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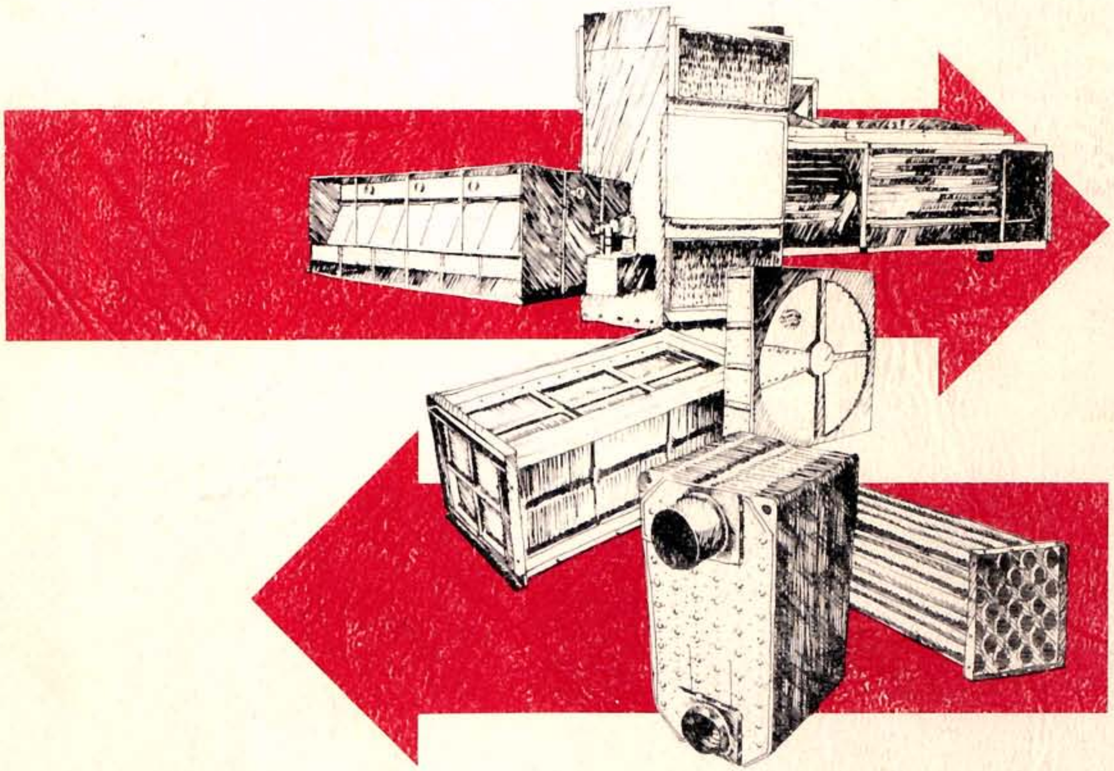
SMACNA Inc. 7/78 15d AIR TREATMENT EQUIPMENT / ENERGY RECOVERY

ENERGY RECOVERY EQUIPMENT AND SYSTEMS



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AIR-TO-AIR

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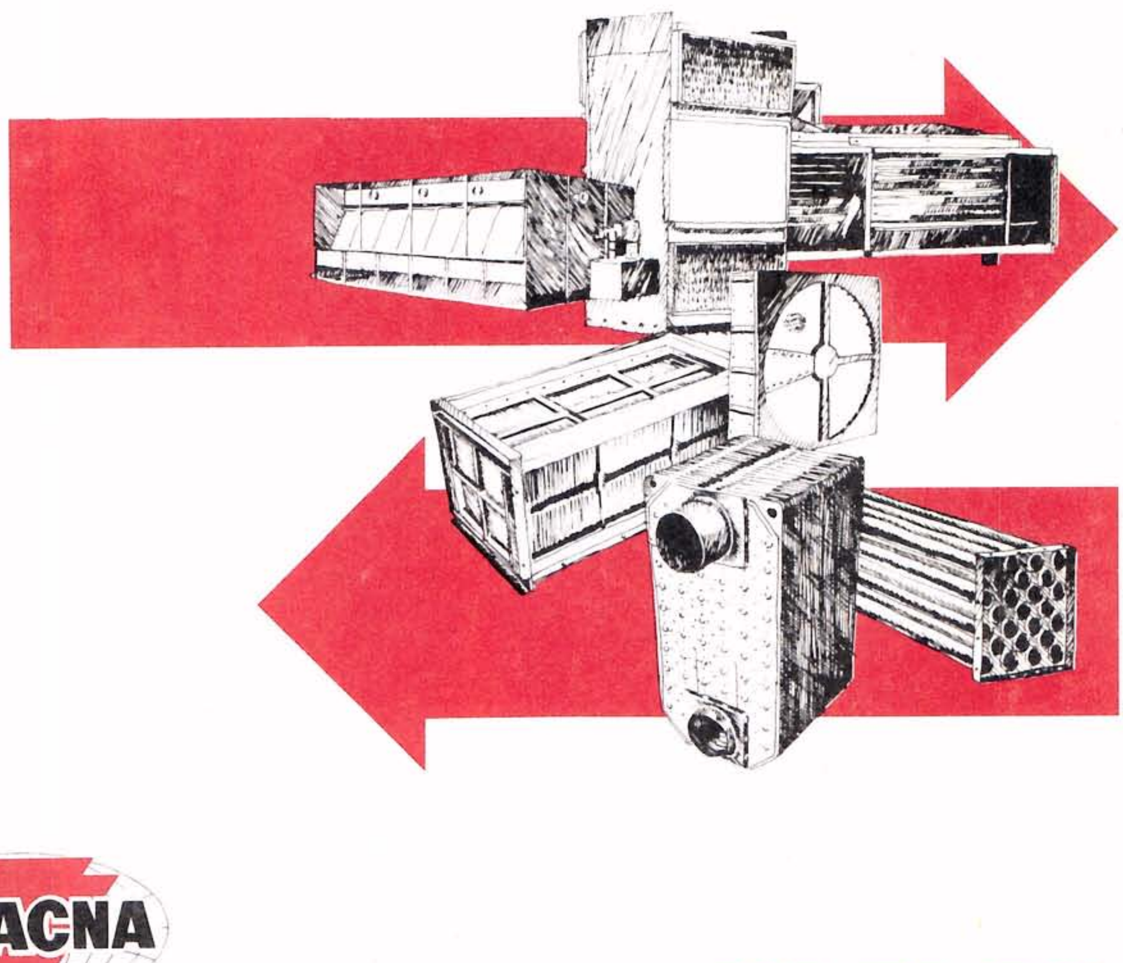

SMACNA

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AIR-TO-AIR

ENERGY RECOVERY EQUIPMENT AND SYSTEMS
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SHEET METAL AND AIR CONDITIONING CONTRACTORS'
NATIONAL ASSOCIATION, INC.
8224 Old Courthouse Rd.
Tysons Corner
Vienna, Virginia 22180

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SMACNA Energy Recovery Committee

CONTRACTORS

G.E. Daniel, Chairman
Daniel & Jones Sheet Metal
Pinellas Park, Florida

William Barnhizer
The Waldinger Corporation
Des Moines, Iowa

John J. Hucker, P.E.
Peabody & Wind Engineering Co.
Philadelphia, Pennsylvania

James A. McVean
F-M Sheet Metal Fabricators
Fort Worth, Texas

Lester W. Lang
Lackland Sheet Metal Company
St. Louis, Missouri

W. David Bevirt, P.E.
SMACNA, Inc.
Vienna, Virginia

MANUFACTURERS

Raymond Barnum
Harrison Radiator Division
of General Motors Corporation
Lockport, New York

Kenneth R. Bush
Trane Company
LaCrosse, Wisconsin

Dr. Nicholas H. DesChamps
DesChamps Laboratories, Inc.
East Hanover, New Jersey

Kon Dravnieks
Energy Recovery Company
Germantown, Wisconsin

Joseph A. Pash
Kathabar Systems, Midland Ross Corporation
New Brunswick, New Jersey

Owen L. Riddell
Barber-Colman Company
Huntsville, Alabama

OTHER CONTRIBUTORS

ASHRAE Technical Committee 5.5-
Air-to-Air Energy Recovery
Justin H. Smith, Chairman

John A. Sharp
Dow Chemical U.S.A.

William E. Rice
University Mechanical & Engineering Contractors

John J. McGuinness
Lewis G. Harriman III
Cargocaire Engineering Corporation

Derald G. Welles
Modine Heating Recovery, Inc.

FOREWORD

Industry generally lacks knowledge about energy recovery systems. Energy prices have increased tremendously since the oil embargo of 1973. Some companies and building owners have made great strides in implementing measures that have significantly reduced energy waste, but most energy conserving programs are still in the planning stages.

The Sheet Metal and Air Conditioning Contractors' National Association, Inc. (SMACNA) in keeping with its policy of disseminating information and providing standards of design and construction, offers this comprehensive and fundamental manual on air-to-air energy recovery equipment and systems as part of the continuing effort to upgrade the HVAC industry and to help the nation's energy conservation program. The SMACNA Energy Recovery Committee is indebted to a large number of individuals and companies that have provided information and proprietary data used in this manual. Until data and information is available from research and test programs conducted by "neutral organizations," the text, tables and charts contained in this manual are the "state-of-the-art" of this fast-growing, but still infant, industry.

SMACNA recognizes that in the future, this manual must be expanded and updated. As need arises, manuals on related subjects may be developed. Continuing effort will be made to provide the industry with a compilation of the latest construction methods and engineering data from recognized sources, supplemented by SMACNA research and the services of local SMACNA Chapters and SMACNA Contractors.

