Second-Law-Based Thermodynamic Analysis of Vapor-Compression Refrigeration Cycles

by

Shamsul Hoda Khan

A Thesis Presented to the

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DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the Requirements for the Degree of

MASTER OF SCIENCE

In

MECHANICAL ENGINEERING

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Second-law-based thermodynamic analysis of vapor-compression refrigeration cycles

Khan, Shamsul Hoda, M.S.

King Fahd University of Petroleum and Minerals (Saudi Arabia), 1992
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This thesis, written by

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MASTER OF SCIENCE IN MECHANICAL ENGINEERING

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Date: 23-2-92
In the name of God, Most Gracious, Most Merciful
This thesis is dedicated to

My parents, brothers, wife and son

for the sacrifices they made to educate me
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THESIS ABSTRACT

NAME OF STUDENT : SHAMSUL HODA KHAN
TITLE OF STUDY : Second law-based thermodynamic analysis of vapor-compression refrigeration cycle.
MAJOR FIELD : Mechanical Engineering

Thermodynamic analysis of vapor-compression refrigeration cycles are investigated by both the first law and second law of thermodynamics. Second law analysis (in terms of energy measure of irreversible loss which is typically described as irreversibility) is carried out for both basic as well as two-stage vapor-compression refrigeration cycles. This analysis is performed on each of the system components to determine their individual contribution to the overall system losses. It is found that most of the irreversibility is in the compressor due to low compressor efficiency. Irreversibilities of the expansion valve as well as condenser are also significant. These irreversible losses increases with the increase of condenser and evaporator temperature differences.

As the refrigerant, R-12 is the major contributor to the ozone depletion, therefore in this thesis a new refrigerant R-134a is also investigated, which is considered as a leading candidate to replace R-12. Computer subroutines are modified to calculate the properties of the refrigerant R-134a. A comparative study is also done between these two refrigerants which indicate that R-12 can be replaced by R-134a, without any significant loss in overall system performance.

The staging of compressors of a refrigerating system produces a saving in work. One of the design parameters of the two-stage refrigeration system is the selection of interstage pressure. In this study, it is found that the optimum interstage pressure for a two-stage, vapor-compression refrigeration cycle is very close to the saturation pressure corresponding to the arithmetic mean of the condensing and evaporating temperatures. These results are compared with the existing practice.

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عنوان الرسالة: تحليل ترمودينامي حراري قائم على القانون الثاني لحودرة تبريد تعمل بضغط البخار
اسم الطالب: شهاب الهدي خان
التخصص: الهندسة الإلكترونية
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تم استقصاء تحليل ترمو دينامي حراري لدورات التبريد التي تعمل (العامل) لضغط البخار باستخدام كل من القانون الأول والثاني للترموديناميكا الحرارية. إن التحليل القائم على القانون الثاني (باستخدام مقياس الطاقة للنقد الفيبرأ قابل للانكماش والذي يصور غطيا بالانكماش) قد تم انتهجه لكل من دورات التبريد العاملة لضغط البخار الأساسية وثنائية المراغ، وقد تم قسم هذا التحليل لكل جزء في الدورة لتحديد مشاركة كل جزء مستغرقا في الفاقد الكلي من الدورة، وقد وجد أن الجزء الأعظم من الانكماش يوجد في "الضغط" نظراً للفاقد الضغط المتفاوتة، إن الانكماش في مسما التحد ونذل في المكتف.

إذا هذا الفاقد الانكماش يزيد مع زيادة فرق درجات الحرارة خلال كل من المكثف والبخار، نظرلاً لأن وسط التبريد (R-12) يعتبر هو العامل الرئيسي في (R-134a) تأكل طبقات الأوزون فقط تم خلال هذا البحث دراسة وسط التبريد الجديد وهو مرشح رئيسي كديل لوسط التبريد (R-12) وقد تم تطبيق برامج كمبيوتر فرعى لحساب خواص وسط التبريد الجديد (R-134a). وقد تم عمل دراسة مقارنة بين وسطي التبريد هذين وقد أوضحنا دراسة واسعة أنه يمكن استبدال وسط التبريد (R-12) بوسط التبريد الجديد (R-134a) لفأقد محسوس في الإدماج الكلي للدورة (النظام الحراري).

إذا تقسيم الضغط في نظم التبريد إلى مراحل يوفر في الطاقة، وبعد اختيار قيمة الضغط بين مرحلتي الضغط في الدورات ثنائية المراغ أحد متغيرات التصميم، وقد وجد خلال هذه الدراسة أن القصة المثلى لهذا الضغط قريبة جداً من ضغط التشبع المقابل للضغط الحسابي لدرجتي حرارة المكثف والبخار، وقد وجد أن نتائج هذا البحث يمكن مقارنتها بما هو موجود عملياً.

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يناير 1992 م
CHAPTER 1

INTRODUCTION

In thermodynamic analysis, we apply the principle of conservation of mass, conservation of energy (first-law of thermodynamics) and the second-law of thermodynamics. In the first-law analysis of vapor-compression refrigeration cycles, the main purpose is to determine, from the specific enthalpies of the refrigerant at the various state points in the cycle, the mass flow of the refrigerant, the heat flow and the flow of rotary shaft work for each piece of equipment. As a criterion of performance, the coefficient of performance of the cycle is calculated. The scope of the first-law analysis for the basic vapor-compression refrigeration cycle can be summarized by the following equations, which exemplify the concepts of the conservation of mass and energy [2].

Mass flow: \[ m = \frac{Q_{\text{evap}}}{\Delta h_{\text{evap}}} \] (1.1)

Heat rejection: \[ Q_{\text{cond}} = m(\Delta h)_{\text{cond}} \] (1.2)

Compressor Work: \[ W_{\text{comp}} = m\Delta h_{\text{comp}} \] (1.3)

Energy balance: \[ Q_{\text{evap}} + W_{\text{comp}} = Q_{\text{cond}} \] (1.4)

Coefficient of Performance: \[ COP = \frac{Q_{\text{evap}}}{W_{\text{comp}}} \] (1.5)
It should be noted that the first-law of thermodynamics does not distinguish between the heat and work; and thus cannot be used on its own to identify the sources of thermodynamic losses in a thermodynamic cycle. On the other hand, second-law of thermodynamics may be used in identifying and quantifying thermodynamic losses in any thermodynamic cycle.

In analyzing vapor-compression refrigeration cycles, the concepts of the reversible process is basic to the application of the second law of thermodynamics. From the basic concepts, the performance of a thermal system may be expressed in terms of component irreversibility rates, which provides a convenient method for optimizing the cycle. The irreversible loss in any thermodynamic process can be defined as the difference between the reversible work and the actual work.

With the concept of irreversibility, we can calculate the thermodynamic losses in vapor-compression refrigeration cycles. The principle processes involved in this cycle are; compression, throttling and heat transfer. The analysis start with a recognition of the individual system irreversibilities; namely, compressor, condenser, expansion valve and evaporator. The analysis illustrate the effect of the departure of a real cycle from the reversed Carnot cycle. The refrigeration cycles which are analyzed in this thesis are defined in terms of temperature and pressure of the refrigerant at various state points in the cycle, as well as the temperature of the refrigeration load and the environment. Analyses are given for both the basic vapor-compression cycle and for a two-stage vapor-compression refrigeration cycle with different type of arrangements.
The staging of compressors of a refrigerating system produces a saving in work. One of the design parameters of the two-stage refrigeration system is the selection of interstage pressure, for which the total work is minimum. In the compression of a perfect gas, the optimum interstage pressure is the pressure corresponding to the geometric mean of the suction and the discharge pressures. However, for real gases the intermediate pressures are usually different from the geometric mean value [5].

Almost all applications of refrigerating compressors require some kind of capacity control. The need for refrigeration capacity control arises from the fact that the system capacity must match the load under a variety of operating conditions. The capacity can be controlled by several methods. The most common are intermittent running, variable speed, hot-gas bypass and suction valve unloading. It should be noted that almost all methods of capacity control do not offer a linear decrease in power consumption with the refrigeration capacity. The identification and quantification of thermodynamic losses in capacity control methods would be useful for design and analysis of such systems.
Overall Objectives Of The Study

Thermodynamic analysis of vapor-compression refrigeration cycles are studied with the following objectives:

1. *Simple vapor-compression cycle*: To demonstrate the significance of second-law-based analysis by considering different refrigerants and including compressor efficiencies, as well as the effect of superheating and subcooling.

2. *Two-stage vapor-compression refrigeration cycle*: To find the optimum interstage saturation pressure (or temperature) for various two-stage refrigeration / air-conditioning applications by considering the effect of various parameters, such as, i) compressor efficiency, ii) effect of superheating, and iii) subcooling.

3. *Capacity control*: To demonstrate thermodynamic losses in a hot gas bypass refrigeration / air-conditioning capacity control scheme by using second-law-based thermodynamic analysis.

To implement these studies, the thermodynamic properties of the refrigerants are needed. A computer program written by Karstounes and Erth [23] is used to calculate the thermodynamic properties of R-12, R-22 and R-502. However, a considerable interest has developed over the last decade to replace chlorofluorocarbon refrigerants, which contribute to the depletion of ozone in the stratosphere. For this reason a refrigerant R-134a is considered in this analysis, which is a safe alternative for R-12 [25]. The existing computer program is also modified to include the thermodynamic properties of the refrigerant R-134a.
CHAPTER 2

LITERATURE REVIEW

In 1953 Threlkeld [1] described a thermodynamic study of vapor-compression heat pump cycle. In his work thermodynamic analysis is carried out for both heat pump operating with a constant temperature heat source and sink as well as for constant temperature heat source and variable temperature heat sink. The ratio of coefficient of performance of the cycle to the coefficient of performance of the reversible Carnot heat pump is used in his work to define the heat pump cycle efficiency. It has been shown that R-11 yields the highest theoretical efficiency as compared to R-12, R-21, R-113, R-114 and Ammonia cycles.

Wilson [2] demonstrated second law analysis of vapor compression refrigeration systems with the help of numerical examples. In his example problems he has shown that:

1. The reversed Carnot cycle, having heat transfer through negligible differences, is an ideal standard and is the criterion for minimum requirements of work input to the compression refrigeration cycle.

2. Departures from this ideal standard, such as fluid friction and heat transfer through finite temperature difference, will create entropy, degrade available energy and cause an increase in entropy of the universe.
3. Such departures require that the work input to the refrigeration cycle be greater than the ideal minimum.

4. This increased work input is equal to the product of the absolute temperature of the atmosphere and the increase in entropy of the universe, \( T_d\Delta S_{\text{m}} \), and is also equal to the sum of all the available energies that are degraded by irreversibilities produced by the refrigerant in its cyclical operation and heat transfer through finite temperature differences.

Thermodynamic analysis of vapor-compression refrigeration systems have been presented by Swers et al [3], which is also discussed in ASHRAE Handbook of Fundamentals [4]. The analysis illustrate the effect of the departure of a real cycle from the reversed Carnot cycle in terms of irreversible losses. In these references, the effects of irreversibilities in a vapor compression refrigeration cycle are compiled for a range of operating conditions, and the irreversible losses of each component are optimized. Analysis are described for both the basic vapor-compression refrigeration cycle and for a two-stage, vapor-compression refrigeration cycles. But for two-stage vapor compression, the interstage pressure is considered as the geometric mean pressure, which is true for the perfect gas with complete intercooling.

According to Soumerai [6], the selection of the compressors and other components of a two stage refrigeration system can be simplified when several refrigerant properties are grouped together and expressed in terms of the intermediate pressure, which can be selected arbitrarily. It is then possible to design a system that will balance at this assumed intermediate pressure by properly selecting the compressor displacement ratio. He also demonstrated the
effect of compressor displacement ratios and volumetric efficiencies on the intermediate pressure of the cycle.

For multistage refrigeration systems, many research workers have investigated the optimum interstage pressure, for which the work required is minimum. According to Keshwani [7], the optimum interstage pressure corresponding to minimum work is given by

\[ P_{\text{int}} = (1 + a)^{\frac{1}{n}} (P_{\text{evap}} \cdot P_{\text{cond}})^{1/2} \] (2.1)

It is assumed that the mass of flashed vapor is invariant with intermediate pressure, however, for real applications the mass of flashed vapor is directly related to the intermediate pressure.

This investigation was extended by Lal [8] after allowing for variation in mass of the flashed vapor with intermediate pressure. His analysis is complicated and it requires generation of whole superheated refrigerant properties. To simplify Lal's work, Charan and Verma [9] assumed linear variation in the mass of flashed vapor with interstage pressure and finally found that the optimum interstage pressure corresponding to the minimum work, however, their expression is in terms of a set of differential equations, for which a closed form solution is not possible.

The discrete maximum principle discussed by Katz [10] has been used by Arora and Dhar [11] to solve the problem of optimum intermediate pressure allocation in multistage compression systems. They first derived the expression
for the interstage pressure with intercooling between the stages. Their result shows that the optimum interstage pressure lies very close to the geometric mean pressure value, however, when the flash intercooler is incorporated, they found a considerable difference between the geometric mean pressure and the optimal pressure. It should be emphasized that their equations are complicated and cannot be solved, explicitly.

Auracher [12-13] illustrated the application of exergy (or available energy) to refrigeration process optimization. According to him, it is exergy rather than energy which is of concern in all thermodynamic processes. The exergy losses are described for various components of a refrigeration cycle and the need for introducing the second law efficiency of refrigeration systems is highlighted, which is similar to that proposed earlier by Threlkeld [11]. He discussed that the optimization of a plant design can best be guided by determination of the sources and magnitude of its exergy losses, with an example problem.

Ait-Ali [14] and Ait-Ali and Wilde [15], discussed that the irreversible work of heat transfer due to a finite-temperature approach represents a major fraction of the actual compressor work in low-temperature refrigeration cycles. They showed that this work is monotonic decreasing with the number of stages in constant boiling temperatures cascades. For a finite number of stages, the minimum compressor work is obtained when every refrigerant inter-stage temperature is the geometric mean of the two adjacent ones. This optimality rule when applied to a classical cascade natural gas liquefaction cycle (which comprises of a three-stage methane subcooling cycle, a three-stage ethylene liquefaction cycle and a two stage pre-cooling cycle) leads to a 7% saving in
refrigeration work as compared to the non-optimum design.

Prasad [16-17] also found that the interstage temperature of a two-stage refrigeration cycle is given by the geometric mean of the condensing and evaporating temperatures, which can be expressed as

\[ T_{int} = (T_{cond} \times T_{evap})^{1/2}. \]  

In the above equation all the temperatures are expressed in absolute units. It should be noted that his method uses approximate technique by ignoring superheat horn in comparison to the total work of compression. Prasad, also indicated that the above assumption is only true for R-12 system because the superheat horn for R-12 is negligible as compared to the total work of compression.

Behringer [18], found that, for two stage Ammonia cycle with liquid subcooling and desuperheating between the stages to saturation temperature, the optimum intermediate temperature is given by

\[ T_{opt} = T_m + 5K, \]  

where \( T_m \) is the saturation temperature in \( ^\circ K \) corresponding to the pressure for equal pressure ratios in the first and second stage.

Khalid [19], discovered that the optimum interstage pressure for two stage vapor compression refrigeration cycle is very close to the arithmetic mean of the condensing and evaporating temperature. That is;
\begin{equation}
T_{\text{int}} = \frac{T_{\text{cond}} + T_{\text{evap}}}{2}.
\end{equation}

He has shown that the maximum coefficient of performance and minimum irreversibility of the system occurs very much near the arithmetic mean temperature. It is also shown that the optimum performance of the refrigeration cycle with mechanical subcooling occurs when the subcooler compressor (saturation) temperature corresponds to the arithmetic mean of the condensing and evaporating temperatures. In his analysis compressor is considered as a reversible and adiabatic as well as the effect of superheating is neglected.

