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Performance Assessment of a Heat Recovery Unit Utilizing Turbine with Variable Inlet Guide Vanes Configuration for Application in Passenger Vehicles

Thaddaeus Julius[1], Tanimu Kogi Ibrahim[1], Emmanuel Okon Asukwo[2], Ezeaku Ikeokwu Innocent[3]

[1]Federal University, Wukari, Nigeria
[2]Nigeria Maritime University, Okerenkoko, Nigeria
[3]Abia State University, Uturu, Nigeria
Email: thaddeus.julius@fuwukari.edu.ng

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Abstract

This study explores the potentials of employing an Organic Rankine Cycle (ORC) system with variable inlet guide vanes (VIV) turbine geometry designed on a GT-Suite platform for effective exhaust heat recovery (EHR) application onboard passenger vehicles. The ORC model simulation was based on vehicle speed mode using R245fa as working fluid to assess the thermal performance of the ORC system when utilizing modified turbine geometry. Interestingly, the model achieved a very improved performance in contrast to the model without a modified turbine configuration. The results revealed the average 2.32 kW ORC net output, 4.93% thermal efficiency, 6.1% mechanical efficiency, and 5.0% improved brake specific fuel consumption (BSFC) for the developed model. As determined by the performance indicators, these promising results from the model study show the prospect of EHR technology application in the transportation sector for reduction in exhaust emissions and fuel savings.

Keywords

Organic Rankine Cycle System, Variable Inlet Guide Vanes, Heat Recovery System, Mechanical Efficiency, Thermal Efficiency, Improved BSFC

1. Introduction

The transport industry is a huge contributor to challenges facing our environment today, including air pollution and global warming [1]. These environmental issues are caused by exhaust gas release during fuel combustion in internal combustion engines (ICEs), which still thrives in most vehicle propulsions. Today's concerns about these negative impacts of combustion exhaust emissions; increasingly stringent emission regulations benchmarked on the quantity of CO_2 expulsion from ICEs; and the need for sustainable and efficient deployment of fossil fuel resources in ICEs control the present trends of research and development (R & D) in automobile industries. Therefore, recapturing some portion of the exergy in this waste heat stream, which otherwise is exhausted to the environment, not only mitigate exhaust emissions and their adverse effects but as well lead to thermal efficiency improvement of the model through the generation of additional power to augment the existing one without extra-fuel consumption by the IC engine [2].

Reports have that 27% of the main causes of greenhouse gas (GHG) release in the United States (US) is from the transport industry, and 41% of this proportion is from passenger cars and motorcycles as published by American Transportation Research Institution (ATRI) [3]. Furthermore, ERTRAC's R & D forecast asserts that ICEs will still power more than 60% of new commercial vehicles in Europe up till 2040 owing to the need for energy density in some vehicle propulsions; thus, the expectation that ICEs will still control the transportation market for some time. Therefore, vehicle research trends will include ICE hybrids, extended configurations, waste heat recovery, and clean energy resources [4]. The exhausts released from ICEs of automobile vehicles comprise Nitrogen Oxide (NO_x), which causes air pollution, Particulate Matter (PM), which causes breathing challenges in people with pre-existing conditions; and CO_2, that is the major contributor to global warming threats. Technologies thus exist for reusing the thermal exergy in the exhaust of these vehicles such as Thermoelectric Generators, Turbo-compounding and ORC Systems being considered as a promising candidate technology for this application owing to its capability of operating effectively with limited heat sources, relatively increased thermal efficacy, and the simplicity of its architecture. ORC systems effectively achieve this heat recapturing by simply absorbing the exergy in the exhaust of these engines to evaporate the organic working fluid, that further expands to generate mechanical power, which is then converted to electricity employing an electric generator to supply power to the electrical appliances on board the vehicle or store in batteries for future use.

Recently, investigations on the deployment of ORC systems for EHR in vehicles are being reported in several online journals, and a few of such articles consulted are described in this project. In 2005, BMW's turbo steamer was reported as the first demonstration prototype system to use the Rankine cycle device to recover heat from exhaust stream to increase the automobile combustion process's efficiency. The cogeneration concept reduced fuel consumption in the 4-cylinder SI engine's stationary operating conditions by 10% [5]. Later, in 2007, Honda reported a Rankine cycle system installed in a hybrid vehicle for possible exhaust heat recovery from the engine. Testing the vehicle at 100 km/h constant speed achieved a thermal efficacy boost of 13.2% [6]. However, the company in 2008 reported that it is not going ahead with production until higher efficiency is realized [7].

