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Thermoelectric Generator Plate-based EV Charging Solutions

Ram Kumar Tadgude[1,*], Harshajeet Suresh Patil[2], Harshada Anil Yewale[3], Dipali Bhausaheb Dighe[4], Mithil Vishnu Marewar[5]

Abstract

The International Institute of Refrigeration estimates that air conditioning and refrigeration use 15% of the world's electricity and contribute to the emission of CO2, HCFCs, and other harmful chemicals. Global warming is one of the many negative effects of using such refrigerants on our environment. The efficiency of the car is impacted by the increased fuel consumption for air conditioning. Thermoelectric air conditioning can be utilized to solve the emission issue and balance the supply and demand for energy consumption. The development and use of plate-based thermal electricity generator (TEG) systems to facilitate electric vehicle (EV) charging are examined in this article. TEG technology harnesses waste heat from various sources, converting it into electrical energy, which can significantly enhance the sustainability and efficiency of EV charging systems. By integrating plate-based TEGs into existing infrastructure, such as roads and parking areas, this approach capitalizes on ambient temperature differentials to generate power. The article examines the efficiency of different TEG materials, design considerations, and potential energy output under varying environmental conditions. It also talks about the monetary and environmental benefits of installing thermoelectric generators in cities, emphasizing how they can lessen dependency on conventional energy sources and encourage the usage of renewable energy in the EV ecosystem. According to the investigations, plate-based TEG technologies are a promising development for EV charging in the future, helping to create a more sustainable and environmentally friendly transportation ecosystem.

Keywords: TEG Plates, compressor, condenser, EV.

INTRODUCTION

The function of an automobile air conditioning system is to provide coolness. Its connected issue raises fuel usage and results in environmental issues. Additionally, the vehicle's efficiency is impacted.

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Today's Ev's biggest issue is charging, thus we're developing new car systems to solve it. The thermoelectric module uses an exhaust gas heating medium to produce power, which is then stored in the battery.

There is an urgent need for inventive, sustainable, and effective charging solutions due to the electric vehicle (EV) field's rapid growth.

Traditional charging methods predominantly rely on grid electricity, which can be sourced from fossil fuels, thereby undermining the environmental benefits that EVs aim to provide. In this context, thermoelectric generators (TEGs) offer a promising alternative by converting waste heat into electrical energy, effectively harnessing energy that would otherwise be lost.

The See beck effect, in which the variation in temperature across a thermoelectric material causes an electric potential, is used by thermoelectric equipment.

This capability enables TEGs to utilize heat from various sources, such as engine waste heat, solar energy, or even ambient temperature variations, making them versatile and adaptable for multiple applications. By integrating plate-based TEGs into EV charging infrastructure—such as roadways, parking lots, and other urban environments—this technology can generate supplementary power for charging stations, enhancing overall energy efficiency and reducing dependency on traditional power grids.

This introduction sets the stage for a comprehensive exploration of plate-based thermoelectric generator solutions, examining their operational principles, material advancements, design strategies, and potential applications in the EV charging landscape. Furthermore, the article will address the economic implications and environmental advantages of adopting TEG technology, ultimately positioning it as a crucial component of a sustainable transportation future. As the push for cleaner energy solutions intensifies, the integration of thermoelectric generators into EV charging systems represents a significant step towards optimizing energy use and minimizing carbon footprints in urban areas.

Objective

Examine the construction and design of a vapour compression refrigeration system to comprehend the purpose and use of the various refrigeration components.

INFORMATION AND WORKING

Teg plate

A thermoelectric generator (TEG), usually referred to as a Seebeck generator, is a solid state device that directly converts heat flow (temperature differential) into electrical energy by use of the Seebeck effect [1], a kind of thermal effect. Thermoelectric generators are smaller and have no moving parts, just as heat engines.

Power facilities might employ thermoelectric generators to turn waste heat into more electrical power, while cars could use automotive thermoelectric generators (ATGs) to boost fuel economy. To provide the necessary temperature differential to power space probes, radioisotope thermoelectric generators employ radioisotopes.

Condenser

The condenser is a crucial part of any refrigeration system, as was previously noted. The refrigerant is superheated when it enters a conventional refrigerant condenser. By rejecting heat to an external medium, it is first de-superheated and subsequently condensed. Depending on the condenser's intended functionality and the external medium's temperature, the refrigerant may exit a condenser as a filled or sub-cooled liquid (Figure 1).

