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SYSTEM COMPONENT BALANCING IN SPLIT TYPE AIR-CONDITIONERS

YÜKSEK LİSANS TEZİ

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NOMENCLATURE

A	: Area (m^2)
b	: Slope $\Delta i / \Delta T$ for saturated air
cp	: Constant pressure specific heat capacity (J/kg -^{0}C)
C	: Flow stream capacity (W / ^{0}C)
D	: Diameter (m)
E	: Exchanger effectiveness
f	: Fraction of exchanger
f'	: Friction factor
G	: Mass velocity (kg / m^2 s)
i	: Enthalpy (J / kg)
h	: Heat transfer coefficient (W / m^2 ^{0}C)
j	: Colburn factor
k	: Thermal conductivity (W / m ^{0}C)
L	: Length (m)
LMHD	: Log mean enthalpy difference (J / kg)
LMTD	: Log mean temperature difference (^{0}C)
ΔP	: Pressure drop (Pa)
$\dot{m}$	: Mass flow rate (kg /s)
N_r	: Number of rows of tubes
NTU	: Number of transfer units
Pr	: Prandtl number
P_s	: Fin pitch (mm)
Q	: Heat transferred (W)
R	: Radius (m)
Re	: Reynolds number
s	: Fin spacing (mm)

S_L	: Longitudinal tube spacing (mm)
S_T	: Transverse tube spacing (mm)
St	: Stanton number
T	: Temperature (^{0}C)
THR	: Total heat rejection (W)
U	: Overall heat transfer coefficient ($W / m^2 \ ^0C$)
V	: Fluid velocity (m / s)
y	: Fin thickness (mm)
x	: Vapor quality
Xtt	: Lockhart Martinelli parameter
Δx	: Tube wall thickness (m)
Δx_r	: Quality increment
η	: Efficiency
ρ	: Density (kg / m^3)
μ	: Dynamic viscosity (kg / m s)
σ	: Ratio of minimum free flow area to frontal area

Subscripts

a	: Air
ac	: Acceleration
act	: Actual
ain	: Air inlet
aout	: Air outlet
asub	: Air in the subcooled region
atr	: Air at refrigerant temperature
b	: At average free stream condition
bs	: At base condition
c	: Cold fluid
cin	: Cold fluid entering
dp	: Dew point

dbin	: Inlet air dry bulb
e	: Exit
f	: Fin
hf	: Hot fluid
h	: Hydraulic
hin	: Hot fluid entering
i	: Inlet
iavg	: Inlet average
iht	: Inside heat transfer
is	: Inside surface
l	: Liquid
m	: Mean
min	: Minimum
max	: Maximum
os	: Outside surface
ow	: Outside wet surface
p	: Constant pressure
r	: Refrigerant
rb	: Return bends
rin	: Refrigerant inlet
rout	: Refrigerant outlet
rsc	: Subcooled refrigerant
sr	: Moist air at refrigerant temperature
swm	: Saturated air at mean water film temperature
sl	: Saturated liquid
st	: Straight
sub	: Subcooled region
t	: Total
tp	: Two phase
v	: Vapor
tube	: Coil tube
wet	: Moisture condensation conditions

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ABSTRACT

The design principle of finned tube heat exchangers currently used in refrigerating and air conditioning systems is the determination of components. While making the selection of the components in traditional thermal design methods, each component is considered and designed separately. But the characteristic of each component has an interrelationship to the others, and the system formed by combining each component must perform properly at both design conditions and every condition that can be expected during operation.

The important point in the design is establishing the proper relationship or "balance" between the vaporizing and condensing sections of the system. This means that whenever an evaporator and a condensing unit are connected together in a system, a balance must be established between the rates of vaporization and condensation. That is, the rate at which the vapor is removed from the evaporator and condensed by the condensing unit is always equal to the rate at which the vapor is produced in the evaporator by the boiling action of the liquid refrigerant. When the system components are properly selected, the point of balance will occur at the system design conditions. On the other hand, when the selected components do not have equal capacities at the design conditions, system balance will be established beyond the system design conditions, and the system will not perform satisfactorily.

The efficiency of the system usually depends on the point at which the system reaches stabilized conditions or balances under operating

conditions. Because of many variables involved, the calculation of the system balance points extremely complicated. Therefore the purpose of this study is to develop an algorithm, to determine the system balance for a direct expansion type air conditioner.

ÖZET

Soğutma ve iklimlendirme sistemlerinde halen kullanılmakta olan ısı değiştiricilerin dizayn esaslarından birisi bileşenlerin seçimidir. Klasik termal dizayn metodlarında bileşenlerin seçimi yapılırken, her bileşen diğerlerinden farklı olarak düşünülür ve ayrı dizayn edilir. Fakat bileşenlerin herbirinin karakteristiği diğerleri ile yakın ilişkidedir. Bileşenlerin biraraya getirilmesi ile meydana gelen sistem, hem dizayn şartlarında ve hem de çalışma esnasında meydana gelebilecek her şartta uygun bir şekilde çalışmalıdır.

Dizayndaki diğer önemli bir nokta, sistemin buharlaşma ve yoğuşma olan bölümleri arasında uygun bir ilişkinin veya "dengenin" kurulmasıdır. Bunun anlamı; evaporatör ve kondenser ünitesi genel bir sistem içinde birlikte bağlandığında, buharlaşma ve yoğuşma miktarları arasında bir denge şartının kurulması gerektiğidir. Yani evaporatörden uzaklaştırılan ve yoğuşma ünitesi tarafından yoğuşturulan buhar miktarı her zaman sıvı soğutucu akışkanın buharlaşması yolu ile evaporatörde üretilen buhar miktarına eşit olmasıdır. Sistem bileşenleri, sistem dizayn şartlarında eşit kapasitelere sahip olacak şekilde seçildiğinde, denge noktası dizayn şartlarında meydana gelecektir. Diğer taraftan, dizayn şartlarında bileşenlerin eşit kapasitelere sahip olacak şekilde seçilmemesi durumunda ise sistem dengesi dizayn şartlarından başka çalışma şartlarında kurulacak ve sistem uygun bir performans gösteremeyecektir.

Sistem verimi genellikle, sistemin denge koşullarına veya çalışma şartları altındaki dengelere ulaştığı noktaya bağlıdır. Birçok değişkenin var olması sebebi ile sistemin denge noktalarının hesaplanması son derece karmaşıktır . Bu yüzden çalışmanın amacı; split direk genişlemeli tip hava

iklimlendiricisi için sistem denge noktalarını ve sistem kapasitesini tespit eden bir algoritma geliştirilmesi olarak saptanmıştır.

Alkol,vücütla temas ettiği zaman bir soğutma hissi uyandırır.Soğutma prensibi de bunun gibidir. Alkol,vücuda sürüldüğü zaman bir soğuma hissi uyandırır. Bunun sebebi alkolun buharlaşırken vücuttan bir miktar ısı çekmesidir. Soğutma prensibi de böyledir.

Sıvılar buhar haline geçerken çevresinden ve çevresindeki nesnelerden bir miktar ısı çeker. Tam tersi olarak buhar da sıvı hale geçerken sahip olduğu enerjiyi dışarıya verir.

Biz likit alkolü bir kabın içine koyarsak,alkol yavaş yavaş buharlaşırken çevreden bir miktar ısı çeker ve bulunduğu yeri soğutur. Tabii ki böyle bir kap içindeki alkol hemen kaybolacaktır.

Alkol, buharlaşırken çevresini soğutur fakat buharlaştıktan sonra tekrar kullanılması mümkün değildir.

Alkolü tekrar sıvı haline getiremez miyiz? Eğer getirebilirsek,belli bir miktar alkol kullanarak odayı soğutabiliriz. O zaman problem alkolü tekrar sıvı hale döndürmek. Bu durumda,biz de buharların doğal özelliği olan "Yüksek basınç ve yüksek sıcaklıkta buharlar sıvı hale döner."

Kompresör içinde sıkıştırılan alkol yüksek basınç ve yüksek sıcaklıkta ısı değiştirgecine gönderilir ve bir fan sayesinde yoğuşturulur. Sıvı hale gelir. Klimalarda ise alkol yerine F-22 gazı kullanılır.

Klimalar ısı taşıyıcısıdırlar. Yazın klimayı çalıştırdığınızda soğuk hava oda içine üflenirken sıcak hava dışarı atılır. Burada klima iç mahalden sıcaklığı alarak dışarıya taşır ve atar. Bu yüzden klimalara ısı taşıyıcısı adı verilir.

Klimanın içindeki soğutucu akışkan evaporatör (iç ünite) tarafına sıvı halde gönderilir ve sonra buharlaşır.

Bu yolla,sıvılaşmış soğutucu akışkan buharlaştırılır ve soğuk hava evaporatörün arkasında bulunan bir fan ile odaya verilir.

Soğutucu akışkan bir kompresör ve kondenser (dış ünite) yardımı ile devamlı kullanılmak üzere ters çevrilir.

Klimanın iç yapısı dört ana elemandan oluşur:

1. Evaporatör (İç ünite)
   Burada soğutucu akışkan buharlaştırılıp soğuk hava odaya verilir.

2. Kompresör
   Buhar halindeki soğutucu akışkanın basıncı arttırılır.Bu sayede ilerde sıvılaştırma için ilk adım atılmış olur.

3. Kondenser
   Yüksek sıcaklıktaki sıkıştırılmış buhar sıvı hale getirilir.

4. Kılcal

Sıvı akışkanın basıncı azaltılarak uçucu bir hale getirilir.

Evaporatör içindeki sıvı akışkan oda içindeki sıcak hava ile temas eder ve akışkanın (F-22) buharlaşmasına sebep olur. Akışkan buharlaşırken çevresinden de ısı çeker. Oda içindeki sıcak hava bir fan tarafından emilir,soğutulur ve tekrar odaya gönderilir. Isı iletkenliğini arttırmak için bakır boru kullanılmış,ayrıca bu bakır borular ince kanatçıklar ile yüzey alanı arttırılmıştır.

Soğutma çevrimini tekrar ettirmek için buhar halinde evaporatörden gelen akışkanın sıvılaştırılmadan önce sıcaklığının ve basıncının arttırılması gerekir.Kompesör sayesinde bu olay hızlandırılır ayrıca gazın çevrimini tamamlaması sağlanır.

Yüksek basınçta buhar halindeki akışkan kondenserde dış hava (30C-35C) sayesinde sıvılaştırılır. Yapı olarak kondenser;evaporatörün aynısıdır. Bu yüzden ikisinin yerleri değiştirilerek ısıtma da,soğutma da gerçekleştirilebilir.

"Kılcal"da sıvı akışkanın basıncı düşürülerek daha uçucu hale getirilir,sıvı akışkan (F-22) son derece dar bir borudan geçilmeye zorlanır. Bu borudan geçerken akışkanın sürtünmeden dolayı kaybettiği enerji basıncının düşmesine sebep olur. Basınç aniden düşürüldüğü için bir kısım akışkan buhar haline çevrilir,kalan sıvı ise soğuyarak sıcaklığın düşmesine sebep olur. İşin esası sıvı akışkan kılcaldan geçerken basıncı ve sıcaklığı düşer ve uçucu bir hale gelir.

Soğutma esnasında,soğuk havayı iç ünite ile sıcak havayı ise dış ünite ile atarız. Eğer ünitelerin yerini değiştirirsek sıcak havayı odadan içeri vermiş oluruz.

Ünitelerin yerini değiştirmek yerine gazın akış yönünü değiştiririz. Böylece istediğimiz zaman soğutma,istediğimiz zaman ısıtma konumunu seçebiliriz. Bunun için dört yollu bir valf kullanırız.

Split tip klimalar iki üniteden oluşur. İç ünite ve dış ünite. Dış ünite yukarıda da belirtildiği gibi kompresör ve kondenser, iç ünite ise evaporatörden oluşur. Çoğunlukla gündelik hayatta pencere tipi klimalar bilinir ki bunlar tüm bu elemanlar üzerinde tek bir ünite halindedir.

Sağutma sisteminin veya heat-pump sisteminin esas amacı ısıyı düşük sıcaklıktan yüksek sıcaklığa naklettirmektir. Bunu yaparken sistemin tüm elemanlarıyla uyum içinde çalışması gerekmektedir.

Dizayn aşamasında incelikle evaporatör dizaynı gelir ki bu başlıbaşına bir ünitedir. Daha sonra kondenser dizaynı yapılarak devam edilir. Kondenser dizaynı - dizayn sistemi olarak evaporatörün aynısıdır yalnız ufak tefek farklılıklar ile geliştirilmiştir. Konderser ünitesi kondersere uygun bir kompresör seçimi yapılır. Kondensör ve evaporatör dizaynında NTU metodu kullanılmıştır.

Dizayn ısı transfer katsayılarını hesaplayarak; bu hesaplar zorlanmış taşınım (hava tarafı için) ve iç soğutucu akışkan için yapılarak belirlenir. Daha sonra kanat verimi hesapları yapılır.

Burada önemli olan ısı değiştici yüzeyinin ıslak, kuru veya ıslak - kuru oluşudur. Bu üç çeşide göre hesaplar devam ettirilir.