In 2011, steady-state testing was conducted on a TOYOTA 8A-FE gasoline engine operated at varying work environments. The ORC model achieved a thermal efficacy of 14.44% [8]. The second-generation turbo steamer of BMW has been again revealed in 2012 to have been modified to facilitate effective incorporation of the Rankine cycle module for EHR [9]. In 2013, Domingues *et al.* reported a Rankine cycle model simulation employing water, R123, and R245fa as working fluids. The study achieved an increase of 2.12% and 5.79% for thermal and mechanical efficiencies, respectively [10]. While in 2014, Abbe *et al.* studied an ORC system for EHR application in vehicles, which realised a 3.5% boost in fuel savings [11].

In 2015, Rosset *et al.* studied the potentials of an ORC turbomachinery model for WHR applications in ICEs utilizing R1233zd as a working fluid. The system achieved 2.3 kW electrical output and a 10% boost in fuel savings [12]. Same year, Cipollone *et al.* proposed an ORC model for EHR applications in ICEs tested at different operating points employing R236fa. The model achieved cycle efficiency of 4.8% [13]. A review of EHR systems from exhausts of passenger vehicles and trucks using ORC techniques was reported by Zhou *et al.* [14]. The integration of an ORC system to an engine model for EHR was reported by Arise *et al.* in 2016. The model achieved significant boost in fuel savings with CO_2 reductions of 4% on standard driving cycles [15]. In 2016, the development of an ORC system for WHR from the exhausts of automobile engines was reported by Rongqi *et al.* using the Active Disturbance Rejection Control (ADRC) approach. The model works steadily and reliably with control error lower than 0.1% and fluctuation of superheating of less than 1˚C relative to traditional ORC system [16].

In 2017, Galindo *et al.* carried out a 1D modelling of an ORC module designed with a swash-plate expander for EHR from a 2 L SI engine employing ethanol as working fluid. The model achieved an expander output of 800 W, offering a potential of 2.5% improvement of fuel efficacy and a 23.5 g/kW reduction in BSFC when operating at 120 km/hr NEDC [17]. Investigations on the economic analysis of ORC application in the transportation sector were reported in 2017. The investigations revealed that ORC mass and volume must be reduced by 13% and 59%, respectively, for trains, while higher reductions are required for vehicular applications [18] [19]. In 2017, Amin *et al.* studied the deployment of a WHR system to a hybrid electric vehicle using three different driving cycles. The model simulation achieved a significant decrease in the overall fuel use of the vehicle [20]. In same 2017, Zhao *et al.* designed a recuperative ORC module to compare its performances to a traditional ORC model as applied to EHR from vehicle engines. The study showed that cooling water temperature reduces with a boost in net output for the recuperative ORC to the traditional

ORC. It also reveals that compared with basic ORC, the back pressure and engine performance are not affected when recuperation is added. However, more charged of refrigerant and longer response time are disadvantages [21].

In 2019, Yue *et al.* modelled an ORC system for EHR application in vehicles. The model achieved a maximum power output and fuel-saving rate of 14.7 kW and 0.23, respectively [22]. In same year, Volkswagen and MAN Diesel developers incorporated EHR systems to two of their conventional vehicles: Golf 7, 2-1 TFSI EA888, and Demo truck D2676 LF25 Euro VI, respectively. The models demonstrated a 75% evaporator efficacy and a 3% decrease in fuel use [23]. The feasibility and potentials of the EHR model for applications in long-haul trucks were reported in 2021. The numerical and experimental studies of the ORC model revealed an encouraging result with great feasibilities. The studies achieved 2.99 kW and 3.67 kW for net outputs achieved from numerical and experimental investigations, respectively [24] [25].

