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Research Article

Experimental investigation of axial finned tube evaporator thermal distillation system using for diesel engine waste heat recovery process

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ABSTRACT

The study aims to improve the waste thermal energy retrieval from flue gas of an internal combustion engine (ICE). The recovered waste heat energy was used for distillation by using a thermal distillation system. The performance of the thermal distillation unit was investigated by varying the evaporator (boiler) type and engine load (25, 50, 75 %). Four different types of boilers were used including one smooth copper tube and other three were two, three and four axial finned copper tube evaporators. The impact of boiler type and engine load on the net retrieved energy and exergy, net energy and exergy efficiency, and distillate yield rate of thermal distillation unit was also examined. The results showed that the net extracted heat energy and exergy for axial finned tube evaporator was approximately 26.823 – 45.513 % and 7.614 – 25.203 W higher than that of smooth tubes evaporator at 25 and 75 % engine load, respectively. The distillation yield was found to be ~ 2.35 liter/ hour in the case of four axial finned tube boiler at 75 % engine load.

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INTRODUCTION

In today's world, technological breakthroughs have improved the quality of life for everyone. These advancements have raised the global energy demand, resulting in an increase in energy prices [1]. The net global energy consumption has reached to 132 PW h, out of which 72 % of energy is lost in ecosystem as waste energy [2]. Nearly three-quarters of waste heat is attributed to exhaust and effluent losses that can be retrieved, while remaining one-fourth is unrecoverable [3].

Waste heat energy also contributed to global warming and had an adverse impact on our ecosystem. To deal with current energy crises, global warming and hike in energy prices, scientific community is focused to hunt novel pathways to improve energy efficiency [4]. Over the last decades, policymakers have implemented several rules and regulations to reduce or eliminate greenhouse gases (GHGs) [5]. Such initiatives have aided rapid development in this subject. One of the cost effective and convenient ways to improve energy efficiency is to recover waste heat energy. Consequently, the recovery of dissipated heat energy has

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emerged as one of the hottest research topics. Farhat et al. [6] studied on waste heat recovery methodology and applications for industrial, residential and internal combustion engine waste thermal energy. It was recommended that the optimized waste recovery system should be used for better efficiency and effectiveness of systems. Varshil and Deshmukh [7] also reviewed the recovery of lost energy from compression ignition engines. It was concluded that exergy efficiency of single loop and combined loop waste heat recovery system were 35% and 46%, respectively.

Hussein et al. [8-12] conducted theoretical, experimental and numerical studies on nano-fluid for energy applications. As a result, the efficiency of renewable energy systems was improved by using nanotechnology. Hussein et al. [13-14] also investigated the performance of double filled glazing unit by using phase change material (PCM). It was observed that the thermal energy storage capacity varies with different thermos-physical properties of PCM. Benabderrahmane et al. [15] presented central corrugated insert in parabolic trough receiver. It was found that average tube side Nusselt number and friction factor was enhanced by 3.7 and 1.8 times relative to smooth absorber, respectively. Ghodbane et al. [16] developed a solar-driven ejector air conditioning system. It was found that the coefficient of performance of ejector air conditioning subsystem was 60.66%.

Research in field of waste heat recovery (WHR) has been accelerated, particularly with regard to fossil fuel systems in numerous industries including aluminum industry, ceramic industry, desalination, and many more [17–20]. WHR units not only reduces the fuel consumption but also prevents adverse environmental impacts including GHGs emission and carbon footprints [21,22]. There are some important strategies have been proposed for WHR such as heat-to-heat, heat-to-work and heat-to-power [23]. Efficiency of WHR is maximum in the case of heat-to-heat strategy. This is attributed to the increase in thermodynamic energy losses when converting heat-to-heat, then to work and finally, to power. Among these strategies, heat-to-heat WHR strategy is the easiest way to recover heat with the help of heat exchangers [24]. The efficiency of WHR also depends on variety of situations including continuous or discontinuous supply, temperature, the mode of heat transfer, type and composition of fuel, type of heat exchanger and distance between source and sink [3]. In spite of the difficulties, there is a tremendous potential in WHR units to meet emergent energy demand.

Recently, WHR units are also used to power desalination units [25]. Such approach not only improve the energy efficiency but also resolve the global problem of fresh water scarcity. Feria-Diaz et al. [26] studied on an optimized thermal desalination system to hybridized with the existing desalination system. The hybridized desalination system was operated with the help of renewable energy. As a result, a reduction in dependences on electrical and combustion energy reduces. Dumka and Mishra [27] performed experimentation to recover the coolant waste thermal energy with the help of double slop long still (DSLS). Recovered thermal energy was used for water distillation. It was found that evaporative heat transfer coefficient reduces by approximately 64.40% from entrance to exit of DSLS. Morciano and co-workers [28] developed waste heat driven passive membrane desalination system. The waste energy was retrieved from ICE fueled with diesel. The results revealed that ~ 2.61 L/m^2h fresh water was produced from the seawater. The type of heat exchanger plays a crucial role on the overall efficiency of waste heat driven desalination system. Sertkaya and Sari [29] performed an experimental investigation on longitudinal finned and radial finned tube heat exchanger for natural heat energy transfer. Heat exchanger performance was estimated on different inclination of the tube. It was found that finned and inclined heat exchanger performed better than smooth tube. G. Krishnayatra and co-workers [30] also investigated the performance of axial finned tube heat exchanger with various geometric parameters. It was concluded that effectiveness of the fin increases with increasing fin spacing and fin thickness. These studies revealed that the axial finned heat exchangers are suitable WHR units. Moreover, impact of heat exchanger type on efficiency of waste heat driven distillation unit is scarce in literature. Therefore, it is crucial to conduct a systematic study to investigate the role of axial finned tube evaporator on distillation productivity. In the present study, four different types of heat exchangers were used to examine performance of waste heat driven distillation unit. One of these heat exchangers was smooth copper tube and other three are axial finned copper tubes with two, three and four fins. The impact of engine load and the number of axial fins on the distillation productivity was also determined.