Compressor

The most crucial and frequently most expensive part of any vapour compression refrigeration system (VCRS), accounting for between 30 and 40 percent of the total cost, is the compressor. To keep temperature in the evaporator and pressure low, a compressor in a VCRS continuously extracts the refrigerant vapor, allowing the refrigerant to boil and dissipate heat from the cooling regions. Next, by rejecting heat to the cooling medium in the capacitor, the compressor must increase the pressure of the refrigerant to a point where it can evaporate.

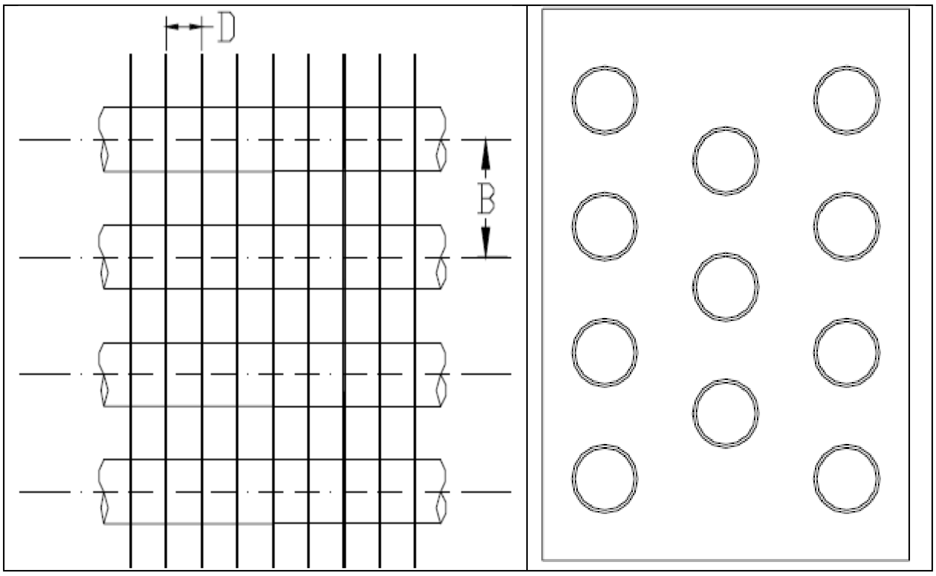

Figure 1. Shell and tube type condenser specification and staggered arrangement.

Evaporator

The primary part of a refrigeration system, the evaporator removes heat from water, air, or any other substance that has to be cooled by the refrigerant that evaporates. Absorbing heat from the item that is cooled by the refrigerant, which boils or evaporates in this component, is the main purpose of a system of refrigeration.

The evaporation process that takes place in the heat is referred to as an evaporator.

Refrigerant

They're R134a. Since no single component refrigerant matches, refrigerant blends were suggested as a substitute during the hunt for a new alternative. This is because combining two or more refrigerants might create a new working fluid with the needed properties. The issue with refrigerant blends is that, in certain situations, its properties may not be identical to those of the original refrigerant. In the appropriate temperature range, for instance, R12 will hardly ever meet the pressure.

INDENTATIONS AND EQUATIONS

Selection of Compressor

As per refrigerant and application the standard compressor is
Model: - KCE444HAG 1phase.
Refrigerating capacity (Q_0) =1077 watt
Power consumption=450 watt

Given Conditions for Actual Design: -
Ambient temperature=35°c
Evaporating temperature=7.2°c
Condensing temperature=54.4°c
Suction gas temperature=35°c
Suction pressure for R134a=40 psi (2.75 bar)
Discharge pressure for R134a=196 psi (13.51bar)
Pressure ratio=196/40=4.9

Design of Condenser

Heat rejection (Q_K)=Q_0+W=1527watt =1.1×1527=1679.7
=1700 watt

Properties of R134a at 54.4℃
(Assuming temperature =55℃)
Cp_g=1.266$KJ/Kg\,K$ Cp_f=1.602 $KJ/Kg\,K$ V_g=0.01353m^3/Kg
V_f=9.36×10^{-4} m^3/Kg μ_f=1.355× 10^{-4} Kg/ms μ_g=1.634× 10^{-5} Kg/ms
K_f=0.0676 W/mK K_g=0.01777 W/mK

Problem Statement

Design of air-cooled condenser for 1700-watt heat rejection with 55°C condensing temperature and 7.2°C evaporating temperature. The face velocity is 90m/min, inside and outside tube diameters are 9.12mm and 9.52mm and fin efficiency is 0.75.

