Waste Heat Recovery from Combustion Engines Based on the Rankine Cycle

Vinay Kumar Yadav[1] Dilsher Khan[2] Anil Kumar Kurchania[3]
[1]Assistant Professor [2]M. Tech Scholar [3]Professor
[1,2,3]Department of Mechanical Engineering
[1,2,3]RNTU Bhopal, M.P, India

Abstract— Maximum amount of energy in the fuel used in *automotive* internal combustion engines, these days is lost in form of waste heat. Theoretically some of this energy can be converted to increase fuel efficiency by employing a *WHRS* (waste heat recovery system) to the engine. This paper *investigates* into the design and functioning of *WHRS* based on *Rankine Cycle* for to improve the design of such kinds of systems. For the purpose, performance of some working fluids through a zero-dimensional model of the Rankine cycle. One expander model was employed to rank the comparative influence of boundary conditions.

Keywords: Waste Heat Recovery, Combustion Engines, Rankine Cycle

I. Introduction

Latest technology trends related to combustion engines & vehicle design are mostly driven by the pressing need to lower the overall consumption of fossil fuels and the resulting emissions of CO_x & SO_x. The greenhouse effect alters natural marine and terrestrial carbon cycles, lowering the nature's capacity for CO_2 storage. In year 2005, the transport sector was mostly responsible for slightly more than 23% of the world's CO_2 emissions, with road transport governing for 17%. The largest share of the globe's CO_2 emissions (45%) originated from fossil fuels used for energy generation. Overall CO_2 emissions have increased by 80% ever since 1970, contributing to an average atmospheric temperature increase of around 0.8°C over the same period.

A. *The Rankine Cycle*

This cycle is the model of a perfect steam-power cycle. It was first clarified by the Scottish designer William. While this hypothetical Rankine cycle depends on a succession of reversible procedures, genuine frameworks include irreversible changes of state in the siphon and the extension gadget. The corresponding isentropic efficiency η is for devices is defined according to following equations. The effect of irreversibility on the pumping and expansion processes is indicated in Fig. 1

$$\eta_{is,\,pump} = w_{01,\,is}/w_{01}$$
$$\eta_{is,\,exp} = w_{23}/w_{01,\,is}$$

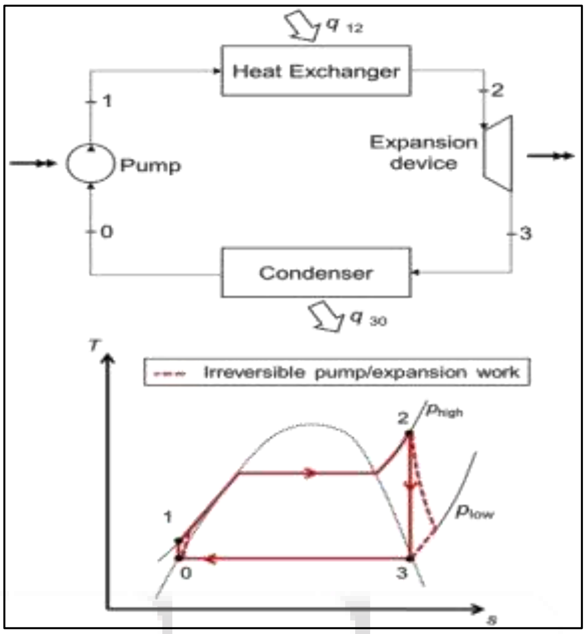


Fig. 1: Layout of an illustrative Rankine cycle-based heat recovery system (top) and a temperature-entropy (T-s) diagram showing the different stages of the cycle (bottom)

The thermal efficiency η_{th} of a Rankine cycle may be evaluated by considering its network output wnet and the heat input q_{12}, Eq. 3

$$\eta_{th} = w_{net}/q_{12} = [w_{23}]-[w_{01}]/q_{12}$$

Notably, Option has developed a heat-recovery system for ships that promises fuel savings of 5 – 10%.

