Thermodynamic Water Heating System

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Abstract

A thermodynamic water heating system is an innovative technology that provides an efficient and cost-effective alternative to traditional water heating systems. This system utilizes the principles of thermodynamics to heat water using a closed-loop system consisting of a compressor, heat exchanger, and refrigerant fluid. The refrigerant fluid absorbs heat from the surrounding environment, which is then transferred to the water via the heat exchanger. This process enables the system to provide hot water even in low ambient temperatures. One of the advantages of a thermodynamic water heating system is its ability to work in any weather condition, making it a reliable source of hot water.

Overall, a thermodynamic water heating system is an innovative and sustainable solution for providing hot water in homes and buildings. Its unique closed-loop system and ability to operate in any weather condition make it an attractive option for those looking to reduce their energy consumption and costs.

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1 Introduction

This report aims to provide an overview of thermodynamic solar systems, their functionality, and the potential benefits they offer. The report will examine the technology behind these systems, how they work, and the different types of applications for which they can be used. Additionally, the report will consider the economic and environmental advantages of utilizing thermodynamic solar systems in buildings, including reduced energy costs and carbon emissions.

The use of renewable energy sources has become increasingly important in recent years, as concerns about climate change and rising energy costs continue to grow. Thermodynamic solar systems offer a promising solution for sustainable energy generation and can significantly reduce a building's reliance on traditional energy sources. With this report, we hope to provide readers with a comprehensive understanding of thermodynamic solar systems and encourage wider adoption of this innovative technology.

2 Literature Survey

We read a literature on energy balance calculation for a heat exchanger, From the article we got idea about the concepts of heat exchangers and basic energy balance calculations. Energy balance calculation involves accounting for the energy inputs and outputs of the heat exchanger, and determining the heat transfer rate.

The article explains the two basic types of energy balance calculations: the overall energy balance and the individual energy balance. The overall energy balance considers the entire heat exchanger as a single entity, while the individual energy balance considers each section of the heat exchanger separately.

The document given in the reference table contains a technical data sheet for R407C, a refrigerant used as a replacement for R22 in refrigeration and air conditioning systems. It provides detailed information on the physical and thermodynamic properties of the refrigerant, its chemical composition, and compatibility with different materials and lubricants. The document also includes information on the safety and handling of R407C. It concludes by summarizing the key properties and advantages of R407C as a replacement for R22 in refrigeration and air conditioning systems.

The document is a brochure for the Arctic Open Frame compressor series, which are compressors designed for industrial and commercial refrigeration applications. The brochure introduces the series and its features, provides an overview of the different models and their specifications, and highlights the advanced technology used in the compressors. The brochure also includes information on installation and maintenance, and concludes by emphasizing the benefits of choosing the Arctic Open Frame compressor series for its energy efficiency, reliability, and advanced technology. Now, we are describing about the technology.

2.1 Solar Water Heater

Traditional collectors that heat a refrigerant or water directly, are used in solar hot water systems. The water or refrigerant can be moved through the system passively using gravity or actively through a controller pump. These collectors need full sunshine.

2.2 Thermodynamic solar panels

Unlike conventional solar panels, thermodynamic solar panels operate on a different principle. Thermodynamic solar panels employ a thermodynamic cycle to produce electricity, contrary to conventional solar panels, which use the photovoltaic effect to convert sunlight directly into electricity.

A heat pump, which draws heat from the air, water, or ground, initiates the process. After that, this heat is transferred to a working fluid, like a refrigerant. A pressure difference caused by the working fluid expanding as it warms up drives a turbine. Electricity is produced by the turbine and can either be used right away or stored in batteries for later use.

As long as there is a temperature difference between the heat source and the environment, the heat pump will continue to operate and generate energy. The heat pump enables thermodynamic solar panels to produce energy even under low-light scenarios, in contrast to conventional solar panels that need direct sunlight to function.

Thermodynamic solar panels do not require placement in direct sunlight, unlike photovoltaics or conventional thermal solar panels. They may draw heat from the surrounding air as well as absorb heat from direct sunlight. Consequently, despite the fact that thermodynamic solar panels are technically solar panels, they resemble air source heat pumps more in some ways. The advantage here is that if you live in a cold region, they will likely perform optimally in full sunlight because the ambient air temperature may not be warm enough to satisfy your home's heating demands. Thermodynamic solar panels can be installed on roofs or walls, in full sun, or in complete shade.

Thermodynamic solar panels, in contrast to solar hot water systems, are still an emerging technology and have not undergone as much testing. The effectiveness of thermodynamic solar panels was tested in 2014 in Blyth, United Kingdom, by a single autonomous lab, Narec Distributed Energy. The experiments were conducted from January to July in Blyth, which has a reasonably temperate climate with a lot of rainfall.

The results indicate when the heat lost from the heat-exchanging tank is taken into account the thermodynamic SAHP (Solar Assisted Heat Pump) system's coefficient of performance, or COP, is 2.2. When heat pumps attain COPs above 3.0, they are often regarded as being highly efficient. This study showed that thermodynamic solar panels weren't very effective in 2014 in a moderate climate, but they are more effective in warmer regions.

2.3 Advantages of Thermodynamic Solar Panels

One of the main advantages of thermodynamic solar panels is their ability to generate electricity in low-light conditions. This makes them ideal for use in areas with long, dark winters or in locations where sunlight is scarce.

Another advantage is their ability to generate electricity even when the temperature is low. Traditional solar panels become less efficient as the temperature drops, but thermodynamic solar panels continue to produce electricity as long as there is a temperature difference between the source of heat and the environment. This makes them ideal for use in cold climates or at high altitudes.

Thermodynamic solar panels also have a higher energy density than traditional solar panels. This means that they can produce more electricity per unit of surface area, making them ideal for use in space-constrained applications.

2.4 Disadvantages of Thermodynamic Solar Panels

While thermodynamic solar panels have many advantages, they also have some disadvantages. One of the main disadvantages is their cost. Thermodynamic solar panels are more complex and expensive to install than traditional solar panels. They require a heat pump, which can be expensive to operate, and the system must be carefully designed to ensure maximum efficiency.

Another disadvantage is that they require a constant energy source to drive the heat pump. This means that they are not entirely self-sufficient and cannot generate electricity when the energy source is not available.

Finally, thermodynamic solar panels are less efficient than traditional solar panels under optimal conditions. While they are more efficient in low-light and low-temperature conditions, they are less efficient than traditional solar panels when the sun is shining brightly.

2.5 How Much Do Thermodynamic Panels Cost?

The thermodynamic cost of the panel, its components, and installation for a four-person family is a reasonable $6300 to $7500. The method guarantees a consistent return rate over time with minimal to no maintenance costs.

Only the compressor of the thermodynamic water heating system requires electricity, therefore operating costs are likewise minimal. The compressor consumes 360 watts per hour under normal circumstances, which is comparable to the use of a bigger refrigerator. The compressor's energy consumption, however, varies continually with the weather, time of day, month, and water use, yet it always operates efficiently.

2.6 Cost-Effective Water Heating Using Thermodynamic Panels

Typically, the compressor usage only costs $10–12 per month, or $120–150 per year, with a typical domestic hot water demand. Even intensive use, such as running the thermodynamic heating system constantly, would only cost $17 per month or $211 per year. The operating costs might theoretically be eliminated if a solar panel was employed in addition to the thermodynamic panels to produce the necessary electricity.

The weekly electricity usage can be reduced below $5 a month or $60 a year without much trouble if the compressor is configured to only run during specific hours, such as when the ambient air is the hottest (11 am to 5 pm).

A straightforward way to lower heating and gas bills is to install a thermodynamic water heating system. Only a heat collector panel and a compressor require to be installed and linked to your pre-existing hot water tank, and the system becomes fully operational.

The thermodynamic panel's technique is based on basic heat exchange. Similar to air-to-water heat pumps, household hot water is heated in a tank by transferring heat from the surrounding air through a specific working fluid and a compressor. As a result, your kitchen and bathroom sinks, tubs and pools, and other appliances now have a very affordable source of hot water.

2.7 Savings with Thermodynamic Panels Compared with Other Heating Systems

The unit price, installation expenses, and running costs are calculated, and the savings from installing thermodynamic panels are shown in the accompanying table 2.

2.8 Maintenance Expenditure

Thermodynamic water heating systems require minimum maintenance due to the straightforward, environmentally friendly product design. There is no requirement for refilling because the refrigerant fluid cycles in a closed cycle and does not deteriorate over time.

One or two check-ups every ten years would be sufficient to keep the system in good working order over the long term. If the compressor breaks down, it can be repaired or replaced for a price starting at $200.

The system's water tank should be maintained similarly to other water tanks: Depending on how hard the water is, it needs to be flushed anywhere from once a year to every few months. To avoid the accumulation of limescale, this needs to be done.

Depending on the weather, the thermodynamic panels themselves might need to be cleaned occasionally. They don't need any special cleaning products, and washing merely serves to keep objects away from the panel so that it can perform as effectively as possible by receiving the full ambient temperature. If frequent precipitation is received the panels are cleansed, this step is not necessary.

As a result, thermodynamic water heating is a great way to cut your expenses and carbon footprint without having to spend a lot of time or money maintaining the system.