Given: -B=30 mm, C=20 mm, D=2.11mm, E=0.20mm, ID=9.12mm, OD=9.52mm, FPI=12

i. Bare tube area,

$$A_{po} = \left(\frac{D-E}{BD}\right)\pi d_0$$
$$=\left(\frac{2.11-0.20}{30\times 2.11}\right)\pi\times 9.52$$
=0.9260 m²per m²face area per row

ii. Fin area,

$$A_f = \frac{2}{D}\left(c - \frac{\pi\, {d_0}^2}{4B}\right)$$
$$=\frac{2}{2{,}11}\left(20 - \frac{\pi\, 9.52^2}{4\times 30}\right)$$
=16.70 m² per m²face area per row

iii. Minimum flow area,

$$A_c = \frac{D-E}{D}\left(1 - \frac{d_o}{B}\right)$$
$$=\frac{2.11-0.20}{0{,}15}\left(1 - \frac{9.52}{30}\right)$$
= 0.6341 m² per m²face area per row

iv. Total heat transfer area,

$$A_0 = A_{po} + A_f$$
=0.9260+16.70
=17.626 m² per m²face area per row

v. Inside heat transfer area,

$$A_{pi} = \frac{\pi\, d_i}{B}$$
$$=\frac{\pi\times 9.12}{30}$$
= 0.9550 m² per m²face area per row

vi. Hydraulic diameter,

$$D_h = \frac{2C\,A_c}{1000 A_0}$$
$$=\frac{2}{1000}\left(\frac{20\times 0.6341}{17.626}\right)$$
$$=1.4390\times 10^{-3}\ m$$

vii. Area ratios

$$\frac{A_{p0}}{A_{pi}} = \frac{17.626}{0.9550} = 18.4556$$

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$\frac{A_{p0}}{A_f} = \frac{0.9260}{16.70}$=0.0554

vii. Condenser heat rejection

For mass flow rates,

Suction pressure =2.75 bar…… (Assuming 2.72 bar)

Specific volume at suction pressure

V_1 =0.07440 m^3/Kg

$\dot{m}$= (piston displacement /specific volume) =V_p/V_1

From ASHRAE Rated conditions: -

Displacement (cc/rev) =12.05

Number of revolutions per minute =2800 rpm

Piston displacement,

$V_p = 12.05 \times 2800$=0.03374m³/min

$\dot{m}=\frac{V_P}{V_1}$

$\dot{m}$= 0.03374/0.07440

= 0.4534 kg/min

=7.5582$\times 10^{-3}$kg/sec

For inside: -

Condensation heat transfer coefficient

From Above R134a property table

1. $Pr_f = \frac{Cp_f \mu_f}{k_f}$

$=\left(\frac{1.602\times10^3\times1.355\times10^{-4}}{0.0676}\right)$

=3.2110

2. $Re_g = \frac{4\dot{m}}{\pi\, d_i \mu_g}$

$= \frac{4}{\pi}\left(\frac{7.5582\times10^{-3}}{9.12\times1.634\times10^{-8}}\right)$

= 64577.5564

3. $Re_f = \frac{4\dot{m}}{\pi\, d_i \mu_f}$

$=\frac{4}{\pi}\left(\frac{7.5582\times10^{-3}}{9.12\times1.335\times10^{-7}}\right)$

=7904.0994

The condensation heat transfer coefficient inside the tube is found from various correlations to get an idea about the ranges by which it can vary

Zecchin's correlation:

Nu=$\frac{h_i d_i}{k_f}$ =0.05$\left(Re_{eq}\right)^{0.8} \times \left(Pr_f\right)^{1/3}$

Were, $Re_{eq} = (1-x)Re_f + xRe_f\sqrt{\frac{\rho_f}{\rho_g}}$

The refrigerant enters as vapour and leaves as saturated liquid. Hence the quality is x=1 at inlet and x=0 at outlet. If we assume that the quality varies linearly with the length of the condenser, then the average of (1-x) Ref and xRe_f is $0.5Re_f$. Therefore, the equivalent Reynolds number Re_{eq} becomes

$$Re_{eq} = 0.5\, Re_f\left[1+\sqrt{\frac{\rho_f}{\rho_s}}\right] = 0.5\, Re_f\left[1+\sqrt{\frac{V_g}{V_f}}\right]$$

$$Re_{eq} = 0.5 \times 7904.0994\left(1 + \sqrt{\frac{0.01353}{9.2336 \times 10^{-4}}}\right)$$

