



A Practical Tool for Evaluating Refrigeration Systems Using R-134a

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Abstract

In this theoretical work, the ideal vapor-compression cycle using Refrigerant-134a (Tetrafluoroethane, a chlorine free refrigerant) is studied considering an evaporator pressure range of 100-200 kPa and a condenser pressure range of 0.8-1.6 MPa. The all four states of the thermodynamic cycle are calculated for a great number of conditions within the above pressure ranges. Thermodynamic properties of the refrigerant R-134a are evaluated using curve fitting polynomials. The deviations of the values generated from fitting curves from the actual tabulated data is less than 0.5%. Results are represented in a graph of COP vs. evaporator pressure for various condenser pressures. The results of this study is believed to be a quick reference especially for designers and engineering students.

Keywords: Thermodynamics, Refrigeration, Vapor-Compression, Refrigerant-134a.

R-134 Kullanan Soğutma Sistemleri için Pratik Hesaplama Aracı

Öz

Bu teorik çalışmada R-134a (Tetrafloretan, klor içermeyen bir soğutucu akışkan) kullanılan buhar sıkıştırımlı soğutma çevrimi 100-200 kPa buharlaştırma basıncı aralığında, 0.8-1.6 MPa yoğusturucu basıncı aralığında olmak üzere incelenmiştir. Termodinamik çevrimin dört aşaması bu basınç aralıklarında detaylı olarak hesaplanmıştır. R-134a soğutucu akışkanının termodinamik özellikleri eğri uydurma polinomları kullanılarak bulunmuştur. Bu eğrilerden elde edilen değerlerin tablo değerlerinden sapması %0.5'ten küçüktür. Sonuçlar farklı yoğusturucu basınçları için buharlaştırma basıncına karşı COP değerleri olarak sunulmuştur. Bu çalışmanın sonuçlarının özellikle soğutma sistemi tasarımcıları ve mühendislik öğrencileri için pratik bir hesaplama aracı olacağı değerlendirilmektedir.

Anahtar Kelimeler: Termodinamik, Soğutma, Buhar-Sıkıştırımlı, Soğutucu R-134a.

Nomenclature

| | |
|-----|--|
| COP | coefficient of performance defined as Q_{evap}/W |
| h | enthalpy, kJ/kg |
| P | pressure, kPa or MPa |
| Q | heat absorbed or rejected per kg of refrigerant, kJ/kg |
| s | entropy, kJ/kg-°C |
| T | temperature, °C |
| v | specific volume, m ³ /kg |
| W | work (done on) per kg of refrigerant, kJ/kg |

Greek Symbols: η isentropic efficiency of compressorSubscripts:*cond* condenser*evap* evaporator*f* liquid phase*g* gaseous phase*H* high, denoting to condenser*in* added in, as in compressor work*L* low, denoting to evaporator*sat* saturation**1. Introduction**

According to Encyclopedia Britannica, refrigeration is a process where heat is removed from an enclosed space or from a substance in order to lower its temperature. Refrigeration is primarily used for storing food at low temperatures in order to prevent the adverse effects of bacteria, yeast, and mold in the developed nations and rich regions in the developing world. Many products can be frozen by permitting them to be preserved for very long time periods with negligible loss in initial properties. Air conditioning has also become a commonly used refrigeration technique in industrialized nations [1].

US Environmental Agency explains refrigeration and air conditioning systems typically using a refrigerant in a vapor compression cycle as devices for cooling and/or dehumidifying a substance or space, like a refrigerator cabinet, room, office building, or warehouse. Systems of refrigeration and air conditioning are chillers, commercial ice machines, household refrigerators and freezers, ice skating rinks, industrial process air-conditioning units, industrial process refrigeration systems, motor vehicle air conditioning, non-mechanical heat transfer systems, residential and light commercial air conditioning and heat pumps, residential dehumidifiers, refrigerated transport, retail food refrigeration, very low temperature refrigeration and water coolers [2].

An important issue about refrigeration systems is that they are one of the main

sources of energy consumption which is a vital factor in environmental concerns. As dictated by Kyoto Protocol, participating countries are to decrease carbon emissions due to the combustion of hydrocarbon content fuels in the energy plants. This approach has also brought about the necessity building more energy efficient ships. One of the components that consume energy in ships is HVAC (Heating, Ventilating and Air Conditioning refrigeration) systems. By optimizing the condenser outlet temperature as main parameter for compressor speed adjustment, an energy saving potential is calculated to be viable [3].