II. Design of Rankine Cycles

Theoretical Rankine cycle may be employed to the boundary conditions of its desired working environment by finetuning some basic process parameters. Following part gives an overview of such parameters, especially emphasizing on WHRS.

A. *Working fluid*

1) *Water*

Conventional working fluid of Rankine cycle employed in power generation. It is considered by a lot of investigators for heat recovery in IC engines.

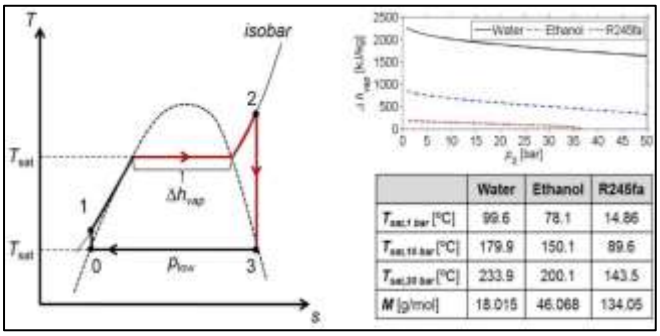
2) *Ethanol*

It is an organic fluid (alcohol) which is frequently used as a suitable alternate working fluid of the water for vehicles.

3) *R245fa*

It is also an organic fluid (hydro fluorocarbon) Which is one of the most important fluids in ORC applications. When considering isothermal vaporization (vap) of the fluid at saturation temperature, Tsat, the relation may be rewritten as presented in following equation-

$$\Delta s = Q/T_{sat}$$



	Water	Ethanol	R245fa
$T_{sat,1\,bar}$ [°C]	99.6	78.1	14.86
$T_{sat,10\,bar}$ [°C]	179.9	150.1	89.6
$T_{sat,30\,bar}$ [°C]	233.9	200.1	143.5
M [g/mol]	18.015	46.068	134.05

Fig. 2: Thermo physical properties for the three representative working fluids.

III. METHODOLOGY

One conventional heavy-duty Diesel engine (VolvoD13US10) was employed as the source of heat for the WHRS evaluated in this paper.

The 'average' formula of diesel is taken as $C_{12}H_{23}$.

$$C_{12}H_{23} + \lambda.17.75.4.762.(0.21.\ O_2 + 0.79.N_2) \rightarrow 12.CO_2 + 11.5.H_2O + \lambda.17.75.3.762.N_2 + 17.75.(\lambda-1).\ O_2$$

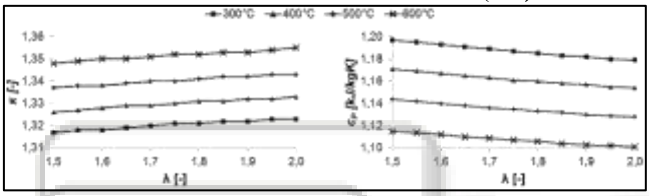


Fig. 2: specific heat ratio (left) and specific heat capacity cp(right) of Diesel exhaust

A. *Simulation Models*

The process used initially in this paper is theoretical and depends on zero- and one-dimensional modelling methods. The results found using such system and component models developed in the first half of the work is then employed to eventually direct the design of the demonstrator test bench.

B. *The Zero-Dimensional System Model*

One zero-dimensional model of the Rankine cycle is developed through EES. No pressure losses are assumed to occur in the cycle. All cycle components have been assumed to be adiabatic. Exhaust gas may be considered as ideal gas. Constant isentropic efficiencies have also been assumed for the pump and expander. Condensation pressure is more than the atmospheric pressure. The temperature of the heat sink is denoted by T_0.

$$N_s = N.\ VFR^{1/2}_{ex}/\Delta h^{3/4}_{is}$$
$$D_s = D.\ VFR^{1/2}_{ex}/\Delta h^{3/4}_{is}$$

