9)

Evaporative Air Conditioner for Automotive Application

by

Zuimdie Guerra

Submitted to the Department of Mechanical Engineering in Partial Fulfillment of the Requirements for the Degree of

MASTER OF SCIENCE
in Mechanical Engineering

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

May 1994

Author ....
Department of Mechanical Engineering
May 6, 1994

Certified by ....
Ernesto E. Blanco
Mechanical Engineering
Thesis Supervisor

Accepted by ....
Ain A. Sonin
Chairman, Graduate Thesis Committee
Mechanical Engineering

EVAPORATIVE AIR CONDITIONER FOR AUTOMOTIVE APPLICATION

by

ZUIMDIE GUERRA

Submitted to the Department of Mechanical Engineering on May 6, 1994 in the partial fulfillment of the requirements for the Degree of Master of Science in Mechanical Engineering

ABSTRACT

About 90% of the automobiles sold in the USA have an air-conditioning unit. The standard for this industry is the vapor-compressor system. This system has two limitations. First, the compressor requires about 10% of the engine's power. This consumption is reflected in a lower fuel economy and a higher pollution rate. Second, the system uses chlorofluorocarbon fluids as refrigerant. The fluorocarbons have raised concern about their role in the destruction of the global environment and their discontinuation is imminent. This issues has prompted an urgency in the scientific and engineering community to find new environmentally safe refrigerants and more efficient systems. This thesis proposes a theoretical design of an evaporative air conditioning system to address both issues.

The theoretical evaporative system designed satisfies the two main objectives with a lower fuel consumption, and less environmental pollution than that of the compressor system. The designed system is feasible to build but it exercises no commercial advantage over the vapor-compressor system in terms of cost, weight, and volume.

Thesis Supervisor: Prof. Ernesto E. Blanco
Title: Professor of Mechanical Engineering

Table of Contents

List of Figures

List of Tables

Chapter 1: Overview

1.1 Introduction

About 85% of all new vehicles sold in the USA have an air-conditioning (A/C) unit (Pierce 1975). There are approximately 170 million cars and light trucks accounting for 144.5 millions of vehicles with a vapor-compressor A/C units. The first limitation of this system is the compressor requirement of about 7% of the engine's power reflected in lower fuel economy and higher pollution rate. The vapor-compressor system consumes about 22.5 gal of typical annual fuel use. About 20 gal of gasoline are used for air-conditioning per 10,000 miles driven per year by the average driver. The additional 2.5 gal comes from the air conditioner weight of 25 lb (about 10 gal of gas are required per 10,000 miles for each 100 lb of weight). The 22.5 gal at 42 gallons per barrel represents 77.2 million barrels per year that are used nationwide on automotive air conditioning (Fischer 1990).

The other limitation of the system is its use of chlorofluorocarbon (CFC) fluids as the refrigerant. The fluorocarbon's discontinuation is imminent because of their role in the destruction of the global environment. The Montreal Protocol of 1987 calls for a 50% reduction in production of CFC-12, the refrigerant used in vehicles A/C systems, by major producers by 1998. The primary concern is the effect in ozone concentrations because all the chlorine of the chlorofluorocarbon gases is released into the stratosphere. The long lifetime of CFC-12, about 130 years, contributes to the rapidly increasing concentrations of that gas, .44 ppdv in 1988 increasing at a rate of 4% per year (Green 1989; Thurlow 1990). Also, chlorocarbons absorb infrared radiation in the window region of the atmospheric spectrum with potential to contribute significantly to increasing surface temperature. Some alternative refrigerants have been identified but their high global warming potential make them only an interim solution. An environmentally acceptable alternative cooling method has became important.

These issues have prompted the scientific and engineering community to look for alternative mobile air-conditioning (MAC) systems that are environmentally safe and efficient. Studies on alternative automotive air-conditioning are being done in work-actuated and heat-actuated systems. The most promising systems in terms of possible future implementation, for both categories, are discussed in Appendix A (Mei, Chen and Kyle 1992). The systems of the appendix, and the evaporative cooling system (in Section 1.2) are compared in Table I-1 based on the most relevant criteria. The comparison points out the Sterling cycle cooling system as the best candidate for future development among the work-actuated systems and the evaporative cooling system for the heat-actuated systems. Since the heat-actuated systems have a good energy saving potential and research and development for these systems is very rare, this thesis makes an exhaustive analysis of the evaporative system to study its viability for automotive applications.

This study will evaluate the system with a thermodynamical analysis to find the set of conditions that has to be met at each stage of the process to achieve the desired cooling effect. These conditions will be used on the preliminary design of the system's components to determine the feasibility for automotive applications. A cost analysis of the preliminary design will determine the feasibility for commercial applications.

Table I-1. Comparison of the Air Cooling Systems

Criteria \ System	Hermetic	Brayton	ROVAC	TE	Sterling	Ejector	Absorption	Desiccant	Evaporative	Hydride
COP	+	S	+	—	+	—	—	—	—	
Refrigerant	S	+	+	+	S	S	S	S	+	
Weight	+	+	+	+	+	+	S	+	S	DATUM
R&D	—	+	—	—	+	—	S	—	S	
Waste Heat	—	—	—	—	—	S	S	S	S	
Heat Pump	S	S	S	+	+	S	S	S	+	
Results:	2 + 2 S 2 -	3 + 2 S 1 -	3 + 1 S 2 -	3 + 3 -	4 + 1 S 1 -	1 + 3 S 2 -	5 S 1 -	1 + 3 S 2 -	2 + 3 S 1 -	

Coefficient of Performance (COP)
Research and Development (R&D)

Legend:
S - same as datum + - better than datum — - worst than datum

Table I-2. Global Warming Potential for Trace Greenhouse Gases. (Fisher 1991)

Trace gas *	Estimate Lifetime (year)	Global Warming Potential Integration Time Horizon (year) +	
		100	500
Carbon dioxide	---	1	1
Nitrous oxide	150	290	190
CFC-11	60	3500	1500
CFC-12	130	7300	4500
HCFC-134a	16	1200	420

* Chlorofluorocarbons (CFCs) and other gases do not include effects through depletion of stratospheric ozone.

\+ Changes in lifetime and variations of radiative forcing with concentration are neglected.

1.2 Evaporative Cooling System

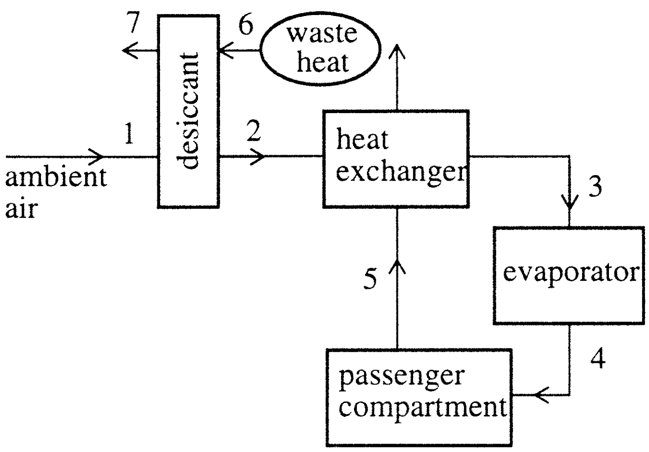


Figure 1-1. Schematic of the Evaporative Air Cooling System.

Cycle:

1→2 ambient air pass through a desiccant becoming hot and dry

2→3 the hot dry air goes to the heat exchanger, heat is removed by the air leaving the passenger compartment

3→4 the warm dry air is cooled and humidified by the evaporation of the water sprayed inside the evaporator

4→5 the cold humid air is delivered to the passenger compartment

6→7 the desiccant is regenerate applying waste heat to adsorb the environmental moisture trap inside the desiccant's lattice

The evaporative cooling system is characterized by the following criteria:

Advantages:

(a) uses water, an environmentally safe substance, as the refrigerant
(b) uses the waste heat of combustion, saving engine power
(c) has few components making it highly reliable and compact, easy to maintain, and easy to decompose in modules
(d) is quiet because it does not need a compressor
(e) has a low operating pressure (atmospheric)
(f) has a winter heater capability

Limitations:

(a) the water used has to be replaced regularly
(b) many pounds of water need to be carried reducing the fuel economy
(c) water needs to be drained in winter to avoid damage of the components
(d) relative low efficiency, but comparable with that of other alternative systems
(e) adsorption heat dissipation problem because of the relative low temperature differential with ambient temperature

Remarks:

The evaporative system is an open-cycle desiccant system with water as the cooling medium.

1.2.1 System Description

The evaporative system is a direct evaporative cooler because the working fluid, ambient air, is cooled directly by adding water to it. Because the system is a direct evaporative cooler, it has to be an open cycle. On an open cycle, the working fluid goes through the cycle once and then is thrown away (is never confined).

1.2.1.1 Operations

The series of operations that constitute the continuous open cycle of the evaporative cooling process include the following phases:

a) dehumidification
b) heat exchange
c) evaporation
d) regeneration

1.2.1.1.1 Dehumidification

The dehumidification process involves the removal of moisture from a stream of air to produce a warm and dry stream. The stream of air gets significantly warmer while it is dried because the moisture gives up its heat to leave the air, in order that the heat content of air remain constant. The air is dried to increase its water adsorption capability.

1.2.1.1.2 Heat Exchanging

The heat exchanging process involves the exchange of heat (without an exchange of moisture) between the dry-warm stream of air and another substance, to produce a colder stream of air. The dry-cooled air can become the equivalent of refrigerate supply air when it is humidified. If air is humidified without being cool by a heat exchanging process, it will revert to the conditions before dehumidification, no colder than it was.

1.2.1.1.3 Evaporation

The evaporation process involves the air conditioning (cooling and humidification) of air. The evaporation of water requires heat that is supplied by the stream of dry air. The dry air cools because it transferred its heat to the water, and becomes humid because it carries the water vapor.

1.2.1.1.4 Regeneration

Regeneration is a process in which the same path heats and cools a substance at different intervals. The process involves the heating of the drying material to liberate the moisture inside it and restitute its water absorption capability. The energy input utilized is the heat from a waste heat source.

1.2.1.2 Components

The evaporative system has four main components to handle the four operations that constitute the evaporative cooling cycle. The four components are:

(a) a desiccant bed
(b) a main stream heat exchanger
(c) a evaporator
(d) a regeneration heat exchanger

1.2.1.2.1 Desiccant Bed

The desiccant bed removes the moisture from the humid-warm ambient air when the air goes through it. The moisture changes from vapor to liquid liberating its internal heat, warming the dried air and the desiccant bed. Air is hot and dry after leaving the desiccant.

1.2.1.2.2 Main Stream Heat Exchanger

The main stream heat exchanger is used to dissipate some of the heat carried by the dry hot air that leaves the desiccant, with the returning air from the passenger compartment. The hot dry air passes through the compact heat exchanger where it has an air-to-air heat exchange with the warm air that leaves the passenger compartment. Heat flows from the hot air to the warm air. No moisture is exchanged between the two airs because they do not mix. The hot dry air is warm and dry after leaving the heat exchanger.

The evaporative cooling cycle is an open cycle because the passenger compartment air is discharged to the ambient air.

1.2.1.2.3 Evaporator

The evaporator produces a rapid evaporation with water spray nozzles to cool and humidify the air. The dry-warm air enters the evaporator and is sprayed with water. The water evaporates humidifying and absorbing heat from the warm air. The temperature is reduced without a change in the air's heat content because the heat is stored inside the water vapor.The air leaves the evaporator cold and humid.

The conditioned (cold and humid) air is delivered to the passenger compartment producing the desired cooling effect.

The evaporative system is a direct evaporative cooler because water is added to conditionate (cold and humid) the air.

1.2.1.2.4 Regeneration Heat Exchanger

The regeneration heat exchanger produces the hot air needed to regenerate the desiccant. The ambient air is heated by the exhaust hot gases. The desiccant, to be continuously effective, has to the regenerated. If the desiccant is not regenerated it will absorb water until it is saturated, after that the desiccant loses its dehumidification capability. The desiccant needs to be heated to release the water absorbed. Waste heat from the combustion process is used to heat the desiccant. The heat vaporizes the trapped water freeing it from the desiccant lattice.

After regeneration the desiccant needs to cool down, without exposure to humidity, to fully regain its absorption capability. A higher desiccant temperature reflects in a detriment of its initial absorbency.

Chapter 2: Thermodynamic Analysis

2.1 Thermodynamics of the Evaporative System

The thermodynamic analysis is a very important factor in the study of the feasibility of a system. The system has to prove it is thermodynamically feasible to be considered because the laws of thermodynamics must not be violated.

2.1.1 Fundamentals

The evaporative system is based in some basic knowledge related to thermodynamics such as the following:

a) humidity ratio
b) relative humidity
c) sensible heat
d) latent heat
e) heat transfer
f) mass transfer
g) dry-bulb temperature
h) wet-bulb temperature
i) enthalpy

2.1.1.1 Humidity Ratio

Humidity Ratio (or Specific Humidity) is the weight of moisture being held by a certain weight of dry air. The measure is independent of temperature. The ratio is dimensionless, the weight of moisture and that of dry air are expressed in identical terms and thus cancel out, leaving a pure ratio.

2.1.1.2 Relative Humidity

Relative humidity is defined as the measured percentage of moisture in a given sample of air compared with the maximum amount of moisture that the same sample of air can hold at the same temperature. The maximum amount of moisture (water vapor content) is known as "100% relative humidity". It occurs at the condensation point (water drop formation) for that temperature, known as "dew point".

The measure of the moisture is relative because the more heat there is in the air, the more moisture the air can hold. When the relative humidity is low, the air can absorb more moisture. Air has the tendency to absorb water until it reaches 100% relative humidity (RH).

2.1.1.3 Sensible Heat

Sensible heat refers to heat flowing into or from a substance changing only the temperature of the substance. The greater the flow of sensible heat, the greater the temperature change.

2.1.1.4 Latent Heat

Latent heat refers to a phase change that a substance goes through (solid-to-liquid or liquid-to-gas) and heat is added to the substance without raising the temperature of the substance. The heat stored in the substance is used to produce the phase change.

Latent heat is a reflection of the air's content of water vapor. When water evaporates, the vapor captures a related latent heat from the air. If water is removed, the latent heat that accompanies the vapor is released into the air.

2.1.1.5 Heat Transfer

Heat transfer is based on the principle that if two substances with differing temperatures are placed in close proximity to each other, the heat in the warmer substance will always transfer to the cooler substance until the temperature of both substances equalize.

The amount of heat transfer that occurs between the exchanging substances is measured in British Thermal Units (BTU). By definition, one BTU is the amount of heat required to raise the temperature of 1 lb. of water 1 degree Fahrenheit.

2.1.1.6 Mass Transfer

Mass transfer is parallel to heat transfer. On evaporative systems, mass transfer consists of the simple evaporation or condensation of water in air when exposed to a water surface. The water vapor flows away from the higher vapor pressure toward the lower until both vapor pressures equalize.

2.1.1.7 Dry-Bulb Temperature

The dry-bulb temperature is the air temperature as measured by the ordinary bare thermometers.

2.1.1.8 Wet-Bulb Temperature

The wet-bulb temperature is that indicated by thermometers whose bulbs are covered by wetted wicks exposed to rapidly moving air. This temperature represents the dry-bulb temperature of that air if its relative humidity were 100%. The wet-bulb temperature is thus lower than the dry-bulb temperature. The higher the air's original humidity, the lower the difference between both temperatures.

2.1.1.9 Enthalpy

Enthalpy is defined as the internal energy per unit mass. In this case enthalpy is the heat content per pound of air. Nondry air (air plus water vapor) behaves like an ideal gas therefore its enthalpy is a function of temperature only. When enthalpy is constant, the wet-bulb temperature is constant.

2.1.2 Psychrometric Chart

The theory of direct evaporative cooling is based on the standard psychrometric chart and theory of adiabatic saturation.

The psychrometric properties of nondry air are plotted on standard psychrometric charts. The psychrometric chart used for the thermodynamic analyses made for this thesis is shown in Figure 2-2.

Given the values of any two psychrometric properties, all the others can be found without computation. The chart arrangement is as follows (see Figure 2-1):

a) lines of constant dry-bulb temperature are vertical
b) lines of constant dew point temperature and humidity ratio(ω)are horizontal
c) lines of constant enthalpy(H) and wet-bulb temperature (coincident properties) slant diagonally down to the right
d) the saturation line of 100% relative humidity curves down from right to left; the temperatures in that line are the coinciding of dew point, dry-bulb, and wet-bulb temperatures in saturated air
e) curves of constant relative humidity(Φ) are spaced below the saturation line, generally parallel to it.
f) lines of constant specific volume are widely spaced and run downward from left to right more steeply than wet-bulb lines

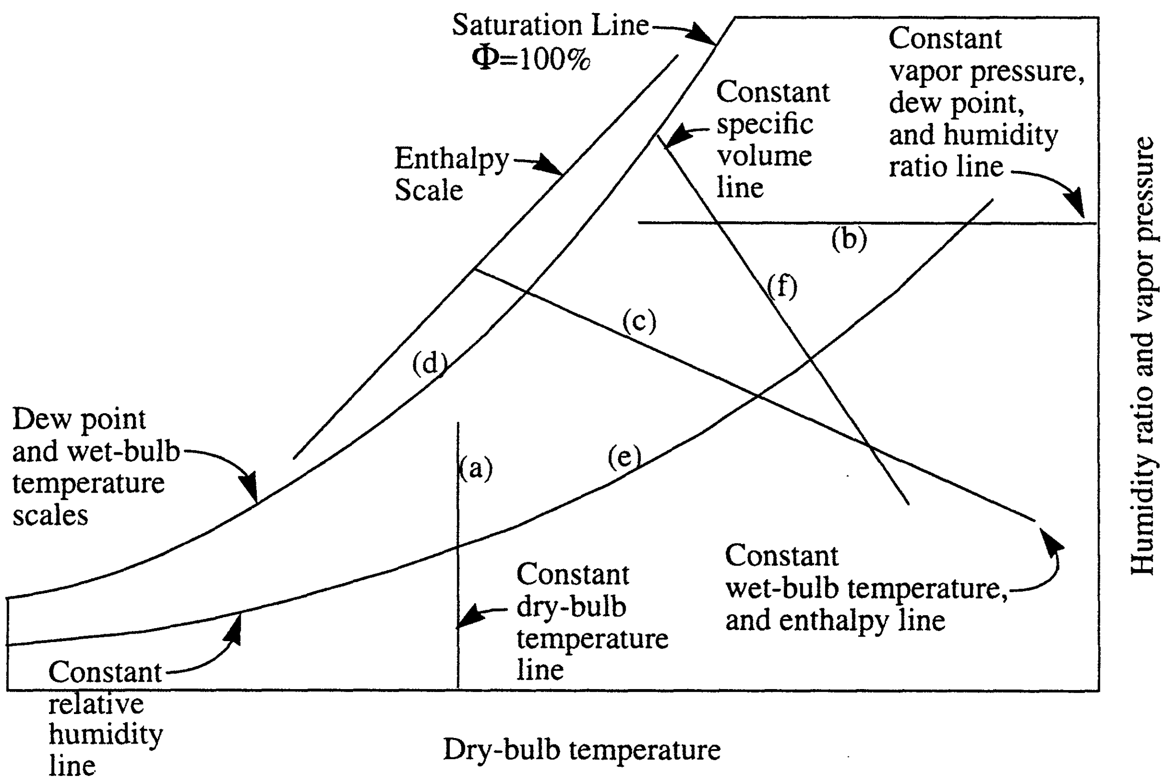


Figure 2-1. Skeleton Outline of the Psychrometric Chart

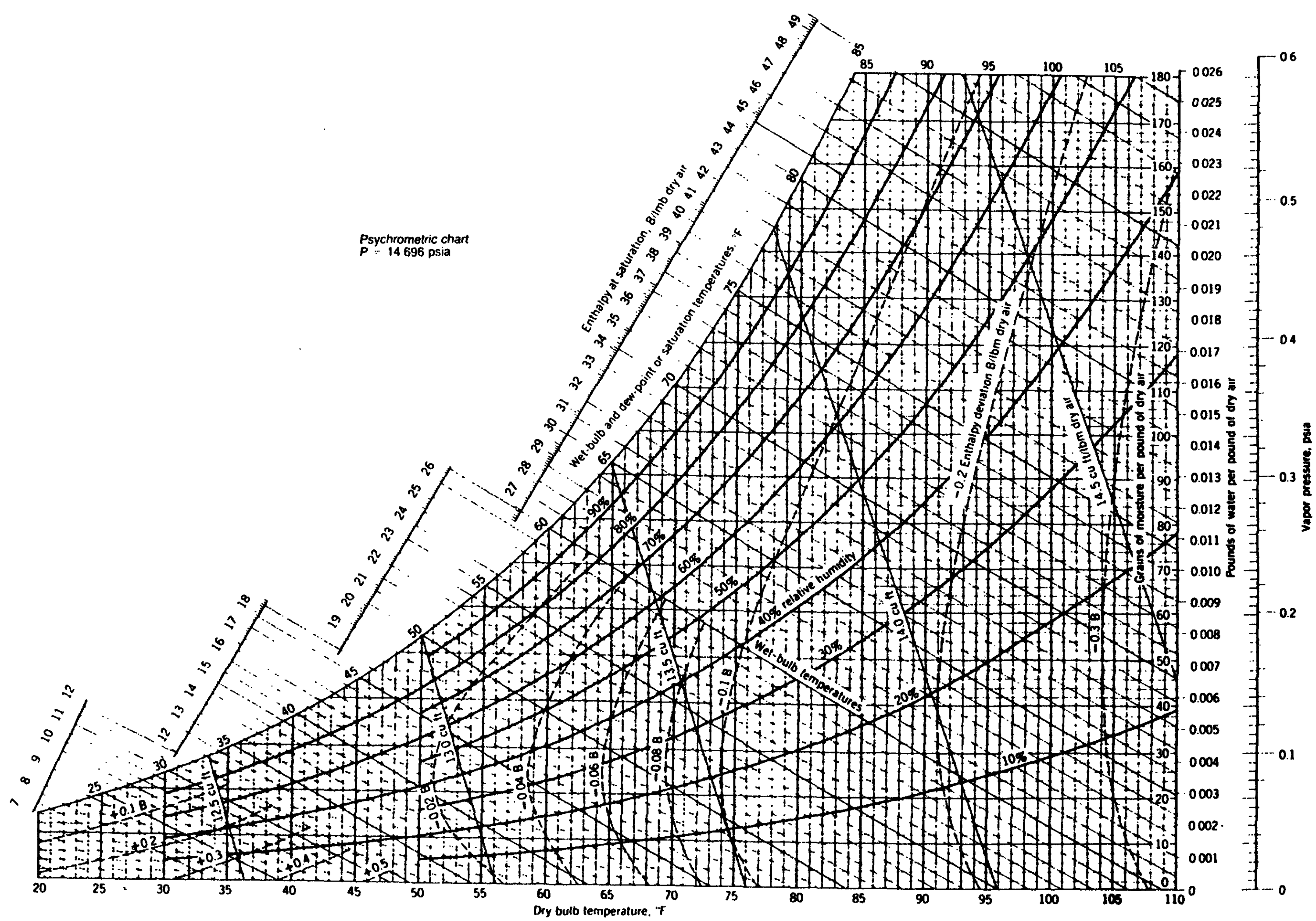


Figure 2-2. Psychrometric Chart (English Units). (Jones and Hawkins 1986)

2.1.3 Psychrometric Processes

Thermodynamical processes trace paths on the psychrometric chart. The processes that are followed by the cooling process of the evaporative system are:

a) heating
b) cooling
c) air-drying
d) ideal adiabatic saturation
e) ordinary adiabatic saturation

2.1.3.1 Heating and Cooling

A simple heating process moves horizontally to the right; a simple cooling process horizontally to the left. The dew point temperature and specific humidity remain constant for both (see Figure 2-3(a) and (b)).

2.1.3.2 Air-Drying

An air-drying process, using moisture-absorbing chemicals, moves diagonally downward to the right along constant wet-bulb or enthalpy lines. Those processes are normally adiabatic (no heat enters from other sources), without loss or gain of enthalpy. (see Figure 2-3(c)).

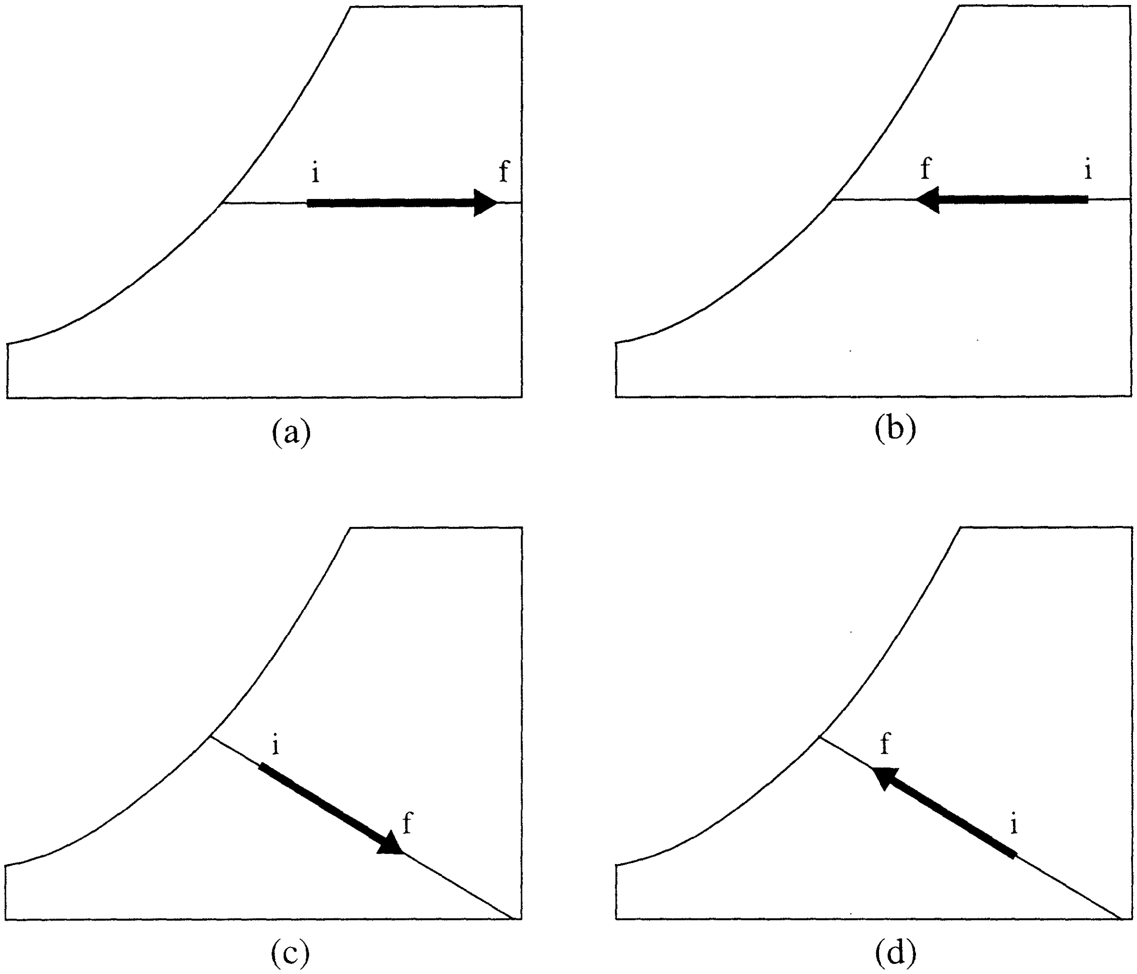
2.1.3.3 **Ideal Adiabatic Saturation** (Ideal Evaporative Cooling)

An ideal adiabatic saturation process moves diagonally upward to the left along constant wet-bulb or enthalpy lines. (see Figure 2-3(d)).

True adiabatic saturation occurs in processes where the initial water temperature approximates the entering air's wet-bulb temperature and are adiabatic (no heat enters from other sources, but the air temperature falls as its sensible heat is converted into latent heat to evaporate the water).All evaporation serves to cool the air, none to cool the water.

2.1.3.4 **Ordinary Adiabatic Saturation** (Ordinary Evaporative Cooling)

An ordinary adiabatic saturation process has two motions. It moves diagonally upward to the left along constant wet-bulb or enthalpy lines to a point indicated by the cooled air's dry-bulb temperature. Then, keeping the dry-bulb temperature constant, the process moves upward to the state point determined by other terminal conditions such as relative and specific humidity or wet- bulb temperature. The second motion represents the gain in the air's enthalpy, relative and specific humidity, and wet-bulb temperature resulting from cooling the water from its entering to final temperature. A complex process like this is illustrated by a straight line connecting its initial and final state points. The resulting process moves along an indeterminable path where air is cooled at increasing, not constant, enthalpy and wet-bulb temperature (see Figure 2-3(e)).



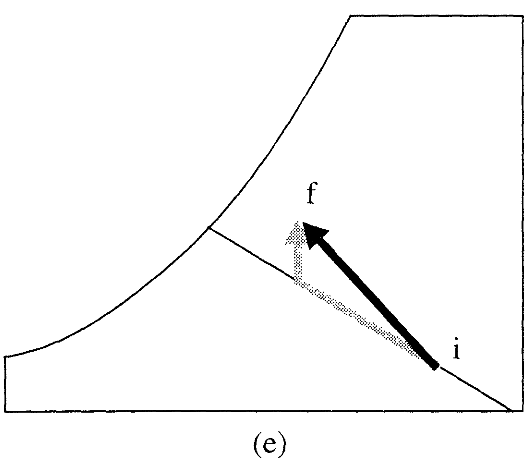


Figure 2-3. Skeleton Outline of the Psychrometric Processes.

(a) Simple heating process goes from its initial state (i) to its final state (f) moving right along the constant humidity ratio line.
(b) Simple cooling process goes from i to f moving to the left along the constant humidity ratio line.
(c) Air drying process goes from i to f moving downward to the right along constant wet-bulb or enthalpy line.
(d) Ideal adiabatic saturation process goes from i to f moving upward to the left along constant wet-bulb or enthalpy line.
(e) Ordinary adiabatic saturation process goes from i to f, first moving upward to the left along constant wet-bulb or enthalpy line, and then moving upward along the constant dry-bulb line to a final state determined by other terminal condition. The resulting process moves along an indeterminable path.

Ordinary adiabatic saturation differs from the ideal in that entering water introduces some sensible heat. In ordinary evaporative processes, the initial temperature of water is warmer than the air's dry-bulb temperature. Air and water, not air alone, are jointly cooled.

2.2 Ideal Evaporative Cooling Process

Thermodynamically the ideal evaporative cooling process is a complex process, consisting of the following:

(a) adiabatic dehumidification at constant enthalpy
(b) cooling at constant humidity
(c) adiabatic saturation at constant enthalpy
(d) constant humidity heating

All these processes are ideal; they follow a determinate path along the property lines of the psychrometric chart as shown on Figure 2-4.

The feasibility of the evaporative system will be determined by analyzing the ideal process. If it is feasible for the ideal case, then the analysis will be extended to a more real case.

2.2.1 Design Conditions

To analyze the evaporative cooling process, the desired initial and final states were defined. Because ambient air is the working fluid, the initial state is determined by the worst conditions expected for the incoming air. The conditions of the incoming ambient air are a temperature of 100°F and 10% relative humidity.

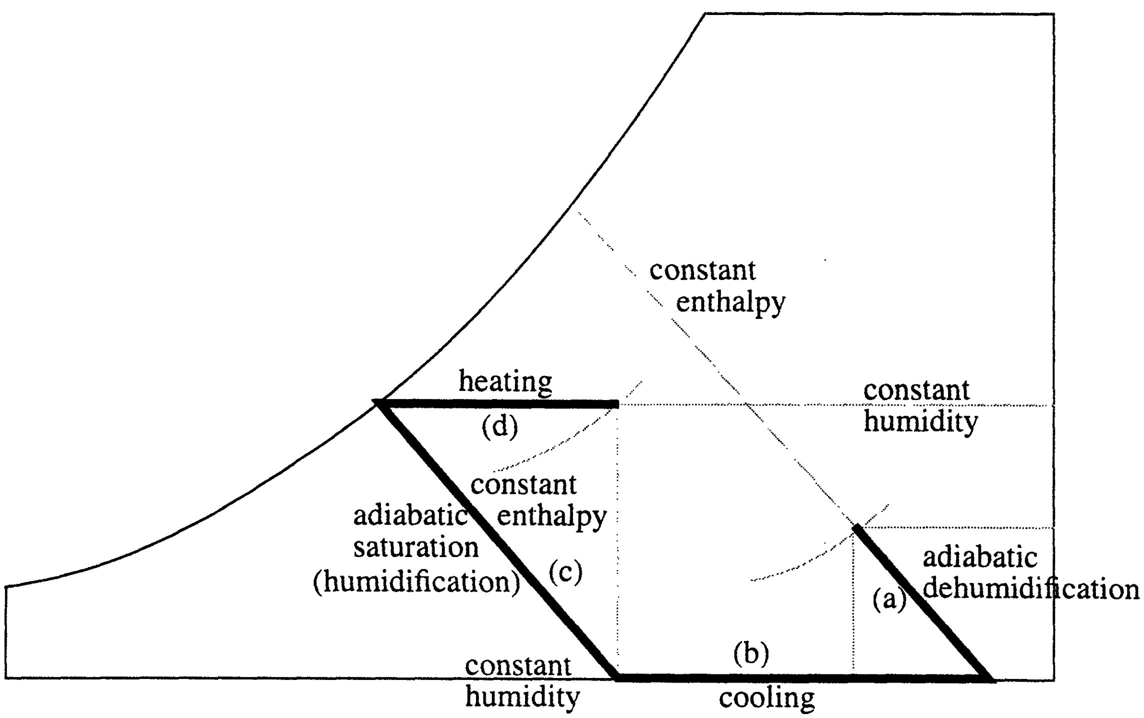


Figure 2-4. Skeleton Outline of the Ideal Evaporative Cooling Process.

The final state is determined by the desired cooling effect inside the passenger compartment. The cooling load used for this purpose is 13,680 Btu/h; the cooling requirement for a compact car with ambient conditions of 100 °F and 20% relative humidity (see Table II-1). The temperature of air at the final state is 77 °F with specific humidity equal to that of the saturate air leaving the evaporator.

2.2.2 Ideal States

In the ideal system all components are 100% efficient and the process is at steady state conditions (values do not change with respect to time) to simplify the analysis. The schematic of the ideal system with states for the ideal steady state process is shown in Figure 2-5. The skeleton psychrometric chart of the process is shown in Figure 2-6.

2.2.2.1 Adiabatic Dehumidification at Constant Enthalpy

Dehumidification occurs between states 1 and 2 when air passes through the desiccant bed.

2.2.2.1.1 State 1

The ideal steady state evaporative cooling process starts with state 1 (initial state) where ambient air enters the system at 100 °F with 10% relative humidity. The other properties of state 1 are found by plotting the state in the psychrometric chart.

Table II-1. Automotive Cooling Requirements. (Ruth 1975)

Car Type	Ambient Condition °C (°F) / RH	City Driving (Cool Down) No Outside Air kW (Btu/h)	City Driving 30 mph 100% Outside Air kW (Btu/h)	Highway Driving 60 mph 100% Outside Air kW (Btu/h)
Subcompact	32 (90) / 50%	3.59 (12250)	3.49 (11910)	3.77 (12850)
	38 (100) / 20%	3.47 (11830)	3.20 (10930)	3.41 (11640)
	43 (110) / 5%	3.86 (13170)	3.79 (12950)	4.08 (13940)
Compact	32 (90) / 50%	4.14 (14140)	4.17 (14220)	4.14 (14120)
	38 (100) / 20%	4.01 (13680)	3.76 (12840)	3.73 (12730)
	43 (110) / 5%	4.42 (15100)	4.51 (15380)	4.48 (15280)
Standard	32 (90) / 50%	5.06 (17270)	5.16 (17620)	5.13 (17520)
	38 (100) / 20%	4.91 (16770)	4.64 (15830)	4.61 (15730)
	43 (110) / 5%	5.37 (18320)	5.55 (18950)	5.52 (18850)

2.2.2.1.2 State 2

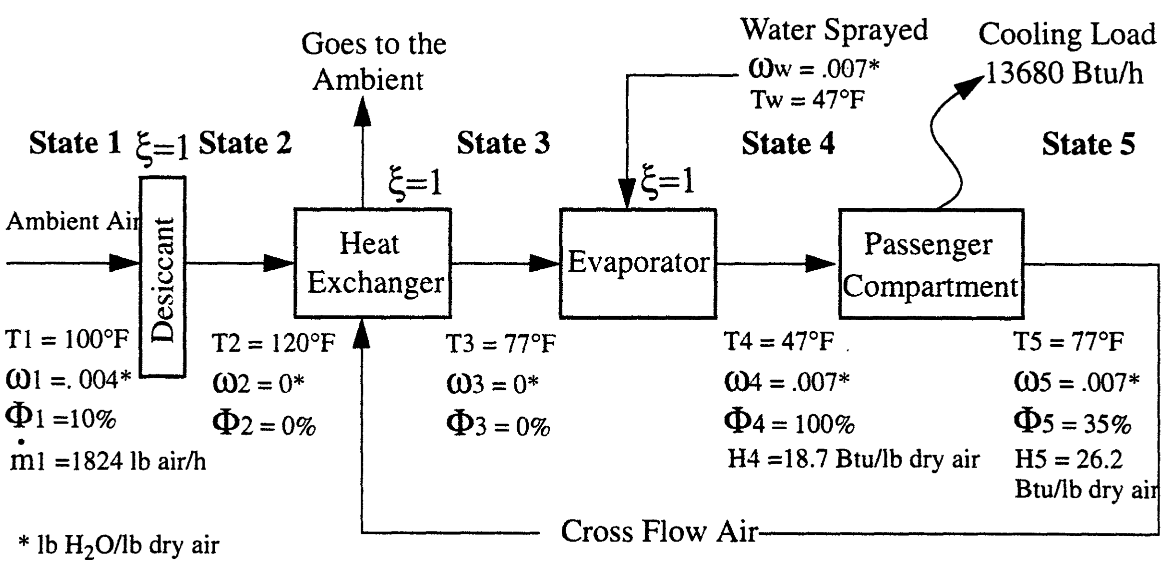
The air from state 1 passes through the desiccant bed. The desiccant removes the humidity of the air leaving at state 2. The properties of state 2 are determined by following the adiabatic ideal dehumidification process until the relative humidity is zero (at this point the humidity ratio is also zero). The temperature of air read from the psychrometric chart is 120 °F.

2.2.2.2 Cooling at Constant Humidity

Cooling occurs between states 2 and 3 when the dry hot air passes through the compact heat exchanger and releases some of its heat to the cold cross flow air that leaves the passenger compartment at state 5 (discussed later). The hot dry air becomes warm at constant humidity because the two flows of air do not mix in the heat exchanger.

2.2.2.2.1 State 3

The warm dry air that leaves the heat exchanger has the same relative humidity as state 2 ($\Phi = 0\%$). Assuming the heat exchanger has 100% efficiency ($\xi = 1$) and the hot air transfers heat to the cold air until its temperature equals the initial temperature of cold air (these conditions are never true for a real heat exchanger but are used in this preliminary analysis to simplify the calculations), then the temperature of state 3 is 77 °F.