REFERENCES

Text material for this manual has been obtained from catalogs, handbooks, and engineering manuals published by the following organizations:

1. Allied Air Products Co., Inc.	Newberg, Oregon
2. American Institute of Chemical Engineers (AIChE)	New York, New York
3. American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc. (ASHRAE)	New York, New York
4. Barber-Colman Company	Huntsville, Alabama
5. Cargocaire Engineering Corporation	Amesbury, Massachusetts
6. Carrier Corporation	Syracuse, New York
7. DesChamps Laboratories, Inc.	East Hanover, New Jersey
8. Dow Chemical U.S.A.	Midland, Michigan
9. Energy Recovery Company	Germantown, Wisconsin
10. Harrison Radiator Division of General Motors Corporation	Lockport, New York
11. Heating/Piping/Air-Conditioning Magazine	Stamford, Connecticut
12. Kathabar Systems, Midland Ross Corporation	New Brunswick, New Jersey
13. Mobil Oil Company	New York, New York
14. Modine Heating Recovery, Inc.	Racine, Wisconsin
15. Monsanto	St. Louis, Missouri
16. National Environmental Balancing Bureau (NEBB)	Vienna, Virginia
17. Q-Dot Corporation	Dallas, Texas
18. Sheet Metal and Air Conditioning Contractors' National Association (SMACNA)	Vienna, Virginia
19. Shell Oil Company	Houston, Texas
20. The Trane Company	LaCrosse, Wisconsin
21. U.S. Department of Commerce—National Bureau of Standards	Washington, D.C.
22. Vari-Cool Product Division of H.&C. Metal Products	Santa Rosa, California

Numbers in parentheses used in the text, figures, and tables refer to the numbers preceding the above organizations.

TABLE OF CONTENTS

SECTION I
ENERGY RECOVERY

INTRODUCTION

The basic objective of all air-to-air energy recovery systems or devices is to reduce the energy consumption and energy costs of a building or process by transferring energy between two airstreams such as the outside air and exhaust air of a building and/or process. A secondary objective is to reduce the size and capital cost of the supporting utility equipment (e.g., boilers, chillers, burners, etc.) of the building or process. In many cases, the proper application of air-to-air energy recovery equipment can result in little or no additional first cost for the building or process, while providing long-term benefits through reduced energy consumption.

There are three basic types of applications of air-to-air energy recovery: process-to-process; process-to-comfort; and comfort-to-comfort.

A PROCESS-TO-PROCESS

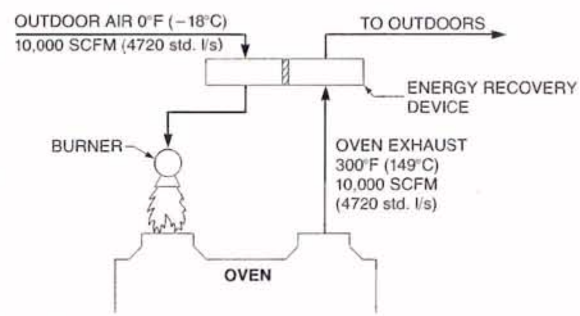
In process-to-process applications, heat is captured from the process exhaust airstream and transferred to the process supply airstream. Economics generally favor higher temperature processes, since more heat is available for recovery, and equipment is available to handle process exhaust temperatures as high as 1600°F (870°C). Typical applications include dryers, kilns and ovens.

When considering application of air-to-air energy recovery from process exhaust, the following should be evaluated:

1. *Effects of Corrosives.* Process exhaust frequently contains particulates or substances which require compatible materials of construction for the energy recovery device and system.
2. *Effects of Condensibles.* The process exhaust may contain a concentration of a substance sufficiently high to cause it to condense upon cooling in the energy recovery device. In some cases, the condensed substance can be recovered.
3. *Effects of Contaminants.* If the process exhaust contains particulate contaminants or condensibles, the energy recovery device should require access for cleaning or use of washdown equipment. Consideration should also be given to air prefiltration and to selection of a recovery device designed to minimize frequency of cleaning.
4. *Effects On Other Equipment.* The removal of heat from some types of process exhaust may reduce the cost of pollution control by allowing less expensive bags in baghouses or improving the efficiency of electronic precipitators. As a result, energy recovery and pollution control can often be coupled with beneficial effects.
5. *Codes and Insurance Requirements.* All local and national codes should be reviewed prior to an in depth analysis of an energy recovery system application. Insurance companies often have restrictive clauses in policies on buildings and equipment that could substantially increase the cost of energy recovery installations.

Process-to-process recovery devices generally recover only sensible heat and do not transfer latent heat; in most cases, latent heat transfer could actually be detrimental to the process.

Modulation of heat recovery is generally not required or desirable in a process-to-process application, since the maximum efficiency of recovery is desired. In some cases involving condensibles, modulation may be required to prevent overcooling of the process exhaust airstream.



ENERGY USAGE
WITH RECOVERY. 2 052 000 BTUH (601 373 W)
WITHOUT RECOVERY. 4,320 000 BTUH (1 266 048 W)

Figure 1-1 PROCESS-TO-PROCESS ENERGY RECOVERY SYSTEM (3)

B PROCESS-TO-COMFORT

In process-to-comfort applications, waste heat is captured from a process exhaust and used to heat building makeup air and/or to heat air for exterior zones of multi-story buildings during the winter months. Typical applications include foundries, strip coating plants, can plants, plating operations, and other processing areas having large exhaust and makeup air volume requirements. Heat from sources such as refrigeration compressors and condensing units, can be used directly to contribute to the heating of many supermarkets without using recovery devices.

Whereas full recovery is generally desired at all times in a process-to-process application, recovery modulation is necessary in process-to-comfort applications to prevent overheating the makeup air during milder weather. During the summer months, no recovery is required. Because energy cost savings are only obtained during the winter months and recovery is modulated during moderate weather, yearly energy savings are generally not as great as in process-to-process applications.

A summer application of process-to-comfort without benefit of energy recovery devices, is being used in many supermarkets. The concentrated cold air around banks of refrigerated display cases that have remote condensing units is collected and fed into the return air of the air conditioning system, thus substantially reducing the load on the space cooling equipment.

When considering application of process-to-comfort energy recovery, the following should be evaluated:

1. *Effects of Corrosives.* Process exhaust frequently contains substances which require compatible materials of construction for the energy recovery device.
2. *Effects of Condensibles.* The process exhaust may contain a concentration of a substance sufficiently high to cause it to condense upon cooling in the energy recovery device.
3. *Effects of Contaminants.* If the process exhaust contains particulate contaminants or condensibles, the energy recovery device should be accessible for cleaning. Consideration should also be given to air prefiltration and to selection of a recovery device having an "open" structure to minimize frequency of cleaning.
4. *Carry-over.* Special care should be used in the selection of energy recovery equipment when the process gasses contain noxious or toxic substances. This same caution also applies to the location of the outside air intakes and discharges.

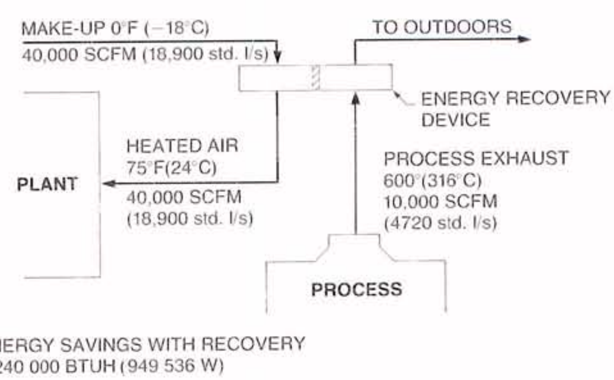


Figure 1-2 PROCESS-TO-COMFORT ENERGY RECOVERY SYSTEM (3)

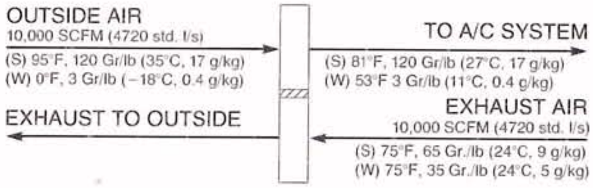
Process-to-comfort recovery devices generally recover only sensible heat and do not transfer latent heat.

C COMFORT-TO-COMFORT

In comfort-to-comfort applications, energy is captured from the exhaust air from a building and used to precondition the outside air to the building. In comfort-to-comfort applications, the energy recovery process is reversible; i.e., the enthalpy of the building supply air is lowered during warm weather, and raised during cold weather.

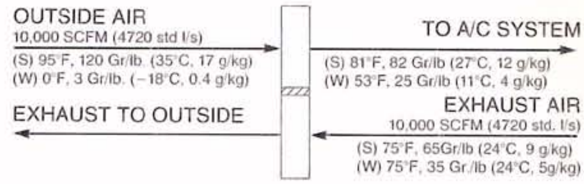
Air-to-air energy recovery devices available for comfort-to-comfort applications generally fall into two categories: sensible-only devices, and total-enthalpy devices.

Sensible-only devices normally transfer sensible heat between supply and exhaust airstreams, while *total-enthalpy or total heat devices* transfer both sensible heat and latent heat (humidity) between supply and exhaust airstreams. Unlike process-to-process and process-to-comfort applications, latent heat transfer is generally desirable in comfort-to-comfort applications. Figure 1-3 depicts a typical comfort-to-comfort application of a sensible-only device of 70% effectiveness, (based on the total amount of sensible heat available) and Figure 1-4 depicts a total-enthalpy device of 70% effectiveness (based on



ENERGY ANALYSIS
ENERGY SAVED (SUMMER) = 151 200 Btu/hr (44 312 W)
ENERGY SAVED (WINTER) = 567 000 Btu/hr (166 170 W)

Figure 1-3 COMFORT-TO-COMFORT SENSIBLE TRANSFER ONLY (3)



ENERGY ANALYSIS
ENERGY SAVED (SUMMER) = 427 500 Btu/hr (125 286 W)
ENERGY SAVED (WINTER) = 729 000 Btu/hr (213 645 W)

Figure 1-4 COMFORT-TO-COMFORT TOTAL ENTHALPY TRANSFER (3)

the total amount of total heat available) operating under the same conditions. Under typical summer design conditions, the total-enthalpy device will recover nearly two times as much energy as the sensible-only device, and under typical winter design conditions the total-enthalpy device will recovery over 25% more energy than the sensible-only device. Figure 1-5 graphically depicts the performance of sensible-only and total-enthalpy devices plotted on a psychrometric chart.