Zubair and Bahel [20] discussed various capacity control methods. They made a comparative study of various compressor capacity control schemes and plotted the percentage of full load power against the system capacity. Haseltine [21] presented a comparison of power and efficiency of a compressor using different capacity control methods. It should, however, be noted that the quantification of thermodynamic losses in various capacity control methods are not discussed in the above references.
CHAPTER 3

THERMODYNAMIC FUNDAMENTALS

In thermodynamic analysis, the principle of conservation of mass, conservation of energy (first law of thermodynamics) and the second law of thermodynamics are applied. In this chapter, these conservation laws are discussed.

3.1 First Law of Thermodynamics

The principle which states that energy is conserved for any thermodynamic system (either open or closed) is defined as the first law of thermodynamics. It should be noted that for the usual thermal systems analysis and design, the principle of conservation of mass is applied along with the conservation of energy. These conservation laws in the notation of Prof. London [22] are expressed as:

\[
\text{Rate of creation of (mass)} = 0, \quad (3.1)
\]

\[
\text{Rate of creation of (energy)} = 0, \quad (3.2)
\]

where the rate of creation of (something) within the thermodynamic system is defined as:
Rate of creation of (something) = Rate of outflow ( ) - Rate of inflow ( )
+ Rate of increase of storage ( ). (3.3)

When applying these principles to a stationary control volume, the above equations can be written as [26]:

\[
\frac{d}{dt} \int_{V} \rho dV + \int_{A} \rho \bar{V} \cdot dA = 0
\]  (3.4)

and

\[
\frac{d}{dt} \int_{V} \rho c dV + \int_{A} (h + \frac{V^2}{2} + gz) \rho \bar{V} \cdot dA + \bar{W}_{c} - \bar{Q}_{c} = 0
\]  (3.5)

In the analysis and design of thermal systems, it is generally assumed that the thermodynamic state and fluid velocity are uniform over the flow area, at any instant of time. As a result of this assumption, Eqs. (3.4) and (3.5) become

\[
\frac{dm_{ex}}{dt} + \Sigma m_{x} - \Sigma m_{l} = 0
\]  (3.6)

and

\[
\frac{dE_{ex}}{dt} + \Sigma m_{x} (h_{x} + \frac{V_{x}^2}{2} + gz) + \bar{W}_{c} - \bar{Q}_{c} - \Sigma m_{l} (h_{l} + \frac{V_{l}^2}{2} + gz) = 0
\]  (3.7)

It should be noted that the refrigeration/air-conditioning systems are designed for long-term steady operation, so it can be modeled as steady-state, steady-flow (SSSF) devices. The following assumptions are generally applied for
a SSSF process [27-28]:

(i) The state of the mass at each point in the control volume does not vary with time.

(ii) The state of mass at each discrete area of flow on the control surface do not vary with time. The rate at which heat and work cross the control surface remains constant.

These assumptions lead to

\[
\frac{dm_{\text{ex}}}{dt} = 0
\]

\[
\frac{dE_{\text{ex}}}{dt} = 0.
\]

Thus, the conservation of mass equation (continuity equation) and the conservation of energy equation (first law of thermodynamics) can be written in the form

\[
\Sigma \dot{m}_e - \Sigma \dot{m}_e = 0
\]

(3.8)

and

\[
\Sigma \dot{m}_e (h_e + \frac{V^2}{2} + gz_e) + \dot{W}_{\text{ex}} - \dot{Q}_{\text{ex}} = \Sigma \dot{m}_e (h_i + \frac{V^2}{2} + gz_i)
\]

(3.9)

For example, if there is only one refrigerant entering and leaving the control volume, we can simplify the above equations as

\[
\dot{m}_i = \dot{m}_e = \dot{m}
\]

(3.10)
and
\[ \dot{m}(h_c + \frac{v_e^2}{2} + gz_e) + \dot{W}_{ce} - \dot{Q}_{ce} - \dot{m}(h_c + \frac{v_c^2}{2} + gz_c) = 0. \] (3.11)

When neglecting the effect of kinetic and potential energies Eq. (3.11) reduces to
\[ \dot{m}h_c + \dot{W}_{ce} - \dot{Q}_{ce} - \dot{m}h_c = 0. \] (3.12)

It should be noted that the first law of thermodynamics discussed above does not distinguish between heat and work; and thus can not be used on its own to identify the sources of loss in a thermodynamic cycle. The first law only tells us that there must be a balance of energy for a given process, i.e., energy in is equal to energy out. Thus, the energy balance analysis is incapable of pinpointing the location of the components and the processes which causes the thermodynamic losses. Second law analysis of each of the processes will locate the losses incurred and the true effectiveness of the process.

3.2 Second Law of Thermodynamics

Second law puts a limitation on the conservation of some form of energy to other. The second law of thermodynamics defines an extensive property of matter called "entropy" which can be expressed as
\[ dS \geq \frac{\delta Q}{T} \] (3.13)
The equal to sign in the above equation belongs to the idealized system where all processes are internally reversible. The inequality sign belongs to the real life system, where internal irreversible losses may be minimized but never reduced to zero. An operationally more convenient formulation of Eq. (3.13) for any thermodynamics system (either open or closed) may be expressed in the notation of Eq. (3.3) as :

\[
\text{Rate of creation of entropy} \geq \sum \frac{\dot{Q}}{T} \text{int} - \sum \frac{\dot{Q}}{T} \text{out} 
\]  

(3.14)

In this formulation \((\frac{\dot{Q}}{T})\text{int}\) terms are for the heat transfer rates into the system from the surroundings, where \(T\) is the absolute temperature just within the system boundaries in the region where the individual \(\dot{Q}_\text{int}\) terms are received. In contrast, the \((\frac{\dot{Q}}{T})\text{out}\) terms are heat transfer rates out of the system to the surroundings, with \(T\) is again the absolute temperature just within the system boundaries in the region where the individual \(\dot{Q}_\text{out}\) terms are being delivered to the surroundings [22].

With the above framework for the second-law, a natural entropy measure of the internal irreversible losses is provided by the strength of the inequality sign. This leads to the definition of entropy generation (or the entropy measure of irreversible loss) for any thermodynamic process as

\[
\dot{S} = \text{Rate of creation of entropy} - [\sum \frac{\dot{Q}}{T} \text{int} - \sum \frac{\dot{Q}}{T} \text{out}] 
\]  

(3.15)

where
\[ \hat{S} \geq 0. \]

When applying the second-law of thermodynamics to a control volume,

\[ \dot{S} = \frac{d}{dt} \int_{\varepsilon_r} p_s dV + \int_{\varepsilon_s} p_v \tilde{V} dA - \int_{\varepsilon_t} \frac{Q_{\varepsilon_t}}{T} dA \quad (3.16) \]

Using the assumptions listed earlier in the development of Eqs. (3.6) and (3.7), the above equation may be written as:

\[ \dot{S} = \frac{dS_{\varepsilon}}{dt} + \sum \dot{m}_{s_s} - \sum \dot{m}_{s_i} - \sum \left( \frac{Q_{\varepsilon}}{T} \right) \geq 0, \quad (3.17) \]

or

\[ \dot{S} = \frac{dS_{\varepsilon}}{dt} + \frac{dS_{\text{net}}}{dt} \geq 0, \quad (3.18) \]

where

\[ \frac{dS_{\text{net}}}{dt} = \sum \dot{m}_{s_s} - \sum \dot{m}_{s_i} - \sum \left( \frac{Q_{\varepsilon}}{T} \right) \quad (3.19) \]

For steady-state, steady flow process with only one refrigerant entering and leaving the control volume,

\[ \frac{dS_{\varepsilon}}{dt} = 0 \]

and Eq. (3.17) reduces to

\[ \dot{S} = \dot{m}(s_s - s_i) - \sum \left( \frac{Q_{\varepsilon}}{T} \right) \geq 0, \quad (3.20) \]

where the equality sign applies to an internally reversible process and the
inequality to internally irreversible processes.

3.3 Irreversibility

A measure of an internal irreversible loss, particularly of interest to the system analyst in order to make decisions, is defined as an energy measure of irreversible loss, which is typically described as the irreversibility. It is defined relative to the entropy generation as [22]

\[ \dot{I} = T_{WF}(\dot{S}) \]  \hspace{1cm} (3.21)

where \( T_{WF} \) is a temperature weighting factor to be specified by the analyst based on a judgement of its relevance to the system being analyzed. In other word, the irreversibility of a process is defined as the difference between the actual work input and the reversible work input.

For steady-state, steady flow process with several refrigerant streams entering and leaving the control volume, we can express Eq.(3.21) as

\[ \dot{I} = T_{WF} \left[ \Sigma \dot{m}_e s_e - \Sigma \dot{m}_i s_i - \frac{Q}{T} \right] \]  \hspace{1cm} (3.22)

It may be noted that irreversibilities are a measure of the effectiveness of any process, e.g., for a reversible process, the irreversibility is equal to zero. The optimization of a plant design can best be guided by the determination of the source and magnitudes of its irreversibilities, which is a direct consequence of the second law of thermodynamics.
CHAPTER 4

THERMODYNAMIC ANALYSIS OF BASIC VAPOR-COMPRESSION REFRIGERATION CYCLES

The preceding chapter introduced basic thermodynamic principles for system analysis. Application of these principles to the basic vapor-compression refrigeration cycle is now considered. Two cases have been selected for the illustration of thermodynamic analysis. The first case is referred to as an ideal vapor-compression refrigeration cycle. The second case is referred to as an actual vapor-compression cycle, which includes the effect of super-heating, subcooling, compressor efficiency and pressure drop in the condenser and evaporator. The overall analysis presented below uses the first law of thermodynamics to determine how well the system performs and the second law of thermodynamics to determine how poorly the system components perform [33].

4.1 An Ideal Vapor-Compression Cycle

The equipment diagram for the ideal vapor-compression cycle is illustrated in Fig. 4.1, while log(p)-h and T-s diagrams are shown in Fig. 4.2-4.3, respectively. The major components of this cycle include compressor, condenser,
Fig. 4.1 Equipment diagram for a basic vapor compression cycle.
Fig. 4.2 Pressure-enthalpy diagram for an ideal vapor-compression cycle.

Fig. 4.3 Temperature-entropy diagram for an ideal vapor-compression cycle.
expansion valve and evaporator. The ideal cycle considers heat transfer in the condenser and evaporator without pressure losses, a reversible adiabatic (isentropic) compressor and an adiabatic expansion valve, connected through piping that has neither pressure loss nor heat transfer with the surroundings. The refrigerant leaves the evaporator at point 1 as a low pressure, low temperature, saturated vapor and enters the compressor where it is compressed reversibly and adiabatically (or isentropically). At point 2, it leaves the compressor as a high temperature, high pressure, superheated vapor and enters the condenser. In the condenser, it is first desuperheated and then condensed at a constant pressure. At point 3, the refrigerant leaves the condenser as a high pressure, medium temperature, saturated liquid and enters the expansion valve, where it expands irreversibly and adiabatically (constant enthalpy). At point 4, it leaves the expansion valve as a low-quality vapor and enters the evaporator, where it is evaporated reversibly at constant pressure to the saturation state at point 1.

4.1.1 First-Law-Based Thermodynamic Analysis

In the first-law-based thermodynamic analysis an energy balance is performed on system components. Applying the steady-state, steady-flow equation with negligible kinetic and potential energy terms for the first law of thermodynamics (Eq. (3.12)) to each of the components of the basic vapor-compression cycle the following relationships are derived [34-36].
1-2 compression work

\[ \dot{W}_{\text{comp}} = \dot{m} (h_1 - h_2) \]  \hspace{1cm} (4.1)

2-3 Heat rejection in the condenser

\[ \dot{Q}_{\text{cond}} = \dot{m} (h_3 - h_4) \]  \hspace{1cm} (4.2)

3-4 Expansion process

\[ h_3 = h_4 \]  \hspace{1cm} (4.3)

4-1 Evaporator load

\[ \dot{Q}_{\text{evap}} = \dot{m} (h_1 - h_4) \]  \hspace{1cm} (4.4)

The evaporator load is generally called as the refrigerating effect or refrigerating capacity. If we know the refrigerating capacity of a plant, then we can calculate the mass flow rate as

\[ \dot{m} = \frac{\dot{Q}_{\text{evap}}}{(h_1 - h_4)}. \]  \hspace{1cm} (4.5)

Considering the refrigerating capacity as one ton, which is defined by the American Society of Refrigeration Engineers [18] as:

1 ton of Refrigeration = 200 Btu / min,

then the mass flow rate for 1-ton refrigeration capacity is

\[ \dot{m} = \frac{200}{(h_1 - h_4)} \]

It should be noted that the performance of a refrigeration system is generally defined by a dimensionless parameter known as the coefficient of performance (COP) as
\[ \text{COP} = \frac{\text{Refrigerating effect}}{\text{Net Work input}}. \]

For an ideal vapor-compression cycle, it may be expressed in terms of enthalpies by using Eqs. (4.1) and (4.4) as

\[ \text{COP} = \frac{h_1 - h_4}{h_2 - h_1} \] (4.6)

4.1.2 Second-Law-Based Thermodynamic Analysis

In analyzing the vapor-compression refrigeration cycle, the concept of reversible process is basic to the application of the second law of thermodynamics. From this basic concept, the performance of a process or system can be expressed in terms of irreversibility (or irreversible losses). With the concept of irreversibility, we can calculate the losses in each of the system component of the ideal vapor-compression refrigeration cycle.

**Compressor**: The compression process is assumed to be reversible and adiabatic. That is;

\[ \frac{\dot{Q}_c}{T} = 0 \]

and

\[ s_1 = s_2 \]

From Eq. (3.22)
\[ \dot{Q} = T_{WF} \left[ \sum m_i \dot{s}_i - \sum m_i \dot{s}_i - \frac{\dot{Q}_{cx}}{T} \right]. \] (4.7)

Thus
\[ \dot{Q}_{comp} = \dot{m} T_{WF} \left[ s_2 - s_1 \right] = 0 \]

**Condenser :- Heat rejection process**

\[ \dot{Q}_{cond} = T_{WF} \left[ \dot{m} (s_3 - s_2) - \frac{\dot{Q}_{cond}}{T_{cond}} \right], \]

or
\[ \dot{Q}_{cond} = \dot{m} T_{WF} \left[ (s_3 - s_2) + \frac{(h_2 - h_3)}{T_{cond}} \right] \] (4.8)

**Expansion Valve :-** The expansion is assumed to be an adiabatic. That is;
\[ \frac{\dot{Q}_{cx}}{T} = 0 \]

and
\[ \dot{Q}_{exp} = \dot{m} T_{WF} \left[ s_4 - s_3 \right] \] (4.9)

**Evaporator :-** Heat transfer from the refrigeration load.

\[ \dot{Q}_{evap} = T_{WF} \left[ \dot{m} (s_1 - s_4) - \frac{\dot{Q}_{evap}}{T_{evap}} \right] \]

or
\[ \dot{Q}_{evap} = \dot{m} T_{WF} \left[ (s_1 - s_4) + \frac{(h_4 - h_1)}{T_{evap}} \right] \] (4.10)
Thus, the total irreversibility of the system can be expressed as

\[ \dot{I}_{\text{total}} = \dot{I}_{\text{comp}} + \dot{I}_{\text{cond}} + \dot{I}_{\text{r.p.}} + \dot{I}_{\text{r.e.}} \]  \hspace{1cm} (4.11)

Thermodynamic analysis is carried out by considering four different refrigerants, R-12 (Dichlorodifluoromethane, \( CCl_2F_2 \)), R-22 (chlorodifluoromethane, \( CHClF_2 \)), R-502 (Azeotrope, \( tHCIF_2 CClF_2CF_3 \)), and R-13-1a. Notice that R-12 is used in domestic refrigerators and water coolers \((T_{\text{r.p.}} = -30^\circ F - 30^\circ F)\), R-22 is used in air-conditioning \((T_{\text{r.p.}} = -20^\circ F - 40^\circ F)\) and R-502 in supermarket frozen food applications \((T_{\text{r.p.}} = -40^\circ F - 20^\circ F)\). A computer program written by Kartsones and Erth [23] and revised by Fischer and Rice [32] is available for R-12, R-22 and R-502 to calculate the thermodynamic properties.