ω - humidity ratio (specific humidity)
Φ - relative humidity
H - enthalpy
T - temperature
$\dot{m}$ - mass flow rate
ξ - efficiency

Figure 2-5. Schematic of the Ideal Steady State Evaporative Cooling Process.

2.2.2.3 Adiabatic Saturation at Constant Enthalpy

The saturation of air occurs between states 3 and 4 when air passes through the evaporator and is sprayed with water until it reaches 100% relative humidity. In this process, the assumptions that there is perfect saturation (efficiency of the saturation process is 100%, $\xi=1$) and that the temperature of the water added is 47 °F (the same as the air leaving the evaporator at state 4) were made to simplify the calculations.

2.2.2.3.1 State 4

The warm dry air that leaves the heat exchanger at state 3 passes through the evaporator where it is humidified with sprayed water until it saturates (Φ=100%). State 4 is determined by the path of constant wet-bulb or enthalpy line at the intersection with the saturation line. The amount of water needed to saturate the air is determined by the relative humidity (.7 lb H_2O/lb dry air). The dry-bulb and wet-bulb temperatures of air are the same at state 4 (47 °F, the wet-bulb temperature of air at state 3). The enthalpy of state 4 (18.7 Btu/lb dry air) is the same at state 3 because the ideal saturation process is at constant enthalpy.

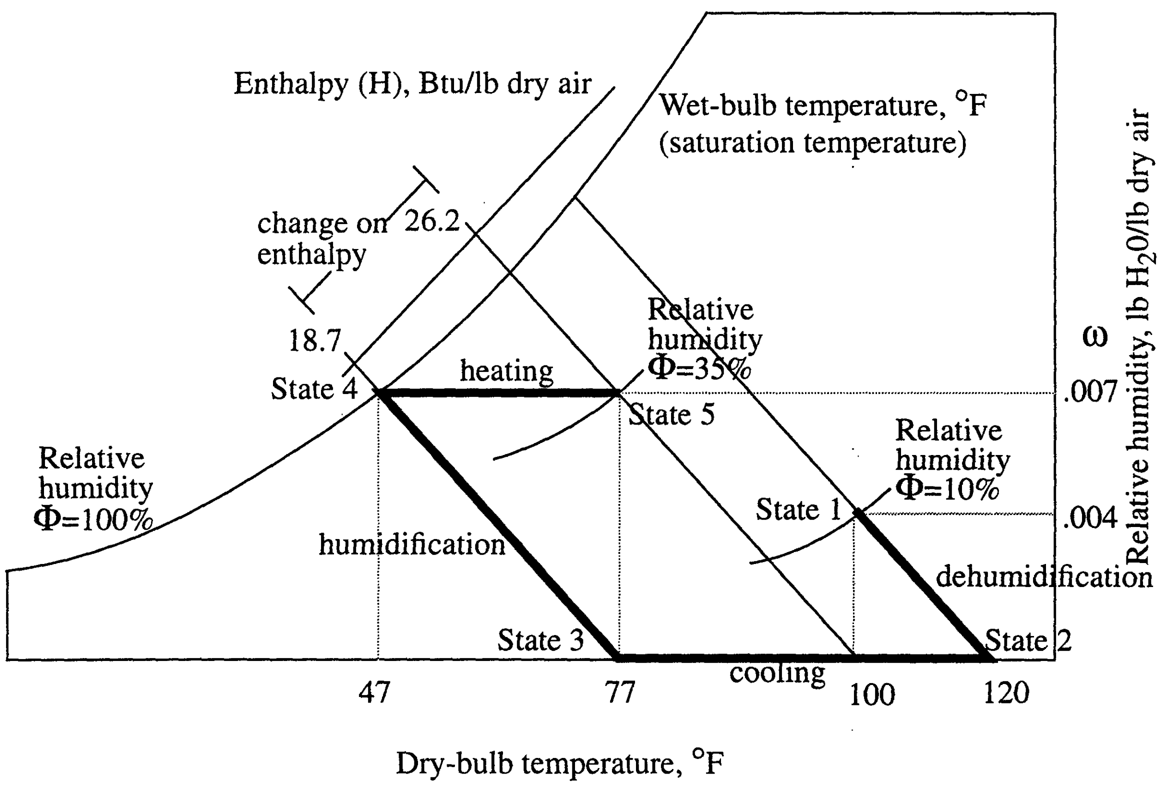


Figure 2-6. Skeleton Psychrometric Chart of the Ideal Steady State Process.

2.2.2.4 Heating at Constant Humidity

The heating process occurs inside the passenger compartment between states 4 and 5. The air at state 4 that enters the passenger compartment is heated because it removes the cooling load (13680 Btu/h). The air of state 4 mixes with the air inside the passenger compartment. This mixture has to remain at a comfortable temperature for the passengers. After air has reached a temperature that is no longer considered comfortable, it is taken out of the passenger compartment at state 5. The air at state 5 is used as the cold cross flow air for the heat exchanger and is then thrown into the ambient.

2.2.2.4.1 State 5

The cold humid air that leaves the evaporator is delivered to the passenger compartment where it absorbs the heat cooling the passenger, producing the desired air-conditioning effect. The air leaves the passenger compartment at state 5. The air, assuming that it does not absorb any moisture inside the passenger compartment, remains at the same humidity of state 4. The temperature was specified by the final state design conditions as 77 °F to keep a comfortable temperature inside the passenger compartment. Although the humidity remains constant, the relative humidity drops to 35% at 77 °F. The enthalpy of air at state 5 (26.2 Btu/h) increases because of the heat absorbed.

2.2.3 Mass Flow Rate

The mass flow rate (m) is needed to determine the water consumption of the evaporative system. The mass flow rate is not a property, it cannot be read directly from the psychrometric chart, and needs to be calculated.

The mass flow rate is going to depend on the amount of heat that we want to remove from the passenger compartment and on the enthalpies of the air arriving and leaving the passenger compartment.

2.2.3.1 Mass Flow Rate Calculation

Cooling Load = Mass Flow Rate (Enthalpy of State 5 - Enthalpy of State 4)

13680 Btu/h = $\dot{m}$ (26.2 Btu/lb dry air - 18.7 Btu/lb dry air)

$\dot{m}$ = 1824 lb air/h

2.2.4 Water Consumption

The water consumption is important because it indicates the amount of water (in weight quantity) needed to operate the system. The system, to be more efficient than the compressor A/C system, needs to weigh less (that is including the weight of the water that is going to be used to create the cooling).

The water consumption of the evaporative system per hour can be calculated with the mass flow rate and the humidity ratio of the air that leaves the evaporator.

2.2.4.1 Water Consumption Calculation

$$\text{Water Consumption} = \text{Mass Flow Rate} * \text{Humidity Ratio of State 4}$$
$$= 1824 \text{ lb air/h} * .007 \text{ lb } H_2O\text{/lb dry air}$$
$$= 12.8 \text{ lb } H_2O\text{/h}$$

2.3 Effect of Component's Efficiency

The purpose of studying the effect of the components' efficiency on the ideal process is to characterize the system behavior. The efficiency of the ideal system depends on the efficiency of the desiccant, the compact heat exchanger, and the evaporator. The study changes the efficiency of each component independently; the rest of the process remains at ideal conditions. The properties used to evaluate the effect are the mass flow rate and the water consumption; they are shown on the graphs of Figure 2-7. The calculations to generate these graphs are in Appendix B.

The air's mass flow rate is important for preventing drafts that occur when stray air currents which are cooler than the compartment temperature reach the bare skin with a velocity that intensifies their cooling effect. Since the air's velocity increases with a rise in mass flow rate, drafts can be prevented with an adequate air velocity (adequate mass flow rate) and by forcing the mixing to occur in unoccupied space. The adequate air velocity should be high enough to mix thoroughly the air entering and the air inside the compartment before touching skin and to induce enough turbulence everywhere for easy convective and evaporative skin cooling to increase comfort.

The water reduction is important because it has to be as low as possible to reduce the weight of the evaporative system, increasing in that way the efficiency of the system.

2.3.1 Effect of the Desiccant

The efficiency of the desiccant is affected by the humidity of the air leaving the desiccant (state 2). The lower the efficiency of the desiccant, the higher the humidity ratio.

To compare the effect of the change in efficiency with the ideal evaporative cooling process, the dehumidification follows the constant enthalpy line and the other processes remain in ideal conditions. The ideal adiabatic dehumidification, instead of ending at a humidity ratio equal to zero, will end at some other value of humidity ratio.

By analyzing the graph of the effect of the desiccant in Figure 2-7, can be seeing that the mass flow rate rises with an increase in humidity ratio, and the water consumption does not change significantly. The mass flow rate changes almost proportionally with the change of humidity ratio.

2.3.2 Effect of the Heat Exchanger

The efficiency of the heat exchanger is affected by the temperature of the stream of air that leaves the heat exchanger toward the evaporator (the temperature of the cross flow air used to cool the stream of air is affected, but it is unimportant for the

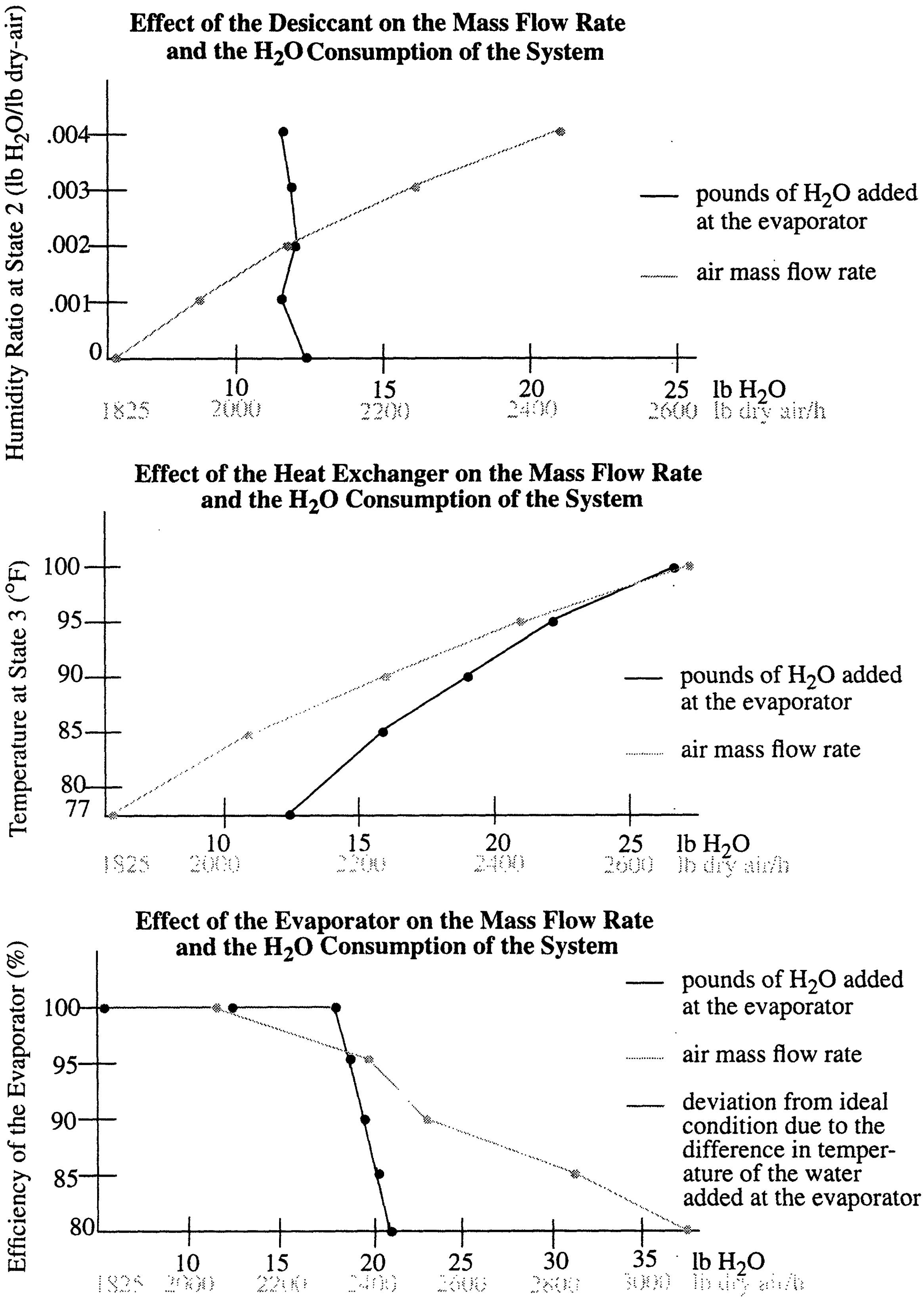


Figure 2-7. Effect of Component's Conditions on the Mass Flow Rate and the Water Consumption.

analysis because the cross flow air is through to the ambient). The lower the heat exchanger efficiency, the hotter the air leaving the heat exchanger (state 3).

The heat exchange cooling process follows the constant humidity line until it gets to the temperature set for state 3. The other processes remain at their ideal conditions to compare the effect of the change in the efficiency of the heat exchanger.

By analyzing the graph of the effect of the heat exchanger in Figure 2-7, can be seeing that the mass flow rate rises with an increase in the temperature of the air that leaves the heat exchanger (state 3), and water consumption also increases with a rise in the temperature of state 3. Both the mass flow rate and the water consumption change almost proportionally with changes in temperature of state 3. Since the water consumption curve has a slope, the water consumption is going to increase very significantly.

2.3.3 Effect of the Evaporator

The efficiency of the evaporator is affected by the temperature of the water that is added, and by the saturating efficiency (the higher the efficiency of evaporation inside the evaporator, the less air escapes without being saturated by water). The saturating efficiency is proportional to the efficiency of the evaporator. The saturating efficiency is not the same as the relative humidity of air. The equation used to calculate saturation efficiency is shown in Appendix B.

Because the process takes into account the effect of the temperature of the added water, it follows an indeterminate path (ordinary saturation process). The other processes remain at their ideal conditions to compare the effect of the change in the efficiency of the evaporator.

Analyzing the graph of the effect of the evaporator of Figure 2-7; the mass flow rate increases with a rise in the evaporator's efficiency, and with a temperature of water added higher than the temperature of air leaving the evaporator; the water consumption also increases for the two conditions. Both the mass flow rate and the water consumption change almost proportionally with the variation of the evaporator's efficiency. The increase in mass flow rate because of the evaporator's efficiency is very significant.

The mass flow rate and the water consumption are greatly increased by the temperature of the added water. The temperature of the water added in the evaporator is less effective in cooling air, while it consumes more water (as shown in the graph by the clear line) because water needs to cool first. The air's comfort-cooling potential is reduced because the air's relative and specific humidities and dew point increase without comparable reduction of the dry-bulb temperature.

2.3.4 Analysis

The conclusions drawn from the comparison of the effect of each component on the mass flow rate and the water consumption are:

(a) any deviation from the perfect efficiency of the components increases the mass flow rate

(b) the mass flow rate increases more steeply with a decrease in the efficiency of the evaporator

(c) the water consumption rate increases more steeply with an increase in the temperature of the air that leaves the heat exchanger toward the evaporator

(d) the difference in temperature of the added water from the temperature of air leaving the evaporator greatly increases both the mass flow rate and the water consumption

This analysis suggests that the evaporative cooling system should:

(a) have components with the highest possible efficiency

(b) have the coldest possible water supply to the evaporator

(c) deliver the coldest and driest possible air to the evaporator

The design of the system has to pay close attention to the choice of an appropriate desiccant and the dissipation of the heat generated by the dehumidification processes.

Chapter 3: Desiccant Stage Theoretical Design

3.1 Desiccant

A desiccant is a hygroscopic chemical that attracts and holds moisture at normal temperatures but releases it as vapor when heated, usually well below the boiling point. Desiccants are of two types:

(a) absorbents
(b) adsorbents

3.1.1 Absorbents

The absorbents are those chemicals that absorb water vapor from the air Their affinity for water is such that they condense it directly from the air.

Absorbents are loose powders or soft solids when dry. When exposed to air they become progressively damper with moisture until they themselves dissolve in it (materials with this properties are know as deliquescent). Absorbents are difficult to use except as strong solutions sprayed through air to dry it. The absorbed moisture dilutes the solution, which is then heated to drive it off and finally cooled to restore absorbency. Those solutions may be corrosives. Liquid desiccants are generally organic liquids, concentrated acids, and alkalis or salts in solution. Some absorbents are calcium chloride and lithium chloride.

3.1.2 Adsorbents

The adsorbents are those chemicals that adsorb water vapor from air and incorporate it into their on structural lattice by virtue of their lower internal surface pressures, less than the partial pressure of water vapor on air. Desiccation continues until the pressure of the adsorbed moisture and air are on equilibrium.

Adsorbents are porous minerals. They operate essentially dry without a visible change in nature. The adsorbents easily adsorb nearly 40% of their weight in moisture, storing it in submicroscopic layers on its enormous internal surface areas without visible change. They can be used for thousands of adsorption and heat-reactivation cycles without deterioration. The adsorbent moisture is released as a vapor when the adsorbent is heated at relatively low temperatures. The adsorbent then needs to cool to restore adsorbency. Some adsorbents are silica gel, molecular sieve, and activated alumina.

Adsorbents are commonly used in two ways to dry the air:

(a) in alternating pairs of porous beds, the air to be dried flows through one while the other is regenerated by heat
(b) in large, slowly revolving porous wheels that resemble rotary heat exchangers, where one sector is drying the air passing through it axially while another sector surrenders its moisture to an axially passing hot air flow that carries the moisture away

3.1.2.1 Desiccant Wheels

Desiccant wheels allow simple continuous operation without the complex ducting and dampering of bed-type systems. Mechanically, desiccant wheels suffer from the same leakage and are housed, driven, and connected to ducting like rotary heat exchangers. The operation is in the middle ranges, not fully saturating or drying the desiccant each cycle, both because reactions are because faster and lower regenerating temperatures are needed. Wheel speeds are adjusted to roughly maximize total water removal. Wheels can revolve in vertical or horizontal planes.

Several companies make rotary dehumidifier wheels with molecular sieve, lithium chloride, silica gel, or similar filler. Some forms of wheels are: granular desiccant confined by screening or granules bonded in sheets; corrugated paper like filling impregnated with lithium chloride; and silica gel bounded to plastic ribbons wound spirally on spacers to form axial air passages, or wound of corrugated aluminum ribbon chemically treated to cover the ribbon with activated alumina. These wheels are more efficient and occupy less space than the desiccant beds.

3.1.3 Desiccant Materials

There are many desiccant materials between absorbents and adsorbents. But the ones that are going to be discussed in terms of their advantages and limitations are those that are most commonly used and have the best characteristics.

3.1.3.1 Lithium Chloride

Lithium Chloride is an absorbent material highly hygroscopic and deliquesce. When dry it is a solid salt, but as desiccant it is use in liquid solution form. Its advantages are:

(a) widely used in industry (most popular liquid sorbent)
(b) maintains the desiccating action over a wide range of concentrations
(c) very high moisture absorption at high air relative humidity
(d) on liquid systems cleans the air stream of any suspended solid and has a biocidal effect on the air (kill bacterias)
(e) low cost

Its limitations are:

(a) undesirable handling characteristics (corrosive and toxic when not diluted)
(b) not suitable for absorption at low relative humidity
(c) regeneration temperature depends on the solution concentration (higher the concentration, higher the regeneration temperature)
(d) slow absorption rate

3.1.3.2 Calcium Chloride

Calcium chloride is an absorbent material with qualities very similar to lithium chloride. It is also a salt when dry, and a liquid solution as desiccant. Its advantages are:

(a) widely used in industry
(b) maintains the desiccating action over a wide range of concentration
(c) high moisture absorption at high air relative humidity
(d) on liquid systems cleans the air stream of any suspended solid and has a biocidal effect on the air (kill bacterias)
(e) low cost

Its limitations are:

(a) undesirable handling characteristics (corrosive and toxic when not diluted)
(b) not suitable for absorption at low relative humidity
(c) regeneration temperature depends on the solution concentration (higher the concentration, higher the regeneration temperature)
(d) slow absorption rate

3.1.3.3 Silica Gel

Silica Gel is a solid adsorbent material that resembles crushed stones and is produced from sodium silicate and sulfuric acid. It is a granular material with great porosity that provides an extensive internal area. Its advantages are:

(a) widely used in industry (traditional solid desiccant, is the most widely known and versatile)
(b) high degree of dehydration until water weight is 20% that of its own weight, continues at lower rate until 50%
(c) does not change nature (remains solid during every stage)
(d) regeneration temperature is lower than 500 ^{o}F
(e) good for a large number of regeneration cycles
(f) fast adsorption rate
(g) non-corrosive

Its limitations are:

(a) low adsorption at low relative humidity
(b) rapid degradation to powder if exposed to water droplets
(c) average cost

3.1.3.4 Activated Alumina

Activated alumina is an adsorbent material with qualities very similar to silica gel. It is a granulated material prepared form aluminium trihydrate that posses considerable porosity. Its advantages are:

(a) widely used in industry
(b) high degree of dehydration until water weight is 15% that of its own weight, continues at a lower rate until 40%
(c) does not change of nature (remains solid during every stage)
(d) regeneration temperature is lower than 400 ^{o}F
(e) good for a large number of regeneration cycles
(f) inert, stable, and non-corrosive

(g) impregnated with calcium chloride, has twice its adsorption capability and is regenerated at temperatures lower than 350 °F
(h) fast adsorption rate
(i) low cost

Its limitations are:
(a) low adsorption at low relative humidity

3.1.3.5 Molecular Sieves

Molecular sieves are adsorbent materials. Any crystalline metal alumino-silicate of the class of minerals known as zeolites belong to this category. Originally they were found naturatelly in modified volcanic ash in relatively small quantities. Synthetic forms of the naturally occurring minerals, as well as many species having no known natural counterpart, have been prepared by a hydrothermal process. Molecular sieves are unusual because they have very uniform pore sizes that allow the material to selectively adsorb or reject molecules according to their molecular size. Molecules having a critical diameter less than the thickness of the molecular layer are held while larger particles are excluded. Its advantages are:
(a) widely used in industry
(b) achieves very low dew points (dry gases to extremely low residual water concentrations)
(c) high desiccant capability even at low relative humidity
(d) adsorbs water until water's weight is 30% that of its own weight
(e) does not change nature (remains solid during every stage)
(f) good for large number of regeneration cycles
(g) fast adsorption rate
(h) non-corrosive

Its limitations are:
(a) regeneration temperature around 600 °F
(b) average cost

3.1.4 Desiccant Selection

A series of evaluative criteria must be employed to select the best desiccant to be used on the evaporative system. To extructurate the criteria and the selection process, a table similar to the one used to select the evaporative cooling system in Section 1.1 is used. Table III-1 presents the materials, criterias, and their comparison. From the information of the table, the best desiccant is activated alumina. But activated alumina does not have high adsorption at low relative humidity. The only desiccant that is good at this property is the molecular sieve, as shown in Figure 3-1.

Changing the adsorption characteristic of activated alumina is harder than changing the negative properties of the molecular sieve. There are different types of molecular sieves with properties that vary by intensity. It is possible to identify a molecular sieve with negative characteristics (regeneration temperature, and cost) that can be comparable with those of activated alumina. The molecular sieve studied in this thesis is the zeolite Na-A of the type 3A (has crystalline cavities of three angstroms).

Table III-1. Comparison of the Desiccant Materials.

Criteria \ Desiccant	Lithium Chloride	Calcium Chloride	Silica Gel	Activated Alumina	Molecular Sieve
High Sorption at Low Relative Humidity	S	S	DATUM	S	+
Sorption Rate	—	—		S	S
Regeneration Temperature	—	—		+	—
High Water Weight % per Desiccant Weight	+	+		—	—
Regeneration Cycles	S	S		S	S
Nature Change	—	—		S	S
Cost	+	+		+	S
Results:	2 + 2 S 3 -	2 + 2 S 3 -		2 + 4 S 1 -	1 + 4 S 2 -

Legend:
S - same as datum + - better than datum — - worst than datum

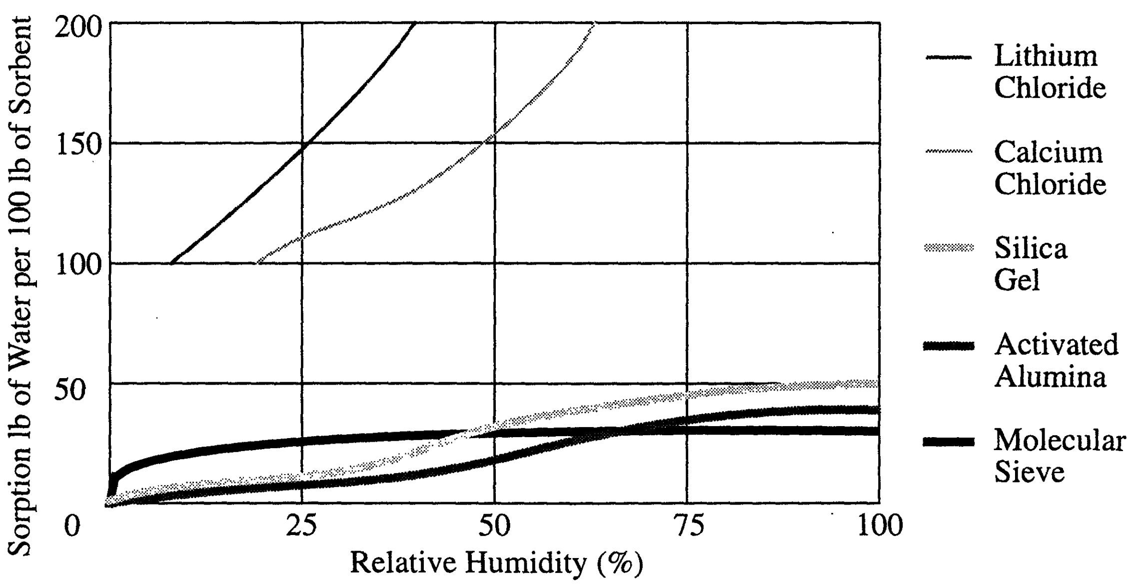


Figure 3-1. Sorption Capability of Desiccants Depending on the Relative Humidity.

3.1.4.1 Desiccant Parameters

The desiccant is going to be used in the way of a desiccant wheel because the wheel arrangement has the advantage of being a more simple and compact system than the desiccant beds. With desiccant wheels the process is continuous without having to switch from one bed to another or having to have more than one pair of desiccant beds. Only one wheel is needed.

Appendix C shows the calculations for the mass flow rate of the main stream of air to satisfy a water consumption of 20 lb H_2O/h at the evaporator and the final conditions inside the passenger compartment of 77 °F and humidity ratio of .008 lb H_2O/lb dry. The dimensions of the desiccant wheel were arbitrarily set on a diameter of 1 ft and a thickness of .25 ft. The weight of the desiccant wheel for these dimensions is 8.64 lb. The calculation of the angular velocity and tangential velocity of the desiccant wheel were based on a mass flow rate of 2500 lb air/h, the weight of the desiccant, and a air humidity ratio of .004 lb H_2O/lb dry. The angular velocity gave .02007 rad/s, and the tangential velocity gave .01004 ft/s. The frequency is 11.5 rev/h. For a higher humidity ratio, the frequency has to be higher.

3.2 Desiccant Case

A case for the desiccant wheel is needed because the desiccant is divided into three different areas according to the processes the desiccant has to go through, and has to be isolated from unwanted humid air. Figure 3-2 shows a schematic of the desiccant case with its three different areas. The three areas of the desiccant case are:

(a) dehumidification area
(b) regeneration area
(c) cool-down area

The desiccant wheel rotates inside the desiccant case in clock wide while the case is static. The case material chosen was aluminum with wall thickness of .0625 in.

3.2.1 Dehumidification Area

The dehumidification area comprises one quadrant of the desiccant wheel case. The case in this section has two nozzles, one at the front and the other at the back. The nozzles are used to guide the air flow from the circular section of the air piping to the quadrant section of the desiccant and the back to the circular piping section.

The air stream that goes through the dehumidification area is the main stream of air.

3.2.2 Regeneration Area

The regeneration area is the quadrant that follows the dehumidification area in the direction of the desiccant wheel rotation. The configuration of this quadrant is the same as the dehumidification area because it also has a stream of air going through the desiccant. In this case the stream of air is hot air to regenerate the desiccant.

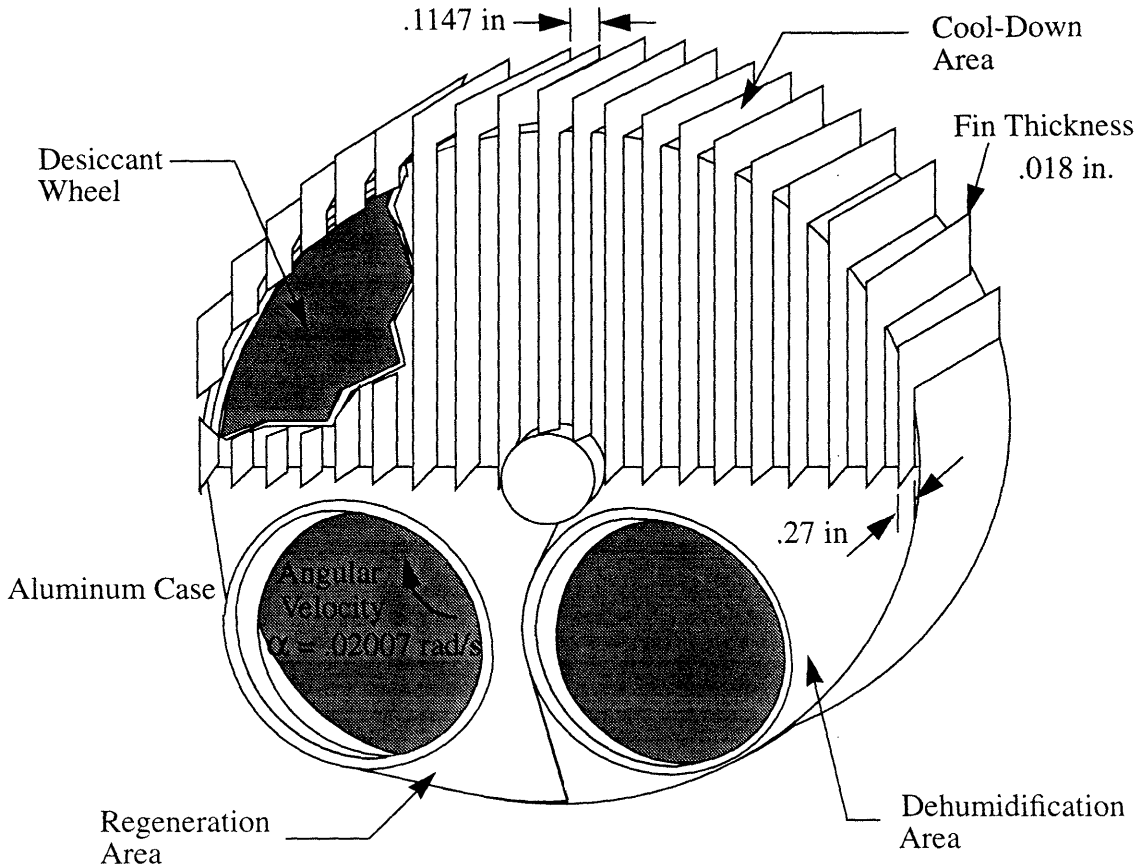


Figure 3-2. Schematic of the Desiccant Case Design.

3.2.3 Cool-Down Area

The cool-down area comprises the next two quadrants after the regeneration area. This area is larger because the desiccant needs time to cool down from the high temperatures of the regeneration process. The desiccant is more efficient with a lower temperature. The desiccant has to exchange heat with the ambient through the case to avoid saturating the desiccant with the water of the ambient air.

To ensure a maximum air flow, the desiccant wheel should be placed horizontally with the cool-down area facing the air flow created by the forward motion of the automobile. To help the heat exchange process through the case, the cool-down area is a fined area with fins in the direction of outside air flow. The fin thickness is .018 in., the spacing between fins is .1147 in., and the fin height is .27 in.

The calculations of the amount of heat that this array can remove are shown in Appendix C. The heat transfer by the cool-down area is based on the assumption that the ambient air velocity is 50 ft/s; the desiccant wheel has the same convection heat transfer coefficient as the finned circular tubes surface CF-8.72; and the temperature is the same as the temperature at the base of the fin (600 °F). For those conditions, the amount of heat removed by the cool-down area is 73597.66 Btu/h. This value is less than the value of heat absorbed by the molecular sieve during regeneration (61500 Btu/h). The calculation of the heat absorbed by the desiccant is based on the conditions of the regeneration air, which is assumed to have a mass flow rate of 500 lb/h, a specific heat at constant pressure and 450 °F of .246 Btu/lb °F, and a change in temperature of 500 °F. According to these calculations, the desiccant case cool-down area as designed can lower the temperature of the desiccant before the wheel cycle stars again.

Chapter 4: Heat Exchangers Theoretical Design

4.1 Heat Exchangers

A heat exchanger is a device that effects the transfer of heat (energy) from one fluid to another. Heat exchangers come in a wide variety of sizes, shapes, and types, and utilize a wide variety of fluids. They are typically classified according to flow arrangement and type of construction. The heat exchanger that will be discussed in this section is the plate-fin compact construction single-pass cross-flow heat exchanger with both fluids unmixed because this system yields the largest heat transfer per volume when both fluids are gases.

Compact heat exchangers are used when a large transfer surface area per unit volume is required and at least one of the fluids is a gas. These devices have dense arrays of finned tubes or plates. They are characterized by a small convection coefficient and the flow is usually laminar.

In cross-flow heat exchangers, the two fluids are arranged to flow perpendicularly to each other. This arrangement offers some advantages in terms of compactness and effectiveness and some disadvantages in terms of fabrication and maintenance. A typical cross-flow exchanger schematic for hot and cold fluid temperature distribution with respect to the initial and final end of the heat exchanger is presented in Figure 4-1. The outlet temperature of the cold fluid can be higher than the outlet temperature of the hot side.

The flow arrangement for each fluid can be mixed or unmixed. A fluid is unmixed if the passageway contains the same discrete portion of the fluid throughout the transverse of the heat exchanger; a fluid is mixed if the fluid from one passageway can mix with fluid from others.

For a single-pass flow, the fluids pass throughout the transverse of the heat exchanger only once; the flow is multipass if the fluids that pass transversely throughout a part of the heat exchanger return and pass through another exchanger again and continue to pass like this for multiple times. The single-pass and multipass for the plate-fin heat exchanger are shown in Figure 4-2.

Cross-Flow
Heat Exchanger

$T_{h,i}$	hot-side inlet temperature	$T_{c,i}$	cold-side inlet temperature
$T_{h,o}$	hot-side outlet temperature	$T_{c,o}$	cold-side outlet temperature
$\dot{m}_h$	hot-side mass flow rate	$\dot{m}_c$	hot-side mass flow rate

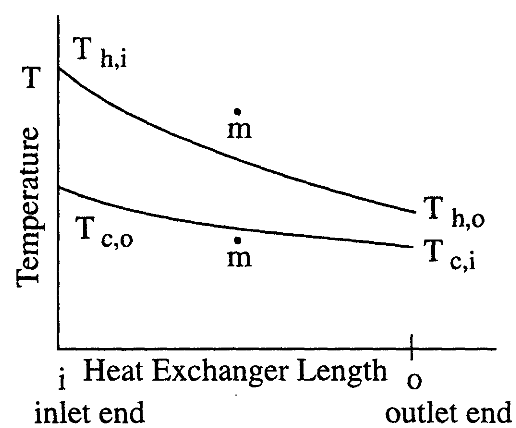


Figure 4-1. Temperature Distribution for a Cross-Flow Heat Exchanger.

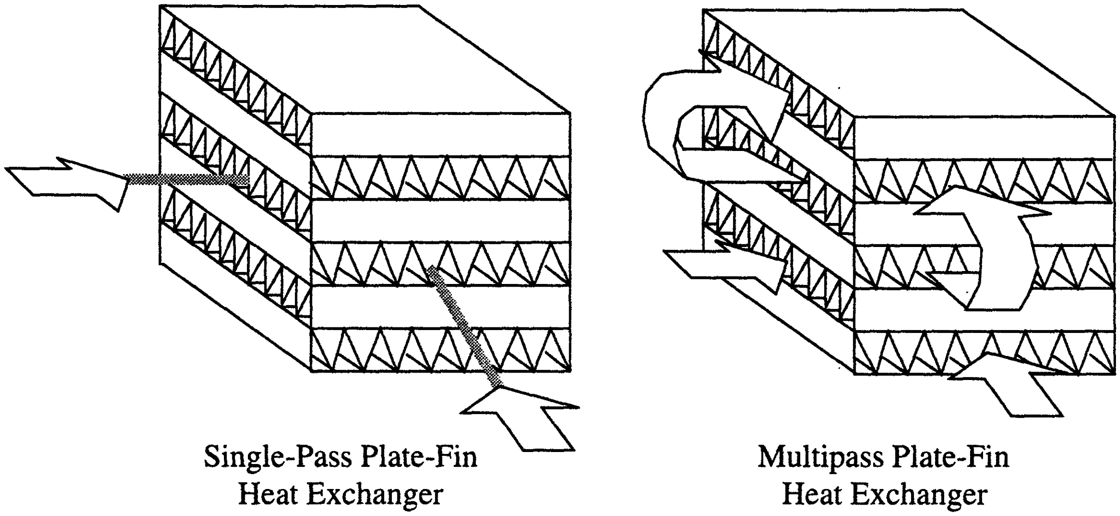


Figure 4-2. Single-Pass and Multipass Flow Arrangement for Plate-Fin Heat Exchangers.

Compact heat exchangers are needed at two stages of the evaporative cycle. The two exchangers are:

(a) main stream heat exchanger
(b) regeneration heat exchanger

4.1.1 Main Stream Heat Exchanger

A heat exchanger on the main stream is used to lower the temperature of the hot dry air that leaves the desiccant wheel. If water is added to that air without lowering its temperature, air will return to its original state without any improvement to its the conditions. The lowering of the air temperature creates a depression of the dry-bulb temperature. This depression generates the difference in enthalpy needed to produce a colder air stream in the evaporator.

The heat transfer occurs between a hot and cold fluid. The hot-side fluid is the hot dry air that leaved the desiccant. The cold-side fluid can be either the air that leaves the passenger compartment, or the outside air with the passenger compartment air. Both fluids are air, therefore a compact plate-fin heat exchanger is needed.

4.1.1.1 Design of the Main Stream Heat Exchanger

The design of a heat exchanger depends on information about both the fluids and the configuration of the heat exchanger. The information needed about the fluids is the inlet temperature and the mass flow rate of both fluids. The mass flow rate can be substituted by the outlet temperature. If the outlet temperature is known, the best method to find the remaining properties of the fluids is the Log Mean Temperature Difference (LMTD) Method (Hodge 1990 p. 93 - 98). If both mass flow rates are known, the best method is the Number of Transfer Units (NTU)

Method (Hodge p. 98 - 108). The configuration information (type, materials, dimensions, and surface geometry data) of the heat exchanger is needed in the design because the heat transfer is different for each configuration.