As evident from the reviewed literature, EHR in vehicles is being studied recently, and the technology is not in the market yet but still in the research stage, and more studies for improved performance and fuel savings are still anticipated. Therefore, investigating this ORC system equipped with a turbine utilizing variable inlet guide vanes (VIV) geometry running on vehicle speed presents another potential and predict real-life performance of the procedure as applied to onboard automobiles. There are scanty or no journal papers that report investigations of ORC system employing VIV modified turbine configuration and at the same time operating on vehicle speed mode, therefore, this research examines the performance of an ORC unit designed with modified turbine configuration and operating on vehicle speed mode for application in passenger vehicles.

2. Methodology

The engine model in this work is centred on a 1.25 L spark ignition engine with specifications presented in **Table 1** [26]. The exhaust heat recovery (EHR) system was modelled on a GT-suite platform to examine the exhaust conditions and the energy recovery prospects of the model dependent on vehicle speeds mode, and the description of the model set up is given in **Figure 1**. The modelling of the system was performed on imposed speeds of 62, 124, 186, and 248 km/hr to as much as possible achieve a real-life performance scenario for eventual consideration in future development and the expression used in this project for converting the engine revolution to vehicle speed was obtained [27] and is described by Equation (1).

$$k = d \times r \times 0.001885 \quad (1)$$

where, k = kilometre per hour (km/hr) d = wheel diameter (cm) r = revolution per minute (rpm).

The performance indicators examined in this work include engine BSFC, BMEP, ORC thermal performance, and VIV as tool for performance optimization as described subsequently.

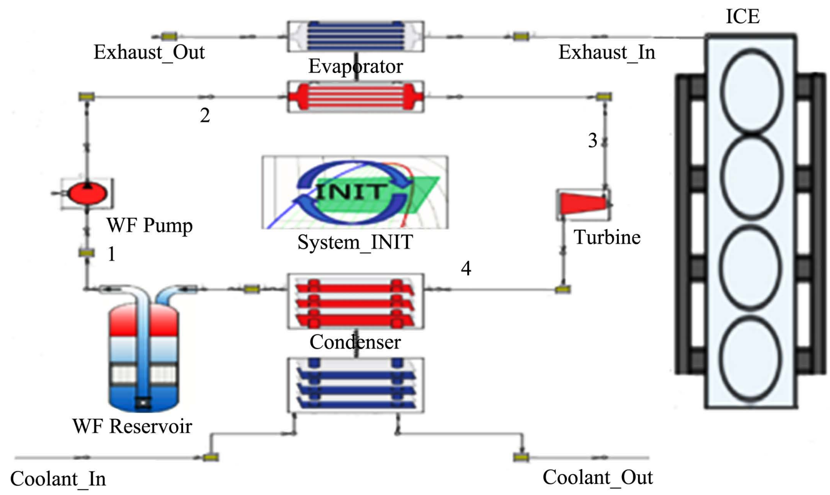

Figure 1. Diagram of the developed ORC model.

Table 1. 1.25 L Zetec-SE DHOC gasoline engine specifications [26].

ITEM	SPECIFICATION
Model	ZETEC-SE DHOC
Displacement (l)	1.25
Stroke (mm)	71.9
Bore (mm)	76.5
Number of Cylinder	4
Number of Valves	4
Maximum Torque	1100 Nm
Maximum Power	60 kW@4000 rpm
Fuel Type	Unleaded Gasoline

2.1. Engine BSFC

Brake Specific Fuel Consumption (BSFC) measures the engine's fuel to produce 1 kW of brake power per hour. The expression for evaluating this engine parameter is given as:

$$\text{BSFC} = \frac{\text{Fuel Flowrate}\left(\dot{m}_f\right)}{\text{Engine Power}\left(\dot{W}_{Eng}\right)}\left(\text{g/kW}\cdot\text{hr}\right) \tag{2}$$

2.2. BMEP

Brake Mean Effective Pressure (BMEP) is a theoretical indicator that measures the effectiveness of an engine, and the expression for evaluating this indicator is given as:

$$\text{BMEP} = \frac{2\pi \times T \times N_R}{V_d}\left(\text{bar}\right) \tag{3}$$

where V_d = swept volume, T = torque, and N_R = number of revolutions.