3 Problem Statement

The thermal solar collectors are highly effective at generating heat energy from the sun's radiation on hot and sunny days. However, these collectors become completely inefficient when there is no sun, such as on cloudy or rainy days, during the nighttime, or in regions with low sunlight intensity. This limits their usability as a reliable source of heat energy throughout the year and in areas with varying weather patterns. As a result, there is a need to find alternative or supplementary sources of heat energy that can be used in conjunction with thermal solar collectors to ensure a consistent and reliable supply of heat energy.

4 Working Principle

A thermodynamic solar panel system involves capturing heat energy from the surrounding air and converting it into usable thermal energy. The system works through a closed loop circuit that contains a fluid, usually a refrigerant or heat transfer fluid, that is circulated through a thermodynamic solar panel, a compressor, a heat exchanger, and an expansion valve.

The thermodynamic solar panel consists of a heat collector plate that absorbs heat from the air, even in the absence of direct sunlight, through natural convection. The fluid in the closed loop circuit is then circulated through the heat collector plate, where it absorbs the heat energy and evaporates.

The evaporated fluid then travels to the compressor, where it is compressed, increasing its pressure and temperature. The hot, pressurized fluid is then passed through a heat exchanger, where the heat is transferred to water or another medium that can be used for space heating, hot water supply, or industrial processes.

After releasing the heat energy, the fluid is passed through an expansion valve, where its pressure and temperature drop, causing it to revert to its liquid state. The liquid fluid then travels back up to the thermodynamic solar panel, where the cycle begins again.

The system can capture heat energy from the air regardless of weather factors, making it highly efficient and reliable. The thermodynamic principles behind this system allow it to operate efficiently even in low-temperature environments, making it suitable for a wide range of applications in various industries, including hotels, hostels, sports complexes, hospitals, and more.

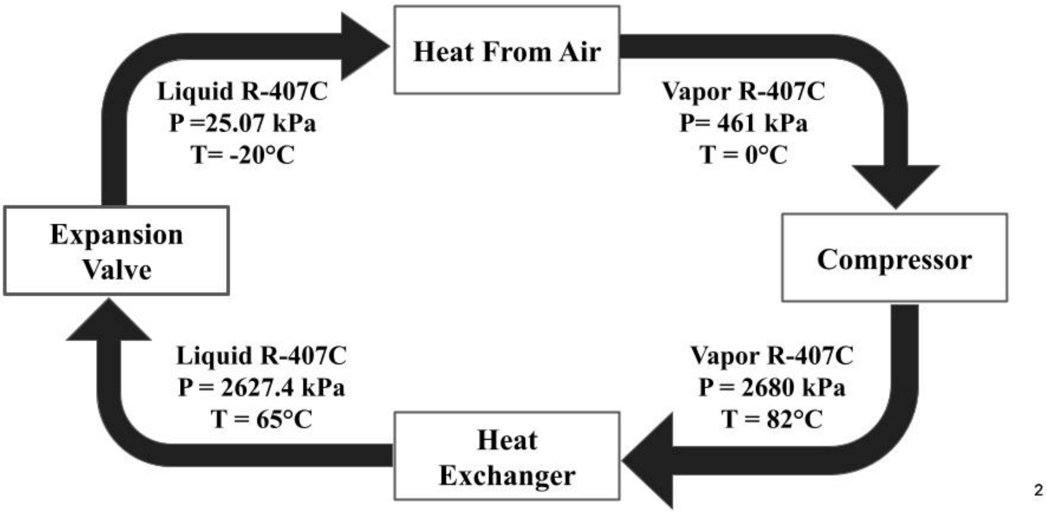

Figure 1: Flow Diagram

(Derived using Figure 18)

5 Coolant R-407C

It is a zeotropic mixture of 1,1,1,2-tetrafluoroethane (CF_3CH_2F), pentafluoroethane (CF_3CHF_2), and difluoromethane (CH_2F_2).

- R407C has a better Coefficient of Performance in tightly sealed systems than R134a because it operates at a higher pressure.
- Direct expansion air conditioning systems and residential & light air conditioning are both suited for R407C. In refrigeration systems for medium temperatures, it is also utilised.
- Refrigerant R407C has a good energy efficiency ratio as well as a low compression pressure and Global Warming Potential(GWP).

Table 1: Properties of R-407C

Molecular Weight (kg/kmol)	86.2
Boiling point $°C$ at 1 atm pressure	-43.8
Saturated liquid density (25°C), kg/m^3	1138
Saturated vapour density (25°C), kg/m3	43.8
Critical temperature (°C)	86.4
Critical pressure, bar	46.3
Liquid heat capacity @ 25°C, (kJ/(kg·K))	1.533
Vapour heat capacity @ 1.013 bar (kJ/(kg·K))	1.107

6 Results and Discussion

6.1 Heat Exchanger

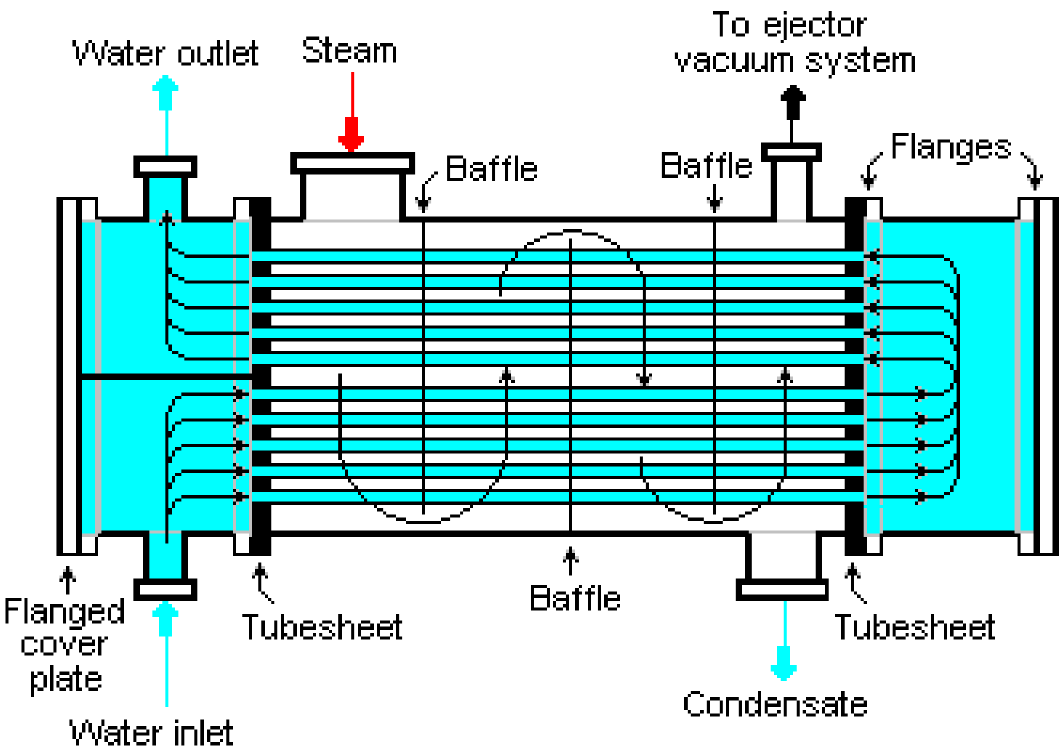

Figure 2: Condenser
https://www.researchgate.net/figure/Desalination-process-and-heat-exchanger-condenser$_fig3_2$21926191

A heat exchanger is a device designed to transfer heat from one fluid to another while keeping them separated by a wall. In chemical processes, it's essential to calculate the heat balance by adding or removing heat via exchangers with hot and cold process streams.

The energy balance equation is a useful tool to determine the heat transfer rate ($\dot{Q}$) if you know the mass flow rates, specific heat capacities, and inlet and outlet temperatures of the two fluid streams. Similarly, it can help calculate the outlet temperatures of the fluid streams if you have the heat transfer rate and inlet temperatures.

The energy balance equation can be used to determine the heat transfer rate ($\dot{Q}$) if the mass flow rates, specific heat capacities, and inlet and outlet temperatures of the two fluid streams are known.It can also be used to determine the outlet temperatures of the fluid streams if the heat transfer rate and the inlet temperatures are known.

6.1.1 Design Calculation

We have assumed $60,000\frac{Kg}{h}$ of water is flowing in tube side of a horizontal condenser.The coolant vapor enter the condenser saturated at $82°C$ and condensation will be completed at $65°C$.Water is available from $0°C$ to $56°C$.The vapor are to be totally condensed.Using **Triangular Pitch** arrangement, starting design assuming overall heat coefficient $= 575\frac{W}{m^{2}°C}$ and we are optimizing the computation until U(Overall heat transfer coefficient) will be within ±30% range.

Now, from the given upper information we are calculating heat duty($\dot{Q}$)

$$\dot{Q} = \dot{m}C_p(\Delta)T \tag{1}$$

Where,

$\dot{m}$ is mass flow rate of water

C_p is molar heat capacity of water is $4.18\frac{kJ}{kg}^{\circ}$

ΔT is $56°C$.