When considering application of comfort-to-comfort energy recovery, the following should be evaluated:

1. *Effect of Particulate Contaminants.* If the building exhaust air contains large amounts of particulate matter, such as dust, lint, animal hair, etc., provision should be made for adequate prefiltration of building exhaust air to prevent plugging of the energy recovery device. The same quality of prefiltration should be provided for any other piece of air-conditioning equipment.
2. *Effect of Gaseous/Vaporous Contaminants.* If the building exhaust air contains gaseous or vaporous contaminants, such as hydrocarbons, sulfur compounds, and water-soluble chemicals, the effect of these compounds on the energy-recovery device should be investigated.
3. *Effect of Indoor Humidity Level During Winter Operation.* If the building is humidity-controlled during the winter months, or if the building is located in a cold climate, winter performance of the energy-recovery device should be thoroughly investigated. Special controls may be required to prevent frosting of the energy recovery device or other operational problems during the winter operation.
4. *Effect of Sensible and Enthalpy Recovery.* When evaluating comfort-to-comfort energy recovery, the costs and benefits of sensible-only and total-enthalpy devices should be investigated, where applicable, to determine which has the greater short term and long term benefits.
5. *Effect of Supply and Exhaust Location.* Where exhaust and supply ductwork are in close proximity, any of a number of energy recovery devices can be designed into the air system. Other energy recovery devices are able to handle large physical separations between the supply air and exhaust air systems.

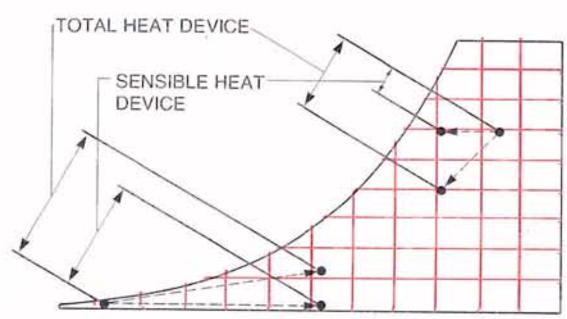


Figure 1-5 SENSIBLE HEAT VS. TOTAL HEAT ENERGY RECOVERY

SECTION II
FUNDAMENTALS OF ENERGY TRANSFER

To be able to evaluate energy recovery equipment ratings and to obtain the necessary expertise to analyze which energy recovery system best suits the application, the fundamental laws of thermodynamics and psychrometrics must be known and used by the system designer. This section will contain a review and explanation of these fundamentals as they relate to energy recovery systems.

A "HEAT" ENERGY

1. Thermodynamics

Heat is one of the several forms of energy which can be converted by various methods to or from energy in mechanical, chemical, electrical and nuclear forms. Thermodynamics is the science of heat energy and its transformations to and from these other forms of energy.

There are two fundamental laws of thermodynamics which can be stated in different, but equivalent ways. For the purpose of energy recovery systems, the laws will be stated as:

(a) *First* — Energy can neither be created nor destroyed (the net increase in the energy content of a particular system in a given period is equal to the energy content of the material leaving the system, plus the work done on the system, plus the heat added to the system).

(b) *Second*—it is impossible for a self-acting machine, unaided by any external agency, to convey heat from a body of lower temperature to one of higher temperature (heat flow always occurs from the higher temperature level to the lower temperature level).

2. Heat Transfer

In environmental systems, as well as in natural processes, heat is transferred by three means:

a. radiation
b. conduction
c. convection

Heat transfer or flow is measured in British thermal units (Btu) per hour in U.S. units and watts (W) in metric units. One watt equals 3.412 Btu/hr.

The differences in heat absorption by various substances is recognized by the term "specific heat." The specific heat (C) of water is 1.0 Btu per pound-degree F (4190 joule per kilogram-kelvin in the metric system) and the specific heat (C_p) of air is 0.2388 Btu/lb°F (1000 J/kg°K).

The science of heat transfer for energy recovery equipment is mainly by convection. Heat may be transferred by this method from material to material, as well as within the same material provided there is contact between the materials. Transfer by conduction is usually greatest in metals while organic solids, such as wood and asbestos retard the energy flow (act as insulators).

The thermal conductivity (k) is the heat transfer coefficient used to express the time rate of heat flow perpendicular to the surface through a homogeneous material in units of Btu-thickness per unit time per unit area per degree temperature difference (Btu -in/hr/sq.ft./°F) or (W/m°K in the metric system).

Heat flow in energy recovery equipment is normally from a fluid (or gas) through a thin wall into another fluid (or gas), or is into a "transfer substance" which moves into or to another cooler fluid (or gas) to deposit its energy. The method of transfer will be covered for each individual type of equipment in sections that follow.

Major factors in the transfer of heat by conduction are:

a. temperature difference
b. size and shape of the transfer surface
c. type of fluid (or gas) and flow velocity
d. conductivity of heat transfer material
e. conductivity of the "boundary layer."

There are also many other factors to consider when designing or using energy recovery equipment or systems, such as film coefficients, fouling, corrosion, condensables, frost or freezing, poor maintenance, etc.

3. Parallel and Counterflow Transfer

Although temperature difference is one of the major factors of heat flow, the transfer surfaces rarely have the same temperature throughout. In most energy exchangers, there is a difference between the entering and leaving temperatures of each fluid or gas, and there is a difference between temperatures of the fluids or gases across the dividing surfaces. It is, therefore, desirable to arrange the flows to produce the highest possible temperature difference across the dividing surfaces or with the exchange media.

There are several fluid flow direction patterns used in energy exchange equipment, such as parallel flow, cross-flow and counterflow. For simplification, the relationship between fluids with parallel flow and counterflow will be covered. Each flow pattern name is derived from the physical relationship of one flow direction to the other.

Fluid A, in Figure 2-1, is the higher temperature fluid which imparts heat through the heat exchanger wall to the lower temperature fluid, B. The maximum temperature that any part of fluid B may approach, but never equal (or become greater than), is the highest temperature of fluid A. There is a distinct relationship between the temperature of either fluid with respect to the other in each case, as illustrated by the graphs in Figure 2-2.

Observe that the temperature difference between the fluids changes from point to point across the exchanger. If the curves on the graphs were straight lines, it would be possible to calculate a useful arithmetical average which could be applied to determine the heat transfer. However, the curvature of the lines representing the differences makes this impossible. Instead, the average is related to a changing scale or logarithmic function and the actual average is called the "log mean temperature difference" or LMTD. This average does not occur at the physical middle of the distance from the entrance to the exit of the heat exchanger as the arithmetical average might, but at a point toward one end of the exchanger, depending upon the flow direction and the temperatures.

Since heat transfer will increase with temperature difference, it is desirable in most cases to select the exchanger on the basis of the highest LMTD. Counterflow produces this condition. Except in special cases, heat exchangers are designed using counterflow patterns to obtain maximum heat transfer with the least heat transfer surface or "dividing wall." An example of such an exception is in outside air coil applications, where maximum heat transfer may be sacrificed to insure the hottest fluid temperature on the cold air entering the side of the coil. Another illustration of this heat transfer relationship is shown in Figure 2-3 for a standard water coil used in environmental systems and run-around coil energy recovery systems.