4.1.1.1.1 Given Information

To design the heat exchanger, the given information is the inlet temperature of the hot- and cold-side and the mass flow rate of the hot-side. The mass flow rate of the cold-side can vary because it can use outside air.

The inlet temperature of air for the hot-side is 115 °F and the cold-side is 77 °F. The mass flow rate of 2500 lb/h, calculated on Appendix C to produce the cooling capacity of 3600 Btu/h, is the same mass flow rate throughout the main stream. Therefore, the mass flow rate of the hot-side air and that of the returning air from the passenger compartment are 2500 lb/h.

The mass flow rate of outside air was set on 5000 lb/h to simplify the calculations for the heat exchanger design and to set a realistic value for the mass flow rate of the cold-side air.

4.1.1.1.2 Design Cases

Three different cases were studied to identify the best fluid conditions for the design of the main stream heat exchanger. The conditions differ on the mass flow rate of the cold-side air. The cases are:

(a) 2500 lb/h - uses only the return stream of air from the passenger compartment.

(b) 7500 lb/h - uses a mixture of the return passenger air (2500 lb/h) and outside air (5000 lb/h).

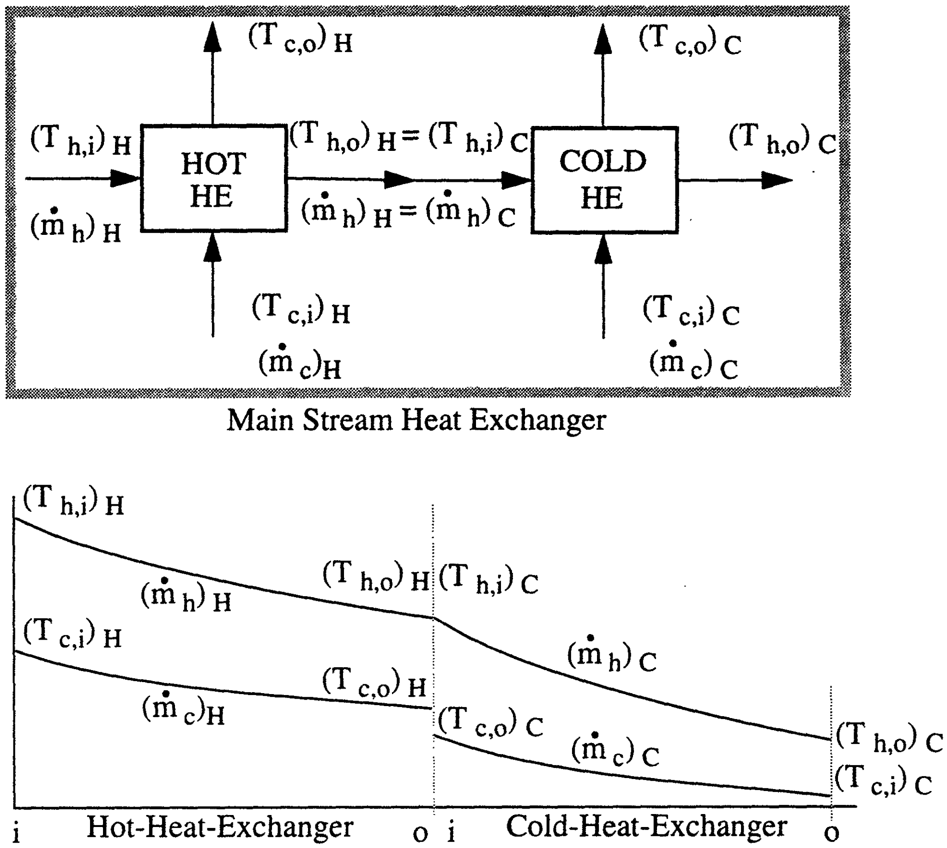
(c) 5000 lb/h and 2500 lb/h - there are two heat exchangers in series; one uses the outside air (5000 lb/h), and the other uses the return passenger air (2500 lb/h).

To decide which case is the best, the outlet temperatures were calculated. Since the inlet temperature and mass flow rate are known for each fluid, the NTU Method was used. The calculations are shown in Appendix D.

The calculations for the case of cold-side mass flow rate of 2500 lb/h resulted in an outlet temperature of 105.5 °F for the cold-side air, and 86.5 °F for the hot-side air. The outlet temperature of the hot-side air is not low enough to supply the cooling capacity of 3600 Btu/h at the passenger compartment. Therefore approach is possible but is not the best.

The calculations for the case of cold-side mass flow rate of 7500 lb/h resulted in an inlet temperature of 92 °F for the cold-side. The calculations to find the outlet temperatures were not made because the hot-side air always leaves the heat exchanger at a higher temperature than the inlet cold-side air. Again, by the consideration that 92 °F is not low enough to supply the cooling capacity of 3600 Btu/h at the passenger compartment, this case was rejected.

The calculations for the case of two heat exchangers in series with cold-side mass flow rate of 5000 lb/h and 2500 lb/h resulted in an outlet temperature at the second heat exchanger of 83.25 °F for the hot-side air. The schematic of the heat exchanger system is shown in Figure 4-3. This system gave the lowest outlet hot-side temperature; therefore this system was chosen for the design.



$(T_{h,i})_H$ - hot-side inlet temperature of hot-heat-exchanger (115 °F)

$(T_{h,o})_H$ - hot-side outlet temperature of hot-heat-exchanger (102 °F)

$(\dot{m}_h)_H$ - hot-side mass flow rate of hot-heat-exchanger (2500 lb/h)

$(T_{c,i})_H$ - cold-side inlet temperature of hot-heat-exchanger (100 °F)

$(T_{c,o})_H$ - cold-side outlet temperature of hot-heat-exchanger (108 °F)

$(\dot{m}_c)_H$ - cold-side mass flow rate of hot-heat-exchanger (5000 lb/h)

$(T_{h,i})_C$ - hot-side inlet temperature of cold-heat-exchanger (102 °F)

$(T_{h,o})_C$ - hot-side outlet temperature of cold-heat-exchanger (77 °F)

$(\dot{m}_h)_C$ - hot-side mass flow rate of cold-heat-exchanger (2500 lb/h)

$(T_{c,i})_C$ - cold-side inlet temperature of cold-heat-exchanger (83 °F)

$(T_{c,o})_C$ - cold-side outlet temperature of cold-heat-exchanger (96 °F)

$(\dot{m}_c)_C$ - cold-side mass flow rate of cold-heat-exchanger (2500 lb/h)

Figure 4-3. Schematic of the Main Stream Heat Exchangers in Series.

Since the first heat exchanger (the hot-heat-exchanger) is in series with a second heat exchanger (the cold-heat-exchanger) at the hot-side, the inlet temperature for the cold-heat-exchanger is the same as the outlet temperature of the hot-side of the hot-heat-exchanger.

Appendix D shows the calculations of the outlet temperatures and the design for the hot-heat-exchanger, as well as the calculations for the cold-heat-exchanger.

4.1.1.1.3 Design Calculations

The design calculations start with the determination of the value of NTU for the maximum value of the effectiveness of the heat exchanger. With NTU, the product UA [overall heat transfer coefficient (U), and heat transfer surface area (A)] is calculated.

Then the velocity of the air and the heat exchanger surface has to be set for the hot- and cold-sides. The same air velocity and heat exchanger surface was set for both surfaces to simplify the calculations and the design.

The velocity was set on 50 ft/s (35 mile/h), and the heat exchanger surface as the plain plate-fin surface 46.45T. The velocity of 50 ft/s was chosen because it was the velocity of the air calculated in Appendix C to produce the cooling capacity of 3600 Btu/h and it seems to be the most likely value for the velocity across the whole evaporative system. The plain plate-fin surface 46.45 T was selected because it has a very high total heat transfer area per volume between plates (1332.45 ft^2/ft^3).

From the velocity and the properties of the surface 46.45 T the product $\bar{h}\eta_t A$ [convection coefficient ($\bar{h}$), temperature effectiveness (η_t), and heat transfer surface area (A)] for the hot- and cold-sides are calculated in terms of the length of the plate-fin surface. From the $\bar{h}\eta_t A$ products and the UA product, the value of the length of the plate-fin surface can be found and the other dimensions can be determined.

The calculation of the dimensions for the hot- and cold-heat-exchangers gave the same values, meaning that both heat exchangers are equal. That is a better strategy for the system part specification. The dimensions give a volume (V) of V= w *h * l = 2.63 in. * 10.3 in. * 13.15 in. [width (w), height (h), and length (l)]. For those dimensions, the heat exchanger consists of an alternating array of 103 rows. The cross-flow rows in the longitudinal direction will consist of 5 plates-fin, one after the other because the fin length in the flow direction was specified as the surface property. The heat exchanger length is a multiple of the width. Figure 4-4 shows a schematic of the heat exchanger and its dimensions.

4.1.2 Regeneration Heat Exchanger

The regeneration heat exchanger is used to exchange heat between the hot exhaust gases of combustion and the outside air to produce the stream of hot air needed to regenerate the desiccant. Is not recommended to use the exhaust gases directly to

$$V = w * h * l = 2.63 \text{ in} * 10.3 \text{ in} * 13.15 \text{ in}$$

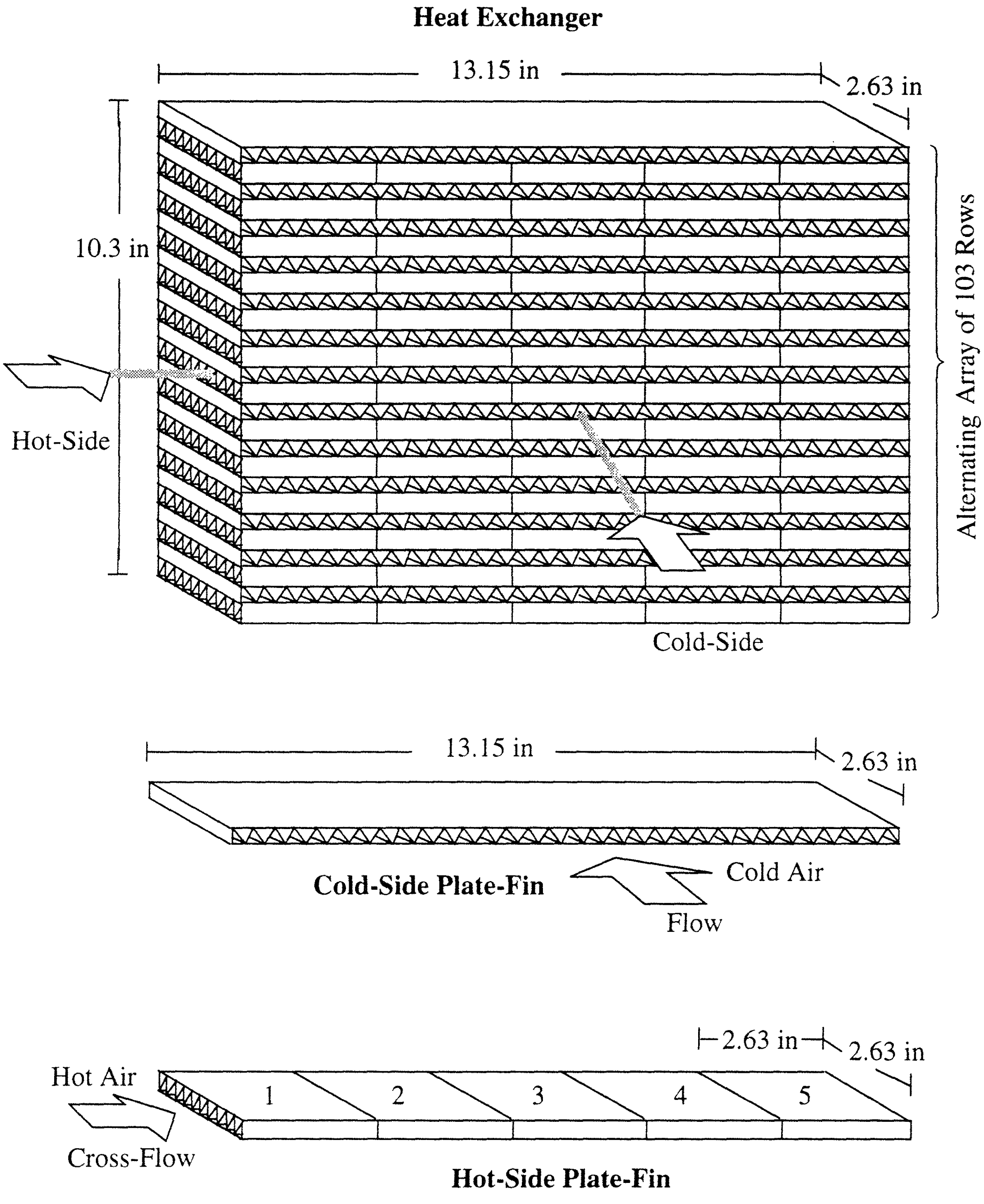


Figure 4-4. Schematic of the Heat Exchanger Design (Hot- and Cold-Heat-Exchangers).

regenerate the desiccant, because these gases have a large amount of water and other impurities. The large amount of water reduces the efficiency of the regeneration process, and the impurities can damage the desiccant.

4.1.2.1 Design of the Regeneration Heat Exchanger

The regeneration heat exchanger (as the main stream heat exchange) has two gases as the working fluids, the mass flow rate and inlet temperatures of which are known. Because of these conditions, the NTU method is going to be used for the design of the compact heat exchanger.

4.1.2.1.1 Given Information

The inlet temperature of air for the hot-side, the exhaust gases side, was set at 700 °F. This temperature varies with the rpm of the engine, therefore the 700 °F represents an average value of the exhaust gases at the engine's manifold. For the cold-side the inlet temperature was set at 80 °F. This temperature is the temperature of outside air. It was set at 80 °F (instead of the 100 °F that has been used as the outside temperature for the design until now) because it will be the worst temperature for this case.

The mass flow rate of 1000 lb/h was set for the hot-side. This mass flow rate, like the hot-side inlet temperature, varies with the rpm of the engine, therefore 1000 lb/h represents an average value of the exhaust gases with the car running at low speed. The mass flow rate of outside air was set at 500 lb/h (half of the mass flow rate of the hot-side) to simplify the calculations for the heat exchanger design.

4.1.2.1.2 Design Calculations

Appendix D shows the calculations of the design for the regeneration exchanger. The first step was to find the outlet temperatures of both sides using the NTU method. Then the value of NTU was determined for the maximum value of the effectiveness of the heat exchanger, and with the NTU, the product UA [overall heat transfer coefficient (U), and heat transfer surface area (A)] was calculated.

The hot- and cold-side air velocity was calculated next. The velocity for the hot-side was calculated assuming a cross-sectional area of the exhaust pipe with a diameter of 3 inches (.25 ft). The resulting velocity for the hot-side was 145 ft/s. For the cold-side, the cross-sectional area was assumed to be one fourth of the area of the desiccant wheel. The diameter of the wheel is one feet. The resulting velocity for the cold-side was 15 ft/s.

The heat exchanger surface was set at the plain plate-fin surface 46.45T. The same heat exchanger surface was used to standardize as much as possible the components of the evaporative system and to simplify the calculations and the design.

From the velocity and the properties of the surface 46.45 T, the product

$\bar{h}\eta_t A$ for the hot- and cold-side were calculated in terms of the length of the plate-fin surface. From the $\bar{h}\eta_t A$ products and the UA product, the value of the length of the plate-fin surface was found and the other dimensions were determined. The calculation of the dimensions for the regeneration heat exchanger gave a volume (V) of V= w *h * l = 2.63 in. * 4.4 in. * 5.26 in. For these dimensions the heat exchanger consists of an alternating array of 44 rows. The cross-flow rows in the longitudinal direction consist of 2 plates-fin, one after the other because the fin length in the flow direction was specified as a the surface property. Figure 4-5 show, a schematic of the heat exchanger and its dimensions.

$$V = w * h * l = 2.63 \text{ in} * 4.4 \text{ in} * 5.26 \text{ in}$$

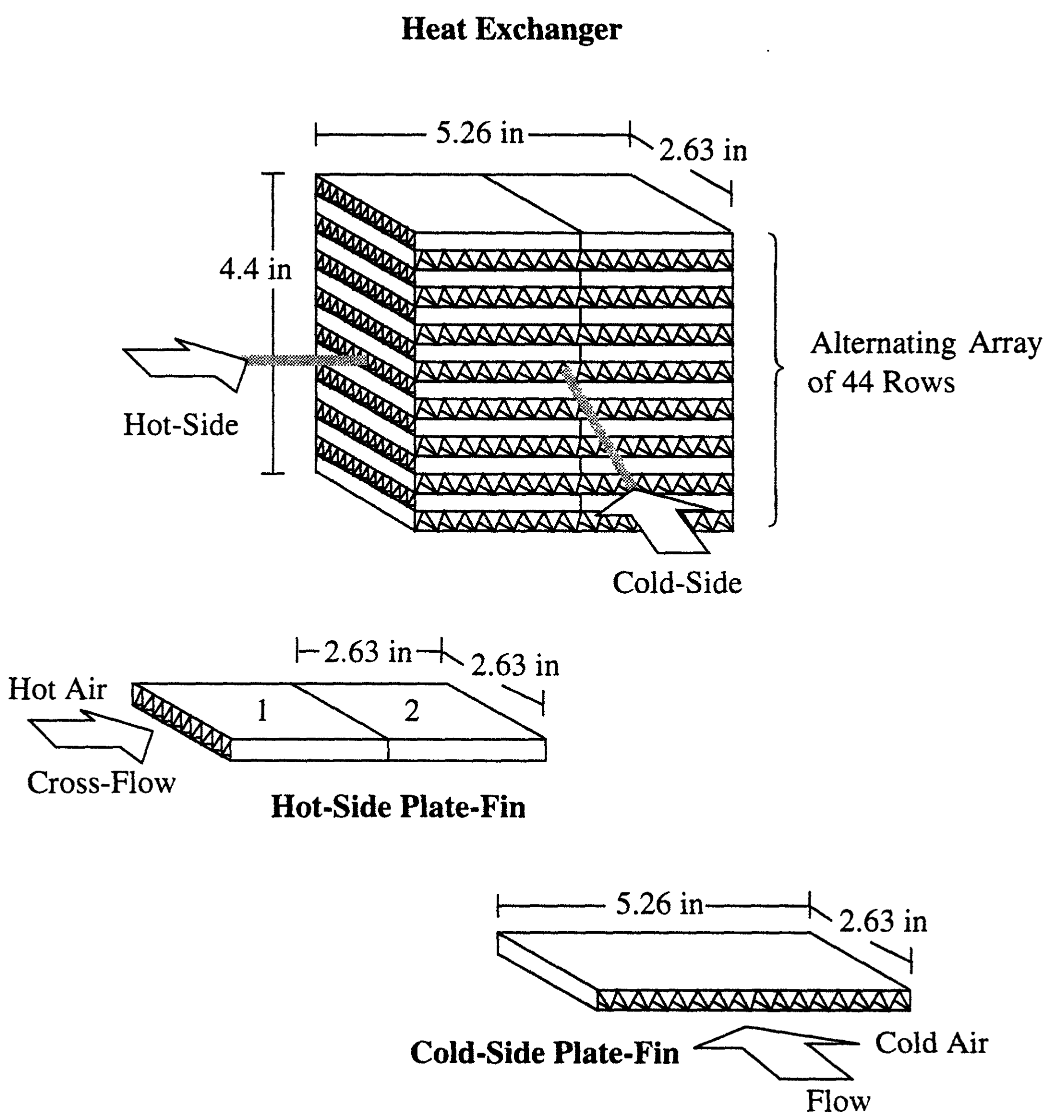


Figure 4-5. Schematic of the Regeneration Heat Exchanger Design.

Chapter 5: Evaporator Theoretical Design

5.1 Evaporator

The evaporator of the system is a direct evaporative cooler. For direct coolers water is added to the supply air in order to create the cooling effect. The system's advantage is its simplicity and compactness. Its limitation is that its cooling capacity is limited to the cooling achievable by adiabatic saturation. The amount of sensible heat removed cannot exceed the latent heat required to saturate the air with water vapor.

5.2 Evaporator Design

To provide the cooling capacity of 13680 Btu/h needed to cool the passenger compartment, the evaporator has to deliver air with a dry-bulb temperature of 55 °F and 90% relative humidity (calculation on Appendix E). The humidity ratio is .008 lb H_2O/lb air to have a water consumption of 20 lb H_2O/h with an air mass flow rate of 2500 lb air/h (see calculation in Appendix C).

The evaporator has to have a 93% efficiency to produce the quality of air needed at the exit of the evaporator (see calculation in Appendix E). This means that the evaporator has to be designed to achieve a high air saturation. The evaporator has to have a spray nozzle able to spray fine drops of water, and the walls has to be covered with porous dense pads of synthetic fiber soaked in water to saturate any mass of air that was not saturated by the water spray nozzle.

5.2.1 Evaporator Case

The evaporator case is a cylinder with a radius of 5.5 inches and longitudinal dimension of 10 in, in the direction of the air flow. The material of the case is plastic because it can insulate the evaporation chamber from the engine heat better than a metal case.

The evaporator case has openings at either end; one for the air inlet, the other for the air outlet. The walls and the openings of the case are covered with wet pads. The wet pads are aspen excelsior pads, which have a thickness of 2 inches for the air openings, and .25 inches for the walls of the case.

5.2.2 Water Spray Nozzle

The water spray nozzle location is at the top of the evaporator case. The nozzle selected is a water-atomizer that can operate with an air pressure as low as 1 PSIG, eliminating the need of an air compressor. The atomizer's water flow rate capacity is 2.41 gal/h $\approx$ 150 ml/min (see calculation in Appendix E).

A schematic of the evaporator with its components is shown in Figure 5-1.

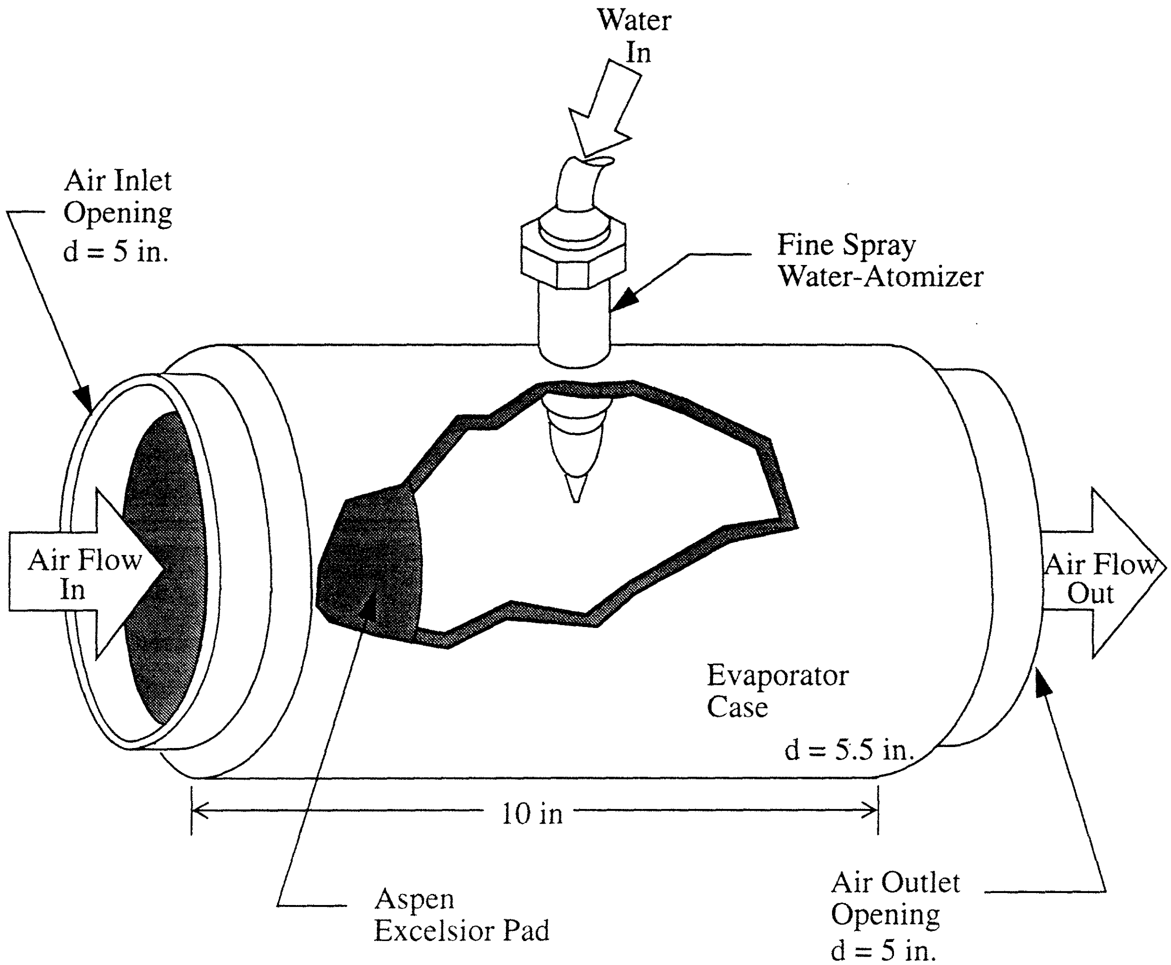


Figure 5-1. Schematic of the Evaporator Design.

Chapter 6: Miscellaneous Components of the Theoretical Design

6.1 Miscellaneous Components

The main components of the evaporative system are the desiccant, the heat exchangers and the evaporator, but there are a series of other components that support the functions of the main components and improve the efficiency of the system. These miscellaneous components are:

(a) air filters
(b) water tank
(c) desiccant wheel motor
(d) circulation pumps
(e) piping
(f) ventilation conduits
(g) heater

6.1.1 Air Filters

The system uses two air filters, one for the main stream, and the other for the regeneration stream. The purpose of the filters is to remove impurities and dust from the air stream.

The main stream needs a filter because, after being cooled, that air is going to be delivered to the passenger compartment. It is desirable to have the best quality of air for the passengers' comfort. The location of this filter is at the opening of the ambient air inlet, in front of the main stream fan. The filter serves a double purpose: it provides clean air not only for the passengers, but also to all the components on the main stream, preventing rapid deterioration of the efficiency of the components.

The regeneration stream has a filter that provides clean air to the desiccant to prevent its rapid deterioration from the contamination with dust particles. The filter is placed at the opening of the ambient air inlet, in front of the regeneration fan.

The filter selected for both streams is a dry-type paper air cleaner. This kind of filter can remove over 99% of air dust with a flow restriction of less than .5 Pa (7.25×10^{-5} psi).

6.1.2 Water Tank

The water tank is a refillable 7.5 gal (60.25 lb H_2O) plastic tank. The water stored inside the tank can cool the passenger compartment for about three hours, based on a water consumption of 20 lb H_2O/h. Although the water tank is too large to fit in the engine compartment, it can be placed inside the trunk of the vehicle.

6.1.3 Desiccant Wheel Motor

The desiccant wheel must rotate with an angular velocity of 11.5 rpm and a torque of 1.08 lb ft (see Appendix F). To produce this rotational motion, a motor has to be used. The motor should be an electric motor with the following characteristics: an frequency of .2 rpm, a voltage of 12 V dc, a power of 5 watts, and torque of 200 oz. in. Due to these unusual requirements the motor will have to be customized.

6.1.4 Circulation Pumps

The system needs circulation pumps to ensure its optimum operation. The circulation pumps needed are:

(a) main stream fan
(b) return stream fan
(c) hot-heat-exchanger fan
(d) regeneration stream fan
(e) water pump

Fans are pumps that use air as their working fluid and have a relatively small increase in head (the losses that the pump has to restitute).

6.1.4.1 Main Stream Fan

A fan is needed on the main stream to provide a continuous supply of 2500 lb air/h regardless of the velocity of the automobile. This fan could be omitted, but the supply of air would then be uncertain, since it would be dependant on the vehicle's velocity. An uncertain supply would hamper the performance of the system and reduce the comfort of the passengers.

The fan is located at the ambient air inlet. The fan has a ventilation requirement of 600 cfm against a fan static pressure of 3 inches of H_2O. The calculations, based on pressure losses for a pipe line of 15 ft, are shown in Appendix F. Base on the calculations, an electrically driven 12 V dc centrifugal fan of 600 cfm, with a static pressure of 3 inches of H_2O, and a power consumption of 240 watts, is needed. Due to the unusual characteristics of this fan, it will have to be customized.

6.1.4.2 Return Stream Fan

Another fan is needed on the return line to remove the warmed air from the passenger compartment and reuse it to cool the cold-heat-exchanger. This returning stream has to remove 2500 lb air/h from the passenger compartment. The fan could be omitted, but the supply of air to the heat exchanger would be uncertain, and the efficiency of the cold-heat-exchanger would be reduced.

The fan location is at the passengers' return air inlet. Because its function is the reverse of the main stream fan, the same fan will be used.

6.1.4.3 Hot-Heat-Exchanger Fan

A fan is also needed for the hot-heat-exchanger to ensure a continuous supply of air (5000 lb air/h $\approx$ 1200 cfm) to the hot-heat-exchanger. Without this fan the heat exchanger efficiency is affected. The fan is located in front of the heat exchanger. The fan selected for this work is the condenser fan designed for compressor systems used by production vehicles (see Appendix H). This fan produces an average of 1060 cfm with a maximum of 2472 cfm. The power consumption is between 150 W and 300 W with a voltage fluctuation of between 6 and 12 V dc.

6.1.4.4 Regeneration Stream Fan

The regeneration stream, like the main stream fan, is needed to ensure a continuous supply of air (500 lb air/h). This fan can be omitted also, but the efficiency of the desiccant regeneration will be affected.

This fan is located at the return air inlet. The fan has a ventilation requirement of 120 cfm against a fan static pressure of .1 inches of H_2O. The calculations, based on pressure losses for a pipe line of 10 ft, are shown in Appendix F. The resulting design is an electrically driven 12 V dc centrifugal fan of 120 cfm, a static pressure of 3 inches of H_2O, a and power consumption of 120 watts. This fan will also have to be customized.

6.1.4.5 Water Pump

A water pump is needed to drive the water inside the tank to the evaporator's atomizer. This pump will be in constant use; it forces the water from the tank at the trunk into the evaporator at the engine compartment. It is located between the water tank and the atomizer.

The pump has to supply a requirement of 150 ml/min of water with a head of .1 ft. The calculations are shown in Appendix F. These calculations are based on the pressure losses for a pipe line of 10 ft. The pump should be a 12 V dc electrical one with a water supply of 150 ml/min, and power consumption of 10 W. Due to its unusual characteristics, this pump will also have to be customized.

6.1.5 Piping

The piping is used to connect and conduit the fluid from one component to the other. There are two kinds of piping, depending on the fluid carried: air piping and water piping.

The air piping is a flexible rubber exhaust hose with a diameter of 5 in. This kind of piping is used for every passage of air on the main stream, return stream and regeneration stream.

The water piping is a thin flexible plastic tube with a diameter of .25 in. This tube connects the plastic tank to the water-atomizer.

6.1.6 Ventilation Conduit

The main ventilation conduit of the system is the distribution system that delivers the cold air to the passenger compartment. The evaporative system will use the same distribution system used by the compressor system (see Appendix H).

Other conduits that are also needed. The main stream line needs one conduit for the ambient air inlet at the start of the line. The ambient air inlet selected is the blower case of the compressor system (see Appendix H). The location of this case, the main stream air inlet case, is at the base of the windshield. This case contains the main stream fan and filter.

The return air line needs a conduit to take away the warmed air from the passenger compartment. The return air inlet is similar to the blower case of the compressor system because the air opening is to the passenger compartment instead of the outside. The return stream air inlet case contains the return stream fan. A possible location for the air opening is at the front passengers feet.

The regeneration stream line needs two conduits, one for the ambient air inlet, and the other at the line outlet behind the desiccant to prevent outside air from contact the desiccant and to direct the hot air away from the desiccant wheel. The regeneration air inlet is similar to the blower case of the compressor system with its opening to the outside. The regeneration stream air inlet case is also located at the base of the windshield.

The cold-heat-exchanger has a case to direct the two air flows through its core. The hot-heat exchanger has two conduits to direct the main stream air flow in and out of its core. The regeneration-heat-exchanger also has a case which directs the two air flows through its core.

6.1.7 Heater

A heater is needed to add heat capability to the system. To simplify the design, the heater selected is the one used by production vehicles (see Appendix H). The heater includes the heater case.

Chapter 7: Evaporative System Theoretical Design

7.1 Evaporative Cooling System

Theoretically the evaporative cooling system can produce a cooling capacity of 13680 Btu/h keeping the passenger compartment at 77 °F and 40% relative humidity with a water consumption of 20 lb H_2O/h and an electric power consumption of about 765 W (≈ 1 hp). The characteristics of the system are given in Table VII-1.

The low COP of the system (about .2) does not mean that the system is inefficient because the COP calculation takes into account the heat that was going to be wasted anyway. The real consumption of the system is the electrical power consumed (≈ 1 hp = 2545 Btu/h) giving a COP of 5.4. The calculations for both COPs are shown in Appendix G.

7.1.1 Thermodynamics of the Theoretical Design

The thermodynamics of the evaporative system predict its performance. The evaporative system will be able to produce the required cooling effect based on its ability to meet the state point of the evaporative cooling process. The calculations of the thermodynamical process are shown in Appendix G.

7.1.1.1 State Points

The state points are the properties defined for the main stream of air. The skeleton psychrometric chart of the process is shown in Figure 7-1. The schematic of the evaporative system with the states is shown in Figure 7-2.

Table VII-1. Characteristics of the Evaporative Cooling System.

CHARACTERISTIC	VALUE
Passenger Compartment Interior	77 °F; Φ = 40%
Ambient Air	100 °F; Φ = 10%
Evaporator Exit	55 °F; Φ = 90%
Main Stream Mass Flow	2500 lb air/h
Hot-Heat-Exchanger Mass Flow	5000 lb air/h
Cold-Heat-Exchanger Mass Flow	2500 lb air/h
Cooling Capacity	13680 Btu/h
Water Consumption	20 lb H_2O/h
Electrical Power	765 W ≈ 1 hp = 2545 Btu/h
Heat Consumption	≈ 70920 Btu/h
Regeneration Stream	500 lb air/h; 600 °F
Coefficient of Performance (COP)	≈ .2
Electrical Power COP	≈ 5.4

7.1.1.1.1 State 1

The conditions of state 1 are defined by the conditions of the incoming ambient air of the main stream. The conditions of this air are unknown because they depend on the weather. For the design, these conditions were set on the worst possible case. The ambient air has a temperature of 100 °F, a relative humidity of 10% (humidity ratio of .004 lb H_2O/lb dry air), and a mass flow rate of 2500 lb air/h. Any air with better conditions than the air at state 1 will allow the system to have a better performance.

7.1.1.1.2 State 2

The air is dehumidified by the desiccant following an adiabatic dehumidification process to arrive at state 2. The air at state 2 has a temperature of 115°F, relative humidity of 1%, and humidity ratio of .0005 lb H_2O/lb dry air.

7.1.1.1.3 State 3

Air is cooled at constat humidity by the main stream heat exchangers. Air arrives at state 3 with a temperature of 84°F. The humidity ratio remains on .0005 lb H_2O/lb dry air but the relative humidity changes to 3%.

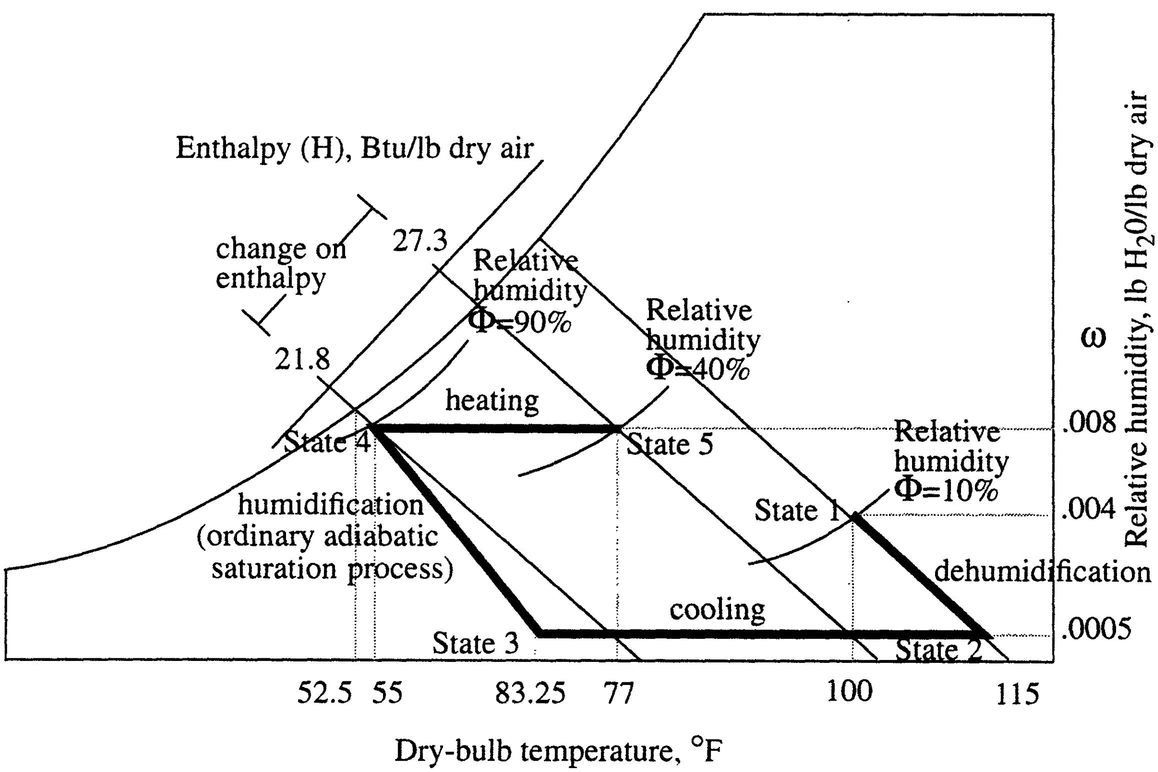


Figure 7-1. Skeleton Psychrometric Chart of the Evaporative Cooling Process.

7.1.1.1.4 State 4

The air passes through the evaporator where it is further cooled adding water to the air (water consumption rate is 20 lb H_2O/h). The air follows an ordinary adiabatic saturation process and leaves state 4 with a temperature of 55 °F, 90% relative humidity, and a humidity ratio of .008 lb H_2O/lb air.

7.1.1.1.5 State 5

The air that leaves the evaporator goes to the passenger compartment. There the air heats up as it removes the heat load produced by the heat of the sun, the ambient air, and the passengers. The air cooled until its temperature is 77 °F. There is no addition of humidity in this heating process, therefore the humidity ratio remains at .008 lb H_2O/lb air but the relative humidity drops to 40%.