Bütün bunları yapabilecek ve soğutma çevrimi veya heat-pump sistemi en yüksek performansla ve en efektif elemanlarla çalıştırmak için dizayn şartlarını ve ısı değiştirici boyutlarını elde etmeye yarayan bir algoritma da geliştirilmiştir.

Chapter One

INTRODUCTION

1. INTRODUCTION

Split type air-conditioners are mainly composed of two units.Indoor unit and outdoor unit. Outdoor unit consists of compressor and condenser,indoor unit consists of evaporator. Basically an air-conditioner is as we know-window type air-conditioners-all components in one unit. We can consider split type air-conditioners as a divided window type air-conditioner. Between indoor unit and outdoor unit of split type air-conditioners, the refrigerant piping is made by installing insulated copper tubes.

Basic aim of split type air-conditioners is to minimize the noise effect by leaving the compressor and condenser fan motor outside the residences. And also to provide more space to the indoor unit with an esthetic design. Therefore we are faced with various models of split type air-conditioners for different places. Most of them have remote control , timer and programming , dehumidification and capacity control functions.

In addition to cooling function with a four way valve-reversing valve- you can easily use your split type air-conditioners as a heat pump.

The term heat pump as applied to HVAC system in which refrigeration equipment is used such that heat is taken from a heat source and given up to the conditioned space when heating service is wanted and is removed from the space and discharged to a heatsink when cooling and dehumidification are desired. The thermal cycle is identical with that of ordinary refrigeration, but the application is equally concerned with the cooling effect produced at the evaporator and the heating effect produced at the condenser. In some applications both the heating and cooling effects obtained in the cycle are utilized.

Unitary heat pumps (as opposed to applied heat pumps) are shipped from the factory as a complete preassembled unit including internal wiring ,controls, and piping. Only the ductwork, external power wiring , and condensate piping are required to complete the installation. For the split unit is also necessary to connect the refrigerant piping between the indoor unit and outdoor sections.In appearance and dimensions , casing of unitary heat pumps closely resemble those of conventional air-conditioning units having equal capacity.

Capacities of unitary heat pumps range from about 1.5 to 25 tons or 5 to 90 kW although there is no specific limitation. This equipment is almost universally used in residential and the smaller commercial and industrial installations. The multi unit type of installation with a number of individual units of 2 to 20 tons or 7 to 70 kW of cooling capacity is particularly advantageous to obtain zoning and to provide simultaneous heating and cooling. It may also be used for heat reclaiming to conserve energy by connecting the units to a very common water circuit.

Large central heat pumps of modern design with in the capacity range of about 30 to 1000 horsepower or 20 to 750 kW of compressor-motor rating are now operating in a substantial number of buildings. A single or central system is generally used throughout the building but in some instances the total capacity is divided among several separate heat pump systems to facilitate zoning.

1.1. Heat Pump Types

The air-to-air heat pump is the most common type. It is particularly suitable for factory-built unitary heat pumps and has been widely used for residential and commercial applications. Figure 8-19 is typical of the refrigeration circuit employed. Outdoor air offers a universal heat-source, heat-sink medium for the heat pump. Extended surface, forced-convection heat transfer coils are normally employed to transfer the heat between the air and the refrigerant.

In some air-to-air heat pump systems, the air circuit may be interchanged by means of dampers (motor driven or manually operated) to obtain either heated or cooled air for the conditioned space. With this system one heat exchanger coil is always the evaporator and the other is always the condenser. The conditioned air will pass over the evaporator during the cooling cycle and the outdoor air will pass over the condenser. The change from cooling to heating is accomplished by positioning the dampers.

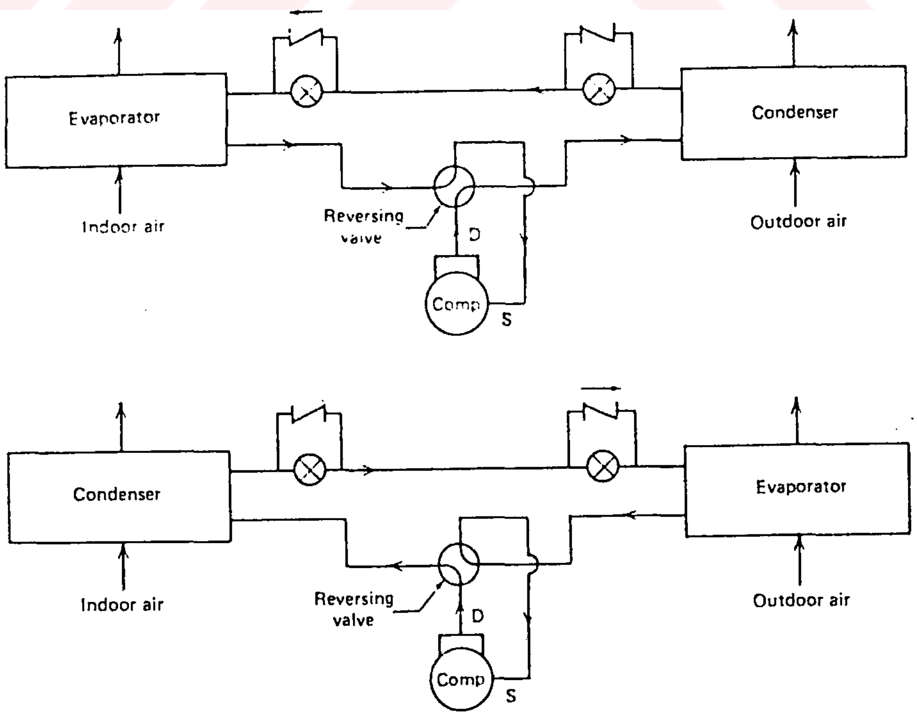


Figure 1.1 Schematics of a simple heat pump in cooling and heating modes.

Air-to-water heat pumps are commonly used in large buildings where zone control is necessary and are also sometimes employed for the production of hot or cold water in industrial applications as well as heat reclaiming. Heat pumps for hot water heating are commercially available in residential sizes.

A water-to-air heat pump uses water as a heat source and sink and uses air to transmit heat to or from the conditioned space.

A water-to-water heat pump uses water as the heat source and sink for both cooling and heating operation. Heating-cooling change over to perform the switching in the water circuits.

Water may represent a satisfactory and in many cases an ideal heat source. Well water is particularly attractive because of its relatively high and nearly constant temperature, generally about 50 F or 10 C in northern areas and 60 F or 16 C and higher in the south. However, abundant sources of suitable water are not always available and the application of this type of system is limited. Frequently sufficient water may be available from wells, but the condition of the water may cause corrosion in heat exchangers or it may induce scale formation. Other considerations to be made are the coats of drilling, piping and pumping, and the means for disposing of used water.

Surface or stream water may be utilized but under reduced winter temperatures the cooling spread between inlet and outlet must be limited to prevent freeze-up in the water chiller , which is absorbing the heat.

Under certain industrial circumstances waste process water such as spent warm water in laundries and warm condenser water may be a source for specialized heat pump operations.

Water-refrigerant heat exchangers generally take the form of direct expansion water coolers either of the shell-and-coil type or of the shell-and-tube type. They are circuited so that they can be used as a refrigerant evaporator during the cooling cycle.

1.2. Closed Loop Systems

In many a building may require cooling in interior zones while needing heat in exterior zones. The needs of the north zones of a building may also be different from those of the south. In many cases closed loop heat pump system is a good choice.

Individual water-to-air heat pumps in each room or zone accept energy from or reject to a common water loop, depending on whether that area has a call for heating or for cooling. In the ideal case the loads will balance and there will be no surplus or deficiency of energy in the loop. If cooling demand is such that more energy rejected to the loop than is required for heating, the surplus is rejected to the atmosphere by a cooling tower. In the other case an auxiliary furnace furnishes any deficiency.

Earth as a heat source and sink by heat transfer through buried coils has not been extensively used. This may be attributed to the high installation expense, the ground area requirements, and the difficulty and uncertainty of predicting performance.

The ground has been used successfully as a source-sink for heat pumps. The most common installation uses the vertical heat exchanger shown in Fig. 8-22. Water from the heat pump is pumped to the bottom of the "well" and slowly rises back up through the annulus. It exchanges heat with the surrounding earth before being returned back to the heat pump. Tests and analysis have shown rapid recovery in earth temperature around the well after the heat pump cycles off. Proper sizing of the well depends upon the nature of the earth surrounding the well, the water table level, and the efficiency of the heat pump.

Although still largely in the research stage, the use of solar energy as a heat source either on a primary basis or in combination with other sources is attracting increasing interest. Solar energy is discussed in detail in Chapter 16. Heat pumps may be used with solar systems either in a series or a parallel arrangement, or a combination of both. In the parallel arrangement, either the heat pump or the solar system is operating to meet the needs of a building. In this system the solar energy might be used first. If the storage tank temperature drops below a level that can maintain comfort for the space, the heat pump comes on automatically. This has the advantage that an air-to-air heat pump can be used and heat is easily rejected during the cooling cycle. A disadvantage is that this system cannot utilize the solar energy available and stored at temperatures below approximately 100 F.

If the solar system and the heat pump are connected in series, the heat pump in the heating mode gets its source energy from the solar storage tank, which is usually at a higher temperature than the outdoor temperature. In addition, the solar storage tank does not have to be maintained at a high temperature in order to furnish useful energy. With a moderate source temperature the heat pump and the collectors can operate with a high efficiency. If the tank temperature drops below some specified amount, energy must be added to the tank from supplemental sources to prevent lowered performance and eventual freezing of the fluid in the tank.

The series arrangement requires a water source heat pump, a type which some major manufacturers have been slow the develop in residential sizes. A water source heat pump must reject heat to water when operating in the cooling mode. the tank must be capable of rejecting heat to the environment if it is remain at an acceptable temperature. This requires a cooling tower or a source of cooling water such as a well, a stream, or a pond. Thermal simulations of series systems indicate that large storage is required to avoid extensive use of back-up fuels. This large storage in turn requires large collector areas for significant solar input. This large collector area creates an expensive system that is difficult to justify on an economic basis.

1.3. HEAT RECOVER SYSTEMS

In large commercial applications considerable heat energy is generated internal and may require removal even during the coldest weather. This condition usually occurs within the central spaces, which do not have exterior walls. It is necessary to exhaust considerable quantities air from large commercial structures because of the introduction of outdoor ventilation air. Considerable savings in energy can be realized if the heat energy from the interior spaces and the exhaust air can be recovered and used in heating the exterior parts of the structure. Heat energy may also be recovered from waste water.

Redistribution of heat energy within a structure can be accomplished through the use of heat pumps of the air-to-air or water-to-water type. Another approach is the use of the dual path systems described earlier.

Recovery of heat energy from exhaust air is accomplished through the use or air-to-air heat exchangers, rotating (periodic type) heat exchangers, and air-to-water heat exchangers connected by a circulating water loop. Sometimes spray systems are used ; they may contain desiccants to enhance latent heat transfer.

Finned tube heat exchangers are widely used in industry, particular, in the area of heating, cooling, and air-conditioning. In recent years studies on finned tube heat exchanger performance have continuosly increased. Some models of cooling and dehumidification performance analysis of finned tube heat exchangers and air-cooled condensers are presented in the literature.

Starting with one of the earliest recognized models, developed by Colburn and Hougen (1934), the heat and mass transfer performance of a cooling coil has been the subject of interest for many researches (Goodman - 1936 - , Bras - 1977 -). After making certain key assumptions, Threlkeld (1970) solved the simultaneous heat and mass transfer equation to predict the cooling and dehumidification performance of a cooling coil. He also suggested a combination of analytical and graphical solution to predict the exit air enthalpy and humidity ratio for known inlet air state, inlet coolant condition, and coil surface data. Elmahdy and Mitolas (1977) conducted an experimental investigation that showed good agreement with Threkeld's work. Goldstein (1983) developed a model which can be used to analyse a mathematically complete analysis of a plate fin heat exchanger. Heat transfer relationships in single phase liquid and vapor and two phase flow for refrigerants inside the tubes are included, as well as enthalpy analysis of the air side for a typical HVAC coil. The predicted exchanger performance is compared with the experimental data for various plate fin coils operated in the dry and wet modes. Oskarson (1990) developed evaporator models for operation with wet and frosted finned surfaces. Hill and Zeter (1991) developed a linear subgrid for the cooling and dehumidification coil models. Hiller and Glicksman modeled a condenser by using the Effectiveness - NTU approach. Ellison and Creswick (1978) retained Hiller and Glicksman 's model for a condenser in their steady state simulation of air to air heat pump. Ellison (1981) modeled an air cooled refrigerant condenser by complex refrigerant tube circuiting. Each tube was regarded as a lump (constant average properties for the whole tube). Anand and Tree (1982) developed a steady state simulation scheme for externally finned single tube heat exchanger in which the outer fluid is air and the inner fluid is a condensing refrigerant. Thermodynamic and flow properties of the refrigerant and temperature of the air flowing over the heat exchanger are predicted as a function of distance using finite difference method.

As can be seen, while many papers have been written about inside and outside heat transfer coefficients, fouling effect, two phase flow, etc. little attention has been given to determine the system balance and capacity in a heat pump system.