All liquids boil and condense at distinct temperatures that depend on their pressures. All physical changes occur within the limits of their freezing points and critical temperatures. During the boiling the liquid must absorb the latent heat of evaporation and during the condensing the absorbed latent heat must be rejected again. The basic refrigeration cycle utilizes the boiling and condensing of a refrigerant at different temperatures or in other words, at different pressures. Heat is added in the fluid at the lower temperature and pressure and provides the latent heat to cause it to boil and to change to a vapor. This vapor is then compressed to a higher pressure and a corresponding saturation temperature where its latent heat can be rejected so that it changes back to a liquid. The total refrigeration effect will be the heat transferred to the refrigerant during the boiling i.e. inside the evaporating tank [4].

As explained above, refrigeration is a major application area of thermodynamics. During refrigeration heat is removed from a lower temperature region and transferred to a higher temperature one. Refrigerators, in general, are devices producing cooling effect. Thermodynamic cycles operating on these principles are

called refrigeration cycles. The vapor-compression refrigeration cycle is the most commonly used refrigeration cycle. In a vapor-compression refrigeration cycles, the refrigerant is evaporated and condensed in an alternating nature. It is then compressed in the vapor phase by mechanical means. Reversibility in thermodynamics refers to a characteristic of certain processes that can be reversed, and the system is returned to its initial state, without leaving any net effect. The Carnot Heat Engine is a totally reversible cycle consisting of two reversible isothermal (constant-temperature) and two isentropic (constant-entropy) processes. As stated in Carnot principles, A Carnot heat engine has the maximum thermal efficiency for given temperature limits. Therefore it is taken as a standard to compare the efficiencies of actual power cycles. Due to its reversibility, all four above processes can be reversed, causing also to change the directions of heat and work exchange with the environment. In this case it is called a reversed Carnot cycle. A Refrigerator operating on the reversed Carnot cycle is called a Carnot refrigerator. The Carnot

refrigerator and the reversed Carnot cycle are shown in Figure 1.

The refrigerant absorbs heat (Q_L) at constant-temperature from a source (T_L). It is then compressed at constant-entropy to a higher temperature (T_H) where heat (Q_H) is rejected at again constant-temperature (T_H). From there, expansion occurs at constant-entropy to the initial lower temperature (T_L). The process 3-4 is called condensing where the refrigerant changes from a saturated vapor to a saturated liquid at temperature whereas the process 1-2 is called evaporation where the refrigerant changes from a mixture of liquid-vapor to a saturated vapor at lower temperature. Other two processes are the compression process (2-3) where the mixture is compressed to a saturated vapor at higher temperature and the expansion process (4-1) where the saturated liquid is brought to mixture at a lower temperature, initiating the refrigeration. However, in real life applications, although the processes 1-2 and 3-4 are viable, processes 2-3 and 4-1 are difficult to achieve due to mixtures of liquid and vapor, since compressors

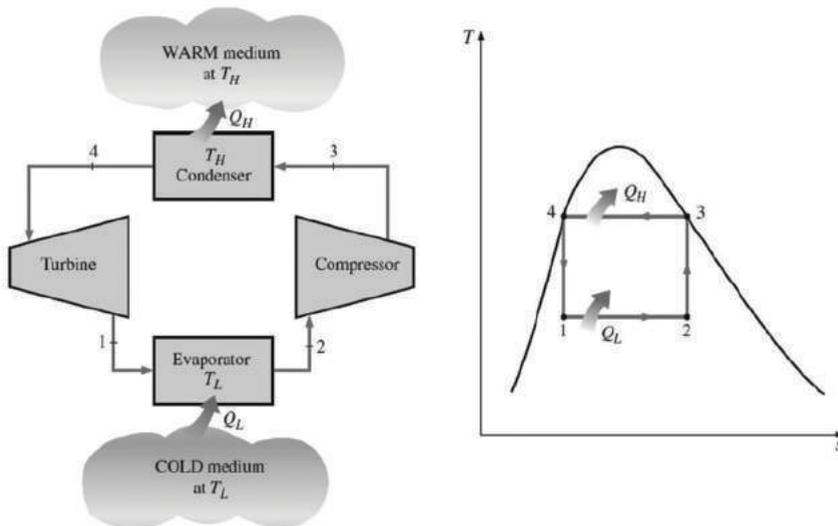


Figure 1. The Carnot Refrigerator and The Reversed Carnot Cycle [5]

and turbines (expanding devices) cannot handle two phase fluids in an efficient way. In order to solve these difficulties, the reversed Carnot cycle could be executed outside the saturation region. However, this would bring up the difficulty in maintaining constant-temperature conditions at the processes 1-2 and 3-4. Although the reversed cycle is a benchmark for comparing the actual cycles, it is therefore not a realistic refrigeration model.

