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"Power Generation From Heat Lost In I.C. Engine"

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Abstract: As we know that in 'Internal Combustion Engine' total heat supplied from fuel is not utilized to develop useful work. Out of total heat 20-30% heat is wasted in cooling medium such as water and 40-50% is wasted in exhaust gases. So, we have inherent scope to utilize this waste heat. In conventional system we are taking benefits of heat lost through exhaust gases with the help of turbocharger to supply compressed air to inlet manifold. So, there is no any system which uses heat wasted from cooling medium.

With this paper, we tried to use that wasted heat from cooling medium for developing power with the help of turbine. In this system we are going to replace cooling water by appropriate refrigerant. Therefore, we are passing liquid refrigerant through cooling jacket of engine where this liquid refrigerant will absorb the heat from engine cylinder and get converted into vapour thus engine will get cooling effect. Therefore, we get vapour refrigerant with high temperature and pressure which would be useful to run a steam turbine. Before the turbine we used nozzle to convert pressure energy into kinetic energy in order to obtain required power from turbine. In the turbine the expansion of vapour refrigerant takes place and work get obtained from generator. The expanded refrigerant then further converted into liquid by passing through radiator/capillary tube. In radiator / capillary tube at proper pressure the refrigerant of vapour phase gets converted into liquid phase. This liquid refrigerant is further recycled for engine cooling (closed loop system). Thus, we have developed an efficient system for utilizing waste heat from cooling medium.

***Index Terms* – Waste Heat Recovery from IC Engine, Mathematical Calculations, Thermal Efficiency of IC engine, Overall efficiency of IC engine.**

I. INTRODUCTION

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. It means approximately 60 to 70% energy losses as a waste heat through exhaust (30% through engine cooling system and remaining through exhaust gases).

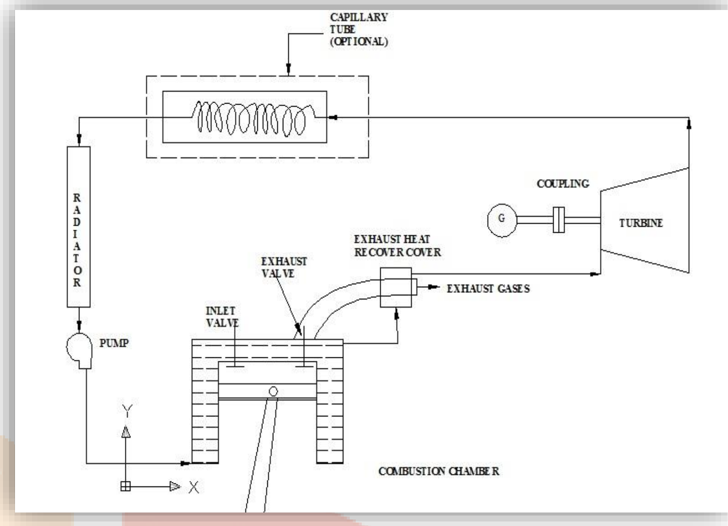


Fig. No.1: System Layout & Flow Direction of R113 Refrigerant.[1]

Main aim of this invention is to utilize the lost or waste heat during cooling engine jacket of an Internal Combustion engine for generation of power. Use this power to drive the auxiliary equipment's such as pump or generator which will improves efficiency and performance of whole system than conventional water-cooling system.

This invention relates to a combined engine cooling and refrigeration system utilizing a common working fluid such as a refrigerant. If an internal combustion engine confined fluid cooling system and a refrigeration system were combined for operation although having divergent purposes, particularly but not necessarily in the automotive field, in such a way as to use the same working fluid i.e., the refrigerant, simplicity and numerous other advantages are obtainable which have not heretofore been available.