In this study, a model for a split direct expansion type air conditioner is developed. This model is intended to be a tool for a detailed design and also a scheme of air conditioner optimization. The purpose of the model is to design evaporator and condenser coils, which have the highest overall heat transfer coefficient and the lowest pressure drop as for as possible, according to the given set of conditions and the compressor used. By means of the algorithm developed, the points of system balance have been determined under given operating conditions and thus the system capacity has been obtained.

While carrying out this study, first of all the ideal vapor compression refrigeration cycle is explained, and then it is compared with the actual refrigeration cycle. At the end of Chapter 2, the definition of the mathematical model is presented. Chapter 3 focuses on mathematical description of the model. All recognized correlations and equations are reviewed. The algorithm of computer program is explained in Chapter 4 and then a simplified calculation flow chart which shows how to set-up the computer program to perform the iterative calculations is presented.

Chapter Two
MODEL STUDY

2. MODEL STUDY

2.1 The Ideal Vapor Compression Refrigerating Cycle

The purpose of a refrigeration cycle or heat pump is to transfer heat from a low temperature level (the heat source), to a high temperature level (the heat sink).

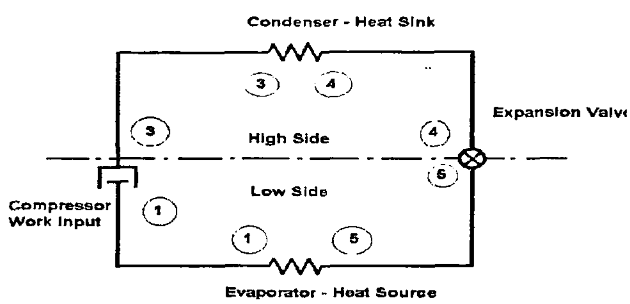


Figure 2.1. Simple vapor compression cycle (Adapted from H.Soumerai, Practical Thermodynamic Tools For Heat Exchanger Design Engineers, 1987)

In a simple vapor compression, refrigerant which changes from liquid to vapor as it absorbs heat in the evaporator is employed. A compressor sucks the vapor generated in the evaporator at a low pressure, compresses and discharges the hot vapor to liquify in the condenser as heat is discarded to the sink. High pressure liquid refrigerant is expanded and fed to the evaporator to complete the refrigeration cycle.

The ideal pressure - enthalpy diagram for a vapor compression refrigeration cycle is shown in Figure 2.2.

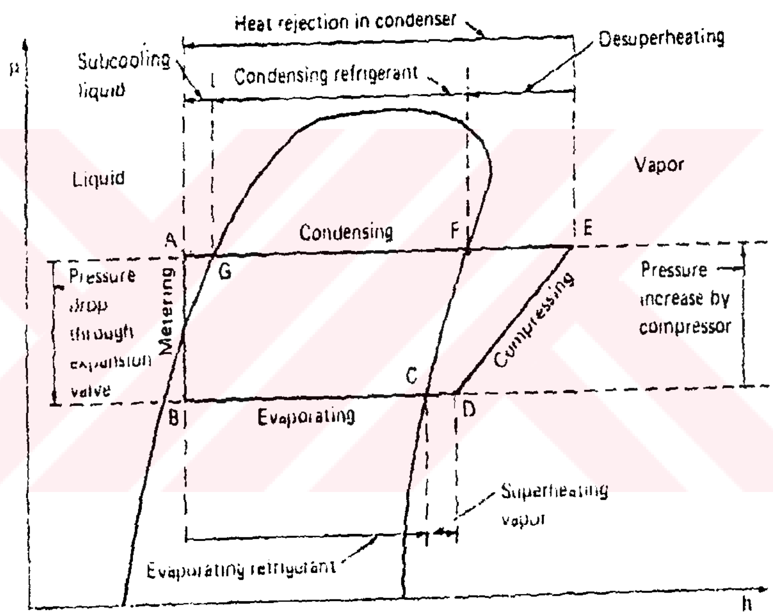


Figure 2.2. Ideal pressure - enthalpy diagram for a refrigeration cycle (Adapted from Eric C. Guyer, Handbook of Thermal Design, 1987)

Pressure of subcooled liquid, at point A, decreases in as it goes through the expansion valve. Some of the liquid flashes into vapor and cools the fluid entering the evaporator at point B. As the refrigerant passes from point B to C, the remaining liquid receives heat and changes from a liquid to vapor, but pressure does not increase. Superheating normally occurs between point C, where the vapor passes the saturation curve, and point D. As the vapor passes through the compressor, point D to E, its temperature, pressure, and enthalpy increase significantly because of the work input for compression. Line EF indicates that the vapor desuperheated within the condenser before it attains a saturated condition and begins to condense. Line FG represents the change from saturated vapor to saturated liquid within the condenser. Line GA represents subcooling before the liquid flows through the expansion device.

2.2. Relationships Between The Ideal and Actual Cycles

The ideal cycle departs from actual cycle in several respects, which are highlighted in paragraphs given below. The reason for this is that certain assumptions are made for the ideal cycle, but are invalid for actual cycle. For example, in ideal cycle, the drop in pressure in the lines and across the evaporator, condenser, etc., resulting from the flow of refrigerant through these parts is assumed to be isentropic - compression.

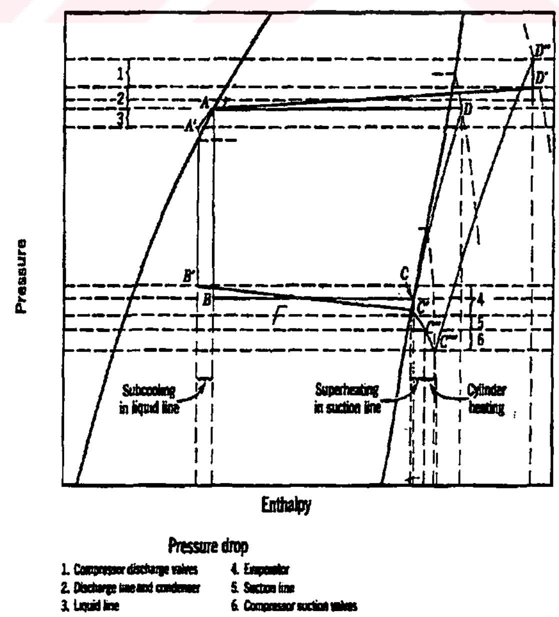


Figure 2.3. P-h diagram of actual refrigerating cycle illustrating effects of subcooling, superheating, and pressure loses. An ideal cycle is also drawn for comparison (Adapted from Ray J. Dossat, Principles of Refrigeration, FourthEdition, 1967)

A P -h diagram of an actual cycle, illustrating the pressure loss occuring in the various parts of the system, is shown in Figure 2.3.

Line C' - C" represents the pressure drop occured while the suction vapor is flowing through the suction line from the evaporator to the compressor inlet. Line C"-C"' represents the pressure drop that the suction vapor undergoes in flowing through the suction valves and passages of the compressor into the cylinder. Line C"'-D" represents the compression process for the cycle. The vapor in the cylinder is compressed to a pressure considerably above the average condensing pressure. This is necessary in order to force out of the cylinder through the discharge valves against the condensing pressure and against the additional pressure caused by spring - loading of the discharge valves. Line D"' - D' represents the drop in pressure required to force the discharge valves against the spring - loading and to force the vapor out through the discharge valves and passages of the compressor into the discharge line. Line D' - A represents the pressure drop resulting from the flow of the refrigerant through the discharge line and condenser. Line A - A' represents the pressure drop resulting from the flow of the refrigerant through the receiver tank and liquid line.

2.3. Definition of The Mathematical Model

2.3.1 Parts of the heat pump system

A simple flow diagram of the system model which will be used in this study is shown in Figure 2.4.
The principal parts of the system model are :

(1) Evaporator whose function is to provide a heat transfer surface
(2) Suction line which conveys the low pressure vapor from the evaporator to the suction inlet of the compressor

(3) Hermetic compressor whose function is to remove the vapor from the evaporator and to raise the pressure and temperature of the vapor to a point where the vapor can be condensed

(4) Hot - gas or discharge line which delivers the high pressure, high temperature vapor from the discharge of the compressor to the condenser

(5) Condenser whose purpose is to provide a heat transfer surface through which heat passes from the hot refrigerant vapor to the outdoor air

(6) Receiver tank provides storage for the condensed liquid so that a constant supply of liquid is available to the evaporator as needed

(7) Liquid line carries the liquid refrigerant from the receiver tank to the flow control device

(8) Refrigerant flow control device (thermostatic expansion valve) whose function is to meter the proper amount of refrigerant to the evaporator and to reduce the pressure of the liquid entering the evaporator so that the liquid will vaporize in the evaporator at the desired temperature

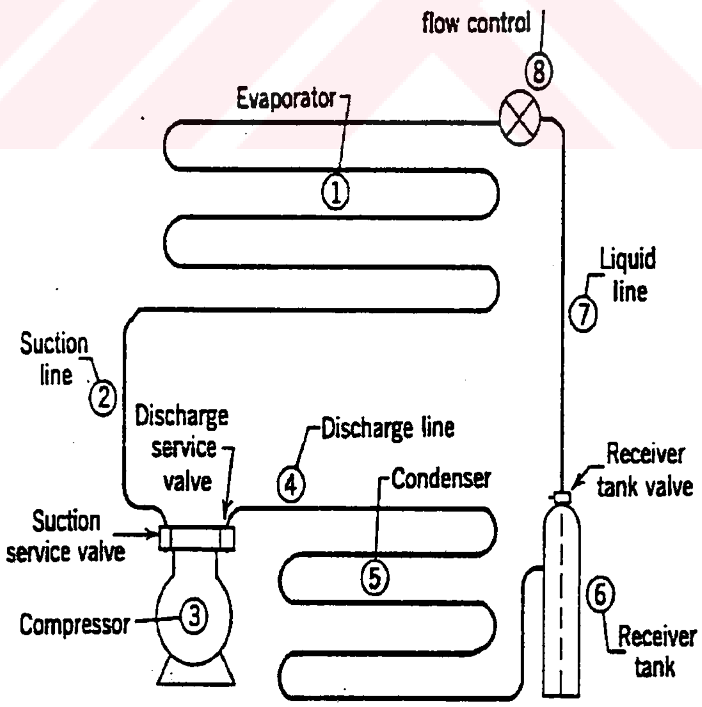


Figure 2.4. Flow diagram of the system model showing the principal parts(Adapted from Ray J. Dossat, Principles of Refrigeration, Fourth Edition, 1967)

Furthermore, the system model has :

- Filter drier is installed to clean out any dirt or foreign material and remove the moisture that might clog the expansion valve, and is mounted in liquid line, usually after the receiver tank
- Strainers are used in liquid and suction lines in order to protect automatic valves and the compressor from any foreign material
- Low presure control device is used to protect the compressor against flooding of suction line with liquid refrigerant. The high pressure control device is used to protect the compressor motor from overloading caused by breakdown of the condenser fan motor or any other condition that causes the head pressure to increase to an unsafe point
- Defrost control device, thermostats, solenoid valve, four - way reversing valve, check valves

2.3.2 Assumptions

Most heat pump coils consists of tubes with fins attached to their outer surface. The purpose of the fins is to increase the area on the air side, where the convection coefficient is generally much lower than on the refrigerant side. Refrigerant flows inside the tubes, and air flows over the outside of the tubes and fins. When a refrigerant evaporates in the tubes, the coil is called a direct expansion coil.

In this study the external surface of the tubes is called primary and the fin surface is called secondary surface. The primary surface consists of rows of the tubes that are staggered. The whole of the tubes that are perpendicularly on the same level according to the direction of air flow is called row. The inside surface of the tubes is smooth. The individual tubes passes in the coil are interconnected by return bends to form the serpentine arrangement of multipass tube circuits. The coils have aluminum smooth fins and copper tubes. Fin tube joints are secured by mechanical expansion (Fig.2.5). The cross-sectional area of the air stream at the entrance of the coil is called the face area. The face velocity of the air is the volume or rate of air flow divided by the face area. The surface area of the coil is the heat transfer area in contact with the air. The air and refrigerant, only R22, move in cross-flow (perpendicular to each other), as shown by the finned heat exchanger of Figure 2.5. The fins prevent motion in a direction that is transverse to the main flow direction.

