Design of a Low capacity Evaporator of a Refrigeration Unit

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Overview

This project has been assigned to students in their first course in thermodynamics, in an attempt to satisfy the ABET requirement of enhancing the design contents of engineering courses. Although, refrigeration cycle is studied in thermodynamics classes and textbooks\textsuperscript{1,2,3} the details regarding the performance of each component of the cycle and its effect on the other parts of the refrigeration system is not considered. Moreover, little is said regarding the choice of working fluid and selection or design of individual components of the cycle. To help students understand how these components work together and how their inter-related performance affect the overall coefficient of performance of the cycle, they were required to choose an environmentally acceptable working fluid and to design the evaporator unit of a small capacity refrigeration unit (0.4 ton of refrigeration). Since students involved with this project do not have a fluid mechanics background, pressure drop in the system except for the case in the expansion valve is ignored. Due to the textbook\textsuperscript{1} used in their thermodynamics class students are familiar with the different mechanisms of heat transfer and are somewhat knowledgeable about convective heat transfer coefficient. This project not only teaches students how to manage and solve an open-ended problem, but also helps them to practice conservation of mass, first law and second law of thermodynamics. Students also learn more about conductive and convective modes of heat transfer by implementing the design steps for sizing the evaporator unit. Students used Mathcad software to perform necessary calculations.

Project Statement and Specification

The evaporator component of a small refrigeration unit of 0.4-ton capacity is to be designed. The ultimate goal of this project is to specify the size, length and number of tubing (or number of turns in a tube) of a double pipe heat exchanger, used as the evaporator unit of the refrigeration cycle. Water is used as a heat source in the evaporator and as a coolant in the condenser unit. It is assumed that the water inlet temperature to the evaporator is 15\degree C and that the refrigerator is sitting in a 25\degree C environment. Owing to concerns regarding the effects of chlorine on the earth’s protective Ozone layer and in
compliance with the Montreal protocol, the choice of the refrigerant is limited to R-22, R-134a, or ammonia. The compressor available for this design is a hermetically sealed reciprocating compressor with following specifications:

- Rotational Speed = 2800 r.p.m.
- Swept Volume = 15 cm³/rev
- Clearance Volume = 5% of the swept volume
- Polytropic Exponent = 1.05

Ignoring the effect of pressure drop across the intake valve, the temperature rise in the intake system, and the mechanical condition of the valves and piston seals, the volumetric efficiency of a reciprocating compressor can be estimated as follow:

\[ \eta_v = 1 - \frac{V_c}{V_s} \left( \frac{1}{(pr)^n} - 1 \right) \]  

(1)

Where:
- \( \eta_v \) = Volumetric Efficiency
- \( V_c / V_s \) = Clearance Volume / Swept Volume
- \( pr \) = Compression Pressure Ratio
- \( n \) = Polytropic Exponent

It is assumed that the convection heat transfer on the refrigerant side during 2-phase evaporation is \( h_{\text{boil}} = 20000 \text{ W/(m².oK)} \), whereas the convection heat transfer coefficient for water and refrigerant (in the vapor phase) can be estimated from the following equations:

\[ h_{\text{water}} = 1707.16 V^{0.8}_{\text{water}} D_e^{0.2} \]  

(2)

\[ h_{\text{refrigerant}} = 4.37 V^{0.8}_{\text{refrigerant}} D_p^{0.2} \]  

(3)

In the above equations the convection heat transfer coefficients are calculated in \( \text{W/(m².oK)} \). \( V \) is the velocity of fluids in m/s. \( D_p \) and \( D_e \) are pipe inside diameter and annulus hydraulic diameter respectively in meters. Assume that water is running through the annulus. Although the convection heat transfer coefficient for refrigerant depends on the type of the refrigerant used and is a function of the temperature of the refrigerant, it is assumed that the above equation for heat transfer coefficient of refrigerant renders a relatively good approximate value of it, regardless of the design choice of the working fluid. For the sake of simplicity, assume that the pressure drop in the evaporator, condenser and compressor-connecting pipe is negligible. In the process, the mass flow rates of water and refrigerant, the power requirement to run the system, the monthly cost of electricity, the rate of heat loss from compressor, the rate of cooling in the condenser, the factor of safety with regard to Hoop’s stress in the selected pipes, and unit’s coefficient of performance are also to be evaluated.
Design procedure and Implementation