7.1.2 Components Integration

The components already discussed in the preceding chapters integrate the evaporative system. The components designed for the system have feasible dimensions to be placed inside the restricted volume of the vehicle. The only moving parts of the system are the circulation pump and the desiccant wheel with its motor.

Table VII-2 gives a list of the components and their specifications. The components are listed according to their location in the different streams following the flow direction. Figure 7-3 shows a schematic of the integrated evaporative system.

Appendix H shows drawings of the compressor system and its components.

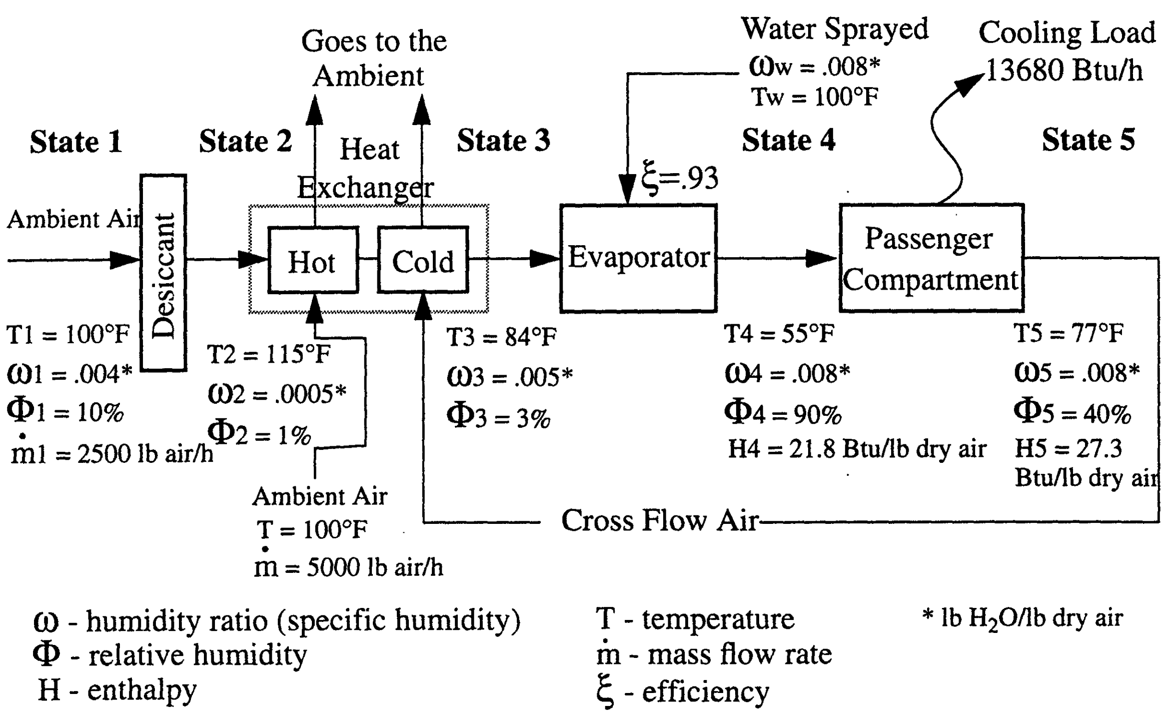


Figure 7-2. Schematic of the Evaporative Cooling System With the State Processes.

Table VII-2. List of Components of the Evaporative Cooling System

COMPONENT	SPECIFICATION
main stream	
Main Stream Air Inlet Case	Plastic case
Air Filter	Dry-type paper air filter
Main Stream Fan	Electrically driven(12 V dc Max.) centrifugal fan 600 cfm, 240 W, static pressure = 3 in H_2O
Air Pipe	Flexible corrugate plastic piping; d = 5 in.
Desiccant Case	Aluminum; t_{wall} = .0625 in. Fins: t = .018 in., h = .27 in., spacing = .1147 in.
Desiccant Wheel *	Molecular sieve 3A; d = 1 ft, t = 3 in.
Desiccant Wheel Motor	Electric motor; 12 V dc, torque = 200 oz. in., 11.5 rev/h (.2 rpm), 5 W
Hot-Heat-Exchanger	Plain plate-fin surface 46.45T; V=2.63 in. * 10.3 in. * 13.15 in.; plastic conduits
Hot-Heat-Exchanger Fan	Electrically driven (12 V dc Max.) axial fan 1060 cfm to 2472 cfm, 150 W to 300 W
Cold-Heat-Exchanger *	Plain plate-fin surface 46.45T; V=2.63 in. * 10.3 in. * 13.15 in.; plastic case
Evaporator *	Plastic case; d = 5.5 in. aspen excelsior pads: $t_{opening}$=2in., t_{wall}=.25in. water-atomizer: flow rate capacity= .004 gal/min
Heater Case	Plastic case with heater
return stream	
Return Stream Air Inlet Case	Plastic case
Return Stream Fan	Electrically driven(12 V dc Max.) centrifugal fan 600 cfm, 240 W, static pressure = 3 in. H_2O
Air Pipe	Flexible corrugate plastic piping; d = 5 in.
Cold-Heat-Exchanger *	-------------
regeneration stream	
Regeneration Stream Air Inlet Case	Plastic case
Air Filter	Dry-type paper air filter
Regeneration Fan	Electrically driven(12 V dc Max.) centrifugal fan 120 cfm, 120 W, static pressure = .1 in. H_2O
Air Pipe	Flexible corrugate plastic piping; d = 5 in.
Regeneration-Heat-Exchanger	Plain plate-fin surface 46.45T; V= 2.63 in. * 4.4 in. * 5.26 in.; plastic case
Desiccant Wheel *	-------------
Air Outlet conduit	Plastic conduit
water stream	
Water Tank	Plastic tank, 7.5 gal
Water Pipe	Flexible plastic piping; d = .25 in.
Water Pump	Electrically driven (12 V Max.) pump; 150 ml/min of water,10 W, head = .1 ft
Evaporator *	-------------

* This components are shared by two stream lines.

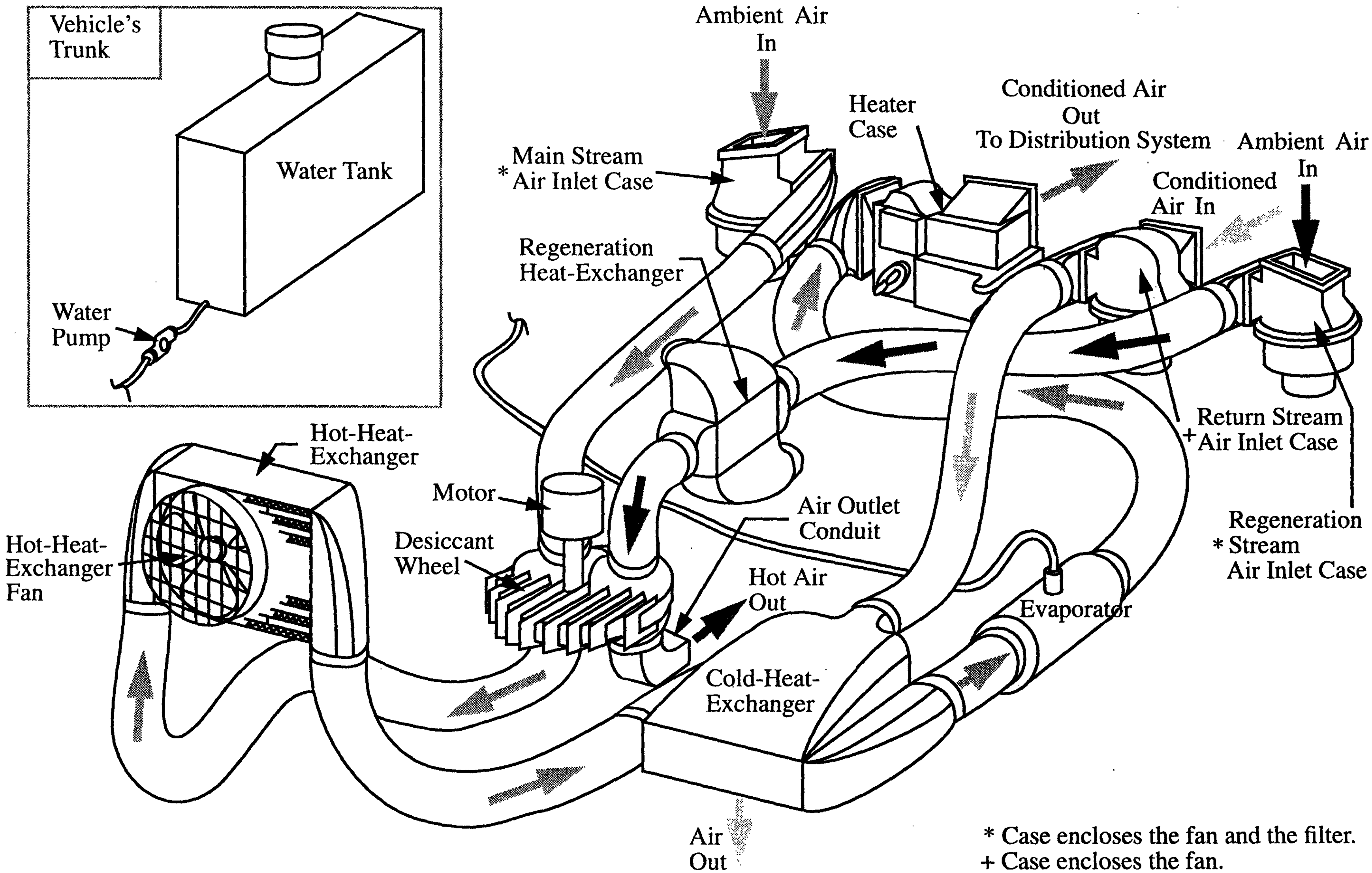


Figure 7-3. Schematic of the Evaporative Cooling System Theoretical Design.

Chapter 8: Analysis of the Evaporative System

8.1 Cost Analysis

A preliminary cost analysis of the theoretical design of the evaporative system is needed to determine the economic feasibility for commercial application of the system. If the system is not economically feasible it will not be able to compete against the compressor system. The cost analysis is based on three aspects:

(a) initial cost
(b) maintenance cost
(c) operating cost

8.1.1 Initial Cost

The initial cost is derived from the total cost of the system's components. The list of components and their approximate cost is shown in Table VIII-1. According to that estimate the total cost of the evaporative system is \$2781.00. The catalog pages for some of the components are shown in Appendix I.

8.1.2 Maintenance Cost

The maintenance cost is measured by the life and reliability of the components. The components that need to be maintained are the filters, fans, water pump, desiccant, and the desiccant motor.

The fans, the water pump, and the desiccant motor can have reliability problems because these components have moving parts but they are expected to last as long as the system does.

The air filters have a long life and reliability but they need to be cleaned or replaced when they get saturated with dust particles. The expected time for the filters to be changed or cleaned is one year.

The desiccant needs to be changed because some detriment to its capabilities is caused by the continuous cycles it has to go through. The expected life of the desiccant is one year to keep the maintenance of the system to once a year.

The total cost of maintenance per year is estimated at \$36. This takes into account the cost of replacing the filters (\$6) and the desiccant (\$30).

8.1.3 Operational Cost

The operational cost is measured by the cost of water consumption, and the fuel consumption cost, as reflected by the electric power requirement, and weight of the water and the system.

The water consumption is 20 lb H_2O/h. The average car in the U.S. is driven about 10,000 miles/year (Fischer 1990). Assuming that this average milage represents about one hour of driving per day, the system will be operating for 365 hours per year with an annual water consumption of 7300 lb H_2O (879.5 gal H_2O). Assuming a

Table VIII-1. List of Components and Their Estimated Cost

COMPONENT	COST
main stream	
Main Stream Air Inlet Case	$150.00
Air Filter	$3.00
Main Stream Fan	$40.00
Air Pipe	$25.00
Desiccant Case	$95.00
Desiccant Wheel *	$30.00
Desiccant Wheel Motor	$35.00
Hot-Heat-Exchanger	$500.00
Hot-Heat-Exchanger Fan	$20.00
Cold-Heat-Exchanger *	$700.00
Evaporator *	$150.00
Heater Case	$200.00
return stream	
Return Stream Air Inlet Case	$150.00
Return Stream Fan	$40.00
Air Pipe	$20.00
Cold-Heat-Exchanger *	----------
regeneration stream	
Regeneration Stream Air Inlet Case	$150.00
Air Filter	$3.00
Regeneration Fan	$40.00
Air Pipe	$20.00
Regeneration-Heat-Exchanger	$300.00
Desiccant Wheel *	----------
Air Outlet conduit	$15.00
water stream	
Water Tank	$45.00
Water Pipe	$5.00
Water Pump	$50.00
Evaporator *	----------
TOTAL	$2781.00

* This components are shared by two stream lines.

Table VIII-2. Cost Analysis

Initial Cost	$2781.00
Annual Maintenance Cost	$36.00
Annual Operational Cost	$117.00
Total Annual Cost	$153.00
Total Cost for Five Years	$3546.00

cost of water of 10 cents per gallon, the total cost of water is about 88 dollars.

The electrical power consumption of the system is about 746 watts ($\approx$ 1 hp). For air-conditioning, the compressor system uses about 20 gal of gasoline for every 10,000 miles (Fischer 1990). The power demand of a constant-speed air conditioner is about 4 hp (Hurter 1974). If 20 gal of gasoline are consumed for 4 hp, 1 hp consumes 5 gal of gasoline. Taking into account a 70% conversion efficiency for the change of the shaft horsepower to electrical power, the annual fuel consumption of the evaporative system is about 7.25 gal of gasoline.

The weight of the system and the water has to be included under the fuel consumption. About 10 gal of gasoline are required per 10,000 miles for each 100 lb of weight (Fischer 1990). The weight of water is about 62.25 lb (7.5 gal H_2O). The weight of the system was estimated to be 130 lb. Therefore, the fuel consumption for the total weight of 192.25 lb is about 19.25 gal of gasoline.

The total fuel consumption is 19.25 gal of gasoline. If the gasoline price is $1.50 per gallon, the annual cost of fuel is about 29 dollars.

The total operational cost per year is about 117 dollars. This takes into account the cost of the water ($88), and the fuel consumption ($29).

8.1.4 Economical Aspect of the Evaporative System

The initial cost of the evaporative cooling system is estimated at $2781. The cost of the compressor system ranges from $725 to $1685 ($1200 average). Compared with the cost of the compressor system ($1200), the cost of the evaporative cooling system ($2751) is higher.

The annual cost of the evaporative system is $153, taking into account the operational cost ($117) and the maintenance cost ($36). The annual cost of the compressor system is the cost of the fuel consumption. Twenty gallons of gasoline are consumed to produce the air-conditioning, and 10 gal are consumed by the 100 lb system. At $1.50 per gallon, the compressor system has an annual cost of $45 for the 30 gal of gasoline. The annual cost of the evaporative system ($153) compared with the annual cost of the compressor system ($45) is higher.

Economically, the evaporative system is not better than the compressor system.

8.2 Performance Analysis

To evaluate the performance of the evaporative system, it is compared with the compressor air conditioning system. The evaluation criteria are:

(a) fuel consumption
(b) cost
(c) cooling capacity
(d) system care
(e) environmental acceptability
(f) weight
(g) volume

Table VIII-3 shows the comparison of the criteria between the two systems.

8.2.1 Fuel Consumption

The fuel consumption of the systems is compared because it is a more accurate term for comparison than the COP. The COP misleads power consumption of a heat-actuated system because these systems use energy that is wasted when work-actuated systems are used. The fuel consumption is more accurate because it takes into account the power demand and the weight of the system. Also, the fuel consumption reflects the pollution caused by the combustible emissions; the higher the fuel consumption, the higher the pollution rate.

The evaporative system consumes 19.35 gal of gasoline. Compared with the fuel consumption of the compressor system (30 gal of gasoline), the evaporative system consumes less fuel, therefore it is better.

8.2.2 Cost

The cost comparison includes the initial, maintenance, and operational costs of each system for ten years. The cost of the evaporative system is about $3546, and the cost of the compressor system is about $1425 for an initial cost of $1200. Thus, the cost of the evaporative system is higher.

8.2.3 Cooling Capacity

The evaporative system design is for a cooling capacity of 13680 Btu/h. The design cooling capacity of the compressor system is 20000 Btu/h (1.5 tons). Therefore the cooling capacity of the evaporative system is lower than that of the compressor system.

8.2.4 System Care

The amount of system care is measured by the degree of inconvenience the driver is put to in order to keep the system running.

To keep the evaporative system working properly, the driver has to replace the consumed water. The water consumption time before refilling is about 3 hours because the water consumption is 2.41 gal H_2O/h (20 lb/h) from a 7.5 gal H_2O (62.25 lb) water tank. The water tank can be refilled almost any where because water is accessible. The tank can be refilled at home at the driver's convenience without having to be refilled at any special place such as a gasoline station.

For long trips it is assumed that the driver would like to stop approximately every three hours to rest, eat, or refill the gasoline tank. The refilling of the water tank can be done at a gasoline station, or the driver can carry extra water. The extra water is not considered in the analysis because long drives are not as often as short drives.

Although the inconvenience of the evaporative system is not great, it is more inconvenient than the compressor system, which needs no refilling on the run.

8.2.5 Environmental Acceptability

The restrictions on car emissions for 1994 are: .41 grams of hydrocarbons (HC) per mile, 3.4 grams of carbon monoxide (CO) per mile, and 1 gram of nitrogen oxides (NO_x) per mile. If the average milage is assumed to be 30 miles per gallon, then for every gallon of gasoline, 12.3 grams of HC, 102 grams of CO, and 30 grams of NO_x are produced. Since the fuel consumption is less for the evaporative systems, the system is less harmful to the environment.

Another environmental difference between the systems is the environmental acceptability of the refrigerant.The evaporative system uses water and air as the working fluids. The evaporative system is better than the compressor system, which uses CFC-12 (Refrigerant-12) or Refrigerant-134a as working fluid, because water and air are fully environmentally acceptable, while CFC-12 is environmentally harmful and Refrigerant-134a has a high global warming potential. In liquid form, CFC-12 and Refrigerant-134a cause injuries when they contact the skin, and cause suffocation in gas form.

Table VIII-3. Comparison of the Evaporative Cooling System With the Compressor Cooling System.

Criteria \ System	Evaporative	Compressor
Fuel Consumption	+	DATUM
Cost	—	
Cooling Capacity	—	
System Care	—	
Environmental Acceptability	+	
Weight	—	
Volume	—	
Results:	2 + 5 -	

Legend:
S - same as datum + - better than datum — - worst than datum

8.2.6 Weight

The weight of the evaporative system is estimated in 130 lb. The evaporative system also has a water weight of about 62.25 lb for a total weight of 192.25 lb. The weight of the compressor system is between 85 lb and 110 lb (100 lb average). Thus, the evaporative system weighs more than the compressor system.

The increase in weight is reflected in an increase the fuel consumption. The value of the fuel consumption was calculated, taking into account the total weight of each system. Therefore, although the evaporative system has a higher weight it has a lower fuel consumption.

8.2.7 Volume

The approximate volume of the evaporative system is about 12 ft^3. The volume of the compressor system is about 8 ft^3. The evaporative system occupies a larger volume than the compressor system. A larger volume is a disadvantage because new vehicles have less space for equipment.

8.3 Justification of the Evaporative System

The evaporative system is justified mainly by its environmental acceptability. It has no CFC-12 or Refrigerant-134a that can leak into the ambient, and it produces less emissions. The importance of this criterion is very subjective but with the new environmental laws it is becoming more relevant. The other advantage of the evaporative system is that it has a lower fuel consumption.

The cost of the evaporative system is higher. The cooling capacity is lower but a cooling capacity as high as 1.5 tons is rarely needed. The evaporative system requires more care than the compressor system but this care is comparable to the care needed to refill the gasoline tank. The evaporative system weight and volume are higher than these for the compressor system but the fuel consumption of the evaporative system is still less than the one for the compressor system.

Chapter 9: Closing

9.1 Summary

The two flaws that can be held against the vapor-compressor air-conditioning unit of production vehicles are its fuel consumption and its pollution of the environment. Studies on alternative automotive air-conditioning are being done in work-actuated and heat-actuated systems. The evaporative system was chosen for this study because it, as a heat-actuated system, has a good energy saving potential, and uses water, an environmentally safe fluid, as refrigerant.

A thermodynamic analysis of the system concluded that in the design of the system it is important to pay close attention to the choice of an appropriate desiccant and the dissipation of the heat after the dehumidification processes. For the best operation of the system, the components should have the highest possible efficiency, the coldest possible water should be supplied to the evaporator, and the coldest and driest possible air should be delivered to the evaporator.

The evaporative system has four main components: the desiccant bed, the main stream heat exchanger, the regeneration heat exchanger, and the evaporator.

The desiccant bed was designed to be a molecular sieve desiccant wheel. A Molecular sieve was chosen as the desiccant material because it has high adsorption at low relative humidity. The shape of the desiccant is a wheel because it allows a simple continuous operation without the complex ducting and dampering of bed-type systems. The desiccant wheel is encased to divide it into three process areas and isolate it from unwanted humid air. The three areas of the desiccant wheel case are: the dehumidification area, the regeneration area, and the cool-down area.

The main stream heat exchanger was divided in two heat exchangers to produce the coldest possible air: the hot-heat-exchanger and the cold-heat-exchanger. The hot-heat-exchanger uses the ambient air as cooling fluid. The cold-heat exchanger uses the return stream of air from the passenger compartment as the cooling fluid.

The evaporator has a cylindrical case with a fine spray water-atomizer to spray the water. The walls are covered with porous dense aspen excelsior pads, a synthetic fiber that is soaked in water, to saturate any mass of air that was not saturated by the atomizer.

The evaporative system has other miscellaneous components that were included in the theoretical design. These are: the air filters, the water tank, the desiccant wheel motor, the circulation pumps, the piping, the ventilation conduits, and the heater.

The theoretical performance of the system was calculated with the thermodynamical process of the theoretical system design. The system should be able to produce a cooling capacity of 13680 Btu/h, delivering air at 70 °F and 40% relative humidity with a consumption of 20 lb H_2O/h, 1hp of power, and 70920 Btu/h of waste heat.

The cost and performance analysis indicated that the evaporative system has a lower environmental pollution and fuel consumption than that of the compressor system. The evaporative system has less environmental pollution because its refrigerant, water, is environmentally acceptable. The system also consumes less fuel because it has less moving parts than the compressor system, and uses the waste heat of the combustion process as the main energy input.

The evaporative system is feasible to build but economically it is not competitive with the compressor system because its cost and care compare badly with the compressor system. Also, the evaporative system weighs more and occupies a larger volume than the compressor system.

9.2 Remarks

The design of the evaporative system presented in this thesis is in the theoretical stage. The hardware of the system has never been built, therefore some performance problems can arise in a prototype system. The expected areas of problems are:

(a) steady state conditions - the system may take a long time to arrive at steady state conditions

(b) ordinary adiabatic saturation process - the process follows an indeterminate path, therefore the theoretical path is not precise

A prototype of the evaporative cooling system needs to be built to test the actual performance of the system. A better comparison between the evaporative and compressor system can be made with a knowledge of the real performance of the system. Then, a final judgement of economic feasibility can be made with the endorsement of test data.

9.3 Recommendations

Some improvements to the design of the evaporative system that can be explored are:

(a) electronic system - an electronic system can improve the efficiency of the evaporative system. This system could have sensors and controls to monitor and adjust the mass flow through the three air streams and the water stream according to the velocity of the outside air, and the rpm of the desiccant wheel according to the outside humidity.

(b) anti-freeze - an anti-freeze can be used to prevent the freezing of water in winter. An air cooling capacity is desirable in winter to remove the blear on the wind-shields of the vehicle. The anti-freeze used has to be environmentally acceptable and unharmful to the passengers.

(c) dehumidification unit - other approaches to the design of the dehumidification unit with other types of desiccants can be studied. Also, an improvement on the dehumidification process can be studied. Since the evaporative system weight is mostly the weight of the water needed to humidify the air, and the dehumidification process removes large amounts of water from air, the efficiency of the system can be improved if the removed water could be saved and used as the water added in the evaporator. The evaporative system could have a commercial feasibility with a higher efficiency and without having to carry a large amount of water and having to refill the water tank as often.

9.4 Conclusion

The evaporative system satisfies the two main objectives of this thesis: less fuel consumption, and less environmental pollution. The theoretical design of the system demonstrate that the evaporative system is feasible to build. But the economic analysis of the system suggested that it is less commercially feasible than the compressor system in terms of cost, weight, and volume.

The building and testing of the evaporative system is absolutely necessary because the economical comparison of the system was based on a theoretical design and performance, and not on the design and performance of a real system. With a real performance, the areas that compared badly can be modified. Also, the design of the system does not take into account a possible breakthrough in technology that can significantly improve the system. An improved system may have better commercial possibilities than the theoretical system discussed in this thesis.

Appendix A

Work-Actuated Systems:

The work-actuated systems generate the refrigerating effect by utilizing the engine shaft power or the electricity generated through the vehicle electric generating system.

A) Refrigerant Vapor Compression Cycle Hermetic System

Advantages:

(a) a hermetic motor/compressor assembly means no shaft seal refrigerant leakage
(b) the compressor is smaller and less complex
(c) the packaging is more flexible because the compressor does not requires a drive belt; the compressor and motor are self-contained
(d) full cooling capacity can be achieved at any engine speed
(e) conditioned air temperature can be controlled without reheat

Limitations:

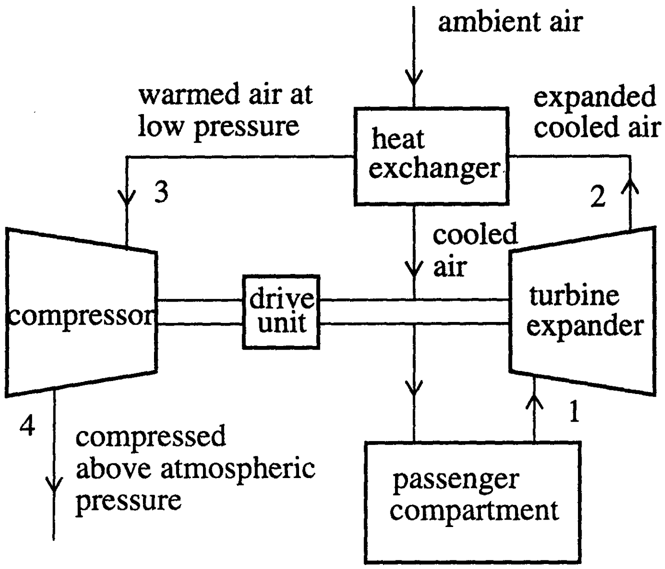
(a) fresh air intake had to be limited to around 30% to reduce the power consumption
(b) the electrical system needs at least 48 V of dc to drive the motor
(c) the hermetic system requires a control strategy, compressor, and electric system different from those used in vapor-compressor A/C.

Remarks:

The hermetic system is less efficient than conventional A/C system. The hermetic system can easily be installed on electric cars.

B) Reverse Brayton Air Cycle

Schematic of Brayton open-cycle system



Cycle:

1→2 passenger compartment air is expanded in the turbine lowering the air temperature

2→3 the cold air exchange heat with ambient air, the cooled ambient air is delivered to the passenger compartment

3→4 the warmed air at low pressure is compressed above ambient atmospheric pressure and vented

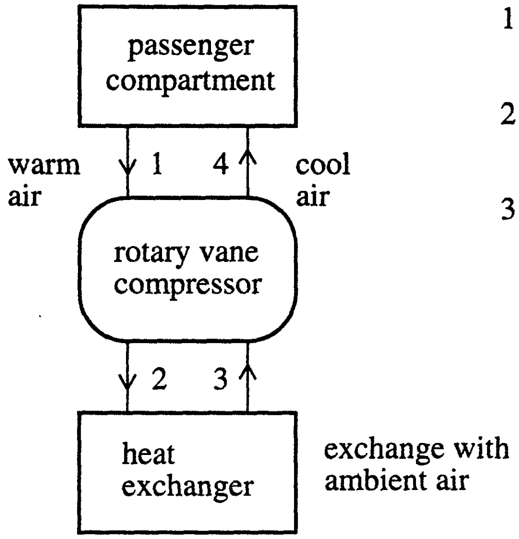
Advantages:
(a) needs no refrigerant
(b) potentially light weight and compact size

Limitations:
(a) needs a high speed compressor and expander drive
(b) has a low COP

Remarks:
For real applications, a closed Brayton air cycle system with a regenerator is probably more efficient than the open cycle. The Brayton cycle is very sensitive to the compressor efficiency.

C) Rotary-Vane Compressor Air Cycle (ROVAC)

Schematic of ROVAC system



Cycle:

1→2 air from the passenger compartment pass through the rotary vane and is compressed

2→3 the compressed air is send to an outdoor heat exchanger

3→4 the air is expanded by the rotary vane compressor, the air becomes cold because of the isenthalpic effect during expansion

Advantages:

(a) needs no refrigerant

(b) compress and expand air simultaneously

Limitations:

(a) is not energy competitive with other cooling methods

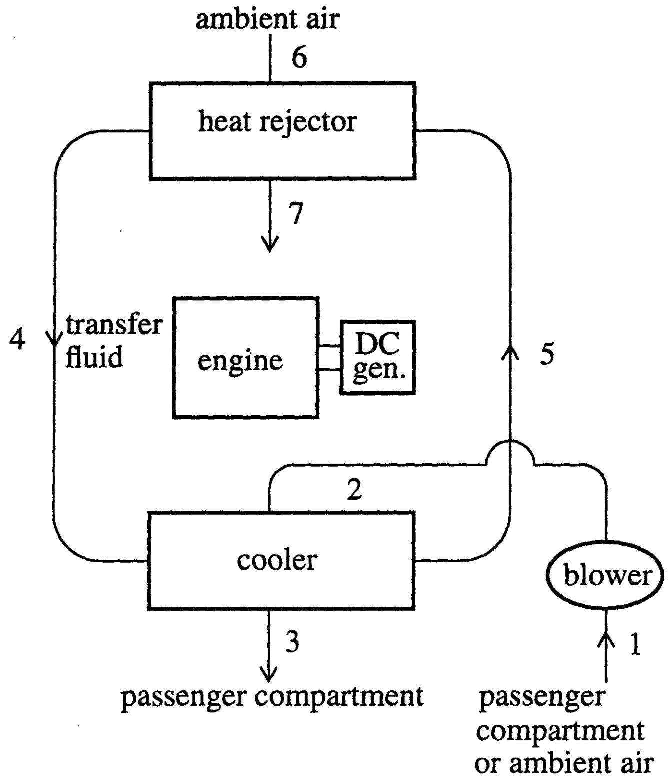
(b) there is not enough detailed description of the operating conditions of the system

Remarks:

Work need to be done in the future to further the analysis of the system.

D) Termoelectric (TE) Cooling System

Schematic of thermoelectric mobile A/C system



Cycle:

1→2 air from the passenger compartment or ambient air is taken in

2→3 the air flows though the TE evaporator to be cooled and then into the passenger compartment

4→5 the TE hot side is cooled by the circulating fluid, which is pumped to the other TE heat exchanger (condenser), where the circulating fluid is cooled

6→7 ambient air cools the condenser TE hot side

Two TE heat exchangers are needed, one acting as the condenser and the other as the evaporator, with a circulating fluid to transfer heat from evaporator to condenser.

Advantages:
(a) needs no refrigerant
(b) solid-state cooling
(c) fast response
(d) high initial cooling capacity
(e) adjustable cooling capacity
(f) no moving parts except a fluid circulating pump
(g) ability to operate as a heat pump when the dc current direction is reversed
(h) system is very rugged and needs little maintenance

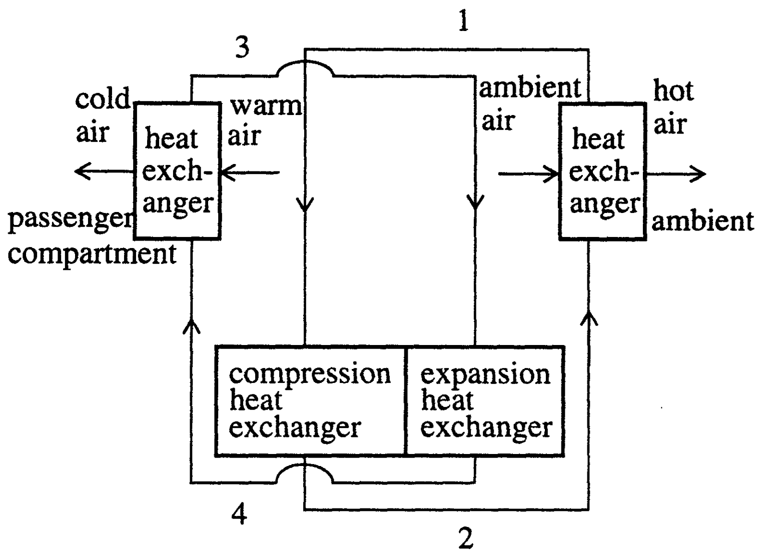
Limitations:
(a) low COP
(b) needs a major TE material breakthrough to be practical

Remarks:

TE cooling theory is based on the Peltier Effect of certain materials. The heat evolved or absorbed per unit time, at the junction between two dissimilar materials is proportional to the current flow provided by a battery. Currently is used for special applications only, like cooling of railway passenger cars in France and space conditioning of underwater vehicles. The TE system is suitable for electric cars because its simplicity.

E) Sterling Cycle Cooling System

Schematic of Sterling mobile A/C system



Cycle:

1→2 constant temperature compression

2→1 constant volume heat exchange

3→4 constant temperature expansion

4→3 constant volume heat exchange

Advantages:
(a) high theoretical COP
(b) heat pump capability
(c) can have modulated cooling capacity
(d) needs non-CFC refrigerant

Limitations:
(a) equipment has not yet been developed to the point where it is competitive with conventional A/C systems in terms of cost, performance, or reliability.
(b) difficulty of achieving a constant temperature compression or expansion in a machine operating at a reasonable speed

Remarks:

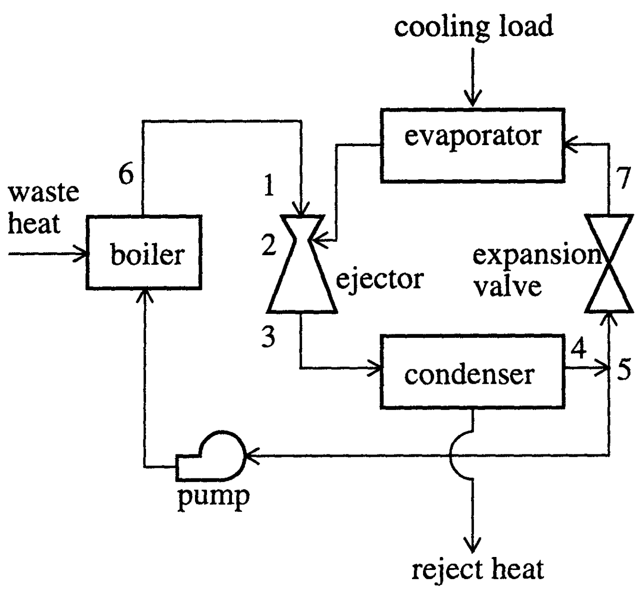
Most Stirling models are proprietary. There is inadequate experimental data for mobile air conditioning application.

Heat-Actuated Systems:

The heat-actuated systems utilize the waste heat of combustion to produce the cooling effect. The use of waste heat represents a very good energy saving potential. Research and development of this kind of system is very rare, which presents a great challenge in technology assessment and potential development.

A) Ejector Cooling System

Schematic of ejector cooling system



Cycle:

1→2 high-pressure gas passes through the ejector to create a low pressure on the evaporator partially flooded with refrigerant, the low pressure of the refrigerant causes low-temperature boiling, the latent heat of vaporization provides the cooling effect

2→3 the low-pressure refrigerant is compressed at the divergent part of the ejector where refrigerant velocity is reduced and pressure is increased

3→4 the refrigerant is condensed

5→6 part of the refrigerant is pumped to the boiler to produce high-pressure gas

5→7 part of the refrigerant is delivered to the evaporator

Advantages:
(a) lightweight and compact
(b) highly reliable
(c) minimum moving parts
(d) low maintenance cost
(e) engine cooling capability

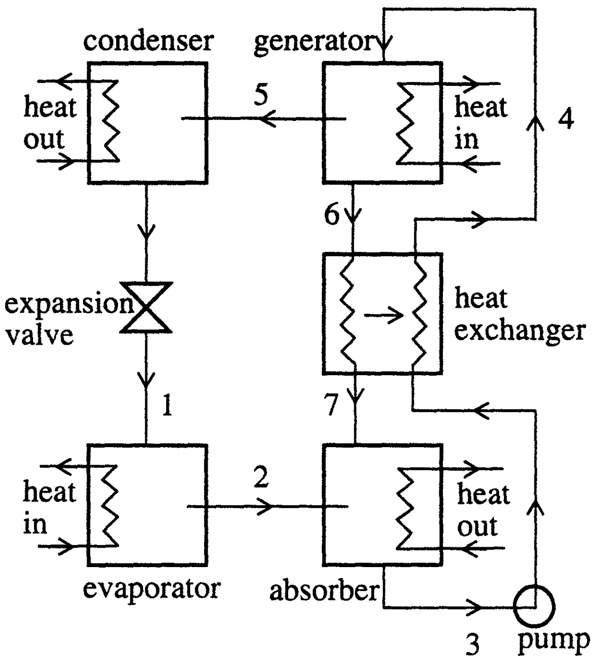
Limitations:
(a) low COP
(b) cars in the future might not have enough waste heat to power the ejector system

Remarks:

The ejector system actuates with low-temperature waste heat. The use of waste heat together with the reduced weight of the system can save over 70% of the fuel consumption compared with actual mobile A/C systems. The theory of the ejector system is similar to that of steam jet refrigeration.

B) Absorption Cooling System

Schematic of single-stage absorption cooling system



Cycle:

1→2 evaporated refrigerant is absorbed by another fluid at the evaporator

2→3 the working fluid is absorbed by the refrigerant creating a low pressure

3→4 the refrigerant-rich working fluid is pumped through the heat exchanger

4→5 heat is added in the regenerator to separate the refrigerant from the absorbent

5→1 the high-pressure refrigerant goes to the condenser, then to the expansion device, and finally to the evaporator

6→7 the refrigerant-lean fluid goes to the absorber to absorb evaporated refrigerant

The absorption needs a pair of fluids (absorbent/refrigerant) to operate.