EXPERIMENTAL DETAILS

A 7 horse power (HP) single cylinder water cooled compression ignition engine was used in the present investigation. A dynamometer was connected to the engine through a shaft to measure the engine load precisely. Digital anemometer was installed at the air intake port of the engine to estimate the air consumption in terms of velocity. The diesel container was placed at appropriate elevation and

Table 1. Technical detail of test engine

Parameters	Specifications/ dimensions
Type of engine	Kirloskar
Cooling	Water
Diameter of cylinder (mm)	87.5
Stroke length (mm)	110
Governing speed (rpm)	1500
Rated power (kW)	5.22
Compression ratio	17.5:1
Working fluid	Diesel

connected to the engine through burette. The consumption of fuel was measured through burette. Technical detail of engine is presented in Table 1.

Shell and tube evaporators were used to retrieve low grade waste thermal energy of exhaust gas. Shell and tube of evaporators were made of mild steel and copper materials, respectively. Total four evaporators were used in the present study. One evaporator was made of smooth copper tube. Three other evaporators were modified with longitudinal finned (two, three and four axial fins) on the tube surface. Thickness and height of the axial fin was 2 and 5 mm, respectively. Smooth and modified tube of evaporators were arrayed at the bottommost of the shell; so that, it can properly submerge in water. Coolant fluid (water) was allowed to pass through shell side, while exhaust gas through tube side as working fluids. Figure 1 presents the horizontal submerge tubes with different modified tube surface. A shell and two pass U-tube condenser were installed and connected through tube to evaporator. Shell and tube of condenser were made of mild steel and copper, respectively. Dimension of boilers and condenser used in the present work are tabulated in Table 2.

Methodology

Experimental setup was equipped with thermocouples, pressure gauge, rotameter and tachometer. K- type of thermocouples were positioned at appropriate places to measure and record temperature. All thermocouples have been calibrated with the help of thermal bath and multimeter. Pressure gauge was installed at the center of the boiler to measure the steam pressure. Used pressure gauge was calibrated by pressure hand pump and digital pressure gauge. Two separate flowmeters (rotameters) were mounted to determine coolant fluid flow rate in boiler and condenser. Each rotameter was standardized by glass beaker. Diesel fueled ICE speed was measured in RPM through a digital non-contact tachometer. The engine speed was calibrated with non-contact tachometer. Schematic view and photograph of the test unit is presented in Figure 2-3.

The experiment was initiated by starting an engine manually and then, the electrical load was applied. Initially engine exhaust was discharged into the surrounding environment by opening valve-1 and closing valve-2. This was done to avoid flue gas stream dust accumulation on surface of evaporator tubes. Later, valve-1 was closed and valve-2 was opened to flow flue gas through copper tubes of the

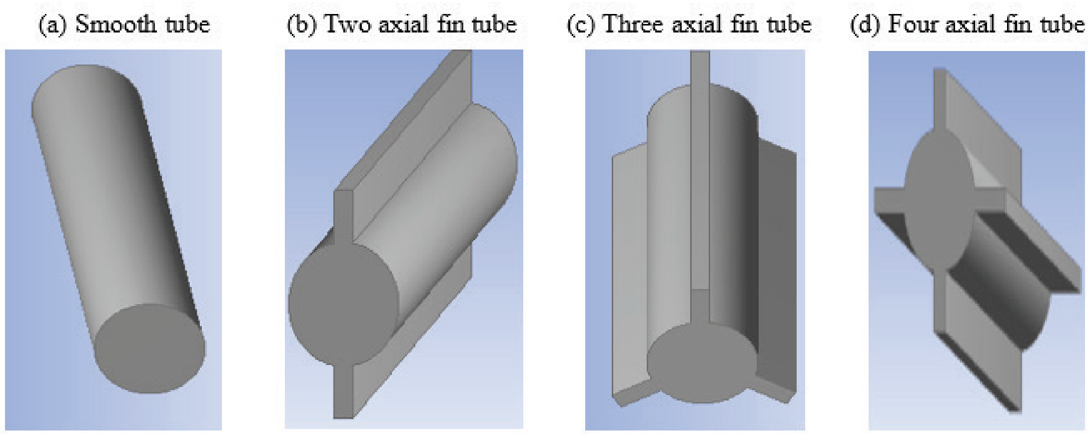

Figure 1. Representation of (a) smooth tube, (b) two axial fin tube, (c) three axial fin tube and (d) four axial fin tube evaporators (boilers).