An objective of the present invention is to provide a combination of an engine cooling system and an auxiliary refrigeration system in one operative system which is simple in construction, low in cost of construction and maintenance, and effective under widely divergent weather conditions. A feature of the invention is a closed engine cooling and auxiliary refrigerating system in which one fluid functions as the working fluid. Another feature is a closed engine cooling and auxiliary refrigerating system having a compressor drive turbine driven by waste heat of an Internal combustion engine, the compressor and turbine being operated with the engine coolant and the latter also serving as a refrigerant. Another feature is a closed system of engine cooling and auxiliary refrigeration in which a condenser serves the customary functions of combination of an engine radiator and a refrigeration system condenser.

II. LITERATURE REVIEW

Several research papers explore Waste Heat Recovery (WHR) systems in Internal Combustion Engines (ICEs), focusing on various technologies like thermoelectric generators (TEGs), Rankine cycles (RCs), and Organic Rankine Cycles (ORCs) to improve engine efficiency and reduce emissions. These WHR systems aim to capture and utilize the heat energy that is typically lost from the engine's exhaust, potentially increasing overall engine efficiency and reducing fuel consumption.

1. Experimental study on waste heat recovery system of an internal combustion engine using thermoelectric technology. IOPscience (2019)
2. Comprehensive Review Of ORC's Application: Waste Heat Recovery System In IC Engine. Keyur AJWALIA (2021). Stanford University, Stanford, California.
3. IC engine waste heat recovery systems. A paper in IRJET (2020), Jabir PP1, Jishnuprakash2, Sajid K3, Sakkeer Hussain Chalil4, Salmanul Faris PK5, Sanjay KN6, Sarathkumar7.

III. METHODOLOGY

3.1. Selection of Refrigerant: The important parameter of our system is to select required refrigerant which will fulfill the following properties

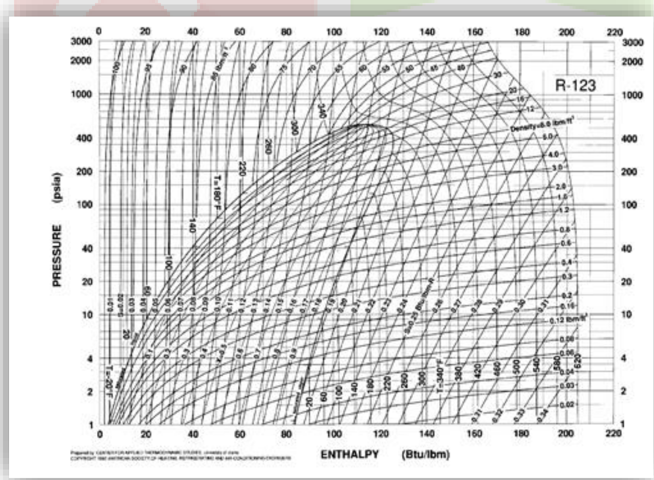
3.1.1. Properties Required for Selection of Refrigerant:

1). Boiling point: - It should be optimum to ensure that it must be brought to its initial liquid state. And it should have in a range of 40-50˚C (considering all weather conditions) it into liquid phase under radiator pressure. **2)**. **Pressure at working temperature: -** The minimum pressure required in order to run the steam turbine is in range of 3-10 bar. **3). Critical temperature: -** critical temperature must be high enough so that it will make easy to convert into liquid phase.

3.1.2. Tri-chloro-Tri-fluoro Ethane (R113):

Chemical formula- C2Cl3F3, Boiling point- 48˚C, Critical temperature- 213˚C

Please refer the refrigerant R113 properties charts for saturated liquid and saturated vapour.



Appearance	Liquid
Colour	Colourless
Odour	Slightly ethereal
Molecular Weight	152.9
Boiling Point	27.9°C
PH	Neutral
Freezing Point	107°C
Specific Gravity	1.48 (21°c)
Solubility	Water 3.9 g/l(25°C)
Vapour Density	5.3 (25°C)