= 19080.1978

Nu =0.05((19080.1978) ^0.8)×(3.2210) ^ (1/3)

=196.0272

\ $h_i = Nu\frac{k_f}{d_i} = \frac{196.0272 \times 0.0676}{9.12 \times 10^{-3}}$

$h_i = 1453.0086 \mathrm{W/m^2k}$

Shah's correlation:

$h_i = h_f(0.55 + \frac{2.09}{{p_r}^{0.38}})$

$h_f = \frac{0.0676 \times 0.023 \times 7904.0994^{(0.8)} \times 3.2110^{\frac{1}{3}}}{9.12 \times 10^{-3}}$

h_f= 330.2491

For refrigerant R134a, p_{cr}= 40.560 bar

Condenser pressure at temperature$54.4^0 c$, p_c= 13.517 bar

$p_r = \frac{p_c}{p_{cr}} = \frac{13.517}{40.56}$ = 0.3332

h_i = 330.2491[0.55+$\frac{2.09}{0.3332^{(0.38)}}$]

h_i = 1229.6339 W/m^2kType equation here.

The value of h_i is also calculated by Dean Acker's and Crosser's correlation

But the feasible maximum value of the hi is from zecchin's correlation is i.e. 1453.0086W/m²k

**Air side heat transfer coefficient: -

Nu=0.1$(Re^{0.65}) \times \left(Pr^{1/3}\right)$

Re=$\frac{U_{max} D_h}{v}$

Nu=$\frac{h_0 D_h}{k}$

From above Property table for air

$$U_{max} = \frac{1.5}{A_c}$$

$= \frac{1.5}{0.6341}$

=2.3655 m/s

Re = $\frac{2.3655 \times 1.4390 \times 10^{-3}}{16.96 \times 10^{-6}}$

=200.7048

PR = $\frac{1005 \times 19.12 \times 10^{-6}}{0.02756}$

=0.6972

Nu=0.1×((200.7048) ^0.65)×((0.6972) ^ (1/3))

=2.7825

h_o =Nu$\frac{k}{D_h} = \frac{2.7825 \times 0.02756}{1.4390 \times 10^{-3}}$

=53.2909 Watt/m²k

Find Overall heat transfer coefficient

$$\frac{1}{U_0} = \frac{A_o}{A_{pi}} \times \frac{1}{h_i} + \frac{A_o}{A_{pi}} \times \frac{1}{h_s} + \frac{1 - \emptyset}{h_o\left(A_{po}/A_f + \emptyset\right)} + \frac{1}{h_o}$$

Fouling factor$\left(1/_{h_s}\right) = 0.00009$ m²k/w
Fin efficiency(∅)=0.75

$$\frac{1}{U_o} = \frac{18.4556}{1453.0086} + \frac{18.4556 \times 0.00009}{1} + \frac{1 - 0.75}{53.2909(0.0554 + 0.75)} + \frac{1}{53.2909}$$

=0.038

U_o= 26.3157w/m²k

LMTD= (Δt_2-Δt_1/ ln(Δt_2/Δt_1))

LMTD= $\frac{42-35}{\ln\left(\frac{54.4-35}{54.4-42}\right)}$

=15.6397 $°c$

$Q_k = U_o \times A_{ot} \times LMTD$

1680 =26.3157 × A_{ot} × 15.6397

A_{ot} =4.0819 m^2

Number of rows = $\frac{A_{ot}}{Actual\ area \times A_o}$

= $\frac{4.0819}{0.07 \times 17.626}$

=3.2

≅ 3

Evaporator Design

Heat absorption (Q_0)= 1077 W

= 1100 W

Properties for air side at mean temperature 30^0c: -

T_{mean} =30°C $\rho = 1.165$ kg/m^3

μ=18.63 × 10^{-6} Ns/m^2 v=16 × 10^{-6} m^2/s

C_P=1005 J/kgK k=0.02675 w/mK

Properties for refrigerant R134a at 7.2°C temperature:-

Cp_g =0.920 KJ/KgK Cp_f =1.360 KJ/KgK $V_g = 0.05284 m^3/Kg$

$V_f = 7.6923 \times 10^{-4} m^3/Kg$ μ_f= 2.454×10^{-4} Kg/ms μ_g= 1.107×10^{-5} Kg/ms

$K_f = 0.09155\ w/mK$ $K_g = 0.01242\ w/mK$

Problem statement

Design of air-cooled evaporator for 1100-watt absorption capacity with 55°c condensing temperature and 7.2°c evaporating temperature. The face velocity is 90m/min, inside and outside tube diameters are 9.12mm and 9.52mm and fin efficiency is 0.75 (Figure 2).

