Design of Variable Geometry Waste Heat Recovery Turbine for High Efficiency Internal Combustion Engine

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ABSTRACT

This study was carried out for a 1.25L Zetec-SE DOHC engine model but its application is generic to gasoline engine (light-duty engine) applications. An ORC model with a radial turbine sub-model is implemented in a light-duty gasoline I.C engine model, to evaluate the impact of the ORC with VGT on the engine fuel consumption, net power output and finally the ORC system efficiency as compared to ORC with FGT. The results showed that VGT can improve ORC system efficiency and net power output by an unweighted point of 5.6% and 3.07kW respectively at partial to high load conditions while benefits are even higher at the lower loads therefore making it an attractive technology given its ability to recover low-grade heat and the possibility to be implemented in decentralized lower-capacity power plants

Keywords: Variable Geometry Turbine; Waste Heat Recovery; ICE Efficiency; Gasoline Engine; Organic Rankine Cycle; GT Power

Notations

BSFC *Brake specific fuel consumption*
CO_2 *Carbon dioxide*
FGE *Fixed Geometry Expander*
ORC *Organic Rankine Cycle*
TC *Turbocompounding*
VGE *Variable Geometry Expander*
WHR *Waste Heat Recovery*

HCV *Heavy duty Commercial vehicles*

NEDC *New European driving cycles*

WLTC *World harmonised light vehicle test cycle*

FTP *Functional threshold power*

TEG *Thermo-electric generation*

1. INTRODUCTION

1.1 Background

Recent trend about the best ways of using the deployable sources of energy in to useful work in order to reduce the rate of consumption of fossil fuel as well as pollution. From all the available sources, the internal combustion engines are the major consumer of fossil fuel around the globe. Out of the total heat supplied to the engine in the form of fuel, approximately, 30 to 40% is converted into useful mechanical work. The remaining heat is expelled to the environment through exhaust gases and engine cooling systems, resulting in to entropy rise and serious environmental pollution. Diesel engines for heavy duty commercial vehicles (HCV) convert in average only approximately 40% of the primary energy into mechanical power, with the residual part released to the environment, while in gasoline-powered vehicles, over 62 percent of the fuel's energy is lost (J.S Jadhao, D.G Thombare, et al. 2013). Internal combustion engines (ICE) are very inefficient at converting the fuel's chemical energy to mechanical energy, losing energy to engine friction, pumping air into and out of the engine, and wasted heat, so it is required to utilize waste heat into useful work. The heat of the exhaust gas can be converted into mechanical power for the vehicle by applying a thermodynamic process. A suitable process is the Rankine process. Today waste heat recovery can be an attractive approach to reduce fuel consumption and operating costs. Additionally, the CO2 emission can be lowered accordingly.

The recovery and utilization of waste heat not only conserves fuel, usually fossil fuel but also reduces the amount of waste heat and greenhouse gases damped to environment. It is imperative that serious and concrete effort should be launched for conserving this energy through exhaust heat recovery techniques. Such a waste heat recovery would ultimately reduce the overall energy requirement and also the impact on global warming. The Internal Combustion Engine has been a primary power source for automobiles and automotive over the past century. Presently, high fuel costs and concerns about foreign oil dependence have resulted in increasingly complex engine designs to decrease fuel consumption. Moreover, increasingly stringent emissions regulations are causing engine manufacturers to limit combustion temperatures and pressures lowering potential efficiency gains (T. Endo, S. Kawajiri et al. 2007). As the most widely used source of primary power for machinery critical to the transportation, construction and agricultural sectors, engines have consumed more than 60% of fossil oil. On the other hand, legislation of exhaust emission levels has focused on carbon monoxide (CO), hydrocarbons (HC), nitrogen oxides (NOx), and particulate matter (PM). Energy conservation on engine is one of best ways to deal with these problems since it can improve the energy utilization efficiency of engine and reduces emissions (K. N. Gopal, Rayapati S. et al. 2010). Given the importance of increasing energy conversion efficiency for reducing both the fuel consumption and emissions of engine, scientists and engineers have done lots of successful research aimed to improve engine thermal efficiency, including supercharge, lean mixture combustion, etc. However, in all the energy saving technologies studied. Engine exhaust heat recovery is considered to be one of the most effective. Many researchers recognize that Waste Heat Recovery from engine exhaust has the potential to decrease fuel consumption without increasing emissions, and recent technological advancements have made these systems viable and cost effective (H. Özcan and M.S. Söylemez 2006).

This paper gives the design of variable geometry waste heat recovery turbine for high efficiency internal combustion engines.

2. LITERATURE REVIEW

2.1. WHR Technologies

Waste heat is heat, which is generated in a process by way of fuel combustion or chemical reaction, and then "dumped" into the environment even though it could still be reused for some useful and economic purpose. This heat depends in part on the temperature of the waste heat gases and mass flow rate of exhaust gas. Waste heat losses arise both from equipment inefficiencies and from thermodynamic limitations on equipment and processes For example, consider internal combustion engine approximately 30 to 40% is converted into useful mechanical work. The remaining heat is expelled to the environment through exhaust gases and engine cooling systems (P. Sathiamurthi. 2011). It means approximately 60 to 70% energy losses as a waste heat through exhaust (30% as engine cooling system and 30 to 40% as environment through exhaust gas). Exhaust gases immediately leaving the engine can have temperatures as high as 842-1112°F [450-600°C]. Consequently, these gases have high heat content, carrying away as exhaust emission. Efforts can be made to design more energy efficient reverberatory engine with better heat transfer and lower exhaust temperatures; however, the laws of thermodynamics place a lower limit on the temperature of exhaust gases (S. Karellasa, A. D. Leontaritisa, et al. 2012). Fig. 2.1 show total energy distributions from internal combustion engine.

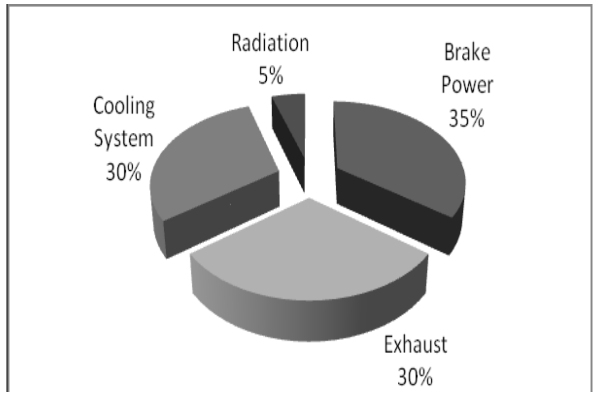

Figure 1. Total Fuel Energy Content in I.C Engine

(J.S Jadhao, D.G Thombare, et al. 2013)

The Benefits of 'waste heat recovery' can be broadly classified in two categories

1. Direct Benefits:

Recovery of waste heat has a direct effect on the combustion process efficiency. This is reflected by reduction in the utility consumption and process cost.

2. Indirect Benefits:

a) Reduction in pollution: A number of toxic combustible wastes such as carbon monoxide (CO), hydrocarbons (HC), nitrogen oxides (NOx), and particulate matter (PM) etc., releasing to atmosphere. Recovering of heat reduces the environmental pollution levels.

b) Reduction in equipment sizes: Waste heat recovery reduces the fuel consumption, which leads to reduction in the flue gas produced. This results in reduction in equipment sizes.

c) Reduction in auxiliary energy consumption: Reduction in equipment sizes gives additional benefits in the form of reduction in auxiliary energy consumption (H. Teng, G. Regner, et al. 2007).

The development of energy-efficient technologies for waste heat recovery has taken on an accelerated pace in recent years. The largest sources of waste heat in industries are the exhaust gases, hot streams and water. The energy of these three types of waste heat could be recovered and used in three different ways, that is, power recovery to generate electricity, processes and building heating through heat exchangers or heat pumps, and processes and building cooling or refrigeration by thermal-driven systems. For recovery technologies for gaseous waste heat, the method and its efficiency are mainly determined by the gaseous temperature. The waste heat sources are generally divided into three categories: low temperature (<230°C), medium temperature (230-650°C) and high temperature (>650°C) (Energy Guide for industry in Asia. 2006). The low-temperature gaseous waste heat accounts for about 50% of all the waste heat in an industry, as reported by statistic data (BCS Inc. 2008). However, most waste heat recovery technologies aim for the heat resources with medium and high temperature, while the technology for low-temperature gaseous waste heat still needs to be developed.