During the past few years, there has been much debate on the effect of the refrigerants, chlorofluorocarbons (CFCs) in particular, on the depletion of the stratospheric ozone layer, which will cause atmospheric warming or the greenhouse effect. It is believed that CFCs are a major contributor to the ozone depletion that has occurred in the stratosphere [24]. It is hypothesized that the chlorine portion of the molecule would be split off by the action of sunlight and free chlorine would react with ozone to reduce the concentration of the species. It is felt that the release of CFCs into the atmosphere should be controlled on a global basis in order to protect the ozone layer. In this regard, United States of America (USA) has approved a new federal law which requires that the production of CFCs refrigerants first be reduced and finally, by the end of 1999, prohibited entirely [29]. The ozone depletion potentials of R-12, R-502 and
R-22 are 1.0, 0.3 and 0.05 respectively [24]. Therefore the greatest emphasis has been on the elimination of R-12. Either hydrofluorocarbons or hydrochlorofluorocarbons are seen to be one of the solutions to the CFCs / ozone issue. R-134a ($CF_3CH_2F$) does not contain any chlorine atom and has emerged as a leading candidate to replace R-12. It is under active worldwide development by a number of chemical manufacturers working with the refrigeration / air conditioning industry [30].

In order to evaluate the performance of R-134a, a complete set of thermodynamic properties is required. Wilson and Basu [25] determined the thermodynamic properties of R-134a by performing experimental measurements. They have developed equations to describe the thermodynamic properties of R-134a, which can be tabulated by using these equations. To calculate the thermodynamic properties of R-12, R-22 and R-502, computer subroutines are available. With the help of equations and constants developed by Wilson et al. [25] and Downing [31], a modification is carried out in the existing subroutines to include the calculation of the thermodynamic properties of R-134a. The equations and constants for R-134a are provided in the Appendix A. Using these revised subroutines, saturation properties of R-134a for volume, pressure, entropy and enthalpy are tabulated in Appendix B. Superheated properties of R-134a are also generated, which are shown in Appendix C. These calculated values are exactly same as given by Wilson and Basu [25] and are in good agreement with the values presented by Melinder et al. [30]. As discussed earlier that R-134a is the potential replacement of R-12 in future, therefore a comparative study of thermophysical properties are also presented in the
Appendix D.

A computer program for the cycle analysis of ideal vapor-compression cycle with revised subroutines is used to calculate the performance result, for chosen refrigerants. The performance results of systems having R-12, R-22, R-502 and R-134a are summarized in Tables 4.1-4.4, respectively. These results are plotted in Figs. 4.4-4.13, for different refrigerants.

Fig. 4.4 is a plot of coefficient of performance as a function of condenser and evaporator temperatures for a R-12 ideal vapor-compression system. The coefficient of performance of the system decreases as the difference between the condensing and evaporating temperature increases. It also shows that the coefficient of performance increases as the evaporating temperature increases. Similar pattern can be seen for other refrigerants, i.e., R-22, R-502 and R-134a in Figs. 4.6, 4.8 and 4.10, respectively.

Fig. 4.5 is a plot of the total and components irreversible losses as a function of condenser temperatures for a R-12 refrigeration system. Almost all of the irreversible losses are in the expansion valve because of the high temperature and pressure drop from the condenser to evaporator. The condenser has some irreversibility due to desuperheating the refrigerant. Other components do not have any irreversible losses. The total and components irreversible losses increases with the increase in condenser temperature. For example, increasing condensing temperature from 100°F to 160°F, the increase in total irreversibility is 3.8 times. A similar pattern can be seen for other refrigerants i.e., R-22 and R-134a in Figs. 4.7 and 4.9, however, for R-502, an increase in irreversibility is more pronounced as shown in Fig. 4.11. This large
TABLE 4.1

*Coefficient of performance and distribution of irreversible losses for an ideal R-12 vapor-compression cycle.*

<table>
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<th>$T_{evap}$ (°F)</th>
<th>$T_{cond}$ (°F)</th>
<th>COP</th>
<th>$i_{comp}$ Btu/min</th>
<th>$i_{cond}$ Btu/min</th>
<th>$i_{expa}$ Btu/min</th>
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<th>$i_{Total}$ Btu/min</th>
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</tbody>
</table>

* $T_0 = 75^\circ F$, $\eta_{comp} = 100\%$, Capacity = 1 ton.
### TABLE 4.2

_Coefficient of performance and distribution of irreversible losses for an ideal R-22 vapor-compression cycle.*_

<table>
<thead>
<tr>
<th>$T_{\text{evap}}$ (°F)</th>
<th>$T_{\text{cond}}$ (°F)</th>
<th>COP</th>
<th>$i_{\text{comp}}$ (Btu/min)</th>
<th>$i_{\text{cond}}$ (Btu/min)</th>
<th>$i_{\text{expa}}$ (Btu/min)</th>
<th>$i_{\text{evap}}$ (Btu/min)</th>
<th>$i_{\text{Total}}$ (Btu/min)</th>
<th>$m$ (lb/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20.0</td>
<td>100.0</td>
<td>4.81</td>
<td>0.00</td>
<td>0.76</td>
<td>7.10</td>
<td>0.00</td>
<td>7.86</td>
<td>2.98</td>
</tr>
<tr>
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<td>120.0</td>
<td>3.56</td>
<td>0.00</td>
<td>1.19</td>
<td>12.12</td>
<td>0.00</td>
<td>13.31</td>
<td>3.30</td>
</tr>
<tr>
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<td>0.00</td>
<td>1.79</td>
<td>19.65</td>
<td>0.00</td>
<td>21.44</td>
<td>3.71</td>
</tr>
<tr>
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<td>160.0</td>
<td>2.04</td>
<td>0.00</td>
<td>2.67</td>
<td>31.43</td>
<td>0.00</td>
<td>34.10</td>
<td>4.31</td>
</tr>
<tr>
<td>40.0</td>
<td>100.0</td>
<td>6.98</td>
<td>0.00</td>
<td>0.38</td>
<td>4.05</td>
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</tr>
<tr>
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<td>120.0</td>
<td>4.87</td>
<td>0.00</td>
<td>0.69</td>
<td>7.69</td>
<td>0.00</td>
<td>8.38</td>
<td>3.20</td>
</tr>
<tr>
<td>40.0</td>
<td>140.0</td>
<td>3.55</td>
<td>0.00</td>
<td>1.13</td>
<td>13.35</td>
<td>0.00</td>
<td>14.49</td>
<td>3.60</td>
</tr>
<tr>
<td>40.0</td>
<td>160.0</td>
<td>2.63</td>
<td>0.00</td>
<td>1.81</td>
<td>22.36</td>
<td>0.00</td>
<td>24.17</td>
<td>4.15</td>
</tr>
</tbody>
</table>

* $T_o = 75^\circ$F, $\eta_{\text{comp}} = 100\%$, Capacity = 1 ton.
TABLE 4.3

Coefficient of performance and distribution of irreversible losses for an ideal R-502 vapor-compression cycle.*

<table>
<thead>
<tr>
<th>$T_{\text{evap}}$ (°F)</th>
<th>$T_{\text{cond}}$ (°F)</th>
<th>COP</th>
<th>$i_{\text{comp}}$ (Btu/min)</th>
<th>$i_{\text{cond}}$ (Btu/min)</th>
<th>$i_{\text{expa}}$ (Btu/min)</th>
<th>$i_{\text{evap}}$ (Btu/min)</th>
<th>$i_{\text{Total}}$ (Btu/min)</th>
<th>$m$ (lb/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-40.0</td>
<td>100.0</td>
<td>1.80</td>
<td>0.00</td>
<td>0.73</td>
<td>41.37</td>
<td>0.02</td>
<td>42.12</td>
<td>5.63</td>
</tr>
<tr>
<td>-40.0</td>
<td>120.0</td>
<td>1.35</td>
<td>0.00</td>
<td>1.03</td>
<td>65.42</td>
<td>0.02</td>
<td>66.46</td>
<td>6.81</td>
</tr>
<tr>
<td>-40.0</td>
<td>140.0</td>
<td>0.96</td>
<td>0.00</td>
<td>1.56</td>
<td>107.16</td>
<td>0.02</td>
<td>108.74</td>
<td>8.77</td>
</tr>
<tr>
<td>-40.0</td>
<td>160.0</td>
<td>0.60</td>
<td>0.00</td>
<td>2.88</td>
<td>200.49</td>
<td>0.02</td>
<td>203.38</td>
<td>13.00</td>
</tr>
<tr>
<td>-20.0</td>
<td>100.0</td>
<td>2.38</td>
<td>0.00</td>
<td>0.39</td>
<td>27.88</td>
<td>0.02</td>
<td>28.29</td>
<td>5.28</td>
</tr>
<tr>
<td>-20.0</td>
<td>120.0</td>
<td>1.76</td>
<td>0.00</td>
<td>0.57</td>
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<td>0.02</td>
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<td>6.31</td>
</tr>
<tr>
<td>-20.0</td>
<td>140.0</td>
<td>1.27</td>
<td>0.00</td>
<td>0.90</td>
<td>74.84</td>
<td>0.02</td>
<td>75.76</td>
<td>7.96</td>
</tr>
<tr>
<td>-20.0</td>
<td>160.0</td>
<td>0.82</td>
<td>0.00</td>
<td>1.70</td>
<td>137.53</td>
<td>0.02</td>
<td>139.25</td>
<td>11.29</td>
</tr>
</tbody>
</table>

*T* = 75°F, $\eta_{\text{comp}}$ = 100%, Capacity = 1 ton.
### TABLE 4.4

*Coefficient of performance and distribution of irreversible losses for an ideal R-134a vapor-compression cycle.*

<table>
<thead>
<tr>
<th>$T_{\text{evap}}$ (*F)</th>
<th>$T_{\text{cond}}$ (*F)</th>
<th>COP</th>
<th>$l_{\text{comp}}$ Btu/min</th>
<th>$l_{\text{cond}}$ Btu/min</th>
<th>$l_{\text{expa}}$ Btu/min</th>
<th>$l_{\text{evap}}$ Btu/min</th>
<th>$l_{\text{Total}}$ Btu/min</th>
<th>$m$ lb/min</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>100.0</td>
<td>3.44</td>
<td>0.00</td>
<td>0.14</td>
<td>13.87</td>
<td>0.00</td>
<td>14.01</td>
<td>3.48</td>
</tr>
<tr>
<td>0.0</td>
<td>120.0</td>
<td>2.59</td>
<td>0.00</td>
<td>0.18</td>
<td>22.80</td>
<td>0.00</td>
<td>22.99</td>
<td>3.98</td>
</tr>
<tr>
<td>0.0</td>
<td>140.0</td>
<td>1.95</td>
<td>0.00</td>
<td>0.25</td>
<td>36.69</td>
<td>0.00</td>
<td>36.94</td>
<td>4.69</td>
</tr>
<tr>
<td>0.0</td>
<td>160.0</td>
<td>1.43</td>
<td>0.00</td>
<td>0.38</td>
<td>59.82</td>
<td>0.00</td>
<td>60.20</td>
<td>5.79</td>
</tr>
<tr>
<td>20.0</td>
<td>100.0</td>
<td>4.74</td>
<td>0.00</td>
<td>0.06</td>
<td>8.36</td>
<td>0.00</td>
<td>8.42</td>
<td>5.31</td>
</tr>
<tr>
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<td>120.0</td>
<td>3.46</td>
<td>0.00</td>
<td>0.09</td>
<td>14.73</td>
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</tr>
<tr>
<td>20.0</td>
<td>140.0</td>
<td>2.56</td>
<td>0.00</td>
<td>0.13</td>
<td>24.78</td>
<td>0.00</td>
<td>24.91</td>
<td>4.39</td>
</tr>
<tr>
<td>20.0</td>
<td>160.0</td>
<td>1.88</td>
<td>0.00</td>
<td>0.22</td>
<td>41.44</td>
<td>0.00</td>
<td>41.67</td>
<td>5.35</td>
</tr>
</tbody>
</table>

* $T_c = 75^\circ F$, $\eta_{\text{comp}} = 100\%$, Capacity = 1 ton.
Fig. 4.4 Coefficient of performance for an ideal vapor-compression cycle.
Fig. 4.5 Irreversible losses for an ideal vapor-compression cycle.
Fig. 4.6 Coefficient of performance for an ideal vapor compression cycle.
Fig. 4.7 Irreversible losses for an ideal vapor-compression cycle.
Fig. 4.8 Coefficient of performance for an ideal vapor-compression cycle.
Fig. 4.10 Coefficient of performance for an ideal vapor-compression cycle.
Fig. 4.11 Irreversible losses for an ideal vapor-compression cycle.
Fig. 4.12 Comparison of coefficient of performance between R-12 and R-134a.
Fig. 4.13 Comparison of irreversible losses between R-12 and R-134a.
increase in irreversible losses in the case of R-502 system is basically due to the large temperature difference between the condenser and evaporator.

A comparative study for R-12 and R-134a refrigeration systems are also shown in Figs. 4.12 and 4.13, respectively. Fig. 4.12 shows the comparison of coefficient of performance and Fig. 4.13 shows the comparison of total irreversibility for R-12 and R-134a refrigeration systems. The coefficient of performance of R-12 is a little more than R-134a and this difference increases slightly with the increase of condenser temperature. It should, however be noted that the total irreversibility of a R-134a refrigeration system is more than R-12 system and this difference become significant at higher condensing temperatures.

4.2 An Actual Vapor-Compression Cycle

So far the thermodynamic analysis of vapor-compression refrigeration systems have been made by ignoring the actual losses. In actual practice, the refrigerant undergoes an irreversible compression process, the pressure drop inside the condenser and evaporator, subcooling during condensation and superheating inside the suction line deviates the cycle from the standard basic vapor-compression cycle. To demonstrate the actual basic vapor-compression refrigeration cycle, the processes have been represented on the log(p)-h and T-s diagrams shown in Figs. 4.14 and 4.15, respectively. The deviations are due to

1. **Pressure drop and superheating of the vapor in the suction line**: The suction pipe usually passes through warm surrounding, and heat transfer to
Fig. 4.14 Pressure-enthalpy diagram for an actual vapor-compression cycle.

Fig. 4.15 Temperature-entropy diagram for an actual vapor-compression cycle.
the vapor can take place, causing a rise in its temperature. This is equivalent to wasting a part of the available refrigerating capacity. By insulating the suction pipe, the waste energy can be reduced but not entirely eliminated. Due to superheating, specific volume at the inlet of the compressor is increased. This results in an increase in the specific work of the compressor. While the pressure drop in the suction pipe increases the pressure ratio across the compressor, the volumetric refrigerating effect is reduced and the specific work of compression is increased.