Given: -B=30 mm, C=20 mm, D=2.11mm, E=0.20mm, ID=9.12mm, OD=9.52mm, FPI=12

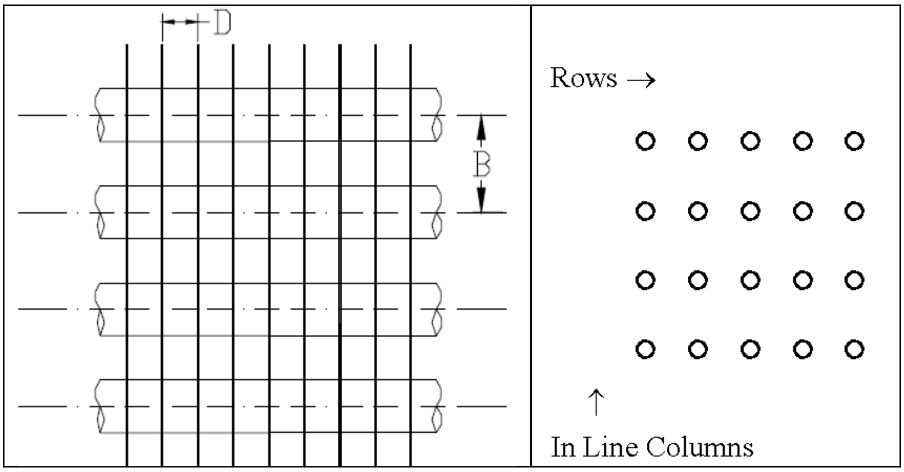

Figure 2. Shell and tube type evaporator inline arrangement Area calculation for evaporator is same as condenser

i. Bare tube area,

$$A_{po} = \left(\frac{D-E}{BD}\right)\pi d_0$$

$$=\left(\frac{2.11-0.20}{30\times2.11}\right)\pi\times 9.52$$

=0.9260 m²per m²face area per row

ii. Fin area,

$$A_f = \frac{2}{D}\left(c - \frac{\pi\, d_{0}{}^{2}}{4B}\right)$$

$$= \frac{2}{2{,}11}\left(20 - \frac{\pi\, 9.52^2}{4\times30}\right)$$

=16.70 m² per m²face area per row

iii. Minimum flow area,

$$A_c = \frac{D-E}{D}\left(1 - \frac{d_o}{B}\right)$$

$$=\frac{2.11-0.20}{0{,}15}\left(1 - \frac{9.52}{30}\right)$$

= 0.6341 m² per m²face area per row

iv. Total heat transfer area,

$$A_0 = A_{po} + A_f$$

=0.9260+16.70

=17.626 m² per m²face area per row

v. Inside heat transfer area,

$$A_{pi} = \frac{\pi\, d_i}{B}$$

$$= \frac{\pi\times9.12}{30}$$

=0.9550 m² per m²face area per row

vi. Hydraulic diameter,

$$D_h = \frac{2C\, A_c}{1000A_0}$$

$$=\frac{2}{1000}\left(\frac{20\times0.6341}{17.626}\right)$$

$$=1.4390\times 10^{-3}\ m$$

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vii. Area ratios

$$\frac{A_{p0}}{A_{pi}}=\frac{17.626}{0.9550}=18.4556$$

$$\frac{A_{p0}}{A_f}=\frac{0.9260}{16.70}=0.0554$$

For inside:
Condensation heat transfer coefficient
From Above R134a property table

1. $Pr_f=\frac{Cp_f\mu_f}{k_f}$

$=\left(\frac{1.360\times10^3\times2.454\times10^{-4}}{0.0915}\right)$

$=3.6474$

2. $Re_g=\frac{4\dot{m}}{\pi\, d_i\mu_g}$

$=\frac{4}{\pi}\left(\frac{7.5582\times10^{-3}}{9.12\times1.107\times10^{-8}}\right)$

$= 99124.5626$

3. $Re_f=\frac{4\dot{m}}{\pi\, d_i\mu_f}$

$=\frac{4}{\pi}\left(\frac{7.5582\times10^{-3}}{9.12\times2.454\times10^{-7}}\right)$

$=5116.1741$

The condensation heat transfer coefficient inside the tube is found from various correlations to get an idea about the ranges by which it can vary

Zecchin's correlation:

$\mathrm{Nu}=\frac{h_id_i}{k_f}=0.05\left(Re_{eq}\right)^{0.8}\times\left(Pr_f\right)^{1/3}$

Were, $Re_{eq}=(1-x)Re_f+xRe_f\sqrt{\frac{\rho_f}{\rho_g}}$

The refrigerant enters as vapour and leaves as saturated liquid. Hence the quality is x=1 at inlet and x=0 at outlet. If we assume that the quality varies linearly with the length of the condenser, then the average of (1-x) Ref and xRe_f is $0.5Re_f$. Therefore, the equivalent Reynolds number Re_{eq} becomes

$$Re_{eq}=0.5\,Re_f\left[1+\sqrt{\frac{\rho_f}{\rho_s}}\right]=0.5\,Re_f\left[1+\sqrt{\frac{V_g}{V_f}}\right]$$

$$Re_{eq}=0.5\times5116.1741\left(1+\sqrt{\frac{0.05284}{7.6923\times10^{-4}}}\right)$$

$= 23751.6592$

Nu =0.05((23751.6592) ^0.8)×(3.6474) ^ (1/3)

$=243.6985$

\ $h_i=Nu\frac{k_f}{d_i}=\frac{243.6985\times0.09155}{9.12\times10^{-3}}$

$h_i=2446.3374\ \mathrm{W/m^2k}$

Shah's correlation:

$h_i = h_f(0.55 + \frac{2.09}{{p_r}^{0.38}})$

$h_f = \frac{0.0915 \times 0.023 \times 5116.1741^{(0.8)} \times 3.6474^{\frac{1}{3}}}{9.12 \times 10^{-3}}$

h_f= 329.3351 W/m²k

For refrigerant R134a, p_{cr}= 40.560 bar

Evaporator pressure at temperature7.2^0c, p_e= 3.7 bar

$p_r = \frac{p_e}{p_{cr}} = \frac{3.7}{40.56} = 0.0912$

$h_i = 329.3351[0.55 + \frac{2.09}{0.0912^{(0.38)}}]$

h_i = 1891.0955 W/m^2k

But the feasible value of the hi is from shah's correlation is i.e. 1891.0955 W/m²k

Air side heat transfer coefficient

$Nu = 0.1(Re^{0.65}) \times (Pr^{1/3})$

$Re = \frac{U_{max} D_h}{v}$

$Nu = \frac{h_0 D_h}{k}$

$$U_{max} = \frac{1.5}{A_c}$$

$= \frac{1.5}{0.6341}$

=2.3655 m/s

$Re = \frac{2.3655 \times 1.4390 \times 10^{-3}}{16 \times 10^{-6}}$

=212.7471

$Pr = \frac{1005 \times 18.63 \times 10^{-6}}{0.02675}$

=0.6999

Nu=0.1×((212.7471) ^0.65)×((0.6999) ^ (1/3))

=2.8937

$h_o = Nu \frac{k}{D_h} = \frac{2.8937 \times 0.02675}{1.4390 \times 10^{-3}}$

=53.7918 W/m²k

Find Overall heat transfer coefficient

$$\frac{1}{U_0} = \frac{A_o}{A_{pi}} \times \frac{1}{h_i} + \frac{A_o}{A_{pi}} \times \frac{1}{h_s} + \frac{1 - \emptyset}{h_o(A_{po}/A_f + \emptyset)} + \frac{1}{h_o}$$

Fouling factor$\left({}^{1}/_{h_s}\right)$ = 0.00009 m²k/w

Fin efficiency($\emptyset$)=0.75

$$\frac{1}{U_o} = \frac{18.4556}{1891.0955} + \frac{18.4556 \times 0.00009}{1} + \frac{1 - 0.75}{53.7918(0.0554 + 0.75)} + \frac{1}{53.7918}$$

$=0.0383$

U_o=26.1096 W/m²k

$\text{LMTD}=\frac{35-25}{\ln\left(\frac{35-7.2}{25-7.2}\right)}$

$=22.4328\ °c$

$Q_k = U_o \times A_{ot} \times LMTD$

$1100 = 26.1096 \times A_{ot} \times 22.4328$

$A_{ot} = 1.890\ m^2$

$\text{Number of rows} = \frac{A_{ot}}{Actual\ area \times A_o}$

$= \frac{1.890}{0.07 \times 17.626}$

$=1.6$

$\cong 2$

FIGURES AND TABLES

TEG Plate Construction and Setup

The figure 3 illustrates the construction of the thermoelectric generator (TEG) plates used for EV charging. It highlights the arrangement of p-type and n-type semiconductor materials, showing how the plates are designed to harness heat differentials for energy production.