So, we get $\dot{Q} = 3,901.33$ kW

After that, we have calculated ΔT_{LMTD} and corrected ΔT_{LMTD} by using F_t correction factor:

$$\Delta T_{LMTD} = \frac{\Delta T_1 - \Delta T_2}{ln\frac{\Delta T_1}{\Delta T_2}} \tag{2}$$

$$\Delta T_{LMTD} = \frac{(82-56)-(65-0)}{ln\frac{82-56}{65-0}} \tag{3}$$

Now, using 3 we got $\Delta T_{LMTD} = 42.56°C$

We have to calculate R and S for the given temperature difference to find out F_t correction factor from the graph.

$$R = \frac{T_1 - T_2}{t_2 - t_1} = 0.30 \tag{4}$$

$$S = \frac{t_2 - t_1}{t_2 - t_1} = 0.68 \tag{5}$$

From graph, we got $F_t = 0.91$
So, corrected ΔT_{LMTD} is $F_t * \Delta T_{LMTD}$ i.e. $0.91 * 42.56 = 38.72°C$
We have assumed that overall heat coefficient $U_{assumed} = 575\frac{W}{m^2 °C}$
Now, we calculated overall area:

$$OverallArea = \frac{\dot{Q}}{U_{assumed} * \Delta T_{LMTD}} = 175.18m^2 \tag{6}$$

Surface area of a tube $= \pi d_o L_{effective}$
Where, tube dimension:
$d_o = 20mm = 0.78inch$
$d_i = 16.8mm = 0.66inch$
$L = 4.88m = 6feet$
So, surface area of one tube is $0.30m^2$
Now, the total number of tubes:

$$\frac{OverallArea}{SurfaceAreaofonetube} \approx 580 \tag{7}$$

Tube bundle diameter Calculation:

$$Tubebundlediameter = d_o(\frac{TotalNumberoftubes}{k_1})^{\frac{1}{n_1}} = 671.01 \tag{8}$$

Clearance = 17 mm (Fixed and U tube)
Shell diameter = Tube Bundle Diameter + Clearance Area = 688.01 mm
Now, calculating the value of $\frac{L}{D_s}$ which is equal to $\frac{4.88}{0.688} \approx 7.09$.
So, it's in between 5 and 10 and according to rule of thumb we can proceed further.

Tube Side Coefficient Calculation

$T_{water_{mean}} = 28°C$
Tube cross sectional area:

$$\frac{\pi}{4} * 16.8^2 = 221.67mm^2 \tag{9}$$

Number of tubes per pass:

$$\frac{580}{2} = 290 \tag{10}$$

So, total tube cross section area:

$$290 * 221.77 = 64284.52mm^2 = 0.064284m^2 \tag{11}$$

Tube velocity:

$$\frac{Massflowrate}{Density * Crosssectionareaofpass} = 0.26\frac{m}{s} \quad (12)$$

But velocity should be between 1.0 to 2.5. So we have to increase tube passes from 2 to 4 keeping shell pass unchanged and in this case F_t will not change. Bundle diameter will change. We have to take K_1 and n_1 values as 0.175 and 2.285 respectively.

$$BundleDiameter = d_o(\frac{N_1}{K_1})^{\frac{1}{n_1}} = 694.51mm = 0.6945m \quad (13)$$

Clearance = 17mm and Shell Diameter = 711.51 mm. So, L/D_S = 6.86 which is lie between 5 to 10. So, it has no issues.

Number of tubes per pass:

$$\frac{580}{4} = 145 \quad (14)$$

Total tube cross section area = 0.032142 m^2

Tube velocity = 0.522 m/s using all above details and it is not coming in the range.

Now increasing to tube passes from 4 to 6, keeping shell pass unchanged.

Bundle Diameter:

$$20(\frac{582}{0.0743})^{\frac{1}{2.499}} = 723.14mm \quad (15)$$

For this bundle diameter we $k_1 = 0.0743$ and $n_1 = 2.499$

Clearance = 19 mm

Shell diameter = 723.14 mm

$$\frac{L}{D_S} = \frac{4.88}{0.742147} = 6.57 \quad (16)$$

it is in the range of 5 to 10.

Number of tubes fer pass:

$$\frac{582}{6} = 97 \quad (17)$$

Cross sectional area of all tubes:

$$91 * 221.67mm^2 = 21501.99mm^2 = 0.02150199m^2 \quad (18)$$

Tube velocity:

$$\frac{60,000}{360 * 993 * 0.021501}\frac{m}{s} = 0.7806\frac{m}{s} \quad (19)$$

Now, increase the tubes 6 to 8 keeping shell unchanged.

Bundle Diameter:

$$20(\frac{584}{0.0365})^{\frac{1}{2.675}} = 745.87mm \quad (20)$$

Clearance = 17 mm

Shell Diameter = 764.87 mm

$$\frac{L}{D_s} = \frac{4.88}{0.764} = 6.30 \tag{21}$$

It's in the range of 5 to 10.
Number of tubes per pass:

$$\frac{584}{8} = 73 \tag{22}$$

So, Total cross-sectional area of all tubes:

$$73 * 221.67mm^2 = 16181.91mm^2 = 0.16181m^2 \tag{23}$$

Tube Velocity:

$$\frac{6000}{3600 * 993 * 0.16181} = 1.03 \tag{24}$$

We are proceeding with the value of tube velocity from 24 and heat exchanger with 1-8 shell side and tube side passes respectively.

Now, calculating inner heat transfer coefficient:

$$h_i = \frac{4200(1.35 + 0.02t)(u_t)^{0.5}}{(d_i)^{0.2}} \tag{25}$$

$$= \frac{4200(1.35 + 0.02 * 28)(1.03)^{0.5}}{(16.8)^{0.2}} \tag{26}$$

$$= 4630.622\frac{W}{m^2 °C} \tag{27}$$

Shell Side Coefficient Calculation

Number of tubes in entire row:

$$N_r = \frac{D_b}{P_t'} \tag{28}$$

$$P_t' = 0.87 * P_t \tag{29}$$

$$N_r = \frac{745.87}{0.87 * 1.25 * 20} = 34.29 \approx 34 \tag{30}$$

$$N_r' = \frac{2}{3}N_r = 22.66 \tag{31}$$

Heat Capacity($\dot{Q}$):

$$\dot{Q} = W_C * (Enthalpy of coolent vapour - Enthalpy of coolent liquid) \tag{32}$$

$$W_C = \frac{\dot{Q}}{Enthalpyofcoolentvapour - Enthalpyofcoolentliquid} \tag{33}$$

$$W_C = \frac{3907.866}{(423 - 314.6)} = 36.05\frac{kg}{s} \tag{34}$$

Tube Loading:

$$T_H = \frac{W_c}{L_{eff} * N_t} \tag{35}$$

$$= \frac{36.05}{(4.88 - 0.05) * 584} = 0.012\frac{kg}{s.m} \tag{36}$$

Now, calculating outer heat transfer coefficient:

$$h_o = 0.95 * K_L * [\frac{g * \rho_L * (\rho_L - \rho_V)}{\mu_L * T_H}]^{\frac{1}{3}} * N_r{}'^{-1/6} \tag{37}$$

Where, $\rho_L = 833.8\frac{kg}{m^3}$
$\rho_v = 194.2\frac{kg}{m^3}$
$\mu = 95.96 * 10^{-6} Pa.s$

$$h_o = 0.95 * 0.13 * [\frac{9.81 * 833.8 * (833.8 - 194.2)}{95.96 * 10^{-6} * 0.01278}]^{\frac{1}{3}} * 22.666^{-1/6} \tag{38}$$

$$h_o = 1194.68\frac{W}{m^{2\circ}C} \tag{39}$$

Overall Heat Transfer Coefficient

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{1}{coefficienth_o} + \frac{d_o * \ln\frac{d_o}{d_i}}{2K_w} + \frac{d_o}{d_i} * \frac{1}{coefficienth_i} + \frac{d_o}{d_i}\frac{1}{h_i} \tag{40}$$

$$\frac{1}{U_o} = \frac{1}{1194.68} + \frac{1}{5000} + \frac{20}{16.8}[\frac{1}{4630.6222} + \frac{1}{4500}] \tag{41}$$

$$\frac{1}{U_o} = 0.00155868 \tag{42}$$

$$U_o = 641.56\frac{W}{m^{2\circ}C} \tag{43}$$

$$Error = \frac{641.5685 - 575}{575} = 11.57\% \tag{44}$$

Error is 11.57% which is acceptable.