Fig. No.02: Pressure – Enthalpy of R113

Table No.01: Properties of R113

3.1.3. Di-chloro-tri-fluoro-ethane (R123):

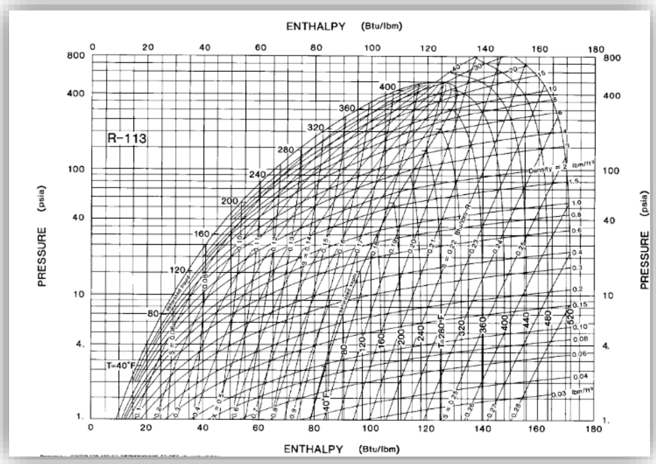


Fig. No.03: Pressure – Enthalpy Chart of R123

• Appearance:	Colourless liquid
• Physical state:	liquid
• Molecular weight:	187.35
• Chemical formula:	cc12fcc1f2
• Odor:	faint ethereal and sweetish odor
• Specific gravity (water = 1.0):	1.47 @ 70˚f (21.1˚c)
• Solubility in water (weight %):	0.31% @ 70˚f (21.1˚c)
• Ph:	neutral
• Boiling Point:	117.7˚f (47.6˚c)
• Melting Point:	-35˚c (-31˚f)
• Vapor pressure:	5.6 psi @70˚f (21.1˚c)
• Vapor density (air = 1.0):	6.5
• Evaporation rate:	>1 compared to: ether = 1
• % Volatiles:	100
• Flash point:	none

Table No.02: Properties of R123

Please refer the refrigerant R123 properties charts for saturated liquid and saturated vapour

3.1.4. Di-chloro-methane Or Methylene Chloride (R30):

Similarly studied the R30 refrigerant and out of 3, selected the R113 refrigerant.

IV. SELECTION OF COMPONENTS:

4.1 Nozzle: Nozzle is defined as the passage of varying cross section, through which heat energy of steam is converted to kinetic energy, its major function is to produce steam jet with high velocity to drive steam turbine. Nozzle is the device which is used for converting fluids pressure energy into kinetic energy. If a fluid is passed through a tapered pipe its velocity is higher at the outlet as compared to inlet

4.2 Velocity of Steam:

Steam enters the nozzle with high pressure and low initial velocity (it is so small as com-pared to the final velocity that it is generally neglected) and leaves it with high velocity and low pressure. This is due to the reason that heat energy of steam is converted into kinetic energy as it (steam) passes through the nozzle. The final or outlet velocity of steam can be found as follows: Let, C = Velocity of steam at the section considered (m/sec), h1 = Enthalpy of steam entering the nozzle, h2 =Enthalpy of steam at section considered, hd = Heat drop during expansion of steam is the nozzle = (h1 — h2), Considering 1 kg of steam and flow to be frictionless adiabatic, we have, Gain in kinetic energy = Adiabatic heat drop. $C^2 = 2\ hd$, $C = \sqrt{(2x\ 1000\ hd)}$, where is in kJ, $= 44.72\ \sqrt{hd}$

In practice, there is loss due to friction in the percent of total heat drop. Due to this, total heat drop is minimized. Let heat drop after deducting friction loss be K*hd. The velocity, $C = 44.72\ \sqrt{hd}$.

4.3 **Turbine:** Turbine is the unit where actual output of the system is obtained. The high pressure and temperature vapour made to expand into turbine in order to obtain work output. The low pressure and temperature vapour is the outlet from turbine. Following factors are to be considered. Power output required from turbine, type of refrigerant used, maximum and minimum temperature obtained, other operating parameters like cost, size and efficiency. The steam turbine is a prime-mover in which the potential energy of the steam is trans-formed into kinetic energy, and later in its turn is transformed into mechanical energy of rotation of the turbine shaft. The turbine shaft, directly or with the help of a reduction gearing, is connected with the driven mechanism. Depending on the type of the driven mechanism a steam turbine may be utilized in most diverse fields of industry, for power generation and for transport. Transformation of the potential energy of steam into the mechanical energy of rotation of the shaft is brought about by different means.