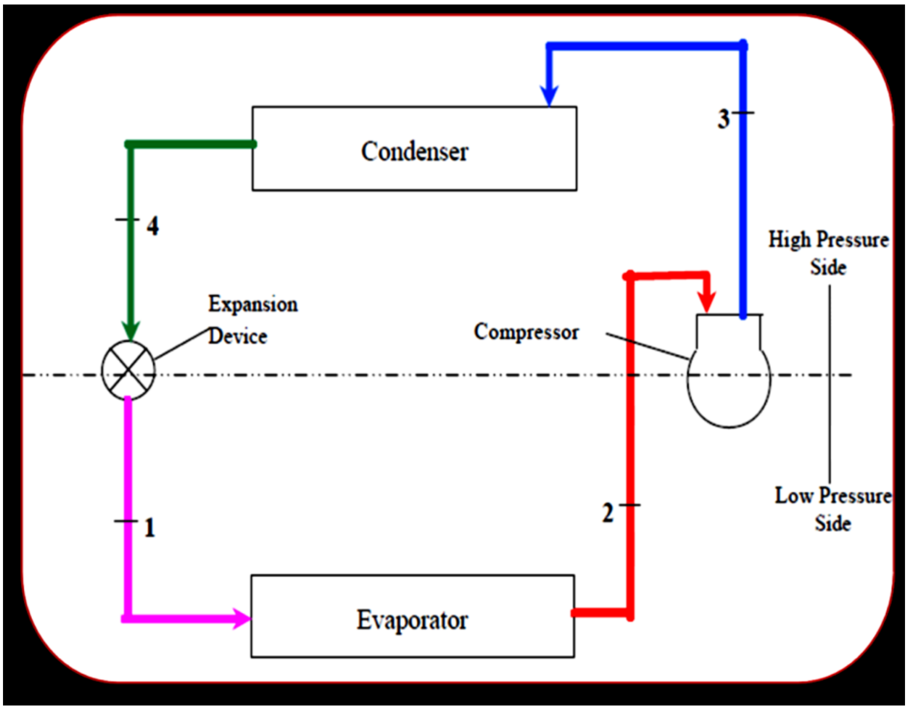

Figure 3. Construction.

Evaporator Inline Arrangement

Diagram of the evaporator's inline configuration, demonstrating the cooling process essential for TEG efficiency. It includes the bare tube and fin area calculations, flow rates, and hydraulic diameter to explain the role of the evaporator in the TEG setup (Figure 4).

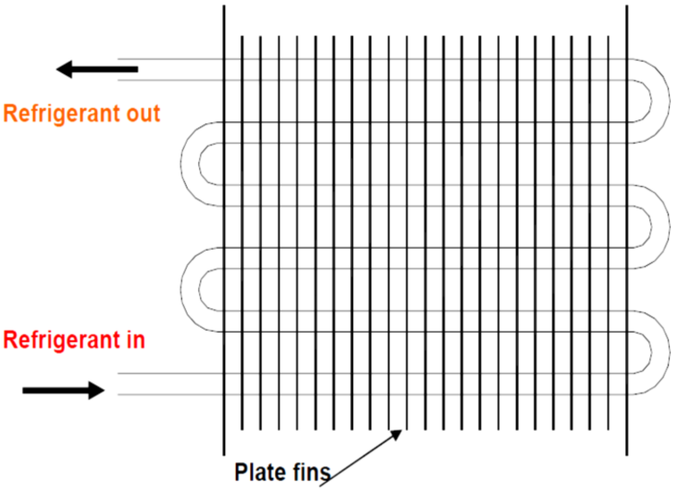

Figure 4. Evaporator.

Compressor Model and Specification

Illustration and specifications of the compressor model KCE444HAG used in the TEG-based charging system. The figure lists key parameters such as refrigerating capacity, power consumption, and operating conditions (Figure 5).