Table 2. Evaporator and condenser dimension details

Parameters	Evaporator		Condenser	
	Dimension (cm)	Material	Dimension (cm)	Materials
Shell diameter	15	Mild steel	10	Mild steel
Shell length	80	Mild steel	40	Mild steel
Tube diameter	1.2	Copper	1.2	Copper
Tube length	50	Copper	60	Copper
Number of tube	7		-	-
Feed water tube diameter	0.6	Copper	-	-

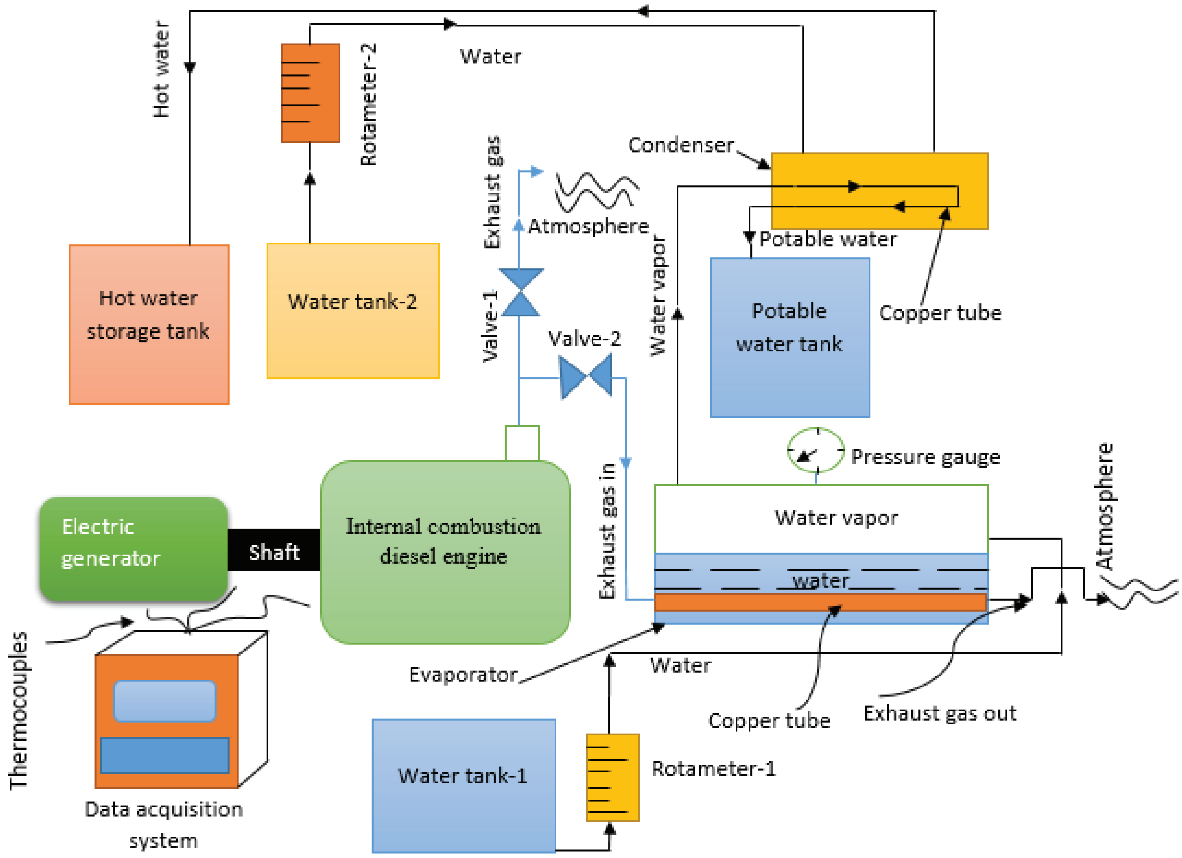

Figure 2. A schematic view of test unit.

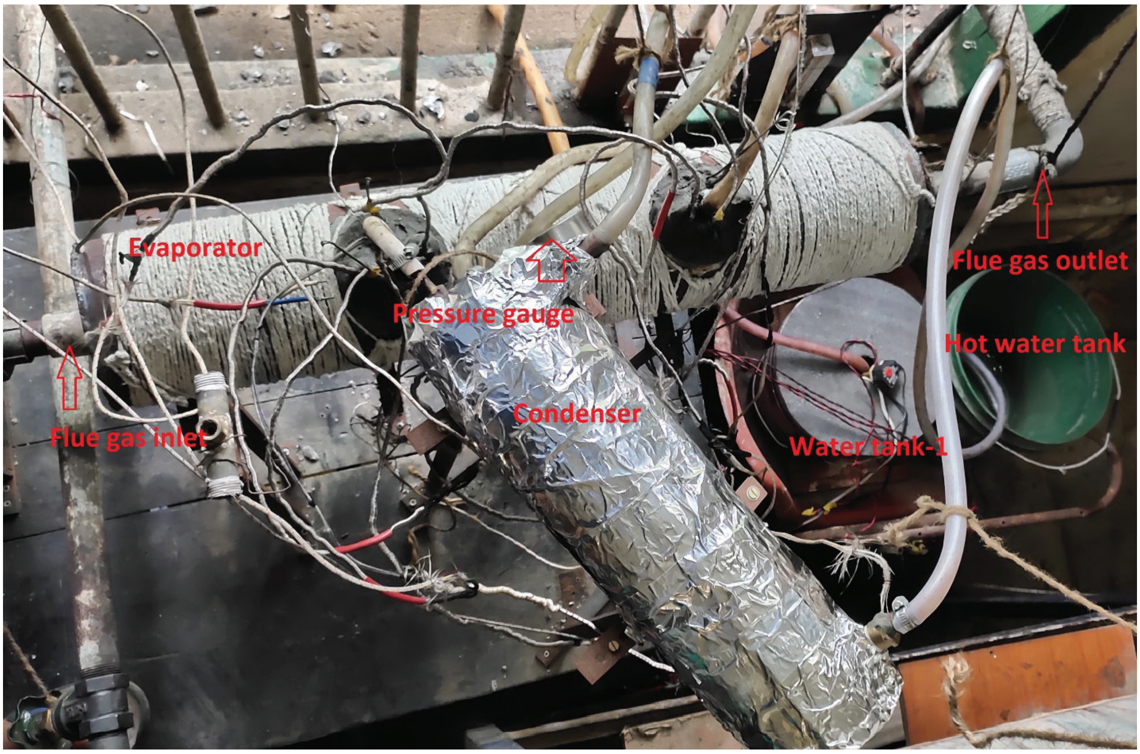

Figure 3. Photograph of experimental system.