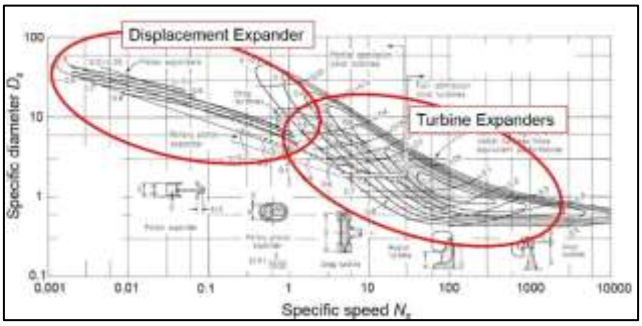


Fig.3 NS-DS turbine chart (imperial units), axis labels traced for better readability

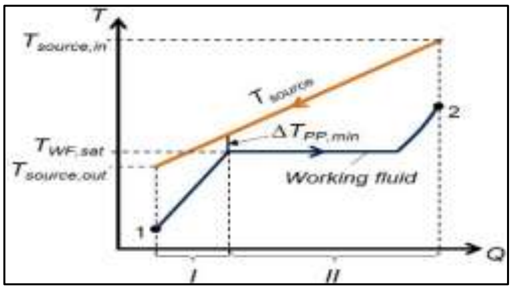


Fig. 4 Energy balance between a heat source and a working fluid (phase I – preheating; phase II - vaporization/superheating)

C. *Expander Efficiency Terminology*

Three measures of efficiency have been employed in connection with the expansion device here in this paper.

The *isentropic efficiency* η_{is}

$$\eta_{is} = w_{exp}/w_{exp,\ is} = w_{exp}/\ \Delta h_{is}$$

Mechanical efficiency

$$\eta_m = \eta_m.\ \eta_m\ w_{exp,\ shaft}/w_{exp}$$

The *overall expander efficiency* η_o is the product of the above two efficiencies

$$\eta_o = \eta_m.\ \eta_m\ w_{exp,\ shaft}/w_{exp} = w_{exp,\ shaft}/\ \Delta h_{is}$$

Number of expansion cylinders	2
Cylinder bore	90 mm
Cylinder stroke	60 mm
Inlet valve opening (IVO)	-17 °CA ATDC
Inlet valve closing (IVC)	34 °CA ATDC
Inlet valve diameter	23mm
Inlet valve max. lift	3.1 mm
Outlet port opening (EVO)	110 °CA ATDC
Outlet port closing (EVC)	-110 °CA ATDC
Outlet port diameter (1 per cylinder)	32 mm
Compression ratio (geometric)	21
Expansion ratio (TDC □ EVO)	15.67
TDC clearance	3 mm

Table 1: Geometric Data for the Reciprocating Piston Expander

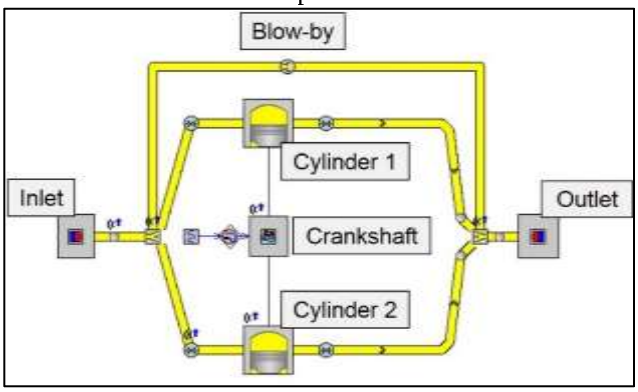


Fig. 5 1: D GT-SUITE model of the piston expander

$$FMEP = FMEP\ constant + A.p\ cyl,\ max + B.C_p,\ m + C.C^2pm$$

$$CD = m/m_{theor} = AE/AR$$

IV. EXPERIMENTAL SETUP

While designing and building the experimental test rig, it was intended to create a resource in order to validate the shown 1D models. An additional and even more important aim was to raise the understanding of the WHRS's behavior in "real world" conditions. Most published studies on heat recovery systems.