WHR technologies can be classified into three mainstreams, namely; thermo-electric generation (TEG), turbocompounding (TC), and organic Rankine cycle (ORC). Experimental studies proved that fuel savings of 3.9 up to 4.7% could be achieved by using thermo-electric generation (Stobart R. and Milner D. 2009; Stobart R., Wijewardane A. et al. 2010); however this technology is currently highly expensive and faced with a longer development time. On the other hand, mechanical turbocompounding can potentially improve brake specific fuel consumption up to 6% (Wilson 1986), while electrical turbocompounding contributes to fuel economy by up over 5% (Hopmann U. 2004; Katsanos C. O., Hountalas D. T. et al. 2013). The main disadvantage of TC is the increase of backpressure and finally the higher pumping losses, which compromise fuel savings from the recovery of exhaust gas heat (Mamat A.M.I. 2012). Last but not least, the Organic Rankine Cycle is probably the most promising candidate for conversion of exhaust heat into power due to its performance and practical elements of cost and ease of maintenance. The heat exchanger of the ORC system produces less backpressure compared to the TC technology, while the thermal efficiency can reach up to 13% at maximum engine power of a heavy duty diesel vehicle (Sekar R. and Cole R.L 1987). Organic Rankine Cycle (ORC) as compared to steam Rankine cycle on a same working condition have these advantages: First, higher thermal efficiency and more net power are achieved by ORC. Second, organic fluids often have lower vaporization heat and can follow the heat source to be cooled better than water at the same boiling temperature, thus reducing temperature differences and irreversibility at the evaporator, and therefore downsizing system volume and weight, which is significant for vehicle applications. Finally, turbines for organic cycles can provide higher efficiencies at part loads as well and are usually less complex due to the lower enthalpy drop of the fluid (H. Tian, Gequn s. et al. 2012).

Several other theoretical and experimental investigations have also been reported in recent years. For instance, Zhang et al. investigated the parameter optimization and the performance comparison of the fluids in a subcritical ORC and a trans-critical power cycle in low temperature binary geothermal power system. The results indicate that adoption of R123 can help to achieve the highest thermal efficiency and exergetic efficiency of 11.1% and 54.1%, respectively in a subcritical ORC system, whereas R125 presents an excellent economic and environmental performance in a trans-critical power cycle (Zhang S., Wang H., et al. 2011). Yamamoto et al. investigated the ORC apparatus through numerical simulation and an experiment. In their studies, HCFC-123 and water were chosen as working fluids. The results show that a better Rankine Cycle performance is found in the case when HCFC-123 is used (Yamamoto T., Furuhata T. et al. 2001). Kang analysed the operational characteristics and performance of the ORC system by using a radial turbine. The results show that the maximum average cycle, turbine efficiencies and electric power are found to be 5.22%, 78.7% and 32.7 kW, respectively (Kang S.H. 2012). Qiu et al. studied on the biomass-fired ORC-based micro-CHP system and achieved an electricity generation efficiency of 1.41% and CHP efficiency of 78.69% (Qiu G., Shao Y. et al. 2012). Pei et al. presented a study on 1 kW-scale ORC system by using a specially designed and manufactured turbine. The experiment results indicate that the turbine isentropic efficiency is 65% and the cycle efficiency is 6.8% (pie G., Li J. et al. 2011). Bracco et al. built a laboratory prototype of an ORC-based cogenerator using working fluid R245fa. The experiment results indicate that system has revealed promising performances, with a global electric efficiency of about 8% (Bracco R., Clemente S. 2013). Yu et al. investigated the combined system of diesel engine with bottoming ORC (DE-ORC). The results indicate that the expansion power, recovery efficiency and exergetic efficiency are 14.5 kW, 9.2% and 21.7%, respectively. And the thermal efficiency of diesel engine can be improved up to 6.1% (Yu G., Shu G. et al. 2013).

Some work on selection of the working fluids has also been reported in the past decade. For instance, Bahaa Saleh et al. compared the thermodynamic characters of 31 pure working fluids used in ORC system for the heat source, of which temperature is between 30°C and 100°C, by using the BACKONE equation of state. The results have shown that the cycle efficiency of dry and isentropic fluid is much higher than that of wet fluid (Saleh B., Koglbauer G. et al. 2007). Hung investigated the irreversibility and efficiency of some working fluids consisting of Benzene, Toluene, p-Xylene, R113 and R123. The results indicate that p-Xylene has the highest efficiency and the lowest irreversibility in recovering of the mid-temperature waste heat, while R113 and R123 have presented a better performance for the low temperature waste heat resources (Hung T.C. 2001). Wang et al. studied the relationship

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between the system performance, the pressure ratio and the mass flow rates of several organic working fluids. The results show that R123 owns the maximal thermal efficiency and net output at the same mass flow rate or heat input among the several working fluids (Wang X.D., Zhao L. 2011). Xu et al. selected 11 working fluids which met the ORC requirement from 61 kinds of fluid and studied the saturated properties and the isentropic and thermal efficiency. 8 working fluids (R507A, R290, R600a, R600, R134a, n-Pentane, Isopentane, and R404A) were considered to be favorable for the ORC and their applicable conditions were also identified (Xu J., Dong A. 2011). Heberle and Brüggemann compared the exergetic efficiency of different working fluids in series and parallel circuits of an ORC. The results indicate that the working fluid such as isopentane with high critical temperature is preferred to be in series circuits, while fluid like R227ea with low critical temperatures is suitable for parallel circuits (Heberle F.and Brüggemann D. 2013). Chys et al. examined the cycle efficiency and electricity production of several mixture working fluids using into the ORC system for the low-temperature heat source. The results indicate that the use of suitable zeotropic mixtures as working fluids has a positive effect on the ORC performance. For heat sources at 150^{o}C and 250^{o}C, a potential increase of 16% and 6% in cycle efficiency is found. The electricity production at optimal thermal power recuperation can be increased by 20% (Chys M., Broek M.V.D. et al. 2012).

2.2. Organic Rankine Cycle

The Rankine cycle system for the WHR helps to increase the efficiency of the engine and improve fuel economy. There are several factors that can be modified within the Rankine cycle to improve its efficiency such as increasing the heat source energy, working fluid selection, turbine design variation and its design optimisation. A more detailed analysis could look into the efficiencies of the components within the Rankine cycle such as the heat exchangers, working fluid pump, condenser and plumbing of the cycle. The overall power produced by the Rankine cycle is a mechanical type where it can directly be supplied to the engine shaft through a belt or a gear box or alternatively produce electricity by means of a combination with a generator. When a generator is used to produce electricity, the control of the expander can be achieved by varying its speed and/or bypassing the working fluid.

The interest for low grade heat recovery grew dramatically in the past decades. An important number of new solutions have been proposed to generate electricity from low temperature heat sources and are now applied to much diversified fields such as engine exhaust gases, solar thermal power, biological waste heat, domestic boilers, etc. Among the proposed solutions, the Organic Rankine Cycle (ORC) system is the most widely used. Its two main advantages are the simplicity and the availability of its components. In such a system, the working fluid is an organic component, better adapted than water to lower heat source temperatures. Unlike with traditional power cycles, local and small scale power generation is made possible by this technology (Quolin, V. Lemort et al. 2010). Organic Rankine Cycle (ORC) could be used to recover waste heat from ICE exhaust gas. Compared to steam Rankine cycle on a same working condition, there are some advantages of ORC as follows. First, higher thermal efficiency and more net power are achieved by ORC. Second, organic fluids often have lower vaporization heat and can follow the heat source to be cooled better than water at the same boiling temperature, thus reducing temperature differences and irreversibility at the evaporator, and therefore downsizing system volume and weight, which is significant for vehicle applications. Finally, turbines for organic cycles can provide higher efficiencies at part loads as well and are usually less complex due to the lower enthalpy drop of the fluid (Hua T., Gequen S. et al. 2012).

The ORC is not merely the subject of laboratory studies as more than one hundred ORC plants are now operating to generate electricity commercially, and the ORC has also applied to diverse fields including industrial waste heat, solar thermal power, geothermal heat, biomass combustion heat, engine exhaust gases and so forth (Quolin, V. Lemort et al. 2010). ORC manufacturers such as ORMAT, Turboden, BNI, Adoratec, UTC, and Electratherm have been present on the market since the beginning of the 1980s. All of them use the turbine as an expander, except Electratherm, which uses a screw expander (Quolin, V. Lemort et al. 2010). Large-scale ORC plants have been successfully demonstrated, such as in the geothermal plant in Altheim, and in the biomass-fired CHP plants in Admont, Lienz and Heidelberg (Pie G., Li J., 2011). In Europe more than 120 ORC plants are in commercial operation, with sizes ranging from 0.2 to 2.5 MW, using biomass combustion heat (Bini R., Guercio A. et al. 2009).