In the ORC module, the EHR process was modelled employing R245fa as working fluid owing to its high performance and commonly used in ORC for ICE applications, and the working fluid from the reservoir is pressurized in the pump *Process* 1 - 2, then sent to the evaporator. The combustion heat in the exhaust stream vaporizes the high-pressure working fluid in the evaporator heat exchanger; *Process* 2 - 3 then expands to create mechanical power at the turbine's shaft, *Process* 3 - 4. Finally, the low-pressure working fluid exiting from the turbine condenses back to liquid in the condenser *Process* 4 - 1, and the cycle repeats.

2.3. Optimization via VIV

In optimizing the efficacy of the designed EHR system, the concept of VIV is to direct the flow at an optimum direction to the turbine rotor for improved energy extraction, a characteristic option only available to radial turbine over its axial counterpart. The attribute also allows the turbine to operate both as a small- or large-scale turbine, thereby enhancing efficacy and reducing the warm-up period. The turbine wheel extracts energy from the fluid flow and converts it to mechanical energy that produces electricity utilizing a generator at the other end of the turbine shaft. The VIV optimization procedure was modelled by employing the guide vanes position at 20%, 40%, 60%, and 80% openings (Rack Positions) on the turbine stator.

2.4. Performance Evaluation

An outline of the equations governing the created ORC module used in evaluating the thermal effectiveness of the model is given as follows:

Heat input to the system via the evaporator

$$\dot{Q}_{in} = \dot{m}_{exh}\left(h_{exh\_in} - h_{exh\_out}\right) = \dot{m}_{WF}\left(h_3 - h_2\right) \tag{4}$$

where $\dot{m}_{exh}$ = mass flowrate of exhaust gas, $\dot{m}_{wf}$ = mass flowrate of working fluid, $h_{exh\_in}, h_{exh_{out}}, h_2$ and h_3 are enthalpies of exhaust and working fluid as defined in **Figure 1**.

Heat is expelled from the system through the condenser.

$$\dot{Q}_{out} = \dot{m}_w C p_w \Delta T_w \tag{5}$$

where $\dot{m}_{WF}$ and $\dot{m}_w$ are working fluid and cooling water mass flowrates, h_4 and h_1 are enthalpies at the various points in **Figure 1**, Cp_w is the specific heat capacity of the cooling water and ΔT_w is the change in temperature of the cooling water.

The electricity output available at the turbine is given as:

$$\dot{W}_{tur} = \eta_{mech}\eta_{gen}\dot{W}_{tur} - \frac{\dot{W}_{elec}}{\eta_M} \tag{6}$$

where $\dot{W}_{elec}$, $\dot{W}_{tur}$ and $\dot{W}_p$ are electricity output, turbine power out and pump consumption, η_{mech}, η_{gen} and η_M are mechanical efficiency, generator efficiency and mechanical efficiency of the pump.

The expression gives the electricity consumption by the pump:

$$\dot{W}_{elec} = \dot{m}_{wf}\left(h_2 - h_1\right) \tag{7}$$

where h_1 and h_2 are the enthalpies of the working fluid in and out of the working fluid pump.

The ORC thermal efficiency is given as:

$$\eta_{th} = \frac{\dot{W}_{tur} - \dot{W}_{elec}}{\dot{Q}_{in}} \times 100 \tag{8}$$

The engine BSFC improvement from the ORC system is given as:

$$\text{BSFC Improvement} = \frac{\text{Fuel Flowrate}}{\text{Engine Brake power} + \text{ORC Net}_{\text{FIV/VIV}}}\left(\text{g/kW}\cdot\text{hr}\right) \tag{9}$$

The expression employed in this study for evaluating the mechanical efficiency of the system is given by Equation (10).

$$\eta_{mech} = \frac{\dot{W}_{net}}{\text{Brake Power}} \tag{10}$$

3. Results Discussions

3.1. Engine Model Performance

This section provides the engine model effectiveness in forms of exhaust temperatures, exhaust mass flow rates, and engine BSFC from modelling the system on vehicle speed mode. The simulation was done at a cycle time of 1000 sec in each case, and the corresponding exhaust conditions are used as input variables to the ORC loop, which investigates the ORC EHR potentials. Figure 2 presents the exhaust temperature distribution as realized in the engine model presented as functions of vehicle speed and BMEP. The model achieved exhaust temperature values ranging from 393˚C to 796˚C at vehicle speeds of 62 to 248 km/hr and BMEP of 2 to 9.8 bar. This result reveals that the combustion exhaust temperature increases with increased engine speed and load; thus, the higher the operating vehicle speed, the better the available heat in the exhaust stream for recovering.