The above mentioned issues against practical use of the reversed cycle can be eliminated by replacing the turbine with a throttling device to reduce the pressure of the liquid, thus lowering the temperature for initiating the cooling effect and by evaporating the mixture completely (i.e. to saturated vapor) before the compression stage. In this case, state 3 is no longer a saturated one, but a superheated state. Therefore process 3-4 takes place in two steps: de-superheating the vapor from a higher temperature to saturation temperature, then changing the vapor phase to liquid at constant-temperature.

The cycle outlined here is called an ideal

vapor-compression refrigeration cycle. The Ideal refrigerator and the Ideal Carnot cycle are shown in Figure 2. As indicated previously, the most common used cycle for refrigeration in general is the vapor-compression refrigeration cycle. The cycle contains four different processes:

Process 1-2: Constant-entropy (isentropic) compression (compressor)

Process 2-3: Constant-pressure (isobaric) heat rejection (condenser)

Process 3-4: Constant-enthalpy (isenthalpic) throttling (e.g. expansion valve)

Process 4-1: Constant-pressure (isobaric) heat absorption (evaporator)

In an ideal cycle, as mentioned previously, the refrigerant at state 1 (compressor inlet) is a saturated vapor. It is then compressed to the condenser pressure in a constant-entropy process. Because compression occurs outside the saturation line (T-s diagram), state 2 is a superheated vapor, therefore attaining a temperature well above the surroundings. This temperature difference causes the heat transfer to take place more easily than it does for the saturated case. Releasing

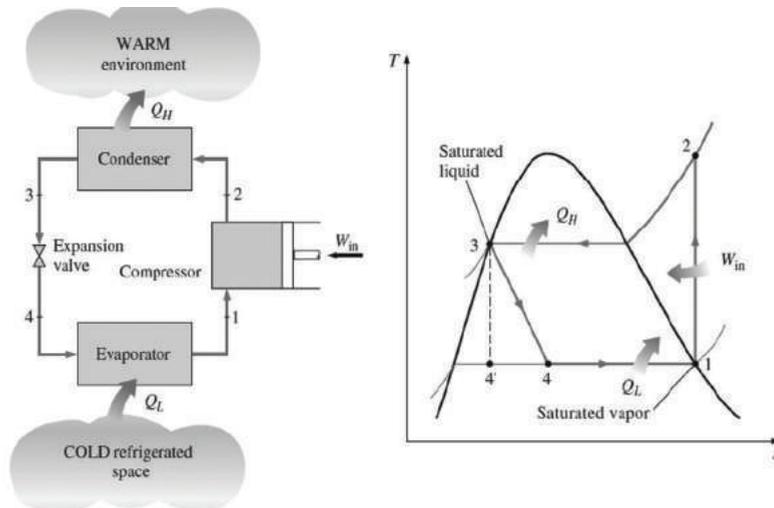


Figure 2. The Ideal Vapor-compression Refrigerator and The Ideal Cycle [5]

the superheat, the rest of the process is same as the reversed Carnot cycle, i.e. the refrigerant changes phase from saturated vapor to saturated liquid at state 3. The intermediate state where the refrigerant is saturated vapor is sometimes called 2'. The higher-pressure liquid refrigerant at state 3 is then throttled causing the liquid to start evaporating easily due to liquid particles sprayed into the lower pressure of evaporator. Meanwhile, temperature of the refrigerant is also reduced to the value corresponding to the saturation pressure. The refrigerant is now a mixture of liquid and vapor phases for which evaporation continues until the refrigerant inside the evaporator undergoes a phase change completely to a saturated vapor. With this evaporation taking place, the latent heat required for the phase change of the refrigerant from a mixture to vapor is absorbed from the refrigerated space, thus achieving the cooling function of the cycle. The cycle is then completed with the refrigerant leaving the evaporator [5].

