapour quality than turbo machinery expanders. It follows that superheat is not recommended in cycles using dry and isentropic fluids [15]. To overcome the low vapour quality in last expansion stages in Rankine Cycle the superheat is recommended.

The layout of the Organic Rankine Cycle is much simpler than that of the Rankine Cycle. In the Organic Rankine Cycles, the water-steam drum is eliminated and a single heat exchanger can be used instead of the three-part heat exchanger (economizer, preheater and superheater). Reheating and turbine bleeding are not recommended for some working fluids while a recuperator can work as a preheater and be installed between the expander outlet and the pump outlet [15]. Figure 4 shows the cycle layout for an ORC using a recuperator and figure 5 shows the T-S diagram for ORC.

Following are the six main thermodynamic processes for an Organic Rankine Cycle uses a recuperator (internal heat exchanger IHE).

Process (1-2) Pump: the condensate working fluid is pumped from condenser pressure to evaporator pressure.

Process (2-3) Recuperator: heat transfer process between undercooled working fluid at pump outlet and superheated vapour at expander outlet.

Process (3-4) Evaporator: after the working fluid leaves the recuperator enters the evaporator to absorb more thermal energy from heat source. Here the working fluid changes the phase from under cooled liquid to saturated or superheated vapour.

Process (4-5): Expander: the saturated or superheated vapour enters the expander and the absorbed thermal energy in recuperator and evaporator converts to useful work. The working fluid leaves the expander as superheated vapour.

Process (5-6) Recuperator: heat transfer occurs between the high temperature vapour at expander outlet and low temperature fluid at pump outlet.

Process (6-1): the cooled working fluid at recuperator outlet enters the condenser. The working fluid stars condensation and more heat is rejected with help of a heat sink.

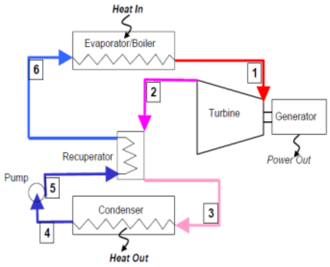

Fig.4: Organic Rankine Cycle using recuperator

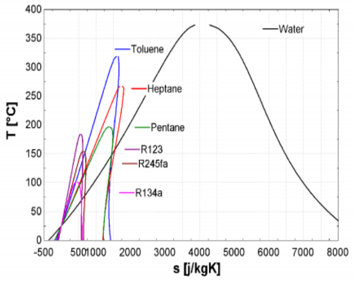

Fig.5: T-S diagram showing saturation curves of water

III. COMPARISONS WITH THE STEAM RANKINE CYCLE

Figure 4 shows in the T-s diagram the saturation curves of water and of a few typical organic fluids in ORC applications. Main difference can be stated as:

•The slope of the saturated vapour curve (right curve of the dome) is negative for water, while the curve is much more vertical for organic fluids. As a consequence, the limitation of the vapour quality at the end of the expansion process disappears in an ORC cycle, and there is no need to superheat the vapour before the turbine inlet.

•Superheating. As previously stated, organic fluids usually remain superheated at the end of the expansion. Therefore, there is no need for superheating in ORC cycles, contrary to steam cycles. The absence of condensation also reduces the risk of corrosion on the turbine blade, and increases its lifetime up to 30 years instead of 15-20 for steam turbines (Bundela & Chawla, 2010).

•Low temperature heat recovery. Due to the lower boiling point of the organic working fluids, heat can be recovered at a much lower temperature. This allows for, among others, power generation from geothermal heat sources.

•Components size. The size of the components is very dependent on the volume flow rate of the working fluid because pressure drops increase with the square of the fluid velocity. This leads to the necessity of increasing the heat exchangers hydraulic diameter and the pipe diameter to reduce this velocity. The turbine size is roughly proportional to the volume flow rate.

•Turbine inlet temperature. In steam Rankine cycles, due to the superheating constraint, a temperature higher than 450°C is required at the turbine inlet to avoid droplets formation during the expansion. This leads to higher thermal stresses in the boiler and on the turbine blades and to higher cost.

•Pump consumption. Pump consumption is proportional to the liquid volume flow rate and to the pressure difference between outlet and inlet. It can be evaluated by the Back Work Ratio (BWR), which is defined as the pump consumption divided by the turbine output power. In a steam Rankine cycle, the water flow rate is relatively low and the BWR is typically 0.4%. For a high temperature ORC using toluene, typical value is 2 to 3%. For a low temperature ORC using HFC-134a, values higher than 10% can be stated. Generally speaking, the lower the critical temperature, the higher the BWR.

•High pressure. In a steam cycle, pressures of about 60 to 70 bar and thermal stresses increase the complexity and the cost of the steam boiler. In an ORC, pressure generally does not exceed 30 bar. Moreover, the working fluid is not evaporated directly at the heat source (e.g. a biomass burner) but by the intermediary of a heat transfer loop. This makes the heat recovery easier since thermal oil is at ambient pressure, and avoids the necessity of an on-site steam boiler operator.

•Condensing pressure. In order to avoid air infiltrations in the cycle, high condensing pressures are advisable. It is not the case for water, whose condensing pressure is generally lower than 100 mbar absolute. Low temperature organic fluids such as HFC-245fa, HCFC-123 or HFC-134a meet this requirement since they condense at a pressure higher than the atmospheric pressure. However, fluids with a higher critical temperature such as hexane or toluene are subatmospheric at ambient temperature.

•Fluid characteristics. Water as working fluid is very convenient compared to organic fluids. Its main assets are:

Cost-effectiveness and availability

Non-toxicity

Non-flammability

Environment friendly: low Global Warming Potential (GWP), null Ozone Depleting Potential (ODP).