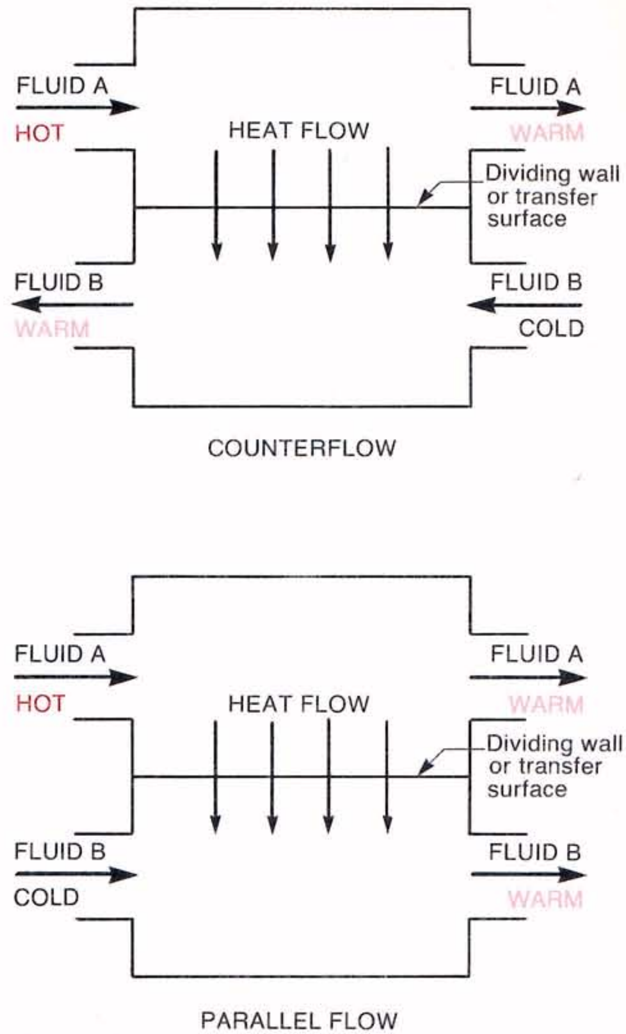


Figure 2-1 HEAT TRANSFER FLOW PATTERNS

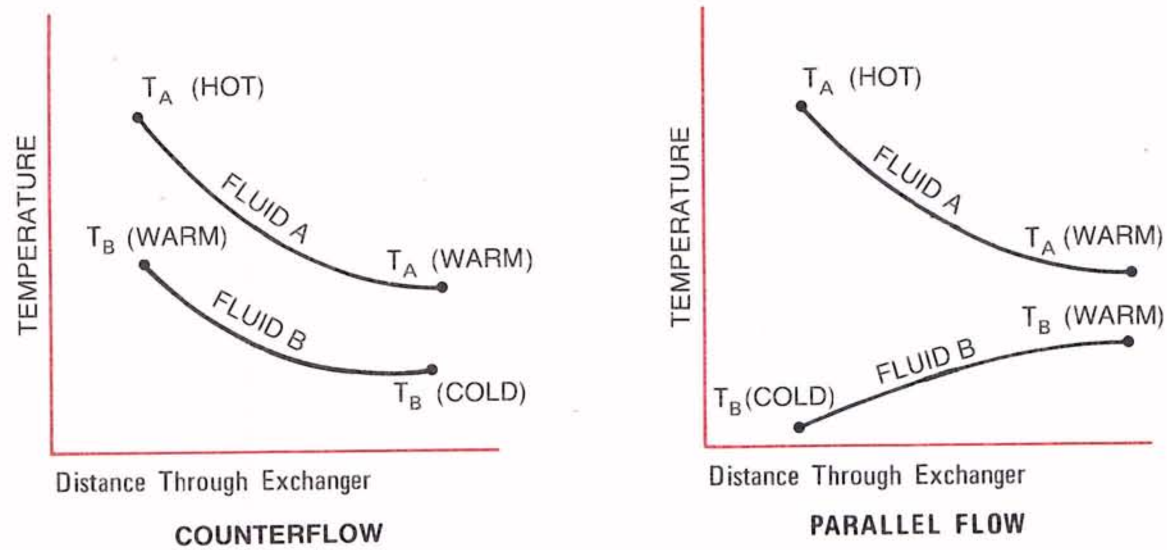


Figure 2-2 PARALLEL AND COUNTERFLOW HEAT TRANSFER CURVES

B PSYCHROMETRICS

1. Introduction

Psychrometrics deals with the thermodynamic properties of moist air and the application of these properties to the environment and the environmental system.

Thermodynamics is the science of heat energy and its transformation or change from one form of energy to another.

Since air is the final environment and one of the system's major fluids, whatever affects air affects the system and the environment. Whatever happens to the air and the moisture it contains, under both natural circumstances and artificial conditions imposed by the systems and the environment, is of concern to the designer. He must understand the language of psychrometrics and be able to use the psychrometric chart and tables as tools to change the existing conditions to those desired or required. All basic terms used in this manual will be found in Section XVIII — "Glossary".

2. Properties of Air

Dry air is an unequal mixture of gases consisting principally of nitrogen, oxygen, and small amounts of neon, helium, and argon. The percentage of each gas will normally be the same from sample to sample, although carbon dioxide and sulfur dioxide (pollutants) might be present in varying quantities.

Air in our atmosphere, however, is not dry but contains small amounts of moisture in the form of water vapor. Thus another gas is added to the mixture, and the percentage may vary from sample to sample. This air-water mixture is the "moist air" referred to in the subject of psychrometrics. The amount of water vapor in atmospheric air normally represents less than 1% of the weight of the moist air mixture. If an average of the weight of air is approximately 0.075 pounds per cubic foot, the moisture contained therein will weigh less than 0.00075 pounds per cubic foot. This would seem to be an insignificant amount to cause so much concern. Normally, atmospheric air contains only a portion of the water it is able to absorb (partially saturated or superheated). If the proper conditions occur, air will absorb additional moisture until it can absorb no more, and it is then "saturated."

Air and water vapor behave as though the other were not present. Each will act as an independent gas and exert the same pressure as if it were alone. The barometric pressure is the sum of the two pressures — the partial pressure of the air, plus the partial pressure of the water vapor.

The dry air component of the moist air exists as only a gas under all environmental conditions and can not be liquified by pressure alone, therefore acting as a perfect gas. However, water vapor does coexist with water as a liquid at all environmental temperatures, and it can be liquified by pressure. As it is not a

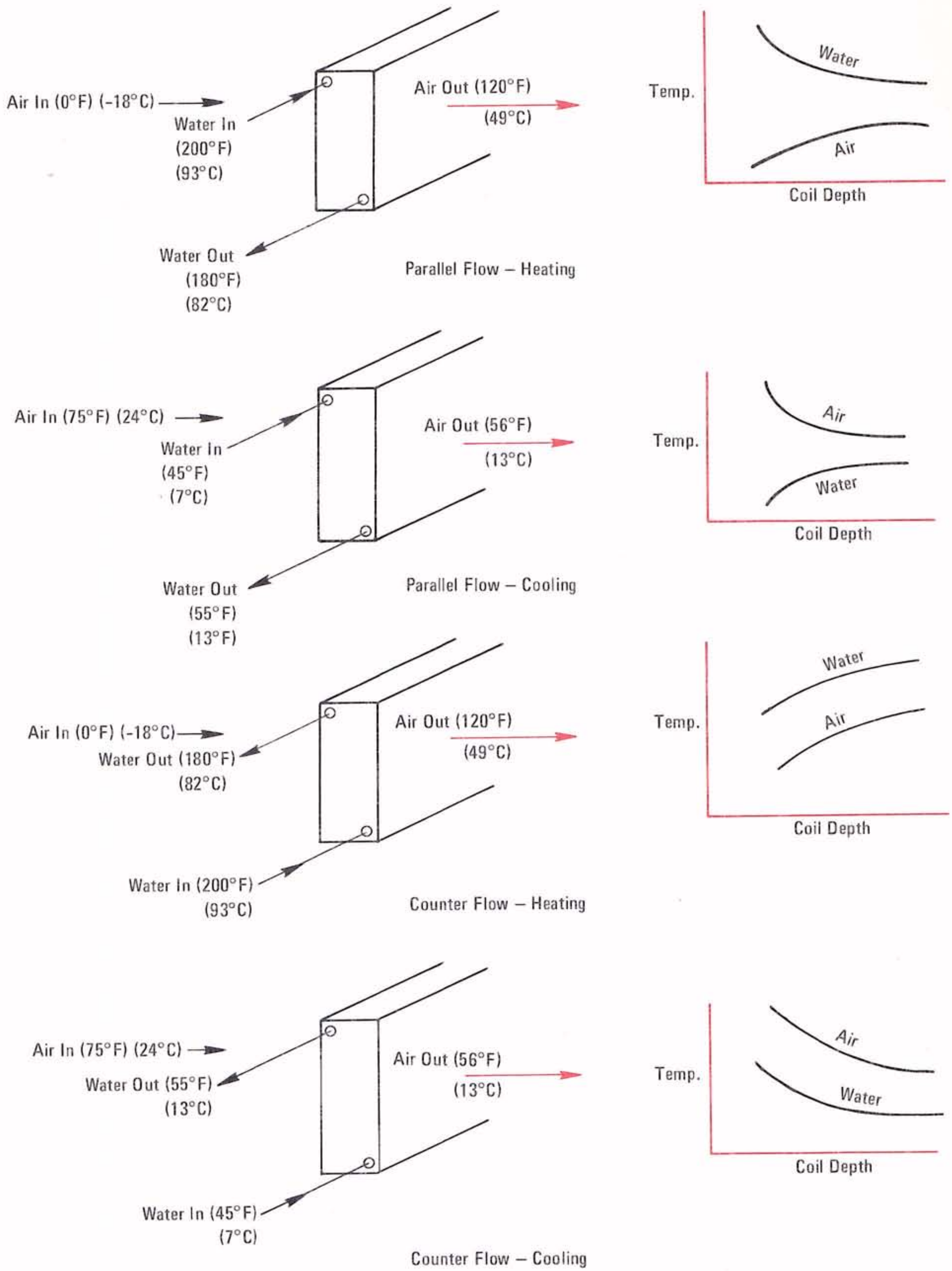


Figure 2-3 HEAT TRANSFER RELATIONSHIP

perfect gas, the properties of water vapor are determined experimentally.

3. Air-vapor Relationship

Throughout the *normal* ranges of atmospheric pressures and temperatures, the air and water vapor mixture behaves as a perfect gas provided no condensation or evaporation takes place. A "perfect gas" is one where the relationship of pressure, temperature and volume may be defined and predicted by the equation:

$$\frac{PV}{T} = R \qquad \textbf{Equation 2-1}$$

where:

P = pressure (lb/sq ft)
V = volume of 1 lb (cu ft)
T = absolute temperature ($460 \times t°F$)
$R = \frac{1545}{\text{molecular weight}}$

The actual value of R as a number has little meaning here, but the fact that R remains constant for any given "perfect" gas is extremely important. It is possible to equate the P, V, T values for two different conditions 1 and 2 for the same gas:

$$\frac{P_1V_1}{T_1} = R = \frac{P_2V_2}{T_2} \text{ or } \frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

If it is assumed that atmospheric pressure in lb/ft^2 remains essentially constant at a given elevation on earth, only varying by a one or so inches of mercury, then:

$$P_1 = \frac{T_1}{V_1} = P_2 = \frac{T_2}{V_2} \text{ or } \frac{T_1}{V_1} = \frac{T_2}{V_2} \text{ or } \frac{V_1}{T_1} = \frac{V_2}{T_2}$$

and:

$$V_2 = \frac{V_1T_2}{T_1}$$

"Standard air" is a fixed reference for air conditioning processes (from SMACNA Manual for the Balancing and Adjustment of Air Distribution Systems), and has the following values for the T, P, V properties:

T = 70°F
P = 29.92 inches of mercury (at sea level)
V = 13.35 cubic feet per pound (or a density)

$d = \frac{1}{V}$, or .075 lb/cu ft)

EXAMPLE NO. 1

Using the previous equation, any condition other than standard may be calculated if one of the final conditions is known, For example, if one pound of standard air were heated to 700°F, as in a process application, the new volume may be determined:

$$V_2 = \frac{V_1T_2}{T_1} = 13.35 \times \frac{460 + 700}{460 + 70} = 13.35 \times \frac{1160}{530}$$

$V_2 = 29.2$ cu ft (per pound)

Similar calculations may be made for any value of temperature so long as the pressure remains constant.