Refrigerants are the fluids absorbing high amount of heat during evaporation, due to their higher evaporation enthalpies. Refrigerants providing a cooling effect during the evaporation are commonly used in a variety of refrigeration systems, including house hold refrigerators, air conditioners, process cooling, etc. The chemical substances directly responsible for refrigeration as they change phase are called the primary refrigerants which can be classified into the following five main groups: halocarbons, hydrocarbons (HCs), inorganic compounds, azeotropic mixtures, and non-azeotropic mixtures [6].

On the other hand, secondary refrigerants are fluids carrying heat from a substance refrigerated to the evaporator of a refrigeration system. The secondary refrigerants experience a change in temperature when they absorb the heat and liberate it at the evaporator, but they do

not change phase. Most common secondary refrigerants include antifreeze solutions (e.g. ethylene glycol) used in chilled water systems and brine solutions (e.g. calcium chloride) used in industrial applications.

However, heat absorption, or in other words thermodynamic properties is not the only criterion for selecting a refrigerant. Physical and chemical properties such as thermal conductivity and viscosity and environmental impacts such as ozone depletion are equally important [7]. Other criteria for determining the type of the refrigerant is another environmental effect called GWP, i.e. green-house effect, flammability and toxicity. The reason why a working fluid (refrigerant) called R-134a is selected is due its relatively very favorable properties. The primary present concern regarding the refrigerants' influence on the environment is due to the release of chlorine (Cl) in the upper atmosphere. Chlorine gas is believed to react with the ozone (O_3), thus depleting it. Depletion of ozone levels in the upper atmosphere caused by its reaction with chlorine can reduce the screening of harmful ultraviolet rays from the sun. Research has shown that, prolonged exposure to ultraviolet rays may give a rise to health problems such as skin cancers, as well as other adverse effects on living creatures. These refrigerants containing chlorine, thus having the potential of depleting the ozone are called chlorofluorocarbons, or CFCs. The types of CFCs are R-11, R-12, R-113, R-114, and R-115. They are sometimes referred to with the prefix CFC, such as CFC-11, CFC-12, and so on. The United States banned CFC production 1995 and refrigerating devices containing these refrigerants are being replaced or modified for use with less harmful refrigerants. Another group of refrigerants containing chlorine as well as a hydrogen molecule in their molecular chain are less persistent than the CFCs, thus representing a reduced threat to the

ozone layer. These refrigerants are called hydro chlorofluorocarbons or HCFCs. The types of HCFCs are R-22, R-123, R-124, R-141b, and R-142b. Starting on January 1, 2004, the United States restricted HCFC refrigerant production. A worldwide ban is also scheduled by 2030. The refrigerants with no chlorine content are considered to be the least harmful to the ozone layer. They are called the hydro fluorocarbons or HFCs. The types of HFCs are R-125, R-134a, R-143a, and R-152a. However, owing to their global warming potential, there exists some pressure for reduction of their use. As can be understood, no refrigerant is completely innocent and replacing a refrigerant could pose another environmental problem. Besides, differences in physical or thermodynamic properties such as boiling points and refrigerating efficiencies or safety and lubricant and elastomer compatibility could bring up further difficulties. Until now, R-134a has proven to be the best refrigerant as a substitute for R-12 which is widely used in automotive and commercial refrigeration applications. R-123, an HCFC, has been promoted as a candidate to replace R-11, at least temporarily. R11 is a very common refrigerant for systems using centrifugal compressors.

Refrigerant are also classified according to their carbon content: The methane and the ethane series. Except for R-22 (an HCFC) and R-23 (an HFC) from methane series, all others are either flammable or toxic or fully halogenated. R-23 is not widely used as a refrigerant due to its thermodynamic properties. On the other hand, R-22 is the most popular refrigerant in many applications, yet it will undergo future government restrictions in manufacture due to its unfavorable environmental impact, though small. Among the ethane series, R-123, R-124, R-125, and R-134a are the only partially halogenated compounds that are non-toxic and non-flammable.

R-123 is an HCFC and will face regulatory restrictions. The thermodynamic properties of R-124 and R-125 make them of low interest. On the other hand, R-134a has already attracted the most interest for its very favorable properties and it is promising to be one of the more popular future refrigerants [8]. Therefore, R-134a appears to be the best available choice among many refrigerants mentioned above, for a variety of applications.