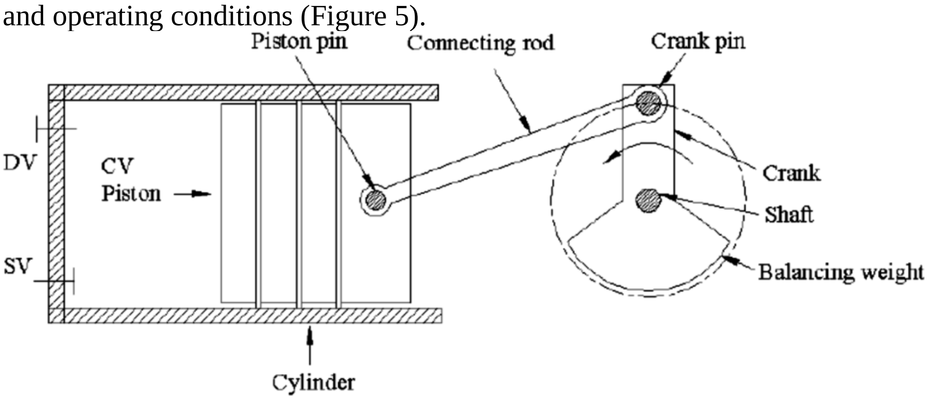

Figure 5. Compressor.

Condenser Design and Configuration

Detailed view of the shell-and-tube type condenser arrangement, showing the staggered tube arrangement, dimensions, and cooling flow paths. It specifies materials, dimensions, and operational parameters that maximize heat rejection from the TEG system (Figure 6).

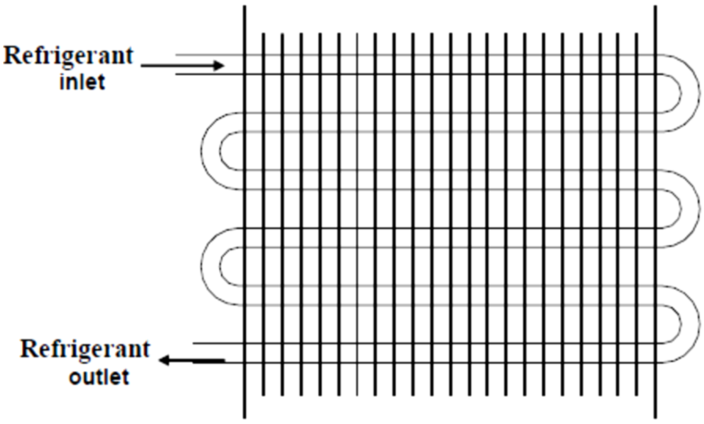

Figure 6. Condesor.

TEG Plates Applied on Heating Surfaces

Description: Layout showcasing the application of TEG plates on different heated surfaces within the EV charging setup. This figure visually demonstrates how waste heat is converted into electric energy through the TEG plates (Figure 7).

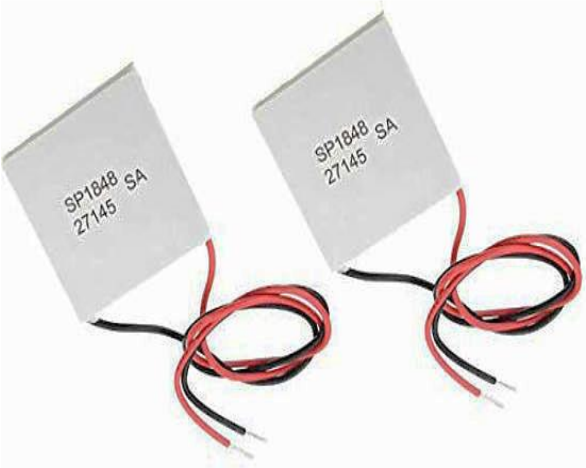

Figure 7. Teg Plates.

Table 1 shows the details of the refrigerant.

Table 1. Refrigerant Details.

S.L.	Properties	R-134a
1	Boiling Point	-14.9°F or -26.1°C
2	Auto-Ignition Temperature	1418°F or 770°C
3	Ozone Depletion Level	0
4	Solubility In Water	0.11% by weight at 77°F or 25°C
5	Critical Temperature	252°F or 122°C
6	Cylinder Color Code	Light Blue
7	Global Warming Potential (GWP)	1200

CONCLUSION

1. Our experimental setup leads us to the conclusion that, by putting TEG Plates in a certain number to a heated surface, we may produce electricity anywhere there is a heating surface.
2. We need extra hot surfaces and TEG plates in accordance with our needs since we want more electricity from this configuration.

Acknowledgements

Although thermoelectric property was discovered about two centuries ago thermoelectric device save only been commercialized during current years. The applications of thermoelectric vary from small refrigerator. In this System we implement TEG plates on Refrigeration cycles Evaporator and sucks its heat and generate electricity and that electricity we can use for battery charging of car and any other purpose.

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