As mentioned before the objective of this project was to introduce thermodynamics students to design process and its iterative nature. To achieve this goal, students in thermodynamics class were divided into groups of three, with each group being responsible for its own detail design of the evaporate unit of the cycle. Although students were guided (through handouts and 2 class lecture periods) to make appropriate engineering decisions for their respective designs, none of the design decision such as cycle’s temperatures and pressures were not given to them. In fact the key final decisions such as, what refrigerant to use, what pressure ratio across compressor to pick, what the state of the refrigerant at the compressor inlet and condenser exit to be, were made by each group independently. As a result each group of students came up with their own unique design.

What is described below is the illustration of one of the many viable solutions to this open ended problem. To back the decisions made in the process, numerical values of those corresponding choices and their related thermodynamic properties are incorporated in this document.

In the following, each important step in the design process is described under a separate title, while for the sake of identification of each state point, a schematic diagram of a typical refrigeration cycle together with its T-S diagram are provided in the Appendix A. Process 2-3 in the T-S diagram indicates the polytropic process in the compressor while 2-3s refers to an isentropic one. Occasional references to the state points in the T-S diagram are made in the following narrative.

Choice of Refrigerant

One of the objectives of this endeavor was to introduce our students to environmentally responsible design. Therefore the very first attempt in their design process was to choose an environmentally acceptable refrigerant. Prior to the international ban on the usage of refrigerants containing chlorine in their chemical formula, R-12, known as Freon, was the most prevalent working fluid in the commercial units. Since the working fluid is predominantly in 2-phase state in the evaporator and condenser and because the pressure of the refrigerant in the heat...
exchanger tube dictates the thickness of the tube wall, for low weight consideration of the
unit, R-134a is selected as the refrigerant for the evaporator design.

Selection of an Appropriate Pressure Ratio across the Compressor

The following discussion in the class room, during a lecture period, guided students in
selecting an appropriate pressure ratio across the given compressor. Numerical values
given below were not provided to the students, and are mentioned here for the sake of
further illustration of the points made.

Figure 3 in Appendix B is a plot of the volumetric efficiency (Equation 1) of the
reciprocating compressor vs. the compression pressure ratio of the compressor. It is seen
that as the compression ratio across the compressor increases, the volumetric efficiency
of the compressor decreases. This will result in a lower mass flow rate of refrigerant in
the cycle, which directly affects the cooling capacity of the cycle, which in turn will
effectively lower the coefficient of performance for the cycle. On the other hand at low
compression ratios a higher volumetric efficiency for the compressor is achieved, with an
effective lower temperature differences between the condenser and evaporator units. In
light of the Carnot Coefficient of performance

\[ \beta_{\text{carnot}} = \frac{T_{\text{evaporator}}}{T_{\text{condenser}} - T_{\text{evaporator}}} \]

low compression pressure ratio brings about a higher coefficient of performance for the cycle.
At the same time, a lower compression ratio results in a lower temperature in the
condenser. Since the direction of heat transfer in the condenser is from the condenser to
the environment at 25°C, one needs to attain an average temperature for the refrigerant in
the condenser 10° to 15°C above the surrounding temperature, to have an effective rate of
heat transfer between the condenser and the environment. In order to have a relatively
light and efficient refrigeration unit, with a reasonable coefficient of performance (yet
with an effective rate of heat transfer with the surrounding in the condenser unit), a
proper choice of the compression ratio is an important step in this project. Students were
also advised on the iterative nature of design, by recommending that their pressure ratio
choice together with other design parameters should be checked and possibly altered so
that the design specification of 0.4 ton of refrigeration is met. Following those steps
described above a compression pressure ratio of pr = 2.9 is chosen, for illustrative
purpose, in this document.