(2) *Polytropic compression with friction losses*: The compression is no longer isentropic and there are irreversibilities due to friction and heat transfer. It should be emphasized that the performance of the compressor plays a significant role on overall performance of the refrigeration system.

(3) *Pressure drop in the condenser and evaporator*: The pressure drop in an evaporator results in a decrease in refrigerating effect and increase in pressure ratio across the compressor. In the condenser, the pressure drop increases the refrigerating effect.

(4) *Subcooling*: The subcooling of the liquid refrigerant in the subcooler or condenser is desirable. It increases the refrigerating effect.

In addition to the equations described earlier in the case of the ideal vapor-compression cycle, the following will be required to compute the thermodynamic analysis of an actual vapor-compression cycle, i.e.,
\[
COP = \frac{(h_1 - h_0)}{(h_2 - h_{1,1})}
\] (4.12)

\[
\dot{\text{\textit{Q}}}_{\text{pipe}} = \dot{m} T_{WF} \left[ (s_{1,1} - s_4) + \frac{(h_1 - h_{1,1})}{T_1} \right]
\] (4.13)

\[
\dot{\text{\textit{Q}}}_{\text{comp}} = \dot{m} T_{WF} \left[ s_2 - s_{1,1} \right]
\] (4.14)

\[
\dot{\text{\textit{Q}}}_{\text{cond}} = \dot{m} T_{WF} \left[ (s_3 - s_4) + \frac{(h_2 - h_3)}{T_{\text{cc}}} \right]
\] (4.15)

\[
\dot{\text{\textit{Q}}}_{\text{exp}} = \dot{m} T_{WF} \left[ s_4 - s_3 \right]
\] (4.16)

\[
\dot{\text{\textit{Q}}}_{\text{evap}} = \dot{m} T_{WF} \left[ (s_1 - s_4) + \frac{(h_3 - h_1)}{T_{\text{EE}}} \right]
\] (4.17)

where,
\[
T_{\text{EE}} = \text{Evaporator Temperature} + 10^\circ R
\]
\[
T_{\text{cc}} = \text{Condenser Temperature} - 10^\circ R
\]
\[
T_1 = \frac{(T_1 + T_{1,1})}{2} \text{ }^\circ R
\]

Thus, the total irreversibility of the system is given by
\[
\dot{\text{\textit{I}}}_{\text{total}} = \dot{\text{\textit{Q}}}_{\text{pipe}} + \dot{\text{\textit{Q}}}_{\text{comp}} + \dot{\text{\textit{Q}}}_{\text{cond}} + \dot{\text{\textit{Q}}}_{\text{exp}} + \dot{\text{\textit{Q}}}_{\text{evap}}
\] (4.18)

The computer program for the cycle analysis of an actual vapor
compression cycle are written to calculate the performance results for the desired refrigerants. The results of system analysis for R-12, R-22, R-502 and R-134a refrigeration systems are presented in the Tables 4.5-4.8, respectively. These performance characteristics are also plotted in Figs. 4.16-4.23. It should be noted that 65 % compressor efficiency assumed in this analysis is representative of reciprocating compressors [39], which includes the pressure drop across the suction and discharge side of the compressor.

Fig. 4.16 compares the coefficient of performance of an actual cycle with an ideal cycle as a function of condenser temperatures, for a R-12 refrigeration system. The coefficient of performance of the system decreases for both actual and ideal cycle as the condenser temperature increases. The difference in actual and ideal cycle is significant at lower condenser temperature and this difference decreases as the condenser temperature increases. Similar pattern is noticed for other refrigerants viz. R-22, R-502 and R-134a refrigeration systems as shown in Figs. 4.18, 4.20 and 4.22, respectively.

As a result of second-law-based thermodynamic analysis, the irreversibilities of all the components of the vapor-compression refrigeration cycle are plotted against the condenser temperature at fixed evaporator temperature. Fig. 4.17 compares the irreversible losses of an actual cycle with an ideal cycle for the R-12 system. The total irreversibility of the actual cycle is much greater than the ideal cycle. The largest irreversibility occurs in the compressor due to low-compressor efficiency and it increases, at higher condenser temperatures. The next largest irreversibility occurs in the expansion valve, which increases as the difference between the condenser and evaporator temperature increases. Losses
<table>
<thead>
<tr>
<th>( T_\text{evap} ) ((^\circ)F)</th>
<th>COP</th>
<th>( T_\text{cond} ) ( (%) )</th>
<th>( j_\text{cond} )</th>
<th>( j_\text{evap} )</th>
<th>( j_\text{Total} )</th>
<th>( q_\text{cond} )</th>
<th>( q_\text{evap} )</th>
<th>( q_\text{Total} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00</td>
<td>100.0</td>
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<td>2.08</td>
<td>1.76</td>
<td>34.41</td>
<td>51.97</td>
<td>18.03</td>
<td>34.41</td>
</tr>
<tr>
<td>0.00</td>
<td>120.0</td>
<td>1.62</td>
<td>2.19</td>
<td>1.62</td>
<td>35.57</td>
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<td>35.57</td>
</tr>
<tr>
<td>0.00</td>
<td>140.0</td>
<td>1.28</td>
<td>2.51</td>
<td>1.28</td>
<td>36.57</td>
<td>52.85</td>
<td>18.00</td>
<td>36.57</td>
</tr>
<tr>
<td>0.00</td>
<td>160.0</td>
<td>1.00</td>
<td>2.51</td>
<td>1.00</td>
<td>37.57</td>
<td>53.08</td>
<td>18.00</td>
<td>37.57</td>
</tr>
</tbody>
</table>

Note: \( T_\text{evap} \) = 75\(^\circ\)F, \( T_\text{cond} \) = 65\%, Superheating = 40\(^\circ\)F, Sub-cooling = 10\(^\circ\)F, Pressure drop in condenser, evaporator and suction pipe = 5\%, Capacity = 1 ton.
TABLE 4.6

Coefficient of performance and distribution of irreversible losses for an actual R-22 vapor-compression cycle.*

<table>
<thead>
<tr>
<th>$T_{evap}$ ($^\circ$F)</th>
<th>$T_{cond}$ ($^\circ$F)</th>
<th>COP</th>
<th>$i_{pipe}$ Btu/min</th>
<th>$i_{comp}$ Btu/min</th>
<th>$i_{cond}$ Btu/min</th>
<th>$i_{expa}$ Btu/min</th>
<th>$i_{evap}$ Btu/min</th>
<th>$i_{Total}$ Btu/min</th>
<th>m lb·min</th>
</tr>
</thead>
<tbody>
<tr>
<td>20.0</td>
<td>100.0</td>
<td>2.84</td>
<td>1.64</td>
<td>20.27</td>
<td>10.57</td>
<td>4.68</td>
<td>6.17</td>
<td>43.33</td>
<td>2.81</td>
</tr>
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<td>2.15</td>
<td>1.79</td>
<td>25.56</td>
<td>12.60</td>
<td>8.44</td>
<td>6.32</td>
<td>54.71</td>
<td>3.08</td>
</tr>
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<td>1.99</td>
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<td>15.09</td>
<td>14.04</td>
<td>6.51</td>
<td>69.22</td>
<td>3.42</td>
</tr>
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<td>14.87</td>
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<td>2.38</td>
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<td>2.74</td>
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<td>120.0</td>
<td>2.86</td>
<td>1.70</td>
<td>19.69</td>
<td>9.78</td>
<td>5.00</td>
<td>5.87</td>
<td>42.05</td>
<td>3.00</td>
</tr>
<tr>
<td>40.0</td>
<td>140.0</td>
<td>2.14</td>
<td>1.88</td>
<td>25.14</td>
<td>11.78</td>
<td>9.11</td>
<td>6.05</td>
<td>53.96</td>
<td>3.32</td>
</tr>
<tr>
<td>40.0</td>
<td>160.0</td>
<td>1.63</td>
<td>2.13</td>
<td>31.64</td>
<td>14.39</td>
<td>15.48</td>
<td>6.29</td>
<td>69.93</td>
<td>3.75</td>
</tr>
</tbody>
</table>

* $T_0 = 75^\circ$F, $\eta_{comp} = 65\%$, Superheating = 25$^\circ$F, Sub-cooling = 10$^\circ$F, Pressure drop in condensor, evaporator and suction pipe = 5%, Capacity = 1 ton.
**TABLE 4.7**

*Coefficient of performance and distribution of irreversible losses for an actual R-502 vapor-compression cycle.*

<table>
<thead>
<tr>
<th>$T_{evap}$ ($^\circ$F)</th>
<th>$T_{cond}$ ($^\circ$F)</th>
<th>COP</th>
<th>$I_{pipe}$ Btu/min</th>
<th>$I_{comp}$ Btu/min</th>
<th>$I_{cond}$ Btu/min</th>
<th>$I_{expa}$ Btu/min</th>
<th>$I_{evap}$ Btu/min</th>
<th>$I_{Total}$ Btu/min</th>
<th>$m$ lb/min</th>
</tr>
</thead>
<tbody>
<tr>
<td>-40.0</td>
<td>100.0</td>
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<td>2.45</td>
<td>51.78</td>
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<td>30.29</td>
<td>8.30</td>
<td>120.2</td>
<td>5.07</td>
</tr>
<tr>
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<td>2.89</td>
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<td>32.37</td>
<td>47.59</td>
<td>8.72</td>
<td>156.7</td>
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</tr>
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<td>75.33</td>
<td>9.35</td>
<td>212.5</td>
<td>7.32</td>
</tr>
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<td>116.9</td>
<td>52.93</td>
<td>125.8</td>
<td>10.45</td>
<td>310.8</td>
<td>9.68</td>
</tr>
<tr>
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<td>1.35</td>
<td>2.29</td>
<td>40.96</td>
<td>21.61</td>
<td>19.85</td>
<td>7.61</td>
<td>92.31</td>
<td>4.79</td>
</tr>
<tr>
<td>-20.0</td>
<td>120.0</td>
<td>1.08</td>
<td>2.67</td>
<td>52.03</td>
<td>25.42</td>
<td>32.37</td>
<td>7.97</td>
<td>120.4</td>
<td>5.58</td>
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<td>67.27</td>
<td>30.93</td>
<td>52.34</td>
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<tr>
<td>-20.0</td>
<td>160.0</td>
<td>0.55</td>
<td>4.16</td>
<td>91.77</td>
<td>40.34</td>
<td>87.77</td>
<td>9.39</td>
<td>233.4</td>
<td>8.70</td>
</tr>
</tbody>
</table>

* $T_o = 75^\circ$F, $\eta_{comp} = 65\%$, Superheating = 65$^\circ$F, Sub-cooling = 10$^\circ$F, Pressure drop in condenser, evaporator and suction pipe = 5\%, Capacity = 1 ton.*
### TABLE 4.8

*Coefficient of performance and distribution of irreversible losses for an actual R-134a vapor-compression cycle.*

<table>
<thead>
<tr>
<th>$T_{\text{evap}}$ (°F)</th>
<th>$T_{\text{cond}}$ (°F)</th>
<th>COP</th>
<th>$I_{\text{pipe}}$ Btu/min</th>
<th>$I_{\text{comp}}$ Btu/min</th>
<th>$I_{\text{cond}}$ Btu/min</th>
<th>$I_{\text{exp}}$ Btu/min</th>
<th>$I_{\text{evap}}$ Btu/min</th>
<th>$I_{\text{Total}}$ Btu/min</th>
<th>m lb/min</th>
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<tr>
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<td>9.72</td>
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</tr>
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<td>1.87</td>
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<td>6.78</td>
<td>74.31</td>
<td>3.64</td>
</tr>
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<td>45.60</td>
<td>14.69</td>
<td>26.57</td>
<td>7.06</td>
<td>96.07</td>
<td>4.20</td>
</tr>
<tr>
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<td>2.56</td>
<td>58.05</td>
<td>17.23</td>
<td>42.56</td>
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</tr>
<tr>
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<td>1.56</td>
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<td>6.07</td>
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<td>3.09</td>
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<td>1.75</td>
<td>28.19</td>
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<td>10.15</td>
<td>6.25</td>
<td>56.85</td>
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</tr>
<tr>
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<td>1.56</td>
<td>2.00</td>
<td>35.95</td>
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<td>17.44</td>
<td>6.50</td>
<td>73.79</td>
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<td>2.35</td>
<td>46.07</td>
<td>13.83</td>
<td>29.01</td>
<td>6.84</td>
<td>98.11</td>
<td>4.67</td>
</tr>
</tbody>
</table>

* $T_o = 75°F$, $\eta_{\text{comp}} = 65 \%$, Superheating = 40°F, Sub-cooling = 10°F,
Pressure drop in condensor, evaporator and suction pipe = 5 %, Capacity = 1 ton.
Fig. 4.18 Comparison of coefficient of performance for an actual vs ideal cycle.
Fig. 4.18 Comparison of coefficient of performance for an actual vs ideal cycle.
Fig. 4.19 Comparison of irreversible losses for an actual vs ideal cycle.
Fig. 4.20 Comparison of coefficient of performance for an actual vs ideal cycle.
Fig. 4.22 Comparison of coefficient of performance for an actual vs ideal cycle.
Fig. 4.23 Comparison of irreversible losses for an actual vs ideal cycle.
in the condenser are significant and have a little effect due to variation in the condenser temperature. These losses are due to pressure drop and desuperheating inside the condenser. Losses in the evaporator are small and losses in the suction pipe is the smallest, and they remain relatively constant with the condenser temperature. The total irreversible losses increases with the increase in the condenser temperature for both cycles. It is noted that due to sub-cooling, the irreversibility of the expansion valve of an actual cycle is less than the ideal cycle. A similar pattern can be seen for other refrigerants, i.e., R-22, R-502 and R-134a refrigeration systems in Figs. 4.19, 4.21 and 4.23, respectively.

A comparative study between R-12 and R-134a refrigeration systems is also carried out for an actual cycle and the results are shown in Figs. 4.24-4.27. Fig. 4.24 shows the comparison of coefficient of performance and Fig. 4.25 shows the comparison of total irreversibility for R-12 and R-134a refrigeration systems. The results are similar to as shown earlier in the case of an ideal cycle in Figs. 4.12-4.13. Fig. 4.26 compares the volumetric refrigerating effect (Btu/ft²). Volumetric refrigerating effect of R-12 is more than the R-134a and, the difference is almost constant with the condenser temperature. As the volumetric refrigerating effect of the R-134a is lesser than R-12, so the bigger size of the compressor will be required for the same refrigeration capacity in the case of a R-134a refrigeration system. Fig. 4.27 compares the discharge temperature of the two refrigerants. The discharge temperature of R-12 is more than the R-134a and the difference increases with the increase in the condenser temperature. Some of the comparison amongst the different refrigerants are also carried out recently by Muir [24]. The results of this analysis follows.
Actual Cycle

Sub-cooling = 10°F
Superheating = 65°F
Comp. efficiency = 65%

Fig. 4.24 Comparison of coefficient of performance between R-12 and R-134a.
Fig. 4.25 Comparison of irreversible losses between R-12 and R-134a.
Fig. 4.26 Comparison between R-12 and R-134a for volumetric refrigerating effect.
Fig. 4.27 Comparison between R-12 and R-134a for discharge temperature.
exactly the same pattern as given by Muir [24]. One of the major problems in using R-134a is of the lubricating oil. Chemical test showed [24] that the lubricating oil used with R-134a have some detrimental effect on the motor insulation material. This problem may be eliminated by introducing special lubricants that are compatible with the R-134a refrigerant.
CHAPTER 5

THERMODYNAMIC ANALYSIS OF TWO-STAGE VAPOR COMPRESSION REFRIGERATION CYCLES

The basic vapor-compression refrigeration cycle is discussed in the previous chapter. From the thermodynamic analysis of basic vapor-compression cycle it is apparent that the performance of the system decreases as the difference in the temperature between the condenser and evaporator increases. In addition to the degradation in the system performance, the compressor discharge temperature become very high due to high compression ratios, low compressor volumetric efficiency and an increase in the size of compressor due to large displacement volume per unit of refrigeration capacity.