Tube Side Pressure Drop

$$\Delta P_t = N_p[8j_f \frac{L}{d_i}(\frac{\mu}{\mu_w})^{-0.14} + 2.5]\frac{\rho\mu_t^2}{2} \tag{45}$$

$$R_e = \frac{\rho\mu_t d_i}{\mu} \tag{46}$$

$$R_e = \frac{999 * 1.03 * 16.8 * 10^{-3}}{0.6 * 10^{-3}} = 28638.12 \tag{47}$$

$j_f = 3.8 * 10^3$, putting vlues of R_e and j_f in equation 45

$$\Delta P_t = 4[8 * 3.8^{-3}\frac{4.88}{0.016} + 2.5]\frac{993 * 1.03^2}{2} \tag{48}$$

$$\Delta P_t = 23872.71\frac{N}{m^2} = 23.872kPa \tag{49}$$

Shell Side Pressure Drop

Baffle spacing = Shell diameter

$$G_s = \frac{129781.5}{3600} * \frac{1}{A_s} = 308.83\frac{kg}{s.m^2} \tag{50}$$

Where,

$$A_s = \frac{P_t - D_o}{P_t} * D_s * BaffleSpacing \tag{51}$$

$$A_s = \frac{1.25d_o - d_o}{1.25d_o} * D_s^2 \tag{52}$$

Where, D_s = 0.764 m. So, Value of A_s is 0.1167392 m^2.

$$d_e = \frac{1.1}{d_o}[P_t^2 - 0.917d_o^2] \tag{53}$$

Where, $P_t = 25$ mm and d_o = 20 mm, So value of d_e = 0.014201 m.

$$R_e = \frac{G_s d_e}{\mu} \tag{54}$$

By using, value of G_s and d_e we got $R_e = 548220.72$, $\rho_t = 2.7 * 10^{-2}$ (25% baffle cut)

$$\mu_s = \frac{G_s}{\rho_v} = 1.59\frac{m}{s} \tag{55}$$

$$\Delta P_s = 8 * j_t * \frac{D_s}{d_e}\frac{L_{eff}}{L_B} * \frac{\rho * \mu^2}{2}(\frac{\mu}{\mu_w})^{0.14} \tag{56}$$

$$\Delta P_s = 8(2.7 * 10^{-2}) * \frac{0.764}{0.014201}\frac{4.88 - 0.05}{0.764} * \frac{194.2 * 1.59^2}{2} = 18034.139Pa \tag{57}$$

Taking 50% of ΔP_s for vapour velocity:

$$\Delta P_s = \frac{18034.139}{2} = 9017.06Pa \tag{58}$$

6.2 Compressor

An air compressor is a type of mechanical device that increases gas pressure by reducing its volume. Among gas compressors, air compressors are a commonly used variety.

While pumps and compressors share the common feature of increasing fluid pressure and moving it through a pipe, the key distinction is that compressors alter the density or volume of gases, which is generally not feasible with liquids due to their limited compressibility. For this reason, compressors are frequently utilized for gases.

A rotary screw compressor is a positive displacement compressor that uses two helical screws to compress refrigerant gas. The compressor is specifically designed to handle R407C refrigerant, which is commonly used in air conditioning and refrigeration systems. It provides efficient and reliable compression of the refrigerant and can operate at high pressures and temperatures. The rotary screw compressor offers continuous compression without pulsation or surging, resulting in stable operation and minimal wear and tear.

Size of Compressor that can be used for required condition is **Model SLT-10000C** which can work for required flow rate of 36kg/s .Dimension of the rotary compressor that can be used will be **12m x 5m x 5.3m** .

6.2.1 Compressor Energy Calculation

Using Energy Equation, Input - Output + Consumption = Accumulation

$$\{\dot{m} * C_p(T) * T\} - \{\dot{m} * C_p(T) * (T + dT)\} + d\dot{Q} = 0 \tag{59}$$

$$\dot{m} * C_p(T) * dT = d\dot{Q} \tag{60}$$

Here,

$\dot{m}$ = mass flow rate

$C_p(T)$ = specific heat at constant pressure w.r.t temperature

dT = Change in Temperature

$\dot{Q}$ = Heat transfer per unit time

Integrating both sides,we get

$$\dot{m} * C_p(T) * \Delta T = \dot{Q} \tag{61}$$

$$\dot{m} * C_p(T) * (T_f - T_i) = \dot{Q} \tag{62}$$

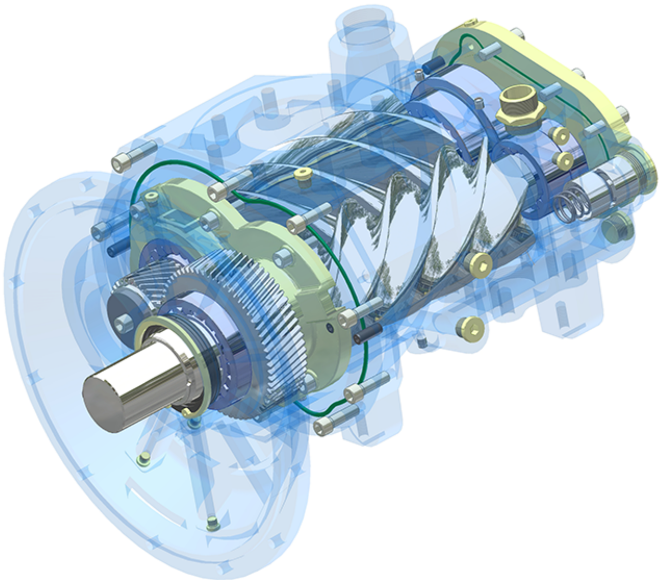

Figure 3: Schematic diagram of a scroll structure. (a) A scroll set. (b) the geometry of a scroll chamber
https://www.compair.com/en-in/technologies/screw-compressor

We Know ,

$\dot{m} = 36$ kg/sec

$C_p(T) = 1.64$ KJ/kgK

Initial Temperature(T_i) $= 0°C$

Oulet Temperature(T_f) $= 82°C$

So,After applying the Values,

we get

$36 * 1.64 * (82 - 0) = \dot{Q}$

$$\dot{Q} = 4857.423 KJ/sec \tag{63}$$

6.3 Expansion Valve

Expansion valves are mechanical devices employed in refrigeration systems to regulate the flow of refrigerant. Their primary function is to facilitate the conversion of liquid refrigerant at high pressure in the condensing unit to low-pressure gas refrigerant in the evaporator.

In refrigeration systems, the term "low side" refers to the section that operates at low pressure, while the term "high side" is used to describe the section that operates at high pressure.

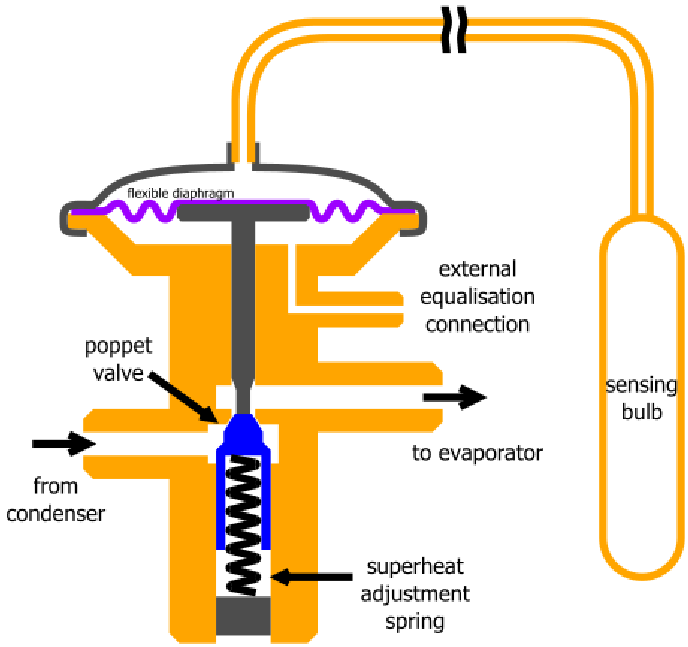

Figure 4: Thermostatic Expansion valve
https://en.wikipedia.org/wiki/Thermal$_{e}xpansion_{v}alve$

6.3.1 Expansion Valve Energy Calculation

Using Energy Equation, $Input - Output + Consumption = Accumulation$

$$\{\dot{m} * C_p(T) * T\} - \{\dot{m} * C_p(T) * (T + dT)\} + d\dot{Q} = 0 \tag{64}$$

$$\dot{m} * C_p(T) * dT = d\dot{Q} \tag{65}$$

Here,

$\dot{m}$ = mass flow rate

$C_p(T)$ = specific heat at constant pressure w.r.t temperature

dT = Change in Temperature

$\dot{Q}$ = Heat transfer per unit time

Integrating both sides,we get

$$\dot{m} * C_p(T) * \Delta T = \dot{Q} \tag{66}$$

$$\dot{m} * C_p(T) * (T_f - T_i) = \dot{Q} \tag{67}$$

We Know ,

$\dot{m}$ = 36 kg/sec

$C_p(T)$ = 1.53 KJ/kgK

Initial Temperature(T_i)=65°C

Outlet Temperature(T_f)$= -20°C$
So,After applying the values,
we get
$36 * 1.53 * ((-20) - 65) = \dot{Q}$

$$\dot{Q} = -4193.66KJ/sec \tag{68}$$

6.4 Thermodynamics Solar Panel

6.4.1 Design Calculation

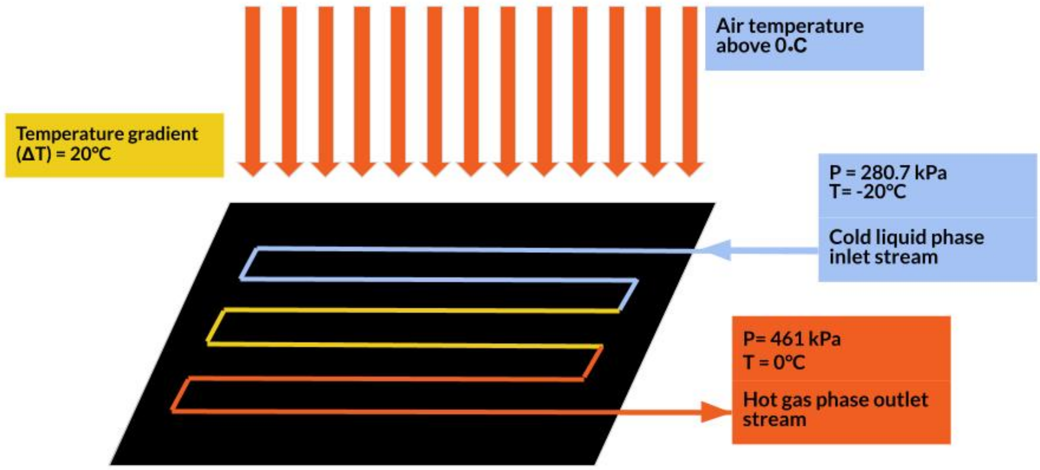

Figure 5: Thermodynamic Solar Panel Flow Diagram

Here the inlet coolant is distributed into 4 pipes

Assuming the panel surface temperature equal to T_1 as of the outer surface temperature of the pipe and the temperature at any point along the pipe is same over a length "d" perpendicular to the flow (mass flow coolant).