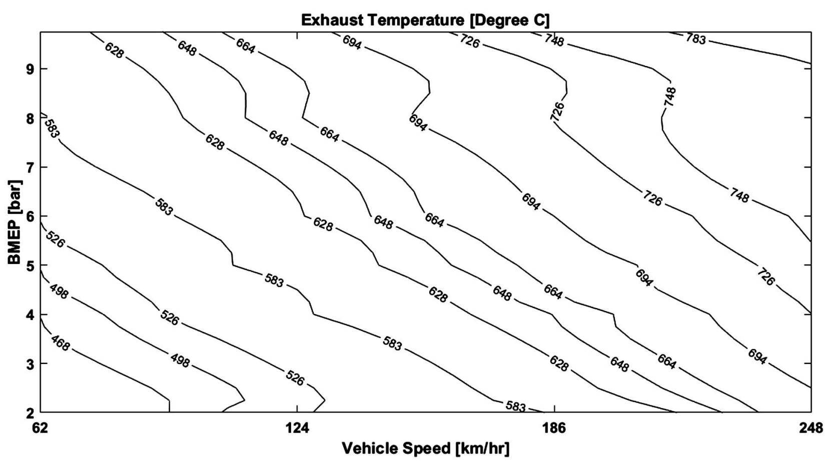

Figure 2. Exhaust temperature distribution.

Figure 3 shows the flow rate distribution of the engine exhaust achieved from the model simulation. The mass flowrate another critical component used in assessing the exhaust feasibility of EHR units for automobile applications. The flowrates obtained in this case fall with 18 to 260 kg/hr. The flow rates result reveal that the higher the vehicle speed and load, the better the flow rates of the exhaust stream. Thus, when running at cruising speeds, one can be sure of a better effectiveness of the EHR unit.

As revealed in the journals consulted, only about 30% to 40% of the fuel content is converted to useful brake power during combustion, whereas the residual portion of the energy is expelled to the surrounding in the forms of exhaust and engine cooling systems. Figure 4 presents the relationship between the engine brake power and fuel energy lost via the exhaust stream, which is available for reuse. The model realized the average 10 to 32 kW engine brake power and 6 to 42 kW of available exergy in the exhaust stream. The findings show that both the brake power and exergy in the exhaust increase with increasing speed and load. Comparing the two parameters gained from the model, it is obvious that at higher vehicle speeds, around 200 km/h above, the SI engine losses more of the fuel energy to the exhaust stream than the brake power for useful work.

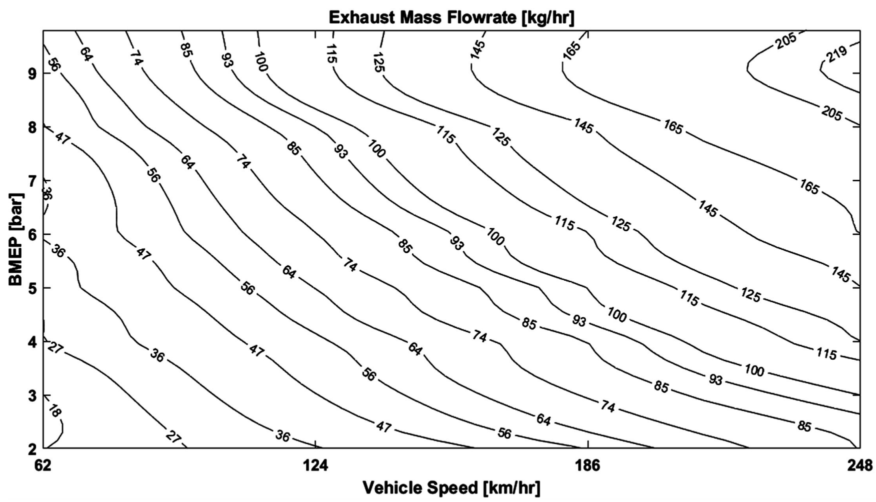

Figure 3. Exhaust mass flowrate distribution.