The solution to this problem is to divide the compression process into two-stages. As we know that for vapor-compression refrigeration system, energy input to the system is the electric work required to drive the compressor. For the same refrigerating effect, the work input to the compressor may be reduced by compressor staging with intercooling between the stages, thereby increasing the coefficient of performance of the system. Due to intercooling between the stages, the discharge temperature of the gas will decrease and due to low compression ratio between stages, the compressor volumetric efficiency will also increase.

In this chapter the thermodynamic analysis of a two-stage, vapor-
compression refrigeration cycle and a system with mechanical subcooling are carried out for both ideal and actual cases. It should be noted that the mechanical subcooling is a modified version of the two-stage cycle. However, for two-stage vapor-compression system there exists an optimum interstage pressure for which the energy requirement is minimum. The optimum interstage pressure for a two-stage vapor compression system and a system with mechanical subcooling are studied in the following sections. Finally the results of the analysis are discussed.

5.1 An Ideal Two-Stage Vapor-Compression Cycle

Fig. 5.1 shows the equipment diagram for two-stage, vapor compression cycle; log(p)-h and T-s diagrams are shown in Figs. 5.2-5.3, respectively. The major components of this cycle include low pressure (LP) compressor, flash intercooler, high pressure (HP) compressor, condenser, expansion valves and evaporator. The refrigerant from the evaporator at point 5 (saturated) is compressed by the LP compressor to the flash intercooler at an interstage pressure. The separated vapor due to throttling (during the process 1-2) and desuperheating of the compressed vapor at state 6 yield the increased mass of vapor entering the HP compressor at state 7. High pressure superheated vapor leaves the HP compressor at state 8 and enters the condenser. The vapor condensed in the condenser at state 1 before being throttled to the flash intercooler pressure at state 2. The liquid after the flash intercooler at state 3 enters the expansion valve, where it expands to the evaporator pressure. At point 4 it leaves the expansion valve and enters the evaporator, where it receives
Fig. 5.1 Equipment diagram for a two-stage vapor-compression cycle.
Fig. 5.2 Pressure-enthalpy diagram for an ideal two-stage vapor-compression cycle.

Fig. 6.3 Temperature-entropy diagram for an ideal two-stage vapor-compression cycle.
heat from the surroundings.

Thermodynamic analysis with the help of conservation laws described earlier in chapter 3 is carried out for the two-stage cycle as follows:

Refrigerant mass flow rate through the evaporator for 1-ton of refrigeration capacity is

\[
\dot{m}_3 = \frac{200.0}{(h_3 - h_2)}. \tag{5.1}
\]

Energy balance applied to an adiabatic flash intercooler gives the mass flow rate through the high pressure compressor, i.e.,

\[
\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_1 h_2 + \dot{m}_3 h_6 \tag{5.2}
\]

and

\[
COP = \frac{\dot{m}_3 (h_3 - h_4)}{\dot{m}_3 (h_6 - h_3) + \dot{m}_1 (h_6 - h_2)}. \tag{5.3}
\]

From the second-law-based thermodynamics analysis described in chapter 3 (refer to Eq. (3.22)), we get

\[
\dot{\mathcal{I}}_{\text{comp,fp}} = \dot{m}_3 T_{WF} [s_6 - s_5], \tag{5.4}
\]

\[
\dot{\mathcal{I}}_{\text{flash}} = T_{WF} [\dot{m}_3 (s_3 - s_6) + \dot{m}_3 (s_2 - s_3)], \tag{5.5}
\]

\[
\dot{\mathcal{I}}_{\text{comp1HP}} = \dot{m}_1 T_{WF} [s_8 - s_7]. \tag{5.6}
\]
\[ \dot{I}_{\text{cond}} = \dot{m}_s T_{\text{WF}} \left( s_1 - s_2 \right) + \frac{(h_k - h_s)}{T_{\text{cond}}}. \]  
\[ (5.7) \]

\[ \dot{I}_{\text{exp1}} = \dot{m}_s T_{\text{WF}} \left[ s_2 - s_1 \right]. \]  
\[ (5.8) \]

\[ \dot{I}_{\text{exp2}} = \dot{m}_s T_{\text{WF}} \left[ s_4 - s_3 \right]. \]  
\[ (5.9) \]

\[ \dot{I}_{\text{comp}} = \dot{m}_s T_{\text{WF}} \left[ (s_3 - s_4) + \frac{(h_k - h_s)}{T_{\text{comp}}}. \right. \]  
\[ (5.10) \]

Thus, the total irreversibility of the system may be written as

\[ \dot{I}_{\text{total}} = \dot{I}_{\text{comp1}} + \dot{I}_{\text{Flash}} + \dot{I}_{\text{comp2}} + \dot{I}_{\text{cond}} + \dot{I}_{\text{exp1}} + \dot{I}_{\text{exp2}} + \dot{I}_{\text{comp}} \]  
\[ (5.11) \]

Using Eqs. (5.1)-(5.11) a computer program is developed to carry out the thermodynamic analysis of the ideal two-stage, vapor-compression refrigeration systems for refrigerants R-12, R-502 and R-134a. In this program revised subroutines described earlier in chapter 4 are used to calculate the thermodynamic properties of these refrigerants.

5.2 An Actual Two-Stage Vapor-Compression Cycle

As discussed earlier in the case of an actual basic vapor compression cycle, the cycle deviates from the ideal cycle due to the following reasons:

(i) Superheating of the vapor in the suction line.
(ii) Polytropic compression with friction losses.

(iii) Pressure drop in the condenser and evaporator.

(iv) Subcooling of the liquid in the condenser.

To demonstrate the actual two-stage, vapor-compression refrigeration cycle all the above effects are considered in the analysis. The actual processes have been represented on the log(p)-h and T-s diagrams shown in Figs. 5.4-5.5, respectively.

The thermodynamic analysis of an actual, two-stage, vapor-compression refrigeration cycle is similar to an ideal two-stage vapor-compression cycle, with additional irreversible losses due to suction-line heating. The coefficient of performance and the irreversible losses are given by

\[
COP = \frac{\dot{m}_3 (h_{s1} - h_i)}{\dot{m}_3 (h_e - h_i) + \dot{m}_4 (h_4 - h_i)}, \tag{5.12}
\]

\[
\dot{L}_{pipe} = \dot{m}_3 T_{WF} \left[ (s_s - s_{s1}) + \frac{(h_{s1} - h_i)}{T_{t}} \right], \tag{5.13}
\]

and Eqs. (5.7) and (5.10) may now be written as

\[
\dot{L}_{cond} = \dot{m}_7 T_{WF} \left[ (s_1 - s_b) + \frac{(h_4 - h_i)}{T_{CC}} \right]. \tag{5.14}
\]
Fig. 5.4 Pressure-enthalpy diagram for an actual two-stage vapor-compression cycle.

Fig. 5.5 Temperature-entropy diagram for an actual two-stage vapor-compression cycle.
\[ \dot{I}_{\text{evp}} = \dot{m}_2 T_{WF} \{ s_{5,1} - s_{2} \} + \frac{(h_4 - h_{s,1})}{T_{EE}}, \] (5.15)

where,

\[ T_{EE} = \text{Evaporator Temperature} + 10^\circ R, \]
\[ T_{CC} = \text{Condenser Temperature} - 10^\circ R, \]
\[ T_s = (T_1 + T_{s,1}) / 2 \quad ^\circ R, \]
\[ T_{WF} = T_0 = 544.7^\circ R, \]

while the other irreversible losses are given by the equations described earlier in section 5.1.

Thus, the total irreversibility of the system is given by

\[ \dot{I}_{\text{total}} = \dot{I}_{\text{pipe}} + \dot{I}_{\text{comp,L}} + \dot{I}_{\text{flash}} + \dot{I}_{\text{comp,HP}} + \dot{I}_{\text{cond}} + \dot{I}_{\text{exp1}} + \dot{I}_{\text{exp2}} + \dot{I}_{\text{evp}} \] (5.16)

The computer program for the cycle analysis of an actual two-stage vapor-compression cycle is written to calculate the performance results for the desired refrigerants.

### 5.3 Mechanical Subcooling System

The performance of refrigeration/air-conditioning system can also be improved by adding subcooler loop in order to reduce the liquid refrigerant temperature leaving the condenser, when the difference between the condensing
and evaporating temperature is large. An auxiliary compressor is used in place of high pressure compressor to compress the refrigerant vapor generated in the subcooler [37-38].

Fig. 5.6 shows an equipment diagram for mechanical subcooling system, while log (p)-h diagram for an ideal system is shown in Fig. 5.7. It consists of the main compressor which sucks vapor form the evaporator (5) and delivers to the condenser pressure (6), where it is condensed (1). Now some of the liquid refrigerant is extracted from the main liquid line and is expanded in an expansion valve to the intermediate pressure (2), which results in evaporation of an extracted liquid refrigerant and subcooling of the main liquid line in the subcooler (3). The refrigerant gas generated in the subcooler is compressed to the condenser pressure (8) by utilizing an auxiliary compressor. On the other hand, the subcooled liquid, on leaving the subcooler enters the expansion valve and throttled to the evaporator pressure (4).

For an ideal mechanical subcooling system with 1-ton of refrigeration capacity, the analysis is carried out as follows.

\[
\dot{m}_3 = \frac{200.0}{(h_5 - h_4)}. \tag{5.17}
\]

Energy balance applied to the subcooler gives the mass flow rate through the auxiliary compressor.
Fig. 5.6 Equipment diagram for a mechanical sub-cooling system.
Fig. 5.7 Pressure-enthalpy diagram for an ideal mechanical subcooling system.

Fig. 5.8 Pressure-enthalpy diagram for an actual mechanical subcooling system.
\[ m_3 = \frac{\dot{m}_3 (h_3 - h_4)}{(h_3 - h_4)}, \] (5.18)

\[ \dot{m} = \dot{m}_3 + \dot{m}_7 \] (5.19)

and

\[ COP = \frac{\dot{m}_3 (h_3 - h_4)}{\dot{m}_3 (h_3 - h_4) + \dot{m}_7 (h_3 - h_4)}. \] (5.20)

From the second-law-based thermodynamics analysis, we get

\[ \dot{I}_{\text{Comp}} = \dot{m}_3 T_{WF} \{ s_3 - s_4 \}, \] (5.21)

\[ \dot{I}_{\text{rec}} = T_{WF} \{ \dot{m}_3 (s_3 - s_4) + \dot{m}_4 (s_4 - s_3) \}, \] (5.22)

\[ \dot{I}_{\text{Abscomp}} = \dot{m}_7 T_{WF} \{ s_7 - s_7 \}, \] (5.23)

\[ \dot{I}_{\text{cond}} = \dot{m} T_{WF} \{ (s_1 - s_M) + \frac{(h_M - h_1)}{T_{\text{cond}}} \}, \] (5.24)

\[ \dot{I}_{\text{epo1}} = \dot{m}_7 T_{WF} \{ s_2 - s_1 \}, \] (5.25)

\[ \dot{I}_{\text{epo2}} = \dot{m}_3 T_{WF} \{ s_4 - s_3 \}, \] (5.26)

\[ \dot{I}_{\text{eop}} = \dot{m}_3 T_{WF} \{ (s_5 - s_4) + \frac{(h_4 - h_1)}{T_{\text{eop}}} \}, \] (5.27)

where state (M) indicates the mixing point of the discharge of the auxiliary and
Thus, the total irreversibility of the system may be written as

$$i_{\text{total}} = i_{\text{comp}} + i_{\text{aux}} + i_{\text{auxcomp}} + i_{\text{cond}} + i_{\text{exp1}} + i_{\text{exp2}} + i_{\text{sup}}$$  \hspace{1cm} (5.28)

In Fig. 5.8, log(p)-h diagram of an actual mechanical subcooling system is shown. To demonstrate the actual mechanical subcooling system the effects of superheating, compressor efficiency and subcooling are considered in the analysis. The thermodynamic analysis of an actual mechanical subcooling system is similar to the an ideal system. The coefficient of performance and the irreversible losses of an actual system are given by

$$COP = \frac{\dot{m}_3 (h_{s3} - h_4)}{\dot{m}_1 (h_e - h_2) + \dot{m}_1 (h_e - h_i)}.$$  \hspace{1cm} (5.29)

$$i_{\text{pipe}} = \dot{m}_3 T_{WF} (s_3 - s_{s3}) + \frac{(h_{s3} - h_4)}{T_i}.$$  \hspace{1cm} (5.30)

and Eqs. (5.24) and (5.27) may now be written as

$$i_{\text{cond}} = \dot{m} T_{WF} (s_i - s_{s4}) + \frac{(h_i - h_{s4})}{T_{CC}}.$$  \hspace{1cm} (5.31)

$$i_{\text{sup}} = \dot{m}_3 T_{WF} (s_{s4} - s_4) + \frac{(h_e - h_{s4})}{T_{EE}}.$$  \hspace{1cm} (5.32)
where,

\[ T_{fe} = \text{Evaporator Temperature} + 10^\circ R, \]
\[ T_{cc} = \text{Condenser Temperature} - 10^\circ R, \]
\[ T_s = \frac{(T_1 + T_{sl})}{2} ^\circ R, \]
\[ T_{WF} = T_0 = 544.7^\circ R, \]

While the other irreversible losses are given by the equations described earlier in the case of ideal mechanical subcooling system.

Thus, the total irreversibility of the system is given by

\[ \dot{I}_{\text{total}} = \dot{I}_{\text{pipe}} + \dot{I}_{\text{comp}} + \dot{I}_{\text{nec}} + \dot{I}_{\text{auxcomp}} + \dot{I}_{\text{cond}} + \dot{I}_{\text{rep}1} + \dot{I}_{\text{rep}2} + \dot{I}_{\text{rep}} \quad (5.33) \]

A computer program is developed to carry out the thermodynamic analysis of an ideal and actual mechanical subcooling system for refrigerants R-502 and R-134a.

### 5.4 Optimum Inter-Stage Pressure

One of the design parameters of the two-stage, vapor-compression refrigeration system is the selection of an intermediate pressure, for which the total work is minimum. For the two-stage, compression of an ideal gas, this occurs at the geometric mean of the suction and discharge pressures. However, for real gases the intermediate pressures are usually different from the geometric mean value [5]. Several research workers have tried to predict the optimum
interstage pressure, which are already discussed in chapter 3.

In the present analysis, the optimum interstage pressure is found based on the maximum COP by varying the interstage pressure from the evaporator to condenser pressure. Comparison is made with the results reported in the literature. The analysis is carried out for both two-stage and mechanical subcooling, vapor-compression refrigeration cycles.

Using the equations given in the section 5.1-5.3, a computer program is developed to get the optimum interstage pressure based on the maximum COP.