The ORC is structurally similar to a typical Rankine cycle but uses organic fluids as a working fluid instead of water. Organic fluids are suitable for the ORC because their specific vaporization heat is much lower than that of water. This enables the ORC to produce electricity by using low-temperature heat sources the layout of the components of a system working on ORC utilizing exhaust

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flue gas heat as thermal source is shown in Fig.2 below. The Rankine cycle is a closed loop cycle where heat is transferred to a working fluid at constant pressure. It consists of four main components, namely; evaporator, expander, condenser and pump. The working fluid is vaporized in the evaporator and then expands in the expander that drives a generator to produce electricity. Finally, the working fluid is condensed at constant pressure and pumped again to the evaporator. In recent years, a great number of studies deal with the implementation of ORC systems in vehicle powertrains. Yang et al. found that the implementation of an ORC system operating with R245fa improves bsfc from 2.5% to 7.4% (Yang K. and Zhang H. 2015). In another study, the engine water and the exhaust gas were employed to predict ORC efficiency of around 9.6%, while the total engine thermal efficiency was increased by 9.0% (Shu G.Q., Yu G. et al. 2013). The efficiency of the ORC system is a function of its specification, including the available heat sources employed, the heat exchanger design, the working fluid selected and the expander type chosen and its design, to name the most important.

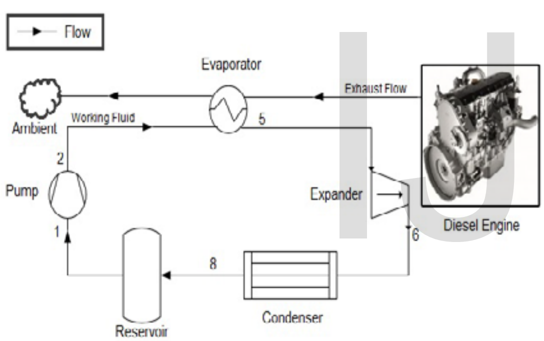

Figure 2. The Schematic Diagram of an Organic Rankine Cycle (ORC)

(A. Karvountzis-Kontakiotis, A. Pesiridis et al. 2016)

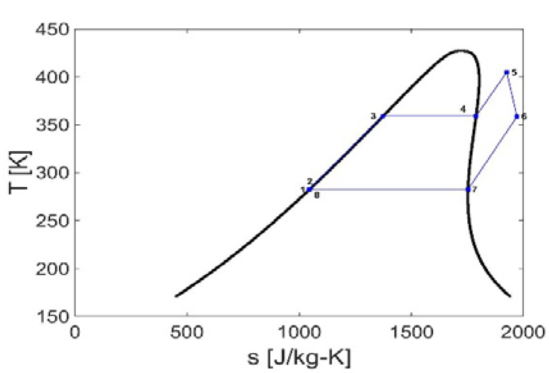

Figure 3. The Corresponding T-S Diagram of the ORC

Among the ORC system components, the expander is the most crucial and expensive component in Organic Rankine Cycle (ORC) systems (Wong C. S., Meyer D. et al. 2013). Expanders can be classified into two main groups, namely; positive displacement expanders (Screw, Scroll, Piston and Rotary Vane) and turbomachine expanders (Axial or Radial). The selection of the appropriate expander depends on the application; however for waste heat recovery applications scroll expanders and radial turbines are the most common solutions in literature (Rosset K., Mounier V. et al. 2015; Zhen L., Guohong T. et al. 2015). ORC efficiency is increased at higher pressure ratios; therefore radial turbines appear as more suitable for vehicular applications where mass flow rates are in the low to middle range and pressure ratios are middle to high. In terms of manufacturing cost, it is less expensive compared to axial turbines as they can be converted from standard production designs, being less sensitive to their blade profile (Paltrinieri N. 2014), while radial turbine geometry allows higher peripheral speeds than the axial turbines and therefore a higher enthalpy drop per stage (Quoilin, Broek et al. 2013). On the disadvantages of radial turbines, they are inefficient at part load, don't operate efficiently at variable speeds (Petchers N. 2003) and their efficiencies drop when operating under off-design conditions (Teng H., Regner G. et al. 2007).

2.3. Variable Geometry Turbine

Variable geometry machines are of particular interest to advanced diesel powertrains for future trucks, since they are viewed as the key enabler for the application of the EGR system in order to meet legislated, current and future, emissions standards. This is due to the fact that VGT systems have the potential to provide accurate control of the pressure difference across the engine, as well as very quick response during engine transients, (Z. Filipi, Y. Wang et al. 2001).

The VGT technology was originally considered as a method to eliminate turbo lag, as well as to improve low speed boost and torque (Watson N. and Banisoleiman, K. 1986). VGT's potential to reduce emissions through increased transient air/fuel ratio and improved transient fuel/air mixing was also noticed (Watson N. and Banisoleiman, K. 1986). The control strategies ranged from a simple increase of turbine area with engine speed (Hashimoto T., Oikawa, T. et al. 1986), through transient strategies developed from steady-state data to systematic development of a multivariable controller (Duffy K. P., Miller R. L. et al. 1999). Nevertheless, production applications have been limited due to the increased cost of the new technology and the fact that continuous improvement of conventional turbochargers made the potential gains less tangible. Recently, however, the VGT technology is getting renewed attention as an enabling technology for applications of exhaust gas recirculation to reduce emissions of heavy-duty diesels. Howley et al. (Hawley J. G., Wallace F.J. et al. 1999) reported experimental studies examining the potential for increasing the limiting torque curve and improving coordination between the VGT setting and EGR setting for optimized emissions. The response of two alternative designs of the VGT system to step pedal inputs was assessed experimentally in the context of controller and actuator design and system dynamic performance, as reported by Brace et al. (Brace C. J., Cox A. et al. 1999). The study of issues related to model-based control of EGR on a light-duty diesel engine, together with experimental results of the transient air/fuel ratio control with open loop, PI and model-based control systems was reported by Duffy at al. (Duffy K. P., Miller R. L. et al. 1999). Analysis of the control aspects of the joint effect of VGT and EGR on engine emissions and a multivariable feedback controller was proposed by Stefanopoulou et al. (Stefanopoulou A., Kolmanovsky. et al. 2000).

One of critical concerns in a variable geometry turbine (VGT) design program is shock wave generated from nozzle exit at small open conditions with high inlet pressure condition, which may potentially lead to forced response of turbine wheel, even high-cycle fatigue issues and damage of inducer or exducer. Though modern turbine design programs have been well developed, it is difficult to eliminate the shock wave and all the resonant crossings that may occur within the wide operating range of a VGT turbine for automotive applications. Variable geometry turbines have been widely used in commercial diesel vehicle applications, due to advantages of improving engine performance and reducing engine emissions. When engine speed is high, the engine back pressure is greatly increased if the nozzles are closed down for exhaust braking. Under the engine exhaust braking condition, the area of the nozzle geometry throat is small, resulting in a high pressure drop across the nozzles (Chen H. 2006). Meanwhile, shock waves may be generated on nozzle vane surface near trailing edge. Because of the relative movement of nozzle vanes and turbine blades, the shock wave periodically strikes on the leading edge of turbine blades, generating a strong aerodynamic excitation on the blades. If the frequency of aerodynamic excitation force matches with the natural frequency of the turbine blades, this forced response will be a high cycle fatigue failure concern of the turbine wheel. Kawakubo (Kawakubo T. 2010), researched unsteady rotor-stator interaction of a radial inflow turbine with variable nozzle vanes and indicated that the nozzle shock wave impinges on the impeller blades periodically. So the shock wave is partially responsible for the damage of downstream turbine wheel.

The implementation of a variable geometry turbine can potentially mitigate many of the performance disadvantages of a radial turbine expander in an ORC system. A recent study on the aerodynamic evaluation of a VGT for organic Rankine cycle showed that turbine power and efficiency is improved in a higher range of mass flow rate and expansion ratios compared to the fixed geometry turbine (Wong S.C. and Krumdieck S. 2015). However in another study the implementation of a variable geometry turbine in a low temperature ORC system that uses geothermal heat showed little benefit in terms of average power output compared to a fixed geometry turbine. (Read M., Kovacevic A. et al. 2015). However, the literature review performed for this work has failed to uncover a detailed study to evaluate the impact of variable geometry radial expander (VGE) performance for organic Rankine cycle waste heat recovery in vehicular applications. Additionally, the evaluation of this technology in terms of fuel consumption and emissions at partial engine load conditions is crucial, as internal combustion engines will only infrequently operate at the ORC design point.