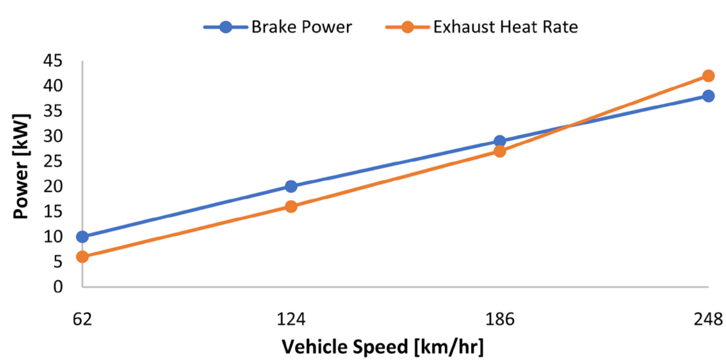

Figure 4. Engine brake power vs. power lost in exhaust.

3.2. Influence of VIV on the Performance of the EHR System

The ORC unit was optimized employing a VIV turbine to determine the extent of this concept's impact on the effectiveness of the EHR unit relative to that with fixed inlet vanes (FIV) turbine. Figure 5 presents the BSFC improvements from the deployment of the ORC model using regular FIV and VIV turbines for EHR application in the passenger vehicle using a SI engine. The model achieved the best BSFC performances at 124 km/hr. The BSFC results realized in this project include 253 g/kW·hr from the engine model and the BSFC improvement of 250 g/kW·hr and 248 g/kW·hr due to the deployment ORC system with FIV and VIV turbine configurations. The model demonstrates maximum BSFC improvement of 3.7% and 5.0% for the turboexpander with FIV and VIV, respectively.

Mechanical efficiency is another potential impact on the engine performance achieved from installing an EHR system.

Figure 6 compares the mechanical efficacy of the SI engine realized from modelling the ORC unit employing turbine with and without variable inlet guide vanes geometry for optimal flow direction. The results reveal mechanical efficiencies of 0.3% to 6.1% for the ORC model, employing a turbine using VIV as

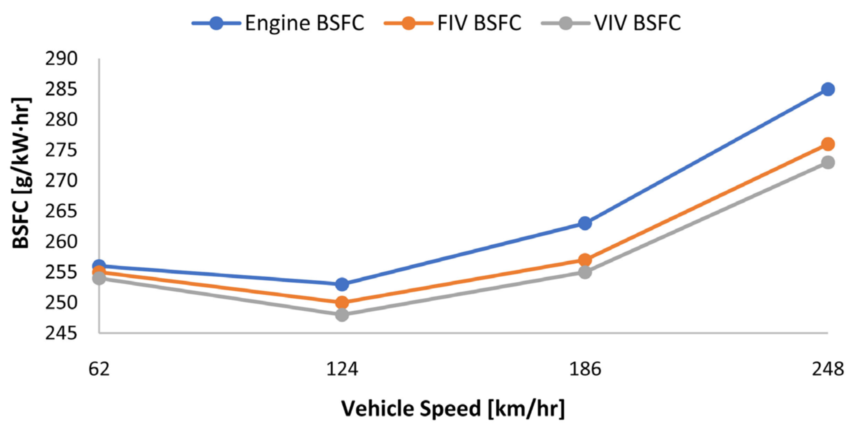

Figure 5. Comparison of BSFCs achieved due ORC system.

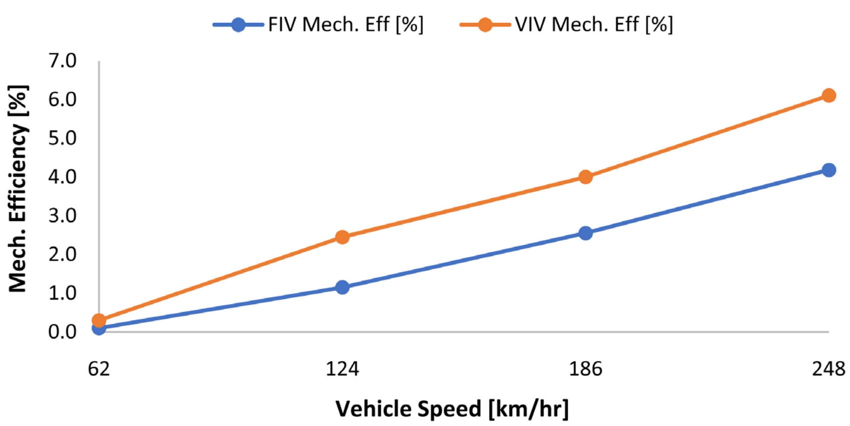

Figure 6. Mechanical efficiency due to ORC system.