5.5 Results and Discussion

The performance results show the effect of inter-stage temperature on the coefficient of performance and irreversibility. To show the predicted improvement in the performance characteristics of the two-stage vapor-compression refrigeration cycle with respect to the single-stage vapor compression cycle, the curves are plotted in the normalized form. The normalized COP may be defined as:

\[ COPN = \frac{COP \text{ of multi-stage cycle}}{COP \text{ of single-stage cycle}} \]  \hspace{1cm} (5.34)

Similarly, normalized irreversibility may also be defined. To get the clear understanding, the intermediate saturated temperature is plotted in the dimensionless form, i.e.,
\[ 0 = \frac{T_{\text{int}} - T_{\text{evap}}}{T_{\text{cond}} - T_{\text{evap}}} \]  

(5.35)

5.5.1 An Ideal Two-Stage Cycle

The influence of saturated, intermediate temperature on the coefficient of performance and irreversibility rates for different refrigerants for an ideal, two-stage vapor compression cycles are shown in the Figs. 5.9-5.14.

Fig. 5.9 is a plot of the normalized coefficient of performance as a function of saturated intermediate temperature, for R-134a. In this figure each curve corresponds to a constant condenser temperature. The normalized coefficient of performance of the system first increases and then decreases with the increase of saturation intermediate temperature. It shows that the system performance peaks at a saturation intermediate temperature nearly halfway between the condensation and evaporation temperatures. It also shows that the performance of the two-stage system as compared to the basic system increases as the difference between the condensing and evaporating temperature is raised. Notice that the basic system is operating at the same condenser and evaporator temperatures as that of the corresponding two-stage system. Similar pattern can be seen for other refrigerants, i.e., R-12 and R-502 refrigeration systems, in Figs. 5.11 and 5.13.

Fig. 5.10 is a plot of normalized total irreversible losses as a function of saturated intermediate temperature for a R-134a refrigeration system. It shows that the system irreversible losses are minimum at a saturation intermediate
Fig. 6.9 Normalized COP vs saturation interstage temperature for an ideal two-stage vapor-compression refrigeration cycle.
Fig. 5.10 Normalized irreversible losses vs saturation interstage temperature for an ideal two-stage vapor-compression cycle.
Fig. 6.11 Normalized COP vs saturation interstage temperature for an ideal two-stage vapor-compression refrigeration cycle.
Fig. 5.12 Normalized irreversible losses vs. saturation interstage temperature for an ideal two-stage vapor-compression cycle.
Fig. 6.13 Normalized COP vs saturation interstage temperature for an ideal two-stage vapor-compression refrigeration cycle.
Fig. 5.14 Normalized irreversible losses vs saturation interstage temperature for an ideal two-stage vapor-compression cycle.
temperature nearly the mean of the condenser and evaporator temperatures. As expected, the losses of the system decreases as the difference between the condensing and evaporating temperature increases. A similar pattern can be seen for other refrigerants, i.e., R-12 and R-502 refrigeration systems, in Figs. 5.12 and 5.14.

As the results are plotted in normalized form, so we can not get the absolute values of the COP and total irreversibility. To get the absolute values of coefficient of performance and irreversible losses of an ideal two-stage cycle, the values of single stage cycle (base system) are given in the Table 5.1, for R-134a, R-12 and R-502 refrigeration system. With the help of Eq. (5.34), the absolute values of an ideal two-stage vapor compression cycle can be found.

5.5.2 An Actual Two-Stage Refrigeration Cycle

The influence of saturated intermediate temperature on the coefficient of performance and irreversibility rates for different refrigerants for an actual, two-stage vapor-compression refrigeration cycles are shown in the Figs. 5.15 - 5.24. The plots shown in these figures are for different condensing temperatures, with return gas temperature at the inlet of compressor is 65 °F, compressor efficiency of 65 % and liquid subcooling in the condenser is 5 °F. The pressure drop in the condenser, evaporator and suction pipe is 5 % each.

Fig. 5.15 is a plot of the normalized coefficient of performance as a function of the saturated intermediate temperature for the R-134a refrigeration system, with different condensing temperatures. The normalized coefficient of
TABLE 5.1

Coefficient of performance and total irreversible losses for an ideal single stage vapor-compression cycle.*

<table>
<thead>
<tr>
<th>Refrigerants</th>
<th>$T_{\text{evap}}$ ($^\circ\text{F}$)</th>
<th>$T_{\text{cond}}$ ($^\circ\text{F}$)</th>
<th>COP</th>
<th>$i_{\text{Total}}$ Btu/min</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-134a</td>
<td>-20.0</td>
<td>100.0</td>
<td>2.58</td>
<td>22.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>120.0</td>
<td>1.98</td>
<td>34.9</td>
</tr>
<tr>
<td></td>
<td></td>
<td>140.0</td>
<td>1.51</td>
<td>54.4</td>
</tr>
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<td></td>
<td></td>
<td>160.0</td>
<td>1.10</td>
<td>87.4</td>
</tr>
<tr>
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<td>-20.0</td>
<td>100.0</td>
<td>2.68</td>
<td>19.4</td>
</tr>
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<td></td>
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<td>120.0</td>
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</tr>
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<td></td>
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<td>160.0</td>
<td>1.26</td>
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</tr>
<tr>
<td>R-502</td>
<td>-40.0</td>
<td>100.0</td>
<td>1.80</td>
<td>42.9</td>
</tr>
<tr>
<td></td>
<td></td>
<td>120.0</td>
<td>1.35</td>
<td>67.7</td>
</tr>
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<td></td>
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<td>140.0</td>
<td>0.96</td>
<td>110.7</td>
</tr>
<tr>
<td></td>
<td></td>
<td>160.0</td>
<td>0.60</td>
<td>207.2</td>
</tr>
</tbody>
</table>

* $T_\text{WF} = T_\text{a} = 85^\circ\text{F}$, $\eta_{\text{comp}} = 100 \%$, Capacity = 1-ton.
Fig. 6.15 Normalized COP vs saturation interstage temperature for an actual two-stage vapor-compression refrigeration cycle.

- Return gas temp. = 66°F
- Comp. efficiency = 86%
- Sub-cooling = 6°F

R-134a

\( T_e = 160°F \)
\( T_e = 140°F \)
\( T_e = 120°F \)
\( T_e = 100°F \)

Normalized COP
Fig. 5.16 Normalized irreversible losses vs saturation interstage temperature for an actual two-stage vapor-compression cycle.
Fig. 5.17 Normalized COP vs saturation interstage temperature for an actual two-stage vapor-compression refrigeration cycle.
Fig. 6.18 Normalized irreversible losses vs saturation interstage temperature for an actual two-stage vapor-compression cycle.
Fig. 5.19 Normalized COP vs saturation interstage temperature for an actual two-stage vapor-compression refrigeration cycle.
R-502
Te=-40°F

Sub-cooling = 5°F
Comp. efficiency = 85 %
Return gas Temp. = 66°F

Fig. 5.20 Normalized irreversible losses vs saturation interstage temperature for an actual two-stage vapor-compression cycle.
Fig. 5.21 Normalized COP vs saturation interstage temperature for an actual two-stage vapor-compression refrigeration cycle.
Fig. 5.22 Components irreversible losses vs saturation interstage temperature for an actual two-stage vapor-compression cycle.
Fig. 5.23 Components irreversible losses vs saturation interstage temperature for an actual two-stage vapor-compression cycle.
Fig. 5.24 Components irreversible losses vs saturation interstage temperature for an actual two-stage vapor-compression cycle.
performance of the system first increases and then decreases with the increase of saturation intermediate temperature. It shows that the system performance peaks again at a saturation intermediate temperature nearly halfway between the condensation and evaporation temperature as discussed earlier in the ideal case of two-stage refrigeration cycle. But when comparing to the ideal case the maximum COP slightly shifted towards the lower intermediate saturation temperature because of the higher irreversible losses in an actual cycle. It also shows that the performance of the system as compared to the basic system increases as the difference between the condensing and evaporating temperature increases. As compared to the ideal cycle the performance improvement is somewhat less, for example, from Fig. 5.15 at 140°F condensing temperature, the normalized COP at arithmetic mean intermediate saturation temperature is 1.235. While for an ideal case (refer to Fig. 5.9) the value is 1.392. Similar pattern can be seen for other refrigerants, i.e., R-12, R502 and R-22 refrigeration systems in Figs. 5.17, 5.19 and 5.21.

Fig 5.16 is a plot of normalized total irreversible losses as a function of saturation intermediate temperature for the R-134a refrigeration system. It shows that the system irreversible losses are minimum at a saturation intermediate temperature nearly the arithmetic mean of the condenser and evaporator temperatures. As expected the total irreversible losses of the system decreases as the difference between the condensing and evaporating temperature increases. A similar pattern can be seen for other refrigerants, i.e., R-12 and R-502 refrigeration systems, in Figs. 5.18 and 5.20.

To get the absolute values of the coefficient of performance and irreversible
losses of an actual two-stage cycle, the values of single stage cycle (base system) are given in Table 5.2, for different refrigerants.

As a result of the second-law-based thermodynamics analysis, the irreversibilities of all the components of the actual two-stage vapor compression refrigeration cycle are plotted against the saturation interstage temperature. Fig. 5.22 shows the components irreversible losses for the R-134a refrigeration system. The largest irreversibilities occurred in the compressors. At lower saturation interstage temperatures, the irreversibilities of LP compressor are low and HP compressor are high. As the saturation interstage temperature increases the irreversibilities of HP compressor decreases and LP compressor increases. But the total irreversibilities of two compressors are minimum at a saturation interstage temperature nearly at the arithmetic mean of the condenser and evaporator temperatures. Next largest irreversibilities occurred in the two expansion valves. Similar to the compressor, as one expansion valve irreversibility increases, the other decreases with respect to the saturation interstage temperature but the minimum value remains in the midway of the condenser and evaporator temperatures. The losses in the condenser and evaporator are significant but relatively negligible as compared to compressors and expansion valves and have little effect on the interstage saturation temperature. Losses in the flash intercooler increases with the increase in the interstage saturation temperature. Irreversible losses in the suction pipe is very small as compared to other components. A similar pattern can be seen for other refrigerants, i.e., R-12 and R-502 refrigeration systems, in Fig. 5.23 - 5.24.

To show the usefulness of this analysis, the calculated data are compared
**TABLE 5.2**

*Coefficient of performance and total irreversible losses for an actual single stage vapor-compression cycle.*

<table>
<thead>
<tr>
<th>Refrigerants</th>
<th>$T_{\text{evap}}$ (^{\circ}\text{F})</th>
<th>$T_{\text{cond}}$ (^{\circ}\text{F})</th>
<th>COP</th>
<th>$i_{\text{total}}$ Btu/min</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-134a</td>
<td>-20.0</td>
<td>100.0</td>
<td>1.36</td>
<td>93.4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>120.0</td>
<td>1.06</td>
<td>118.1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>140.0</td>
<td>0.81</td>
<td>152.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>160.0</td>
<td>0.61</td>
<td>205.5</td>
</tr>
<tr>
<td>R-12</td>
<td>-20.0</td>
<td>100.0</td>
<td>1.40</td>
<td>90.1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>120.0</td>
<td>1.11</td>
<td>111.1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>140.0</td>
<td>0.88</td>
<td>138.9</td>
</tr>
<tr>
<td></td>
<td></td>
<td>160.0</td>
<td>0.68</td>
<td>178.2</td>
</tr>
<tr>
<td>R-502</td>
<td>-40.0</td>
<td>100.0</td>
<td>0.91</td>
<td>146.7</td>
</tr>
<tr>
<td></td>
<td></td>
<td>120.0</td>
<td>0.69</td>
<td>191.9</td>
</tr>
<tr>
<td></td>
<td></td>
<td>140.0</td>
<td>0.51</td>
<td>262.7</td>
</tr>
<tr>
<td></td>
<td></td>
<td>160.0</td>
<td>0.34</td>
<td>396.3</td>
</tr>
</tbody>
</table>

* $T_{\text{wi}} = T_0 = 85^\circ\text{F}, \, \eta_{\text{comp}} = 65 \%, \, \text{Sub-cooling} = 5^\circ\text{F},$

Return gas temperature at the inlet of compressor $= 65^\circ\text{F},$

Pressure drop in condensor, evaporator and suction pipe $= 5 \%$

Refrigeration capacity of the system $= 1$-ton.
with the experimental data obtained at Copeland Corporation [39]. The system used in the experimental program is a two-stage vapor-compression refrigeration system using R-22 and R-502 as refrigerants.

The comparison between the calculated and experimental values for the coefficient of performance, for R-502 and R-22 are shown in Figs. 5.25-5.26, respectively. From these figures it can be seen that the experimental values are very close to the calculated values. The results obtained by the present analysis is within ±5 % of experimental values discussed in reference [39]. This variation is mainly due to the fixed compressor efficiency used in the present analysis. However, for a real compressor the compressor efficiency is a function of the evaporator and condenser pressure.

5.5.3 Mechanical Subcooling System

The influence of saturated, intermediate temperature on the coefficient of performance and irreversibility rates for refrigerants R-134a and R-502, for an ideal and actual, mechanical subcooling systems are shown in the Figs. 5.27-5.34.

Fig. 5.27 is a plot of the normalized coefficient of performance as a function of the saturated intermediate temperature, for an ideal R-134a refrigeration systems. In this figure each curve corresponds to a constant condenser temperature. The normalized coefficient of performance of the system first increases and then decreases with the increase of saturation intermediate temperature. It shows that the system performance peaks at a saturation
R-502

Te = -20°F
Tc = 120°F

Return gas temp. = 66°F
Comp. efficiency = 65 %
Sub-cooling = 5°F

Fig. 5.26 Comparison between calculated and experimental values for coefficient of performance.
Fig. 5.26 Comparison between calculated and experimental values for coefficient of performance.

Te = -20°F
Tc = 120°F

Return gas temp. = 65°F
Comp. efficiency = 65 %
Sub-cooling = 6°F
Fig. 5.27 Normalized COP vs Saturation interstage temperature for an ideal mechanical subcooling system.
Fig. 5.28 Normalized irreversible losses vs saturation interstage temperature for an ideal mechanical subcooling system.
Fig. 5.29 Normalized COP vs saturation interstage temperature for an ideal mechanical subcooling system.
Fig. 5.30 Normalized irreversible losses vs saturation interstage temperature for an ideal mechanical subcooling system.
Fig. 5.31 Normalized COP vs saturation interstage temperature for an actual mechanical subcooling system.
Fig. 5.32 Components irreversible losses vs saturation interstage temperature for an actual mechanical subcooling system.
Fig. 5.33 Normalized COP vs saturation interstage temperature for an actual mechanical subcooling system.
Fig. 5.34 Components irreversible losses vs saturation interstage temperature for an actual mechanical subcooling system.
intermediate temperature nearly halfway between the condensation and evaporation temperatures. It also shows that the performance of the two-stage system as compared to the basic system increases as the difference between the condensing and evaporating temperature is raised. Similar pattern can be seen for a R-502 refrigeration system, in Fig. 5.29.

Fig. 5.28 is a plot of normalized total irreversible losses as a function of saturated intermediate temperature for a R-134a refrigeration system. It shows that the system irreversible losses are minimum at a saturation intermediate temperature nearly at the arithmetic mean of the condenser and evaporator temperatures. As expected, the losses of the system decreases as the difference between the condensing and evaporating temperature increases. A similar pattern can be seen for a R-502 refrigeration system, in Figs. 5.30.

Fig. 5.31 is a plot of the normalized coefficient of performance as a function of the saturated intermediate temperature for an actual R-134a refrigeration system, with different condensing temperatures. The normalized coefficient of performance of the system first increases and then decreases with the increase of saturation intermediate temperature. It shows that the system performance peaks again at a saturation intermediate temperature nearly halfway between the condensation and evaporation temperatures similar to that discussed earlier in the case of an ideal, two-stage refrigeration cycle. But when comparing to the ideal case the maximum COP slightly shifted towards the lower intermediate saturation temperature because of the higher irreversible losses in an actual cycle. It also shows that the performance of the system as compared to the basic system increases as the difference between the condensing and evaporating
temperature increases. Similar pattern can be seen for R502 refrigeration system in Figs. 5.33.