More and more ORC plants, especially for geothermal power plants, solar power plants and biomass CHP plants have been installed in many counties including Italy, USA, Japan, Canada, Austria, Germany and elsewhere (Tchanche B.F, Papadakis G. et al. 2009). However, the ORC systems using low-temperature exhaust as heat source are still under development. Experimental studies on the ORC for waste heat recovery from low-temperature flue gas have not been reported either. This present study explores the impact of a variable geometry turbine (VGT) in an ORC system for waste heat recovery from an internal combustion engine (ICE). An integrated in-house model has been developed for this reason, which includes the engine map row data from a gasoline engine, the ORC model and the variable geometry radial expander model. In order to evaluate the potential benefit on fuel consumption and NOx emissions, the model was employed at various engine speed and bmep operating points.

3. MODELLING

3.1. Gasoline Engine Modelling

3.1.1. Gasoline ICE Model

Large quantity of hot flue gases is generated from internal combustion engine etc. If same of this waste heat could be recovered, a considerable amount of primary fuel could be saved. This depends upon mass flow rate of

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exhaust gas and temperature of exhaust gas. The internal combustion engine energy lost in waste gases cannot be fully recovered. However, much of the heat could be recovered and losses be minimized by adopting certain measures such as WHR for power generation. This modelling work focuses on heat recovery from exhaust gas and conversion into mechanical power (then to electrical power) with the help of organic Rankine cycle (ORC). The engine mode for this simulation was based on a Ford Fiesta 1.25l 16v Zetec-SE DOHC engine which basic characteristics are given in Table 1 (Ford Workshop Manuals). The Ford Fiesta 4-cylinder 16-valves gasoline engine manufactured by Ford appears to be a reasonable choice to apply a waste heat recovery system on a gasoline ICE, considering its high exhaust flow rate and the level of exhaust gas energy available for recovery.

Table 1. 1.25L Zetec-SE DHOC Engine Specifications

Item	Specification
Main Specifications	
Displacement	1242cc
Bore	71.9mm
Stroke	76.5mm
Number of Cylinder	4
Number of Valve/Cylinder	4
Fuel Type	Unleaded Gasoline
Performance	
Maximum Torque	110Nm
Maximum Power	55kW
At Engine Speed	4000rpm

The simulation is a vital part in the engine phase in this current economy. Many car manufacturers do excessive testing in software based analysis such as in 1-D simulation and 3-D simulation before actual experimental testing. This speeds the process of car production and cuts the costs of the testing phase, investigate quickly the effects of a change in a real life situation of a car that take place over several years, and finally used to investigate situation in a car that would be dangerous in real life.

This engine model was designed using a GT Power commercial engine simulation software, in order to develop the required engine maps and exhaust conditions that will be used for the ORC simulation. The data input was set at different load (bmep) and RPM range to replicate the different conditions like engine would face in the daily use such as hilly road conditions as shown in fig.5 below. This makes the simulation as realistic as possible while fig.4 shows the layout used for the simulation of the 1.25l zetec engine.

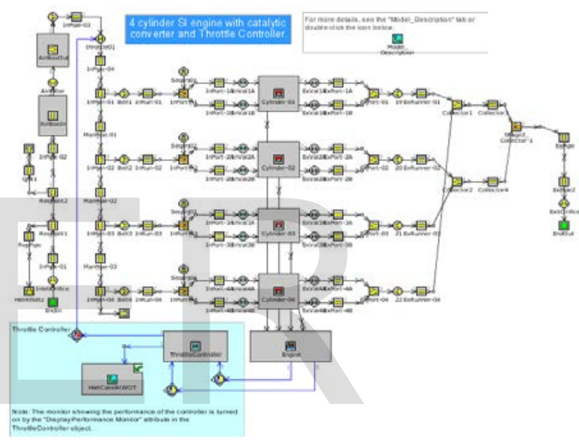

Figure 4. S.I Engine Simulation Model

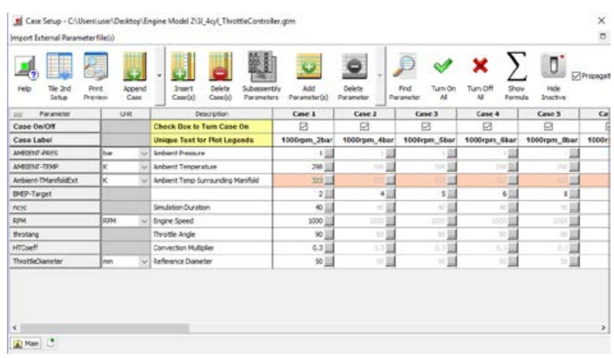

Figure 5. The Calibration of the Engine Model

3.1.2 Impact of Pressure drop on BSFC

Typically brake specific fuel consumption (bsfc) is mapped as a function of brake mean effective pressure load (bmep) and engine speed. In addition to any dependence on combustion system design, these maps depend upon induction and exhaust system details, calibration, engine friction and auxiliary load characteristics. Minimum brake specific fuel consumption figures are typically around 240g/ kW-hr at mid to high speed range conditions. (The formulae presented are consistent with the use of SI units unless stated otherwise. Numerical data for specific fuel consumption are given in g/kW-hr, and engine speed in rev/min, following common practice.) To separate the various effects on fuel consumption it is advantageous in the first instance to consider indicated conditions over the compression and power strokes of the engine cycle. This isolates the characteristics of the combustion system as far as possible, (P.J. Shayler and J.P. Chick, 2016).

Brake Specific Fuel Consumption (BSFC) can be defined as the amount of fuel the engine requires to produce one kilowatt power per hour.

BSFC = $\frac{Fuelrate}{Power} = \frac{\dot{m}_f}{W_{eng}}$**(g/kW-hr)……………………… (1)**

BSFC increases with increasing inlet air temperature and decreases with both engine load (bmep) and engine speed. The decreasing of bsfc with engine occurs until reaching around (2500rpm), then increases at around (2700rpm) and above.

Brake Mean Effective Pressure (BMEP) is another very effective yardstick for comparing the performance of an engine of a given type to another of the same type, and for evaluating the reasonableness of performance claims or requirements.

Bmep = $\frac{W_b}{V_d} = \frac{2\pi.T.N_R}{V_d}$……………………………………… (2)

Where T = Torque, V_d= swept volume, N_R=Number of Revolution.

Performance map is used to display the bsfc over the engines full load and speed range. Engine manufacturers plot engine performance maps as an aid to vehicle manufacturers in choosing suitable engine for their vehicles. The model engine performance map is plotted and used in this work to validate the suitability of the engine model for WHR using ORC system.

The final calibrated engine model, fig.5 was simulated at various bmep and speed ranges to determine the effect of bmep and speed on break specific fuel consumption (bsfc), the exhaust waste heat, and the exhaust flowrate of the engine model which relates to its real life situations.

3.2. Organic Rankine Cycle (ORC) Modelling

3.2.1. ORC for Waste Heat Recovery (WHR)

The ORC model uses standard thermodynamic relations to calculate the fluid conditions at each point in the cycle indicated in Figure 2. The isentropic efficiency of the turbine and the pump was fixed at 0.85 and 0.75, respectively, for the calculations described in this paper. Some other assumptions made for the process include the following:

1. Each process is considered steady-state.
2. Pressure drops and heat losses through pipe lines are neglected.
3. Potential and kinetic energy of flowing fluid are considered negligible.
4. The fluid entering in the pump is considered to be saturated liquid at the correspondent condenser pressure (quality = 0).
5. The condenser pressure is determined starting from the condenser temperature, which is an investigation parameter.

The ORC system consists of a dry expansion evaporator driven by exhaust heat, an air cooled condenser, a reservoir, a solution pump and a single stage turbine (Fig. 2). The pump supplies the working fluid to the evaporator, where the working fluid is heated and vaporized by the exhaust heat. The generated high pressure vapour flows into the turbine and produces power there, and then, the low pressure vapour is led to the condenser and condensed by air. The condensed working fluid flows into the reservoir and is pumped back to the evaporator, and a new cycle begins. The typical T–S process for the investigated ORCsystem is as shown in Fig.3.

The detailed equations used for each component are listed in the following sub-sections as shown in figure 2.