Selection of Evaporator Pressure and Inlet Condition to the Compressor

Once the pressure ratio for the compressor is chosen then the evaporator pressure is
selected so that an effective rate of heat transfer between the water, as a heat source, and
the evaporator is achieved. By choosing the evaporator saturation temperature (T_{1} = -1.23
°C ; 16.23°C below the water inlet temperature) the corresponding saturation pressure in
the evaporator is looked from saturation table (P_{1} = P_{2} = 2.8 bar for R-134a). With the pr
= 2.9, the condenser pressure is then calculated (P_{3} = P_{4} = 8.12 bar). To avoid the
possibility of what is known as a wet compression in the compressor, the intake condition
to the compressor is considered to be superheated vapor, therefore, a temperature, 2° to 3
°C above the saturation temperature at the evaporator pressure is recommended (T_{2} = 1°

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C). To avoid the possibility of having liquid in the compressor as the refrigerating load alters, usually a higher temperature is considered as inlet to the compressor. With the inlet condition known (T₂ = 1°C and P₂ = 2.8 bar) the inlet specific volume is looked up (v₂ = 0.07278 m³/kg) from superheated vapor table.

Mass Flow Rate of the Refrigerant

Using equation 1, volumetric efficiency of the compressor is found (\(\eta_v = 91.22\%\)). With the knowledge of compressor rotational speed, volumetric efficiency, the swept volume of the compressor, and the inlet specific volume of the charge to the compressor, the mass flow rate of the R-134a can be calculated (\(\dot{m}_{\text{ref}} = 8.77 \times 10^{-3} \text{ kg/s}\)).

Inlet State at the Evaporator and Expansion Valve

Using the first law of thermodynamics, and the refrigerant capacity of 0.4 ton the inlet enthalpy to the evaporator is evaluated (h₁ = 87.45 kJ/kg). Since the heat transfer is negligible across the expansion valve with the absence of work first law for expansion valve leads to value of enthalpy at the inlet to the expansion valve (h₄ = h₁ = 87.45 kJ/kg).

Evaluating the Cycle’s Coefficient of Performance

With inlet and exit pressure known in the compressor and (v₂ = 0.07278 m³/kg, P₂ = 2.8 bar), the Polytropic equation:

\[
P_2 v_2^{1.05} = P_3 v_3^{1.05} \quad (4)
\]

renders the specific volume at state 3 (v₃ = 0.02640 m³/kg). The power input to the compressor is found using the equation for work per unit charge of a polytropic process and the charge mass flow rate:

\[
\dot{W}_{\text{Comp}} = \dot{m}_{\text{ref}} \frac{P_3 v_3 - P_2 v_2}{(1 - n)\eta_{\text{mechanical}}} = -0.2187 kW \quad (5)
\]

Knowing 2 intensive properties at state 3 (v₃ and P₃) other properties at state 3 are looked up from superheated R-134a table. Application of the First Law for the condenser gives the heat loss from condenser to the surroundings (\(\dot{Q}_H = -1.63 kW\)). The cycle coefficient of performance is evaluated then as:

\[
\beta_{\text{cycle}} = \frac{\dot{Q}_C}{\left| \dot{W}_{\text{comp}} \right|} = \frac{0.4\text{ton} \times 212 \text{ kJ/min} \times \frac{1\text{min}}{60\text{s}}}{0.2187} = 6.46
\]
Comparison with Carnot’s Coefficient of Performance

Finding average temperature in the evaporator and condenser using:

\[ T_c = T_{avg\_evap} = \frac{\dot{Q}_c}{s_2 - s_1} = 271.94° K \] (6)

\[ T_H = T_{avg\_cond} = \frac{\dot{Q}_H}{s_4 - s_3} = 305.25° K \] (7)

The Carnot coefficient of performance is readily evaluated to be:

\[ \beta_{carnot} = \frac{T_{avg\_evap}}{T_{avg\_cond} - T_{avg\_evap}} = 8.2 \]

As it is expected \( \beta_{carnot} > \beta_{cycle} \).

Evaporator Pipe Size

The energy loss by the water in the evaporator is equal to energy gain by the charge in the evaporator. That is:

\[ \dot{m}_{water}(h_{e\_water} - h_{i\_water}) = 0.4t_\text{on} \]

Assuming that the water exit temperature is 7° C then \( \dot{m}_{water} = 0.04251 \text{ kg/s} \). Using continuity equation:

\[ \dot{m} = \frac{AV}{v} \] (8)

Where \( V \) is the fluid velocity, \( A \) is the cross section of the tube (or annulus cross section in case of water flowing in the annulus) and \( v \) is the specific volume of the fluid.