2. Theory

The first law of Thermodynamics states that the net energy exchange in a cycle is zero which can be stated as $Q_{net} - W_{net} = 0$. It can be seen from Figure 2 that $Q_{net} = Q_H - Q_L$ and $W_{net} = W_{in}$ since the only work involved is compressor work. The performance of any system can be evaluated by comparing the desired output with the required input, i.e. $Performance = (Desired\ output) / (Required\ input)$ which is called the efficiency for a work producing or a work consuming device and is always less than unity. On the other hand, in the case of a refrigeration system it is usually larger than unity and called the Coefficient of Performance and shown as $COP = Q_L / W_{in}$. Since all four components found in a basic refrigeration system are steady flow devices with negligible change in kinetic and potential energy in the working fluid, the resulting heat and work transactions can be calculated using the enthalpy change through the device. It should be noted that the expansion process is isenthalpic where the enthalpy remains unchanged and the compression process is isentropic where the entropy is constant, assuming the compressor is adiabatic and reversible, i.e. it is perfectly insulated against any heat loss to environment and there exists no losses (frictional etc.) within the compressor. Summarizing all:

$Q_L = h_4 - h_1$ with $h_4 = h_3$ (state 4 is mixture of liquid and vapor at evaporator pressure)

$W_{in}=h_2-h_1$ with $s_2=s_1$ (state 1 is saturated vapor at evaporator pressure)

$Q_H=h_2-h_3$ (state 2 is superheated vapor and 3 is saturated liquid at condenser pressure)

All the state properties throughout the calculations are evaluated using curve fitting polynomials obtained from the property data tables belonging to R-134a [9]. Once the property evaluations are made using the evaporator and condenser pressures which are two most important parameters of a compression refrigeration system, the above calculations are performed to obtain Q_L , W_{in} , Q_H and eventually COP . Figures 3 to 15 show temperature dependence of pressure, specific volume, entropy

and enthalpy that are obtained from the property data tables which are used to construct the fitting polynomials upto 5th order depending on R^2 of the fitting curves. A wide range of pressure values, 100-200 kPa for evaporator and 0.8-1.6 MPa for condenser, is covered based on the above graphs, performing a vast number of calculations to obtain COP against P_{evap} for various P_{cond} values (Figure 16).