EXAMPLE NO. 2

It is possible to make similar calculations which include pressure variations. An example might be to find the correct volume for an altitude of 5000 feet. Assuming the temperature to be the same at both points for convenience:

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}, \; T_1 = T_2 \text{ and } P_1V_1 = P_2V_2$$

$$\text{or } V_2 = V_1\frac{P_1}{P_2}$$

The standard atmospheric pressure at 5000 feet is 24.90 inches of mercury (from the Tables and Charts Section). Substituting in the equation:

$$V_2 = V_1\frac{P_1}{P_2} = 13.35 \times \frac{29.92}{24.90} = 16.05 \text{ cu ft}$$

The conclusion to be reached from these examples is that large deviations from the "standard air" temperature and pressure values require corrections to be made to the calculations, while relatively small variations may be ignored. In the case of pressure, no correction is normally used in applications below 2000 feet above sea level. Above 2000 feet it becomes necessary to make the correction, since the air has a significant reduction in its ability to carry heat. Notice in Example No. 2, at sea level the

density $= \frac{1}{13.35} = 0.075$ lb/cu ft and at 5000 feet the

density $= \frac{1}{16.05} = 0.0625$ lb/cu ft, or a reduction of

17%.

By a series of calculations and laboratory measurements, a long list of values are obtained and are listed in Tables 17-1 and 17-2 "Air Density and Correction Factors." To be meaningful, these values must have a reference, which has been stated to be standard atmospheric pressure at sea level: 29.92 inches of mercury (101.3 kPa). Since we have established that variations caused by pressure are not serious below an altitude of 2000 feet, the values obtained by maintaining the 29.92 inches of mercury are adequate for design of energy recovery systems.

However, as seen in Example No.1., temperature variations in energy recovery systems can cause wide variations in the air density:

$$\text{density} = \frac{1}{29.2} = 0.034 \text{ lb/cu ft, or a reduction}$$

of 54%. For this reason, many engineering catalogs use standard cfm (scfm) or cfm corrected to "standard air" conditions.

SECTION III
USE OF THE PSYCHROMETRIC CHART

A THE PSYCHROMETRIC CHART

A psychrometric chart is a modified Mollier's chart for moist air (the mixture of dry air and water vapor discussed earlier). It is a series of graphs or curves arranged in such a way, that by knowing a specific value of each of two different properties, a point can be obtained that will determine the values for all other properties under the same conditions. Charts are available for different elevations above sea level, higher or lower tempreatures, and many other variations (see Figures 3-9 to 3-13).

Referring to the sample chart in Figure 3-1, note that the groups or "families" of curves have been labeled to indicate which part or parts of the graph are for the different properties of moist air. Definitions of the properties may be found in the "Glossary" — Section XVIII.

1. Basic Grid, Humidity Ratio (Specific Humidity)

The horizontal parallel and equidistant grid lines with the scale displayed along the right side of the chart indicate the "grains of moisture per pound of dry air" or "pounds of moisture per pound of dry air." The bottom line of the chart is zero humidity and represents totally dry air. The chemical industry prefers to use "mol-ratios", and "grams of moisture per kilogram of dry air" is used in the metric system.

0.001 lb of moisture/lb of dry air =
1 gram of moisture/kg of dry air

2. Enthalpy (Total Heat)

Enthalpy lines are slanted from top-left to the bottom/right. Enthalpy is designated by the letter "h" and, as all values on the chart, is referred to a pound of dry air. This is the only value which does not change through various processes, such as heating-cooling, compression-expansion, and humidification-dehumidification. Other constant value lines are superimposed upon this basic grid by plotting and are neither equidistant nor parallel, although they may appear to be so. Some industrial charts are drawn with the grid being formed by parallel and equidistant humidity ratio and dry bulb temperature lines. Enthalpy lines then are neither parallel nor equidistant.

In Europe, psychrometric charts are shown in agreement with the Mollier diagram. In comparison with charts used in U.S.A., they are mirror-viewed and then turned 90°. (See Figure 3-13)

3. Dry Bulb Temperature

Constant dry bulb temperature lines on the chart are nearly vertical. They diverge slightly towards the top of the chart. The scale is at the bottom, along the "dry air" line (zero humidity ratio) from left to right, and values are given in °F (Fahrenheit) in the English system and in °C (celsius) in the metric system.

4. Saturation Line, Dew Point & Wet Bulb

The curved upper border-line of the chart which runs from the bottom/left to the mid-top is an experimentally plotted saturation line. It shows the *maximum* amount of water vapor (in pounds) which can be associated with a pound of dry air at a given dry bulb temperature. Air is said to be saturated (with moisture) at this point. The temperature at this point is known as the *saturation temperature*.

It can be seen from the saturation line that at higher temperatures air can hold more moisture than at lower temperatures. Conversely, the capacity to hold moisture is less at lower temperatures. Therefore, if the already saturated moist air is being cooled, the excess moisture instantly begins to separate from the moist air by *condensation* either in the form of *fog* (the left side of the saturation curve is the fog area) or as *dew* if it condenses on a cold surface.

The point of saturation is also called the *dew point* and the saturation temperature is called the dew point or wet bulb temperature. The saturation temperature values (also dew point or wet bulb temperatures) are shown in °F along the saturation line where it is crossed by the same value dry bulb lines.

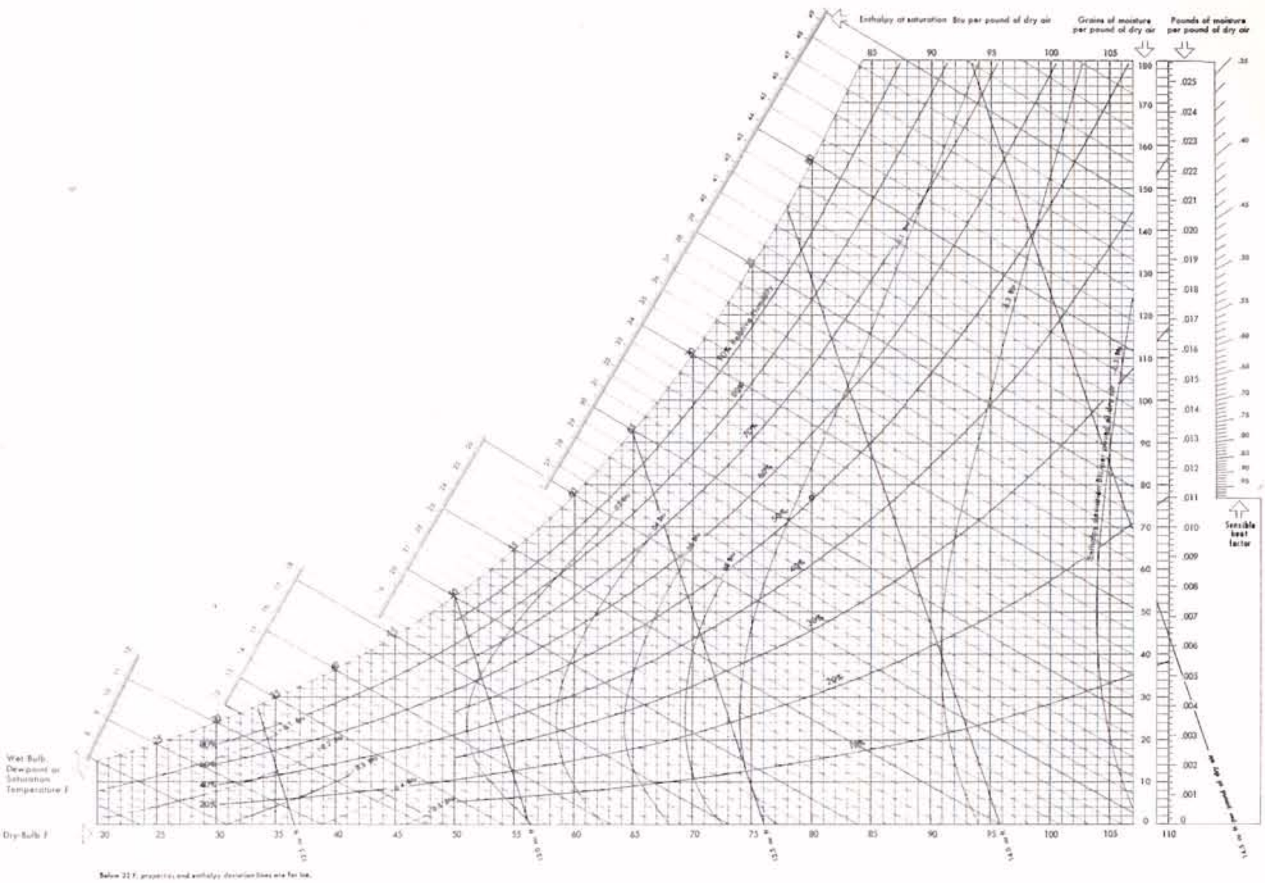


Figure 3-1 SAMPLE PSYCHROMETRIC CHART (6)

5. Specific Volume

Specific volume lines run at a steep angle from top/left to bottom/right. The numerical values, along the bottom of the chart at the end of these lines, are given in cubic feet per pound of dry air in English units (and in cubic meters per kilogram in the Metric system). The conversion to metric is:

1 cu ft/lb = 0.0624 m^3/kg

The slant and spacing of the specific volume lines show that moist air becomes lighter (by expansion) with an increase in temperature and with an increase in moisture content (moist air is lighter than dry air at the same temperature). The specific volume of dry air can be found at the intersection with the bottom, zero humidity ratio line. The volume of the water vapor can be found by subtracting the volume of the dry air from the volume of the moist air.

6. Relative Humidity

The curved constant relative humidity (RH) lines lay between the zero humidity ratio line at the bottom and the curved saturation line above. The curvature decreases as curves approach the dry air line.

Relative humidity expresses the proximity of the subject moist air to that of saturated air at the same temperature. The saturation line represents 100% RH and the bottom line of the chart is 0% RH.

Another term used to define proximity of the moist air to saturation is the *degree of saturation* which is the ratio of the humidity-ratio of the moist air to that of the saturated moist air at the same temperature and pressure. The difference between both meanings is small but still noticeable within the comfort conditions. While the degree of saturation of 50% lays on the dry bulb temperature line directly in the middle between 0% and 100%, the 50% RH point is slightly below the mid-point.