Chemical stability: no working fluid deterioration in case of hot spot in the evaporator

•Low viscosity: lower friction losses, higher heat exchange coefficients However, steam cycles are generally not fully tight: water is lost as a result of leaks, drainage or boiler blow down. Therefore, a water-treatment system must be integrated to the power plant to feed the cycle with high-purity deionised water.

•Turbine design. In steam cycles, the pressure ratio and the enthalpy drop on the turbine are both very high. This involves using turbines with several expansion stages. In ORC cycles the enthalpy drop is much lower, and single or two-stage turbines are usually used, which reduces their cost. Additional effects of the low enthalpy drop include lower rotating speeds and lower tip speed. The lower rotating speed allows direct drive of the electric generator without reduction gear (this is especially advantageous for low power-range plants), while the low tip speed decreases the stress on the turbine blade and makes their design easier.

•Efficiency. The efficiency of current high temperature Organic Rankine Cycles does not exceed 24%. Typical steam Rankine cycles show a thermal efficiency higher than 30%, but with a more complex cycle design (in terms of number of components or size). The same trend is stated for low temperature heat sources: steam Rankine cycles remain more efficient than ORC cycles.

Therefore the ORC cycle is more profitable in the low to medium power range (typically less than a few MW), since small-scale power plants cannot afford an on-site operator, and require simple and easy to manufacture components and design.

IV. THERMAL CONDUCTIVITY AND VISCOSITY

The thermal conductivity and viscosity of working fluids are two very important parameters in the design of heat exchanger and other Organic Rankine Cycle equipment. The knowledge of thermal conductivity is necessary to estimate the size of heat exchanger while the knowledge of viscosity is required to determine the required work for pumping the working fluid. High thermal conductivity and low viscosity is desirable in order to keep down heat exchangers size and to reduce the needed work for pumping the working fluid. Figures show the thermal conductivity and viscosity for the working fluids which give the highest thermal efficiency. The thermal conductivity is estimated at 373K and saturation liquid while the viscosity is estimated at 298K and saturation liquid. The thermal conductivity and viscosity data are not available for sulfur dioxide SO_2. Water is not considered a working fluid for Organic Rankine Cycle and it is added to these figures just to compare its properties to the properties for other working fluids.

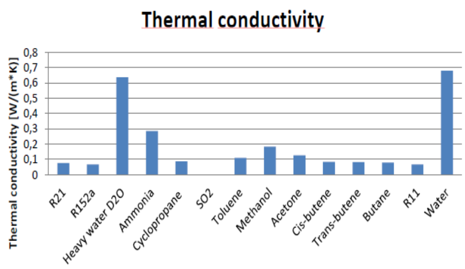

Fig. 6. Refrigerant vs Thermal conductivity

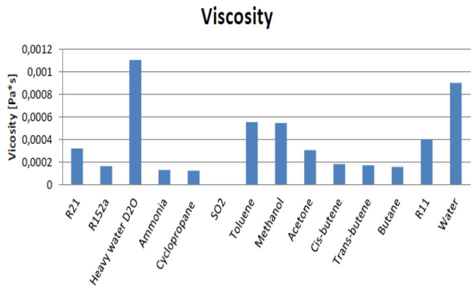

Fig. 7. Viscosity vs Thermal conductivity

The selection of optimal working fluid for Organic Rankine Cycle is not an easy process. There are many different working fluids to choose among and many criteria should be taken in consideration. Some working fluids have good thermodynamic properties but at the same time have undesirable environmental and safety data. Other fluids have a good environmental and safety data but they are not efficient thermodynamically. There is no ideal working fluid can achieve all the desired criteria and the fluid selection process is a trade-off between thermodynamic, environmental and safety properties.

According to results and regarding thermal efficiency, the best working fluids for subcritical cycles are: ammonia, cyclopropane, R152a, sulfur dioxide SO_2, R21, heavy water, toluene, methanol, acetone, cis-butene, trans-butene, butane and R11. Furthermore the best working fluids for trans-critical cycles are: R11, R21, ammonia, cyclopropane and SO_2.

Regarding the environmental and safety criteria, the selected working fluid should have zero ozone depilation potential ODP, non-flammable, non-toxic and global warming potential should be very low. Taking all these criteria in consideration, following working fluids are not optimal: ammonia (flammable and toxic), SO_2 (toxic), R21 (flammable, toxic and ODP=0,04), methanol (flammable), acetone (flammable), trans-butene (highly toxic), butane (highly toxic), R152a (toxic), and R11 (ODP=1 and GWP=4750). The environmental and safety data are not available for toluene and heavy water and the safety group for cyclo propane is also not available. Here the conclusion is that there is no ideal working fluid which can have all desirable criteria and properties at the same time. The trade-off is between high thermodynamic performance and environmental and safety criteria. There are some safe and environmental friendly working fluids like

R124, but their thermodynamic performance is quite low.

V. CONCLUSION

A review of the open literature with respect to WHR finds that IC engine exhaust contains heat with sufficient energy to justify implementation of a secondary cycle. Among the different cycles, ORC's are the most commonly implemented WHR systems in order to generate power from low-grade to medium-grade waste heat. This is because Rankine systems use standard components and operate efficiently using these grades of energy. Turbomachines are preferred for large output systems while displacement-type expanders dominate small-scale efforts. Finally, no single working fluid is best for all ORC's as the selection of the proper working fluid requires consideration of operating conditions, environmental concerns and economic factors. The amount of wasted heat from different parts of a 12 liter compression ignition engine and their potential for waste heat recovery were reviewed. Then, two configurations were introduced for simultaneous heat recovery from exhaust gases and coolant. An optimization process was applied for the configurations in which the goal was to maximize the work generation and thermal efficiency.

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