Advantages:

(a) no moving parts except a fluid circulating pump

Limitations:

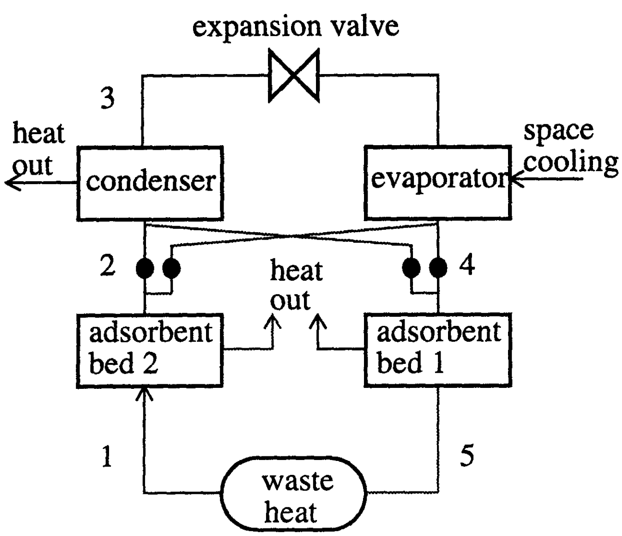
(a) needs proper selection of refrigerant and absorbent
(b) needs very large heat exchangers because heat rejection is at a low temperature
(c) bulky and heavy components
(d) low COP
(e) high operating pressures

Remarks:

The absorption units are the most common heat-driven A/C devices possessing a matured technology for conventional machines. Passenger cars do not have enough waste heat to power an absorption system but it can be feasible for long-haul trucks refrigeration and bus A/C system.

C) Adsorption Cycle (Desiccant) Cooling System

Schematic of closed-cycle desiccant mobile A/C system



Cycle:

1→2 waste heat is applied to a desiccant bed, vapor refrigerant is regenerated, and high pressure is created

2→3 the high-pressure refrigerant is condensed into liquid at the condenser

3→4 the refrigerant goes through an expansion device before being evaporated at the evaporator

4→5 the desiccant bed adsorbs refrigerant vapor and creates a low pressure

After the cooling process is completed, the functions of the two desiccant beds are reversed to start another cooling process.

Advantages:

(a) solid desiccants are usually low-cost materials
(b) the choice of refrigerants is more flexible since solid desiccants can adsorb a number of refrigerant vapors
(c) there are broad choices of solid-fluid pairs
(d) the desiccant can be regenerated by waste heat from car exhaust gas
(e) the cooling COPs (thermal) are higher than those of the absorption systems
(f) the system design is simple

Limitations:

(a) adsorption and regeneration processes pose a heat dissipation problem because of the relatively low temperature differential with ambient air
(b) needs a more frequent cycling for higher cooling capacity

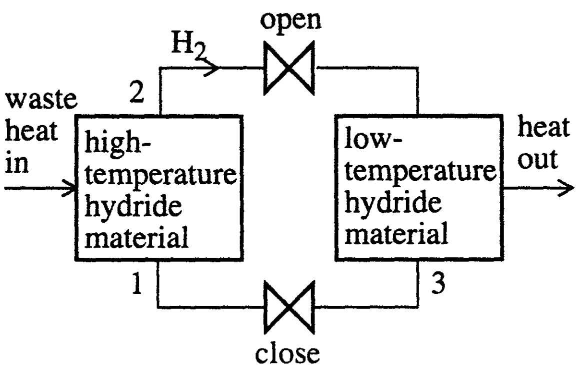
Remarks:

More than two desiccant beds could be needed to insure continuous supply of cooling capacity. Open-cycle desiccant systems with water as the cooling medium are possible.

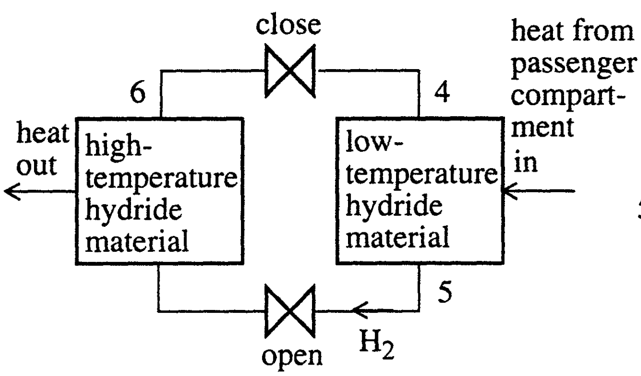
D) Metal Hydride Cooling System

Schematic of metal hydride cooling system

regeneration mode:



absorption mode:



Cycle:

1→2 waste heat is added to high-temperature hydride material (initially at ambient temperature), the temperature and pressure increase to a point where the hydride material starts desorbing hydrogen

2→3 the low-temperature hydride material starts absorbing hydrogen gas, the absorbing heat is dissipate to ambient air

4→5 when the lower valve opens the low-temperature hydride material starts desorbing hydrogen gas because of the low pressure, the temperature drops lower than ambient temperature achieving the cooling effect

5→6 when the high-temperature hydride material is cooled down to ambient temperature the pressure drops

Advantages:

(a) potentially high COP
(b) needs non-CFC refrigerant
(c) fast response rate
(d) hydride materials are not costly

Limitations:

(a) hydride materials are brittle and have a high thermal expansion
(b) the system is heavy because many pounds of hydride material are needed to produce a considerable cooling capacity

Remarks:

The metal hydride system can potentially be operated with waste heat from engine exhaust gas. Hydriding alloys are intermetallic absorbent compounds that can absorb a very large quantity of hydrogen gas, this process is reversible. Metal hydride mobile A/C is feasible for hydrogen fueled cars and buses.

Appendix B

A) Calculations of the Effect of the Desiccant on the Mass Flow Rate and the H_2O Consumption of the Evaporative Cooling System

(a) $\omega 2$ = 0 lb H_2O/lb dry air(Ideal Case)

From the Psychrometric Chart (see Figure B-1):

$H5$ = 28.2 Btu/lb dry air
$H4$ = 18.7 Btu/lb dry air
$\omega 4$= .007 lb H_2O/lb dry air
$\Phi 5$= 36%

Mass Flow Rate Calculation:

$\dot{m}$ = Cooling Load/(H5 - H4)
= (13680 Btu/h)/(26.2 Btu/lb dry air - 18.7 Btu/lb dry air)
= 1824 lb dry air/h

Water Consumption Calculation:

Water Consumption = $\dot{m} * \omega 4$
= 1824 lb dry air * .007 lb H_2O/lb dry air
= 12.8 lb H_2O/h

(b) $\omega 2$ = .001 lb H_2O/lb dry air

From the Psychrometric Chart (see Figure B-1):

$H5$ = 26.6 Btu/lb dry air
$H4$ = 19.5 Btu/lb dry air
$\omega 4$= .0073 lb H_2O/lb dry air
$\Phi 5$= 38%

Mass Flow Rate Calculation:

$\dot{m}$ = Cooling Load/(H5 - H4)
= (13680 Btu/h)/(26.6 Btu/lb dry air - 19.5 Btu/lb dry air)
= 1926.8 lb dry air/h

Water Consumption Calculation:

Water Consumption = $\dot{m} * (\omega 4 - \omega 2)$
= 1926.8 lb dry air * (.0073 - .001 lb H_2O/lb dry air)
= 12.1 lb H_2O/h

(c) $\omega 2$ = .002 lb H_2O/lb dry air

From the Psychrometric Chart (see Figure B-1):

$H5$ = 27.5 Btu/lb dry air
$H4$ = 20.9 Btu/lb dry air
$\omega 4$= .008 lb H_2O/lb dry air
$\Phi 5$= 40%

Mass Flow Rate Calculation:

$\dot{m}$ = Cooling Load/(H5 - H4)

= (13680 Btu/h)/(27.5 Btu/lb dry air - 20.9 Btu/lb dry air)

= 2072.7 lb dry air/h

Water Consumption Calculation:

Water Consumption = $\dot{m}$ * (ω4 - ω2)

= 2072.7lb dry air * (.008 - .002 lb H_2O/lb dry air)

= 12.4 lb H_2O/h

(d) ω2 = .003 lb H_2O/lb dry air

From the Psychrometric Chart (see Figure B-1):

H5 = 27.9 Btu/lb dry air

H4 = 21.8 Btu/lb dry air

ω4= .0085 lb H_2O/lb dry air

Φ5= 42%

Mass Flow Rate Calculation:

$\dot{m}$ = Cooling Load/(H5 - H4)

= (13680 Btu/h)/(27.9 Btu/lb dry air - 21.8 Btu/lb dry air)

= 2242.6 lb dry air/h

Water Consumption Calculation:

Water Consumption = $\dot{m}$ * (ω4 - ω2)

= 2242.6 lb dry air * (.0085 - .003 lb H_2O/lb dry air)

= 12.3 lb H_2O/h

(e) ω2 = .004 lb H_2O/lb dry air

From the Psychrometric Chart (see Figure B-1):

H5 = 28.6 Btu/lb dry air

H4 = 23 Btu/lb dry air

ω4= .009 lb H_2O/lb dry air

Φ5= 45%

Mass Flow Rate Calculation:

$\dot{m}$ = Coming Load/(H5 - H4)

= (13680 Btu/h)/(28.6 Btu/lb dry air - 23 Btu/lb dry air)

= 2442.9 lb dry air/h

Water Consumption Calculation:

Water Consumption = $\dot{m}$ * (ω4 - ω2)

= 2442.9 lb dry air * (.009 - .004 lb H_2O/lb dry air)

= 12.2 lb H_2O/h

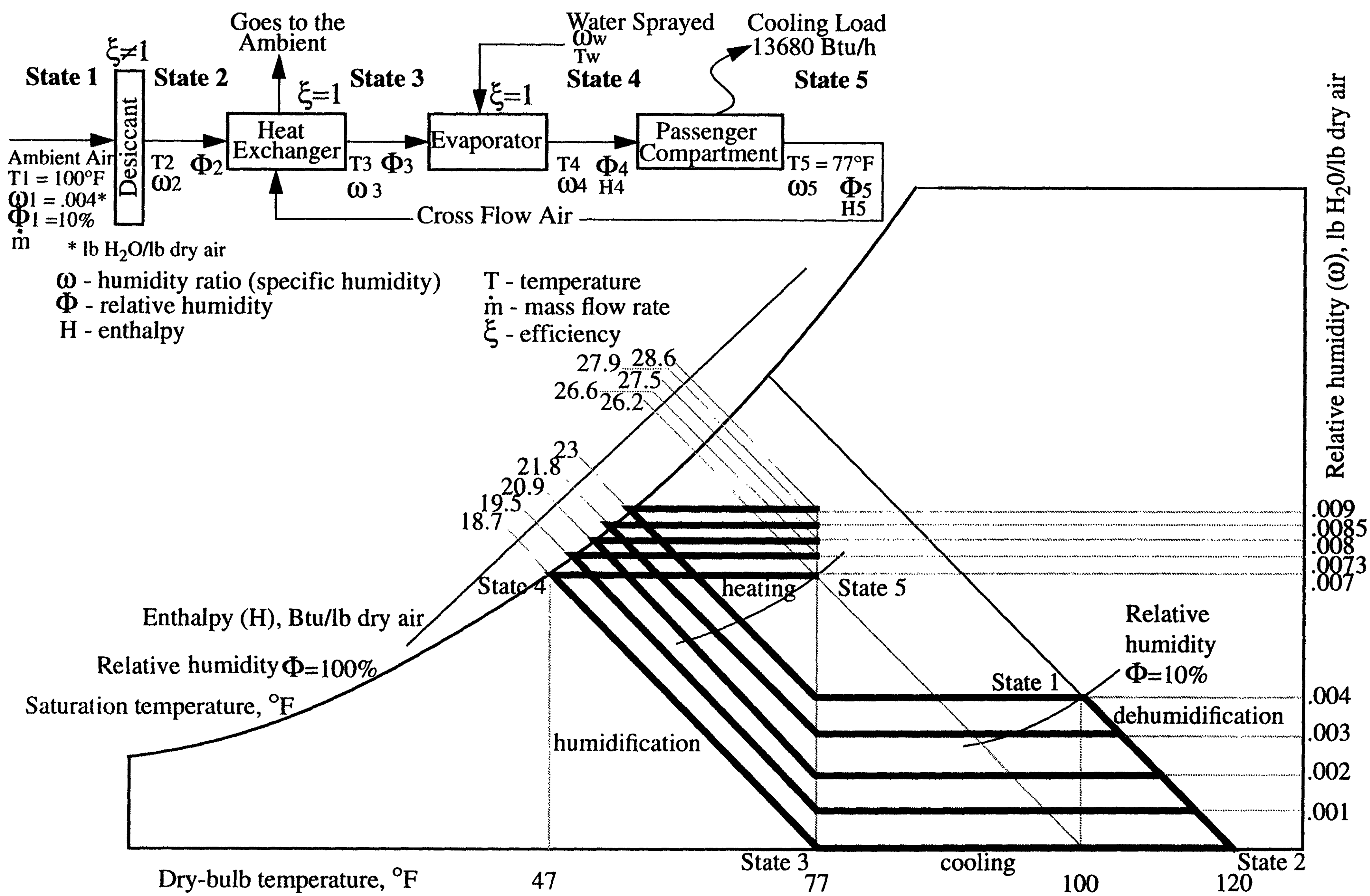


Figure B-1. Skeleton Psychrometric Chart for the Calculations of the Effect of the Desiccant.

B) Calculations of the Effect of the Heat Exchanger on the Mass Flow Rate and the H_2O Consumption of the Evaporative Cooling System

(a) T3 = 77 °F (Ideal Case)

From the Psychrometric Chart (see Figure B-2):
H5 = 28.2 Btu/lb dry air
H4 = 18.7 Btu/lb dry air
$\omega4$= .007 lb H_2O/lb dry air
$\Phi5$= 36%

Mass Flow Rate Calculation:
$\dot{m}$ = Cooling Load/(H5 - H4)
= (13680 Btu/h)/(26.2 Btu/lb dry air - 18.7 Btu/lb dry air)
= 1824 lb dry air/h

Water Consumption Calculation:
Water Consumption = $\dot{m}$ * $\omega4$
= 1824 lb dry air * .007 lb H_2O/lb dry air
= 12.8 lb H_2O/h

(b) T3 = 85 °F

From the Psychrometric Chart (see Figure B-2):
H5 = 27.3 Btu/lb dry air
H4 = 20.6 Btu/lb dry air
$\omega4$= .0078 lb H_2O/lb dry air
$\Phi5$= 40%

Mass Flow Rate Calculation:
$\dot{m}$ = Cooling Load/(H5 - H4)
= (13680 Btu/h)/(27.3 Btu/lb dry air - 20.6 Btu/lb dry air)
= 2041.8 lb dry air/h

Water Consumption Calculation:
Water Consumption = $\dot{m}$ * $\omega4$
= 2041.8 lb dry air * .0078 lb H_2O/lb dry air
= 15.9 lb H_2O/h

(c) T3 = 90 °F

From the Psychrometric Chart (see Figure B-2):
H5 = 27.9 Btu/lb dry air
H4 = 21.8 Btu/lb dry air
$\omega4$= .0085 lb H_2O/lb dry air
$\Phi5$= 42%

Mass Flow Rate Calculation:

$\dot{m}$ = Cooling Load/(H5 - H4)

= (13680 Btu/h)/(27.9 Btu/lb dry air - 21.8 Btu/lb dry air)

= 2242.6 lb dry air/h

Water Consumption Calculation:

Water Consumption = $\dot{m}$ * ω4

= 2242.6 lb dry air * .0085 lb H_2O/lb dry air

= 19.1lb H_2O/h

(d) T3 = 95 °F

From the Psychrometric Chart (see Figure B-2):

H5 = 28.6 Btu/lb dry air

H4 = 23 Btu/lb dry air

ω4= .009 lb H_2O/lb dry air

Φ5= 45%

Mass Flow Rate Calculation:

$\dot{m}$ = Cooling Load/(H5 - H4)

= (13680 Btu/h)/(28.6 Btu/lb dry air - 23 Btu/lb dry air)

= 2442.9 lb dry air/h

Water Consumption Calculation:

Water Consumption = $\dot{m}$ * ω4

= 2442.9 lb dry air * .009 lb H_2O/lb dry air

= 22 lb H_2O/h

(e) T3 = 100 °F

From the Psychrometric Chart (see Figure B-2):

H5 = 29.4 Btu/lb dry air

H4 = 24.3 Btu/lb dry air

ω4= .0099 lb H_2O/lb dry air

Φ5= 50%

Mass Flow Rate Calculation:

$\dot{m}$ = Cooling Load/(H5 - H4)

= (13680 Btu/h)/(29.4 Btu/lb dry air - 24.3 Btu/lb dry air)

= 2682.4 lb dry air/h

Water Consumption Calculation:

Water Consumption = $\dot{m}$ * ω4

= 2682.4 lb dry air * .0099 lb H_2O/lb dry air

= 26.6 lb H_2O/h

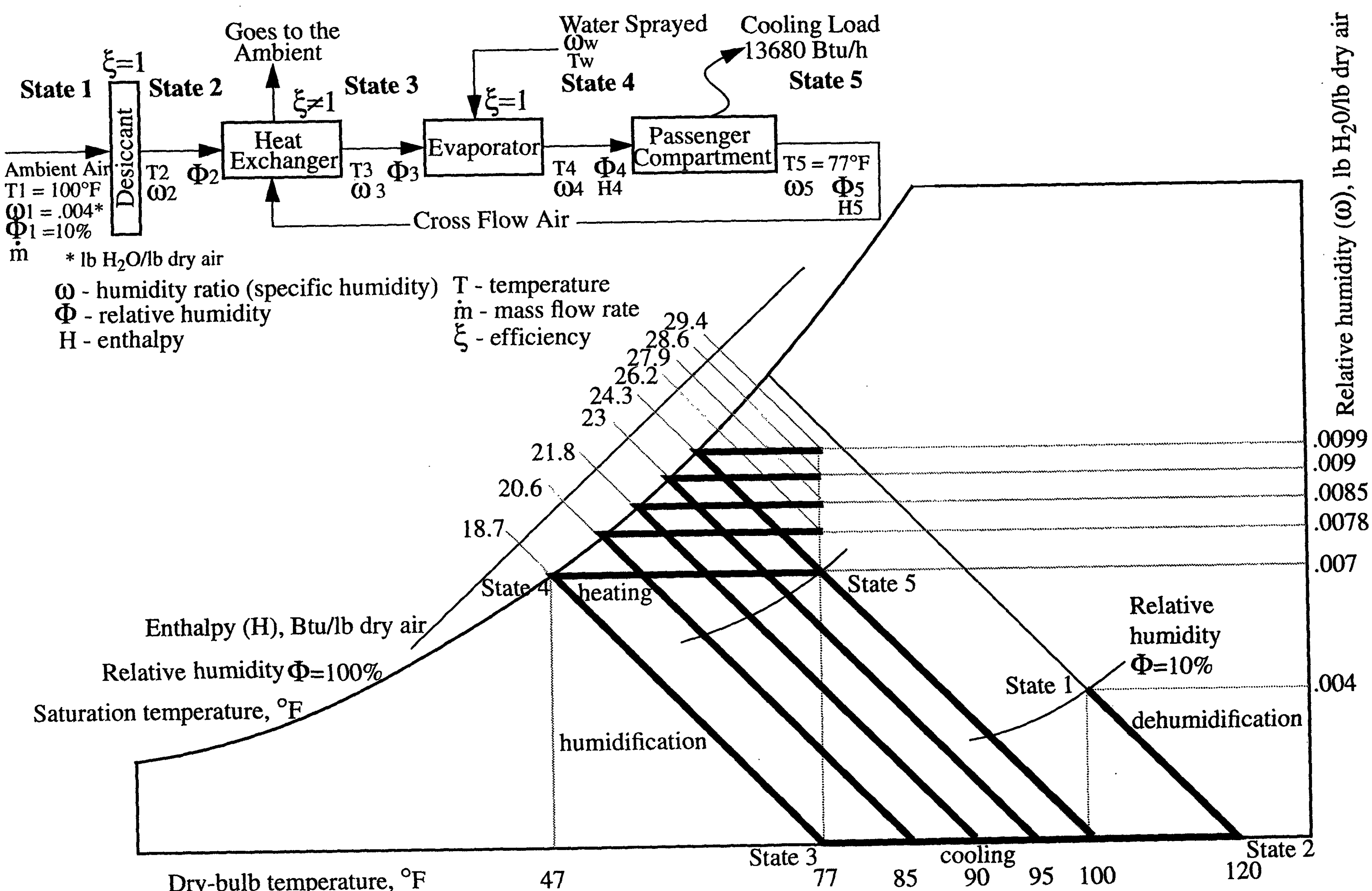


Figure B-2. Skeleton Psychrometric Chart for the Calculations of the Effect of the Heat Exchanger.

C) Calculations of the Effect of the Evaporator on the Mass Flow Rate and the H_2O Consumption of the Evaporative Cooling System

(a) ξ=1; Tw = 47 °F (Ideal Case)

From the Psychrometric Chart (see Figure B-3 and Figure B-4):
H5 = 28.2 Btu/lb dry air
H4 = 18.7 Btu/lb dry air
ω4= .007 lb H_2O/lb dry air
Φ5= 36%

Mass Flow Rate Calculation:
$\dot{m}$ = Cooling Load/(H5 - H4)
= (13680 Btu/h)/(26.2 Btu/lb dry air - 18.7 Btu/lb dry air)
= 1824 lb dry air/h

Water Consumption Calculation:
Water Consumption = $\dot{m}$ * ω4
= 1824 lb dry air * .007 lb H_2O/lb dry air
= 12.8 lb H_2O/h

(b) ξ=1; Tw = 100 °F

From the Psychrometric Chart (see Figure B-3 and Figure B-4):
H5 = 27.7 Btu/lb dry air
H4 = 21.4 Btu/lb dry air
ω4= .0083 lb H_2O/lb dry air
Φ5= 42%

Mass Flow Rate Calculation:
$\dot{m}$ = Cooling Load/(H5 - H4)
= (13680 Btu/h)/(27.7 Btu/lb dry air - 21.4 Btu/lb dry air)
= 2171.43 lb dry air/h

Water Consumption Calculation:
Water Consumption = $\dot{m}$ * ω4
= 2171.43 lb dry air * .0083 lb H_2O/lb dry air
= 18 lb H_2O/h

(c) ξ= .95; Tw = 100 °F

Evaporator's Efficiency Calculation (T3" and T4" are from Figure B-3):
ξ = (T in dry-bulb - T out dry-bulb)/(T in dry-bulb - T in wet-bulb)
ξ = (T3" - T out dry-bulb)/(T3"- T4")
T out dry-bulb = T3" - (ξ / T3"-T4"))
T out dry-bulb = 88 °F - (.95 / (88 °F-52 °F)) = 53.8 °F

From the Psychrometric Chart (see Figure B-4):

T out dry-bulb = T4(c) = 53.8 °F

H5 = 27.1 Btu/lb dry air

H4 = 21.4 Btu/lb dry air

$\omega 4$= .0078 lb H_2O/lb dry air

$\Phi 5$= 40%

Mass Flow Rate Calculation:

$\dot{m}$ = Cooling Load/(H5 - H4)

= (13680 Btu/h)/(27.1 Btu/lb dry air - 21.4 Btu/lb dry air)

= 2400 lb dry air/h

Water Consumption Calculation:

Water Consumption = $\dot{m}$ * $\omega 4$

= 2400 lb dry air * .0078 lb H_2O/lb dry air

= 18.7 lb H_2O/h

(d) ξ= .9; Tw = 100 °F

Evaporator's Efficiency Calculation (T3" and T4" are from Figure B-3):

ξ = (T in dry-bulb - T out dry-bulb)/(T in dry-bulb - T in wet-bulb)

ξ = (T3" - T out dry-bulb)/(T3"- T4")

T out dry-bulb = T3" - (ξ / T3"-T4"))

T out dry-bulb = 88 °F - (.9 / (88 °F-52 °F)) = 55.6 °F

From the Psychrometric Chart (see Figure B-4):

T out dry-bulb = T4(d) = 55.6 °F

H5 = 26.7 Btu/lb dry air

H4 = 21.4 Btu/lb dry air

$\omega 4$= .0075 lb H_2O/lb dry air

$\Phi 5$= 38%

Mass Flow Rate Calculation:

$\dot{m}$ = Cooling Load/(H5 - H4)

= (13680 Btu/h)/(26.7 Btu/lb dry air - 21.4 Btu/lb dry air)

= 2581.1 lb dry air/h

Water Consumption Calculation:

Water Consumption = $\dot{m}$ * $\omega 4$

= 2581.1 lb dry air * .0075 lb H_2O/lb dry air

= 19.4 lb H_2O/h

(e) ξ= .85; Tw = 100 °F

Evaporator's Efficiency Calculation (T3" and T4" are from Figure B-3):

ξ = (T in dry-bulb - T out dry-bulb)/(T in dry-bulb - T in wet-bulb)

ξ = (T3" - T out dry-bulb)/(T3"- T4")
T out dry-bulb = T3" - (ξ / T3"-T4"))
T out dry-bulb = 88 °F - (.85 / (88 °F-52 °F)) = 57.4 °F

From the Psychrometric Chart (see Figure B-4):
T out dry-bulb = T4(e) = 57.4 °F
H5 = 26.2 Btu/lb dry air
H4 = 21.4 Btu/lb dry air
ω4= .007 lb H_2O/lb dry air
Φ5= 35%

Mass Flow Rate Calculation:
$\dot{m}$ = Cooling Load/(H5 - H4)
= (13680 Btu/h)/(26.2 Btu/lb dry air - 21.4 Btu/lb dry air)
= 2850 lb dry air/h

Water Consumption Calculation:
Water Consumption = $\dot{m}$ * ω4
= 2850 lb dry air * .007 lb H_2O/lb dry air
= 20 lb H_2O/h

(f) ξ= .8; Tw = 100 °F

Evaporator's Efficiency Calculation (T3" and T4" are from Figure B-3):
ξ = (T in dry-bulb - T out dry-bulb)/(T in dry-bulb - T in wet-bulb)
ξ = (T3" - T out dry-bulb)/(T3"- T4")
T out dry-bulb = T3" - (ξ / T3"-T4"))
T out dry-bulb = 88 °F - (.8 / (88 °F-52 °F)) = 59.2 °F

From the Psychrometric Chart (see Figure B-4):
T out dry-bulb = T4(f) = 59.2 °F
H5 = 25.8 Btu/lb dry air
H4 = 21.4 Btu/lb dry air
ω4= .0066 lb H_2O/lb dry air
Φ5= 33%

Mass Flow Rate Calculation:
$\dot{m}$ = Cooling Load/(H5 - H4)
= (13680 Btu/h)/(25.8 Btu/lb dry air - 21.4 Btu/lb dry air)
= 3109.1 lb dry air/h

Water Consumption Calculation:
Water Consumption = $\dot{m}$ * ω4
= 3109.1 lb dry air * .0066 lb H_2O/lb dry air
= 20.5 lb H_2O/h

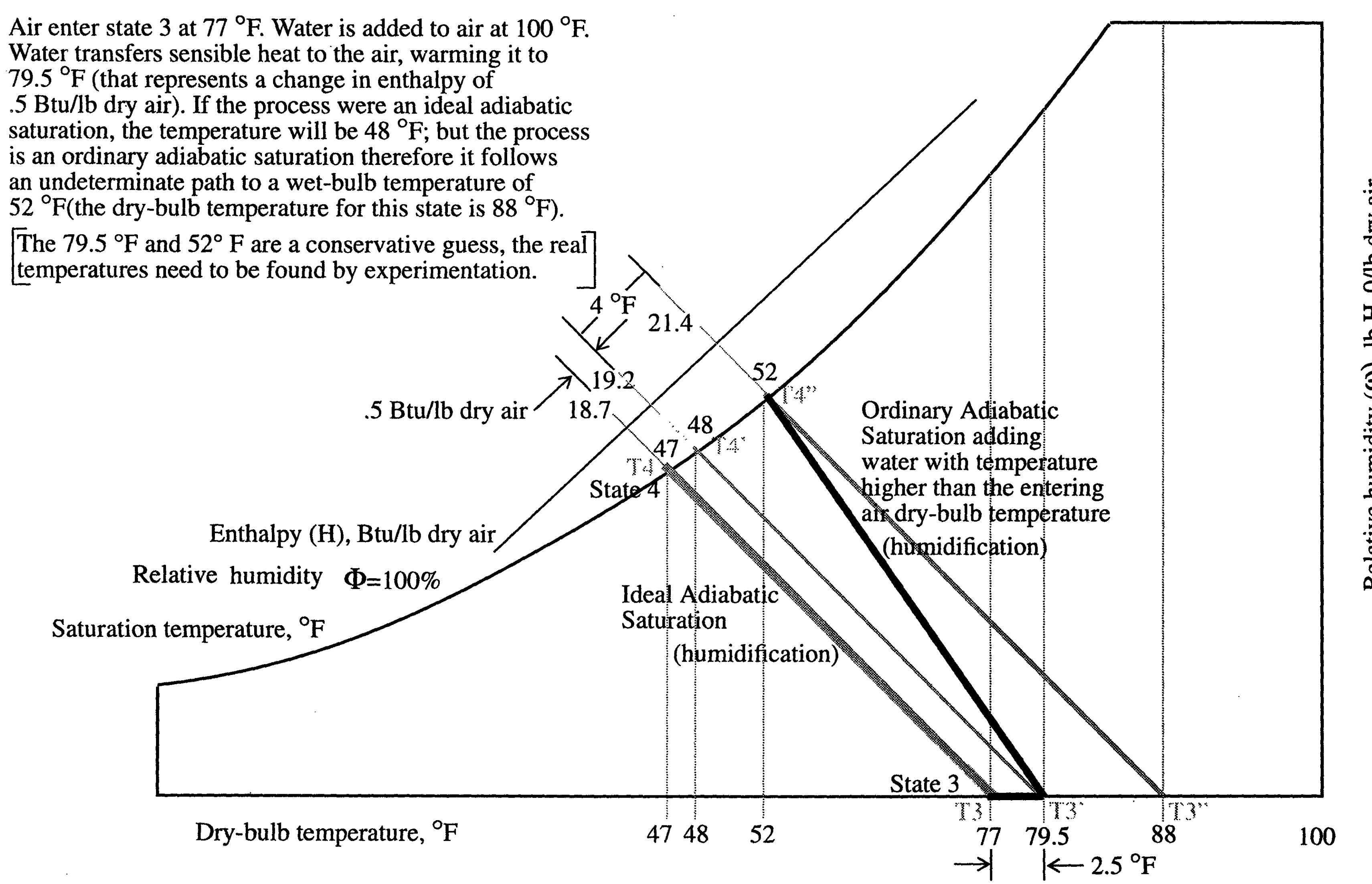


Figure B-3. Skeleton Psychrometric Chart of the Ordinary Evaporative Cooling Adding Water at temperature exceeding the air's entering wet-bulb temperature.

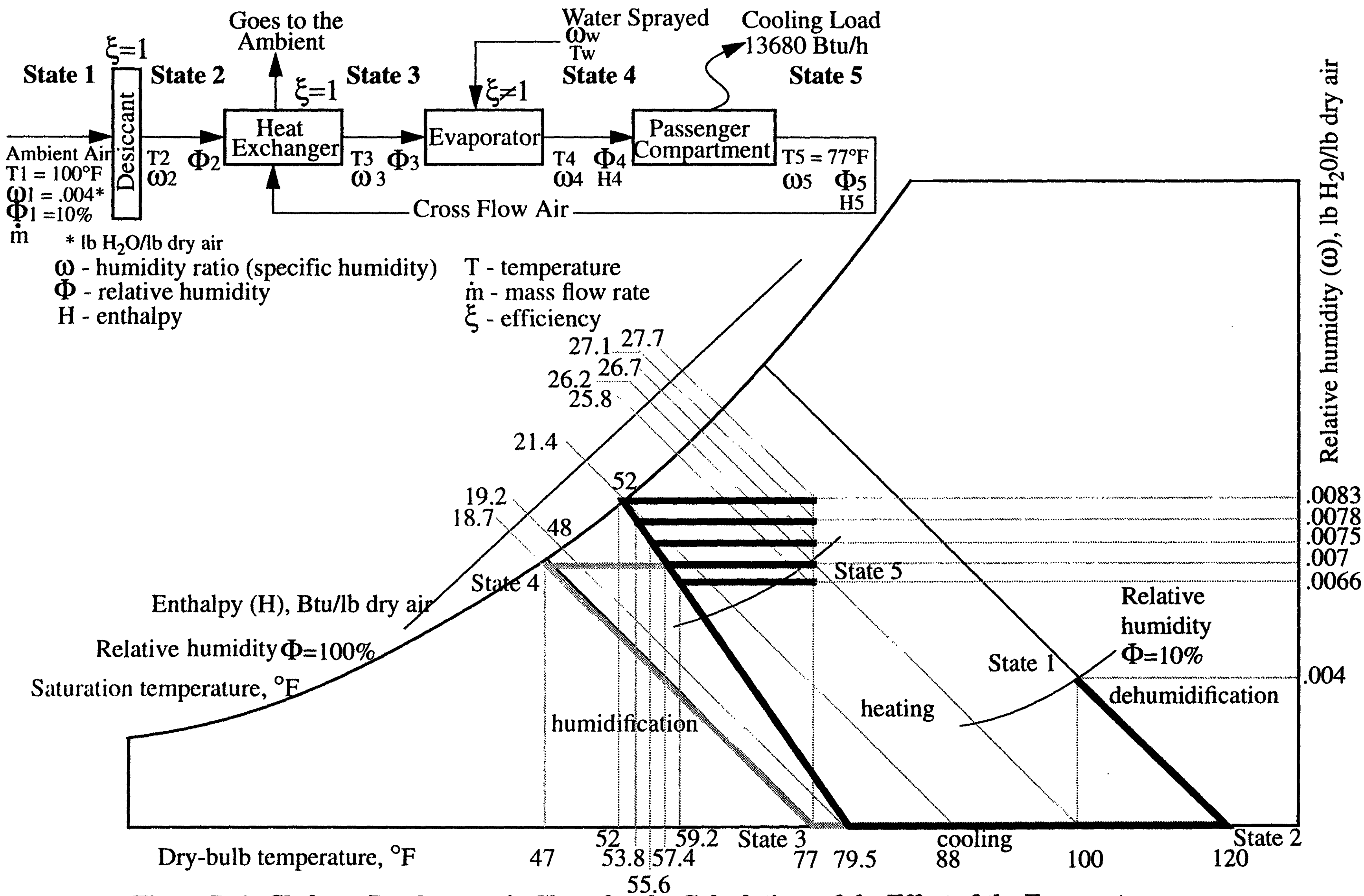


Figure B-4. Skeleton Psychrometric Chart for the Calculations of the Effect of the Evaporator.

Appendix C

A) Calculations for the Mass Flow Rate ($\dot{m}$) for the Main Stream

If the final conditions of air inside the passenger compartment are set at:

$T = 77\ °F$ $\Phi = 40\%$ $\omega = .008\ lb\ H_2O/lb\ dry\ air$

and the Water Consumption (WC) is set at 20 lb H_2O/h

Then:

$$WC = \dot{m} * \omega \quad \Rightarrow \quad \dot{m} = WC / \omega$$
$$= (20\ lb\ H_2O/h) / (.008\ lb\ H_2O/lb\ dry\ air)$$
$$= 2500\ lb\ air/h$$

Since 2500 lb air/h satisfy the conditions, this value is going to be used as the mass flow rate of the main stream of air (stream of air that is going to be cooled).

B) Calculations for the Air Velocity ($\vee$) for the Main Stream

If the cross-sectional area of air is assumed to be .25 of the area of the desiccant wheel, and the diameter of the wheel is 1 ft, then the area is:

$$A = .25\ \{\pi\ [(d/2)]^2\} = .25 * \{3.1416 * [(1\ ft/2)]^2\} = .19635\ ft^2$$

With $\dot{m} = 2500$ lb/h = .694 lb/s and ρ @ 100 °F = .071 lb/ft^3
Therefore:

$$\dot{m} = \rho A \vee \quad \Rightarrow \quad \vee = \dot{m}/(A\ \rho) = (.694\ lb/s)/[(.19635\ ft^2)*(.071\ lb/ft^3)]$$
$$= 49.26\ ft/s$$
$$\approx 50\ ft/s$$

C) Calculations for the Angular Velocity (α) and Tangential Velocity (ϑ) of the Desiccant Wheel

The desiccant wheel dimensions were set on 1 ft diameter (d) and .25 ft thickness (h). For this dimensions the volume is:

$$V = h * \{\pi\ [(d/2)]^2\} = .25\ ft * \{3.1416 * [(1\ ft/2)]^2\} = .19635\ ft^3$$

The density (ρ) of the molecular sieve 3A is 44 lb/ft^3. See Table C-1(UOP Molecular Sieves 1994).
The weight of the desiccant wheel (M) is:

$$\rho = M / V \quad \Rightarrow \quad M = \rho\ V = 44\ lb/ft^3 * .19635\ ft^3 = 8.64\ lb\ of\ desiccant$$

The air that is going to be dehumidified has a $\omega = .004$ lb H_2O/lb dry air and a $\dot{m} = 2500$ lb air/h.

The amount of water carried by this air is:

$$\text{water} = \dot{m} * \omega = .004 \text{ lb } H_2O/\text{lb dry air} * 2500 \text{ lb air/h} = 10 \text{ lb } H_2O/\text{h}$$

If molecular sieve 3A can adsorb 20% of its own weight in water (See Table C-1), then:

$$\frac{X \text{ lb of desiccant}}{10 \text{ lb } H_2O/\text{h}} = \frac{100 \text{ lb of desiccant}}{20 \text{ lb } H_2O/\text{h}}$$

$$\Downarrow$$

$$X = (100/20)*(10) = 50 \text{ lb of desiccant}$$

To take into account that desiccant is being used for a dynamic adsorption, a factor of two is used:

lb of desiccant needed = 2 * 50 = 100 lb of desiccant

Then:

100 lb of desiccant / 8.64 lb of desiccant = 11.57 $\approx$ 11.5

$\therefore$ The desiccant wheel has to rotate 11.5 times per hour.
The frequency (f) is 11.5 rev/h.