boiler, which were submerged in water and released to atmosphere. Feed water was supplied in evaporator from water tank-1 at 30 °C, and flow rate was fixed at 2.5 LPH by Rotameter-1. Exhaust gas waste thermal energy was extracted by water and used for the conversion water into vapors. The saturated steam was accumulated on the surface of boiler coolant fluid and allowed to flow out from the evaporator. Afterwards, the steam was supplied to a condenser copper tube at one end. On the shell side, coolant fluid was supplied to condenser from tank-2 and mass flow rate was regulated through flowmeter-2. Shell side condenser water was heated by recovered heat from the tube side vapors. Later, vapors were allowed to condensed in a condenser copper tube. Finally, potable water (distillated yield) was obtained from other end of the condenser copper tube. Potable water was stored into potable water tank. The complete procedure was repeated for 100 minutes to each load. The total yield of distilled water was calculated by measuring glass beaker. Above experimental procedure was used for each type of boiler and at different engine loads 25, 50 and 75 %.

Data Reduction

Some assumptions were considered throughout experimentation work which are listed below:

(i) Flue gas is supposed as ideal gas.
(ii) The Specific thermal capacity of flue gas at constant pressure (C_{pf}) is temperature dependent.
(iii) Potential and kinetic energies of combustion gas do not change significantly.
(iv) Neglecting thermal energy losses to the environment.
(v) Conduction in longitudinal direction of the tubes is insignificant.
(vi) All parameters are measured after achieving the steady state.

Energy analysis of evaporator

Heat energy retrieved from the combustion gas was calculated at considered load condition by applying the equation:

$$\dot{q}_f = \dot{m}_f \times \Delta H_f \tag{1}$$

Specific thermal capacity of combustion gas can be written as [31]:

$$c_{pf_i} = 1.015 \times 10^3 - 1.512 \times 10^{-1} \times t_{f_i} + 4.544 \times 10^{-4} \times t_{f_i}^2 - 1.785 \times 10^{-7} \times t_{f_i}^3 \tag{2}$$

By using heat capacity of flue gas, the formula for change in enthalpy of combustion gas was modified by Hatami and co-authors [32]. Finally, it is possible to write the modified equation as

$$\Delta H_f = 1.015 \times 10^3 (t_{f_o} - t_{f_i}) - 0.756 \times 10^{-1} (t_{f_o}^2 - t_{f_i}^2) + 1.514 \times 10^{-4} (t_{f_o}^3 - t_{f_i}^3) - 0.446 \times 10^{-7} (t_{f_o}^4 - t_{f_i}^4) \tag{3}$$

A quantity of flue gas is discharged by the diesel engine.

$$\dot{m}_f = \dot{m}_{air} + \dot{m}_{diesel} \tag{4}$$

$$\text{Heat energy enters the boiler tube} = \dot{m}_f \times c_{pf_i} \times t_{f_i} \tag{5}$$

The required heat energy can be calculated by using equation [33]:

$$\dot{q}_{req} = \dot{m}_{ve_o} \times L_v \tag{6}$$

According to low of energy balance, heat lost by combustion gas is equal to heat required for evaporation of water. So, equation (1) is equal to equation (6).

$$\dot{m}_f \times \Delta H_f = \dot{m}_{ve_o} \times L_v \tag{7}$$

Energy analysis of condenser

Heat carried by condensed water is

$$\dot{q}_{cw} = \dot{m}_{cwc_o} \times c_{pcw} (\Delta t)_{cw} \tag{8}$$

Heat gain by coolant fluid is

$$\dot{q}_{cc} = \dot{m}_{cc_i} \times c_{pc} \times (t_{cc_o} - t_{cc_i}) \tag{9}$$

The condenser was designed at an evaporation rate of water in the boiler. Heat energy is loosed by water steam in the condenser to condensed the steam into distilled water. Liberated heat is used for heat up cooling water, and remaining thermal energy was also carried by distilled water. Both fluids of condenser have been used as useful products. According to conservation of energy, energy loss is equal to energy gain. So the equation is [33]:

$$\dot{m}_{ve_o} \times L_v = \dot{m}_{cwc_o} \times c_{pcw} (\Delta t)_{cw} + \dot{m}_{cc_i} \times c_{pc} (t_{cc_o} - t_{cc_i}) \tag{10}$$

Thermal energy loosed by water vapor in condenser to convert into condensed water = $\dot{q}_{vc}$

$$\dot{q}_{vc} = \dot{m}_{cwc_o} L_w + \dot{m}_{cwc_o} \times c_{pcw} (t_{swe} - t_{cwc_o}) \tag{11}$$

$$\text{Net retrieved energy} = \frac{\dot{q}_{cc} + \dot{q}_{vc}}{2} \tag{12}$$

Net energy efficiency of thermal distillation system is:

$$\eta_I = \frac{Net\ recovered\ energy}{thermal\ energy\ supply\ at\ inlet\ of\ evaporator\ tube} \tag{13}$$