Fig. 6: Experimental setup of the heat recovery system in the engine test cell

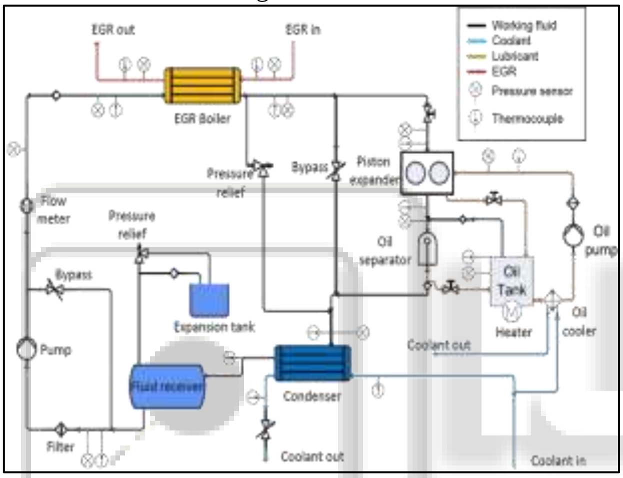


Fig. 7: Schematic layout of the heat recovery system

A. EGR Boiler

The EGR boiler is developed and supplied by Titan X which is one of the industrial partners in this research. Factors employed in its development included the EGR flow and temperature for the test rig.

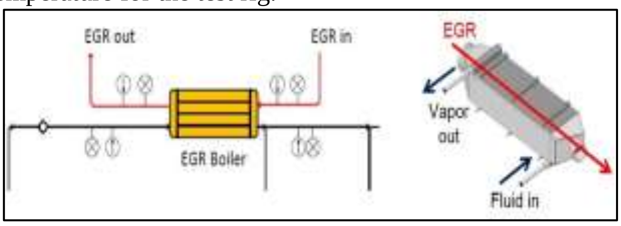


Fig. 12: The EGR boiler in the system layout (left) and as a CAD drawing (right)

B. Expander Subsystem

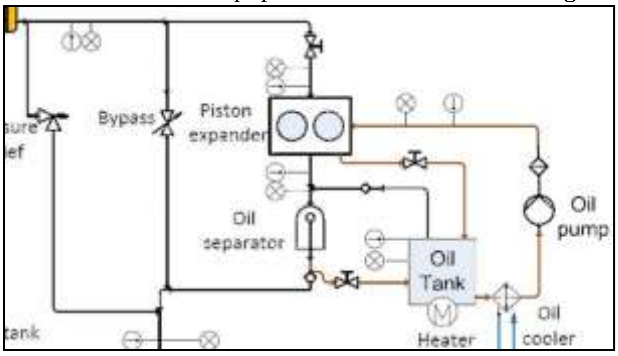
Volumetric/Mechanical efficiency: 43.9/72.5% at 1500 rpm and 120bar.
Pump speed range: 1000-3000rpm.
Max. Fluid temperature: 90°C.

C. Measurement Equipment

The Lab VIEW platform was employed to regulate the Rankine cycle test bench.

0	Sensor Principle	Range	Accuracy
Flow-rate (working fluid)	Carioles	0-2.72 kg/h	0.2% of reading
Flow-rate (coolant)	Turbine flow meter	0.8-80 liter/min	2% of reading
Temperature	Thermo -couple (K-type)	-40-1100 °C	< 2.4 °C @ max. 600 °C
Pressure	Piezo -resistive	0-10 bar, 0-60 bar	0.25% of range