against 0.1% to 4.2% from the ORC model utilizing a fixed inlet guide vane configuration. The findings show an approximately 31% boost in mechanical efficacy of the system realized from an EHR system employing a turbine with a VIV stator configuration. The results also showed that the mechanical efficacy of the SI engine increases with increasing engine speed, as demonstrated in both cases.

3.3. Thermal Effectiveness of the ORC System

The effectiveness of the ORC model is measured in forms of power output, ORC net, and thermal efficacy of the system, all examined as a function of engine speed. Figure 7 describes the evolution of the ORC system's net output with increasing vehicle speeds as demonstrated by the EHR system utilizing turbines with and without variable inlet guide vanes. The results present 0 to 1.59 kW as net output realized from the ORC model utilizing turbine with modified inlet guide vanes, while the introduction of a turbine with variable inlet guide vanes achieved a net output of 0.03 to 2.32 kW corresponding to a maximum boost of about 0.31 when operating at vehicle speed of 62 to 248 km/h. The outcome shows that 0.73 kW can be achieved as an addition when the turbine with VIV stator geometry is employed for the expansion process.

Figure 8 presents the efficacies of the ORC model with and without modification of the turbine stator configurations. As observed in the previous performance indicators of the EHR system, the modified turbine geometry achieved better effectiveness of the ORC model with up to 4.93% cycle efficiency relative to 3.37% shown by the turbine without modified geometry. This result reveals a huge potential for implementing an EHR system with modified stator geometry in the transport industry for energy sustainability and emission control targets.

Furthermore, the variable inlet guide vanes configuration was modelled at different rack positions: 20%, 40%, 60%, and 80% openings that would be automatically determined by the flow characteristic of the working fluid at the entrance of the turboexpander. Figure 9 shows the thermal efficacy realized by the various guide vanes positions of the turboexpander geometry as achieved in this study. The results reveal that the stator configuration at 0.4 vanes position

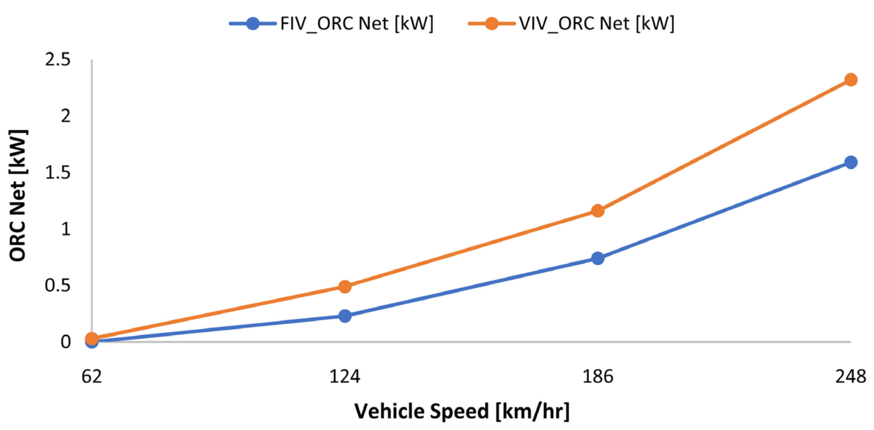

Figure 7. ORC net output.