Humidity of moist air can be determined by several methods, but the humidity ratio is an absolute value and is difficult to measure. Use of the dew point is the most exact, direct and repetitive method requiring no calibration and charts, but the equipment is expensive. Measurement of relative humidity depends on changes in humidity responsive materials. While low cost and quite accurate, these instruments require frequent calibration.

7. Wet Bulb Temperature

The wet bulb method was developed to obtain a more practical way to measure relative humidity and depends on the evaporation of water around the bulb of a mercury thermometer. Since evaporation and, consequently, the depression of the wet bulb temperature depends on the relative humidity of the moist air, it can be used to measure relative humidity by use of conversion tables.

At the saturation point, there is no evaporation. Since the wet bulb depression is zero, the wet bulb and dry bulb temperatures are equal at this point and are also equal to the dew point temperature (saturation temperature). As the wet bulb lines run so close to the enthalpy lines, most psychrometric charts use the same lines for both wet bulb temperatures and enthalpy and show correction of either enthalpy values or wet bulb values by another family of curves.

8. Plotting Conditions

Consider point A, in Figure 3-2, representing summer outdoor design conditions. By finding the point which represents 95°DB and 78°WB, the values for the other properties are:

Dew point temperature = 71.9°F

Relative humidity = 48%

Enthalpy (total heat) = 41.6 Btu/lb dry air

Moisture content = 118 grains/lb dry air or
.0169 lb moisture/lb dry air.

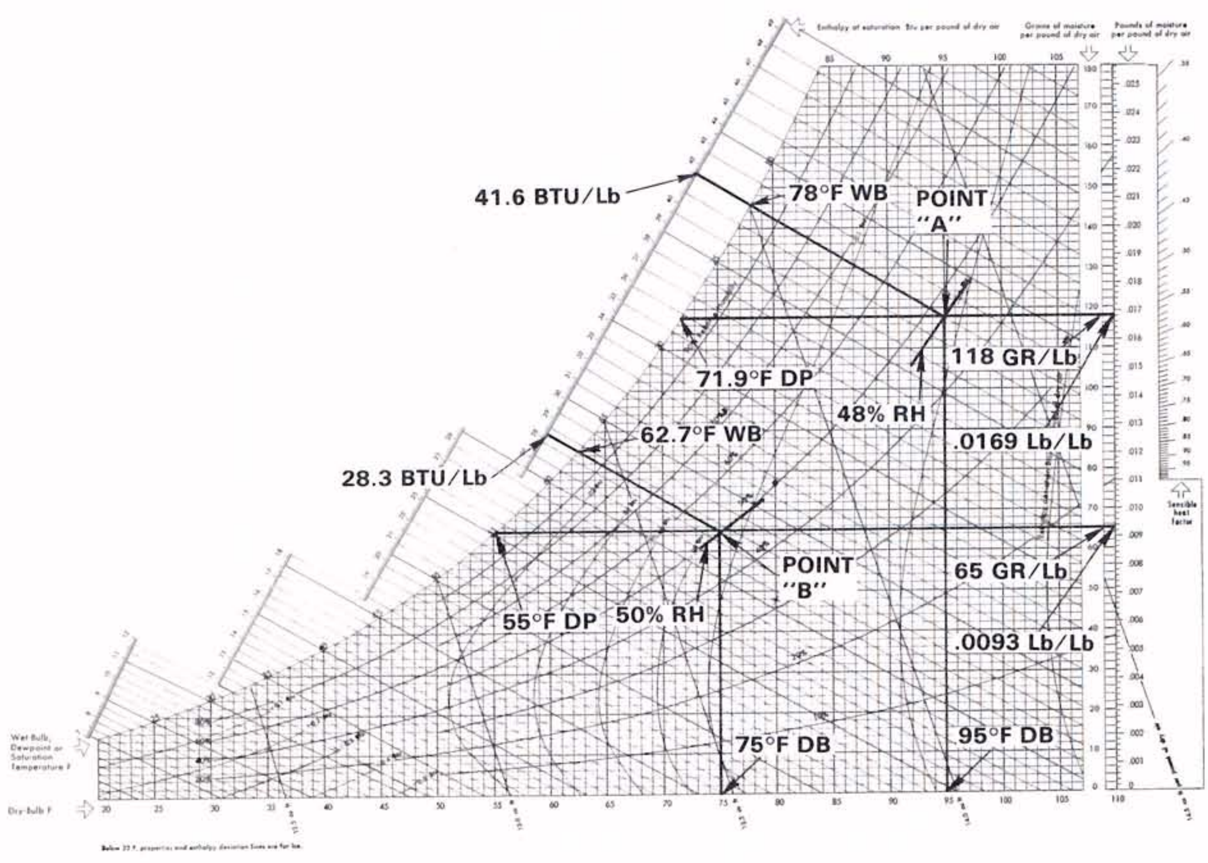


Figure 3-2 PSYCHROMETRIC CHART — TYPICAL CONDITION POINTS

Now consider point B in Figure 3-2 representing summer indoor design conditions. By finding the point which represents 75°DB and 50% RH, the values for the other properties are:

Dew point temperature = 55°F

Wet bulb temperature = 62.7°F

Enthalpy (total heat) = 28.3 Btu/lb dry air

Moisture content = 65 grains/lb dry air or .0093 lb moisture/lb dry air.

In this way, any condition may be plotted on the chart for normal environmental systems. Other charts are available for plotting from tabular data for conditions not found here, but the need for such variations is not common.

B CONDITION CHANGES ON THE PSYCHROMETRIC CHART

From the chart, simple logic leads to the conclusion that there must be some way to change the properties of the air initially at the conditions of point A to the conditions of point B. The means to accomplish such a change are the environmental systems. The designer, by knowing what design conditions must be satisfied, but without knowing the airflow rate, is able to plot the various changes in different portions of the system and the environment. He is then able to determine what systems are capable of accomplishing the necessary results. In addition, he uses the heat values in the design calculations and is able to see immediately if the design conditions are practical or impossible.

But first, the various condition changes must be illustrated and understood. For this purpose, skeleton charts have been used in the related diagrams.

1. Sensible Changes

Sensible heat, by definition, indicates only dry bulb temperature changes. Therefore, any heating system not including humidification is a sensible heat process or system and is represented on the chart as a horizontal straight line. Conversely, any cooling system utilizing a dry coil which does not dehumidify or where the surface of the coil does not fall to or below the air dew point is also a horizontal straight line.

A. Heating

Assume that air which is at 70°F and 20% RH is heated to 105°F. This process is indicated in Figure 3-3. Conditions other than those mentioned here have been determined from the psychrometric chart.

By considering the values, it may be seen that the dew point has not changed and the total moisture content in grains/lb has not changed. The dry bulb temperature has gone up, the heat content has gone up, the wet bulb temperature has gone up, and the ability of the air to absorb moisture has gone up as indicated by the *decrease* in relative humidity. The example is theoretical because moisture from people, cooking, infiltration, etc., will make some contribution to the moisture in the air. However, this is the design diagram for heating without deliberate humidification.

B. Cooling

Now consider the process and ignore the extraneous moisture sources noted above. In ideal conditions, air gives up its heat along the same line to maintain the occupied spaces at the given 70°F DB and 20% RH. A cooling coil, selected to cool air from 105°F DB and 26° DP to 70°F DB would also produce conditions along the same line as long as the coil surface temperature was above 26°F. In most systems, this would be impractical if not impossible, since the coil would immediately clog with frost.

2. Latent Changes

Latent heat, by definition, involves a change of state of a fluid; and in the case of air, this means the addi-

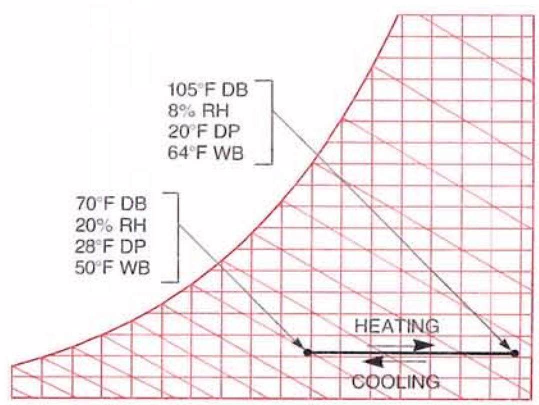


Figure 3-3 SENSIBLE HEATING AND COOLING

tion or removal of moisture. It must be remembered that the dry bulb temperature *does not change* during this addition or removal of moisture. Therefore, a vertical line on the psychrometric chart between any two points at constant dry bulb temperature represents a change in latent heat. This process is indicated on the sample chart in Figure 3-4.

Humidification and dehumidification are defined as the addition or subtraction of moisture from air. Since each is a change of state from liquid to gas and gas to liquid, each occurs at a constant dry bulb temperature, but a varying wet bulb temperature. Note that this is the same process as the addition or subtraction of latent heat and is also the same vertical line on the chart in Figure 3-4 at a constant dry bulb temperature. Humidification and dehumidification are both latent heat processes, and both are shown on the same chart.

In this example, the only constant value is the dry bulb temperature; all other properties increase for humidification and decrease for dehumidification. Note that this process is essentially an illustration and cannot normally be reproduced in environmental systems.

3. Combination Changes

Combination sensible-latent heat processes are the rule in most systems. The addition or subtraction of latent and sensible heat appears as a combination process with all changes occurring simultaneously. The result is neither a horizontal or vertical line but a slanted one tilted in the direction dictated by the process. Referring to Figure 3-5, consider the general rules below based on the two endpoints of the process, the first being the initial condition of the air, and the second being the final condition after the process or a portion of the air treatment has been completed. On the chart, all processes have the same initial point, and the arrow point indicates each arbitrary final point:

Sensible heating is a horizontal line from left to right.

Sensible cooling is a horizontal line from right to left.

Humidification is a vertical line upward.

Dehumidification is a vertical line downward.

Heating and humidification is a line sloping upward to the right.

Cooling and dehumidification is a line sloping downward to the left.

Evaporative cooling is a line sloping upward to the left.