Table 3: Sensor equipment installed on the test rig



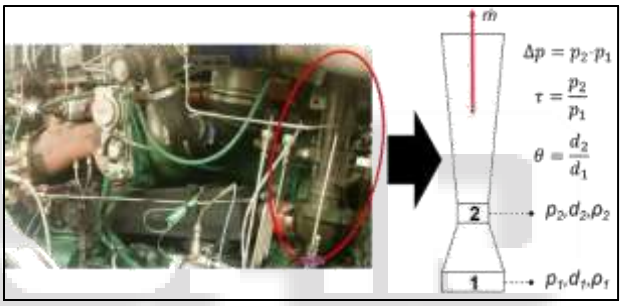


Fig. 13: EGR line with Venturi tube (left) and nomenclature for the Venturi pipe (right)

V. RESULTS AND DISCUSSION

This portion explains the data obtained through the analysis and the experimental work performed within the work. Important results obtained are explained briefly.

A. Zero-Dimensional System Model

The results found in the zero-dimensional model are explained basically arising from this work. The aim of these work is to:

1) Identify desirable working fluids to be used as Rankine cycle operating parameters and vehicle applicability.
2) Compare the expander results based on relevant heat boundary conditions, working parameters and working fluids.

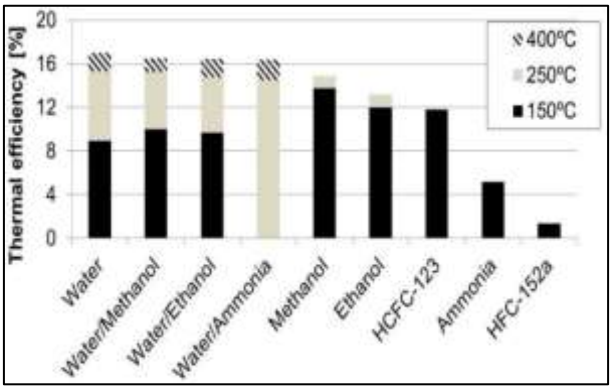


Fig. 8: Thermal efficiencies for chosen working fluids at three different temperatures

VI. BYPASSED SYSTEM

A major discovery of the set is that the WHRS did not cause major EGR cooling, resulting in post-boiler EGR temperatures nearly 200°C as compared to 80°C with the original EGR cooler.

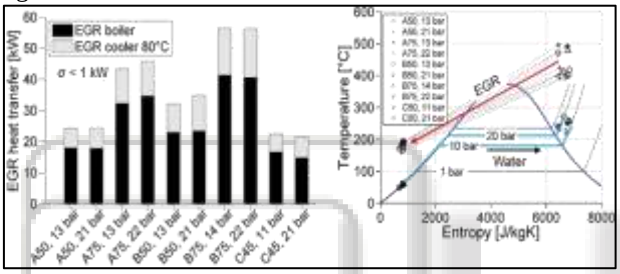


Fig. 9: EGR boiler heat transfer (left) and T-s diagram (right) at various ESC operating points and boiling pressures

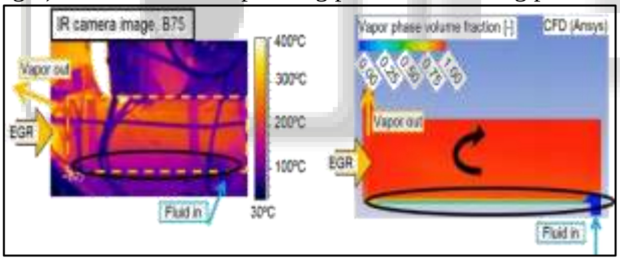


Fig. 10: IR image (left) and CFD result (right) provided by TitanX for the EGR boiler