The angular velocity (α) is:

$$\alpha = 2\,\pi\,f = 2 * (3.1416 \text{ rad/rev}) * (11.5 \text{ rev/h}) = 72.26 \text{ rad/h} = .02007 \text{ rad/s}$$

The tangential velocity (ϑ) is:

$$\vartheta = 2\,\pi\,r\,f = 2 * (3.1416) * (.5 \text{ ft}) * (11.5 \text{ /h}) = 36.128 \text{ ft/h} = .01004 \text{ ft/s}$$

D) Calculations for the Heat Exchange of the Desiccant Case Cool-Down Area

The calculation is made base on the following assumptions:
(a) although the ambient air velocity ($\vee$) can vary, it is assumed to be 50 ft/s
(d) the desiccant wheel case is assumed to have the same convection heat transfer coefficient as the finned circular tubes surface CF-8.72
(c) the temperature at the base of the fin is assumed to be 600 °F

- **Calculation of $\bar{h}$ for the ambient air**

Properties of air @ 100 °F (Hodge 1990):
$C_p = .24$ Btu/lb °F; $\rho = .071$ lb/ft^3; $\mu = 1.285 \times 10^{-5}$ lb/ft s; Pr = .72

Properties of the finned circular tubes surface CF-8.72 (Figure C-1):
$4r_h = .01288$ ft; $A_f/A = .910$; $\sigma = .524$; $\beta = 163\ ft^2/ft^3$
t = .018 in = .0015 ft, aluminum @ 100 °F $\kappa = 133$ Btu/h ft °F
p = 8.72 per in; L = .27 in

$G = \rho\, v / \sigma = (.071\ lb/ft^3 * 50\ ft/s)/(.524) = 6.7748\ lb/ft^2\ s$

$Re = (G\, 4r_h)/\mu = (6.7748\ lb/ft^2\ s * .01288\ ft)/1.285 \times 10^{-5}\ lb/ft\ s = 6790.6$

From Figure C-1 for $N_R = Re \times 10^{-3} = 6.7906 \approx 6.8$ gives $(\bar{h}\, Pr^{2/3})/(G\, C_p) = .0072$

$\bar{h} = [(G\, C_p) * .0072]/Pr^{2/3}$
$= [(6.7748\ lb/ft^2\ s * .24\ Btu/lb\ °F) * .0072]/(.72)^{2/3}$
$= 1.46 \times 10^{-2}\ Btu/ft^2\ °F\ s = 52.46\ Btu/ft^2\ °F\ h$

- **Calculation of η_f for the fin area**

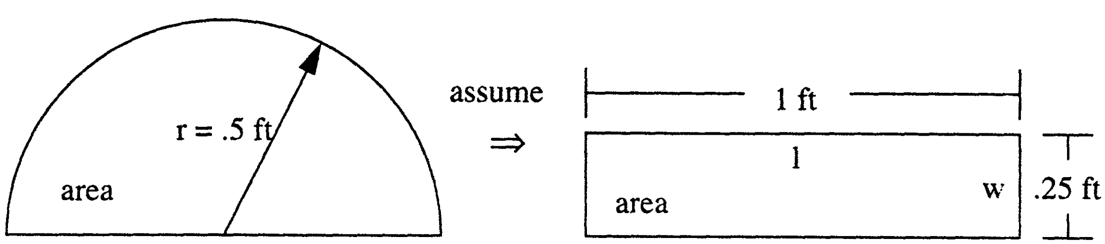
$Lc = L + (t/2) = .27 + (.018\ in/2) = .279\ in = .02325\ ft$
$Ap = t * Lc = (.018\ in) * (.279\ in) = .005\ in^2 = 4.185 \times 10^{-4}\ ft^2$

$Lc^{3/2} * [\bar{h}/(\kappa\, Ap)]^{1/2}$
$= (.02325\ ft)^{3/2} * [(52.46\ Btu/ft^2\ °F\ h)/(133\ Btu/h\ ft\ °F * 4.185 \times 10^{-4}\ ft^2)]^{1/2}$
$= .1088$

From Figure C-2 for $Lc^{3/2} * [\bar{h}/(\kappa\, Ap)]^{1/2} = .1088 \approx .1$ gives $\eta_f = .98$

- **Calculation of the heat transfer rate (q_f) of the fin area**

T_b - temperature of the fin at the base = 600 °F
T_∞ - temperature of the fluid (ambient air) = 100 °F
w - length of the fin (assuming that the average length of the fins is .25 ft)
l - length of the area
t - thickness of the fin = .018 in = .0015 ft
p - fin pitch = 8.72 per in. = 104.64 per ft
P -perimeter of the tip of the fin
N - number of fins



* there are two of this areas, one for the front side of the case and the other for the back of the case

$\theta_b = (T_b - T_\infty) = (600\ °F - 100\ °F) = 500\ °F$

$P = 2\ w + 2\ t = (2 * .25\ ft) + (2 * .0015\ ft) = .503\ ft$

$N = p * l = (104.64\ fin/ft) * (1\ ft) = 104.64\ fins$
The number of fins has to be a hole number
$\therefore N = 104$ fins

$q_{max} = h\ P\ Lc\ \theta_b$
$= (52.46\ Btu/ft^2\ °F\ h) * (.503\ ft) * (.02325\ ft) * (500\ °F)$
$= 306.7533\ Btu/h$

$q_f = N\ \eta_f\ q_{max} = (104) * (.98) * (306.7533\ Btu/h) = 31264.3\ Btu/h$

• Calculation of the heat transfer rate (q_o) of the area without fins

A_o = area without fins

$A_o = [l - (N\ t)] * w = [1ft - (104 * .0015\ ft)] * .25\ ft = .211\ ft^2$

$q_o = h\ A_o\ (T_b - T_\infty) = (52.46\ Btu/ft^2\ °F\ h) * (.211\ ft^2) * (600\ °F - 100\ °F)$
$= 5534.53\ Btu/h$

• Calculation of the total heat transfer rate (q) of the cool-down area

$q = q_f + q_o = 31264.3\ Btu/h * 5534.53\ Btu/h = 36798.83\ Btu/h$

Because there are two of this areas on the desiccant case the total q is:
q total = 2 * q = 2 * 36798.83 Btu/h = 73597.66 Btu/h

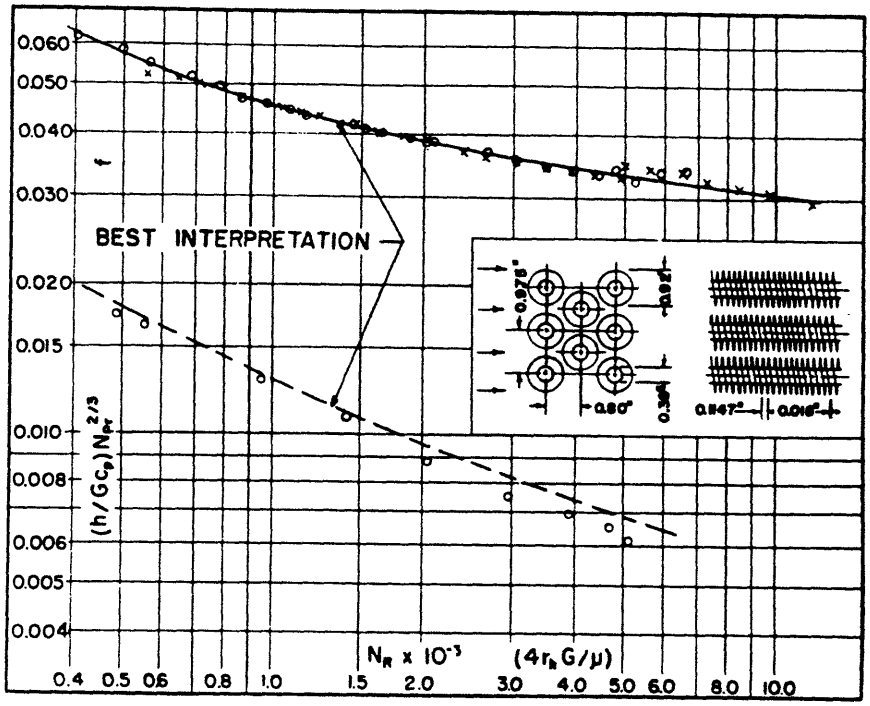
• Calculation of the heat transfer rate (q) of molecular sieve

If the heat transfer of the molecular sieve is the heat absorbed during the regeneration, and the mass flow rate ($\dot{m}$) of the regeneration air is 500 lb/h, the specific heat of air at constant pressure (C_p) @ 450 °F is .246 Btu/lb °F, and the change in temperature (ΔT) is 500 °F, then:

$q = (\dot{m}\ C_p) * (\Delta T) = (500\ lb/h * .246\ Btu/lb\ °F) * (500\ °F) = 61500\ Btu/h$

The desiccant case will be able to cool down the desiccant to a low temperature before the cycle starts again because the heat absorbed by molecular sieve is less than the heat that can be transfer by the cool-down area of the desiccant case.

$q_{\text{molecular sieve}}$		$q_{\text{cool-down area}}$
61500 Btu/h	$<$	73597.66 Btu/h



Fin Pitch, p = 8.72 per in
Flow Passage Hydraulic Diameter, $4r_h$ = .01288 ft
Fin Metal Thickness, t = .018 in; Aluminum
Free-Flow Area/Frontal Area, σ = .524
Heat Transfer Area/Total Volume , β = 163 ft^2/ft^3
Fin Area/Total Area. A_f/A = .910

Figure C-1. Properties of the Finned Circular Tubes, Surface 46.45 T. (Kays and London 1964)

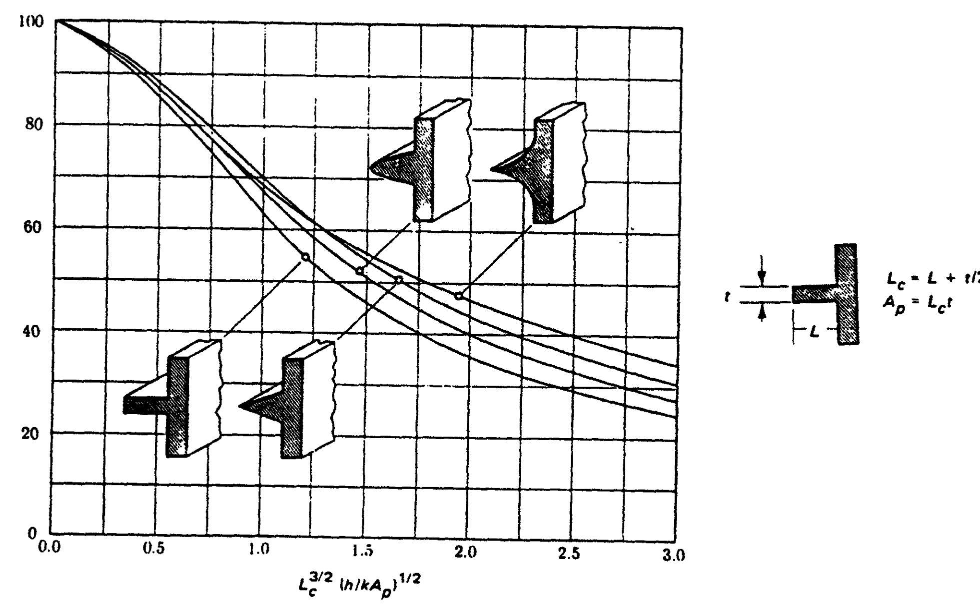


Figure C-2. Fin Efficiencies for Straight Fins. (Hodge 1990)

Table C-1. List of Molecular Sieves and Their Characteristics. (UOP Molecular Sieves)

Basic Types of UNION CARBIDE Molecular Sieves

BASIC TYPE	Nominal Pore Diameter Angstroms	Common Form	Bulk Density lb/cu ft (gm/dm³)	Heat of Adsorption (max) btu/lb H_2O (kcal/kg H_2O)	Equilibrium H_2O Capacity % wt*	Molecules Adsorbed**	Molecules Excluded	APPLICATIONS
3A	3	Powder 1/16-in. Pellets 1/8-in. Pellets 8 x 12 Beads 4 x 8 Beads	30 (480) 47 (750) 47 (750) 44 (705) 44 (705)	1800 (1000)	23 20 20 20 20	Molecules with an effective diameter <3 angstroms, including H_2O and NH_3	Molecules with an effective diameter >3 angstroms, e.g. ethane	The preferred Molecular Sieve adsorbent for the commercial dehydration of unsaturated hydrocarbon streams such as cracked gas, propylene, butadiene, and acetylene. It is also used for drying polar liquids such as methanol and ethanol.
4A	4	Powder 1/16-in. Pellets 1/8-in. Pellets 8 x 12 Beads 4 x 8 Beads 14 x 30 Mesh	30 (480) 45 (720) 45 (720) 45 (720) 45 (720) 44 (705)	1800 (1000)	28.5 22 22 22 22 22	Molecules with an effective diameter <4 angstroms, including ethanol, H_2S, CO_2, SO_2, C_2H_4, C_2H_6, and C_3H_6	Molecules with an effective diameter >4 angstroms, e.g. propane	The preferred Molecular Sieve adsorbent for static dehydration in a closed gas or liquid system. It is used as a static desiccant in household refrigeration systems; in packaging of drugs, electronic components and perishable chemicals; and as a water scavenger in paint and plastic systems. Also used commercially in drying saturated hydrocarbon streams.
5A	5	Powder 1/16-in. Pellets 1/8-in. Pellets	30 (480) 43 (690) 43 (690)	1800 (1000)	28 21.5 21.5	Molecules with an effective diameter <5 angstroms, including n-C_4H_9OH**, n-C_4H_{10}** C_3H_8 to $C_{22}H_{46}$, R-12	Molecules with an effective diameter >5 angstroms, e.g. iso compounds and all 4 carbon rings	Separates normal paraffins from branched-chain and cyclic hydrocarbons through a selective adsorption process.
13X	10	Powder 1/16-in. Pellets 1/8-in. Pellets 8 x 12 Beads 4 x 8 Beads	30 (480) 40 (640) 40 (640) 40 (640) 40 (640)	1800 (1000)	36 28 5 28 5 28 5 28 5	Molecules with an effective diameter <10 angstroms	Molecules with an effective diameter >10 angstroms, e.g. $(C_4F_9)_3N$	Used commercially for general gas drying, air plant feed purification (simultaneous removal of H_2O and CO_2) and liquid hydrocarbon and natural gas sweetening (H_2S and mercaptan removal).

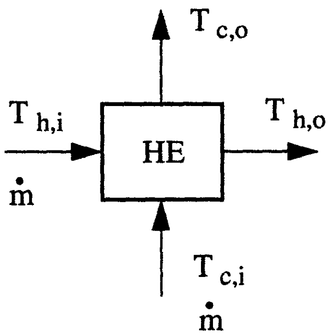
*Lbs. H_2O/100 lbs. activated adsorbent at 17.5 mm Hg, 25°C.

**Each type adsorbs listed molecules plus those of preceding type

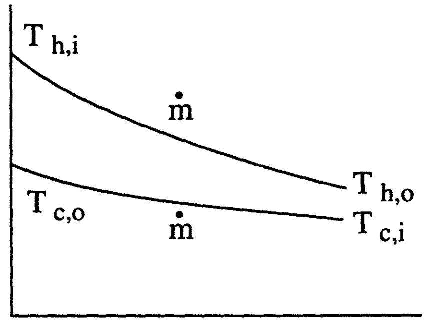
Appendix D

A) Calculations for the Main Stream Heat Exchanger Design

(a) Heat Exchanger with Hot- and Cold-Side Mass Flow Rate (m) of 2500 lb/h



Given:
$T_{h,i} = 115\ ^{\circ}F$
$T_{c,i} = 77\ ^{\circ}F$
$\dot{m} = 2500$ lb/h
Cross-Flow
Heat Exchanger



Assume that $T_{c,o} = 105\ ^{\circ}F$; then

$\overline{T}_c = [(105 - 77)/2] + 77 = 91\ ^{\circ}F$

Properties of air @ 91 °F (Hodge 1990):
$C_p = .23991$ Btu/lb °F

The maximum heat transfer rate (q) between the two fluids is:
$q_{max} = (\dot{m}\, C_p)_{min} * (T_{h,i} - T_{c,i}) = (2500\ \text{lb/h} * .23991\ \text{Btu/lb}\ ^{\circ}F) * (115 - 77)^{\circ}F$
$= 22791.45$ Btu/h

Because air is the working fluid for both sides and the temperature difference between the two streams is not much, the C_p hot-side $\approx C_p$ cold-side.
Then:
$(\dot{m}\, C_p)_h = (\dot{m}\, C_p)_c$

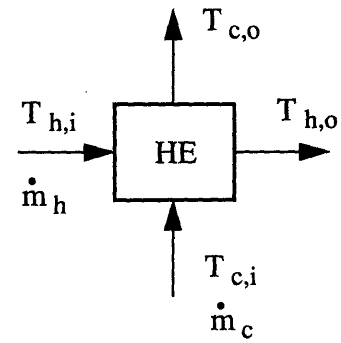
$\therefore\ (\dot{m}\, C_p)_{min}/(\dot{m}\, C_p)_{max} = 1$

From Figure D-1 the maximum effectiveness (ξ) for a cross-flow heat exchanger with both fluids unmixed and $(\dot{m}\, C_p)_{min}/(\dot{m}\, C_p)_{max} = 1$ is .75.
Then:
$\xi = q/q_{max} \quad \Rightarrow \quad q = \xi * q_{max} = .75 * (22791.45\ \text{Btu/h}) = 17093.59$ Btu/h

$T_{h,o} = T_{h,i} - [q/(\dot{m}\, Cp)] = 115\ ^{\circ}F - [17093.59\ \text{Btu/h} / (2500\ \text{lb/h} * .23991\ \text{Btu/lb}\ ^{\circ}F)]$
$= 86.5\ ^{\circ}F$
$T_{c,o} = [q/(\dot{m}\, Cp)] + T_{c,i} = [17093.59\ \text{Btu/h} / (2500\ \text{lb/h} * .23991\ \text{Btu/lb}\ ^{\circ}F)] + 77\ ^{\circ}F$
$= 105.5\ ^{\circ}F$
The assumption of $T_{c,o} = 105\ ^{\circ}F$ is good.

The $T_{h,o} = 86.5^{\circ}F$ is not low enough to supply the cooling capacity of 13680 Btu/h at the passenger compartment. Therefore, this approach is possible but is not the best.

(b) Heat Exchanger with Hot-Side Mass Flow Rate ($\dot{m}$) of 2500 lb/h and Cold-Side Mass Flow Rate of 7500 lb/h



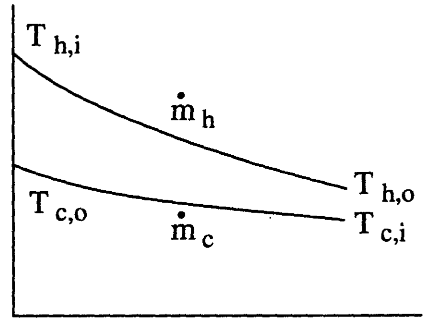
Given:

$T_{h,i} = 115\ ^{\circ}F$

$\dot{m}_h = 2500$ lb/h

$\dot{m}_c = 7500$ lb/h

Cross-Flow
Heat Exchanger



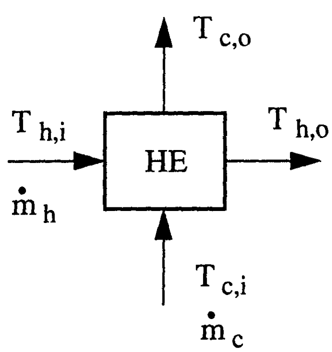
The $\dot{m}$ of the cold-side air is given by the mixture of two streams:
1) outside air @ T = 100 °F and $\dot{m}$ = 5000 lb/h
2) passenger air @ T= 77 °F and $\dot{m}$ = 2500 lb/h

The temperature of the cold-side air is given by:
$T_{c,i}$ = (% mass air * temperature)$_{\text{outside air}}$+ (% mass air * temperature)$_{\text{passenger air}}$
= (2/3 * 100) + (1/3 * 77) = 92.3 °F

The hot-side air always leaves the heat exchanger at a higher temperature than the inlet cold-side air. The $T_{c,i}$ = 92 °F is not low enough to supply the cooling capacity of 13680 Btu/h at the passenger compartment. Therefore, this is not a good approach.

(c) Two Heat Exchangers in Series

1) Hot-Side Mass Flow Rate ($\dot{m}$) of 2500 lb/h and Cold-Side Mass Flow Rate of 5000 lb/h [Hot-Heat-Exchanger]



Given:

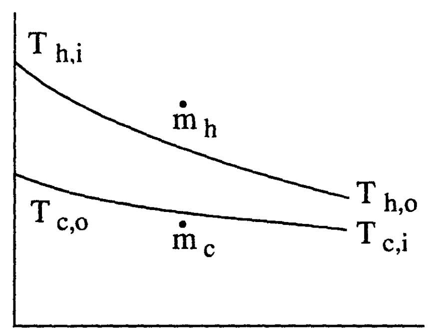
$T_{h,i} = 115\ ^{\circ}F$

$T_{c,i} = 100\ ^{\circ}F$

$\dot{m}_h = 2500$ lb/h

$\dot{m}_c = 5000$ lb/h

Cross-Flow
Heat Exchanger



*The cold-side air is the outside air @ T = 100 °F

Assume that $T_{c,o}$ = 107 °F; then

$\overline{T}_c$ = [(107 - 100)/2] +100 = 103.5 °F

Properties of air @ 104 °F (Hodge 1990):
C_p = .24004 Btu/lb °F; ρ = .07056 lb/ft^3; $\mu = 1.2912 \times 10^{-5}$ lb/ft s; Pr = .70518

The maximum heat transfer rate (q) between the two fluids is:

$$q_{max} = (\dot{m}\, C_p)_{min} * (T_{h,i} - T_{c,i}) = (2500 \text{ lb/h} * .24004 \text{ Btu/lb °F}) * (115 - 100)\text{°F}$$
$$= 9001.5 \text{ Btu/h}$$

Because air is the working fluid for both sides and the temperature difference between the two streams is not much, the C_p hot-side $\approx$ C_p cold-side.
Then:

$$(\dot{m}\, C_p)_h = .5\, (\dot{m}\, C_p)_c$$

$$\therefore\ (\dot{m}\, C_p)_{min}/(\dot{m}\, C_p)_{max} = .5$$

From Figure D-1 the maximum effectiveness (ξ) for a cross-flow heat exchanger with both fluids unmixed and $(\dot{m}\, C_p)_{min}/(\dot{m}\, C_p)_{max}$ =.5 is .9.
Then:

$$\xi = q/q_{max} \quad \Rightarrow \quad q = \xi * q_{max} = .9 * (9001.5 \text{ Btu/h}) = 8101.35 \text{ Btu/h}$$

$$T_{h,o} = T_{h,i} - [q/(\dot{m}\, Cp)_h] = 115 \text{ °F} - [8101.35 \text{ Btu/h} / (2500 \text{ lb/h} * .24004 \text{ Btu/lb °F})]$$
$$= 101.5 \text{ °F}$$
$$T_{c,o} = [q/(\dot{m}\, Cp)_c] + T_{c,i} = [8101.35 \text{ Btu/h} / (5000 \text{ lb/h} * .24004 \text{ Btu/lb °F})] + 100 \text{ °F}$$
$$= 106.75 \text{ °F}$$

The assumption of $T_{c,o}$ = 107 °F is good.

$$\overline{T}_h = [(115 - 102)/2] + 102 = 108.5 \text{ °F}$$

Properties of air @ 108 °F (Hodge 1990):

C_p = .24008 Btu/lb °F; ρ = .07012 lb/ft^3; $\mu = 1.2974 \times 10^{-5}$ lb/ft s; Pr = .70476

From Figure D-1 for a cross-flow heat exchanger with both fluids unmixed with $(\dot{m}\, C_p)_{min}/(\dot{m}\, C_p)_{max}$ =.5 and ξ = .9, gives a Number of Transfer Units NTU = 5.5.
Then:

$$NTU = (U_h\, A_h)/(\dot{m}\, C_p)_{min} \quad \Rightarrow \quad U\, A = NTU * (\dot{m}\, C_p)_{min}$$
$$= 5.5\, (2500 \text{ lb/h} * .24008 \text{ Btu/lb °F})$$
$$= 3301.1 \text{ Btu/h °F}$$

If the velocity of air for the hot- and cold-side air stream is v = 50 ft/s and the heat exchanger uses surface 46.45 T for both sides, the heat exchanger design is:

Plain plate-fin surface 46.45 T is shown in Figure D-2.
Properties of surface 46.45 T:
$4r_h$ = .002643 ft; A_f/A = .837; β = 1332.45 ft^2/ft^3
t = .002 in, stainless steel AISI 304 @ 107 °F κ = 8.609 Btu/h ft °F
b = .1 in = .0083 ft; c = 2.63 in = .2192 ft

• Calculation of $\bar{h}\eta_t A$ for the hot-side air

$G = \rho\ v = .07012\ lb/ft^3 * 50\ ft/s = 3.506\ lb/ft^2\ s$

$Re = (G\ 4r_h)/\mu = (3.506\ lb/ft^2\ s * .002643\ ft)/1.2974 \times 10^{-5}\ lb/ft\ s = 714.2$

From Figure D-2 for $N_R = Re \times 10^{-3} = .7142 \approx .7$ gives $(\bar{h}\ Pr^{2/3})/(G\ C_p) = .0065$

$\bar{h} = [(G\ C_p) * .0065]/Pr^{2/3}$
$= [(3.506\ lb/ft^2\ s * .24008\ Btu/lb\ {}^\circ F) * .0065]/(.70476)^{2/3}$
$= 6.908 \times 10^{-3}\ Btu/ft^2\ {}^\circ F\ s = 24.87\ Btu/ft^2\ {}^\circ F\ h$

$Lc = .5 * b = .5 * (.1\ in) = .05\ in = .004167\ ft$
$Ap = t * Lc = (.002\ in) * (.05\ in) = .0001\ in^2 = 8.3 \times 10^{-6}\ ft^2$

$Lc^{3/2} * [\bar{h}/(\kappa\ Ap)]^{1/2}$
$= (.004167\ ft)^{3/2} * [(24.87\ Btu/ft^2\ {}^\circ F\ h)/(8.609\ Btu/h\ ft\ {}^\circ F * 8.3 \times 10^{-6}\ ft^2)]^{1/2}$
$= .1587$

From Figure D-3 for $Lc^{3/2} * [\bar{h}/(\kappa\ Ap)]^{1/2} = .1587 \approx .16$ gives $\eta_f = .98$

$\eta_t = 1 - [A_f/A * (1 - \eta_f)] = 1 - [.837 * (1 - .98)] = .9833$

If	fin length in cross-flow direction	a
	plate spacing	b = .1 in = .0083 ft
	fin length in flow direction	c = 2.63 in = .2192 ft
then		
	volume between plates	$V = a * b * c$
		$= a * .0083\ ft * .2192\ ft = .0018\ a\ ft^2$

β = Total heat transfer area/ volume between plates = A / V
$\Rightarrow \quad A = \beta\ V = 1332.45\ ft^2/ft^3 * .0018\ a\ ft^2 = 2.4\ a\ ft^2/ft$

$\bar{h}\eta_t A = 24.87\ Btu/ft^2\ {}^\circ F\ h * .9833 * 2.4\ a\ ft^2/ft = 58.69\ a\ Btu/ft\ {}^\circ F\ h$

• Calculation of $\bar{h}\eta_t A$ for the cold-side air

$G = \rho\ v = .07056\ lb/ft^3 * 50\ ft/s = 3.528\ lb/ft^2\ s$

$Re = (G\ 4r_h)/\mu = (3.528\ lb/ft^2\ s * .002643\ ft)/1.2912 \times 10^{-5}\ lb/ft\ s = 722.2$

From Figure D-2 for $N_R = Re \times 10^{-3} = .7222 \approx .7$ gives $(\bar{h}\ Pr^{2/3})/(G\ C_p) = .0065$

$\bar{h} = [(G\ C_p) * .0065]/Pr^{2/3}$
$= [(3.528\ lb/ft^2\ s * .24004\ Btu/lb\ {}^\circ F) * .0065]/(.70518)^{2/3}$
$= 6.948 \times 10^{-3}\ Btu/ft^2\ {}^\circ F\ s = 25.01\ Btu/ft^2\ {}^\circ F\ h$

$Lc = .5 * b = .5 * (.1 \text{ in}) = .05 \text{ in} = .004167 \text{ ft}$
$Ap = t * Lc = (.002 \text{ in}) * (.05 \text{ in}) = .0001 \text{ in}^2 = 8.3 \times 10^{-6} \text{ ft}^2$

$$Lc^{3/2} * [\bar{h}/(\kappa\, Ap)]^{1/2}$$
$$= (.004167 \text{ ft})^{3/2} * [(25.01 \text{ Btu/ft}^2\ {}^\circ\text{F h})/(8.609 \text{ Btu/h ft }{}^\circ\text{F} * 8.3 \times 10^{-6} \text{ ft}^2)]^{1/2}$$
$$= .1591$$

From Figure D-3 for $Lc^{3/2} * [\bar{h}/(\kappa\, Ap)]^{1/2} = .1591 \approx .16$ gives $\eta_f = .98$

$$\eta_t = 1 - [A_f/A * (1 - \eta_f)] = 1 - [.837 * (1 - .98)] = .9833$$

If	fin length in cross-flow direction	a
	plate spacing	b = .1 in = .0083 ft
	fin length in flow direction	c = 2.63 in = .2192 ft
then		
	volume between plates	$V = a * b * c$
		$= a * .0083 \text{ ft} * .2192 \text{ ft} = .0018\, a \text{ ft}^2$

β = Total heat transfer area/ volume between plates = A / V
$\Rightarrow \quad A = \beta V = 1332.45 \text{ ft}^2/\text{ft}^3 * .0018\, a \text{ ft}^2 = 2.4\, a \text{ ft}^2/\text{ft}$

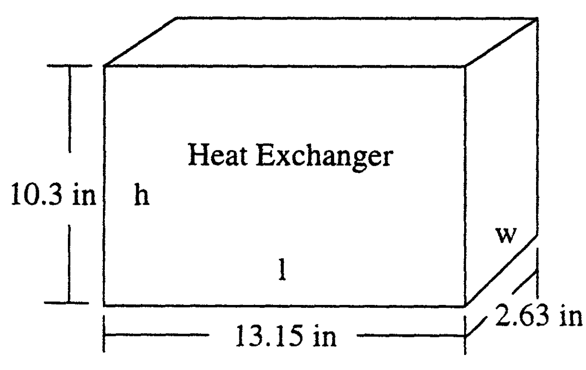
$$\bar{h}\eta_t A = 25.01 \text{ Btu/ft}^2\ {}^\circ\text{F h} * .9833 * 2.4\, a \text{ ft}^2/\text{ft} = 59.02\, a \text{ Btu/ft }{}^\circ\text{F h}$$

- **Calculation of the heat exchanger volume**

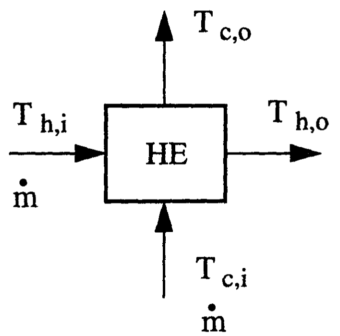
$$U A = 1/\{[1/(\bar{h}\eta_t A)_h] + [1/(\bar{h}\eta_t A)_c]\}$$
$$\Downarrow$$
$$3301.1 \text{ Btu/h }{}^\circ\text{F} = 1/\{[1/(58.69\, a \text{ Btu/ft }{}^\circ\text{F h})_h] + [1/(59.02\, a \text{ Btu/ft }{}^\circ\text{F h})_c]\}$$
$$\Downarrow$$
$$a = 112.18 \text{ ft}$$

To design the heat exchanger volume the width (w) has to be w = c = 2.63 in and the length (l) has to be a multiple of b. If l = 5 c = 5 * 2.63 in = 13.15 in = 1.0958 ft.
Then:
$a / l = 112.18 \text{ ft} / 1.0958 \text{ ft} = 102.36 \approx 103$ rows
103 row @ height b = .1 in gives a total height (h) of 10.3 in

$\therefore V = w * h * l = 2.63 \text{ in} * 10.3 \text{ in} * 13.15 \text{ in}$



2) Hot-Side Mass Flow Rate ($\dot{m}$) of 2500 lb/h and Cold-Side Mass Flow Rate of 2500 lb/h [Cold-Heat-Exchanger]



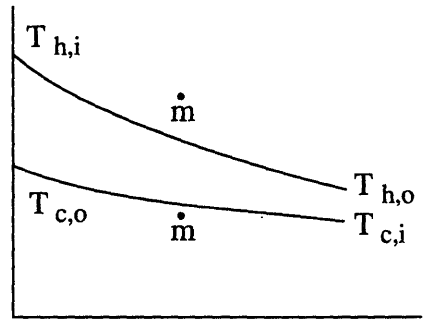
Given:

$T_{h,i}$ = 102 °F

$T_{c,i}$ = 77 °F

$\dot{m}$ = 2500 lb/h

Cross-Flow
Heat Exchanger



*The cold-side air is the return passenger compartment air @ T = 77 °F
*The inside hot-side air is the outside hot-side air of the first heat exchanger @ 102 °F

Assume that $T_{c,o}$ = 96 °F; then

$\overline{T}_c = [(96 - 77)/2] + 77 = 86.5$ °F

Properties of air @ 87 °F (Hodge 1990):
C_p = .23987 Btu/lb °F; ρ = .07722 lb/ft^3; $\mu = 1.2622 \times 10^{-5}$ lb/ft s; Pr = .70644

The maximum heat transfer rate (q) between the two fluids is:
$q_{max} = (\dot{m}\, C_p)_{min} * (T_{h,i} - T_{c,i})$ = (2500 lb/h *.23987 Btu/lb °F) * (102 - 77)°F
= 14991.875 Btu/h

Because air is the working fluid for both sides and the temperature difference between the two streams is not much, the C_p hot-side ≈ C_p cold-side.
Then:
$(\dot{m}\, C_p)_h = (\dot{m}\, C_p)_c$

$\therefore\ (\dot{m}\, C_p)_{min}/(\dot{m}\, C_p)_{max} = 1$

From Figure D-1 the maximum effectiveness (ξ) for a cross-flow heat exchanger with both fluids unmixed and $(\dot{m}\, C_p)_{min}/(\dot{m}\, C_p)_{max} = 1$ is .75.
Then:
$\xi = q/q_{max} \quad \Rightarrow \quad q = \xi * q_{max}$ = .75 * (1499.875 Btu/h) = 11243.9 Btu/h

$T_{h,o} = T_{h,i} - [q/(\dot{m}\, Cp)_h]$ = 102 °F - [11243.9 Btu/h / (2500 lb/h *.23987 Btu/lb °F)]
= 83.25 °F
$T_{c,o} = [q/(\dot{m}\, Cp)_c] + T_{c,i}$ = [11243.9 Btu/h / (2500 lb/h *.23987 Btu/lb °F)] + 77 °F
= 95.75 °F
The assumption of $T_{c,o}$ = 96 °F is good.

$\overline{T}_h = [(102 - 83)/2] + 83 = 92.5\ ^{\circ}F$

Properties of air @ 93 °F (Hodge 1990):

$C_p = .23993$ Btu/lb °F; $\rho = .07205$ lb/ft^3; $\mu = 1.2728 \times 10^{-5}$ lb/ft s; Pr = .70602

From Figure D-1 for a cross-flow heat exchanger with both fluids unmixed with $(\dot{m}\, C_p)_{min}/(\dot{m}\, C_p)_{max} = .5$ and $\xi = .9$, gives a Number of Transfer Units NTU = 5.5. Then:

$$NTU = (U_h\, A_h)/(\dot{m}\, C_p)_{min} \quad \Rightarrow \quad U\,A = NTU * (\dot{m}\, C_p)_{min}$$
$$= 5.5\ (2500\ \text{lb/h} * .23993\ \text{Btu/lb}\ ^{\circ}F)$$
$$= 3299\ \text{Btu/h}\ ^{\circ}F$$

If the velocity of air for the hot- and cold-side air stream is $v = 50$ ft/s and the heat exchanger uses surface 46.45 T for both sides, the heat exchanger design is:

Plain plate-fin surface 46.45 T is shown in Figure D-2.
Properties of surface 46.45 T:
$4r_h = .002643$ ft; $A_f/A = .837$; $\beta = 1332.45$ ft^2/ft^3
t = .002 in, stainless steel AISI 304 @ 107 °F $\kappa = 8.609$ Btu/h ft °F
b = .1 in = .0083 ft; c = 2.63 in = .2192 ft

• **Calculation of $\overline{h}\eta_t A$ for the hot-side air**

$G = \rho\, v = .07205\ \text{lb/ft}^3 * 50\ \text{ft/s} = 3.6025\ \text{lb/ft}^2\ \text{s}$

$Re = (G\, 4r_h)/\mu = (3.6025\ \text{lb/ft}^2\ \text{s} * .002643\ \text{ft})/1.2728 \times 10^{-5}\ \text{lb/ft s} = 748.1$

From Figure D-2 for $N_R = Re \times 10^{-3} = .7481 \approx .75$ gives $(\overline{h}\, Pr^{2/3})/(G\, C_p) = .0061$

$$\overline{h} = [(G\, C_p) * .0061]/Pr^{2/3}$$
$$= [(3.6025\ \text{lb/ft}^2\ \text{s} * .23993\ \text{Btu/lb}\ ^{\circ}F) * .0061]/(.70602)^{2/3}$$
$$= 6.65 \times 10^{-3}\ \text{Btu/ft}^2\ ^{\circ}F\ \text{s} = 23.94\ \text{Btu/ft}^2\ ^{\circ}F\ \text{h}$$

$Lc = .5 * b = .5 * (.1\ \text{in}) = .05\ \text{in} = .004167\ \text{ft}$
$Ap = t * Lc = (.002\ \text{in}) * (.05\ \text{in}) = .0001\ \text{in}^2 = 8.3 \times 10^{-6}\ \text{ft}^2$

$$Lc^{3/2} * [\overline{h}/(\kappa\, Ap)]^{1/2}$$
$$= (.004167\ \text{ft})^{3/2} * [(23.94\ \text{Btu/ft}^2\ ^{\circ}F\ \text{h})/(8.609\ \text{Btu/h ft}\ ^{\circ}F * 8.3 \times 10^{-6}\ \text{ft}^2)]^{1/2}$$
$$= .1557$$

From Figure D-3 for $Lc^{3/2} * [\overline{h}/(\kappa\, Ap)]^{1/2} = .1557 \approx .15$ gives $\eta_f = .98$

$\eta_t = 1 - [A_f/A * (1 - \eta_f)] = 1 - [.837 * (1 - .98)] = .9833$

If	fin length in cross-flow direction	a
	plate spacing	b = .1 in = .0083 ft
	fin length in flow direction	c = 2.63 in = .2192 ft
then		
	volume between plates	V = a * b * c
		= a *.0083 ft *.2192 ft = .0018 a ft^2

β = Total heat transfer area/ volume between plates = A / V

$\Rightarrow$ A= β V = 1332.45 ft^2/ft^3 *.0018 a ft^2 = 2.4 a ft^2/ft

$\bar{h}\eta_t A$ = 23.94 Btu/ft^2 °F h *.9833 * 2.4 a ft^2/ft = 56.5 a Btu/ft °F h

- **Calculation of $\bar{h}\eta_t A$ for the cold-side air**

G = ρ v = .07722 lb/ft^3 * 50 ft/s = 3.861 lb/ft^2 s

Re = (G $4r_h$)/μ = (3.861 lb/ft^2 s *.002643 ft)/1.2622 x 10^{-5} lb/ft s = 808.48

From Figure D-2 for N_R = Re x 10^{-3} = .80848 ≈ .8 gives ($\bar{h}$ $Pr^{2/3}$)/(G C_p) = .0059

$\bar{h}$ = [(G C_p) *.0059]/$Pr^{2/3}$
= [(3.861 lb/ft^2 s *.23987 Btu/lb °F) *.0059]/$(.70644)^{2/3}$
= 6.835 x 10^{-3} Btu/ft^2 °F s = 25.61 Btu/ft^2 °F h

Lc = .5 * b = .5 * (.1 in) = .05 in = .004167 ft
Ap = t * Lc = (.002 in) * (.05 in) = .0001 in^2 = 8.3 x 10^{-6} ft^2

$Lc^{3/2}$ * $[\bar{h}/(\kappa\ Ap)]^{1/2}$
= $(.004167\ ft)^{3/2}$ * $[(24.61\ Btu/ft^2\ °F\ h)/(8.609\ Btu/h\ ft\ °F * 8.3 x 10^{-6}\ ft^2)]^{1/2}$
= .1579

From Figure D-3 for $Lc^{3/2}$ * $[\bar{h}/(\kappa\ Ap)]^{1/2}$ = .1579 ≈ .16 gives η_f = .98

η_t = 1 - [A_f/A * (1 - η_f)] = 1 - [.837 * (1 - .98)] = .9833

If	fin length in cross-flow direction	a
	plate spacing	b = .1 in = .0083 ft
	fin length in flow direction	c = 2.63 in = .2192 ft
then		
	volume between plates	V = a * b * c
		= a *.0083 ft *.2192 ft = .0018 a ft^2

β = Total heat transfer area/ volume between plates = A / V

$\Rightarrow$ A= β V = 1332.45 ft^2/ft^3 *.0018 a ft^2 = 2.4 a ft^2/ft

$\bar{h}\eta_t A$ = 25.61 Btu/ft^2 °F h *.9833 * 2.4 a ft^2/ft = 60.44 a Btu/ft °F h

• **Calculation of the heat exchanger volume**

$U A = 1/\{[1/(\bar{h}\eta_t A)_h] + [1/(\bar{h}\eta_t A)_c]\}$

$\Downarrow$

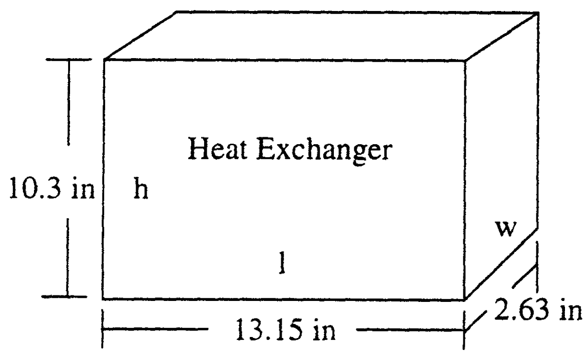
3299 Btu/h °F = 1/{[1/(56.5 a Btu/ft °F h)$_h$] + [1/(60.44 a Btu/ft °F h)$_c$]}

$\Downarrow$

a = 112.97 ft

To design the heat exchanger volume the width (w) has to be w = c = 2.63 in and the length (l) has to be a multiple of b. If l = 5 c = 5 * 2.63 in = 13.15 in = 1.0958 ft. Then:
a / l = 112.97 ft / 1.0958 ft = 103.1 ≈ 103 rows
103 row @ height b = .1 in gives a total height (h) of 10.3 in

∴ V= w *h * l = 2.63 in * 10.3 in * 13.15 in



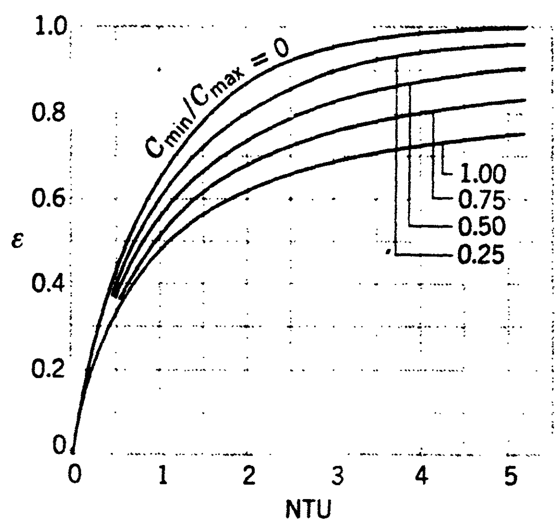


Figure D-1. Heat Exchanger Effectiveness for a Single-Pass, Cross-Flow Heat Exchanger with Both Fluids unmixed (Incropera and DeWitt, 1985).