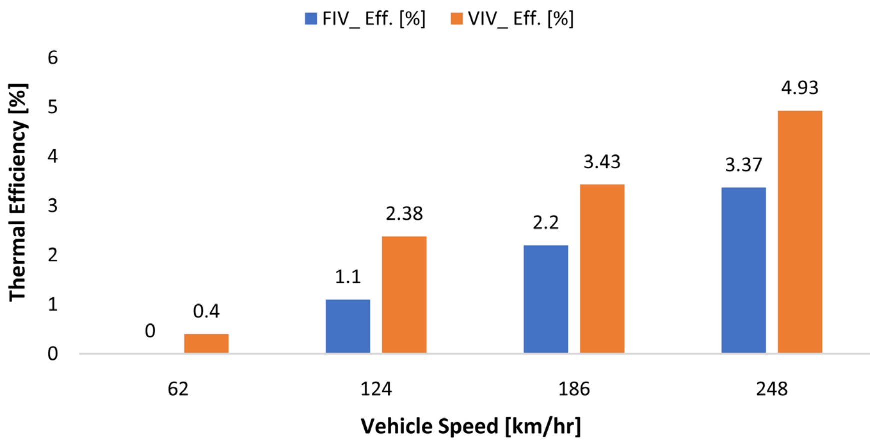

Figure 8. ORC model efficiency.

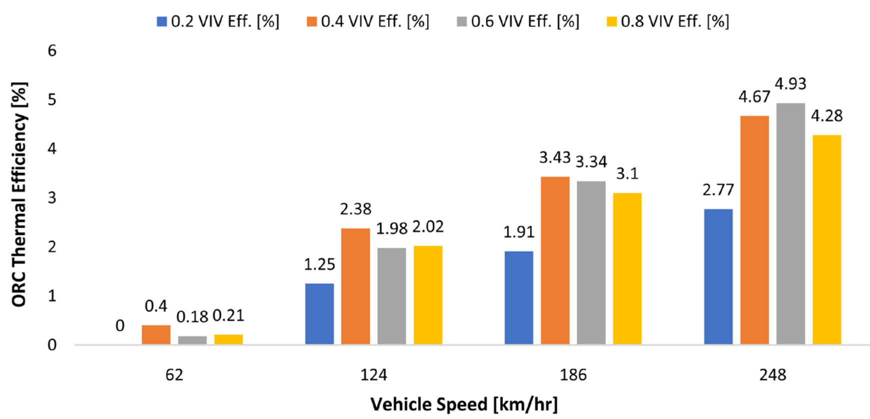

Figure 9. System efficiency achieved at VIV positions.

achieved the optimum averaged value of 2.72% thermal efficiency, followed by 2.61% for the structure with 0.6 vanes position, then 2.40% from the 0.8 vanes position and 1.48% for 0.2 position.

4. Conclusion

In this research, an ORC module with a turbine utilizing VIV configuration was designed for EHR application onboard passenger vehicle with SI engine, and the results compared with that of the ORC unit without a modified guide vanes configuration. The study achieved its primary objective of effectively recovering the thermal exergy in the exhaust stream, which would have exhausted the environment causing pollution and other negative environmental effects. The recovered energy is transformed into electricity to supply electrical power to appliances onboard the passenger vehicle or store in batteries for future use. The comparative findings revealed that the model with modified turbine geometry achieved 2.32 kW, 4.93%, 6.1%, and 5.0% for net output, thermal efficiency, mechanical efficiency, and BSFC improvement, as against 1.59 kW, 3.37%, 4.2%, and 3.7% for net output, system efficiency, mechanical efficiency, and BSFC improvement, respectively for system without modified turbine geometry. The results revealed the huge EHR potentials in employing an ORC system with a VIV turbine configuration in passenger vehicle applications.

Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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Nomenclature

Abbreviations

BTE	brake thermal efficiency
CHP	combined heat and power
EHR	Exhaust Heat Recovery
FIV	fixed inlet guide vanes
GHG	Greenhouse gas
GWP	global warming potentials
ICE	internal combustion engine
NEDC	new European driving cycle
ORC	organic Rankine cycle
PEMFC	Proton Exchange Membrane Fuel Cell
PM	particulate matter
R & D	research and development
rpm	revolution per minute
SI	spark ignition
T-s	temperature-entropy
UK	United Kingdom
US	United States of America
VIV	variable inlet guide vanes
WF	Working fluid
WHR	Waste heat recovery

Greek symbols

$\dot{m}_r$	Mass flow rate [kg/s]
$\dot{W}$	Work done [kW]
h	Enthalpy [kJ/kg]
η	Efficiency [%]
$\dot{Q}$	Thermal power [kW]
ρ	Density [kg/m^3]
ω	Speed [rad/s]

Subscripts

in	Inlet
mech	Mechanical
out	Outlet
th	Thermodynamic
turb	Turbine