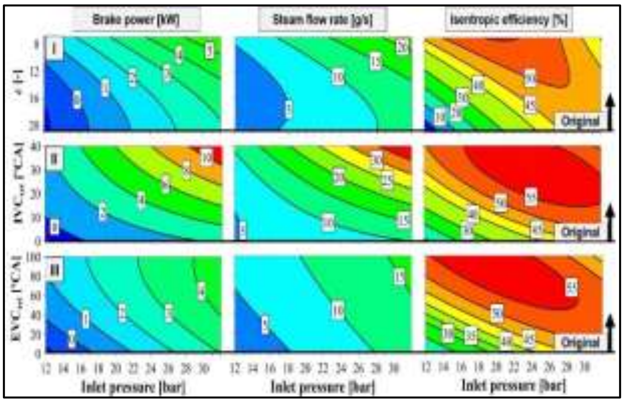


Fig. 11: Influence of compression ratio ε (I), retarded IVC (II) and retarded EVC (III) on expander bp, steam flow rate & isentropic efficiency (speed 600 rpm)

VII. CONCLUSION

The IC engine is the most important heat engine but still has one major drawback: most of the current engine designs loose more than 50% of the supplied fuel energy as heat losses. There is an important need to raise the efficiency due to the danger of climate change and the earth's diminishing fossil fuel reserves, making the use of WHRS for IC engines more and more attractive.

This paper addresses a lot of key design challenges associated with the design of a vehicular WHRS based on the Rankine cycle, including the selection of a suitable working fluid and evaluation of the inter connections between the fluid choice, the cycle parameters and the expansion device.

It was considered important for the project to include an experimental component which could provide data to determine the quality and reliability of the conclusions drawn from the theoretical models. A full-scale demonstrator test rig using a Rankine cycle was designed based on predicted operating parameters from the simulations.

REFERENCES

[1] Edwards S., "Waste Heat Recovery: The Next Challenge for Truck Engine Development", Behr GmbH & Co. KG, Technical Press Day, 2010.
[2] Wang T., Zhang Y., Peng Z. and Shu G., "A review of researches on thermal exhaust heat recovery with Rankine cycle", Renewable and Sustainable Energy Reviews 15(6):2862-2871, 2011, doi:10.1016/j.rser.2011.03.015.
[3] Ismail Y., Durrieu D., Menegazzi P., Chesse P. et al., "Potential of Exhaust Heat Recovery by Turbocompounding," SAE Technical Paper 2012-01-1603, 2012, doi:10.4271/2012-01-1603.
[4] Greszler A., "Diesel Turbo-compound Technology", presented at ICCT/NESCCAF Workshop, USA, February 20th 2008.
[5] Rowe D.M., "CRC Handbook of Thermoelectrics", CRC Press, Florida, USA, ISBN 978-0-8493-0146-9, 1995.
[6] Stobart R., Wijewardane A., and Allen C., "The Potential for Thermo- Electric Devices in Passenger Vehicle Applications," SAE Technical Paper 2010-01-0833, 2010, doi:10.4271/2010-01-0833.
[7] Baker C. and Shi L., "Experimental and Modeling Study of Heat Exchanger Concept for Thermoelectric Waste Heat Recovery from Diesel Exhaust," SAE Technical Paper 2012-01-0411, 2012, doi:10.4271/2012-01-0411.
[8] Hussain Q., Brigham D., and Maranville C., "Thermoelectric Exhaust Heat Recovery for Hybrid Vehicles," *SAE Int. J. Engines* 2(1):1132-1142, 2009, doi:10.4271/2009-01-1327.
[9] Mori M., Yamagami T., Oda N., Hattori M. et al., "Current Possibilities of Thermoelectric Technology Relative to Fuel Economy," SAE Technical Paper 2009- 01-0170, 2009, doi:10.4271/2009-01-0170.

[10] Rankine, W.J.M., “A Manual of the Steam Engine and Other Prime Movers”, Richard Griffin and Company, Publisher to the University of Glasgow, Scotland, 1859.
[11] Turns, S.R., “Thermodynamics: Concepts and Applications”, Cambridge University Press, UK, ISBN 978-0521850421, 2006.
[12] Kökeritz J., “Innovation as business drive-energy efficiency as a drive for sustainable production of power”, presented at Green Tec Conference in Hannover, Germany, April 25th 2012.