The other assumptions which will be used while developing the program algorithm are given below:

- Refrigerant passes through three regions (subcooled, two phase, and superheated) in general so calcultive procedures and heat transfer correlations must be differentin each region.
- In the evaporator design, a mathematical simplification assumes that air side latent heat transfer can occur in the subcooled and two phase zones of the coil and moisture condensation in the superheated region is neglected. This is a good physical assumption because the superheated portion of the coil is usually quite small and the error incurred in neglecting moisture condensation is minimal.
- Inlet air assumed to be distributed uniformly by across the face of the heat exchanger and outlet air conditions for air leaving the subcooled, two phase, and superheated region are different. The average air condition is obtained by weight averaging the outlet conditions in proportion to the fraction of the exchanger in each region.
- Model results are invalid if ice forms on the heat exchanger because the heat exchanger is not in steady state operation if icing occurs
- Fluid properties are assumed to vary in the axial direction only
- The tube is so thin and thermal conductivity of the copper is so high that conduction resistance in the radial direction is negligible
- In order to make the computation more economical, axial heat transfer by conduction in the tube is neglected
- The air side heat transfer coefficient is assumed to be constant

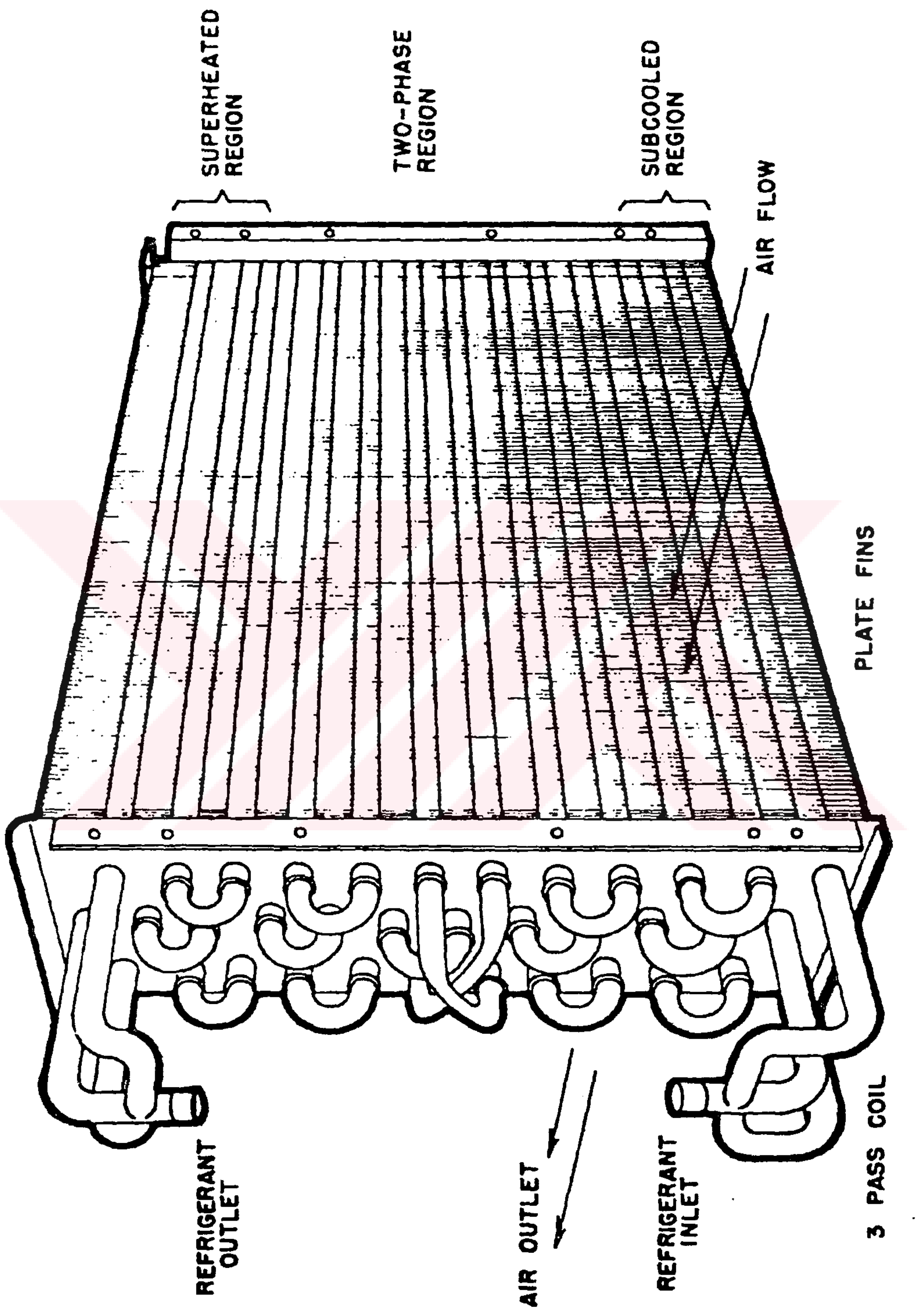


Figure 2.5. Finned tube heat exchanger arrangement (Adapted from S.D. Goldstein, A Mathematically Complete Analysis of Plate Fin Heat Exchangers, ASHRAE Trans, vol 89, page 470, 1983)

- Thermophysical properties of the copper tube and aluminum fins are assumed to be constant
- The liquid properties are a very weak function of pressure, therefore the saturated properties at the given temperature are used as subcooled properties for the refrigerant 22
- The two phase region is assumed to be a homogeneous mixture of liquid and vapor.

2.3.3. The mathematical model and determining the point of system balance

In this section, the variable-conductance model and the Component Balancing Method that are used to determine the calculation of the system balance points is introduced. The calculations of the evaporator and condenser coils are done by using the model. After determining the capacity curves of evaporator and condenser, the system balance points can be obtained by taking into account the compressor selection.

For the case of an evaporator or condenser, the overall heat transfer resistance is expressed as the sum of the convective resistance of the air, the conductive resistance of the heat exchanger, and the convective resistance of the refrigerant. The overall air side resistance is assumed to be constant since air flow rates and heat exchanger geometry do not vary with time. The convective resistance of the refrigerant is dependent on refrigerant properties and phase. Different correlations must be used to find its value in different refrigerant zones [23].

Calculation of direct expansion coil performance requires two different levels. First local heat transfer between the air and refrigerant conditions must be modeled in terms of convection and conduction heat transfer, and when necessary, the coupling of heat and mass transfer. Second, the integration of local heat and mass transfer throughout the coil must be modeled to determine the overall effect on leaving air and refrigerant conditons.

The algorithm advanced calculates the outlet refrigerant temperature, air dry bulb temperature and humidity ratio, and the total and sensible cooling capacity. It accounts for the heat and mass transfer associated with moisture condensation on the finned air side surface of the coil in accordance with ASHRAE methods. It determines also whether the finned surface is completely dry, copmletely wet, or partially dry.

The calculation of evaporator and condenser performance, and the outlet conditions of the fluids, since the inlet temperatures only are given, has been directly achieved by the use of the Effectiveness-NTU Method.

How to determine the point of the system balance is explained below.

First step in design process is to determine the effects of changing the operating conditions on the performance of each component. The next step is to study the performance of the entire system. To make this type of evaluation, the individual performance characteristics of the compressor, condenser, and evaporator must be combined.

Therefore first of all, capacity curves of the evaporator and condenser coils are determined according to the inlet air temperature, evaporating and condensing temperatures.

Fig.2.6 represents a typical capacity and power input curves for a hermetic reciprocating compressor.

The next step in analyzing the performance of the complete system is to study the behaviour of the compressor and the condenser coil operating together. These two components combine to form the condensing unit. The performance of a condensing unit can be found by superimposing the condenser performance characteristics on compressor capacity curves. At any operating point of the condensing unit, identical values of the condensing temperature, evaporating temperature, cooling capacity must exist for both the compressor and condenser. Operating points occur at intersections of common condensing temperatures. The line connecting the operating points represents the performance of the condensing unit.

Once the combined performance of the compressor and the condenser has been found, attention is shifted to the evaporator performance. For any particular evaporator and condensing unit, the points of system balance can be determined graphically by plotting the evaporator capacity and condensing unit capacity on the same graph (Fig. 2.7).

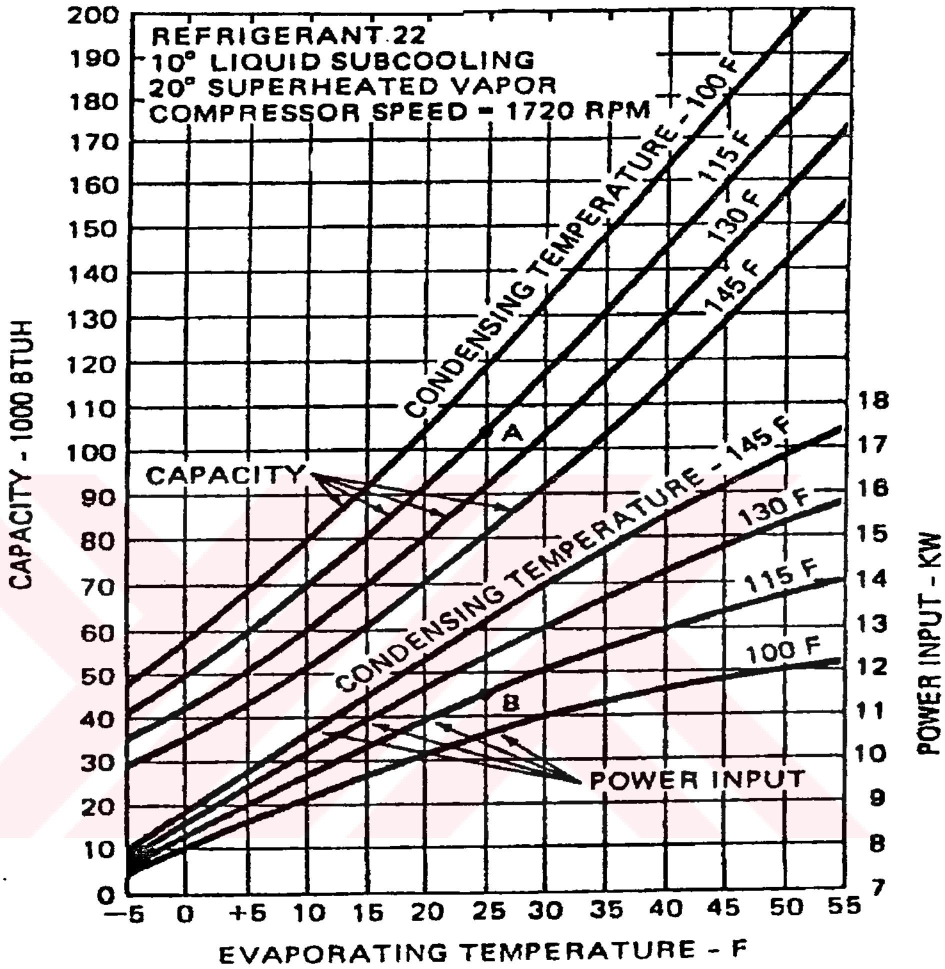


Figure 2.6. Capacity and power input curves of the compressor (Adapted from Eric Guyer, Handbook of Thermal Design, 1987)

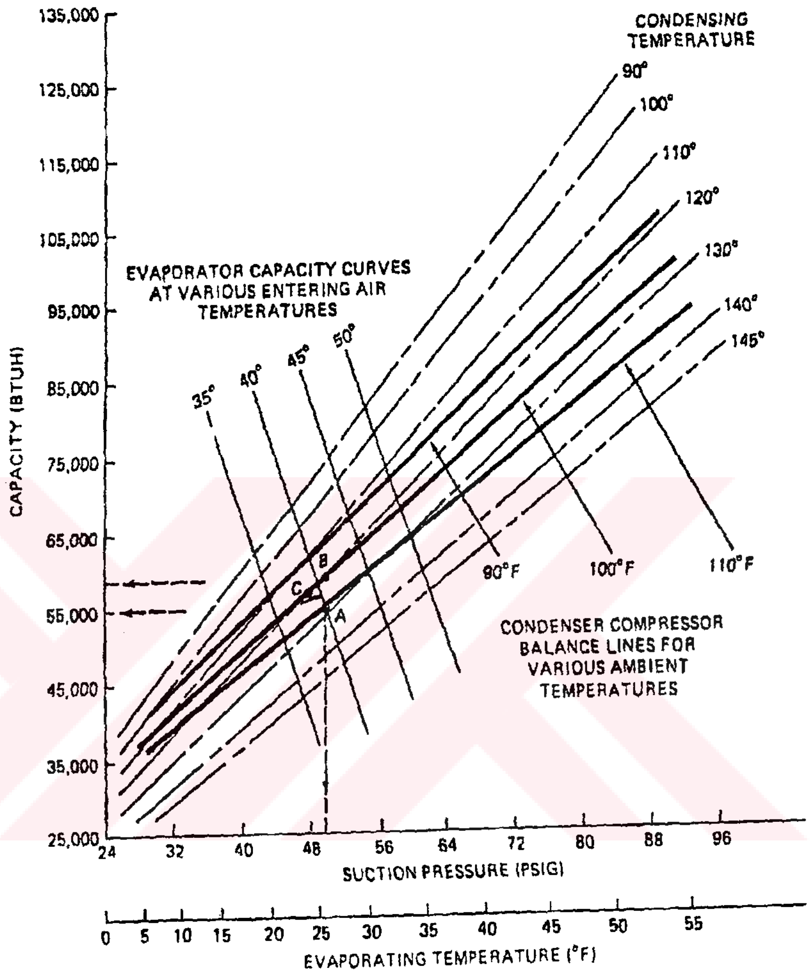


Figure 2.7. System balance chart (Adapted from Eric Guyer, Handbook of Thermal Design, 1987)

Chapter Three
THE DESIGN METHOD

3. THE DESIGN METHOD

3.1 Evaporator Design

Refrigerant passes through three regions (subcooled, two phase, and superheated) as it flows through the exchanger. In the subcooled region, the refrigerant enters in the liquid phase. Refrigerant temperature rises as the liquid moves through the heat exchanger and absorbs sensible and latent heat from the air. At saturated liquid point, the subcooled region ends and the two phase region begins .