3.2.1.1. Pump Process

The compression process in the pump is considered non-isentropic by adopting an isentropic efficiency (η_P) of 0.75. The work done by the pump is given by:

$W_{Pump} = \dot{m}_{refr}\ .\ (h_2 - h_1)/\eta_P$ ………………….. (3)

3.2.1.2. Evaporator Heating Process

The heating process in the evaporator is considered isobaric and is provided by the exhaust gas from the IC engine. The heat transferred from the exhaust gas to the working fluid is:

$Q_{In} = \dot{m}_{refr}\ .\ (h_5 - h_2)$ ……………………….. (4)

3.2.1.3. The Expansion process

The expansion process by the turbine produces work by reducing the working fluid pressure to the condenser one. This expansion process is considered non-Isentropic with an Isentropic efficiency (η_T) of 0.85, (Hung T.C. 2001). Thus, the turbine converts the thermal energy of the superheated working fluid into useful mechanical energy which is given by the equation:

$W_{Turbine} = \dot{m}_{refr}\ .\ (h_5 - h_6)$ …………………... (5)

3.2.1.4. Condenser Cooling Process

The condenser cools the working fluid before being pumped again to high pressure. The heat rejected by the condenser is given by:

$$Q_{Out} = \dot{m}_{refr} \ . (h_6 - h_8) \ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \ (6)$$

The complete process is summarised thus: the power consumed by the pump is given by (1), the heat input from the exhaust gas is given by equation (2), the power generated by the turbine is determined by equation (3) and the rejected heat from the condenser is given by equation (4). The number indexes are schematically described in Fig.3.

3.2.1.5. ORC Efficiency

The thermal efficiency of the ORC cycle is calculated as the ratio between the net power output and the thermal power input available from the engine, as shown in equation (8). Subtracting the power consumed by pump from the power generated by the turbine gives the net power generated by the ORC system and is represented by:

$$W_{net} = W_{Expander} - W_{Pump} \ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \ (7)$$

Finally, the overall ORC efficiency is given by:

$$\eta_{ORC} = \frac{W_{net}}{Q_{In}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \ (8)$$

The efficiency of the turbine is given by the turbine model through an interpolation and extrapolation module, as turbine efficiency varies at different nozzle (stator) positions, expander rotational speeds, pressure ratios and mass flow rates. Then the ORC model calculates the power produced by the turbine as given by equation (5).

This ORC model, (figure 6), was designed from the GT power simulation software and was used to calculate the thermodynamic properties of the organic fluid at liquid and gaseous conditions. The system in this version of the ORC model is assumed to operate at steady state conditions, and for simplicity, the heat and pressure losses in the connecting pipes are neglected.

The WHR system uses a Rankine cycle to recover its energy. Once the turbine design calculations were performed, the turbine map was created for the fixed and variable geometry turbines. Using different loads (bmep) and RPM ranges for the engine as mentioned before; the fixed and VG turbines were compared. For the VG turbine the vane angles were set in four different nozzle positions for each situation to obtain the best suited vane angle for each load and RPM. Fig.6 below shows the ORC model layout for the WHR designed using GT-Power simulation software and fig.7 shows the calibration data used for the ORC model simulation.

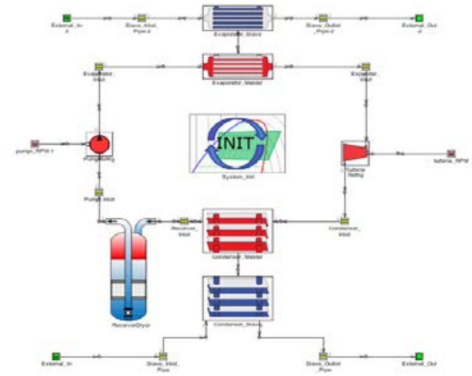

Figure 6. ORC Simulation Model

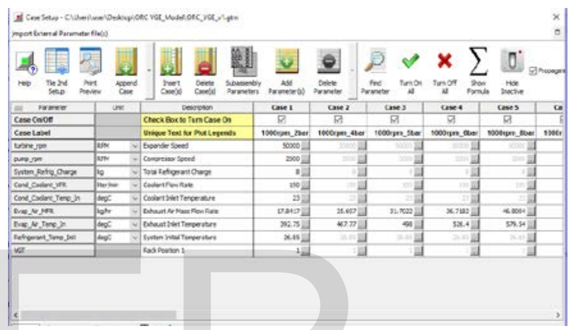

Figure 7. The Calibration of the ORC Model

3.2.2. ORC for various Operating Engine Points (FGT)

Using the exhaust gas temperature and exhaust gas mass flowrate data obtained from the simulation of the 1.25L engine model, the fixed geometry turbine (FGT) was simulated on the designed ORC system to obtain the power recovered from engine exhaust gas

3.3. Variable Geometry Turbine Modelling

The variable geometry turbine (VGT) uses variable vanes to control exhaust flow against the turbine blades. The expander modelling includes the modelling of a turbine which consists of three main components namely volute, stator and rotor. The process includes the procedures of turbine design and analysis respectively. The latter includes the off-design expander analysis through which turbine maps can be developed by varying the pressure ratio, the rotational speed, the mass flow parameter and the nozzle position. The polynomial correlations between all these parameters are imported in the ORC model for integration.

3.3.1. ORC for various Operating Engine Points (VGT)

The ORC system technique developed and tested on a fixed geometry turbine is extended to modelling of the VGT, where the same exhaust gas temperature and mass

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flowrate were simulated on the designed ORC system to determine how this could affect the output power of the ORC system as compared to the FGT. This was made possible by setting vane at four (4) different nozzle positions (Rack positions) of 0.2-0.8 representing 20 to 80% nozzle opened positions.

4. RESULTS

4.1 Engine Modelling

The 1.25L Zetec-SE DHOC engine was simulated at steady-state operation for engine speeds across 1000 rpm to 5000 rpm and targeted bmep of 2 to 9bar conditions. The engine model calibration was based on experimental fuel consumption values from (P.J. Shayler and J.P. Chick, 2016) and the results of the engine modelling include the calculations of the exhaust gas conditions (temperature and mass flowrate) and the engine fuel consumption (bsfc).

4.1.1. Engine Exhaust Temperature

For analysis of exhaust energy, several parameters of the engine exhaust conditions, such as temperature, mass flow rate, would be required for the calculation. In those parameters, the exhaust temperature plays a so important role, such as for the design of recovery system, the choice of working fluids and the optimization of system. Therefore, in order to make full advantages of exhaust energy, it is necessary to have adequate information of the distribution of exhaust gases temperature under different engine operating conditions. As shown in Fig.8 below which is the distribution of the measured exhaust gas temperature as function of engine speed and engine load (bmep) of the engine model simulated.

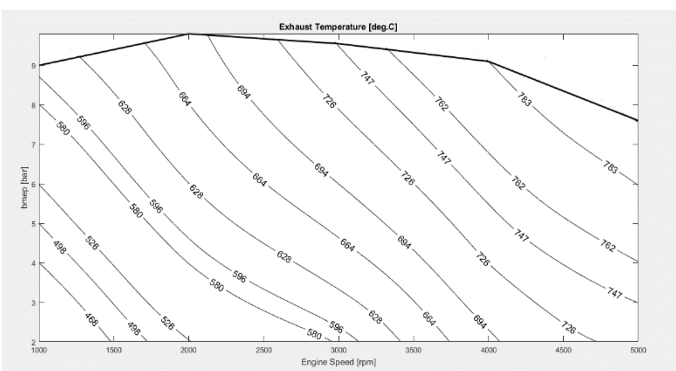

Figure 8. The Distribution of the Engine Exhaust gas Temperature as function of Speed and Bmep

The result shows that the exhaust gas temperature distinctly depends strongly on the engine speeds and loads. At mid-range of the engine bmep (4 - 7bar), the corresponding exhaust gas temperature of the test engine reached 500 – 700^{o}C and could rise up to 783^{o}C at full load. Therefore, the gasoline 1.25L engine gives a temperature distribution ranging from 458^{o}C to 783^{o}C when running the engine on 1000-5000rpm engine speed and 2-9bar loads as obtained from the simulation results.

4.1.2 Engine Exhaust Mass flowrate

From the simulation results of the engine model, it is obvious that both the engine exhaust mass flow rate and temperature increase at higher engine speed and loads operating conditions; this means that the available exhaust gas energy for the WHR system is higher. Figure 9 display the exhaust mass flowrate distribution available for WHR from the engine model.

The Simulation of the 1.25L engine model at same conditions above resulted in the exhaust mass flowrate ranging from 27kg/hr to 226kg/hr as shown in fig.9 which shows a promising potentials for the WHR process.