Now considering an element length "dx" along the pipe and heat transfer over the area corresponding length "dx" is given by,

$$h_A * dx * d * (T_A - T_1) = \frac{2\pi k * dx * (T_1 - T_2)}{\ln \frac{r_o}{r_i}} \tag{69}$$

Here, h_A = Air convective heat transfer coefficient

T_A = Air temperature

k = Aluminium thermal conductivity

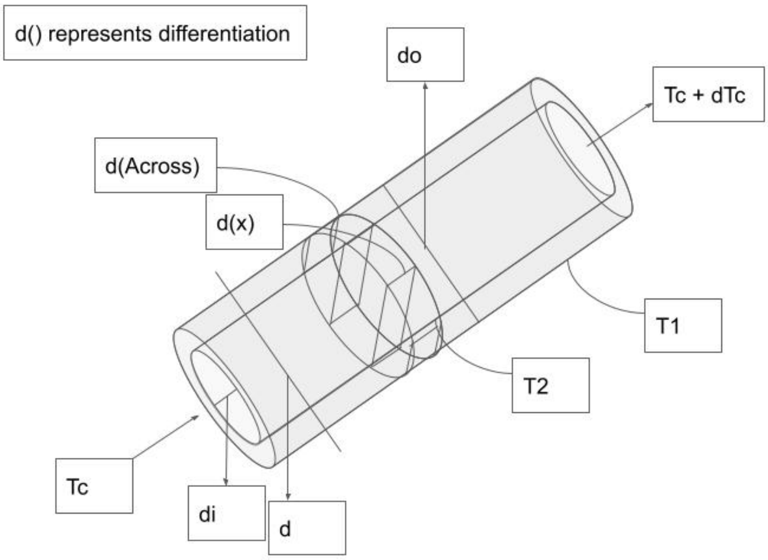

Figure 6: Schematic of a Single Pipe

T_2 = Inner surface temperature of pipe
r_o = Outer pipe radius
r_i = Inner pipe radius
Considering worst case scenario $T_A = 0°C$, then we have

$$h_A * d * (-T_1) = \frac{2\pi k(T_1 - T_2)}{\ln \frac{r_0}{r_i}} \tag{70}$$

Heat transfer equation for inner pipe surface for coolant is given by,

$$\frac{2\pi k(T_1 - T_2)dx}{\ln \frac{r_o}{r_i}} = h_c * (2\pi r_i) * dx * (T_2 - T_c) \tag{71}$$

$$\frac{2\pi k(T_1 - T_2)}{\ln \frac{r_o}{r_i}} = h_c * (2\pi r_i) * (T_2 - T_c) \tag{72}$$

Here, h_c = Coolant convective heat transfer coefficient
T_c = Coolant temperature
From equation 70 T_1 & T_2 are related by

$$T_1 = \frac{2\pi k T_2}{2\pi k + h_A d \ln \frac{r_o}{r_i}} = C_1 T_2 \tag{73}$$

Where

$$C_1 = \frac{2\pi k}{2\pi k + h_A d \ln \frac{r_o}{r_i}} \tag{74}$$

Now, h_c is dependent on temperature of coolant and is given by,

$$h_c = h'_{fc} + h'_{nb} \tag{75}$$

$$h'_{fc} = h_{fc} * f_c \tag{76}$$

$$\frac{h_{fc} * d_i}{k_f} = J_h * Re * Pr^{0.33} * (\frac{\mu}{\mu_w})^{0.14} \tag{77}$$

Here, k_f = fluid (coolant) thermal conductivity
f_c = Correction Factor

$$h'_{nb} = h_{nb} * f_s \tag{78}$$

f_s = Factor

$$h_{nb} = 0.00122 * \frac{k_L^{0.79} * Cp_L^{0.45} * \rho_L^{0.49}}{\sigma^{0.5} \mu_L^{0.29} \lambda^{0.24} \rho_v^{0.24}} (T_2 - Tc)^{0.24} (P_2 - P_c)^{0.75} \tag{79}$$

Where
k_L = Conductivity of coolant $W/m°C$
Cp_L = Liquid heat capacity $J/kg°C$
ρ_L = Liquid density kg/m^3
μ_L = Liquid viscosity $N.s/m^2$
λ = Latent heat J/kg
ρ_v = Vapour density kg/m^3
P_2 = Saturated pressure corresponding to the T_c N/m^2
σ = Surface tension N/m

From equation 72 we get

$$\frac{2\pi k(C_1 T_2 - T_2)}{\ln \frac{r_o}{r_i}} = h_c(2\pi r_i)(T_2 - T_c) \tag{80}$$

$$T_2 = \frac{h_c * r_i * \ln \frac{r_o}{r_i}}{h_c * r_i * \ln \frac{r_o}{r_i} + k(1 - C_1)} T_c \tag{81}$$

Now, since $h_c = h_c(T_2 - Ts)$ (Function of temperature difference), therefore we use a good assumed temperature for $T_2 = 0°C$ & calculate h_c & then calculate T_{2new} using equation 81 and intering the above calcuations for $T_2 = T_{2new}$ until error between T_{2new} & T_2 is less then 5%

We obtain temperature profile of T_2 (wall temperature) for a given temperature profile of T_c.

Now,

$$\dot{m}_c * C_p * dT_c + \lambda(d\dot{m}_c) = \frac{2\pi * k(T_1 - T_2) * dx}{\ln \frac{r_o}{r_i}} = \frac{2\pi * k(dx) * T_2(C_1 - 1)}{\ln \frac{r_o}{r_i}} \tag{82}$$

$$\int_0^l dx = \frac{\ln \frac{r_o}{r_i}}{2\pi k[C_1 - 1]}[\int_{-20^\circ c}^{0^\circ C} \frac{\dot{m}_{CL} * C_{PL} * (dT_c)}{C_2 T_2} + \int_{-20^\circ c}^{0^\circ C} \frac{\dot{m}_{CG} * C_{PG}(dT_c)}{C_2 T_c} + \int_0^{\dot{m}_c} \lambda(d\dot{m}_{CG})] \tag{83}$$

From equation 81

$$C_2 = \frac{h_c r_i * \ln \frac{r_o}{r_i}}{h_c r_i \ln \frac{r_o}{r_i} + k(1 - C_1)} \tag{84}$$