Here, liquid and vapor refrigerant coexist in physical equilibrium. The quality of the mixture increases as refrigerant flows through the two phase region, picking up sensible and latent heat from the surrounding air stream until the refrigerant is completely evaporated. At that point, the two phase region ends and the superheated region begins. Average refrigerant temperature drops slightly across the two phase region because of fluid pressure drop. In the superheated region, refrigerant vapor gains superheat by absorbing sensible heat from the air.

On the air side of the heat exchanger, heat transfer can occur by two mechanisms: dry heat transfer or heat transfer with moisture condensation on exterior surfaces. Air side surfaces of the evaporator can also be partially dry.

The evaporator model is programmed to allow subcooled refrigerant to enter the coil. Evaluating the subcooled region involves the following sequence of the calculations:

First, the inlet enthalpy of the refrigerant must be checked against the saturated liquid enthalpy at the inlet pressure to determine whether a subcooled region exists. If the inlet enthalpy is greater than the enthalpy at the saturated liquid, there is no subcooled region in the coil.

If the subcooled region of the coil is completely dry, the following iterative procedure should be used to determine the amount of exchanger coil used for subcooling. The amount heat transfer occuring in the subcooled region is calculated as,

$$Q_{sub} = \dot{m}_r (i_{sl} - i_{sin}) \quad (3.1)$$

An assumptions of the fraction of the total exchanger area required for the subcooled region is then made and the air outlet temperature from this region is calculated as ,

$$T_{aout} = T_{dbin} - Q_{sub} / (cp_a \, \dot{m}_a) \quad (3.2)$$

The log mean temperature difference (LMTD) is defined as ,

$$LMTD = \frac{T_{dbin} - T_{aout}}{Ln\left[\frac{T_{dbin} - T_{rsub}}{T_{aout} - T_{rsub}}\right]} \quad (3.3)$$

and

$$f_{sub} = \frac{Q_{sub}}{U_{sub} \, .LMTD. \, A_t} \quad (3.4)$$

where f_{sub} is the fraction of exchanger in the subcooled region. If f_{sub} does not equal the fraction originally assumed, the program must iterate until obtaining the correct fraction.

If moisture condensation has occured on the coil, a different analysis must be used to determine the moisture condensation location. If it begins somewhere in the middle of the coil, the forward dry fraction is evaluated as in the previous dry analysis. A wet analysis then follows for the remainder of the coil. The relationships used in the wet analysis are given in the Section 3.5, therefore only a general procedure is presented here.

Values for the average tube and water - film temperatures on the coil must be assumed and an initial assumption must be provided for the fraction of exchanger in the wet subcooled region. The wet air - side coefficient, inside coefficient, the overall wet transfer coefficient, and the air outlet enthalpy from the subcooled region are then calculated.

$$i_{aout} = i_{atr} + (i_{ain} - i_{atr}) \exp\left[\frac{-U_{wet} A_{wet}}{m_{asub} c_{pa}}\right] \qquad (3.5)$$

The log mean enthalpy difference (LMHD) can be computed as:

$$LMHD = \frac{i_{ain} - i_{aout}}{\ln\left[\dfrac{i_{ain} - i_{atr}}{i_{aout} - i_{atr}}\right]} \qquad (3.6)$$

and then the tube temperature is calculated as follows[8]:

$$T_{tube} = T_{sub} + \frac{U_{wet} A_{os} LMHD}{h_{iht} A_{is} c_{pa}} \qquad (3.7)$$

The mean air enthalpy then becomes ,

$$i_{am} = i_{atr} + LMHD \tag{3.8}$$

At this point, all fundamental values for checking the originally assumed variables are available. The next sections show methods of calculating the enthalpy of saturated air (i_{swm}) at the assumed water-film and the corresponding dry bulb temperatures, resulting from this enthalpy and 100 % relative humidity. These new water - film and previously calculated tube temperatures must be compared with those assumed at the outlet. If not within acceptable tolerance, an iteration can be performed on both variables.

The subcooled fraction of the exchanger is compared with the original assumption. If tolerance is not met, iteration becomes necessary. These calculations complete the analysis of the subcooled region. It should be remembered that at any time a coil may become wet somewhere in the direction of air flow through the coil. Therefore, the initialization of dry portion for the subcooled region must be updated for each iteration. The model uses the outlet properties of the subcooled region as inlet properties of the two phase region.

The simulation of a two phase region is equivalent in form of that the subcooled region. The major difference is in evaluation of the heat transfer coefficient.

The superheat calculations follow a different form than the subcooled or two phase calculations if superheat degree is not known. The reason for the change of analysis is because the superheated region does not have a known capacity. In other words, in the subcooled or two phase regions, the capacity was known, which would be calculated from an enthalpy balance.

But in the superheated region the refrigerant outlet statepoint is not known at the beginning .

In the superheated region, determining capacity and evaporator outlet conditions ends the calculations. Finally, the refrigerant side pressure drop is evaluated. If the calculated pressure drop is not equal to the assumed value, the pressure drop is updated and the calculations are repeated until they match.

3.2 Condensing Unit Design

3.2.1 Condenser design

Heat is transferred in an air-cooled condenser in three main phases: (1) Desuperheating (sıperheated region), (2) Condensing (two phase flow), and (3) Subcooling (single phase). Desuperheating and subcooling zones vary 5 to 10 %, depending on the inlet gas temperature and the outlet liquid temperature. Condensing takes place in approximately 85 % of the condenser area at a constant temperature. The drop in the condensing temperature is the result of the fraction loss through the condenser coil.

Heat transfer on the refrigerant side is complex. The refrigerant enters the condenser as superheated vapor and by single phase convection cools to the saturation temperature after which condensation starts. If the tube wall temperature is lower than the saturation temperature, condensation can occur while the vapor is superheated. Eventually, all the vapor condenses to liquid. In most condensers, the condensed liquid is subcooled before leaving the condenser in a single phase convection process.

The simulation of a condenser coil is equivalent in form to the evaporator coil. Therefore it is not repeated here. The major differences occur in the evaluation of the condensing heat transfer coefficient and air side heat transfer coefficient for dry surface condition, having three different regions.

3.2.2. Compressor selection

In this study, while making the selection of the compressor, instead of the capacity curves of a special hermetic compressor such, it is aimed to be reached the capacity curves of hermetic compressor for a general calculation. Therefore the definitions of total heat rejection and cooling capacity are used.

Condensers should be selected for the total heat rejection which equals the sum of the net refrigerating capacity at the evaporator and the heat of compression added to the refrigerant gas by the comptressor. Thus the total heat rejection can be calculated as :

$$\text{THR} = \text{Cooling Capacity} \times \text{THR Factor} \qquad (3.9)$$

Table 3.1 can be used to estimate THR by using the THR factor from this table in the formula.

Table 3.1 THR factors for the suction cooled hermetic compressors

	Condensing Temperature (°C)					
Evap.Temp.	32	38	43	49	54	60
-9.4 °C	1.28	1.32	1.35	1.40	1.46	1.52
-6.7 °C	1.26	1.29	1.33	1.37	1.43	1.49
-3.9 °C	1.24	1.27	1.31	1.35	1.40	1.45
1.1 °C	1.22	1.25	1.28	1.32	1.37	1.42
4.4 °C	1.18	1.21	1.24	1.27	1.31	1.35
10.0 °C	1.14	1.17	1.20	1.23	1.26	1.39

3.3 Effectiveness - NTU Method of The Heat Exchanger Evaluations

The relationships that determine overall heat transfer coefficients can be used in the Effectiveness - NTU method to evaluate heat exchanger designs. This method allows, rigorous, and easily understandable technique for the analysis of heat exchangers. The basis of the scheme is a dimesionless quantity, NTU, which describes the heat transfer " size " or effectiveness of a particular exchanger in terms of readily available parameters.

A calculative procedure for determining the output of an exchanger having refrigerant flowing inside the tubes and air on the outside surface is to define the exchanger geometry, flow rates, and inlet conditions and then calculate the outlet conditions that depend on the effectiveness of the exchanger to transfer heat from the hot fluid to the cold fluid.

If $C=\dot{m}\ c_p$ for the fluid, then, from a thermodynamic point of the view, the exchanger effectiveness is compared to the actual heat transfer rate to the thermodynamically limited, maximum possible heat transfer rate achieved only in a counterflow exchanger of infinite heat transfer area, and

$$E=\frac{Q_{act}}{Q_{max}} \qquad (3.10)$$

The thermodynamic limit on the maximum heat that may be transferred is :

$$C_c<C_h \Rightarrow \quad Q_{max}=C_c\left(T_{hin}-T_{cin}\right) \qquad (3.11)$$

$$C_{hf}<C_c \Rightarrow \quad Q_{max}=C_{hf}\left(T_{hin}-T_{cin}\right) \qquad (3.12)$$

The number of heat transfer units is a nondimensional term for heat transfer " size "of the exchanger called NTU and defined ($U\ A\ /\ C_{min}$), where C_{min}=

minimum flow stream capacity rate of C_{hf} and C_c. The Effectiveness - NTU method can be used to evaluate heat exchangers in both single and two phase regions. However, the technique is modified to account for heat transfer to a constant - temperature fluid, such as in evaporators and condensers.

When NTU and C_{min} / C_{max} (= C) are known, the expression for the exchanger effcetiveness of a cross - flow exchanger with both fluids unmixed is [8],

$$E = \frac{1-\exp(-NTU(1-C))}{(1-C\exp(-NTU(1-C))) \quad (1+0.047C)NTU^{0.036C}} \tag{3.13}$$

for sensible heat transfer to the refrigerant. Unmixed refers to the degree of channeling the fluid in its traverse through the exchanger. Then Q_{act} can be calculated as,

$$Q_{act} = E\,Q_{max} = E\,C_{min}\left(T_{hin} - T_{cin}\right) \tag{3.14}$$

The actual heat transferred in the single phase portion of the exchanger is determined from the geometric considerations and the inlet temperatures of both fluids.

Now

$$Q_{act} = m_r \Delta\, i_r = m_a\, c_{pa}\left(T_{ain} - T_{aout}\right) \tag{3.15}$$

Thus

$$T_{aout} = T_{ain} - Q_{act}/m_a\, c_{pa} \tag{3.16}$$

and knowing the entering refrigerant temperature, pressure and enthalpy

$$i_{rout} = i_{rin} + Q_{act}/m_r \tag{3.17}$$

In condensation and evaporation analysis, the refrigerant is a constant - temperature fluid. Conceptually, this would correspond to an infinite flow rate of the refrigerant. Therefore, the refrigerant is always C_{max} and furthermore $C_{min} / C_{max} = 0$ and

$$E = 1 - \exp(-NTU) \tag{3.18}$$

An energy balance now provides that

$$Q_{act} = E\, Q_{max} \tag{3.19}$$

$$Q_{act} = m_a\, c_{pa} \left(T_{aout} - T_{ain}\right) \tag{3.20}$$

$$Q_{max} = m_a\, c_{pa} \left(T_r - T_{ain}\right) \tag{3.21}$$

Therefore,

$$T_{aout} = T_r + \left(T_{ain} - T_r\right) \exp(-NTU) \tag{3.22}$$

3.4 Heat Transfer Coefficients

An analysis of the heat transfer capacity of a heat exchanger depends on three basic parameters. The first is the thermal driving potential, or log-mean temperature difference (LMTD). The second parameter consists of the geometric properties of the exchanger overall size, fins per inch, etc. The last one is the heat transfer coefficient or resistance to heat flow is a third controlling variable. The heat transfer equation can be written as

$$Q = (UA)LMTD \tag{3.23}$$

where

$$\frac{1}{UA} = \frac{1}{h_{is}A_{is}} + \frac{\Delta x}{k_{tube}A_{tube}} + \frac{1}{h_{os}A_{os}\eta_f} \qquad (3.24)$$

It is assumed that only the external tube surface is finned, since this is the general practice in commercial HVAC heat exchangers.

3.4.1. Forced convection coefficient for air side

The air side average heat transfer coefficient can be determined from the following correlation of Colburn j factor, Prandtl number and air flow properties.

$$h_{os} = j\,G_a\,c_{pa}\,Pr^{-2/3} \qquad (3.25)$$

where G_a is the air mass flux through the minimum flow area.

Research by McQuiston [3] has resulted in the correlation of plate-fin tube transport data that account for geometric variables as well as hydrodynamic effect. The Colburn heat transfer j factor is determined by using the correlations developed by McQuiston.