Exergy analysis of evaporator

Change of exergy of flue gas is:

$$\Delta\emptyset_f = \dot{m}_f \times (\Delta H_f - T_0 \Delta s_f) \tag{14}$$

The mathematical relation to the change in the entropy of the combustion gas was modified by Hatami and co-authors [32] using specific heat of flue gas. Final equation can be written as

$$\Delta s_f = 1.015 \times 10^3 ln\left(\frac{t_{fo}}{t_{fi}}\right) - 1.512 \times 10^{-1}(t_{fo} - t_{fi}) + 2.272 \times 10^{-4}(t_{fo}^2 - t_{fi}^2) - 0.595 \times 10^{-7}(t_{fo}^3 - t_{fi}^3) - Rln\left(\frac{P_{fo}}{P_{fi}}\right) \quad (15)$$

By putting the value of ΔH_f and ΔS_f from equation (3) and (15) in equation (14)

$$\Delta \emptyset_f = \dot{m}_f \times \left[1.015 \times 10^3(t_{fo} - t_{fi}) - 0.756 \times 10^{-1}(t_{fo}^2 - t_{fi}^2) + 1.514 \times 10^{-4}(t_{fo}^3 - t_{fi}^3) - 0.446 \times 10^{-7}(t_{fo}^4 - t_{fi}^4)\right] - T_0 \times \left[1.015 \times 10^3 ln\left(\frac{t_{fo}}{t_{fi}}\right) - 1.512 \times 10^{-1}(t_{fo} - t_{fi}) + 2.272 \times 10^{-4}(t_{fo}^2 - t_{fi}^2) - 0.595 \times 10^{-7}(t_{fo}^3 - t_{fi}^3) - Rln\left(\frac{P_{fo}}{P_{fi}}\right)\right] \quad (16)$$

Exhaust gas enters into the evaporator with exergy =

$$\dot{m}_f(\emptyset_{fi} - \emptyset_0) \quad (17)$$

Water enters into boiler with exergy:

$$\emptyset_{ce_i} - \emptyset_0 = \dot{m}_{ce_i} \times (H_{ce_i} - H_0) - T_0(s_{ce_i} - s_0) \quad (18)$$

Vapor comes out from boiler with exergy:

$$\emptyset_{ve_o} - \emptyset_0 = \dot{m}_{ve_o}(H_{ve_o} - H_0) - T_0(s_{ve_o} - s_0) \quad (19)$$

$$\text{Retrieved exergy} = \emptyset_{ve_o} - \emptyset_{ce_i} \quad (20)$$

Exergy analysis of condenser

Condensed water exergy at exit of condenser

$$\emptyset_{cwc_o} - \emptyset_0 = \dot{m}_{cwc_0}(H_{cwc_o} - H_0) - T_0(s_{cwc_o} - s_0) \quad (21)$$

Coolant fluid exergy at exit of condenser

$$\emptyset_{cc_o} - \emptyset_0 = \dot{m}_{cc_i}(H_{cc_o} - H_0) - T_0(s_{cc_o} - s_0) \quad (22)$$

$$\text{Net retrieved exergy} = (\emptyset_{cc_o} - \emptyset_0) + (\emptyset_{cwc_o} - \emptyset_0) \quad (23)$$

Net exergy efficiency of distillation system is:

$$\eta_{II} = \frac{Net\ recovered\ exergy}{Exergy\ at\ inlet\ of\ evaporator} \quad (24)$$

Uncertainty Analysis

In experimentation, some errors may be involved due to parameter calculation through instrument and human observation. The error reduces accuracy of obtained results. Therefore, calculation of error in calculated parameters as uncertainty (W_R) is necessary. In this study, uncertainties have been calculated by following equation (25) which is given below [34]. The value of uncertainty in all calculated quantities is arranged in Table 3.

Table 3. Uncertainties of different parameters

Parameters	Uncertainty (%)
Boiler and condenser tube Curve and cross-sectional area	± 0.8330
Combustion gas flow rate	± 0.9530
Water vapor flow rate	± 0.1541
Coolant fluid flow rate	± 0.7592
Condensed water flow rate	± 0.1535
Net retrieved energy	± 1.1310
Net retrieved exergy	± 1.7695
Net energy efficiency	± 1.2968
Net exergy efficiency	± 2.6850

$$W_R = \left[\left(\frac{\delta R}{\delta X_1} w_1\right)^2 + \left(\frac{\delta R}{\delta X_2} w_2\right)^2 + \cdots + \left(\frac{\delta R}{\delta X_n} w_n\right)^2\right]^{1/2} \quad (25)$$

RESULTS AND DISCUSSION

Experimental Validation

Figure 4 (a-c) are showing the comparison of the temperature at the inlets and outlets of the evaporators (smooth tube and two axial finned tube) with the literature at 25, 50 and 75 % engine load. The comparison revealed that the experimental results are in agreement with published work [33]. The temperature and time curve characteristics remain the same when compared to the reported literature. However, the magnitude of the temperature was different as found in Figure 4. The difference is attributed to the power of engine used in this investigation. It is clear that the difference in temperature at inlets and outlets of the boilers increased with increase in the ICE load. This resulted in high recovery of waste thermal energy and enhancement in the production yield of potable water.