And therefor we get required length "l" of the pipe

6.4.2 Thermodynamic Panel Energy Calculations

Using Energy Equation, $Input - Output + Consumption = Accumulation$

$$\dot{m} * C_p(T) * dT + d\dot{m} * \lambda = d\dot{Q} \tag{85}$$

Here,

$\dot{m}$ = mass flow rate

$C_p(T)$ = specific heat at constant pressure w.r.t temperature

dT = Change in Temperature

λ = Latent Heat of Vapourisation

$\dot{Q}$ = Heat transfer per unit time

Integrating both sides,we get

$$\dot{m} * C_p(T) * \Delta T + \dot{m} * \lambda = \dot{Q} \tag{86}$$

$$\dot{m} * C_p(T) * (T_f - T_i) + \dot{m} * \lambda = \dot{Q} \tag{87}$$

We Know ,

$\dot{m}$ = 36 kg/sec

$C_p(T)$ = 1.38 KJ/kgK

Initial Temperature(T_i)=$-20°C$

Oulet Temperature(T_f)= $0°C$

λ = 248 KJ/kg

So,After applying the Values,

we get

$36 * 1.38 * (0 - (-20)) + 36 * 248 = \dot{Q}$ $\dot{Q} = 992.188 + 8935.92$

$$\dot{Q} = 9928.11 KJ/sec \tag{88}$$

6.5 Expenditure

6.5.1 Capital Expenditure(CAPEX)

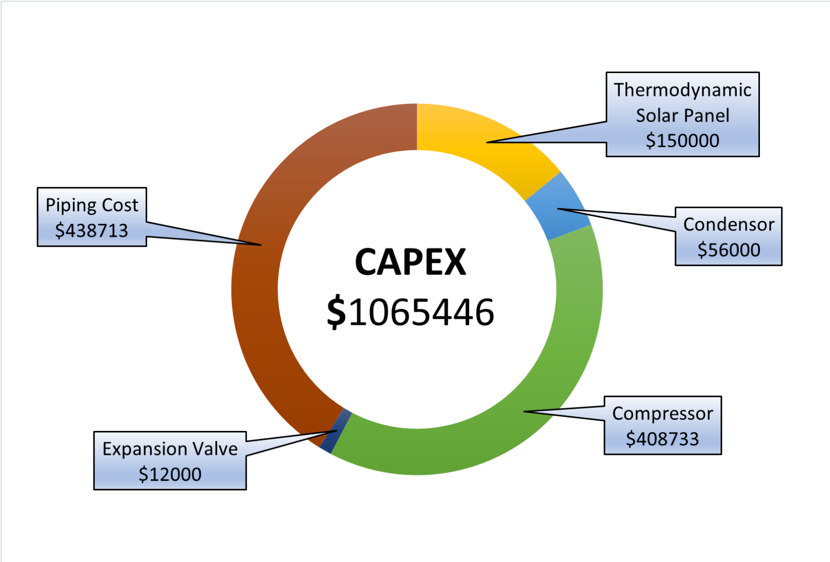

Figure 7: Capital Expenditure

For Thermodynamic Panel,

Price of each solar panel = 7500$

So Cost of 20 solar panel = 150000$

$$C_E = C_B * (\frac{Q}{Q_B})^M \tag{89}$$

For the compressor,

using figure 19,we got

Base cost for 250 kW power = $98400

So for 4857kW will be calculated using (equation 89)

where ,value of M = 0.46

Applying Values,

$$C_E = 98400 * (\frac{4857}{250})^{(0.48)} \tag{90}$$

$C_E = 408733$

For Heat exchanger,We get,

Base cost for 80m^2 heat transfer area =$32800

For 175.18m^2 will be calculated using equation89

where value of M=0.68

Applying Values,

$$C_E = 32800 * (\frac{175.18}{80})^{(0.68)} \tag{91}$$

C_E = \$56000

For thermostatic expansion valve,

We get that for our purpose the required expansion valve will cost around \$12000

Total Cost = \$150000 + \$408733 + \$56000 + \$12000=\$626733

And piping and installation cost will be taken as 70% of total cost

Therefore,Cost of piping and installation = \$626733 * $\frac{70}{100}$ = \$438713

Final cost after piping =\$626733 + \$438713=\$1065446

So, the Capital Expenditure of the process will be **\$1065446**

6.5.2 Operational Expenditure(OPEX)

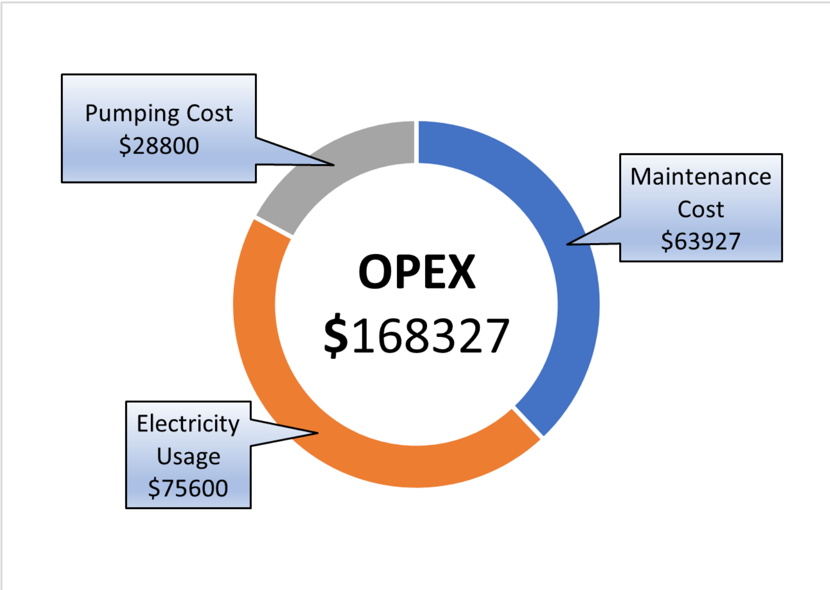

Figure 8: Operational Expenditure

OPEX includes maintenance cost,electricity cost,pumping cost.

So,Maintenance cost is 6% of total capital expenditure

Therefore, Maintenance Cost = \$1065446*$\frac{6}{100}$ = \$63927

For Electricity Cost,

Cost of per kWh in india is \$0.105

So,the cost for 1000kWh will be \$105

If the compressor is running for 8hr per day cost will be \$840

And it is running for 3months cost will be \$75600

For Pumping cost,

The cost for running pump for 3months and each day 8hr will be \$28800

Total Operational cost =\$28800 + \$75600 + \$63927

So, the Operational Expenditure of the process will be **\$168327** for one year

6.6 Break Even Point

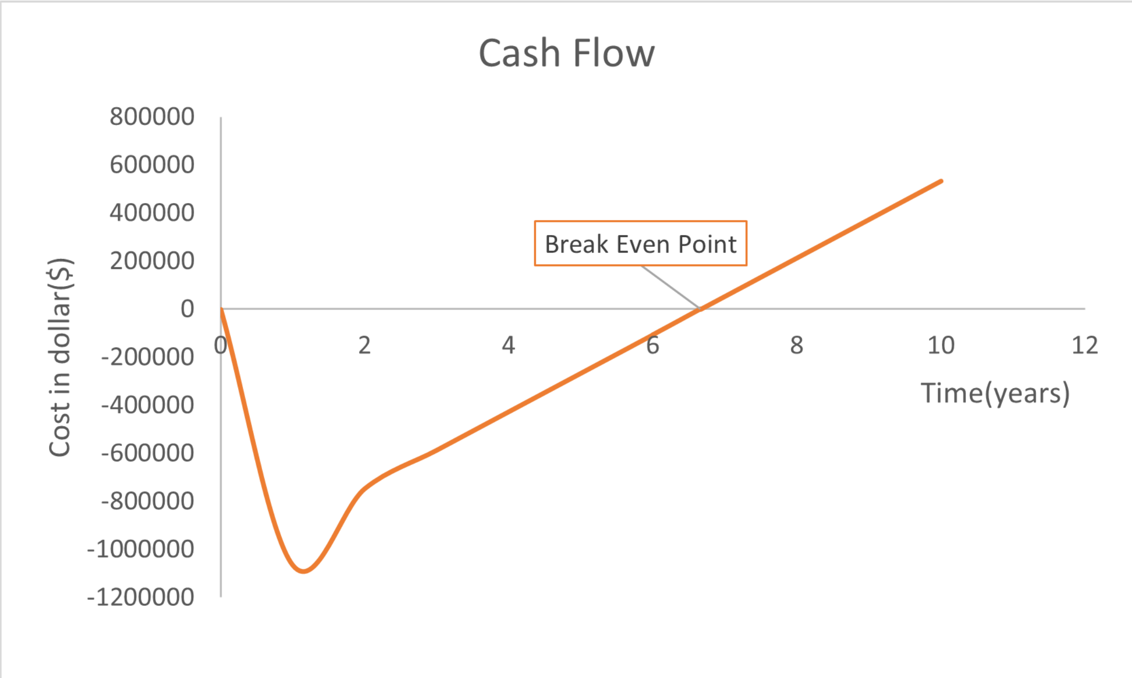

Figure 9: Cost Analysis

Considering Cost of Electricity =\$0.105 per unit
If we us geyser to heat water of same flow rate having efficiency of 95%

$$CostofElectricityused = \frac{\dot{m} * C_p * t * (costofelectricityperunit)}{\eta} \tag{92}$$

$= \frac{60000*40187*8*90*(0.105)}{0.95*3600} = \$328260.8$

$$BreakEvenPoint = \frac{CAPEX}{(OPEX)_{gyser} - (OPEX)_{thermodynamic}} \tag{93}$$

$= \frac{\$1065446}{\$(328260.8-168327)}$
$= 6.66 years$

After **6 years and 8 months** we got our total money we have expended in setup and running