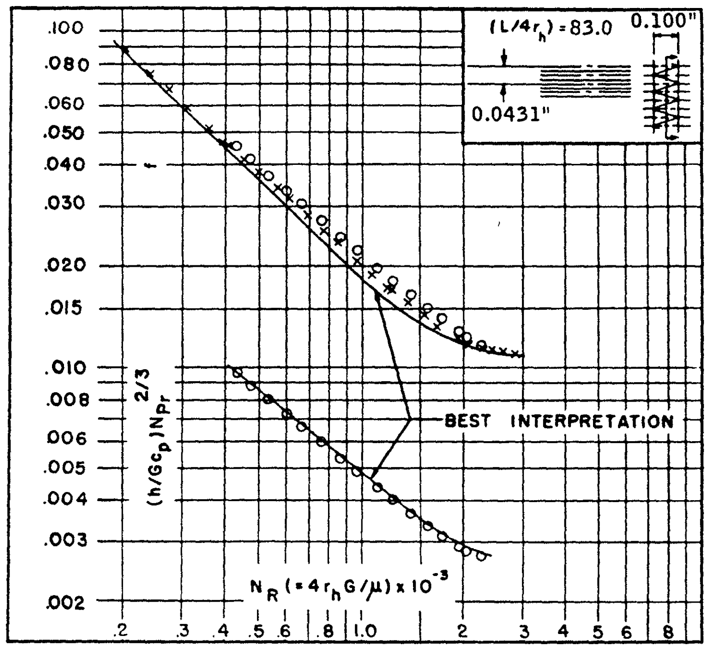


Plate Spacing, b = .1 in
Fin Length Flow Direction, c = 2.63 in
Flow Passage Hydraulic Diameter, $4r_h$ = .002643 ft
Fin Metal Thickness, t = .002 in; Stainless Steel
Total Heat Transfer Area/Volume Between Plates, β = 1332.45 ft^2/ft^3
Fin Area/Total Area, A_f/A = .837

Figure D-2. Properties of the Plain Plate-Fin Surface 46.45 T. (Kays and London 1964)

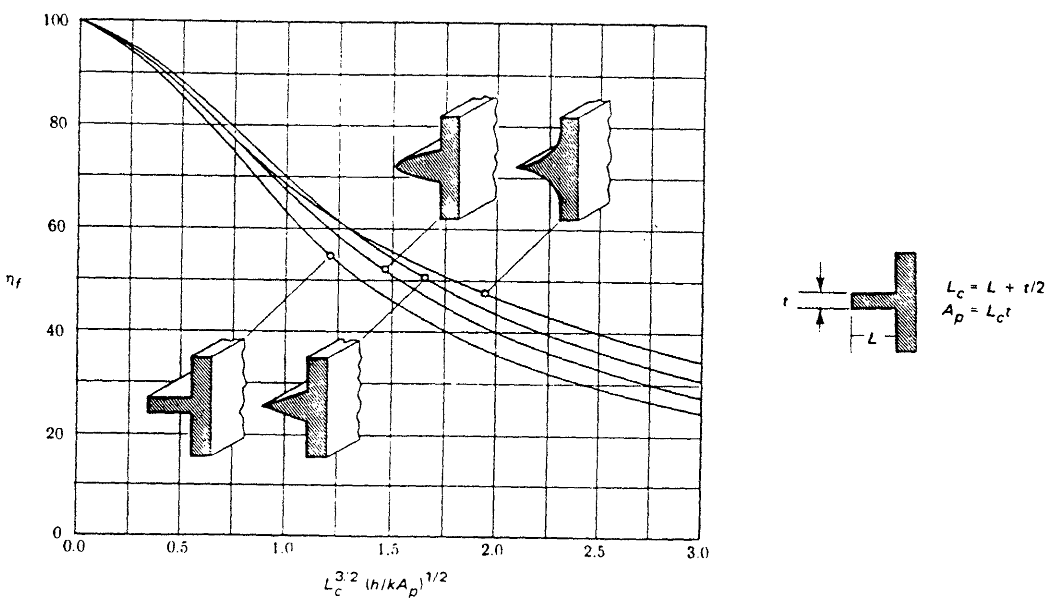
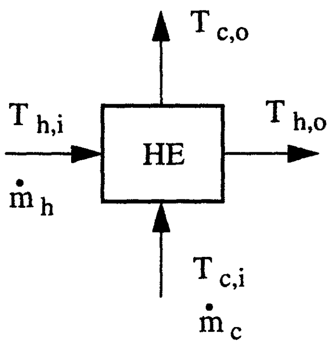


Figure D-3. Fin Efficiencies for Straight Fins. (Hodge 1990)

B) Calculations for the Regeneration Heat Exchanger Design

Hot-Side Mass Flow Rate ($\dot{m}$) of 1000 lb/h and Cold-Side Mass Flow Rate of 500 lb/h



Given:

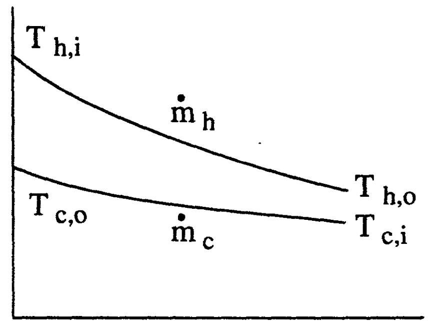
$T_{h,i} = 700\ ^{\circ}F$

$T_{c,i} = 80\ ^{\circ}F$

$\dot{m}_h = 1000$ lb/h

$\dot{m}_c = 500$ lb/h

Cross-Flow Heat Exchanger



*The cold-side air is the outside air @ T = 80 °F because it will be the worst temperature for this case instead of 100 °F

Assume that $T_{c,o} = 638\ ^{\circ}F$; then

$\overline{T}_c = [(630 - 80)/2] + 80 = 359\ ^{\circ}F$

Properties of air @ 359 °F (Hodge 1990):
$C_p = .25418$ Btu/lb °F; $\rho = .04846$ lb/ft^3; $\mu = 1.6926 \times 10^{-5}$ lb/ft s; Pr = .6858

The maximum heat transfer rate (q) between the two fluids is:
$q_{max} = (\dot{m}\, C_p)_{min} * (T_{h,i} - T_{c,i}) = (500 \text{ lb/h} * .25418 \text{ Btu/lb }^{\circ}F) * (700 - 80)^{\circ}F$
$= 78795.8$ Btu/h

Because air is the working fluid for both sides and the temperature difference between the two streams is not much, the C_p hot-side $\approx C_p$ cold-side.
Then:
$(\dot{m}\, C_p)_c = .5\ (\dot{m}\, C_p)_h$

$\therefore\ (\dot{m}\, C_p)_{min}/(\dot{m}\, C_p)_{max} = .5$

From Figure D-1 the maximum effectiveness (ξ) for a cross-flow heat exchanger with both fluids unmixed and $(\dot{m}\, C_p)_{min}/(\dot{m}\, C_p)_{max} = .5$ is .9.
Then:
$\xi = q/q_{max} \quad \Rightarrow \quad q = \xi * q_{max} = .9 * (78795.8 \text{ Btu/h}) = 70916.22$ Btu/h

$T_{h,o} = T_{h,i} - [q/(\dot{m}\, Cp)_h] = 700\ ^{\circ}F - [70916.2 \text{ Btu/h} / (1000 \text{ lb/h} * .25418 \text{ Btu/lb }^{\circ}F)]$
$= 421\ ^{\circ}F$
$T_{c,o} = [q/(\dot{m}\, Cp)_c] + T_{c,i} = [70916.2 \text{ Btu/h} / (500 \text{ lb/h} * .25418 \text{ Btu/lb }^{\circ}F)] + 80\ ^{\circ}F$
$= 638\ ^{\circ}F$
The assumption of $T_{c,o} = 678\ ^{\circ}F$ is good.

$\overline{T}_h = [(700 - 421)/2] + 421 = 560.5\ ^\circ F$

Properties of exhaust gases are assumed to be the air @ 561 °F (Hodge 1990):

$C_p = .24883$ Btu/lb °F; $\rho = .03882\ lb/ft^3$; $\mu = 1.9571 \times 10^{-5}$ lb/ft s; Pr = .68422

From Figure D-1 for a cross-flow heat exchanger with both fluids unmixed with $(\dot{m}\ C_p)_{min}/(\dot{m}\ C_p)_{max}$ =.5 and $\xi = .9$, gives a Number of Transfer Units NTU = 5.5.
Then:

$$NTU = (U_c\ A_c)/(\dot{m}\ C_p)_{min} \quad \Rightarrow \quad U\ A = NTU * (\dot{m}\ C_p)_{min}$$
$$= 5.5\ (500\ lb/h * .25418\ Btu/lb\ ^\circ F)$$
$$= 699\ Btu/h\ ^\circ F$$

- **Calculation of the exhaust gases velocity (v) [hot-side]**

If the cross-sectional area of the exhaust pipe is assumed to has a diameter of 3 in = .25 ft, then the area is:
$A = \pi\ [(d/2)]^2 = 3.1416 * [(.25\ ft/2)]^2 = .04909\ ft^2$
and with $\dot{m}$ = 1000 lb/h = .278 lb/s and ρ @ 561 °F = $.03882\ lb/ft^3$
then:

$$\dot{m} = \rho\ A\ v \quad \Rightarrow \quad v = \dot{m}/(A\ \rho) = (.278\ lb/s)/[(.04909\ ft^2)*(.03882\ lb/ft^3)]$$
$$= 145.88\ ft/s$$
$$\approx 145\ ft/s$$

- **Calculation of the air velocity (v) [cold-side]**

If the cross-sectional area of air is assumed to be .25 of the area of the desiccant wheel, and the diameter of the wheel is 1 ft, then the area is:
$A = .25\ \{\pi\ [(d/2)]^2\} = .25 * \{3.1416 * [(1\ ft/2)]^2\} = .19635\ ft^2$
and with $\dot{m}$ = 1000 lb/h = .1389 lb/s and ρ @ 359 °F = $.04846\ lb/ft^3$
then:

$$\dot{m} = \rho\ A\ v \quad \Rightarrow \quad v = \dot{m}/(A\ \rho) = (.1389\ lb/s)/[(.19635\ ft^2)*(.04846\ lb/ft^3)]$$
$$= 14.598\ ft/s$$
$$\approx 15\ ft/s$$

The heat exchanger uses surface 46.45 T for both sides.

Plain plate-fin surface 46.45 T is shown in Figure D-2.
Properties of surface 46.45 T:
$4r_h = .002643$ ft; $A_f/A = .837$; $\beta = 1332.45\ ft^2/ft^3$
t = .002 in, stainless steel AISI 304 @ 460 °F $\kappa = 18.376$ Btu/h ft °F
b = .1 in = .0083 ft; c = 2.63 in = .2192 ft

- **Calculation of $\bar{h}\eta_t A$ for the hot-side air**

$G = \rho\ v = .03882\ lb/ft^3 * 145\ ft/s = 5.6289\ lb/ft^2\ s$

$Re = (G\ 4r_h)/\mu = (5.6289\ lb/ft^2\ s * .002643\ ft)/1.9571 \times 10^{-5}\ lb/ft\ s = 760.2$

From Figure D-2 for $N_R = Re \times 10^{-3} = .7602 \approx .76$ gives $(\bar{h}\ Pr^{2/3})/(G\ C_p) = .006$

$\bar{h} = [(G\ C_p) * .006]/Pr^{2/3}$
$= [(5.6289\ lb/ft^2\ s * .24883\ Btu/lb\ °F) * .006]/(.68422)^{2/3}$
$= .01082\ Btu/ft^2\ °F\ s = 38.96\ Btu/ft^2\ °F\ h$

$Lc = .5 * b = .5 * (.1\ in) = .05\ in = .004167\ ft$
$Ap = t * Lc = (.002\ in) * (.05\ in) = .0001\ in^2 = 8.3 \times 10^{-6}\ ft^2$

$Lc^{3/2} * [\bar{h}/(\kappa\ Ap)]^{1/2}$
$= (.004167\ ft)^{3/2} * [(38.96\ Btu/ft^2\ °F\ h)/(18.376\ Btu/h\ ft\ °F * 8.3 \times 10^{-6}\ ft^2)]^{1/2}$
$= .136$

From Figure D-3 for $Lc^{3/2} * [\bar{h}/(\kappa\ Ap)]^{1/2} = .136 \approx .14$ gives $\eta_f = .98$

$\eta_t = 1 - [A_f/A * (1 - \eta_f)] = 1 - [.837 * (1 - .98)] = .9833$

If	fin length in cross-flow direction	a
	plate spacing	b = .1 in = .0083 ft
	fin length in flow direction	c = 2.63 in = .2192 ft
then		
	volume between plates	$V = a * b * c$
		$= a * .0083\ ft * .2192\ ft = .0018\ a\ ft^2$

β = Total heat transfer area/ volume between plates = A / V
$\Rightarrow\quad A = \beta\ V = 1332.45\ ft^2/ft^3 * .0018\ a\ ft^2 = 2.4\ a\ ft^2/ft$

$\bar{h}\eta_t A = 38.96\ Btu/ft^2\ °F\ h * .9833 * 2.4\ a\ ft^2/ft = 91.94\ a\ Btu/ft\ °F\ h$

- **Calculation of $\bar{h}\eta_t A$ for the cold-side air**

$G = \rho\ v = .04846\ lb/ft^3 * 15\ ft/s = .7269\ lb/ft^2\ s$

$Re = (G\ 4r_h)/\mu = (.7269\ lb/ft^2\ s * .002643\ ft)/1.6926 \times 10^{-5}\ lb/ft\ s = 113.5$

From Figure D-2 for $N_R = Re \times 10^{-3} = .1135 \approx .1$, assuming that the graph extends to .1 in a linear form, gives $(\bar{h}\ Pr^{2/3})/(G\ C_p) = .03$

$\bar{h} = [(G\ C_p) * .03]/Pr^{2/3}$
$= [(.7269\ lb/ft^2\ s * .25418\ Btu/lb\ °F) * .03]/(.6858)^{2/3}$
$= 7.1275 \times 10^{-3}\ Btu/ft^2\ °F\ s = 25.66\ Btu/ft^2\ °F\ h$

$Lc = .5 * b = .5 * (.1\ in) = .05\ in = .004167\ ft$
$Ap = t * Lc = (.002\ in) * (.05\ in) = .0001\ in^2 = 8.3 \times 10^{-6}\ ft^2$

$Lc^{3/2} * [\bar{h}/(\kappa\ Ap)]^{1/2}$
$= (.004167\ ft)^{3/2} * [(25.66\ Btu/ft^2\ °F\ h)/(18.376\ Btu/h\ ft\ °F * 8.3 \times 10^{-6}\ ft^2)]^{1/2}$
$= .1103$

From Figure D-3 for $Lc^{3/2} * [\bar{h}/(\kappa\ Ap)]^{1/2} = .1103 \approx .11$ gives $\eta_f = .98$

$\eta_t = 1 - [A_f/A * (1 - \eta_f)] = 1 - [.837 * (1 - .98)] = .9833$

If	fin length in cross-flow direction	a
	plate spacing	b = .1 in = .0083 ft
	fin length in flow direction	c = 2.63 in = .2192 ft
then		
	volume between plates	$V = a * b * c$
		$= a * .0083\ ft * .2192\ ft = .0018\ a\ ft^2$

β = Total heat transfer area/ volume between plates = A / V
$\Rightarrow \quad A = \beta V = 1332.45\ ft^2/ft^3 * .0018\ a\ ft^2 = 2.4\ a\ ft^2/ft$

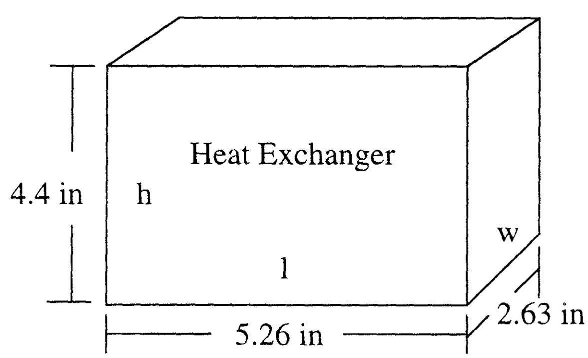
$\bar{h}\eta_t A = 25.66\ Btu/ft^2\ °F\ h * .9833 * 2.4\ a\ ft^2/ft = 60.56\ a\ Btu/ft\ °F\ h$

- **Calculation of the heat exchanger volume**

$U\ A = 1/\{[1/(\bar{h}\eta_t A)_h] + [1/(\bar{h}\eta_t A)_c]\}$
$\Downarrow$
$699\ Btu/h\ °F = 1/\{[1/(91.94\ a\ Btu/ft\ °F\ h)_h] + [1/(60.56\ a\ Btu/ft\ °F\ h)_c]\}$
$\Downarrow$
$a = 19.14\ ft$

To design the heat exchanger volume the width (w) has to be w = c = 2.63 in and the length (l) has to be a multiple of b. If l = 2 c = 2 * 2.63 in = 5.26 in = .4383 ft. Then:
$a / l = 19.14\ ft / .4383\ ft = 43.67 \approx 44$ rows
44 row @ height b = .1 in gives a total height (h) of 4.4 in

$\therefore\ V = w * h * l = 2.63\ in * 4.4\ in * 5.26\ in$



Appendix E

A) Calculations of the Conditions of the Air that Leaves the Evaporator (State 4)

Cooling Load = 13680 Btu/h
Mass Flow Rate ($\dot{m}$) = 2500 lb air/h
Enthalpy of Air After Leaving the Passenger Compartment (H5) = 27.3 Btu/lb air
Enthalpy of Air Leaving the Evaporator (H4)
Humidity Ratio ($\omega 4 = \omega 5$) = .008 lb H_2O/lb dry air

Cooling Load = $\dot{m}$ * (H5 - H4)
13680 Btu/h = 2500 lb air/h * (27.3 Btu/lb air - H4)
H4 = 27.3 Btu/lb air - [(13680 Btu/h)/(2500 lb air/h)]
= 21.8 Btu/lb air

For $\omega 4$= .008 lb H_2O/lb dry air and H4 = 21.8 Btu/lb air this state in the psychrometric chart has $T4_{\text{dry-bulb}}$ = 55 °F, $T4_{\text{wet-bulb}}$ = 52.5 °F, and $\Phi 4$ = 90%

The initial dry-bulb temperature for the ideal adiabatic case of state 4 is 90 °F

B) Calculation of the Efficiency of the Evaporator (ξ)

$$\xi = (T_{\text{in dry-bulb}} - T_{\text{out dry-bulb}}) / (T_{\text{in dry-bulb}} - T_{\text{in wet-bulb}})$$
$$= (90\ °F - 55\ °F) / (90\ °F - 52.5\ °F) = .93$$

The efficiency of the evaporator has to be 93%.

C) Calculation of the Flow Rate Capacity for the Water-Atomizer

The mass flow rate of water is 20 lb H_2O/h = .333 lb H_2O/min.
The density of water @ 100 °F = 62 lb/ft^3

The volume of water per hour is:
D= M / V $\Rightarrow$ V = M / D = (20 lb H_2O) / (62 lb/ft^3)
= .32258 ft^3 = 2.41 gal
$\therefore$ The mass flow rate of water for the atomizer is 2.41 gal/h = .00402 gal/min
$\approx$ 150 ml/min

Appendix F

A) Calculation of the Desiccant's Motor Torque

The weight of the desiccant wheel (8.64 lb) creates a force (F) of 8.64 lb
The diameter (d) of the desiccant wheel is 1 ft, therefore the radius (r) is .5 ft
Torque (T) is given by the equation T = F * r. Since the radius of the desiccant wheel goes from zero to .5 ft, the equation of torque is integrated in terms of r.

$$\text{Torque} = \int_0^r F * r\,dr = [F(r)^2]/2 \Big|_0^r = [F(r)^2]/2 - [F(0)]/2 = [F(r)^2]/2 = [8.64\text{ lb }(.5\text{ ft})^2]/2$$
$$= 1.08\text{ lb ft} = 207.36\text{ oz in} = 1.464\text{ N m}$$

B) Calculations for the Main Stream Fan

- **Calculation of the cfm**

The mass flow rate $\dot{m}$ is 2500 lb/h (41.67 lb/min or .694 lb/s)
The density of air (ρ) at 100 °F = .071 lb/ft^3

volume per minute:
$V = (41.67\text{ lb}) / (.071\text{ lb/ft}^3) = 586.9\text{ ft}^3$

$\therefore$ 586.9 ft^3/min $\approx$ 600 cfm

- **Calculation of the air velocity ($\vee$)**

The cross-sectional area of the air pipe has a diameter (d) of 5 in = .4167 ft.
The area is given by:

$A = \pi\,[(d/2)]^2 = 3.1416 * [(.4167\text{ ft}/2)]^2 = .1364\text{ ft}^2$

therefore:

$\dot{m} = \rho\,A\,\vee \qquad \Rightarrow \qquad \vee = \dot{m}/(A\,\rho) = (.694\text{ lb/s})/[(.1364\text{ ft}^2)*(.071\text{ lb/ft}^3)]$
$= 71.68$ ft/s
≈ 70 ft/s

- **Calculation of the friction factor (f)**

μ @ 100 °F = 1.285×10^{-5} lb/ft s

$Re = (\rho\,d\,\vee) / \mu = (.071\text{ lb/ft}^3 * .4167\text{ ft} * 70\text{ ft/s}) / 1.285 \times 10^{-5}\text{ lb/ft s}$
$= 161167.24$
$\approx 1.6 \times 10^5$

From Figure F-1, assuming a relative roughness of .05 for the air pipe, the friction factor (f) is .072.

- **Calculation of the Static Pressure**

The length of pipe line (l) is 15 ft. The gravitational acceleration (g) is 32.174 ft/s^2.

$$\Delta P = [\rho\, f\, l\, (\vee)^2]/(d\, 2\, g)$$
$$= [.071 \text{ lb/ft}^3 * .072 * 15 \text{ ft} * (70 \text{ ft/s})^2] / (.4167 \text{ ft} * 2 * 32.174 \text{ ft/s}^2)$$
$$= 14.0126 \text{ lb/ft}^2 = .0973 \text{ lb/in}^2 = 2.7 \text{ in } H_2O$$

If the minor losses (due by the filter and the changes in direction of the fluid) are considered as a 5% increase on the ΔP, then the new value is .1022 lb/in^2 = 2.835 in H_2O

If the inlet pressure of the fan is zero and the velocity pressure is not taken into account, the static pressure of the fan is ΔP. Therefore, the static pressure (SP) is equal to ΔP.

$$SP = \Delta P = 14.71 \text{ lb/ft}^2 = .1022 \text{ lb/in}^2 = 2.835 \text{ in } H_2O$$

The fan tables are for air at 70 °F. The density of air (ρ) at 70 °F = .075 lb/ft^3.
The estimated static pressure (ESP) is:

$$ESP = SP * (\rho @ 70\ °F / \rho @ 100\ °F) = 2.835 \text{ in } H_2O * [(.071 \text{ lb/ft}^3)/(.071 \text{ lb/ft}^3)]$$
$$= 2.995 \text{ in } H_2O \approx 3 \text{ in } H_2O$$

C) Calculations for the Regeneration Stream Fan

- **Calculation of the cfm**

The $\dot{m}$ is 500 lb/h (8.33 lb/min or .1389 lb/s), and ρ air @ 100 °F is .071 lb/ft^3

volume per minute:
$V = (8.33 \text{ lb}) / (.071 \text{ lb/ft}^3) = 117.32 \text{ ft}^3$

$\therefore$ 117.32 ft^3/min $\approx$ 120 cfm

- **Calculation of the air velocity** ($\vee$)

The cross-sectional area of the air pipe has a diameter (d) of 5 in = .4167 ft.
The area is given by:

$$A = \pi\, [(d/2)]^2 = 3.1416 * [(.4167 \text{ ft}/2)]^2 = .1364 \text{ ft}^2$$

therefore:

$$\dot{m} = \rho\, A \vee \quad \Rightarrow \quad \vee = \dot{m}/(A\, \rho) = (.1389 \text{ lb/s})/[(.1364 \text{ ft}^2)*(.071 \text{ lb/ft}^3)]$$
$$= 14.35 \text{ ft/s}$$
$$\approx 15 \text{ ft/s}$$

- **Calculation of the friction factor (f)**

μ @ 100 °F = 1.285 x 10^{-5} lb/ft s

$$Re = (\rho\ d\ v) / \mu = (.071\ lb/ft^3 * .4167\ ft * 15\ ft/s) / 1.285 \times 10^{-5}\ lb/ft\ s$$
$$= 34535.8$$
$$\approx 3.5 \times 10^4$$

From Figure F-1, assuming a relative roughness of .05 for the air pipe and an air flow in the transition zone, the friction factor (f) is .076.

- **Calculation of the Static Pressure**

The length of pipe line (l) is 10 ft. The gravitational acceleration (g) is 32.174 ft/s^2.

$$\Delta P = [\rho\ f\ l\ (v)^2]/(d\ 2\ g)$$
$$= [.071\ lb/ft^3 * .076 * 10\ ft * (15\ ft/s)^2] / (.4167\ ft * 2 * 32.174\ ft/s^2)$$
$$= .4528\ lb/ft^2 = .00314\ lb/in^2 = .0871\ in\ H_2O$$

If the minor losses (due by the filter and the changes in direction of the fluid) are considered as a 5% increase on the ΔP, then the new value is .0033 lb/in^2 = .0915 in H_2O

If the inlet pressure of the fan is zero and the velocity pressure is not taken into account, the static pressure of the fan is ΔP. Therefore, the static pressure (SP) is equal to ΔP.

$$SP = \Delta P = .4754\ lb/ft^2 = .0033\ lb/in^2 = .0915\ in\ H_2O$$

The fan tables are for air at 70 °F. The density of air (ρ) at 70 °F = .075 lb/ft^3
The estimated static pressure (ESP) is:

$$ESP = SP * (\rho\ @\ 70\ °F / \rho\ @\ 100\ °F) = .0915\ in\ H_2O * [(.071\ lb/ft^3)/(.071\ lb/ft^3)]$$
$$= .0966 in\ H_2O \approx .1\ in\ H_2O$$

D) Calculations for the Water Pump

- **Calculation of the gpm**

The $\dot{m}$ is 20 lb/h (.3333 lb/min or .0055 lb/s), and ρ water @ 100 °F = 62 lb/ft^3

volume per minute:
$$V = (.3333\ lb) / (62\ lb/ft^3) = .00538\ ft^3 \approx .0402\ gal/min = .0402\ gpm$$

$\therefore$ .0402 gpm = 152.17 ml/min $\approx$ 150 ml/min

- **Calculation of the air velocity** ($\vee$)

The cross-sectional area of the air pipe has a diameter (d) of .25 in = .02083 ft.
The area is given by:

$A = \pi\,[(d/2)]^2 = 3.1416 * [(.02083\ ft/2)]^2 = .00034\ ft^2$

therefore:

$\dot{m} = \rho\, A\, \vee \quad \Rightarrow \quad \vee = \dot{m}/(A\,\rho) = (.0055\ lb/s)/[(.00034\ ft^2)*(62\ lb/ft^3)]$
$= .2602\ ft/s$
$\approx .3\ ft/s$

- **Calculation of the friction factor** (f)

μ @ 100 °F = 4.58×10^{-4} lb/ft s

$Re = (\rho\, d\, \vee) / \mu = (62\ lb/ft^3 * .02083\ ft * .3\ ft/s) / 4.58 \times 10^{-4}\ lb/ft\ s$
$= 845.93$

From Figure F-1, if the air flow is on the laminar zone, the friction factor (f) is given by:

$f = 64 / Re = 64 / 845.93 = .0756$

- **Calculation of the Head**

The length of pipe line (l) is 10 ft. The gravitational acceleration (g) is 32.174 ft/s^2.

$\Delta P = [\rho\, f\, l\, (\vee)^2]/(d\, 2\, g)$
$= [62\ lb/ft^3 * .0756 * 10\ ft * (.3\ ft/s)^2] / (.02083\ ft * 2 * 32.174\ ft/s^2)$
$= 3.1472\ lb/ft^2 = .02186\ lb/in^2 = .6056$ in H_2O

If the minor losses (due by changes in direction of the fluid) are considered as a 5% increase on the ΔP, then the new value is 3.3046 lb/ft^2 = .02295 lb/in^2

The head is:

Head = $\Delta P / \rho = (3.3046\ lb/ft^2) / (62\ lb/ft^3) = .0533\ ft \approx .1\ ft$

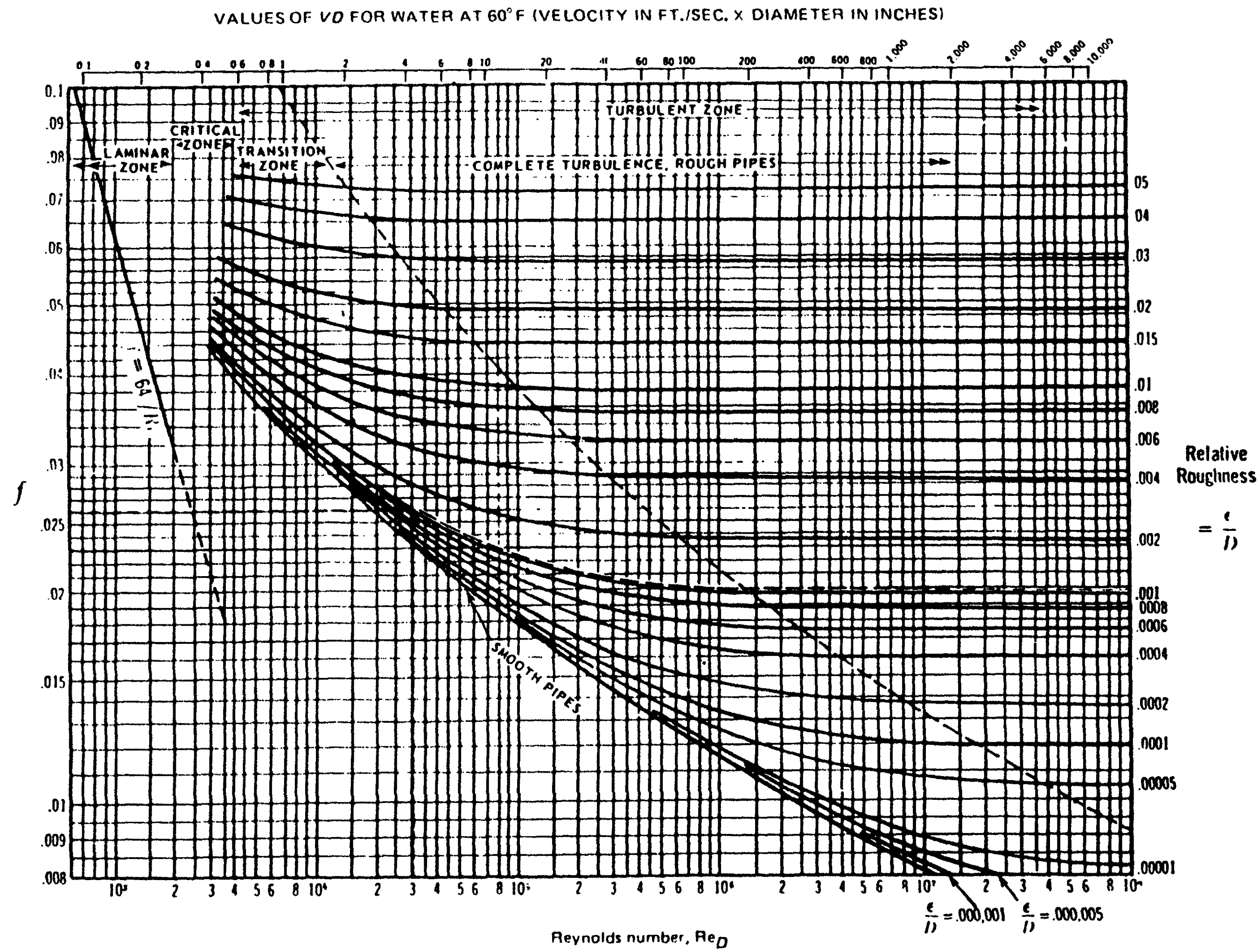


Figure F-1. Heat Loss Representation: Moody Diagram. (Hodge 1990)

Appendix G

A) Calculation of the Electrical Power

The electrical power is the sum of the electrical power of the desiccant motor, the fans, and the pump.

Electrical Power = 240 W+ 240 W+ 120 W+ 150 W+ 10 W + 10 W= 765W = 1.02 hp
$\approx$ 1 hp = 2545 Btu/h

B) Calculation of the Coefficient of Performance (COP)

COP = Cooling Capacity / (Electric Power + Heat Consumption)
= (13680 Btu/h) / [(2545 Btu/h) + (70920 Btu/h)] = .186 $\approx$ .2

Heat Consumption is the regeneration heat supplied by the waste heat of the exhaust gasses of combustion.

C) Calculation of the Electric Power COP

COP = Cooling Capacity / Electric Power
= (13680 Btu/h) / (2545 Btu/h) = 5.375 $\approx$ 5.4

D) Calculations for the States of the Thermodynamical Process

The desired output of the thermodynamical process (state 5) is air at 77 °F, Φ = 40%, ω = .008 lb H_2O/lb air, and H = 27.3 Btu/lb dry air; with a water consumption of 20 lb H_2O/h and cooling capacity of 13680 Btu/h.

To satisfy the water consumption (WC), the mass flow rate of air (m) has to be:

$$WC = \dot{m} * \omega \quad \Rightarrow \quad \dot{m} = WC / \omega$$
$$= (20 \text{ lb } H_2O/h) / (.008 \text{ lb } H_2O/\text{lb dry air})$$
$$= 2500 \text{ lb air/h}$$

The conditions of state 5 were set, therefore the conditions of state 4 can be found. To satisfy the cooling capacity, the enthalpy (H) of state 4 has to be:

$$\text{Cooling Load} = \dot{m} * (H5 - H4)$$
$$13680 \text{ Btu/h} = 2500 \text{ lb air/h} * (27.3 \text{ Btu/lb air} - H4)$$
$$H4 = 27.3 \text{ Btu/lb air} - [(13680 \text{ Btu/h})/(2500 \text{ lb air/h})]$$
$$= 21.8 \text{ Btu/lb air}$$

From state 4 to state 5 the system follows a heating process at constant humidity; therefore $\omega 4 = \omega 5$ = .008 lb H_2O/lb dry air. Since only two properties are needed to define the state of the system in the psychrometric chart, the state is found with $\omega 4$ and H4. For state 4: $T4_{\text{dry-bulb}}$ = 55 °F, $T4_{\text{wet-bulb}}$ = 52.5 °F, and $\Phi 4$ = 90%

If the system follows an ideal adiabatic evaporation process (constant enthalpy line) and if the dehumidification process is assumed to dry the air until it has a ω = .0005 lb H_2O/lb air; then, from the psychrometric chart, the temperature at that point is 87.5 °F.