Net Recovered Energy and Exergy

The net retrieved energy and exergy was determined for smooth tube as well as axial finned (two, three and four axials finned) tube evaporators. Figure 5 – 6 presented the value of net retrieved energy and exergy of all four evaporators at different load conditions (25, 50 and 75 % engine load). At higher engine load, the consumption of fuel by ICE is higher to meet the power requirement. This led increment in temperature and flow rate of combustion gas at entrance of evaporators. Consequently, net thermal energy extraction rate improved by ~ 95.046 % at 75 % engine load compared to at 25% engine load in the case

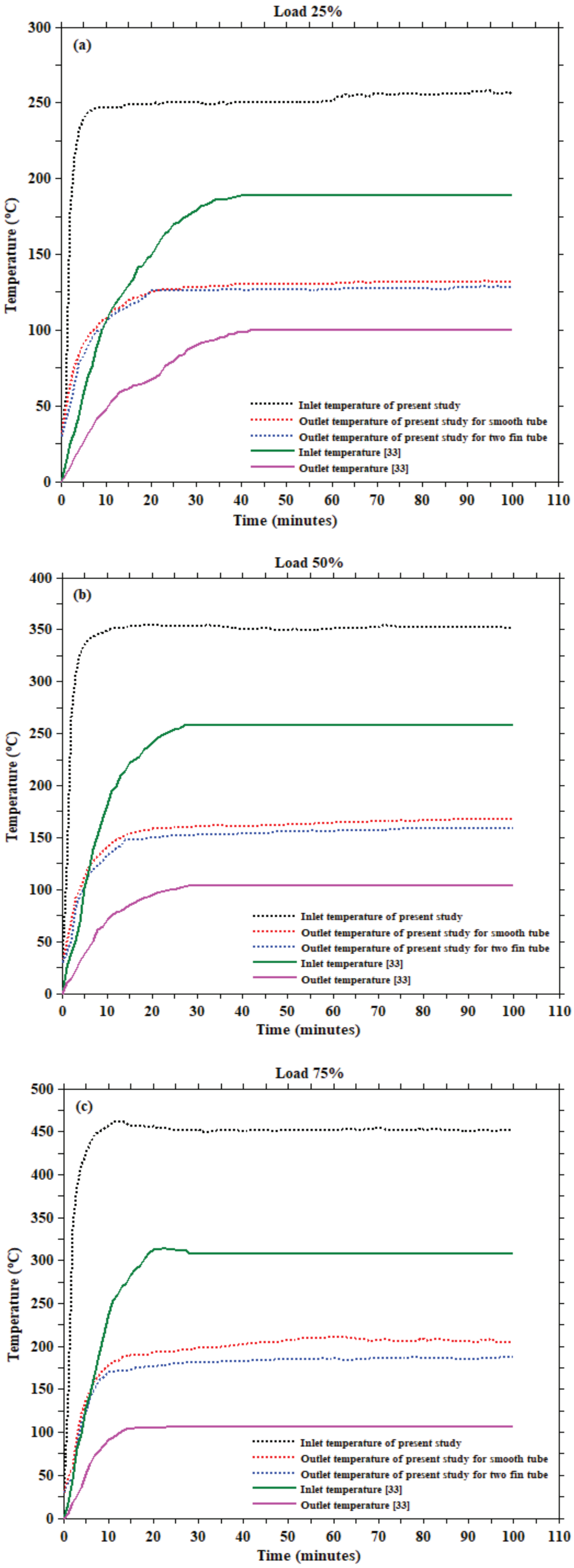

Figure 4. Temperature and time curve for different loads (a) 25% (b) 50% (c) 75%.

of smooth tube boiler. Moreover, heat transfer surface area increased by increasing in number of axial fins on boiler tube. Hence, the energy extraction rate improved in four axial finned tube evaporators. It was clear that the thermal energy extracted with four, three and two axial finned tube evaporators were more than that of smooth tube evaporator by about 26.823 - 45.513 %, 12.202 - 21.979 % and 5.084 - 13.930 %, respectively. Figure 6 demonstrated the variation

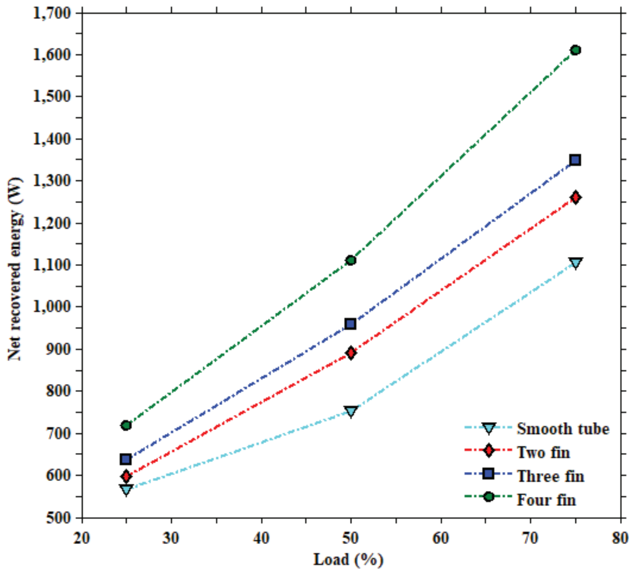

Figure 5. Variation of net extracted energy with load for smooth tube evaporator, two, three and four longitudinal finned tube evaporators.