4.4 Capillary tube: A capillary tube is a long, narrow tube of constant diameter. The word capillary" is a misnomer since surface tension is not important in refrigeration application of capillary tubes. Typical tube diameters of refrigerant capillary tubes range from 0.5 mm to 3 mm and the length ranges from 1.0 m to 6 m. The pressure reduction in a capillary tube occurs due to the following two factors: 1. The refrigerant has to overcome the frictional resistance offered by tube walls. This leads to some pressure drop. 2. The liquid refrigerant flashes (evaporates) into mixture of liquid and vapour its pressure reduces. The density of vapour is less than that of the liquid. Hence, the average density of refrigerant decreases as it flows in the tube. The mass flow rate and tube diameter (hence area) being constant, the velocity of refrigerant increases since :***m* = ρVA,** For domestic refrigerator in general preferable dimensions and material, which are suitable for our system are= Diameter = 2 mm, Length = 3.5 m, Material = Cu.

Fig. No.04: Capillary Tube

4.5 Radiator: Radiator works as a heat exchanger (condenser unit). Vapour refrigerant has to be converted into its liquid phase in order to recycle of the same. Factors to be taken into account: Inlet and outlet temperatures, Type of refrigerant, Size allowed, Work to be done to achieve final phase. Its heat to the airstream which passes around the outside of the tubes. To help spread the heated water over the top of all the tubes, a baffle plate is often placed in the upper tank, directly under the inlet hose from the engine. Sooner or later, almost everyone has to deal with an overheating.

V.RESULT AND DISCUSSION

5.1 Assumptions for Calculations

All the assumptions taken according to trail on single cylinder diesel engine for Cast steel material,

m = mass flow rate = 6.41 kg/min, h = Convective heat transfer = 6500 watt / m^2 k

T1 = Initial temperature of liquid (refrigerant) = 35 ˚C, Initial pressure of refrigerant = 1 atm

Diameter of cylinder = 87.5 mm, Stroke = 110mm, Thickness of wall = 10mm

Area for convection heat transfer = $A = \pi DL$, $= \pi * 0.1075 * 0.11$ = 0.03715 m^2

5.2 Heat Carried by Refrigerant At Cooling Jacket(Q)

Q = h * A *ΔT = 6500 *$\frac{60}{1000}$* (0.03715) * [105 - 35]= 1014.195 kJ/ min,

Q = Sensible heat + Latent heat + Superheated heat

= m* Cp * (Tsat - Tatm) + LH + m* Cp*(Tsup - Tsat)

= $\frac{6.41}{60}$ * 0.92947 * (48 - 35) + [147.2* 6.41] + [x * (m * Cp *ΔT)

1014.176 = 77.45 + x * 943.552 ∴ x = 0.997

We are not getting required temperature so we passed refrigerant around exhaust manifold for getting **110 ˚C** superheated temperature,

∴ Q = 1021 + 5.13 * [110 – 48] = 1339.06 kJ/ min

So, we required to take amount of heat from exhaust manifold for getting supersaturated steam.

q = 1339.06 – 1014.176 = 324.884 kJ/ min

Area required to take above heat = h *A * Δ T, $\frac{324884}{60}$ = 6500 * A * [300 - 48], A = 0.003057 m²

Also, $A = \pi$dL ……. {Assume diameter of exhaust pipe(d)=0.025m}

L = 0.04208 m, L = 42.089 mm

For considering practical variable conditions taking **factor of safety** as **2.35** for calculating length,

∴ L = 100 mm

With taking **95˚C** Supersaturated temperature of refrigerant, we are getting,

L = 31.96 mm

For considering practical variable conditions taking **factor of safety** as **2.35** for calculating length,

∴ L = 75 mm

5.3 Trial No.1_For 95˚c Supersaturated Temperature:

Table No. 3: Properties of R113 for saturated temperature (95˚C)

Saturated Temperature	Saturated pressure	Enthalpy (h) KJ/Kg		Entropy (S) Kj/kg˚k	
(˚C)	(bar)	sat.liq.	Sat.vap.	sat.liq.	Sat.vap.
93.33	3.72	124.31	251.208	0.4175	0.7837
82.60	2.72	113.05	244.70	0.3866	0.757

h1 = 251.208 kj/ kg , Sf2 = 0.41759 KJ/Kg ˚k (liquid), S1 = 0.7837 KJ/Kg ˚k (vapour).