The ratio of total surface area to the outside surface area of the tubes without fins, A/A_t can be calculate as,

$$\frac{A}{A_t} = \frac{4}{\pi}\frac{S_L}{D_h}\frac{S_T}{D}\sigma \qquad (3.26)$$

where

S_L : Longitudinal tube spacing

S_T : Transverse tube spacing

D_h : Hydraulic diameter

σ : Ratio of minimum free flow area to frontal area

The following correlating parameter, JP, for use with the j factors was obtained [3]:

$$JP = Re_D^{-0.4}\left[\frac{4}{\pi}\frac{S_L}{D_h}\frac{S_T}{D}\sigma\right]^{-0.15} \qquad (3.27)$$

The j factors can be calculated as,

$$j = 0.0014 + 0.2618(JP) \qquad (3.28)$$

If the row effect is taken into account, the following equation can be used for surfaces with 4 rows of tubes.

$$\frac{j_n}{j_4} = \frac{1 - 1280 N_r Re_{SL}^{-1.2}}{1 - 5120 Re_{SL}^{-1.2}} \qquad (3.29)$$

where

j_4 is expressed by equation (3.28)

j_n is j factor corresponding to N_r

N_r is number of rows of tubes

Re_{SL} is Reynolds number based on the longitudinal tube spacing

The general correlation for wet surface condition is modification of equation 3.28.

$$J(s) = 0.95 + 4.10^{-5}\, Re^{1.25}\left(\frac{s}{s-y}\right)^2 \qquad (3.30)$$

where

s : Fin spacing

y : Fin thickness

The Colburn j factor for wet surface condition ,

$$j = 0.0014 + 0.2618(JP)J(s) \tag{3.31}$$

3.4.2 Convection coefficient for internal flow

For single phase [8];

$St\,Pr^{2/3}=1.10647\,Re^{-0.78992}$	$Re \le 3500$	(3.32)
$St\,Pr^{2/3}=3.5194\,Re^{1.03804}$	$3500<Re \le 6000$	(3.33)
$St\,Pr^{2/3}=0.0108\,Re^{-0.1375}$	$Re>6000$	(3.34)

The following heat transfer coefficient has been derived by an integral method to quantify the average evaporative heat transfer coefficients as a function of inlet and outlet quality (Tong 1975) [8].

$$h_i = 0.01869 \frac{k_l}{D_i^{0.2}} \left(\frac{G}{\mu_l}\right)^{0.8} Pr_l^{0.4} \left(\frac{\rho_l}{\rho_v}\right)^{0.375} \left(\frac{\mu_v}{\mu_l}\right)^{0.075} \left(\frac{x_e - x_{in}}{x_e^{0.325} - x_{in}^{0.325}}\right) \tag{3.35}$$

A condensing fluid also has a heat transfer coefficient that is a function of quality (Hiller ans Glicksman, 1976). Here, with X_{tt} defined as [8],

$$X_{tt} = \left(\frac{\mu_l}{\mu_v}\right)^{0.1} \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \tag{3.36}$$

There is a function of X_{tt} that is,

$$F(X_{tt}) = 0.15\left[\left(X_{tt}^{-1} + 2.85X_{tt}\right)^{-0.476}\right] \tag{3.37}$$

and F_2 is defined within ranges of Reynolds numbers based on liquid properties as,

$$F_2 = 0.707\,Pr_l\,Re_l^{0.5} \qquad Re_l \le 50 \tag{3.38}$$

$$F_2 = 5Pr_l + 5Ln\left[1 + Pr_l\left(0.09636\,Re_l^{0.585} - 1\right)\right] \qquad 50 < Re_l \le 1125 \tag{3.39}$$

$$F_2 = 5Pr_l + 5Ln\left(1 + 5Pr_l\right) + 2.5Ln\left(0.00313\,Re_l^{0.812}\right) \qquad Re_l > 1125 \tag{3.40}$$

Then

$$h_{ix} = \frac{k_l\,Pr_l\,Re_l^{0.9}\,F(X_{tt})}{D_{is}F_2} \tag{3.41}$$

Twenty quality increments are recommended for this calculation to achieve sufficient accuracy, and then

$$h_{iavg} = \frac{1}{x_i - x_e}\sum_{x_e}^{x_i}\frac{1}{h_{ix}}\Delta x_r \tag{3.42}$$

3.5 Wet Surface Coil Analysis

An evaporator may transfer not only sensible heat from air to the refrigerant but also latent heat form of dehumidification. A predictive mathematical tool is complicated by this moisture condensation mechanism. Heat transfer driving force is no longer a simple LMTD relationship but must be based on an effective or log mean enthalpy difference.

An evaporator generally receives refrigerant in the two phase region. It then discharges this refrigerant as saturated vapor, depending on the refrigerant control mechanism. Energy balance relationships therefore, must be provided to calculate heat transfer to a constant temperature medium as well as to a single-phase fluid.

In the two region, the following equation can be used to ascertain the air-outlet state point when transferring heat to constant temperature refrigerant:

$$i_{aout} = i_{sr} + \left(i_{ain} - i_{sr}\right) \exp\left(\frac{-U_{ow}\, A_t}{\dot{m}_a\, cp_a}\right) \quad (3.43)$$

Total heat transferred from air is:

$$Q_t = \dot{m}_a \left(i_{ain} - i_{aout}\right) \quad (3.44)$$

It must be remembered that evaluation of $b_{w,m}$ and b_r began with an assumption of the pipe-wall and water film temperature.

The next equations can be used to check assumptions for these temperatures. If they are sufficiently in error, an iterative procudure is used. Firstly log mean enthalpy difference is determined.

$$\Delta i_m = \frac{i_{ain} - i_{aout}}{\ln\left(\frac{i_{ain} - i_{sr}}{iout - i_{sr}}\right)} \quad (3.45)$$

Then

$$i_{am} = i_{sr} + \Delta i_{am} \quad (3.46)$$

and

$$T_p = T_r + \frac{U_{ow}A_t}{h_{is}A_{is}c_{pa}}\left(i_{am} - i_{sr}\right) \tag{3.47}$$

Finally, from known quantities [8]

$$i_{swm} = i_{am} - \frac{h_{ow}\eta_{ow}}{b_{wm}h_{is}}\left(1 - \frac{b_r U_{ow}A_{tube}}{h_{is}A_{is}}\right)\left(i_{am} - i_{sr}\right) \tag{3.48}$$

Now $T_{w,m}$ can be determined. If T_p and $T_{w,m}$ are within acceptable tolerance of the assumed values the calculation is complete.

Under certain circumstances, a cooling coil can operate with one or more rows of its external surface dry and with the remainder of its surface wet. When applied to inlet conditions, if $T_{w,m}$ is higher than the inlet dew point temperature T_{dp1}, the initial surface will be dry. Dry analysis can be used for this portion of the coil. The mean fin temperature can be calculated as [8] :

$$T_{fm} = T - \eta_f\left[1 - \frac{U_{os}A_{os}}{h_{is}A_{is}}\right]\left(T - T_r\right) \tag{3.49}$$

When $T_{f,m}$ is equal to the inlet air dewpoint temperature,

$$T = \frac{T_{dp1} - \eta_f\left(1 - \frac{U_{os}A_{os}}{h_{is}A_{is}}\right)T_r}{1 - \eta_f\left(1 - \frac{U_{os}A_{os}}{h_{is}A_{is}}\right)} \tag{3.50}$$

Thus, the $T_{f,m}$ equation allows calculation of the mean temperature for dry section of coil when the air dry-bulb temperature and refrigerant temperature are known. The last equation allows calaculation where condensation just begins, even though the bulk air temperature is above the dew point. The analysis for a wet coil should begin here.

3.6 Pressure Drops For Air Side And Refrigerant Side

3.6.1. Air side

Pressure drops for flow of gases over a bank of tubes can be calculated with the following equation, expressed in Pascals: [25]

$$\Delta P_a = \frac{2 f' G_{max}^2}{\rho} \left(\frac{\mu_w}{\mu_b}\right)^{0.14} \tag{3.51}$$

Where G_{max}: Mass velocity at minimum flow area

ρ : Density evaluated at free stream conditions

N : Number of transverse rows

μ_b : Average free stream viscosity

μ_w : Viscosity evaluated at wall conditions

The empirical friction factor f' is given by McQuiston [3] as

$$FP = Re_D^{-0.25} \left(\frac{A/A_t}{(S_T - 2R)P_s + 1} \right)^{0.25} \left[\frac{(S_T - 2R)P_s}{4(1 - P_s y)} \right]^{-0.4} \left[\frac{S_T}{2R_h} - 1 \right]^{-0.5} \tag{3.52}$$

For dry surface,

$$f' = 4.094 x 10^{-3} + 1.382 (FP)^2 \tag{3.53}$$

For wet surface,

$$F(s) = \left(1 + Re_s{}^{-0.4}\right)\left(\frac{s}{s-y}\right)^{1.5} \qquad (3.54)$$

$$f' = 4.094x10^{-3} + 1.382(FP.F(s))^2 \qquad (3.55)$$

3.6.2. Refrigerant side

Equations given below are used for evaluation of the refrigerant side pressure drop in multipass, plate fin type coils common in HVAC Industry.

Single phase frictional pressure drop calculations are reasonably well understood and amply documented to permit accurate evaluation for any Newtonian fluid. The frictional pressure drop relationship always takes the form:

$$\Delta P = 4f'\frac{L}{D}\rho\frac{V^2}{2} \qquad (3.56)$$

Acceleration pressure drop follows the form:

$$\Delta P_{ac} = vG^2 \qquad (3.57)$$

These relationships are valid for either liquid or vapor flow, but not two phase flow. The recommended method for evaluating two-phase pressure drop in coils with freon refrigerant flowing in the tubes is a relatively simple analysis, developed by J.R.S.Thom, based on earlier work by Martinelli and Nelson.

The general equation for two-phase pressure drop neglecting gravity contributions is[9]:

$$\Delta P_{TP} = v_l G^2 r_{2,x_{out}} + \frac{4f'(L+L_u)}{2D} v_l G^2 r_{3,x_{out}} - v_l G^2 r_{2,x_{in}} - \frac{4f'L_u}{2D} v_l G^2 r_{3,x_{in}} + \frac{4f'L_e}{2D} v_l G^2 r_{5,x_m} \tag{3.58}$$

Where v_l : Specific volume of liquid

L : Tube length

G : Fluid mass velocity

L_u : Length of the tube required to change the fluid from saturated liquid to the actual inlet quality

$$L_u = \frac{x_{in} L}{x_{out} - x_{in}} \tag{3.59}$$

L_e : Equivalent length of return bends

f' : Mody friction factor based on a liquid Reynolds number

$$f' = 64/Re_D \qquad Re=<2300 \tag{3.60}$$

$$f' = 0.316 Re_D^{-0.25} \qquad Re=<2.10^4 \tag{3.61}$$

$$f' = 0.184 Re_D^{-0.2} \qquad Re>2.\ 10^4 \tag{3.62}$$

Fully developed flow and smooth surface conditions are assumed for evaluation of the friction factor.

r_2 : Acceleration multiplier (Table 3.2)

r_3, r_5 : Two phase friciton multipliers (Table 3.3, 3.4)

Table 3.2 Two phase acceleration multiplier (r_2)

	Specific Volume Ratio (ν_g / ν_f)				
Outlet quality	99	38	15	6.7	2.5
0.01	0.4125	0.2007	0.0955	0.0431	0.0132
0.015	0.6201	0.3034	0.1441	0.0649	0.0195
0.02	0.8325	0.4055	0.1930	0.0865	0.0261
0.03	1.268	0.6132	0.2903	0.1297	0.0392
0.04	1.710	0.8242	0.3886	0.1739	0.0562
0.05	2.169	1.040	0.4892	0.2182	0.0657
0.06	2.635	1.258	0.5893	0.2582	0.0787
0.07	3.116	1.479	0.6903	0.3076	0.0924
0.08	3.605	1.704	0.7940	0.3521	0.1055
0.09	4.110	1.934	0.8969	0.3975	0.118
0.10	4.620	2.165	1.001	0.4431	0.1319
0.15	7.365	3.379	1.538	0.6751	0.1996
0.2	10.39	4.678	2.100	0.9139	0.2676
0.3	17.30	7.539	3.292	1.412	0.4067
0.4	25.37	10.75	4.584	1.937	0.5495
0.5	34.58	14.30	5.968	2.490	0.6957
0.6	44.93	18.21	7.448	3.070	0.8455
0.7	56.44	22.46	9.030	3.678	0.9988
0.8	69.09	27.06	10.79	4.312	1.156
0.9	82.90	32.01	12.48	5.067	1.316
1.0	98.10	37.30	14.34	5.664	1.480

Table 3.3 Two phase friction multiplier (r_3)