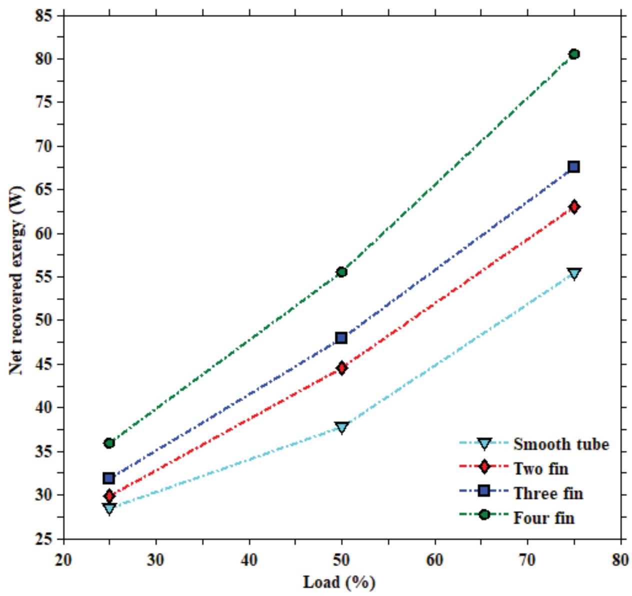

Figure 6. Variation of net extracted exergy with load for smooth tube evaporator, two, three and four longitudinal finned tube evaporators.

of net extracted exergy with respect to engine load for four different types of evaporators. As engine load increased, so, the thermal availability of exhaust gas also increased. The recovery of waste heat exergy increased with increase in engine load for all the considered cases. The net recovered exergy in the case of two and four finned tube evaporates at 25 % and 75 % load were 1.442 – 7.614 W and 7.713 – 25.203 W higher than that of plane tube, respectively. The energy transfer area per unit length increased by increasing the number of fins on evaporating surface. Therefore, it was also found that the retrieved exergy improved with increase in number of axial fins in boiler tube surface. Furthermore, the highest extracted exergy was approximately 80.582 W in the case of four finned tube boiler at 75% load.

Net Efficiency

Figure 7 presented the curves of net energy efficiency for various types of evaporators at 25, 50 and 75 % engine load. With increase in engine load, waste energy disposal also increased at the entrance of evaporators. The recovered waste heat energy act as driving force for the thermal distillation unit. The net energy efficiency for the smooth tube boiler was 21.075 % and 27.653 % at 25 % and 75 % engine load, respectively. It was found that the net energy efficiency improved when surface area of heat transfer increased. Maximum net energy efficiency (40.239 %) was obtained for four axial finned boiler at maximum considered engine load. Furthermore, net exergy efficiency of thermal distillation unit with different evaporators at 25, 50 and 75 % engine load is demonstrated in Figure 8. During the study, it was observed that an increase in temperature of the flue gas raised the exergy destruction. As results, net exergy efficiency reduced with increase in engine load. In addition, number of fins on evaporator tube surface also effects the efficiency of the thermal distillation unit. The, net exergy efficiency for smooth tube evaporator was found to be 5.411 % and 4.518% at 25 % and 75 % engine load, respectively. The highest efficiency (~ 6.863 %) was observed for the four finned tube boiler at 25 % load. This clearly indicated that axial finned tube evaporators are better than smooth tube evaporators.

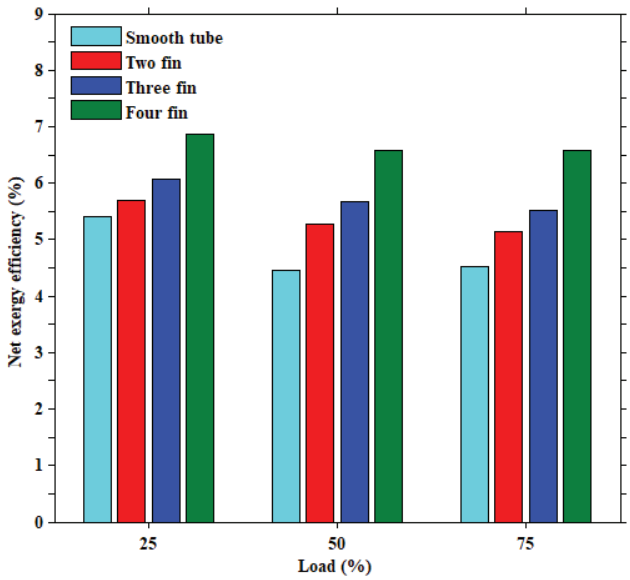

Figure 8. Net exergy efficiency and load bar chart for smooth tube evaporator, two, three and four longitudinal finned tube evaporators.

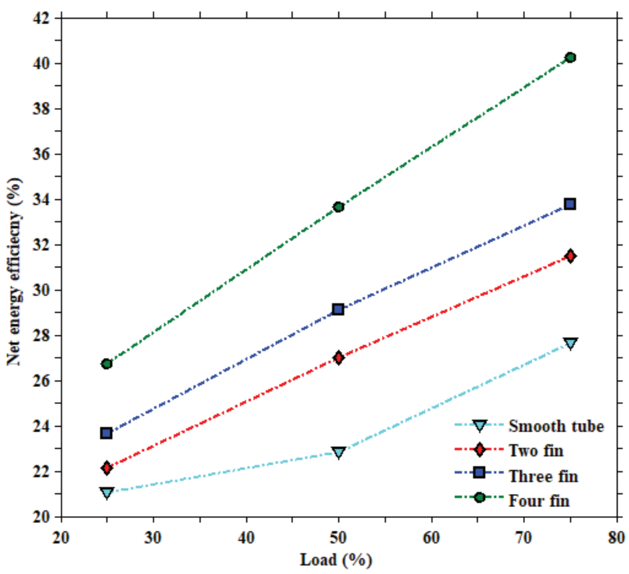

Figure 7. Net energy efficiency and load curve for smooth tube evaporator, two, three and four longitudinal finned tube evaporators.