Sfg2 = Sg2 – Sf2, = 0.7572 – 0.3805, = 0.370706 kJ / kg k, hf2 = 113.053 hg2 = 244.70 KJ/Kg

Isentropically energy is constant S1 = S2, 0.7837 = 0.38656 + x2 (0.370706) x2 = 0.997 ~ 1

h2 = hf2 + x2 * hfg2= 113.052 + 1(244.70 – 113.053), = 244.7 kj/kg, Δ h = 6.508 kj/ kg = 6508 j/kg

C1² = 2 (h1 – h2), = 2*(6508), C1 = 114.0876 m/s

∴ **Outlet velocity from nozzle = C1 = 114.0876 m/s**

5.4. Trial No.02_For 110˚c Supersaturated Temperature:

Table No. 4: Properties Of R113 for saturated temperature (110 ˚C)

Saturated Temperature	Saturated pressure	Enthalpy (h) KJ/Kg		Entropy (S) KJ/Kg ˚k		Density Kg/m3
(˚C)	(bar)	sat liq.	satvap.	sat	liq. satvap.	
110	4.50	141.47	261.8	0.4630	0.777	1340.10
60	90.9466	90.946	230.6834	0.3226	0.7412	1476

h1 = 261.8 kj/ kg Sf2 = 0.322633 KJ/Kg ˚k (liquid), S1 = 0.777028 KJ/Kg ˚k (vapour).

Sfg2 = Sg2 – Sf2, = 0.74292 – 0.3226 = 0.419466 KJ/Kg ˚k, hf2 = 90.9466 , hg2 = 261.80 KJ/Kg

Isentropically energy is constant S1 = S2, 0.777028 = 0.322633 + x (0.419467) x2 = 0.998 ~ 1, h2 = hf2 + x2 * hfg2, = 90.9466 + 1(230.68 – 90.9400), = 230.6834 kj/ kg

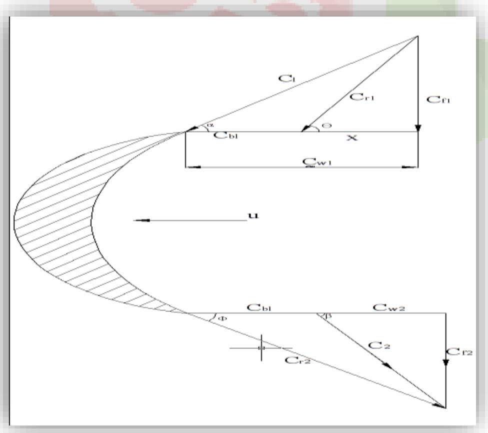
Δ h = (261.8-230.6834) kj/kg = 31.1166 kj/kg, $C1^2$ = 2*(h1 – h2) = 2*(6508)

C1 = 249.46 m/s

∴ **Outlet velocity from nozzle = C1 = 249.46 m/s**

5.5 TURBINE CALCULATION FOR 95˚C: Nozzle angle (α) =20, Inlet velocity of steam to turbine (C1) = 114.087m/s, Blade angle (θ, φ) =30˚,

Fig.No. 05 Velocity Triangle (Turbine Calculation refers to 95 ˚C)



INLET VELOCITY TRIANGLE

OUTLET VELOCITY TRIANGLE

From velocity triangle ,

(1) Cot(α) = Cw1/C1

Cw1 =114.087*cos(20)

= 107.206m/s

(2) sin(α) = Cf1/C1

Cf1 = 114.08*sin(20)

= 39.02m/s

(2) sin(θ) = Cf1/Cr1

Cr1 = 39.02/sin(30)