But the system does not follows an ideal adiabatic process because the water added at the evaporator does not have the temperature of the air exiting the evaporator (55 °F). Water is added at ambient temperature (100 °F). Some of the cooling capability of the evaporation process is used to cool the water to 55 °F. The change in temperature of air, due to the cooling of water, is calculated as follows:

Change in Temperature of Water = 100 °F - 55 °F = 45 °F

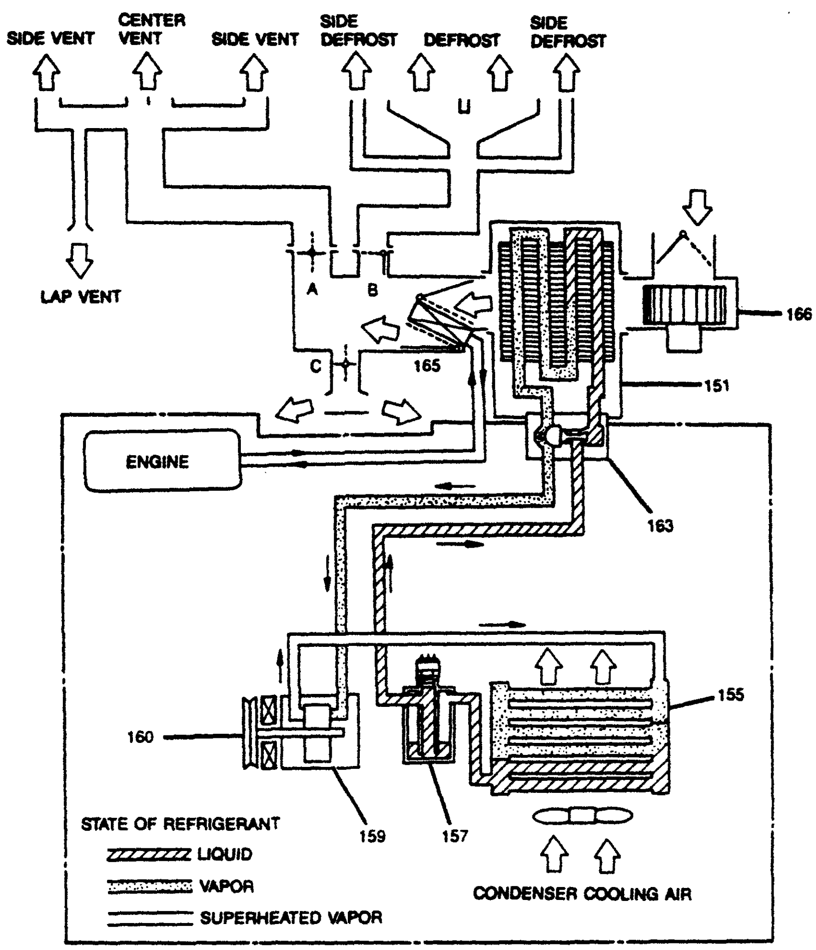
If for 1 Btu, the change in temperature of 1 lb of water is 1 °F; therefore:
45 °F * [1 Btu/(1 lb H_2O * 1 °F)] * (20 lb H_2O) = 900 Btu

If for 1 Btu, the change in temperature of 1 lb of air is 4.7 °F; therefore:
{900 Btu * [(1 lb air * 4.7 °F)/1 Btu]} / 2500 lb air = 1.69 °F $\approx$ 2 °F

Taking into account the 2 °F needed to cool the water, air has to enter the evaporator with a temperature of 85.5 °F. But the temperature has to be even lower to take into account that the process is an ordinary evaporative cooling process. The ordinary process follows an indeterminable path. To fix state 3, the exit temperature of the second main stream heat exchanger (83.25 °F) is taken as the temperature of state 3. With ω = .0005 lb H_2O/lb air, state3 is determined.

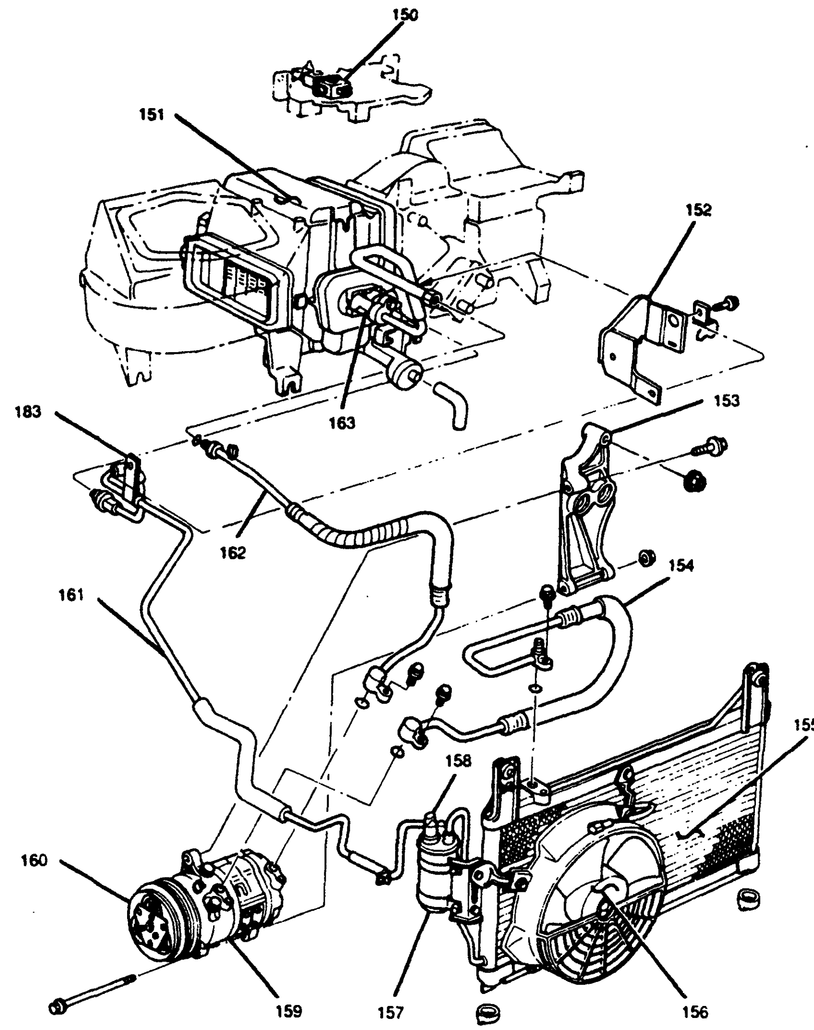
State 2 is determined from state 1 (state 1 is the design conditions, therefore it is know). The system follows a constant enthalpy dehumidification process. As assumed before, if the dehumidification process dry the air until it has a ω = .0005 lb H_2O/lb air; then, from the psychrometric chart, the temperature at state 2 is 115 °F.

Appendix H



A MODE (VENT) CONTROL DOOR
B MODE (DEFROST) CONTROL DOOR
C MODE (HEAT) CONTROL DOOR
151 EVAPORATOR
155 CONDENSER
157 RECEIVER/DRYER
159 COMPRESSOR
160 A/C COMPRESSOR CLUTCH
163 EXPANSION VALVE
165 HEATER CORE
166 BLOWER MOTOR

Figure H-1. Schematic of the Vapor-Compressor Air-Conditioning System.



150 A/C SWITCH
151 EVAPORATOR CASE
152 REFRIGERANT LINE HOLD-DOWN BRACKET
153 COMPRESSOR MOUNTING BRACKET
154 COMPRESSOR DISCHARGE PIPE
155 CONDENSER
156 CONDENSER FAN
157 RECEIVER/DRYER
158 TRIPLE SWITCH
159 COMPRESSOR
160 A/C COMPRESSOR CLUTCH
161 RECEIVER/DRYER-TO-EVAPORATOR PIPE
162 COMPRESSOR SUCTION PIPE
163 EXPANSION VALVE
183 HOLD-DOWN BRACKET

Figure H-2. Drawing of the Components of the Vapor-Compressor Air-Conditioning System.

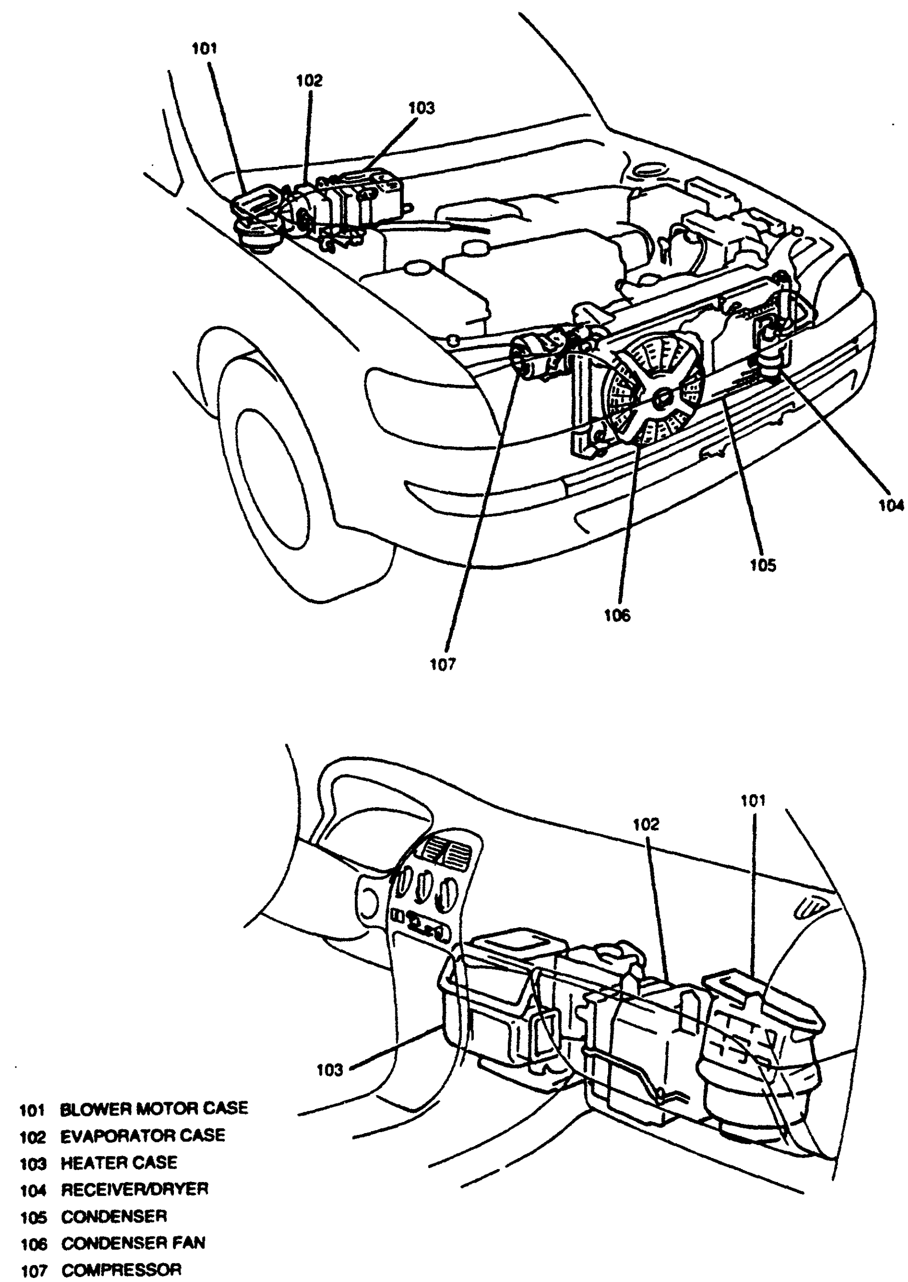


Figure H-3. Placement of the Components of the Vapor-Compressor Air-Conditioning System Inside the Automobile.

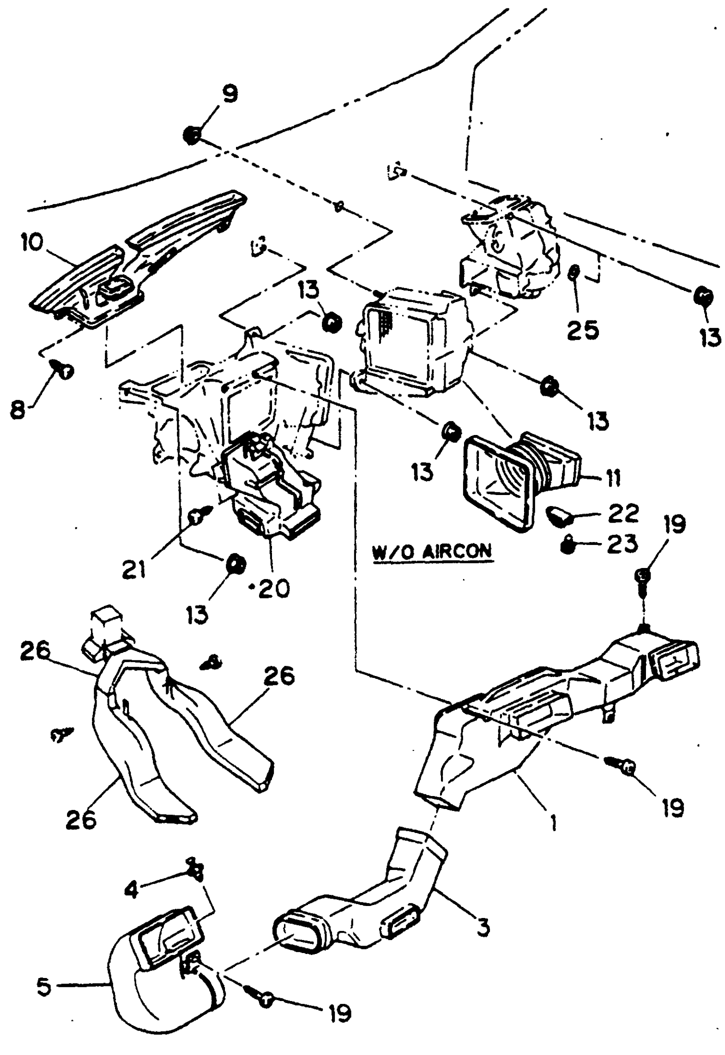


Figure H-4. **Air Distribution System of the Vapor-Compressor Air-Conditioning System.**

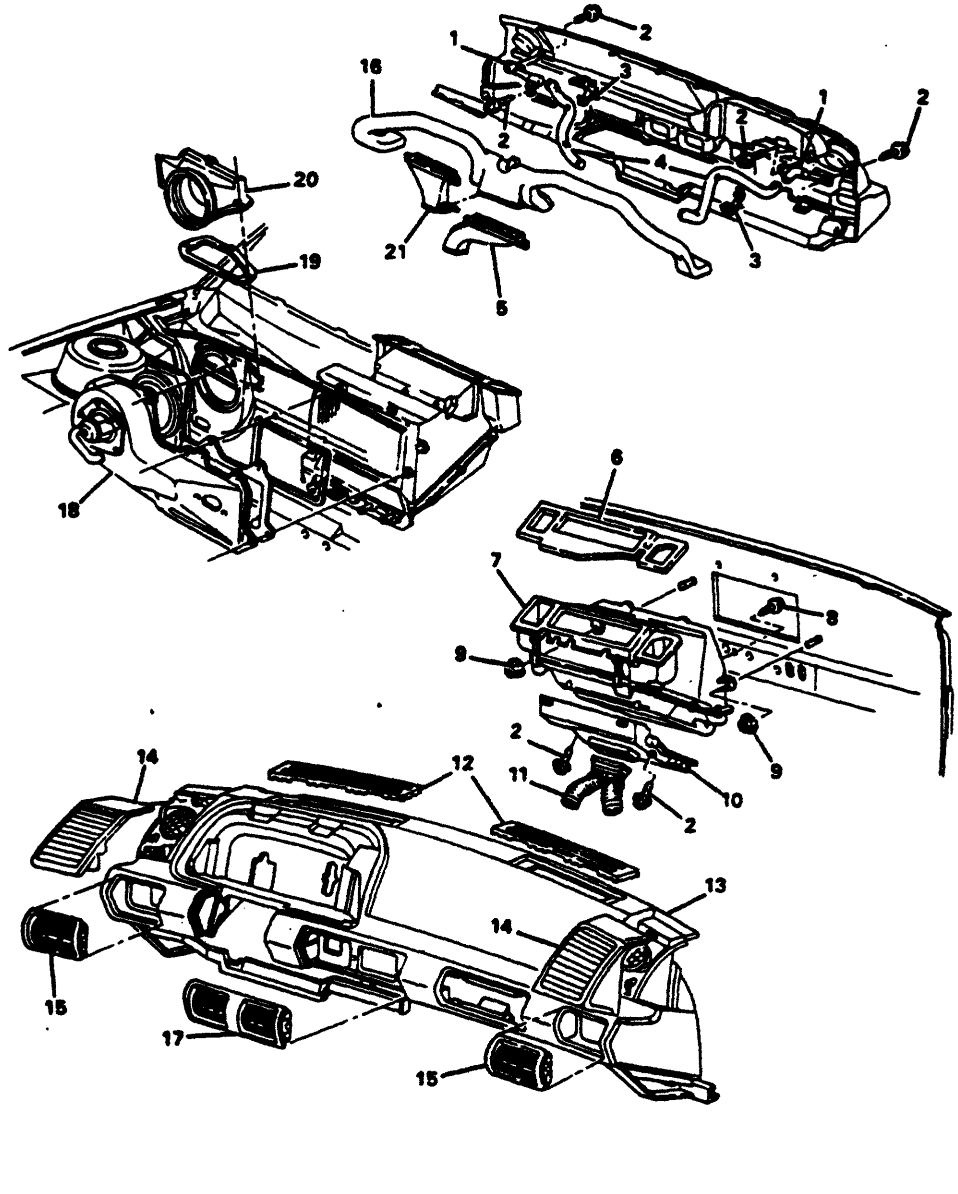
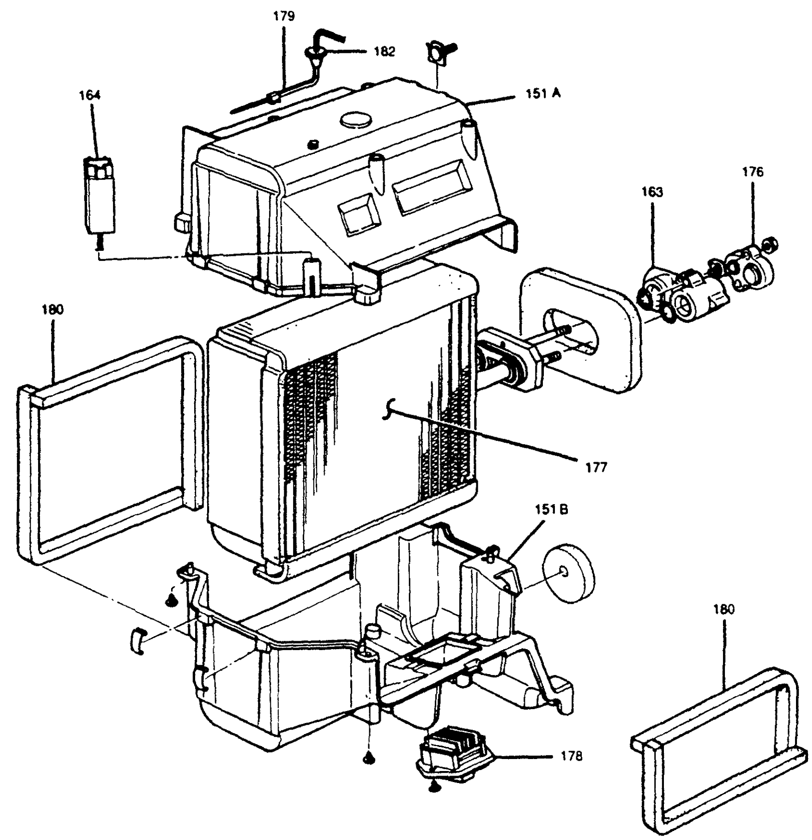
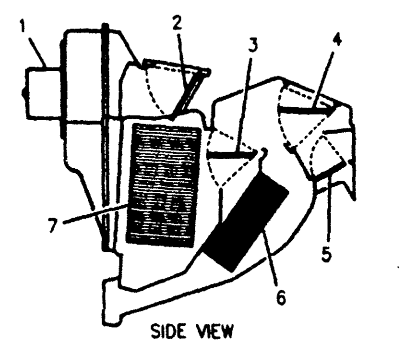


Figure H-4. Air Distribution System of the Vapor-Compressor Air-Conditioning System. (Continuation)



151A EVAPORATOR CASE (UPPER)
151B EVAPORATOR CASE (LOWER)
163 EXPANSION VALVE
164 A/C THERMOSTATIC SWITCH
176 EVAPORATOR PIPE FITTING
177 EVAPORATOR CORE
178 BLOWER MOTOR RESISTOR
179 A/C THERMOSTATIC SWITCH SENSOR
180 EVAPORATOR CASE LINING
182 GROMMET

Figure H-5. Evaporator Assembly.



1 BLOWER
2 AIR INLET VALVE
3 TEMPERATURE VALVE
4 MODE VALVE
5 DEFROST VALVE
6 HEATER CORE
7 EVAPORATOR CORE

Figure H-6. Cross Section of the Heater Module.

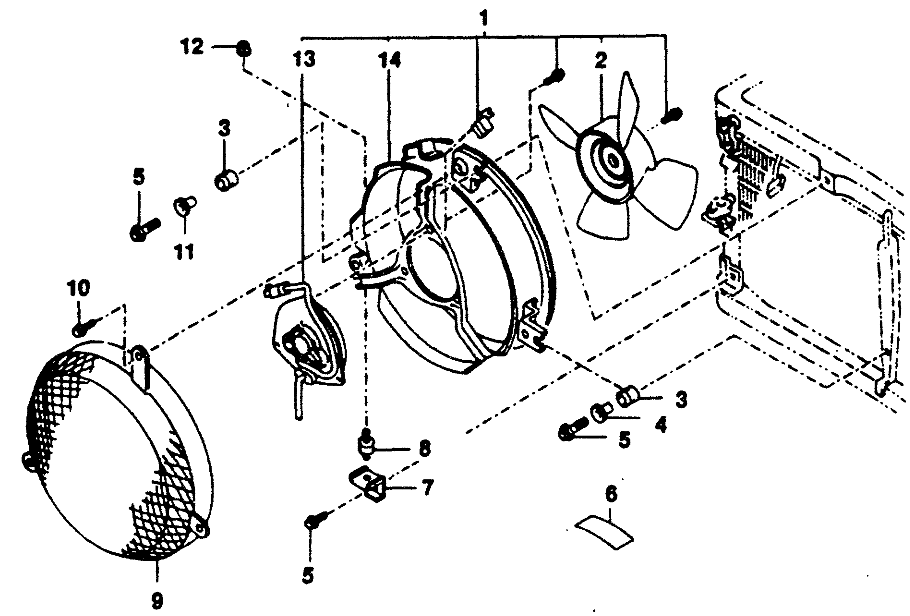
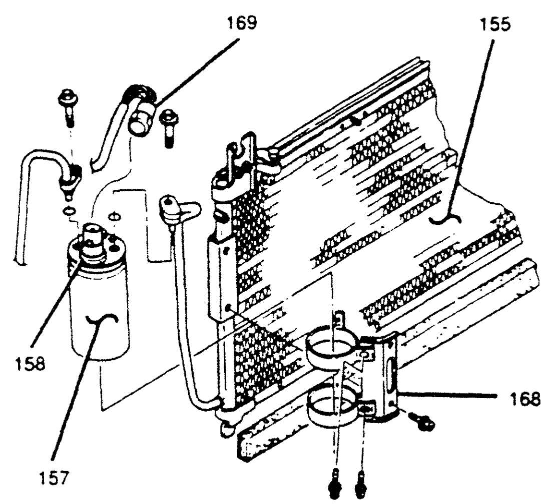


Figure H-7. Condenser Fan Assembly.



155 CONDENSER
157 RECEIVER/DRYER
158 TRIPLE SWITCH
168 RECEIVER/DRYER MOUNTING BRACKET
169 TRIPLE SWITCH ELECTRICAL CONNECTOR

Figure H-8. Condenser Assembly.

Appendix I

T H O M A S   R E G I S T E R   C A T A L O G   F I L E

DELAVAN-DELTA INC. AGRICULTURAL & INDUSTRIAL PRODUCTS OPERATION
20 DELAVAN DRIVE • LEXINGTON, TN 38351 • FAX: 901-968-5085
800 892 6943

Delavan
Industrial Spray Nozzles

Aspirflo Series

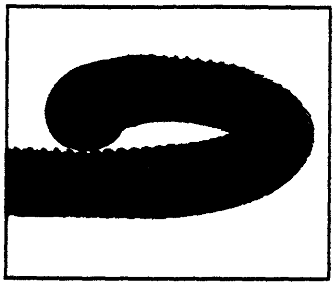
AI, TS, DF, DFA

U.S. Patent No.
5,058,890

Applications:

Aspirflo nozzle "breathes" free air. Ideally suited for cooling applications. Can eliminate air compressor cost in air atomizing systems. Cooling efficiency increases up to 25% over standard pressure nozzles. Can operate on pressures as low as 1 PSIG.

Water-Atomizer: Aspirflo Nozzle (DELAVAN-DELTA INC.)

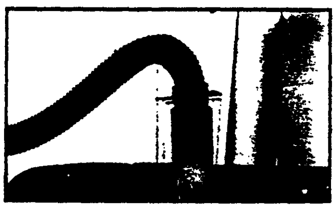


FLARELOCK HOSE: Cars and small trucks
Flared end on each length to permit connection of two or more lengths of same size hose.

FLT250	2½"x 11 ft.	Compact Car
FLT300	3"x 11 ft.	Passenger Car
FLT350	3½"x11 ft.	Small truck, van
FLT400	4"x 11 ft.	Gasoline truck

UNIHOSE: Hose and tailpipe adapter joined in ONE CONTINUOUS PIECE.

UNH2500	2½"x 11 ft.	Compact Car
UNH3000	3"x 11 ft.	Passenger Car
UNH4000	4"x 11 ft.	Gasoline Truck



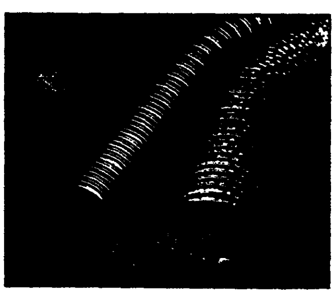
SUPERFLEX HOSE: Accordion-like stretch. Compressed rubber hose to prevent kinking at 180 degree bend over diesel stack.

40-20SF	4"x 20 ft.	Gasoline Trucks
50-12SF	5"x 12 ft.	Diesel Trucks

ACT HOSE: Diesel and Gasoline Trucks.
Regular and **WIRE-INSERTED** Hose for use with overhead exhaust systems.

ACT400	4"x 11 ft.	Gasoline Trucks
* **ACT400W11**	4"x 11 ft.	(inserted w/11' WIRE)
ACT 40-20	4"x 20 ft.	Gasoline Truck
ACT500	5"x 11 ft.	Diesel Trucks
* **ACT500W11**	5"x 11 ft.	(inserted w/11' WIRE)

* Also available w/30" wire (Use W30 suffix)



STAINLESS STEEL HOSE: Dynomometers.
Lightweight and flexible. Withstands 1650°F.

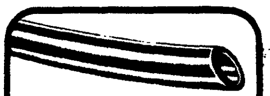
SS4000	4"x 10 ft.	Longer lengths available
SS5000	5"x 10 ft.	Longer lengths available

FABRIC/WIRE HOSE: Lightweight 600°F. hose for use with overhead exhaust systems.

401	4"x 25 ft.	Gasoline Trucks
501	5"x 25 ft.	Diesel Trucks

Air Stream Piping: 50-12SF 5" x 12 ft Diesel Trucks (CRUSHPROOF)

Plastic Tubing



Temp. Range: −20°F to 140°F
Color: Clear

Masterkleer Vinyl Plastic Tubing

Lightweight, flexible tubing is excellent for industrial food handling and laboratory applications . . . ideal for medical and quality control laboratories.

Nontoxic tubing is **FDA-approved** for food, beverages and water lines. Tubing has good chemical resistance and handles oils, mild acids, alkalies, and solvents.

Sold in full coils only, except where indicated with a price.

Note: Not recommended for esters, ketones, and aromatics.

ID In.	OD In.	Wall In.	Max. Pressure @ 70°F	Full Coil Lgth.	No.	NET FT. Cut Lgth.	NET FT. Full Coil
1/16	1/8	1/32	100	100	5233K11	—	$0.05
1/8	7/32	3/64	85	100	5233K12	—	.12
3/16	5/16	1/16	55	100	5233K13	—	.13
3/16	3/8	3/32	70	100	5233K14	$0.26	.20
3/16	7/16	1/8	80	100	5233K15	.28	.23
1/4	3/8	1/16	55	100	5233K16	.17	.13
1/4	7/16	3/32	60	100	5233K17	.23	.18
1/4	1/2	1/8	70	100	5233K18	.43	.33
5/16	7/16	1/16	50	100	5233K19	.25	.19
5/16	1/2	3/32	60	100	5233K21	.33	.26
5/16	9/16	1/8	70	100	5233K22	.49	.37
3/8	1/2	1/16	40	100	5233K23	.24	.19
3/8	9/16	3/32	50	100	5233K24	.36	.28
3/8	5/8	1/8	65	100	5233K25	.53	.41
1/2	5/8	1/16	30	100	5233K26	.35	.28
[illegible]	[illegible]	[illegible]	40	100	5233K27	$0.46	$0.37
[illegible]	[illegible]	[illegible]	45	100	5233K28	.62	.49
[illegible]	[illegible]	[illegible]	40	100	5233K29	.79	.62
[illegible]	[illegible]	[illegible]	35	50	5233K31	.93	.72
[illegible]	[illegible]	[illegible]	26	50	5233K32	1.10	.88
[illegible]	[illegible]	[illegible]	45	50	5233K33	1.49	1.21
[illegible]	[illegible]	[illegible]	40	50	5233K34	2.16	1.72
[illegible]	[illegible]	[illegible]	45	50	5233K35	2.42	1.99
[illegible]	[illegible]	[illegible]	35	50	5233K36	2.51	2.02
[illegible]	2	[illegible]	40	50	5233K37	2.90	2.38
[illegible]	[illegible]	[illegible]	40	50	5233K38	5.70	4.28
2	[illegible]	[illegible]	35	50	5233K39	4.25	3.58
[illegible]	[illegible]	[illegible]	35	50	5233K41	4.38	3.77
[illegible]	3	[illegible]	30	50	5233K42	4.74	4.14
3	[illegible]	[illegible]	20	50	5233K43	5.35	4.76

Water Piping: Masterkleer Vinyl Plastic Tubing No.5233K17 (McMaster-Carr)

Filtrete Media Cleans Better Without Restricting Air Flow.

Filtrete media offers an optimum balance between cleaner air and better filter designs. It offers an unequalled combination of low pressure drop and high efficiency in trapping dust, pollen, road salt, viruses, bacteria and other particles.

Because of its open construction, Filtrete media resists clogging. That means high air permeability and longer service life. In medical applications, for example, Filtrete media allows you to design in unrestricted air flow to the patient for increased safety and comfort. And when used to protect computer disk drives from damaging particles, Filtrete media provides a faster time to clean-up.

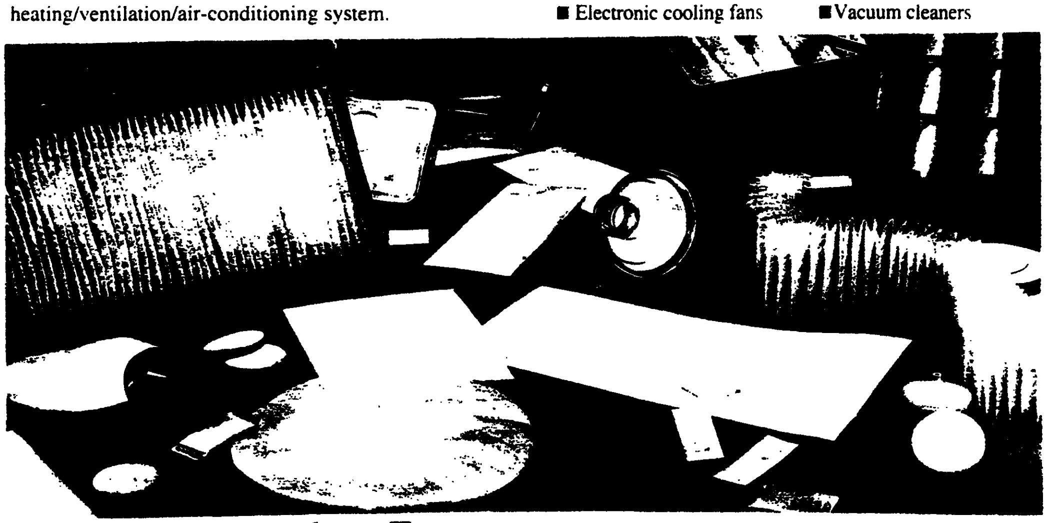
Since it maintains low pressure drop and requires no electrical power, Filtrete media is the ideal choice for vehicle ventilation filters. It effectively cleans interior air with low pressure drop to the vehicle's heating/ventilation/air-conditioning system.

Filtrete media is made of polypropylene, which resists moisture and degradation. You can be confident that its performance will remain constant within a wide range of temperature and humidity changes.

Use Filtrete media when particle capture is critical and space is at a premium. Filtrete media gives you more design flexibility because, in most applications, it takes less media to do the job (often pleating or layering is not necessary, as it is with competitive filter media). In addition, there are several basis weights from which to choose, depending on your application and design.

Put Filtrete Media To Work For You.

- Anesthetic gas equipment
- Incubators
- Suction equipment
- Laser surgery applications
- Disk drive recirculation
- Electronic cooling fans
- Copying machines
- Respirators
- Room air cleaners
- Air conditioners
- Vehicle ventilation systems
- Vacuum cleaners



We Use 3M Technology To Make A Better Filter.

A prime example of electrostatic enhancement in action is our Filtrete™ Air Filter Media. Each and every nonconductive fiber contains permanent, electrostatic charges that enhance particle capture efficiency. With its patented electret construction, Filtrete media captures particles throughout the media, rather than mostly on the media surface.

The result is efficiencies up to 99.9999%, depending on the application — with demonstrated high efficiency against difficult-to-capture, submicron particles. It is best designed for use when high efficiency is needed with low pressure drop.

Because we've created electrostatic fibers that are so efficient in capturing particles, we're also able to give Filtrete media a more open construction. Air molecules encounter fewer fibers as they pass through the filter, resulting in extraordinarily low pressure drops and greater air flow.

The low pressure drop is maintained during use because Filtrete media is a three-dimensional, depth-loading filter rather than a surface-loading filter.

Although the use of electrostatics is not uncommon in filtration, Filtrete media is superior for three reasons: 1) it provides the highest charge density of any electrostatically charged material; 2) these permanently charged rectangular fibers provide more loading surface than do round fibers; and 3) the charged media is combined with depth filtration to provide a higher loading capacity.

Air Filter: Filtrete Air Filter Media (3M FILTRATION PRODUCTS)

Bibliography

[1] *The ABC's of Air Conditioning*. Ford Climate Control Division.

[2] *1993 ASHRAE Handbook, Fundamentals*. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, GA, 1993, chaps. 6 and 19.

[3] *1988 ASHRAE Handbook, Equipment*. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, GA, 1988, chaps. 3 and 4.

[4] Baumeister, T.: *Mark's Standard Handbook for Mechanical Engineers*, Eighth Edition. McGraw-Hill, New York, 1978, chap. 11 pp.3-18.

[5] Brundrett, G. W.: *Handbook of Dehumidification Technology*. Butterworths, London, England, 1987, chap. 2.

[6] Držaj, B.; Hočevar, S.; and Pejovnik, S.: *Zeolites: Synthesis, Structure, Technology and Application*. Elsevier Science Publishing Company Inc., New York, 1985. pp.503-512 and 547-563.

[7] Fischer, S. K.: *National Energy Impacts of CFC Alternatives in Heating, Air-Conditioning and Refrigerating Equipment and Foam Insulation*. CONP-900203 --4 DE90 005450, United State Department of Energy, February 1990.

[8] Fischer, S. K.: *Energy and Global Warming Impacts of CFC Alternative Technologies*. AFEAS/DOE report, Alternative Fluorocarbon Environmental Acceptability Study and the United State Department of Energy, December 1990.

[9] Fourmery, E. A.; and Heggs, P. J.: *Heat Exchange Engineering*, volume 2 *Compact Heat Exchangers: Techniques and Size Reduction*. Ellis Horwood Limited, London, England, 1991, chap. 1.

[10] Green, A. E. S.: *Greenhouse Mitigation*. The American Society of Mechanical Engineers, 1989, pp.49-55.

[11] Hall, C. W.: *Dictionary of Drying*. Marcel Dekker Inc., New York, 1979.

[12] *Handbook of HVAC Design*. McGraw-Hill, New York, 1990, chaps. 34, 39, and 47.

[13] Hodge, B. K.: *Analysis and Design of Energy Systems*, Second Edition. Prentice-Hall Inc., Englewood Cliffs, New Jersey, 1990, chaps. 2,3, and 4.

[14] Incropera, F. P.; and DeWitt, D.P.: *Fundamentals of Heat and Mass Transfer*, Second Edition. John Wiley & Sons Inc., New York, 1985, chaps. 3 and 11.

[15] James, R. W.: *Desiccants and Humectants*. Noyes Data Corporation, Park Ridge, New Jersey, 1973, pp.167-220.

[16] Jones, J. B.; and Hawkins, G. A.: *Engineering Thermodynamics an Introductory Textbook*, Second Edition. John Wiley & Sons Inc., New York, 1986, chap. 11.

[17] Kays W. M.; and London, A. L.: *Compact Heat Exchangers*, Second Edition. McGraw-Hill, New York, 1964.

[18] Langley, B. C.: *Heating, Ventilating, Air Conditioning, and Refrigeration*. Prentice-Hall Inc., Englewood Cliffs, New Jersey, 1990, chap. 22.

[19] *McGraw-Hill Concise Encyclopedia of Science & Technology*. McGraw-Hill, New York, 1989, pp.224-231, 277, and 657-658.

[20] Mei, V. C.; Chen, F. C.; and Kyle, D. M.: *Alternative Non-CFC Mobile Air Conditioning*. ORNL/CON-335, Oak Ridge National Laboratory, Oak Ridge, TN, September 1992.

[21] Mei, V. C.; Chen, F. C.; and Sullivan, R. A.: *Analysis of Non-CFC Automotive Air Conditioning*. CONP-911256 --1 DE92 000738, United State Department of Energy, December 1992.

[22] Palen, J. W.: *Heat Exchanger Sourcebook*. Hemisphere Publishing Corporation, New York, 1986, chap. 2.

[23] Ruth, D. W.: *Simulation of Modelling of Automobile Comfort Cooling Requirements*. ASHRAE Journals, 53-55, May 1975.

[24] Szostak, R.: *Handbook of Molecular Sieves*. Van Nostrand Reinhold, New York, 1992.

[25] Thurlow, G.: *Technological Responses to the Greenhouse Effect*, Report Number 23. The Watt Committee on Energy, 1990, pp.7-15.

[26] Townsend, R. P.: *The Properties and Applications of Zeolites*. The Chemical Society, London, England, 1980, pp.391-421.

[27] Watt, J. R.: *Evaporative Air Conditioning Handbook*, Second Edition. Chapman and Hall, New York, 1986, chaps. 2, 3, 8, and 23.

[28] Williams-Gardner, A.: *Industrial Drying*. George Godwin Ltd., London, England, 1980, chap. 14.

[29] *UOP Molecular Sieves*. UOP Molecular Sieve Adsorbents.

4958-19