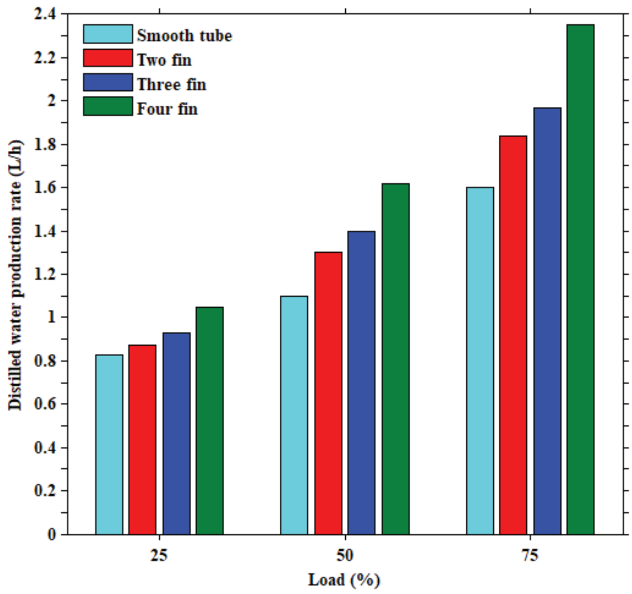

Figure 9. Distillated yield and load bar chart for smooth tube evaporator, two, three and four longitudinal finned tube evaporators.

Distillation Yield

Distillated yield vs. load bar graph for all considered evaporators is illustrated in Figure 9. The water evaporation rate improved in evaporator by increasing in engine load. This resulted in increment of water vapor flow rate though condenser tube. The water flow rate in condenser shell side was also increased to condense the increased amount of vapor as distillated yield. Distillated yield was found to be 0.83 liter/hour at 25% load and 1.6 liter/hour at 75% load in smooth tube evaporator. Moreover, heat transfer rate enhanced with increasing number of fins on evaporating surface. More water molecules received sufficient kinetic energy for conversion into vapor phase. As a result, the evaporation rate of water in evaporator improves by increasing number of axial fins on boiler tube surface. Therefore, distillated yield improves with longitudinal fin tube evaporator as compared to smooth tube evaporator. Distillated yield was maximum (2.35 liter/hour at 75% load) for four longitudinal fin tube evaporator.

CONCLUSION

A thermal distillation system was used to retrieve the waste heat energy from flue gas. The impact of boiler type and considered load on the efficiency of thermal distillation unit was examined. Using all considered types of evaporators, including one smooth, the remaining ones were axial finned tubes with two, three, and four axial fins. A shell and U-tube condenser were used in the study. Thermal distillation system was driven by the recover thermal energy of flue gas from 7 hp diesel engine generator at 25, 50 and 75 % engine load. Net retrieved thermal energy and exergy, Net energy and exergy efficiency, distillated yield was obtained. The following conclusions can be drawn from the present investigation.

1. The net extracted thermal energy was found to be 45.513 % higher for four finned tube boiler as compared to smooth tube boiler at 75 % engine load.
2. The four axial finned tube evaporator shown highest value of net extracted exergy, i.e., 80.582 W at maximum load.
3. The net energy efficiency was found to be 40.239 % at 75 % load, and net exergy efficiency was 6.863 % at 25 % load for four axial finned tube evaporator.
4. Finally, the distillation yield of ~ 2.35 liter/hour was achieved in case of four finned tube evaporator.

An axial finned tube evaporator for thermal distillation has opened up new avenues for the development of a more efficient thermal distillation system. Such distillation system can be used at construction sites, restaurants, hospitals, community halls etc. to supply potable water without consuming power from external sources.

NOMENCLATURE

C_p	Specific heat capacity, kJ/kg-K
H	Enthalpy, kJ/kg
ΔH	Change of energy, kJ/kg
L	Latent heat, kJ/kg
$\dot{m}$	Mass flow rate, kg/s
P	Pressure, bar
$\dot{q}$	Rate of thermal energy, kJ/s
R	Universal gas constant, kJ/kg-K
s	Entropy, kJ/kg-K
Δs	Entropy changes, kJ/kg-K
t	Temperature, K
Δt	Change in temperature, K

Greek Symbols

$\emptyset$	Exergy, kJ/kg
$\Delta\emptyset$	Change of exergy, kJ/s
η_I	Net energy efficiency
η_{II}	Net exergy efficiency

Subscripts

c	Condenser, Coolant fluid, Condensate
e	Evaporator (Boiler)
f	Flue gas
i	Inlet
o	Outlet, Surrounding condition
req	Required
s	Saturation
v	Vapor
w	water

ORIGINALITY OF THE SCIENTIFIC ARTICLE

This study is devoted to recovering waste energy from the combustion gas of a compression ignition engine at different loads using thermal distillation system. Smooth and axial finned tube evaporators have been used during experimentation. Finally, the performance of the thermal distillation unit has been examined.

AUTHORSHIP CONTRIBUTIONS STATEMENT

Satyendra Kumar: Conceptualization, Performing experimentation, Formal analysis, Writing- original draft. ***Prakash Chandra:*** Arranging resources, Supervising experimentation, Editing original draft.

SOURCES OF FUNDING

The fund has not been issued by any organization.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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