	Specific Volume Ratio (ν_g / ν_f)				
Outlet quality	99	38	15	6.7	2.5
0.01	1.49	1.11	1.03	-	-
0.015	1.76	1.25	1.05	-	-
0.02	2.05	1.38	1.08	1.02	-
0.03	2.63	1.62	1.15	1.05	-
0.04	3.19	1.86	1.23	1.07	-
0.05	3.71	2.09	1.31	1.10	-
0.06	4.21	2.30	1.40	1.12	-
0.07	4.72	2.50	1.48	1.14	-
0.08	5.25	2.70	1.56	1.16	1.04
0.09	5.78	2.90	1.64	1.19	1.05
0.10	6.30	3.11	1.71	1.21	1.06
0.15	9.00	4.11	2.10	1.33	1.09
0.2	11.4	5.08	2.47	1.46	1.12
0.3	16.2	7.00	3.20	1.72	1.18
0.4	21.0	8.80	3.89	2.01	1.26
0.5	25.9	10.6	4.55	2.32	1.33
0.6	30.5	12.4	5.25	2.62	1.41
0.7	35.2	14.2	6.00	2.93	1.50
0.8	40.1	16.0	6.75	3.23	1.58
0.9	45.0	17.8	7.50	3.53	1.66
1.0	49.93	19.65	8.165	3.832	1.740

Table 3.4 Two phase friction multiplier for return bends(r_5)

	Specific Volume Ratio (v_g / v_f)				
Outlet quality	99	38	15	6.7	2.5
0.01	2.12	1.46	1.10	-	-
0.015	2.71	1.60	1.16	-	-
0.02	3.22	1.79	1.22	1.06	-
0.03	4.29	2.13	1.35	1.11	-
0.04	5.29	2.49	1.48	1.16	-
0.05	6.29	2.86	1.62	1.21	1.02
0.06	7.25	3.23	1.77	1.26	1.03
0.07	8.20	3.61	1.92	1.31	1.04
0.08	9.15	3.99	2.07	1.37	1.05
0.09	10.1	4.38	2.22	1.42	1.06
0.10	11.1	4.78	2.39	1.48	1.08
0.15	15.8	6.60	3.03	1.75	1.16
0.2	20.6	8.42	3.77	2.02	1.24
0.3	30.2	12.1	5.17	2.57	1.40
0.4	39.8	15.8	6.59	3.12	1.57
0.5	49.4	19.5	8.03	3.69	1.73
0.6	59.1	23.2	9.49	4.27	1.88
0.7	68.8	26.9	10.19	4.86	2.03
0.8	78.7	30.7	12.4	5.45	2.18
0.9	88.6	34.5	13.8	6.05	2.33
1.0	98.86	38.30	15.33	6.664	2.480

Pressure drop for condensation can be calculated as [10],

$$\Delta P_{tp} = \Delta P_{st} + \Delta P_{rb} \tag{3.63}$$

$$\Delta P_{st} = \Phi^2 \Delta P_L \tag{3.64}$$

$$X_{tt} = \left(\frac{1-x_m}{x_m}\right)^{0.9} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_v}\right)^{0.1} \tag{3.65}$$

$$\Phi = 1.467 - 0.51346 \mathrm{Ln} X_{tt} + 0.048789 (\mathrm{Ln} X_{tt})^2 \tag{3.66}$$

$$G_L = (1 - x_m) G \tag{3.67}$$

$$\Delta P_L = \frac{2 f' G_L^2}{\rho_e} \frac{L}{D} \tag{3.68}$$

$$\rho_m = \frac{1}{\frac{x_m}{\rho_v} + \frac{1-x_m}{\rho_l}} \tag{3.69}$$

$$\Delta P_{rb} = \frac{G^2}{2\rho_m} \tag{3.70}$$

where

ΔP_{rb} : Pressure drop in return bends

G_L : Mass velocity of the liquid

ρ_m : A constant depending on x_m

3.7 Fin Efficiency

Fin efficiency is defined as the ratio of actual heat transfer from the fin compared to the heat transfer, if the entire fin was at the temperature of the fin base. The equaitons for calculating the efficiency can be obtained analytically using modified Bessel functions, fin geometric parameters, and the convective heat transfer coefficient.

The fin efficiency is a function of two parameters:

$$F_1 = \left(R_{rat} - R_{tube}\right)\sqrt{\frac{h_{os}}{k_f y}} \qquad (3.71)$$

$$R_{rat} = \frac{R_f}{R_{tube}} \qquad (3.72)$$

$$F_2 = R_{rat} F_1 \qquad (3.73)$$

The fin efficiency for a circular fin of uniform thickness is calculated from two parameters in terms of modified bessel functions of the first and second kind.

$$\eta = \frac{2R_{rat}}{F_1(1-R_{rat}^2)} \frac{K_1(F_2)I_1(F_1) - I_1(F_2)K_1(F_1)}{K_0(F_2)I_1(F_1) - I_0(F_2)K_1(F_1)} \qquad (3.74)$$

Plate fin surfaces can be approximated as a circular fin using an effective circular fin diameter, D_f.

$$R_f = \sqrt{\frac{S_T S_L}{\pi}} \qquad (3.75)$$

Surface effectiveness is defined as the actual heat transfer to the fin and base divided by the heat transfer to the fin and base when the whole fin is at the base temperature T_{bs}. This can be written as,

$$\eta_s = 1 - \frac{A_f}{A_t}(1-\eta) \tag{3.76}$$

and

$$Q = hA_t\eta_s\left(T_{bs} - T_\infty\right) \tag{3.77}$$

In general, the equations for fin efficiency are based on standart assumptions of 1) steady state transfer, 2) constant fin and air properties, 3) constant temperatures at fin base, 4) one-dimensional heat conduction in the fin, 5) neglegible heat transfer from the tip, 6) uniform temperature of air stream, and 7) constant outside surface convection coefficient.

Chapter Four
THE PROGRAM ALGORITHM

4. THE PROGRAM ALGORITHM

The input parameters necessary for the algorithm are described below :

- Inlet temperatures for both the evaporator and condenser coils
- Air volumes for both inside and outside coils
- Outer and inner diameter of tubes
- Fin thickness
- Thermopyhsical properties of the tube and fin material
- Superheating and subcooling degrees
- Geometrical dimensions of indoor and outdoor coils

- Longitudinal and transverse tubes spacings

Calculations start from the evaporator. First, the total area for evaporator is calculated. The following calculations are made for the area. Then whether a subcooled region occurs in the evaporator coil is checked according to the given initial conditions. If the coil has a subcooled region, the evaporator design should begin from the subcooled region. At the outlet it is assumed that the subcooled region on the air side is dry, and the fraction of the exchanger area required to bring the refrigerant to saturated liquid is calculated. At this point the subcooled capacity is determined and the possibility that moisture condensation can occur from the air is evaluated. If the coil is really dry, leaving air temperature can be calculated by the NTU method. When the new and old values of the f_{sc} are compared to each other, if the difference between them is in an acceptable tolerance, calculations for two phase region can begin. If the coil surface is wet, wet surface calculation should be used. The tube and water film temperatures are calculated by using wet surface relationships. They are compared to the assumed values. After calculating f_{sc}, whether the difference between its new and old values is in the tolerance is checked. If the coil surface is partially dry, dry and wet surface analysis are applied separately.

Subcooled region ends and two phase region begins. Calculations are somewhat complicated than the subcooled region because the refrigerant in the two phase region does not always exit the evaporator totally vaporized. Therefore, it is assumed that coil is dry and that there is enough exchanger length to vaporize the refrigerant flow completely. The fraction of the exchanger area in the two phase region is the remainder of the coil after subcooling. Coil capacity is evaluated and the possibility of moisture condensation is determined. If the coil is wet, a wet analysis is performed to calculate whether there is sufficient exchanger area to vaporize the refrigerant completely. If there is, the superheated calculations can begin. If there is insufficient exchanger area to proceed to saturated vapor in a wet coil, the exchanger outlet quality is evaluated. No superheated region follows. If the coil remains dry, an evaluation of whether the dry capacity is sufficient to bring the refrigerant to saturated vapor should be carried out. The superheated region calculations are made according to the dry surface analysis. Then the total coil area is checked. Pressure drops for both air and refrigerant side are evaluated. If there is an acceptable difference between the evaluated value and assumed value for the refrigerant side pressure drop, calculations should continue by increasing the evaporating temperature until the evaporator capacity curve is obtained. Afterthat, similar calculations for the condenser coil, which has different area, are made and then the capacity curves are obtained. The system capacity is determined by means of the system balance chart which is plotted according to these evaporator and condenser coil curves.

Indoor conditions	: 27 °C(DB), 50 % RH
Outdoor conditions	: 35°C(DB), 50 % RH
Superheating degree	: 5 °C
Subcooling degree	: 5 °C
Tube diameter	: 10 mm
Fin thickness	: 0.12 mm
Tube / Fin	: Cu / Al
Arrangement	: Staggered
Longitudinal tubes spacing	: 22 mm
Transverse tubes spacing	: 25 mm
Air volumes	: 2500 m^3/h (outdoor unit), 1250 m^3/h (indoor unit)
Indoor coil geometry	: 14T-4R-386A-2.0P-7NC
Outdoor coil geometry	: 16T-4R-500A-2.5P-16NC