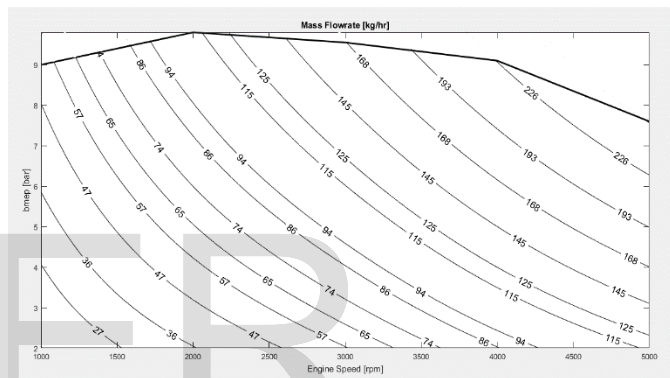

Figure 9. The Distribution of the Engine Exhaust gas flowrate as function of Speed and Bmep

The power generated by the turbine is directly proportional to the amount of exhaust gases flow rate and waste heat temperature and the equation for the power output by the system is given as:

$$\boldsymbol{P = \dot{m}_r C_p T} \quad \ldots\ldots (9)$$

Where P = Power, $\dot{m}_r$= Exhaust Mass Flowrate, C_p= Specific Heat Capacity and T = Exhaust Temperature.

4.1.3 Engine BSFC Map

Fig. 10 shows the experimental fuel consumption map obtained from the simulation of 1.25L gasoline engine model.

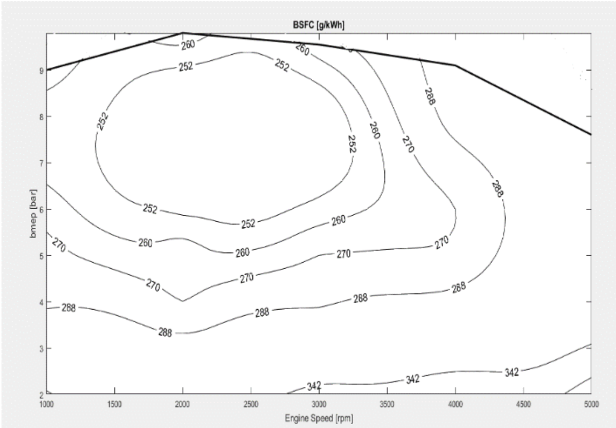

Figure 10. Ford 1.25L Zetec-SE Engine BSFC Map

As seen from the graph of engine's actual bsfc, the best bsfc is a 252g/kWh (the island), obtained round 2,000 to 3,000rpm with 6 to 9bar load and the worst bsfc is when the car is idling, where the engine consumes fuel with little or no loads (little or no bmep) completely.

The engine fuel consumption is briefly presented in table 2. Comparative results between table 2 and fig. 10 show that bsfc is predicted within a 3% error band compared to the experimental values.

Table 2. Simulation BSFC Results

1	Engine Speed [RPM]	Bmep [Bar]	BSFC Experimantal [g/kwh	BSFC Final [g/kwh]
2	1000	2	345	355.837
3	1000	4	285	296.536
4	1000	5	275	281.891
5	1000	6	265	271.952
6	1000	8	255	259.976
7	1000	9	265	256.389
8	2000	2	340	398.201
9	2000	4	270	306.019
10	2000	5	265	288.044
11	2000	6	250	276.05
12	2000	8	240	260.969
13	2000	9.8	265	252.652
14	3000	2	350	422.329
15	3000	4	285	320.064
16	3000	5	270	299.873
17	3000	6	255	286.592
18	3000	8	245	270.163
19	3000	9.55	265	262.551
20	4000	2	360	487.718
21	4000	4	295	357.842
22	4000	5	275	331.604
23	4000	6	270	314.381
24	4000	8	295	292.468
25	4000	9.1	300	284.702
26	5000	2	370	550.562
27	5000	4	325	394.871
28	5000	5	315	363.516
29	5000	6	310	343
30	5000	7.6	330	321.386

4.1.3.1 Impact of FGT on the engine BSFC

The implementation of ORC with FGT in WHR process for the gasoline engine, the engine bsfc map was improved with about 3% which implies the engine fuel consumption could be reduced by 3% when using the technology for the WHR. This also reduces the cost of running the engine and the amount exhaust gas send to the environment. Fig.11 shows the improve bsfc map of the engine with FGT implementation. This improved bsfc was determined using the formula:

$$\textbf{FGT\_BSFC} = \frac{\textit{Fuel Flowrate } (\dot{m}_f)}{\textit{Brake Power} + \textit{FGT\_ORC Power}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (10)$$

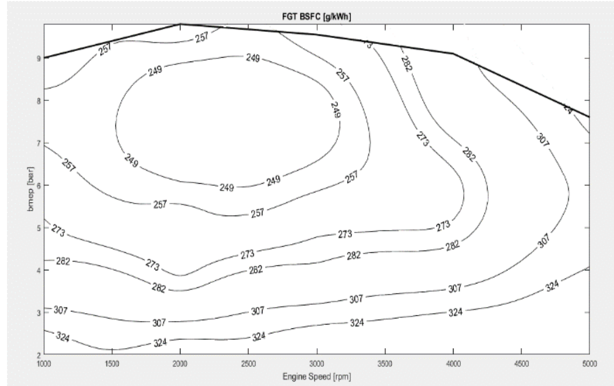

Figure 11. Impact of FGT on the Engine BSFC Map

4.1.3.2 Impact of VGT on the Engine BSFC

In this study, ORC with VGT was further implemented in the simulation process to further improve the engine bsfc from that of FGT. This resulted in around 5% fuel economy as compared to the original fuel consumption of the engine. This proves the potentials that can be explored when ORC with VGT technology is adopted in WHR from gasoline engine. Fig.12 below shows the impact of VGT on the engine bsfc map as obtained from the engine model simulation results. The outcome was obtained using the formula:

$$\textbf{FGT\_BSFC} = \frac{\textit{Fuel Flowrate } (\dot{m}_f)}{\textit{Brake Power} + \textit{VGT\_ORC Power}} \ldots\ldots\ldots (11)$$

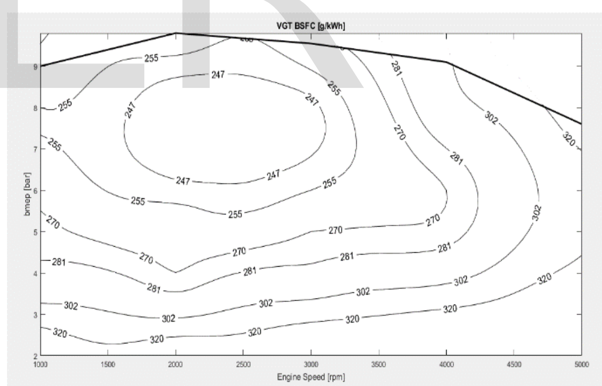

Figure 12. Impact of VGT on the Engine BSFC Map

4.1.3.3 The Percentage difference of FGT and VGT on the Engine BSFC

Fig.13 below shows the typical values of the impacts of implementing FGT and VGT on the engine performance map. This is given in percentage of the difference between the actual engine bsfc map as obtained from the simulation and that obtained by implementing FGT and VGT as calculated using the formula:

$$\textbf{Difference (\%)} = \frac{BSFC_{fgt/vgt} - BSFC_{engine}}{BSFC_{engine}} \quad \ldots\ldots\ldots\ldots\ldots (12)$$

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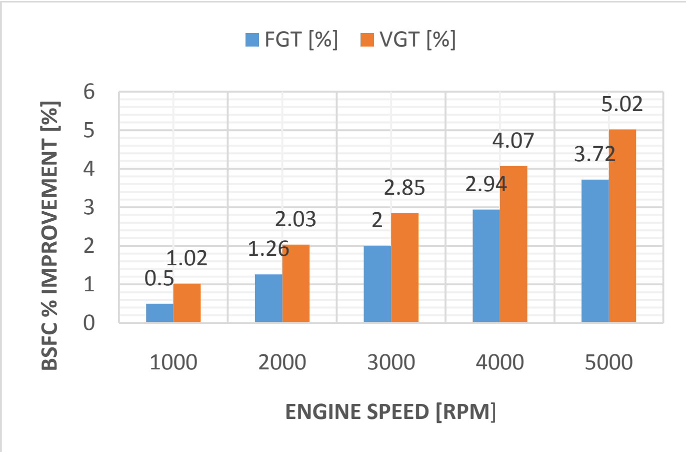

Figure 13. Comparison of FGT and VGT Impact on the Engine BSFC Map

From the results shown in fig.13, it's obviously that by implementing ORC with VGT, the engine bsfc is improved by up to 5.02% more at maximum compared to implementing ORC with FGT which improves the engine bsfc by up to 3.72% maximum. The results gives the potential fuel economy that can be explored in implementing the technology of ORC with VGT in WHR from gasoline internal combustion engines (ICE).