To get the absolute values of the coefficient of performance and irreversible losses of an ideal and actual mechanical subcooling systems, the values of single stage cycle (base system) are given in Table 5.1-5.2, respectively, for different refrigerants.

As a result of the second-law-based thermodynamics analysis, the irreversibilities of all the components of the actual mechanical subcooling system are plotted against the saturation interstage temperature. Fig. 5.32 shows the components irreversible losses for the R-134a refrigeration system. The largest irreversibilities occured in the main compressor. At lower saturation interstage temperatures, the irreversible losses due to an auxiliary compressor are high. As the saturation interstage temperature increases the irreversibilities of an auxiliary compressor decreases and that of the main compressor increases. The irreversible losses due to an expansion valve 1 decreases and that of the expansion valve 2 increases, with the increase of saturation interstage temperature. The losses in the condenser are significant and increases with the increase of interstage saturation temperature. The losses in the evaporator remain constant with respect to interstage saturation temperature. Losses in the Subcooler decreases with the increase in the interstage saturation temperature. Irreversible losses in the suction pipe is very small as compared to other components. A similar pattern can be seen for the R-502 refrigeration system, in Fig. 5.34.
5.5.4 Optimum Inter-stage pressure

The comparison of formulae for optimum interstage pressure given by various investigators are shown in the Figs 5.35 - 5.46 for different refrigerants. These comparisons are made for both, the two-stage and mechanical subcooling vapor compression refrigeration cycle. The following are the (marked) points on the plots.

(i) Maximum coefficient of performance (COP).

(ii) COP at the arithmetic mean of the condenser and evaporator temperature.

(iii) COP at the saturation temperature corresponding to the pressure for equal pressure ratios in the first and second stage +5°C. This is given by Behringer [18].

(iv) COP at the geometric mean of the condenser and evaporator temperature.

(v) COP at the geometric mean pressure.

The result of various investigators for an ideal two-stage vapor compression cycle are compared in Fig. 5.35 for a R-134a refrigeration system. The geometric mean pressure gives the lowest coefficient of performance. COP corresponding to the arithmetic mean of the condensing and evaporating temperature found to be closest to the maximum COP. The other formulae give the value of COP in between the arithmetic mean temperature and geometric
Fig. 5.36 Comparison of formulae for optimum interstage pressure given by various researchers for an ideal two-stage cycle.

Te=140°F
Te=20°F
Te=0°F
Fig. 5.37 Comparison of formulae for optimum interstage pressure given by various researchers for an ideal two-stage cycle.
Fig. 5.38 Comparison of formulae for optimum interstage pressure given by various researchers for an actual two-stage cycle.
Fig. 5.39 Comparison of formulae for optimum interstage pressure given by various researchers for an actual two-stage cycle.
Fig. 5.40 Effect of compressor efficiency on optimum interstage pressure for a two-stage vapor-compression cycle.
Fig. 5.41 Effect of superheating on optimum interstage pressure for a two-stage vapor-compression cycle.
Fig. 5.42 Effect of subcooling on optimum interstage pressure for a two-stage vapor-compression cycle.
Fig. 6.43 Comparison of formulae for optimum interstage pressure for an ideal mechanical subcooling system.
Fig. 5.44 Comparison of formulae for optimum interstage pressure for an ideal mechanical subcooling system.
Fig. 5.46 Comparison of formulae for optimum interstage pressure for an actual mechanical subcooling system.
Return gas temp. = 65°F
Comp. efficiency = 65 %
Sub-cooling = 5°F
Tc=140°F

Fig. 5.46 Comparison of formulae for optimum interstage pressure for an actual mechanical subcooling system.
mean pressure. Although, the arithmetic mean temperature does not give the maximum COP, it is clear from this figure that it gives the best result as compared to the other formulae. When comparing in terms of percentage of error as compared to the maximum COP, geometric mean pressure has 1.9% and arithmetic mean temperature has only 0.2% error. It can be seen from the figure that the maximum COP occurs at higher interstage saturation temperature than arithmetic mean value. A similar pattern can be seen for other refrigerants, i.e., R-12 and R-502 refrigeration systems, in Figs 5.36-5.37.

Fig. 5.38 compares the results of various researchers for an actual two-stage vapor-compression cycle for the R-134a refrigeration system. The geometric mean pressure gives the lowest coefficient of performance. The COP corresponding to the arithmetic mean of the condensing and evaporating temperatures and other formulae found to be closer to the maximum COP. It should be noted that in an actual cycle the maximum COP occurs at lower interstage saturation temperature than the ideal cycle. A similar pattern can be seen for a R-502 refrigeration system, in Fig. 5.39.

To investigate the shifting of the maximum COP point from the higher saturation intermediate temperature to the lower saturation intermediate temperature for an actual cycle, the effects of compressor efficiency, superheating and subcooling are considered separately. In Fig. 5.40 the effect of compressor efficiency on the optimum interstage temperature is shown. There is very little effect of compressor efficiency on the shifting of maximum COP point. In Fig. 5.41 the effect of superheating in the suction line on the optimum interstage temperature is shown. The effect of superheating shifted the
maximum COP point closer to the arithmetic mean intermediate saturation temperature. More significant shift of the maximum COP point may be seen in Fig. 5.42, by considering subcooling in the condenser.

The results of various investigators for mechanical subcooling system are compared in Figs. 5.43 - 5.46 for R-134a and R-502 refrigeration systems. The result of an ideal mechanical subcooling system is similar to the ideal two-stage vapor compression cycle. The geometric mean pressure gives the lowest coefficient of performance and COP at the arithmetic mean temperature is closer to the maximum COP. The maximum COP occurs at higher interstage saturation temperature than the arithmetic mean value. The maximum COP point of an actual mechanical subcooling system is at lower interstage saturation temperature than the arithmetic mean value, as discussed earlier in the case of an actual two-stage vapor-compression cycle.
CHAPTER 6

CAPACITY CONTROL SCHEMES

Refrigeration and air conditioning systems have to be designed for variable capacity. The variable capacity requirement may be due to a variable load or simply due to the fact that the system was designed for the expected peak load, while the working load is most of the time lower than the peak load. For the best performance of the system, the capacity must match the load under a variety of operating conditions. This is performed by some form of capacity control.

Capacity control of a refrigeration system can be attained by various refrigerant flow modulation schemes. These flow control techniques may be categorized into two groups: techniques outside the compressor that may be employed with any compressor, regardless of its design, and means within the compressor that may be built in at the time of its manufacture [40]. In the following sections capacity control methods are discussed in which first four belongs to the first group and the remaining to the second group.

1. On And Off Control :- The most common method of regulating compressor capacity is to start and stop the compressor. For example, typical residential and some commercial refrigeration systems operate by starting and stopping the compressor to maintain the desired set-point conditions within the appropriate
limits. With such a method, there is no capacity modulation as the unit either runs at full capacity or not at all. This intermittent running of the compressor imposes wear and tear on the compressor in addition to difficulties of maintaining steady temperature within the suitable limits. A system may, however include several compressors which may be of different size [20]. Starting and stopping individual compressors allows several steps of capacity to be attained. This method of capacity control is a common practice in almost all supermarket refrigeration installations.

2. Hot Gas Bypass :- In this system a metered flow of discharge, hot gas, is bypassed back to the suction line in a proportion that balances the system capacity with the refrigeration load. This technique permits full modulation of system capacity by loading the compressor artificially with hardly any reduction in power consumption under-part load condition. This method is not efficient as the suction temperature tends to increase as more and more hot gas is passed. However the method is simple and can be used where, very-close temperature and humidity control is required [20].

3. Evaporator Pressure Regulator :- The capacity of the system may be controlled by regulating the evaporator temperature. An evaporator pressure regulator (EPR) valve is used to control the evaporator's outlet temperature for maintaining constant set-point conditions. It should be noted that a decrease in the refrigerating load will result in a reduced evaporator temperature. This decrease in the evaporator temperature initiates the closing of an EPR valve, creating a flow restriction between the evaporator and compressor suction and causing a reduction in the compressor suction pressure while increasing the
evaporator temperature and pressure [20]. The decrease in the compressor suction pressure increases the compression ratio while the specific volume of refrigerant at the inlet of compressor is reduced. Thus, lowering the refrigerating capacity of the system and increasing the compressor power consumption.

4. **Variable Speed** :- The capacity of a refrigerating compressor may be controlled by varying the motor speed, which in turn control the refrigerant flow rate. At reduced speed, the capacity is lower than at high speed. For the most part, speed regulation is limited to a minimum of 50% of the full load speed, resulting only in the two step of capacity control [41]. Nowadays it appears that the efficient method of changing the motor speed, and hence the compressor capacity continuously, is by varying the motor frequency [20].

5. **Clearance Volume Control** :- The pumping capacity of a reciprocating compressor also depends on the re-expansion of the refrigerant gas trapped in the clearance volume. The compressor capacity may be decreased by deliberately increasing the clearance volume; the greater the clearance volume, the greater the capacity reduction. Clearance volume can be artificially increased by incorporating clearance pockets or a small reservoir that are opened when the capacity reduction is required [20]. The refrigerant gas is compressed into these pockets on the compression stroke and re-expands into the compressor cylinder on the return stroke, preventing compression of the additional gas.

6. **Cylinder Unloading** :- This is the most widely used method of controlling the capacity of large, multi-cylinder reciprocating compressors under fluctuating load conditions [20]. Capacity control is accomplished by unloading one or more
cylinders so that they become ineffective. Consequently, the refrigerant vapor is
not compressed and pumped out of the unloaded cylinders to the condenser,
even though the pistons continue to go through their normal strokes.

In this chapter, the thermodynamic analysis with the help of first and
second law is carried out for the hot-gas-bypass capacity control scheme. The
equipment and log(p)-h diagram are shown in Figs. 6.1-6.2. As shown in Fig.
6.1 the hot-gas-bypass system uses a hot-gas-bypass valve to load the
compressor artificially as the evaporator load falls below the compressor
capacity. The valve modulates to maintain a set-suction pressure, by passing
part of the discharge, refrigerant gas, back to the compressor. The amount of
bypass required is determined by the evaporator load; as the evaporator load
decreases, the amount of bypass gas increases [42].

In addition to the performance equations described earlier in the case of the
basic vapor compression cycle, the following equations are required to quantify
thermodynamic losses of hot-gas-bypass capacity control scheme.

Throttling process through an adiabatic bypass valve with negligible change in
kinetic and potential energies (refer to Eq. (3.12)) is given by

\[ h_6 = h_2. \quad (6.1) \]

The first law of thermodynamics (refer to Eq. (3.9)) when applied to the mixing
point at the inlet of the compressor with negligible change in kinetic and
potential energies gives:

\[ \dot{m}_3 h_3 + \dot{m}_6 h_6 = \dot{m}_1 h_1, \quad (6.2) \]
Fig. 6.1 Equipment diagram for hot gas bypass capacity control system
Fig. 6.2 Pressure-enthalpy diagram for a hot gas bypass capacity control system.
where

\[ \dot{m}_1 = \dot{m}_6 + \dot{m}_5, \]  
\[ \text{(6.3)} \]

\[ \text{COP} = \frac{(h_5 - h_4)\dot{m}_5}{(h_2 - h_1)\dot{m}_1} \]  
\[ \text{(6.4)} \]

\[ \dot{q}_{\text{comp}} = \dot{m}_1 T_{WF} [s_2 - s_1] \]  
\[ \text{(6.5)} \]

\[ \dot{q}_{\text{HGB}} = \dot{m}_6 T_{WF} [s_6 - s_3] \]  
\[ \text{(6.6)} \]

\[ \dot{q}_{\text{cond}} = \dot{m}_5 T_{WF} [(s_3 - s_2) + \frac{(h_3 - h_2)}{T_{cc}}] \]  
\[ \text{(6.7)} \]

\[ \dot{q}_{\text{expa}} = \dot{m}_5 T_{WF} [s_4 - s_3] \]  
\[ \text{(6.8)} \]

\[ \dot{q}_{\text{swp}} = \dot{m}_5 T_{WF} [(s_5 - s_4) + \frac{(h_4 - h_3)}{T_{EF}}] \]  
\[ \text{(6.9)} \]

where,

\[ T_{EF} = \text{Evaporator Temperature} + 10^\circ R, \]
\[ T_{cc} = \text{Condensor Temperature} - 10^\circ R, \]
\[ T_{WF} = T_0 = 544.7^\circ R, \]

Thus, the total irreversibility of the system may be expressed as

\[ \dot{\mathcal{I}}_{\text{total}} = \dot{q}_{\text{comp}} + \dot{q}_{\text{HGB}} + \dot{q}_{\text{cond}} + \dot{q}_{\text{expa}} + \dot{q}_{\text{swp}}. \]  
\[ \text{(6.10)} \]
With the help of a computer program a thermodynamic analysis of the hot-gas-bypass capacity control scheme is carried out for systems having different refrigerants. It should be noted that the part-load performance results of different refrigeration systems discussed in Figs. 6.3-6.10, are based on the full-load system capacity of 1-ton at the operating conditions shown in these figures.

Fig 6.3 is a plot of coefficient of performance as a function of the capacity and the evaporator temperature for R-134a refrigeration system. The coefficient of performance of the system decreases as the capacity of the system reduces. This is because of the fact that the compression work remains the same but the refrigerating capacity of the system reduces with the reduced mass flow rate. Similar pattern is noticed for other refrigerants, viz. R-12, R-22 and R-502 refrigeration systems as shown in Figs. 6.5, 6.7 and 6.9 respectively.

As a result of second-law based thermodynamic analysis, irreversible losses of all the components of the hot-gas-bypass system are plotted against the system capacity in Fig. 6.4 for R-134a refrigeration system. The total irreversible losses of the system increases as the capacity of the system decreases. The irreversible losses in the compressor are significant and remain constant with capacity reduction. The losses in the condenser increases as the capacity of the system decreases. The irreversible losses in the expansion valve is higher at full-load system capacity and decreases with the decrease in the system capacity. This happened because of the reduced mass flow rate through the expansion valve at the reduced system capacity. Irreversibility in the evaporator is smallest, while the losses due to hot-gas-bypass valve at full load is zero, but these losses increase rapidly as the capacity of the system decreases, and become
R-134a
Comp. Efficiency = 65 %
Tc=140°F

Fig. 6.3 Variation of COP for hot gas bypass capacity control scheme.
Fig. 6.4 Variation of irreversible losses for hot-gas-bypass capacity control scheme.
R-12
Comp. Efficiency = 65 %
Tc=140°F

Te=20°F
Te=0°F
Te=-20°F

Coefficient of performance

Percentage of Full Load Capacity

Fig. 6.5 Variation of COP for hot gas bypass capacity control scheme.
Fig. 6.6 Variation of irreversible losses for hot-gas-bypass capacity control scheme.
Fig. 6.7 Variation of COP for hot gas bypass capacity control scheme.
Fig. 6.8 Variation of irreversible losses for hot-gas-bypass capacity control scheme.
R-502
Comp. Efficiency = 65 %
Tc=140°F

Te=20°F
Te=0°F
Te=-20°F

Fig. 6.9 Variation of COP for hot gas bypass capacity control scheme.
Fig. 6.10 Variation of irreversible losses for hot-gas-bypass capacity control scheme.
highest at the lowest system capacity. The increase in irreversible losses are due to an increased mass flow rate through the bypass valve at reduced system capacity. A similar pattern can be seen for other refrigerants, i.e., R-12, R-22 and R-502 refrigeration systems in Figs. 6.6, 6.8 and 6.10, respectively.
CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

1. A new refrigerant Tetrafluoroethane (R-134a), which is considered to be a leading candidate to replace Dichlorodifluoromethane (R-12), from several refrigeration / air-conditioning applications is investigated from the standpoint of performance. For this purpose, computer program described by Erth & Kartsounes [23] is modified to calculate the thermodynamic properties (by using the thermodynamic properties equations developed by various investigators based on the experimental data) of refrigerant R-134a. With the help of revised subroutines, thermodynamic properties are generated, which can be used as a reference for any type of thermodynamic analysis. A comparative study is also carried out between R-12 and R-134a refrigerants, which indicates that R-12 can be replaced by R-134a, without any significant loss in overall performance. It is found that the volumetric refrigerating effect of R-134a is less than R-12 for the same operating conditions. Thus, a bigger size of the compressor will be required for the same refrigerating capacity in the case of a R-134a refrigeration system.