(4) tan(θ)=Cf1/x

x = 39.02/tan(30)

= 78.04m/s   x = 67.58m/s

(5) Cbl = Cw1-x

=107.206-67.58   = 39.62m/s

Assume all the blades of turbine are fine finished, so there is **no friction.**

So, Cr1=Cr2 =78.04 m/s and blade velocity is always same.Cbl1=Cbl2 =39.62 m/s

Cos(θ) = (Cbl + Cw) / Cr2,   (Cbl + Cw) = cos(30)*78.64,   = 67.58 m/s

Now , Cw2= 27.96 m/s.

The total power output of turbine = mf * (Cw1+Cw2)* Cbl

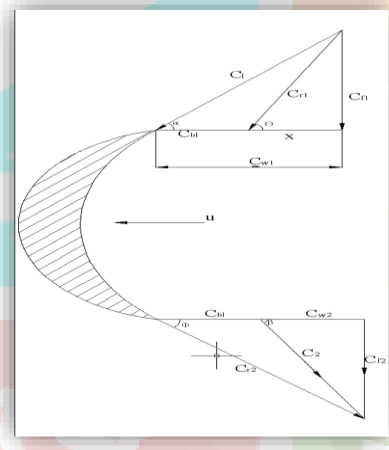
= (6.41/60) * (107.206 + 27.96) * 39.62, = 572.12 watt = (572.12 / 746)

= **0.8 HP**

5.6 TURBINE CALCULATION FOR 110˚C.

Nozzle angle(α) = 20, Inlet velocity of steam to turbine(c1) =249.46 m/s, Blade angle (θ, φ) =30˚,

Fig No.6.Velocity Triangle (Turbine Calculation refers to 110 ˚C)



INLET VELOCITY TRIANGLE

OUTLET VELOCITY TRIANGLE

From velocity triangle ,

(3) Cot (α) = Cw1/C1   (2) sin(α) = Cf1/C1

Cw1 =249.46 *cos(20)   Cf1 = 249.46 *sin(20)

= 234.42 m/s   = 85.32 m/s

(4) sin(θ) = Cf1/Cr1   (4) tan(θ)=Cf1/x

Cr1 = 85.32/sin(30)   x = 85.32/tan(30)

= 170.46 m/s   x = 147.78 m/s

(5) Cbl = Cw1-x, = 234.12 - 147.78, = 86.64 m/s

All the blades of turbine are rough finish , so there is 10% friction loss.

So, Cr1= 0.9 * Cr2 = 153.41 m/s and blade velocity is always same. Cbl1= Cbl2 = 86.64 m/s

Cos (θ) = (Cbl + Cw)/ Cr2, (Cbl + Cw) = cos(30)*153.41, = 46.21 m/s

Now , Cw2= 46.21 m/s.

The total power output of turbine = mf * (Cw1 + Cw2) * Cbl

= (6.41 / 60) * (234.42 + 46.21) * 86.64, = 2597 watt, = (2597 / 746), **= 3.48 HP**

VI. CONCLUSION AND FUTURE SCOPE

1. **With superheated steam (110˚C) of refrigerator R113 power generation observed: 3.48HP.**
2. **With superheated steam (95˚C) of refrigerator R113 power generation observed: 0.8HP.**

By using this experimental modification, it is convenient to use refrigerant as a cooling media in Internal Combustion engine in order to develop the power required to drive the auxiliary equipments such as pump used for lubricating oil pumping is can be directly derived from turbine unit. The power output from turbine is in the form of rotational energy of turbine of shaft, but further it can be stored in battery for more applications by using alternator.

Thus, it can be possible to reduce the load on Internal combustion engine by saving power required to drive auxiliary equipment's. Therefore, power output from turbine is increased which will ultimately results into increasing the thermal and mechanical efficiency of engine.

Future Scope for Improvements:

1. Need to do the real experimental trials with R113 refrigerant.
2. Need to do the simulation of data as applicable.
3. Need to focus on the either to use the capillary or radiator for application.

VII. AKNOWLEDGMENTS

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