4.2. Optimisation of ORC Efficiency using VGT

4.2.1 Impact of VGT on ORC system Efficiency

The exhaust gas conditions obtained from the engine simulation were fed into the ORC model, which was parametrically executed for a realistic range of turbine nozzle positions (0.2 – 1, which correspond to 20% open to 100% open). Fig.14-17 show the distribution of the impact of the nozzle positions on ORC efficiency under various engine conditions. As the nozzles are closed (moving towards 0.2 from an initial value of 1.0), the ORC efficiency is initially on the increased (that is from 0.2-0.6 nozzle positions) and after reaching a maximum the efficiency is decreased. The ORC maximum thermal efficiency is a trade-off between the efficiency of the evaporator and the turbine.

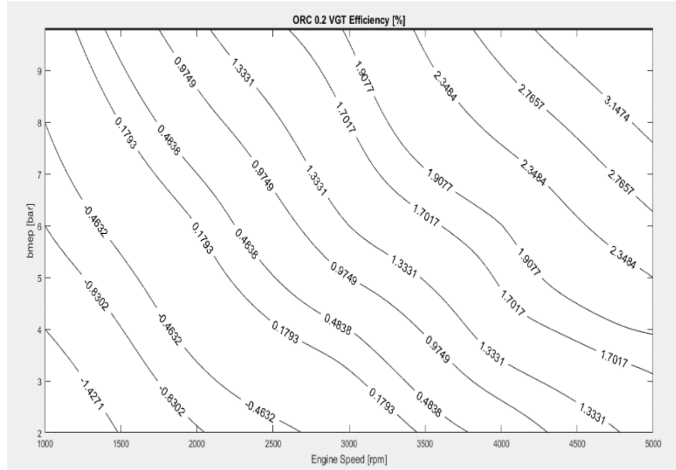

Figure 14. 0.2 VGT Nozzle Position ORC System Efficiencies Distribution

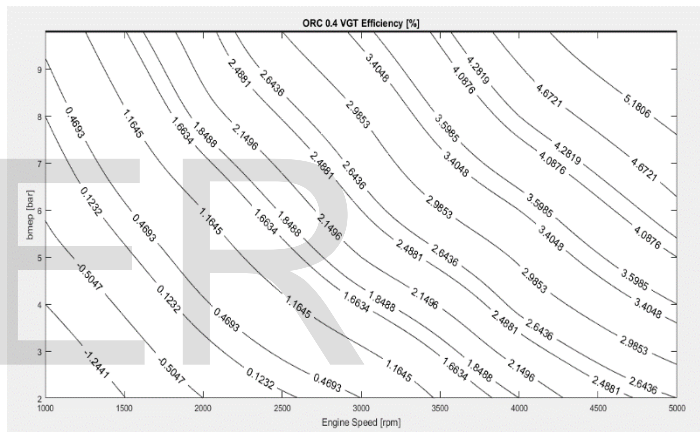

Figure 15. 0.4 VGT Nozzle Position ORC system Efficiencies Distribution

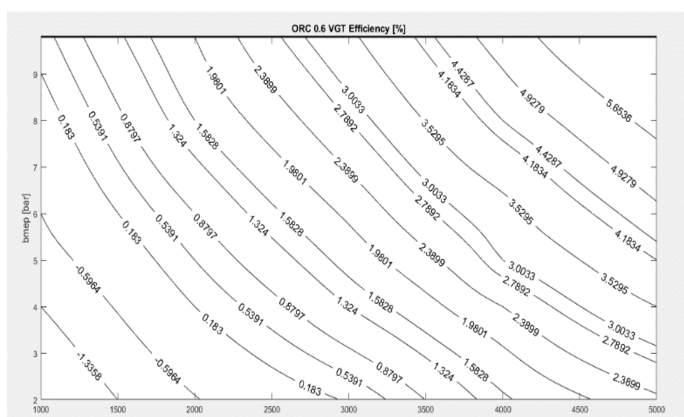

Figure 16. 0.6 VGT Nozzle Position ORC System Efficiencies Distribution

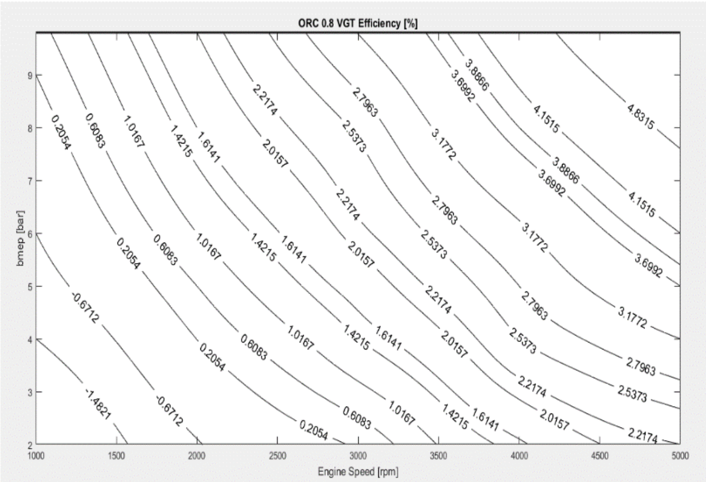

Figure 17. 0.8 VGT Nozzle Position ORC System Efficiencies Distribution

From the simulation results of VGT on ORC system (fig.14-17), it can be seen that in each case of the VGT nozzle positions, the ORC system efficiency increases with the simultaneous increase of speeds and loads of the engine model. This is evident from the fact that it is the exhausted gas mass flowrate and temperature that determine the amount of heat transfer to the organic working fluid which in turn determines the eventual amount of mechanical power generated by the ORC system. Therefore, the power generated by the ORC system is the function of the engine exhaust gas conditions. The results have also shown that at lower speeds (1000 – 2000rpm) and low load, the system appears not to be generating power but at same low speed, the system generates power when with high loads (bmep). These kind of situations can be taken care of with VGT control technology where the actuator will adjust the vane opening in order to balance the speed and load to maintain continuous power generation by the system.

As highlighted earlier, the results show that the ORC system efficiency keeps increasing as the nozzle opening positions moves from 0.2 to 0.6 and start dropping when from 0.8 to 1.0 (FGT), making 0.6±1 VGT nozzle position as best for WHR technology. However, for 0.2 VGT nozzle position, the efficiency is zero at 1000rpm, showing that the system is not generating power at that speed, hence in real situations, the actuator has to adjust the nozzle opening to balance the 1000rpm speed with flowrate that can generate power. Fig.18 below shows the impacts of all the nozzle position of the VGT on the ORC system efficiency and from the results obtained, 0.6 VGT nozzle position appears to have the best positive impact on the ORC system efficiency.

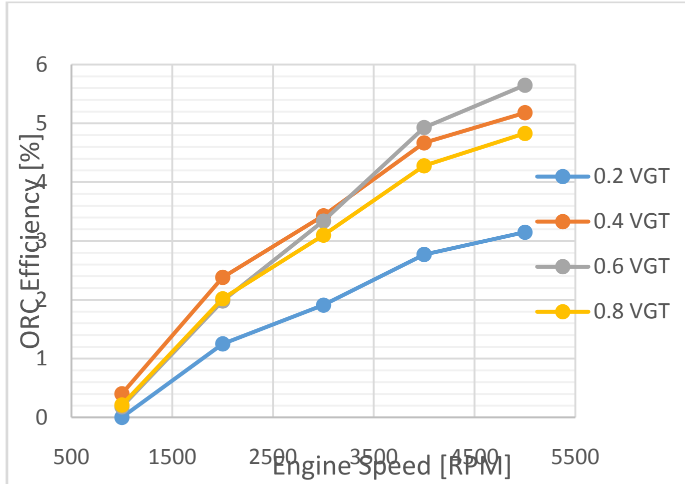

Figure 18. Comparison of the Efficiencies of the Various VGT Nozzle Positions used

4.2.2. ORC System Efficiency using FGT

Fig.19 shows the efficiencies distribution map as obtained from the ORC system with FGT. It can be noted from the map that from 1000 to 3000rpm and from 2 to 8bar the system is not generating power. The results also show that the ORC system with FGT is 4.03% (2.2kW net power recovered) efficient in WHR, which could then be optimize by implementing VGT on the ORC system.