Chemical dehydration or dehumidification is a line sloping downward to the right.

Now consider the specific cooling-dehumidification example which might represent the conditions obtained from a cooling coil using 100% recirculated air. Assume that the entering air and room conditions are 80°F DB and 50% RH. Also assume that the cooling coil can produce leaving conditions of the air at 55°F DB and 54°F WB. The illustration of the process is shown in Figure 3-6.

The line drawn between the initial and final conditions represents the change of air properties produced by the cooling coil and is conveniently drawn straight. In actual fact, the process follows a curve, but the deviation is not usually important to the system analysis. To the coil designer, however, the curvature is critical, since it indicates the heat transfer conditions from point-to-point through the coil depth. The amount of moisture that was removed from the air (condensed on the coil) was 16 grains/lb of dry air (77 − 61 = 16).

The reverse operation, heating and humidifying, could be explained by working the cooling-dehumidifying diagram backwards. However, a more practical application may be obtained by reusing the previous sensible heating-cooling diagram and adding the humidification. Assuming that 70° DB, 20% RH air returns to a heating coil and that a humidifier has been added, the *combination* process, assuming the required leaving conditions to be 105°F DB and 40% RH, is illustrated in Figure 3-7. 114 grains of moisture/lb of dry air was added in the process (136 − 22 = 114).

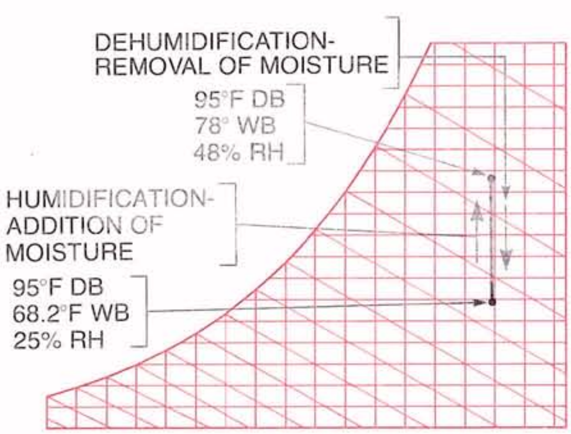


Figure 3-4 HUMIDIFICATION AND DEHUMIDIFICATION

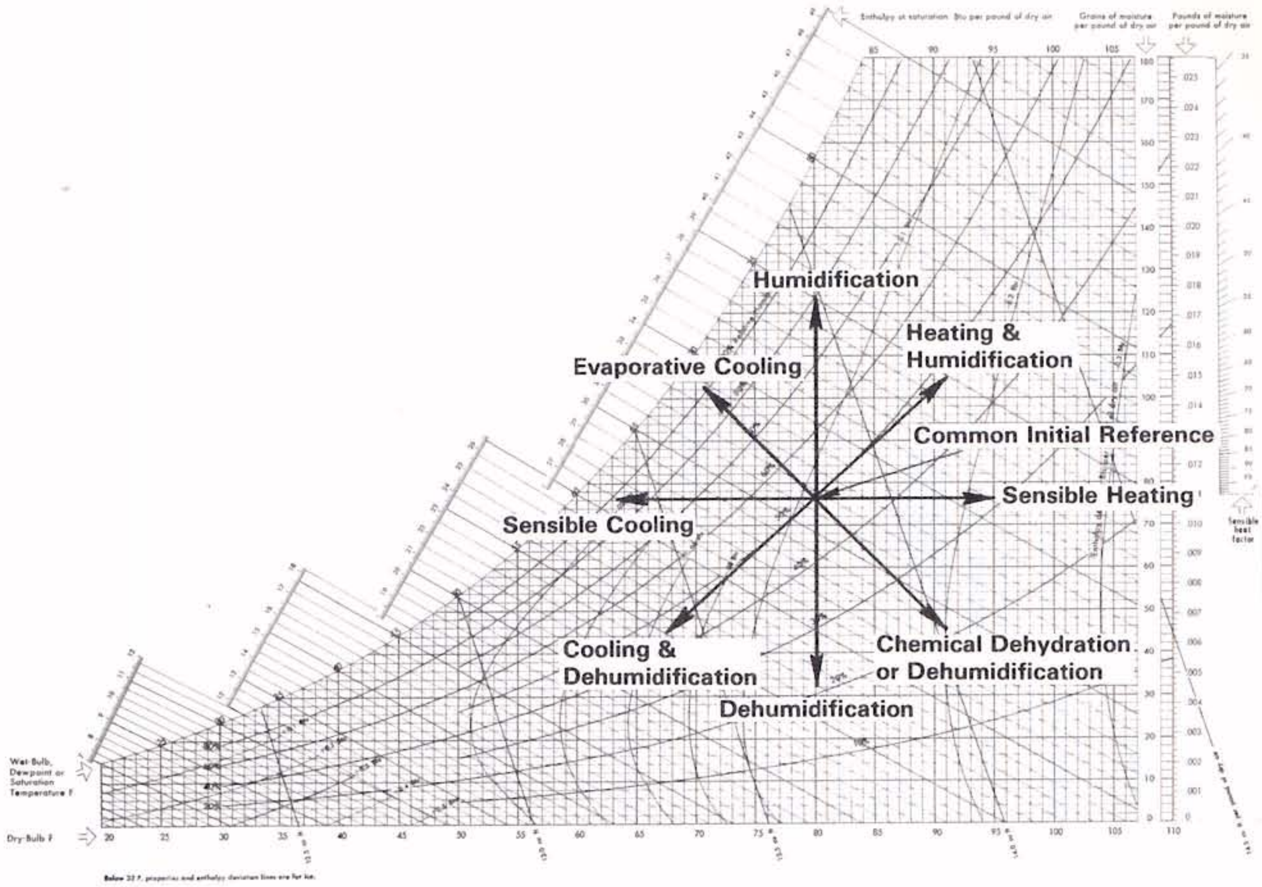


Figure 3-5 PSYCHROMETRIC CHART/PROCESSES

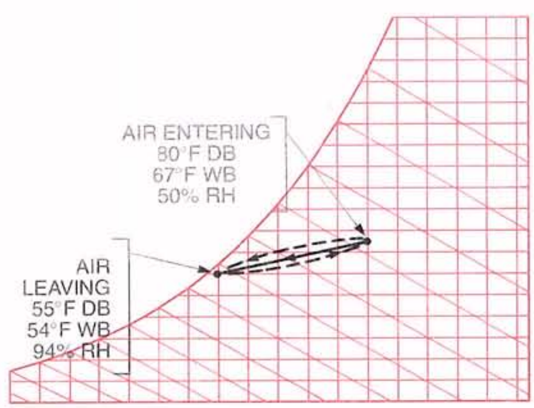


Figure 3-6 COOLING AND DEHUMIDIFYING

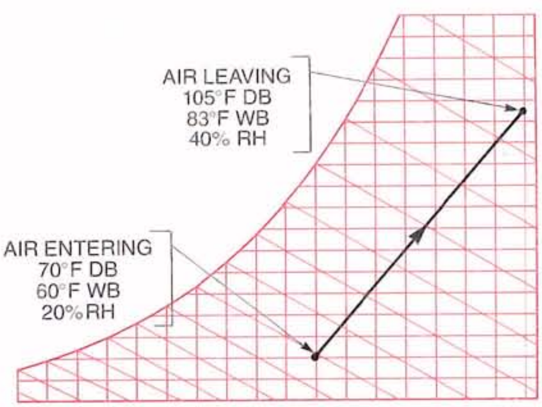


Figure 3-7 HEATING AND HUMIDIFICATION

4. Airstream Mixtures

Mixtures of two or more airstreams are a common requirement of the environmental system. The mixing of outside and return air on the entering side of the air cooling and/or heating equipment is the common method of introducing the outside ventilation air to the system. It is noted that outside air can be one of the greatest heating and cooling loads.

Assume design conditions of:

Outside air: 95°F DB, 78°F WB
Return air (room design): 75°F DB, 50% RH

Also assume that the system is designed to use 25% outside air and 75% returned room air and that the total air quantity is 1000 cfm. Begin with the premise that mixing any two quantities of a fluid originally at different temperatures will produce a mixture equal in total quantity to the sum of the two original quantities and at a resulting temperature which is between the two initial temperatures. The temperature of the mixture is directly proportional to the amounts of the two original samples being mixed and will be closest to the temperature of the largest original amount.

$$25\% \times 1000 \text{ cfm} \times 95^\circ + 75\% \times 1000 \text{ cfm} \times 75^\circ = 100\% \times 1000 \text{ cfm} \times T_{mix}$$

$$T_{mix} = \frac{23800 + 56200}{100\% \times 1000} = \frac{80000}{100\% \times 1000}$$

$$T_{mix} = 80^\circ F.$$

Another way to arrive at the same conclusion:

$$T_{mix} = 95^\circ - .75\,(95-75) = 95^\circ - 15^\circ = 80^\circ F.$$

or

$$T_{mix} = 75^\circ + .25\,(95-75) = 75^\circ + 5^\circ = 80^\circ F.$$

It is all very well to know dry bulb temperature; but the other properties of the mixture are not determined by this method, and they may be necessary. In such case the psychrometric chart provides a quick method using only a ruler and requiring no calculations.

The procedure is illustrated in Figure 3-8

First, plot the two conditions on the chart and draw a straight line between. Then measure the physical distance between the two. Divide this distance in proportion to the mixed air quantities, and plot the mixed air point so that it is *closest to the conditions of the largest original quantity in the mixture.* This is usually closest to the lowest dry bulb point since outside air quantities are usually less than 50% when they are

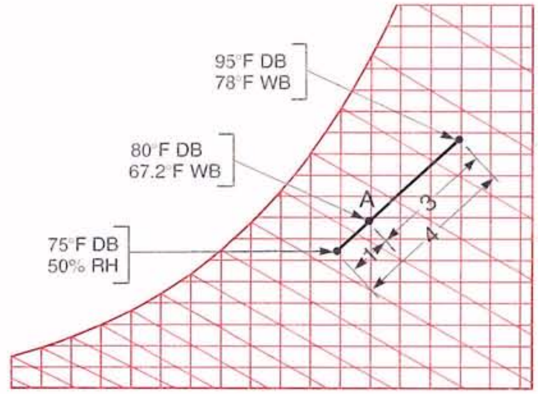


Figure 3-8 MIXING AIR STREAMS ON THE PSYCHROMETRIC CHART

not 100%. In the latter case, no mixture determination is required.