2. Thermodynamic analysis of vapor-compression refrigeration cycles are investigated by both the first law and second law of thermodynamics. Although the first law (energy balance) approach to system analysis determine the overall performance of the system correctly, it fails on its
own to identify the sources of thermodynamic losses. Second law analysis (in terms of energy measure of irreversible loss which is typically described as irreversibility) is carried out for both basic as well as two-stage vapor-compression refrigeration cycles. This analysis is performed on each of the system components to determine their individual contribution to the overall system losses. It is found that most of the irreversible losses are due to the low compressor efficiency as well as in the expansion process. These losses increase with the increase in the difference between the condenser and evaporator temperatures. The performance results are provided for R-12, R-134a, R-22 and R-502 refrigeration systems.

3. It is found that the increase in irreversible losses at high temperature difference can be minimized by compressing the refrigerant in two stages. Although the addition of extra components provide additional sources of irreversible losses, they reduce the overall system irreversibility by minimizing the irreversible losses of the base components. It is found that a two-stage, R-134a refrigeration system when operating at $T_c = -20^\circ F$ and $T_e = 140^\circ F$, has about 24% improvement in system performance compared to the single-stage system. This improvement in performance increases with the increase in condensing temperatures. The coefficient of performance as well as the breakdown of irreversible losses are discussed for two-stage, R-12, R-134a and R-502 refrigeration systems. The performance results for R-22 and R-502 are also compared with the experimental values. It is found that the results obtained in the present analysis are within $\pm 5\%$ of the experimental values.
4. One of the design parameters of the two-stage refrigeration systems is the selection of interstage pressure. In this study, it is found that the optimum interstage pressure for the two-stage, and mechanical sub-cooling, vapor-compression refrigeration cycles is very close to the saturation pressure corresponding to the arithmetic mean of the condensing and evaporating temperatures. This is based on the fact that the major contribution to the irreversibles losses is due to the temperature difference. We note that the heat transfer to the flash intercooler is caused by the temperature difference between the condenser and the intercooler, and the intercooler and the evaporator. It is obvious that the position that would minimize both temperature differences is the arithmetic average of the condenser and evaporator temperatures. The results of various investigators are compared for R-12, R-134a, and R-502 refrigeration systems. For example, when comparing in terms of percentage of error (for R-134a) as compared to maximum COP, it is found that the geometric mean-pressure (generally used for perfect gas) has 1.9% and arithmetic mean temperature has only 0.2% error.

5. Thermodynamic losses are also quantified for the hot-gas-bypass capacity control scheme. The results are discussed for R-12, R-22, R-502, and R134a refrigeration systems. It is shown that to maintain the desired system capacity, the system is subjected to additional irreversible losses due to the bypass-valve. It should be noted that the thermodynamic analysis can also be carried out for other capacity control schemes by following the general guidelines discussed in this thesis.
NOMENCLATURE

\( \Lambda \) = Area, \( \text{ft}^2 \) (\( m^2 \))

\( E \) = Total energy, \( \text{Btu} \) (kJ)

\( e \) = Specific energy, \( \text{Btu/lb} \) (kJ/kg)

\( g \) = Acceleration due to gravity, \( \text{ft/sec}^2 \) (\( m/\text{sec}^2 \))

\( h \) = Specific enthalpy, \( \text{Btu/lb} \) (kJ/kg)

\( \Delta h \) = Change in enthalpy, \( \text{Btu/lb} \) (kJ/kg)

\( i \) = Irreversibility rate, \( \text{Btu/min} \) (kJ/sec)

\( m \) = Mass, \( \text{lb} \) (kg)

\( \dot{m} \) = Mass rate of flow, \( \text{lb/min} \) (kg/sec)

\( n \) = Compression index.

\( p \) = Pressure, \( \text{psia} \) (kPa)

\( Q \) = Heat transfer, \( \text{Btu} \) (kJ)

\( \dot{Q} \) = Rate of heat transfer, \( \text{Btu/min} \) (kJ/sec)

\( \rho \) = Density, \( \text{lb/ft}^3 \) (kg/m^3)

\( S \) = Entropy, \( \text{Btu}^\circ R \) (kJ^\circ K)

\( s \) = Specific entropy, \( \text{Btu/lb}^\circ R \) (kJ/kg^\circ K)

\( \Delta S \) = Change in entropy, \( \text{Btu}^\circ R \) (kJ^\circ K)

\( \dot{S} \) = Entropy creation (or generation), \( \text{Btu/min}^\circ R \) (kJ/sec^\circ K)

\( T \) = Temperature, \( ^\circ F \) (\( ^\circ C \))

\( T_o \) = Surrounding temperature, \( ^\circ R \) (\( ^\circ K \))

\( \theta \) = Dimensionless temperature,
\( \forall \) = Volume, \( \text{ft}^3 (\text{m}^3) \)

\( v \) = Specific volume, \( \text{ft}^3/\text{lb} (\text{m}^3/\text{kg}) \)

\( \dot{V} \) = Velocity, \( \text{ft/sec} (\text{m/sec}) \)

\( W \) = Work, \( \text{Btu} (\text{kJ}) \)

\( \dot{W} \) = Rate of work or power, \( \text{Btu/min} (\text{kW}) \)

\( z \) = Elevation, \( \text{ft} (\text{m}) \)

**Subscript**

\begin{align*}
\text{comp} & = \text{Compressor.} \\
\text{cond} & = \text{Condenser.} \\
\text{c.v.} & = \text{Control volume.} \\
\text{c.s.} & = \text{Control surface.} \\
\text{c} & = \text{State of substance leaving a control volume.} \\
\text{evap} & = \text{Evaporator.} \\
\text{expa} & = \text{Expansion.} \\
\text{i} & = \text{State of substance entering a control volume.} \\
\text{int} & = \text{Intermediate.} \\
\text{opt} & = \text{Optimum.} \\
\text{surr} & = \text{Surrounding.} \\
\text{wf} & = \text{Weighting factor.}
\end{align*}
REFERENCES


APPENDIXES
A. FORMULATION OF THE THERMODYNAMIC PROPERTIES OF REFRIGERANT - 134a

GENERAL EQUATIONS

1) LIQUID DENSITY

\[ \rho_l = \rho_c + \sum_{n=1}^{4} D_n (1 - T_r)^{n\alpha} \]  

(A.1)

where

\[ \rho_l \text{ is in lb/ft}^3 \]
\[ \rho_c = 31.976 \]
\[ D_1 = 51.1671 \]
\[ D_2 = 63.9001 \]
\[ D_3 = -72.21398 \]
\[ D_4 = 49.30054 \]
\[ T_r = \frac{T}{T_c} \]
\[ T_c = 673.65^\circ R \]
\[ T = T_r + 459.7^\circ R \]

2) VAPOR PRESSURE

\[ \ln P_v = A + \frac{B}{T} + CT + DT^2 + \frac{E(F - T)}{T} \ln (F - T) \]  

(A.2)

where

\[ P_v \text{ is in psi.} \]
\[ \Lambda = 22.9899336 \]
\[ B = -7.243876 \times 10^3 \]
\[ C = -1.336296 \times 10^{-2} \]
\[ D = 6.929662 \times 10^{-2} \]
\[ E = 0.1995548 \]
\[ F = 6.747251 \times 10^{-2} \]
3) \textit{EQUATION OF STATE}

\[ P = \frac{RT}{(v-b)} + \frac{A_2 + B_2 T + C_2 e^{kT}}{(v-b)^2} + \frac{A_3 + B_3 T + C_3 e^{kT}}{(v-b)^3} \]

\[ + \frac{A_4}{(v-b)^4} + \frac{A_5 + B_5 T + C_5 e^{kT}}{(v-b)^5} \]  \hspace{1cm} (A.3)

where

\[ R = 0.105178 \text{ psi-ft/lb/\(^\circ\)R} \] \hspace{1cm} \text{\(v\) is in cft/lb}

\[ b = 5.535127\text{E-3} \]

\[ B_2 = 2.352\text{E-3} \]

\[ A_2 = -4.447446 \]

\[ C_2 = -131.4301 \]

\[ A_3 = 8.630833\text{E-2} \]

\[ B_3 = -2.9616\text{E-5} \]

\[ C_3 = 3.856549 \]

\[ A_4 = -1.0017\text{E-3} \]

\[ A_5 = -1.063691\text{E-6} \]

\[ B_5 = 1.079076\text{E-8} \]

\[ C_5 = -3.137839\text{E-4} \]

\[ k = 5.475 \]

4) \textit{HEAT CAPACITY} \hspace{.5cm} (at constant pressure)

\[ C_{p0} = C_{p1} + C_{p2} T + C_{p3} T^2 + C_{p4} T^3 + \frac{C_{p5}}{T} \]  \hspace{1cm} (A.4)

where

\[ C_{p0} = \text{Heat capacity at constant pressure (Btu/lb R)} \]

\[ C_{p1} = -1.2557\text{E-3} \]

\[ C_{p2} = -4.37427\text{E-4} \]

\[ C_{p3} = -1.4871\text{E-7} \]

\[ C_{p4} = 0.00 \]

\[ C_{p5} = 6.8021 \]
DERIVED EQUATIONS

1)  **LATENT HEAT OF VAPORIZATION**

\[ \Delta H_{lv} = JT(v_x - v_f)P \left[ \frac{-B}{T^2} + C + DT - E \left( \frac{1}{T} + \frac{F \ln(F - T)}{T^2} \right) \right] \quad (A.5) \]

where

\[ \Delta H_{lv} = \text{Latent heat of vaporization (Btu/lb)} \]

\[ J = 0.1850497 \]

\[ v_x = \text{saturated volume of vapor (cft/lb)} \]

\[ v_f = \text{saturated volume of liquid (cft/lb)} \]

2) **ENTHALPY OF VAPOR**

\[ H = H_0 + J(pv - RT) + (C_{p1}T + C_{p2} \frac{T^2}{2} + C_{p3} \frac{T^3}{3} + C_{p4} \frac{T^4}{4} + C_{p5} \ln T) \]

\[ + J \left( \frac{A_2}{(v - h)} + \frac{A_3}{2(v - h)^2} + \frac{A_4}{3(v - h)^3} + \frac{A_5}{4(v - h)^4} \right) \]

\[ + Je^{kT}(1 + kTr) \left( \frac{C_2}{(v - h)} + \frac{C_3}{2(v - h)^2} + \frac{C_4}{3(v - h)^3} + \frac{C_5}{4(v - h)^4} \right) \quad (A.6) \]

where

\[ H = \text{Enthalpy (Btu/lb)} \]

\[ H_0 = \text{Enthalpy at reference temperature (\cdot \cdot \cdot 40^\circ F)} \]

\[ = \Delta H \text{ (at reference temperature)} - \Pi \text{ (vapor at reference temperature)} \]

whereas other constants are described in Eqs. (A.3)-(A.4).
3) **ENTROPY OF VAPOR**

\[ S = S_o + \left( C_{p1} T + C_{p2} T + C_{p3} \frac{T^2}{2} + C_{p4} \frac{T^3}{3} - \frac{C_{p5}}{T} \right) + JR \ln \left( \frac{(v - b)P_1}{RT} \right) \]

\[ - J \left( \frac{B_2}{(v - b)} + \frac{B_3}{2(v - b)^2} + \frac{B_4}{3(v - b)^3} + \frac{B_5}{4(v - b)^4} \right) \]

\[ + J \frac{K}{T_e} e^{4\tau} \left( \frac{C_2}{(v - b)} + \frac{C_3}{2(v - b)^2} + \frac{C_4}{3(v - b)^3} + \frac{C_5}{4(v - b)^4} \right) \]

\[ (A.7) \]

where

\[ S = \text{Entropy (Btu/lb} °R) \]

\[ S_o = \text{Entropy at reference temperature (} °40°F) \]

\[ = \frac{\Delta H \text{ (at ref. temp.)}}{T \text{ (at ref.)}} - S \text{ (vapor at ref. temp.)} \]

\[ P_1 = 14.696 \text{ psia.} \]

whereas other constants are described in Eqs. (A.3)-(A.4).
COMPUTER CALCULATIONS OF THERMODYNAMICS PROPERTIES.

The calculation of thermodynamic properties of R-12, R-134a, R-22 and R-502 is divided into six subprogrames. These subprogrames can be easily incorporated into computer programes where values of the thermodynamic properties are required.

Subroutine TABLES :- To provide the values of basic equations constants, corresponding to the desired refrigerants.

Function TSAT :- To evaluate the saturation temperature given the saturation pressure.

Function SPVOL :- To evaluate the specific volume of the vapor phase given the pressure and temperature.

Subroutine SATPRP :- To evaluate the saturation thermodynamic properties (pressure, specific volume of liquid and vapor, enthalpy of liquid and vapor, entropy of liquid and vapor, latent enthalpy of vaporization) given the saturation temperature.

Subroutine VAPOR :- To evaluate the thermodynamic properties of the superheated vapor phase (specific volume, enthalpy and entropy) given the temperature and pressure.

Subroutine TRIAL :- To determine remaining superheated vapor properties, given the pressure and one other property (specific volume, enthalpy or entropy) of a specified refrigerant.
### B. SATURATION PROPERTIES OF R-134A

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19
### B. SATURATION PROPERTIES OF R-134A

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<td>34.84(558.1)</td>
</tr>
<tr>
<td>Liquid Density @ 80 °F, lb/cft (kg/ m$^3$)</td>
<td>74.98(1201.1)</td>
<td>81.45(1304.7)</td>
</tr>
<tr>
<td>Specific Heat of Liquid @ 80 °F,</td>
<td>0.342(1.132)</td>
<td>0.208(0.871)</td>
</tr>
<tr>
<td>Btu/lb °F (kJ/kg °C)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Specific Heat of Vapor at Constant Pressure (1 atm) and @ 80 °F, Btu/lb °F (kJ/kg °C)</td>
<td>0.204(0.854)</td>
<td>0.141(0.590)</td>
</tr>
<tr>
<td>Heat of Vaporization at NBP, Btu/lb (kJ/kg)</td>
<td>93.67(217.8)</td>
<td>71.9(167.24)</td>
</tr>
<tr>
<td>Flammability Limits, vol. %</td>
<td>Nonflammable</td>
<td>Nonflammable</td>
</tr>
<tr>
<td>Ozone Depletion Potential</td>
<td>0.00</td>
<td>1.00</td>
</tr>
</tbody>
</table>