6.7 Process Flow Diagram

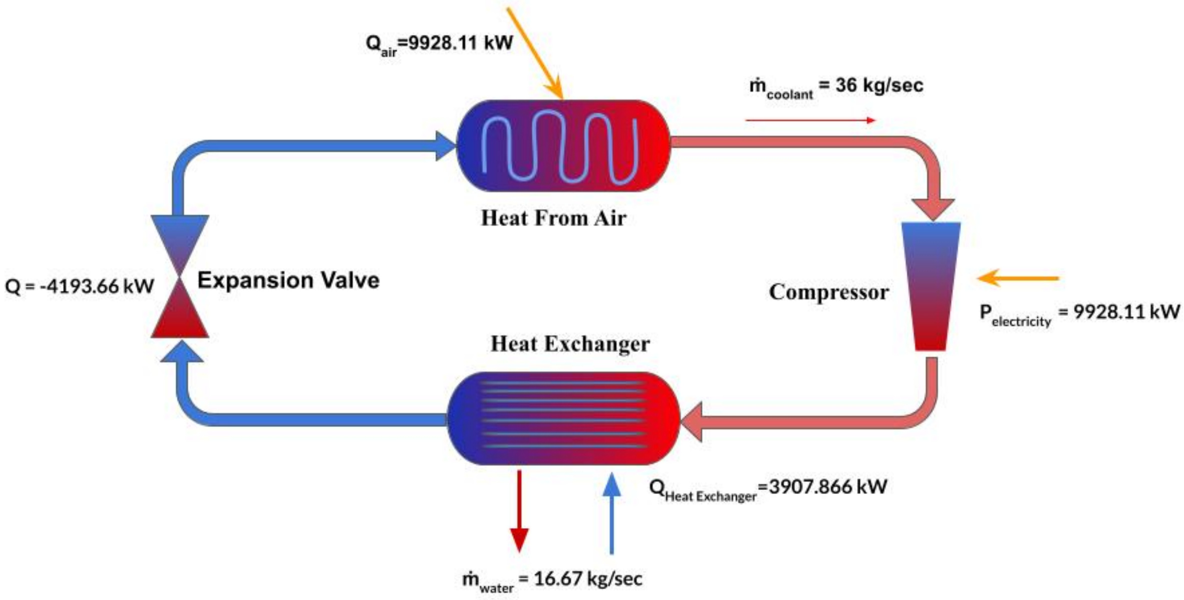

Figure 10: Cost Analysis

7 Conclusion

We have developed a comprehensive thermodynamics water heating system incorporating thermodynamic solar panel, condenser, compressor, valve,,pump, etc.As it is a type of renewable energy technology that has gained popularity in recent years due to their energy efficiency and cost-effectiveness.

To begin, we are absorbing heat from the air using thermodynamic solar panel of dimensions (2m*0.8m) to increase the temperature of coolant R407C from $-20°C$ to $0°C$.Then we are compressing the coolant R407C using scroll compressor to increase the temperature and pressure from $0°C$ and $461kPa$ to $82°C$ and $2680kPa$. Further, it passes to condenser where coolant is losing heat to heat up water of flow rate 62,000Kg/h and raising its temperature to 56°C which is the used for commercial use. After losing heat, coolant passes through expansion valve where the temperature and pressure is further decreased.

Overall, thermodynamic water heating systems are a reliable and sustainable way to provide hot water for residential and commercial use. They offer a great alternative to traditional heating systems and can help reduce our reliance on fossil fuels, saving money and reducing carbon emissions in the process.

8 Future Scope

In the future, emerging technologies will continue to transform the water heating industry, enabling us to create more efficient, reliable, and sustainable systems. Here are some additional points on the future scope:

- Integration of Artificial Intelligence: With the advent of artificial intelligence, water heating systems can be optimized in real-time, resulting in improved energy efficiency, reduced downtime, and lower operating costs.

- Use of Renewable Energy Sources: Renewable energy sources such as wind and solar power can be harnessed to provide energy for water heating systems. This approach will reduce the reliance on fossil fuels, resulting in lower greenhouse gas emissions and a more sustainable future.

- Internet of Things (IoT) Integration: IoT sensors can be integrated into water heating systems, enabling remote monitoring and control. This feature will facilitate predictive maintenance and prevent equipment failure, thereby reducing repair costs and downtime.

- Heat Recovery: Heat recovery systems can be integrated into water heating systems, allowing the heat generated from the system to be repurposed for other applications. This approach will reduce energy consumption and lower operating costs.

- Energy Storage: Energy storage systems can be used to store excess energy generated by water heating systems. This approach will ensure a consistent supply of energy and reduce the reliance on the power grid.

In conclusion, the water heating industry is poised for significant transformation in the near future. By leveraging emerging technologies and sustainable practices, we can create more efficient, reliable, and eco-friendly systems that meet the evolving needs of our society.

9 References:

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https://pubs.acs.org/doi/pdf/10.1021/ed027p584.3

A Code

Code for thermodynamic solar panel for length of tube tube in solar panel

```
#include<bits/stdc++.h>
using namespace std;

class Node {
public:
    float data;
    Node* next;
};

void Data(Node* head,float T){
  Node* headref = head;
  int i = 0;
  float A[7];
  A[0] = 560 + 18.2*pow(T,1) + 0.222
*pow(T,2) + 1.09e-03*pow(T,3);//P
  A[1] = 1.40 + (3.975e-3)*pow(T,1) +
(3.14e-5)*pow(T,2) +
(9.14e-7)*pow(T,3);//Cpl
  A[2] = 93.4-0.333*T;//Kl
  A[3] = 1233 - 3.81*pow(T,1) -
4.08e-03*pow(T,2) + 1.38e-04*pow(T,3)
+ 5.37e-06*pow(T,4);//rhol
  A[4] = 18.9 + 0.644*pow(T,1) +
8.84e-03*pow(T,2) + 5.96e-05*pow(T,3)
+ 1.11e-07*pow(T,4);//rhov
  A[5] = 218 - 2.83*pow(T,1) + 2.20e-2
*pow(T,2);//mul
  A[6] = 214 - 0.907*pow(T,1) -
8.57e-03*pow(T,2) - 1.38e-04
*pow(T,3);//lambdal
```

Figure 11: Code 1

```
*pow(T,3);//lambdal
  while(1){
    headref->data = A[i];
    if(i==6){break;}
    headref->next = new Node();
    headref=headref->next;
    i++;
  }
}

float Cpg(float T){
  float A[3]= {12.268, 11.701, 4.636};
  float B[3]= {-0.0699, 0.0216,
0.0618};
  float C[3]= {0.000394,
0.0000868, -0.0000309};
  float D[3]=
{-0.000000837, -0.000000112, 0};
  float X[3]= {0.381,0.179,0.439};
  float Sum=0;
  for(int i=0;i<3;i++){
   float Cp = 4.184
*((A[i]*T)+(B[i]*(pow(T,2)))+(C[i]*(po
w(T,3)))+(D[i]*(pow(T,4))));
   Sum+=Cp*X[i];
  }
  Sum/=86.2;
  return Sum;
}
```

Figure 12: Code 2

```
  return Sum;
}

float wall_temp(float Ti,float Ts){
  float
To,Tn,ri,k,ro,mcg,dmcg,C1,C3,ha,d,hc,P
,sigma;
  ri=1e-2;//m
  ro=1.25e-2;//m
  k=247;//W/m.K
  ha=35;//W/m^2.K
  d=4e-2;//m
  C1=(2*3.142*k)/(ha*d*log(ro/ri)+2*
3.142*k);
  //Let the fractional phase change of
coolant be given by following equation
(function of temperature varying along
length)
  mcg = pow((exp(Ts/Ti+
1)-1)/(exp(1)-1),4);
  /*The differentiation of mass
changing phase with respect to
temperature is given by following
equation
     (function of temperature varying
along length)*/
  dmcg = 4*pow((exp(Ts/Ti+
1)-1),3)*exp(Ts/Ti+
1)/(Ti*pow((exp(1)-1),4));
  //Let the initial guess for wall
```

Figure 13: Code 3

```
1)/(Ti*pow((exp(1)-1),4));
  //Let the initial guess for wall
temperature To be 0Â°C
  To=0;
  Node* head = new Node();
  //Import the data of coolant for
temperature Ts
  Data(head,Ts);
  int i=0;
  float A[7];
  while(head!=NULL){
    Node* temp = head;
    A[i] = head->data;
    head = head->next;
    delete temp;
    i++;
  }
  sigma=11.8*1e-3;
  C3=0.00122*(pow(A[2]*
1e-3,0.79)*pow(A[1]*
1e3,0.45)*pow(A[3],0.49)/(pow(sigma,0.
5)*pow(A[5]*1e-6,0.29)*pow(A[6]*
1e3,0.24)*pow(A[4],0.24)));
  while(1){
    P = (560 + 18.2*pow(To,1) + 0.222
*pow(To,2) + 1.09e-03*pow(To,3))*1e3;
    hc = C3*pow(To-Ts,0.24)*pow(P-
A[0]*1e3,0.75);
    hc*=0.05;//factor fs of 0.05
    hc+=68599;//hfc'=68599 W/m^2.K
```

Figure 14: Code 4

```
      hc+=68599;//hfc'=68599 W/m^2.K
      //We get new wall temperature
using following equation
      Tn =
(ri*log(ro/ri)*(Ts)*hc)/(k*(hc*ri*log(
ro/ri)/k+1-C1));
      if(Tn==0){break;}
      if((((Tn/To)-1)*
100>-5)&&(((Tn/To)-1)*100<5)){break;}
      To=Tn;
   }
   //Using simpson's rule for 2nd
degree polynomial to solve for length
   //heat absorbed by coolant in liquid
phase (CpL.dT) is given by following
   float sum = (-1)*A[1]*1e3*(36.05/(4*
90))*(1-mcg)*(hc*ri*log(ro/ri)/k+1-
C1)/((Ts+273)*hc*2*3.142*ri*(C1-1));
   //heat absorbed by coolant in gas
phase (CpG.dT) is given by following
   sum+=(-1)*Cpg(Tn)*1e3*(36.05/(4*
90))*mcg*(hc*ri*log(ro/ri)/k+1-
C1)/((Ts+273)*hc*2*3.142*ri*(C1-1));
   //heat absorbed by coolant during
phase transition (lambdaL.(dm/dT)) is
given by following
   sum+=(-1)*A[6]*1e3*(36.05/(4*
90))*dmcg*(hc*ri*log(ro/ri)/k+1-
C1)/((Ts+273)*hc*2*3.142*ri*(C1-1));
   return sum;
```