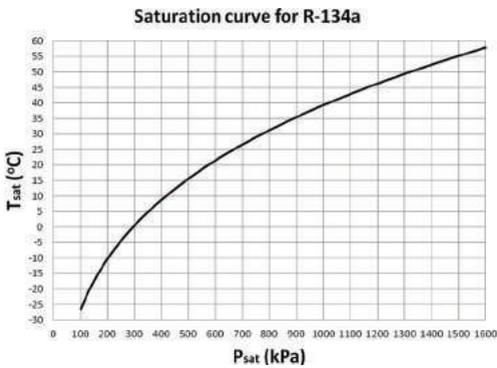


Figure 3. Saturation Temperature vs. Saturation Pressure

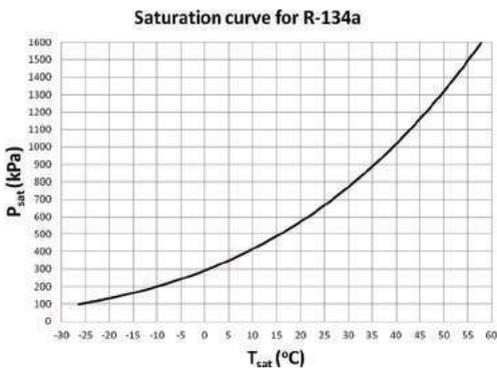


Figure 4. Saturation Pressure vs. Saturation Temperature

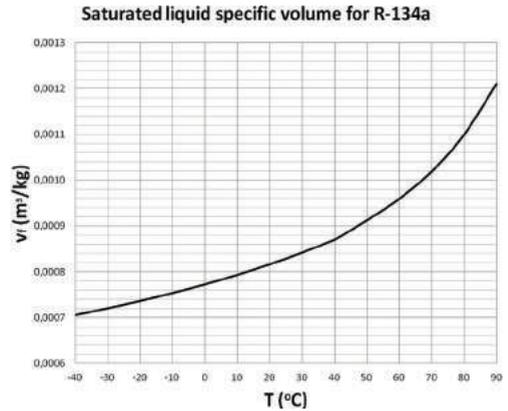


Figure 5. Saturated Liquid Specific Volume vs. Saturation Temperature

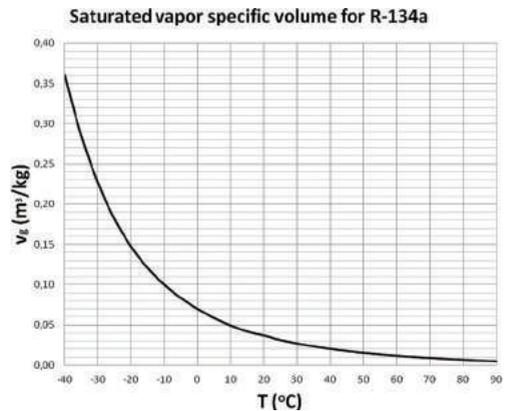


Figure 6. Saturated Vapor Specific Volume vs. Saturation Temperature

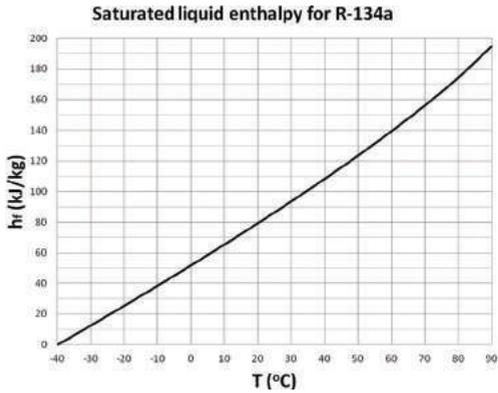


Figure 7. Saturated Liquid Enthalpy vs. Saturation Temperature

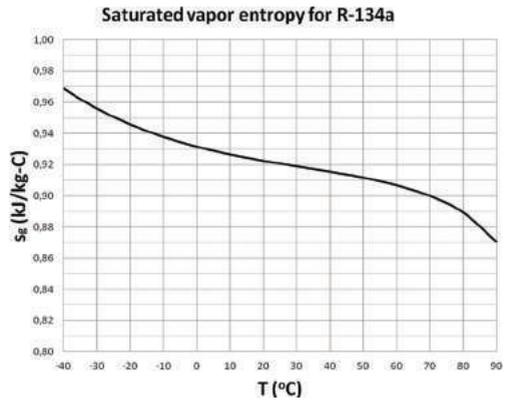


Figure 10. Saturated Vapor Entropy vs. Saturation Temperature

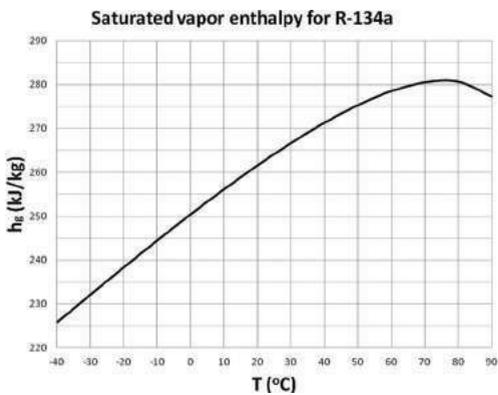


Figure 8. Saturated Vapor Enthalpy vs. Saturation Temperature

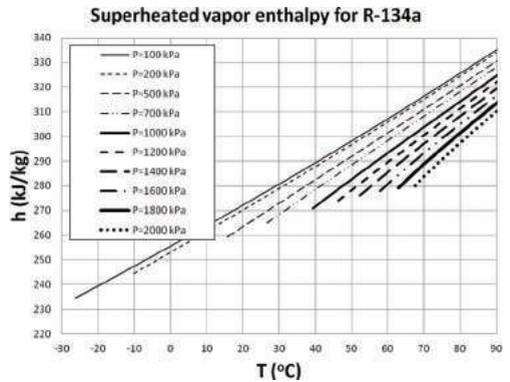


Figure 11. Superheated Vapor Enthalpy vs. Temperature

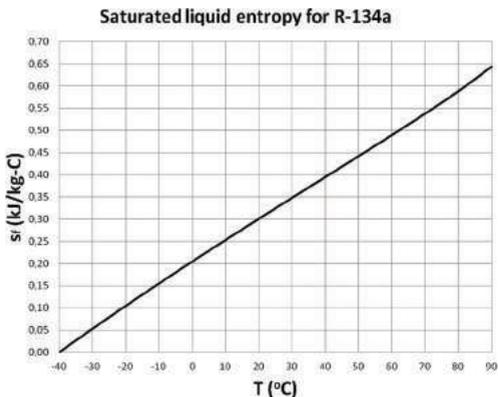


Figure 9. Saturated Liquid Entropy vs. Saturation Temperature

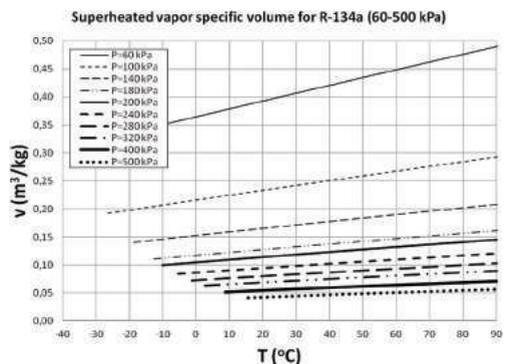