Assuming that the vapor refrigerant velocity (\( V = 6.2 \text{ m/s} \)), with the aid of the tables for dimensions of seamless copper tubing, the M type copper tube of inside and outside diameters \( D_{inner\_ref} = 1.142 \text{ cm} \) and \( D_{outer\_ref} = 1.270 \text{ cm} \) respectively is selected for R-134a in this trial. Water is assumed to have a velocity of 1.13 m/s in this solution. The calculation then indicates that water is flowing in an M type copper annulus with outer pipe having, outside diameter of \( D_{outer\_water} = 1.588 \text{ cm} \) and inside diameter of \( D_{inner\_water} = 1.446 \text{ cm} \). The annulus inner pipe has outside diameter of 1.270 cm and inside diameter of 1.142 cm (please see evaporator pipe diagram in Appendix A) for this particular solution.

Length of the Evaporator

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Since the convective heat transfer coefficient in case of evaporation is different than the one for superheated vapor case, the evaporator is considered to be of two regions, the entry region, \( L_{\text{evaporation}} \), where a 2-phase flow of charge exists and an end region where a superheated charge flows, \( L_{\text{superheat}} \), (please see evaporator pipe diagram in Appendix A). The evaporator is considered as a double pipe counter-flow heat exchanger, with the refrigerant running through the pipe and water flowing in the annulus. Because the tubes are made of copper (high thermal conductivity) and tube wall thickness is very small, conductive resistance of the tubes is neglected. With this assumption the overall heat transfer coefficient for the evaporating length of the evaporator is estimated as:

\[
U_{01} = \frac{1}{\frac{1}{h_{\text{water}}} + \frac{D_{\text{outer_ref}}}{h_{\text{boil}}D_{\text{inner_ref}}}}
\]  

(9)

Where \( h_{\text{water}} \) is calculated from equation 2. The overall heat transfer coefficient for the superheated vapor length of the evaporator is evaluated using the following equation:

\[
U_{02} = \frac{1}{\frac{1}{h_{\text{water}}} + \frac{D_{\text{outer_ref}}}{h_{\text{refrigerant}}D_{\text{inner_ref}}}}
\]  

(10)

Where \( h_{\text{refrigerant}} \) is calculated via equation 3. The amount of heat exchange between water and refrigerant for the evaporative length is:

\[
\dot{Q}_1 = m_{\text{dotref}}(h_{2g} - h_1) = 1.3953kW
\]  

(11)

Where \( h_{2g} \) is the saturated vapor enthalpy at the evaporator pressure. The amount of heat transfer between water and superheated charge in the evaporator therefore is:

\[
\dot{Q}_2 = \dot{Q}_c - \dot{Q}_1 = 1.413 - 1.3953 = 0.018kW
\]  

(12)

Water temperature at the point where the superheated charge region begins can be calculated from the following equation:

\[
T_{\text{water}}_{\text{prime}} = \frac{\dot{Q}_1}{m_{\text{water}}c_{p_{\text{water}}}} + T_{\text{water}}_{\text{out}}
\]  

(13)

Where \( T_{\text{water}}_{\text{out}} = 7^\circ C \) is the water outlet temperature. The length of the evaporative length and superheated length of the evaporator are calculated from equations (14) and (15) respectively as follows:
\[ \dot{Q}_1 = U_{1} \pi D_{\text{outer_ref}} L_{\text{evaporation}} \Delta T_{lm1} \] (14)

\[ \dot{Q}_2 = U_{2} \pi D_{\text{outer_ref}} L_{\text{superheat}} \Delta T_{lm2} \] (15)

Where

\[ \Delta T_{lm1} = \frac{(T_{\text{Water_prime}} - T_1) - (T_{\text{Water_out}} - T_1)}{\ln \left( \frac{T_{\text{Water_prime}} - T_1}{T_{\text{Water_out}} - T_1} \right)} \] (16)

And

\[ \Delta T_{lm2} = \frac{(T_{\text{Water_in}} - T_2) - (T_{\text{Water_prime}} - T_1)}{\ln \left( \frac{T_{\text{Water_in}} - T_2}{T_{\text{Water_prime}} - T_1} \right)} \] (17)

Where \( \Delta T_{lm1} \) and \( \Delta T_{lm2} \) are the Log Mean Temperature for evaporative section and superheated section of the evaporator. Substituting equations (9), and (16), into equation (14) and solve for \( L_{\text{evaporation}} \):

\[ L_{\text{evaporation}} = 0.61 \text{ m} \]