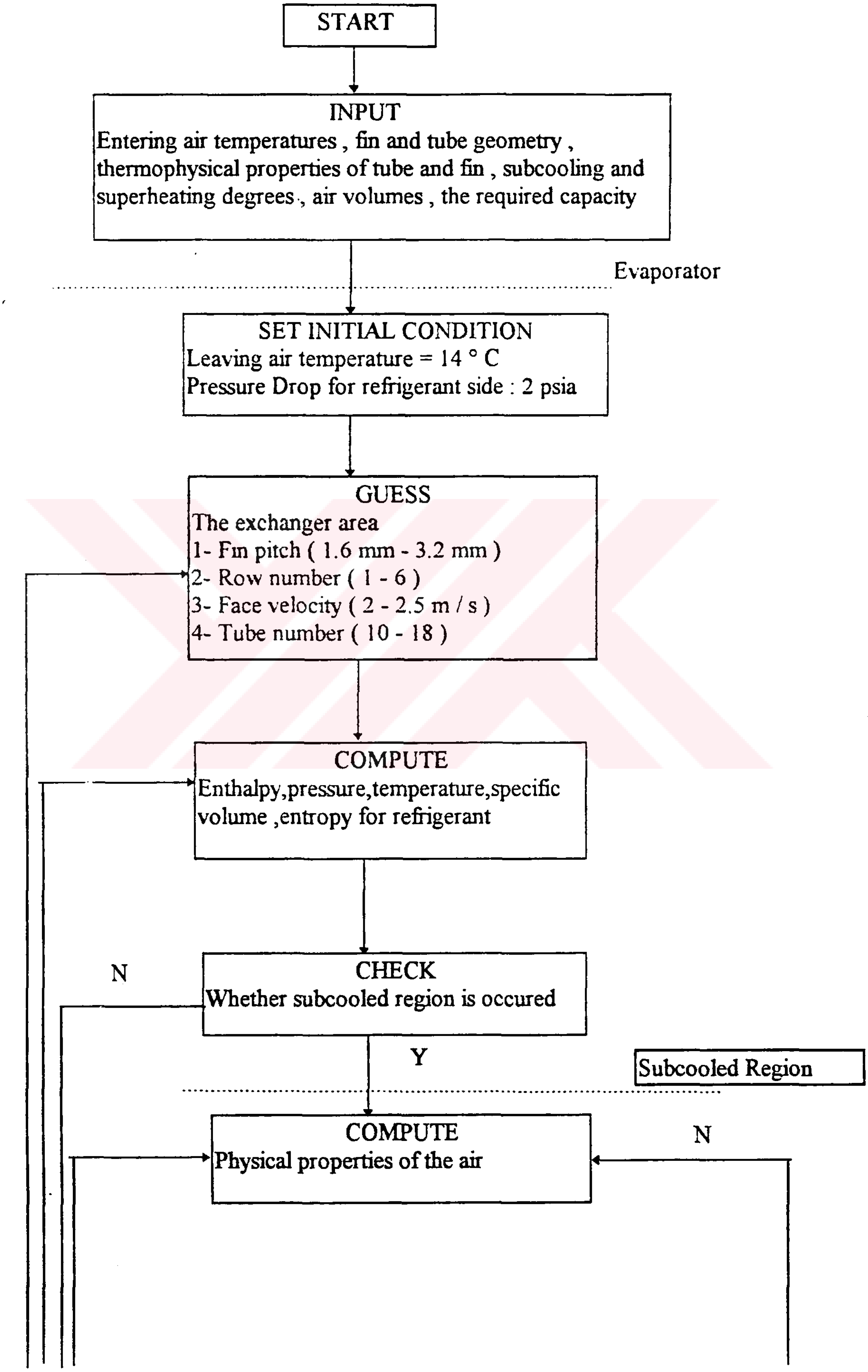
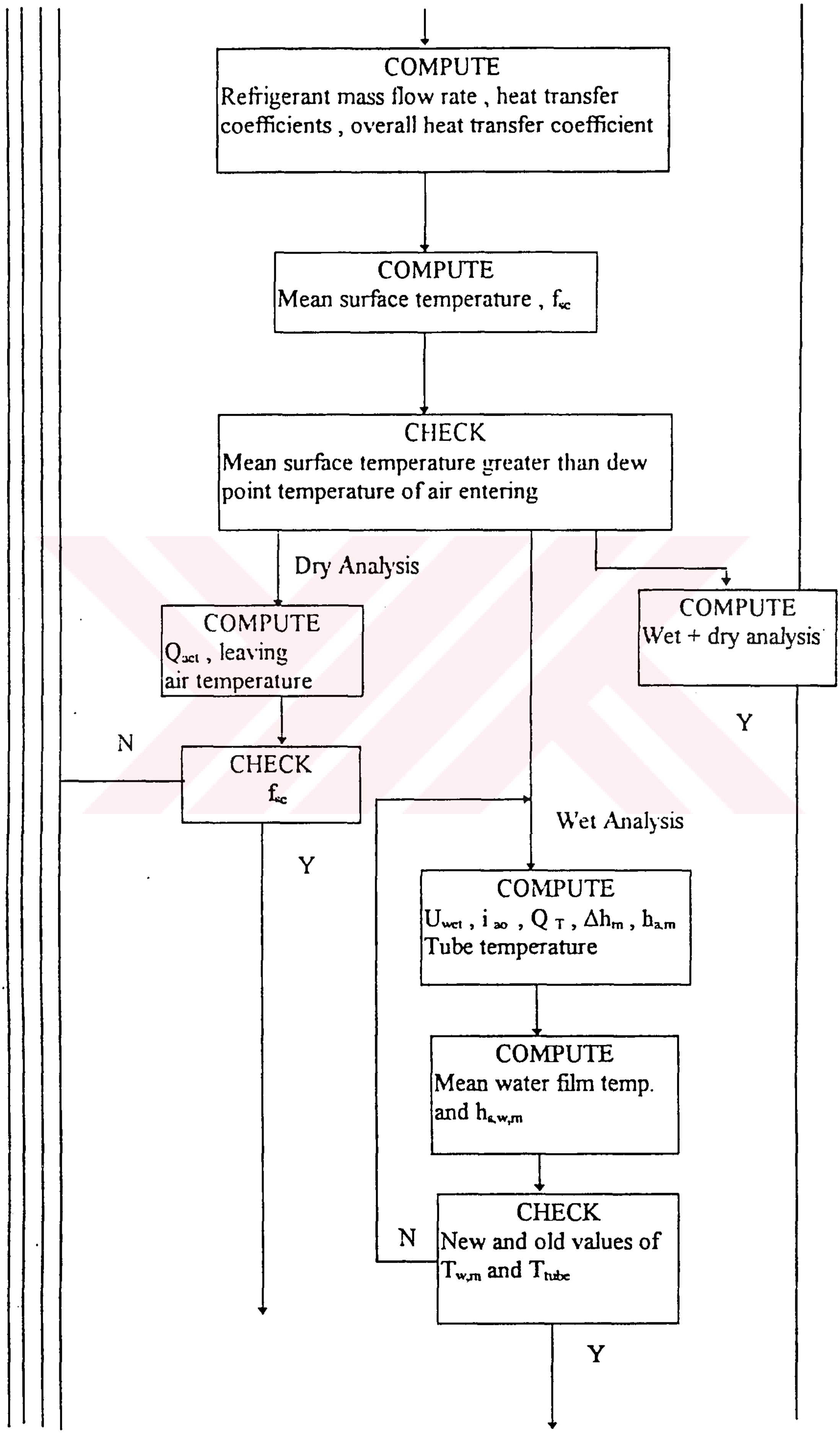
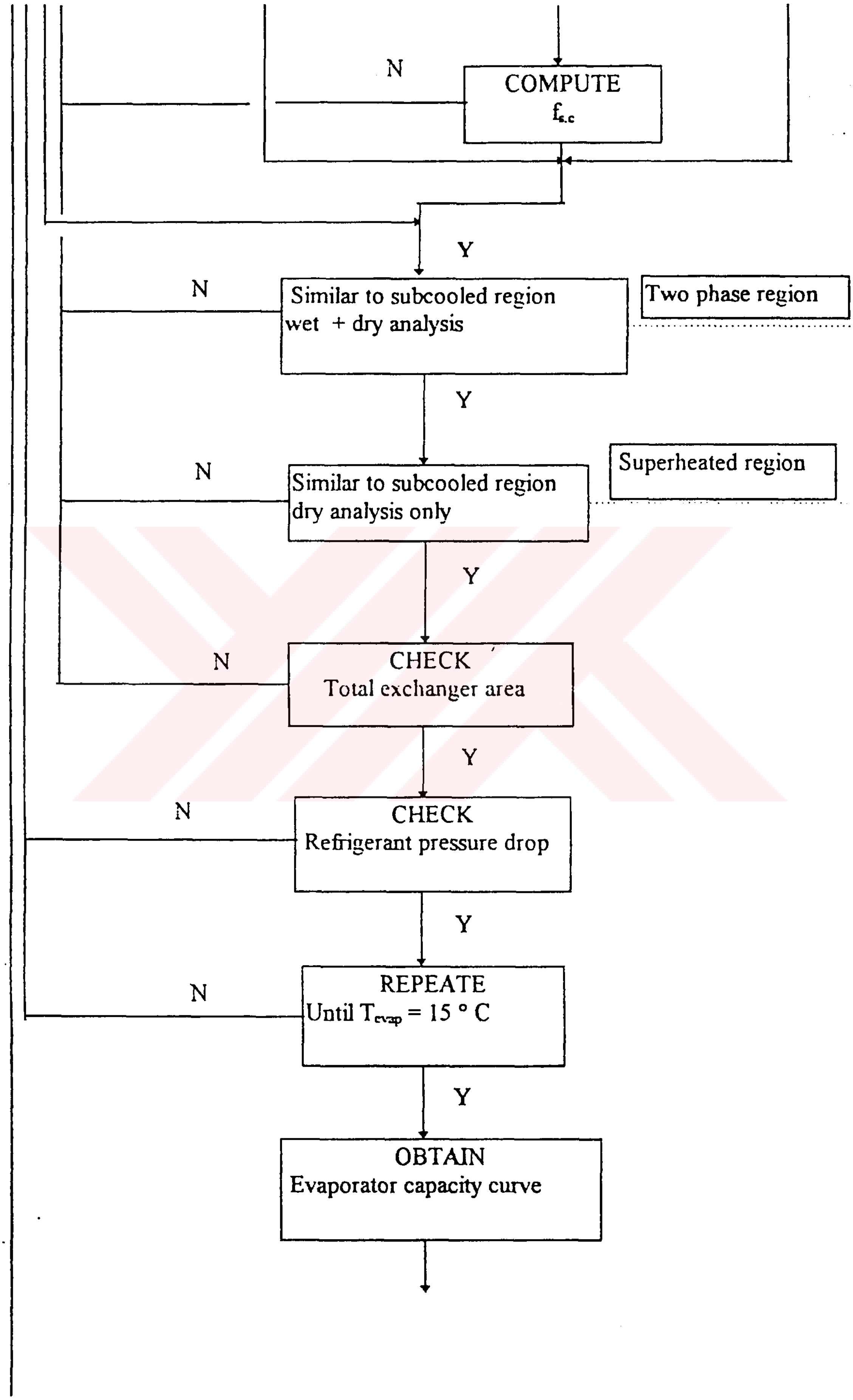
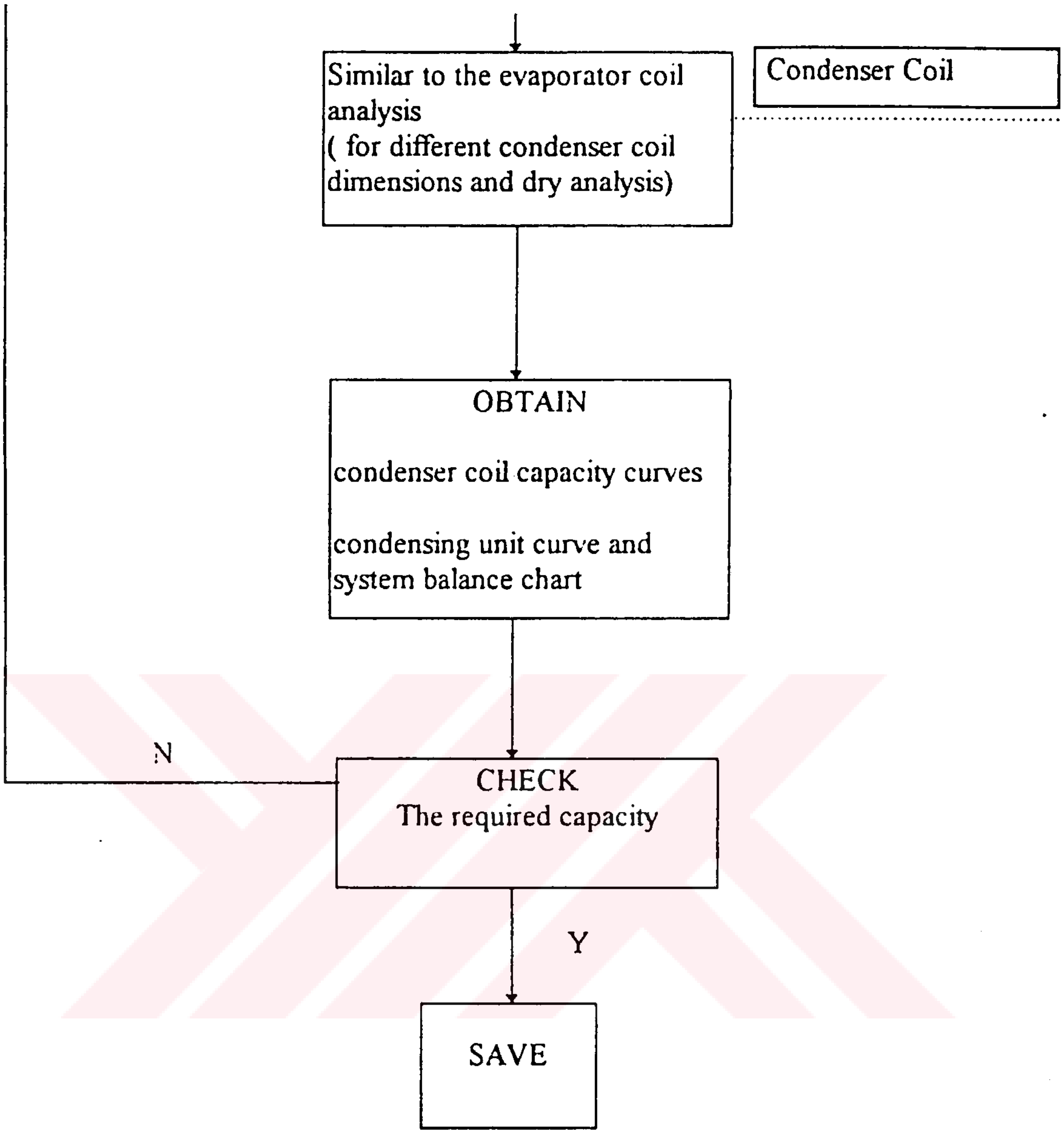


Figure 4.1 Flow chart


COMPUTE
Refrigerant mass flow rate , heat transfer coefficients , overall heat transfer coefficient
COMPUTE
Mean surface temperature , f_{sc}
CHECK
Mean surface temperature greater than dew point temperature of air entering
Dry Analysis
COMPUTE
Q_{act} , leaving air temperature
COMPUTE
Wet + dry analysis
Y
N
CHECK
f_{sc}
Y
Wet Analysis
COMPUTE
U_{wet} , i_{ao} , Q_T , Δh_m , $h_{a,m}$
Tube temperature
COMPUTE
Mean water film temp. and $h_{a,w,m}$
CHECK
New and old values of $T_{w,m}$ and T_{tube}
N
Y


N
COMPUTE
$f_{s.c}$
Y
N
Similar to subcooled region
wet + dry analysis
Two phase region
Y
N
Similar to subcooled region
dry analysis only
Superheated region
Y
N
CHECK
Total exchanger area
Y
N
CHECK
Refrigerant pressure drop
Y
N
REPEATE
Until T_{evap} = 15 ° C
Y
OBTAIN
Evaporator capacity curve


Similar to the evaporator coil analysis
(for different condenser coil dimensions and dry analysis)
Condenser Coil
OBTAIN
condenser coil capacity curves
condensing unit curve and system balance chart
N
CHECK
The required capacity
Y
SAVE

Chapter Five
CONCLUSIONS

5. CONCLUSIONS

One principal design aspect of an air-to air refrigeration system is the selection of the components, including the compressor, condenser, evaporator, refrigerant flow control device. Selection of the best and most economical components from a particular system requires a familiarity with the effects of changing operating conditions on the performance of each component.

The system balance charts are very helpful in checking the performance of a system. These charts are drawn with the temperature scale spaced. This spacing results in easier plotting because the compressor (or condensing unit) capacity lines and coil capacity lines are practically straight.

System balance charts which can be plotted according to the data given in Section 4.0 shows the balance conditions for the condenser - compressor - evaporator balance points at various ambient temperature and entering air temperature to evaporator. If the outdoor ambient temperature remains constant, two parameters which must be controlled to obtain the certain evaporator load are seemed. It can be found the values of the parameters which can be for the steady state condition on the chart.

This control can be automatic and for example can be controlled from the room temperature. The room thermostat can control several steps (generally four steps) of compressor capacity, and the dampers controlling the air flow over the coil can be controlled from the temperature of the air leaving the coil.

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