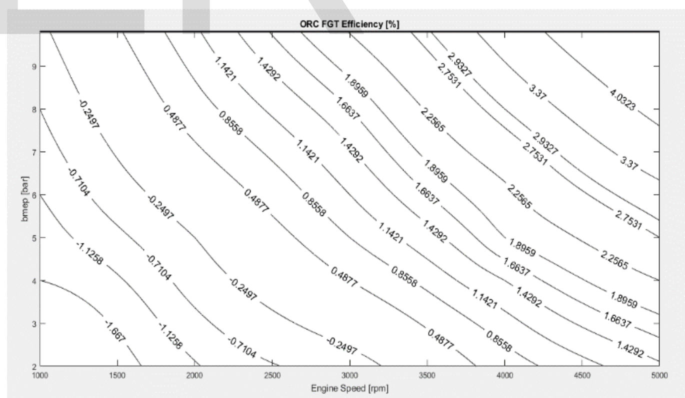

Figure 19. FGT ORC System Efficiency Distribution

4.2.3 Comparison between FGT and VGT for ORC WHR

The product of an ORC system is generated power. This system also uses power, so the cost of power consumed is subtracted from the value of the power generated to determine the net power produced. The net power translate to revenue, so it is the target variable for the ORC system and hence, the results of VGT on ORC system net power generated for WHR is the major concern of this work.

In a VGT, the vanes on the turbine can change their angles, depending on the velocity it is turning at – this provides higher boost pressure at different rpms. FGT – there's only one angle (100% opened) and a constant boost pressure. The simulation results illustrate that the variable geometry turbine achieves better ORC efficiency benefits compared to the fixed geometry turbine even under fixed organic fluid mass flowrate. Fig.20-21 present the impact of the VGT on ORC efficiency and ORC power output compared to a fixed geometry turbine (no moving nozzles). It is observed that the variable geometry turbine achieves higher ORC efficiency and power outputs through all engine operating points. Especially under low to partial load conditions, where ORC efficiency suffers, VGT appears to enhance the ORC system performance. In addition, at high speeds and loads, the extra power of the VGT technology compared to the FGT is almost 1kW (transmission losses neglected as this is a feasibility study), while the ORC efficiency is extended beyond the 5%, which appears in most studies to be a good performance of an ORC system for WHR. Fig.20-21 show the impact of VGT on the ORC system efficiency and turbine power output.

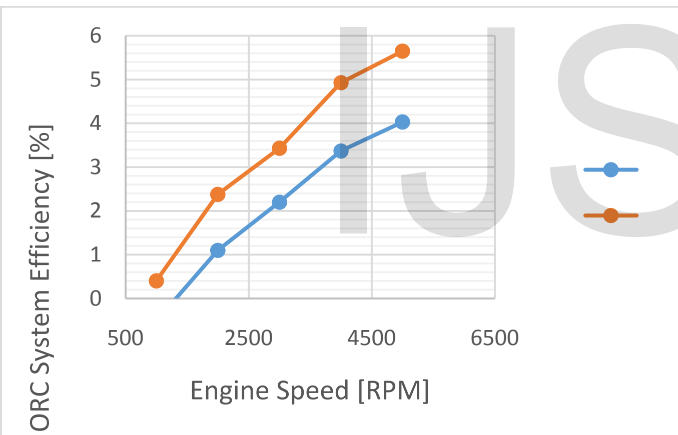

Figure 20. Comparison of FGT and VGT ORC System Efficiencies

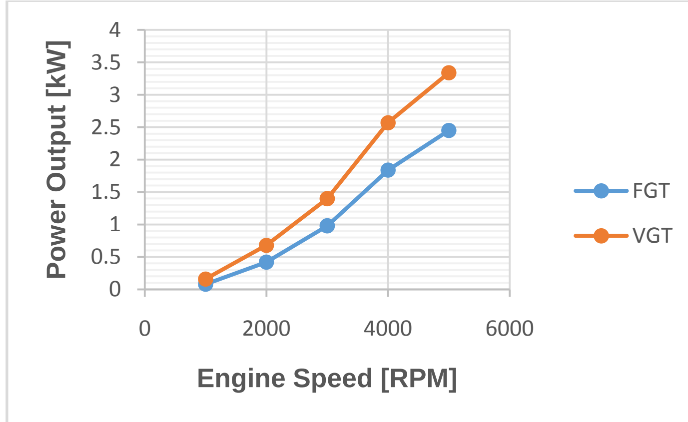

Figure 21. Comparison of FGT and VGT ORC Power Output

The implementation of an ORC system on the gasoline engine vehicle can improve the engine power, fuel consumption and emissions reduction. The target of this paper is to explore the potential for improvement of the ORC system power output when a VGT is implemented. Fig.22 presents the improvement on the engine net power due to the ORC system with and without the VGT technology. At maximum engine speed, the power is increased by 2.2 kW (4% increase) for FGT while by implementation of a VGT technology gives an additional, approximate 1% increase (i.e. 3.07kW with 5.6% increase) on the engine net power output (transmission losses neglected as this is a feasibility study). Regarding the other engine operating points, VGT shows an improvement on the ORC system exit power from 0.15% to 1.6% compared to the FGT.

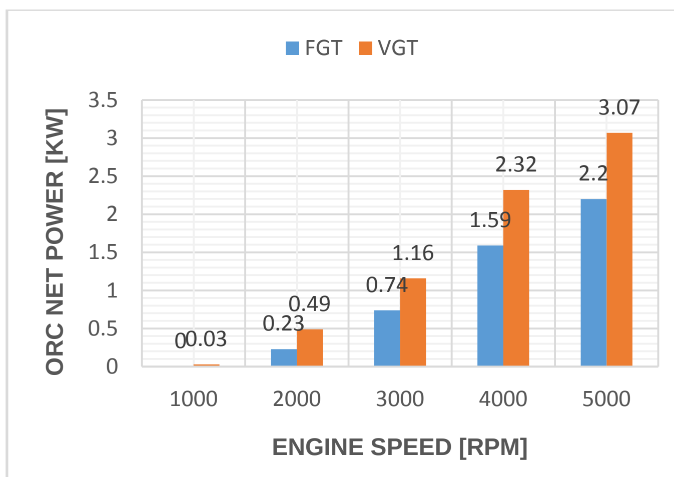

Figure 22. Comparing the ORC Net Power of FGT and VGT

The main purpose of waste heat recovery is the improvement of output energy from the fuel the engine consumed. This engine model simulation work illustrates the surplus energy that could be harnessed from the exhaust gas of gasoline engine through implementing ORC system WHR with VGT. In summary, net power can be improved up to 4% with a fixed geometry expander and up to 5.6% when a VGT is implemented as obtained from this study. In fact, implementation of VGT benefits include maximum energy output from fuel consumed, fuel economy and even reduction in engine emissions (environmental pollution).

CONCLUSION

In this research work, a variable geometry radial turbine (VGT) for waste heat recovery using an organic Rankine cycle for gasoline engine vehicle application has been investigated. A 1.25L Zetec engine model was designed on GT-Power engine software and calibrated using parameters of the actual 1.25l zetec engine as presented in table 1. The calibrated engine model was simulated across 2-9bar and 1000-5000rpm respectively for engine loads and speeds to reflect a real engine situations and the resulted exhaust gas conditions (exhaust temperature and mass flowrate) were used to calibrate and simulate an ORC system model with fixed geometry turbine (FGT) and then with variable geometry turbine (VGT) to compare their impact on WHR and ORC system efficiency from the available exhaust gas conditions.
It was observed that VGT can improve ORC system efficiency and net power output by an unweighted point of 5.6% and 3.07kW respectively at partial to high load conditions while benefits are even higher at the lower loads. This technology can also have an impact on the powertrain of gasoline engine vehicles and many other medium including light duty engines. The energy reclaimed from exhaust gas by power turbine can be fed directly to the engine shaft, allowing the engine to run at a correspondingly reduced input and to deliver the same power to the propeller shaft, or power a generator to provide electricity for electrical equipment for meeting general electrical demand. Both applications can save the fuel consumption and increase the distance travel. Fuel consumption and exhaust emissions are also reduced as added advantages while output power is increased by same magnitude.
Finally, it may be stated the performance of a VGT offers a substantial improvement in terms of relative fuel consumption gain compared to a conventional FGT. This makes it an attractive technology given its ability to recover low-grade heat and the possibility to be implemented in decentralised lower-capacity power plants. Steam cycles need high temperature, high pressures, and therefore high installed power in order to be profitable. The ORC offers a cost effective system for low power, lower temperature heat source applications where a steam plant would be expensive.

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