Substituting equations (10), and (17), into equation (15) and solve for \( L_{\text{superheat}} \):

\[ L_{\text{superheat}} = 0.731 \text{ m} \]

This renders a total length of:

\[ L_{\text{Total}} = L_{\text{evaporation}} + L_{\text{superheat}} = 1.341 \text{ m} \]

If it is assumed that each turn in a compact refrigeration unit is no longer than 0.4 m then the number of turns of the evaporator pipe is:

\[ N = 1.341/0.4 = 3.325 \text{ turn} \]

The factor of safety with respect to high pressure in the condenser pipe is calculated using Hoop’s stress as follow:

\[ \sigma = \frac{(P_2 - P_{\text{atmosphere}}) r_{\text{mean}}}{t} \]
Where $P_{atmosphere}$ is the atmospheric pressure, $t$ is pipe wall thickness, $\sigma_{yield}$ is the copper yield strength, and

$$F.S. = \frac{\sigma_{yield}}{\sigma}$$

For this design:

$$F.S. = 5.37$$

The cost of electricity is calculated to be $10.83 per month for this particular design.

Discussion of the Result

It should be noted once again that in this project the pressure drop in the evaporator, condenser and inlet and exit ports of the compressor is neglected. That is a big source of irreversibility in the cycle is ignored. Had the effect of friction been included in the analysis a much lower coefficient of performance would have resulted and a much longer length for the evaporator would have been obtained. The inlet temperature to the compressor, unlike what we assumed for this particular design solution, usually is higher than 2.3°C above the evaporator saturation temperature to safeguard against wet compression. Higher the inlet temperature, higher the irreversibility, which would further bring down the coefficient of performance and increases the required length of the evaporator. Stray heat would also affect the coefficient of performance adversely. The design however indicates a lower coefficient of performance for the unit than an ideal Carnot cycle ($\beta_{cycle} = 6.46, \beta_{Carnot} = 8.2$)

Conclusion

Although friction was ignored in this work, the project however contains some good educational values for a first course in thermodynamics. First of all it exposes students to challenges of design process and an open-ended problem. Secondly it covers a lot of fundamental concepts in thermodynamics. Thirdly gives students a mature view of how each component in the cycle affects the other components’ performance and how it influences the overall behavior of the system.

Each group of students came up with different design parameters and values for their design, as it is expected from an open ended problem like this.

At the end of this project, majority of the students indicated to the author that this project helped them understand thermodynamics concepts in great detail. They pointed out that
they have a better appreciation of how a design process evolves and how changing a single design parameter will affect the overall picture. They mentioned that they became aware of the iterative nature of the design process by having to change their selected design parameters several times till they finally met the design specifications.

Because of the textbook\textsuperscript{1} used in their thermodynamics class, students were aware of different mechanisms of heat transfer. The author, however, had to devote 2 lecture periods to extend the concepts discussed in the textbook to cover double pipe heat exchanger and introduce students to concepts such as log mean temperature. Students also enjoyed the analogy between flow of heat and electrical charge, when that analogy was used to give them a sense for overall heat transfer coefficient. The author found out, through testing, that students had better absorption of the new concepts after their exposure to this project.

Bibliography

APPENDIX A

Temperature
Evaporator
Condenser
Compressor
Specific Entropy
T-s Diagram
Point 1
Point 4
Point 3s
Point 2
Expansion Valve
Condenser
Compressor
Evaporator
Point 3
Point 3s

Temperature
Specific Entropy
Evaporator Pipe Diagram
### Saturation Tables

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<tr>
<th>Temperature (°C)</th>
<th>R-12 Saturation Pressure (bar)</th>
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**FIGURE 1**
Saturation Curves

![Saturation Curves Diagram]

FIGURE 2
## Compressor Properties

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Volumetric Efficiency vs. Pressure Ratio

\[
\text{Volumetric Efficiency} = 1 - \frac{V_c}{V_s} \left( \frac{1}{\text{Pressure Ratio}} \right) - 1
\]

Where:

\[ n = 1.05 \quad \frac{V_c}{V_s} = 0.05 \]

**FIGURE 3**