Figure 15: Code 5

```
    return sum;
  }

  int main(){
    float Ts,Ti;
    cout<<"Please enter the coolant
inlet temperature in K"<<endl;
    cin>>Ts;
    Ts-=273;
    Ti=(-1)*Ts;
    //Using simpson's rule to solve with
n=1000
    float p = (-1)*Ts/1000;
    float C0=wall_temp(Ti,Ts);
    cout<<"Ts (K)"<<"   "<<"length
(m)"<<endl;
    cout<<Ts+273<<"      "<<C0
*p/3<<endl;
    int i=1;
    while(Ts<=(-1)*p){
      Ts+=p;
      if(i%2==1){
        C0 += 4*wall_temp(Ti,Ts);
        cout<<Ts+273<<"   "<<C0
*p/3<<endl;
      }
      else if(i%2==0){
        C0 += 2*wall_temp(Ti,Ts);
        cout<<Ts+273<<"   "<<C0
*p/3<<endl;
```

Figure 16: Code 6

```
            C0 += 4*wall_temp(Ti,Ts);
            cout<<Ts+273<<"    "<<C0
*p/3<<endl;
        }
        else if(i%2==0){
            C0 += 2*wall_temp(Ti,Ts);
            cout<<Ts+273<<"    "<<C0
*p/3<<endl;
        }
        i++;
    }
    Ts=0;
    C0 += wall_temp(Ti,Ts);
    cout<<Ts+273<<"    "<<C0*p/3<<endl;
    C0*=p/3;
    cout<<"length  =  "<<C0<<" meters";
    return 0;
}
```

Figure 17: Code 7

B Reference Graphs

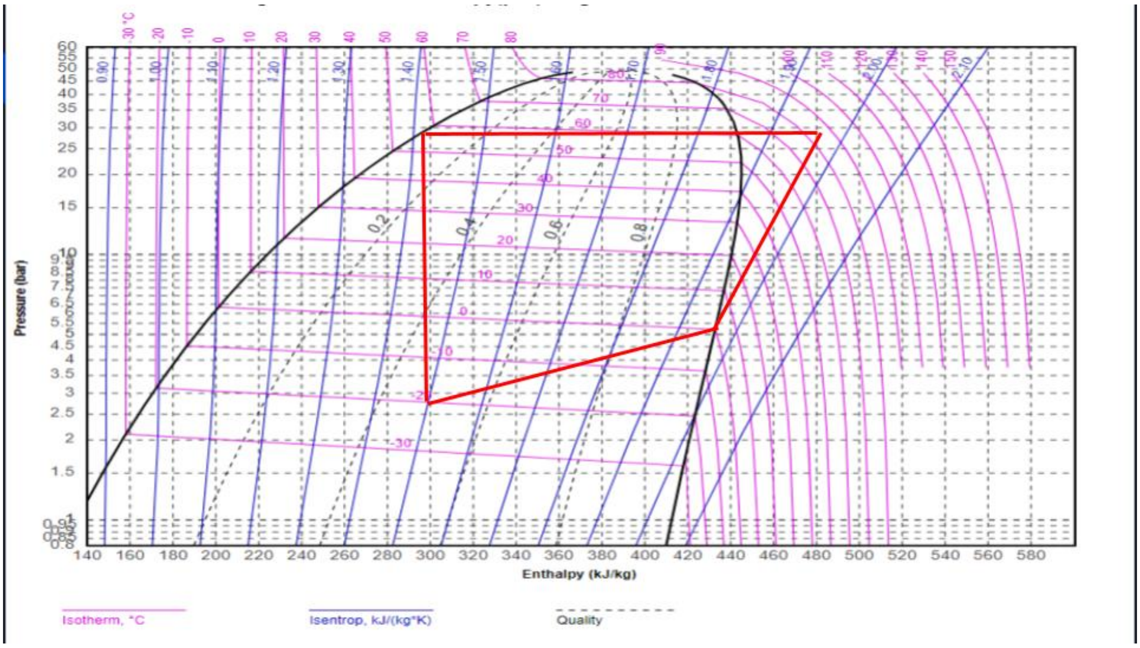

Figure 18: Flow Diagram
https://www.flycarpet.net/en/phonline

C Reference Tables

Figure 19: Equipment Capacity Delivered Capital Cost Correlation

Table 2.1

Typical equipment capacity delivered capital cost correlations.

Equipment	Material of construction	Capacity measure	Base size Q_B	Base cost C_B ($)	Size range	Cost exponent M
Agitated reactor	CS	Volume (m^3)	1	1.15×10^4	1–50	0.45
Pressure vessel	SS	Mass (t)	6	9.84×10^4	6–100	0.82
Distillation column (empty shell)	CS	Mass (t)	8	6.56×10^4	8–300	0.89
Sieve trays (10 trays)	CS	Column diameter (m)	0.5	6.56×10^3	0.5–4.0	0.91
Valve trays (10 trays)	CS	Column diameter (m)	0.5	1.80×10^4	0.5–4.0	0.97
Structured packing (5 m height)	SS (low grade)	Column diameter (m)	0.5	1.80×10^4	0.5–4.0	1.70
Scrubber (including random packing)	SS (low grade)	Volume (m^3)	0.1	4.92×10^3	0.1–20	0.53
Cyclone	CS	Diameter (m)	0.4	1.64×10^3	0.4–3.0	1.20
Vacuum filter	CS	Filter area (m^2)	10	8.36×10^4	10–25	0.49
Dryer	SS (low grade)	Evaporation rate (kg $H_2O \cdot h^{-1}$)	700	2.30×10^5	700–3000	0.65
Shell-and-tube heat exchanger	CS	Heat transfer area (m^2)	80	3.28×10^4	80–4000	0.68
Air-cooled heat exchanger	CS	Plain tube heat transfer area (m^2)	200	1.56×10^5	200–2000	0.89
Centrifugal pump (small, including motor)	SS (high grade)	Power (kW)	1	1.97×10^3	1–10	0.35
Centrifugal pump (large, including motor)	CS	Power (kW)	4	9.84×10^3	4–700	0.55
Compressor (including motor)		Power (kW)	250	9.84×10^4	250–10,000	0.46
Fan (including motor)	CS	Power (kW)	50	1.23×10^4	50–200	0.76
Vacuum pump (including motor)	CS	Power (kW)	10	1.10×10^4	10–45	0.44
Electric motor		Power (kW)	10	1.48×10^3	10–150	0.85
Storage tank (small atmospheric)	SS (low grade)	Volume (m^3)	0.1	3.28×10^3	0.1–20	0.57
Storage tank (large atmospheric)	CS	Volume (m^3)	5	1.15×10^4	5–200	0.53
Silo	CS	Volume (m^3)	60	1.72×10^4	60–150	0.70
Package steam boiler (fire-tube boiler)	CS	Steam generation (kg$\cdot h^{-1}$)	50,000	4.64×10^5	50,000–350,000	0.96
Field erected steam boiler (water-tube boiler)	CS	Steam generation (kg$\cdot h^{-1}$)	20,000	3.28×10^5	10,000–800,000	0.81
Cooling tower (forced draft)		Water flowrate ($m^3 \cdot h^{-1}$)	10	4.43×10^3	10–40	0.63

CS = carbon steel; SS (low grade) = low-grade stainless steel, for example, type 304; SS (high grade) = high-grade stainless steel, for example, type 316.

(Chemical Process Design and Integration, 2nd Edition by Robin and Smith)

Table 2: Cost Calculation Table

Year	Electric Boiler	Electric Immersion Heater	Heating Oil	Bottled Gas	LPG Gas	Mains Gas
1 year	-2,993	-3,644	-1,889	-1,815	-2,888	-3,019
2 years	-2,576	-2,439	-1,734	-522	-2,668	-2,931
3 years	-2,158	-1,233	-1,580	772	-2,448	-2,842
4 years	-1,741	-28	-1,425	2,065	-2,228	-2,753
5 years	-1,324	1,178	-1,271	3,358	-2,008	-2,665
6 years	-907	2,384	-1,117	4,651	-1,787	-2,576
7 years	-490	3,589	-962	5,944	-1,567	-2,487
8 years	-72	4,795	-808	7,238	-1,347	-2,398
9 years	345	6,000	-653	8,531	-1,127	-2,310
10 years	762	7,206	-499	9,824	-907	-2,221
Needs replacement						
11 years	2,769	8,562	-345	11,117	-687	-2,132
12 years	3,186	9,767	-190	12,410	-467	-2,044
13 years	3,604	10,973	-36	13,704	-247	-1,955
14 years	4,021	12,178	119	14,997	-27	-1,866
15 years	4,438	13,384	273	16,290	193	-1,778
Needs replacement						
16 years	4,855	14,590	3,384	19,475	2,306	203
17 years	5,272	15,795	3,539	20,768	2,526	292
18 years	5,690	17,001	3,693	22,062	2,746	381
19 years	6,107	18,206	3,848	23,355	2,966	469
20 years	6,524	19,412	4,002	24,648	3,186	558