Figure 12. Superheated Vapor Specific Volume vs. Temperature

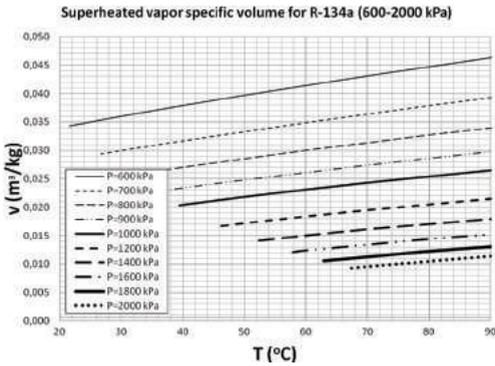


Figure 13. Superheated Vapor Specific Volume vs. Temperature

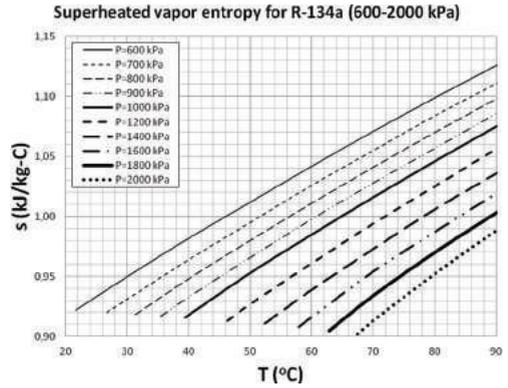


Figure 15. Superheated Vapor Entropy vs. Temperature (600-2000 kPa)

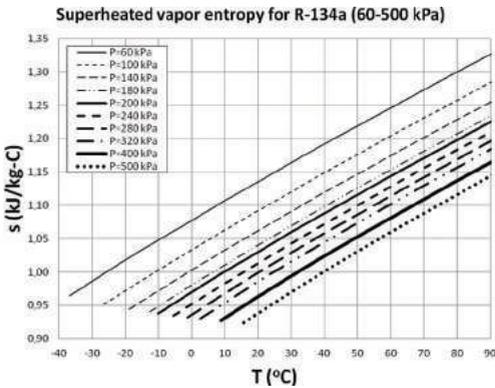


Figure 14. Superheated Vapor Entropy vs. Temperature (60-500 kPa)

3. Results and Discussion

The results obtained herein are obtained for an ideal compression refrigeration system using R-134a as refrigerant. The results are readily usable especially by designers and engineers aiming to perform quick calculations for such systems. If any data is available regarding the level of irreversibility of the compressor, i.e. the isentropic efficiency, η , a more realistic (actual) COP can be obtained by simply multiplying the ideal COP by efficiency, i.e. $COP_{actual} = \eta COP_{ideal}$. Just to have an insight of how practically the results can be obtained,

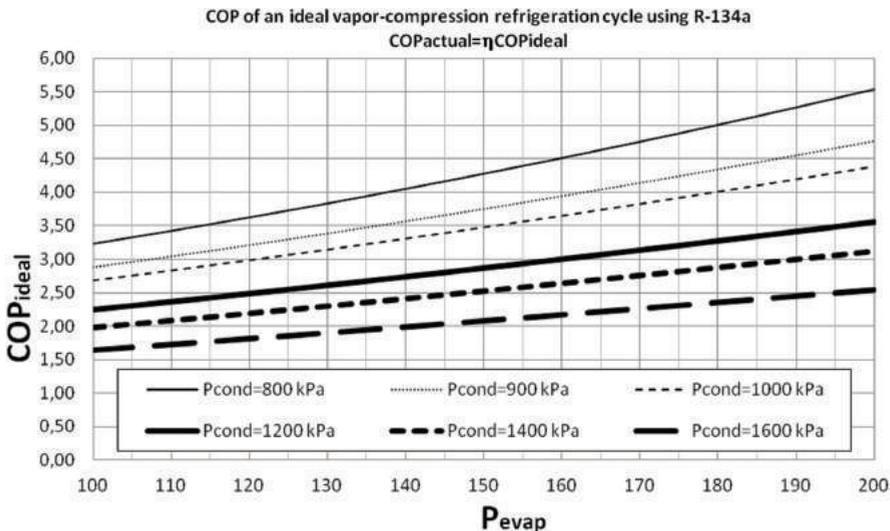


Figure 16. COP of an Ideal Vapor-compression Refrigeration Cycle Using R-134a

an example is included herein:

Example: A refrigerator of 5 tons of refrigerant is to be designed with a freezer temperature of -10°C where the heat is absorbed and an environment temperature of 20°C where the heat is rejected to. COP of the refrigeration system and the power required for the compressor are to be determined.

Solution: As a rule of the thumb, evaporation and condensation temperatures can be approximated to have 15°C difference with medium (air) where heat is absorbed from and rejected to [10]. In this case, using Figure 4, the freezer temperature of -10°C corresponds to -25°C of evaporator temperature whose saturation pressure is 100 kPa and an environment temperature of 20°C corresponds to 35°C of condenser temperature whose saturation pressure is 900 kPa. Then, using Figure 16, $COP_{ideal} = 2.85$ is obtained. If the compressor isentropic efficiency is known to be 85%, then:

$COP_{actual} = 2.42$ is calculated. 5 tons is equal to 17.6 kW of refrigeration which requires a compressor power of 9.7 hp.

One may wish to go through the following details which is still much easier than reading out the property values from the thermodynamic properties tables and interpolating between them:

$T_1 = -25^{\circ}\text{C}$, $h_1 = 235 \text{ kJ/kg}$ (Figure 8) and $s_1 = 0.95 \text{ kJ/kg}\cdot^{\circ}\text{C}$ (Figure 10).

$s_2 = s_1 = 0.95 \text{ kJ/kg}\cdot^{\circ}\text{C}$ and $P_2 = 900 \text{ kPa}$, $T_2 = 46^{\circ}\text{C}$ (Figure 15) and $h_2 = 282 \text{ kJ/kg}$ (Figure 11).

$T_3 = 35^{\circ}\text{C}$, $h_3 = 100 \text{ kJ/kg}$ (Figure 7) and $h_4 = h_3 = 100 \text{ kJ/kg}$

$W_{in} = h_2 - h_1 = 47 \text{ kJ/kg}$ and $Q_L = h_1 - h_3 = 135 \text{ kJ/kg}$, then

$COP_{ideal} = Q_L / W_{in} = 2.87$ and $COP_{actual} = 2.44$ which is practically same as the above value.

4. Conclusion

In this theoretical work, an ideal compression refrigeration cycle is studied. Tabulated properties data of R-134a is

first presented as graphs of properties vs. temperature or pressure, both for saturated and superheated states. Those who do not want to interpolate between table entries which may be very time consuming or those who do not prefer using P-h diagrams in which a significant amount of loss of precision may occur, may find using these graphs practical. A graph of COP vs. evaporator pressure for various condenser pressures is presented. This graph is considered to be a practical tool for refrigeration systems designers or engineering students who aim to get quick results for performance and power requirements of a compression refrigeration cycle using R-134a. It is also very practical for Marine Engineers on board the ships who must take quick decisions when it comes to analyze and remedy a problem within refrigeration systems. Although a specific refrigerant, R-134a, has been chosen for this paper, similar calculators can be created in the future as refrigeration industry comes up with newer refrigerants.

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