The number 4 represents the total distance in any units. The numbers 1 and 3 represent the proportions of the distance based on the proportions of the required mixed air quantity. The point A represents what the mixture conditions should be if the air quantity proportions are correct.

All of the properties of the mixture are immediately available from the psychrometric chart.

It should be noted that the use of the psychrometric chart in the design to determine the properties of the mixture does not establish the air quantity or cfm. This determination must be made independently. The designer must establish the total air quantity required from the sensible heat load and the outside air quantity from the design ventilation requirements.

5. Related Tables and Equations

Chapter 5, "Psychrometrics," and Chapter 6, "Psychrometric Tables," of the 1977 ASHRAE Handbook on Fundamentals has more detailed theory and data on this subject along with the necessary equations and psychrometric tables. Chapter 8, "Physiological Principles, Comfort and Health" includes the "comfort zone" and "effective temperature scale" superimposed on a standard psychrometric chart.

C OTHER CHARTS

There are many variations of psychrometric charts other than the one used in this text, which has a "normal" temperature range of 32°F to 107°F dry bulb. Figure 3-10 is a low temperature range chart (-40°F to 50°F DB) and Figure 3-9 is a high temperature range chart (60°F to 250°F DB). There are high altitude charts for the normal temperature range shown in Figure 3-11 (5000 ft.) and Figure 3-12 (7500 ft.). A European (metric) psychrometric chart is shown in Figure 3-13

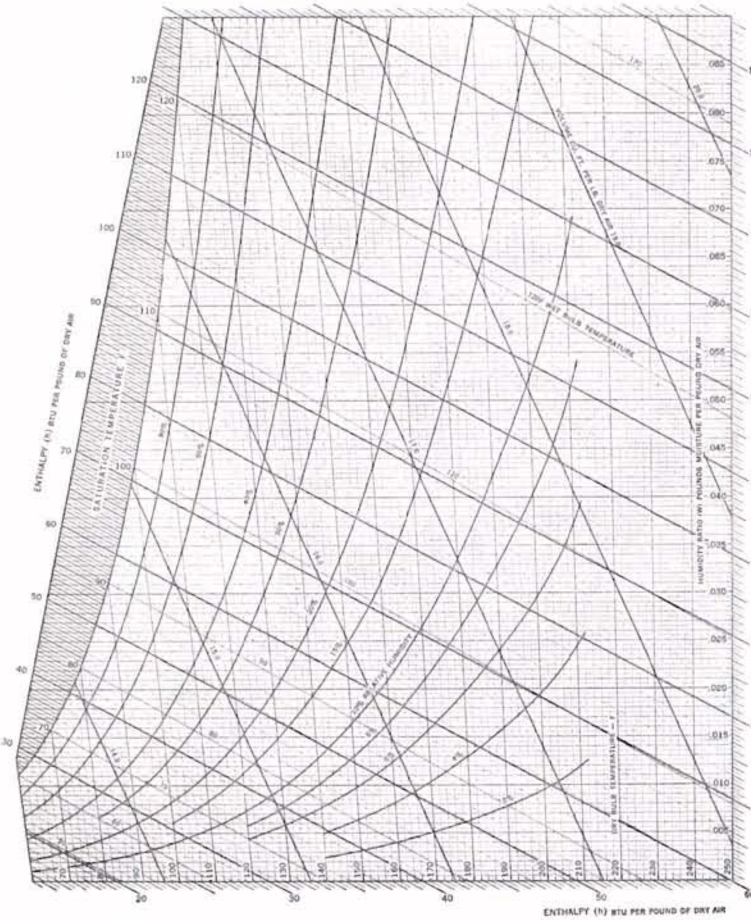


Figure 3-9 HIGH TEMPERATURE RANGE CHART (60°F to 250°F D.B.) (3)

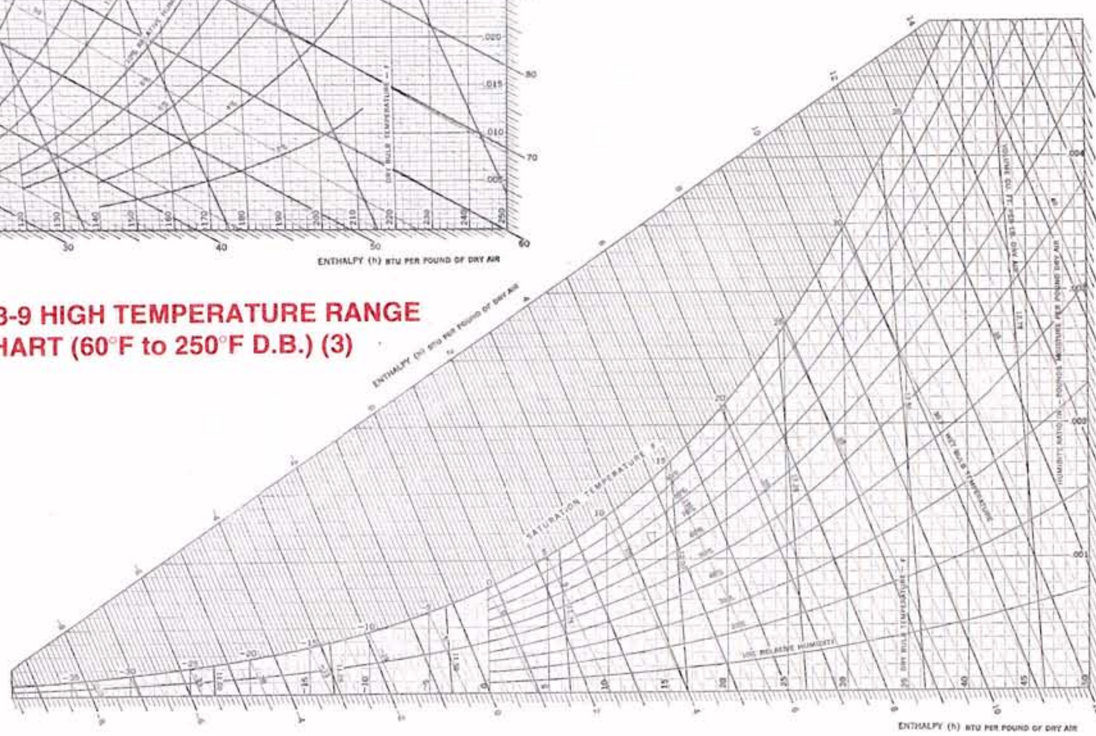


Figure 3-10 LOW TEMPERATURE RANGE CHART (−40°F to 50°F D.B.) (3)

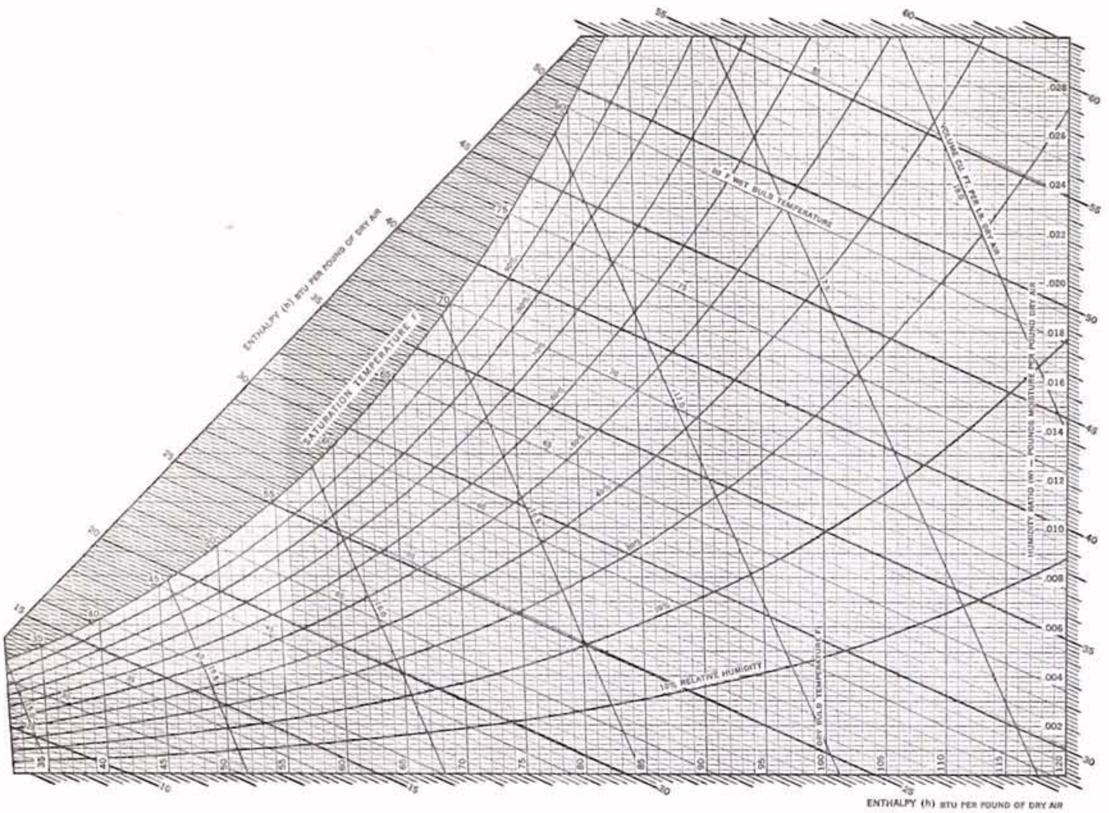


Figure 3-11
HIGH ALTITUDE CHART FOR NORMAL TEMPERATURE RANGE (5000 ft.) (3)

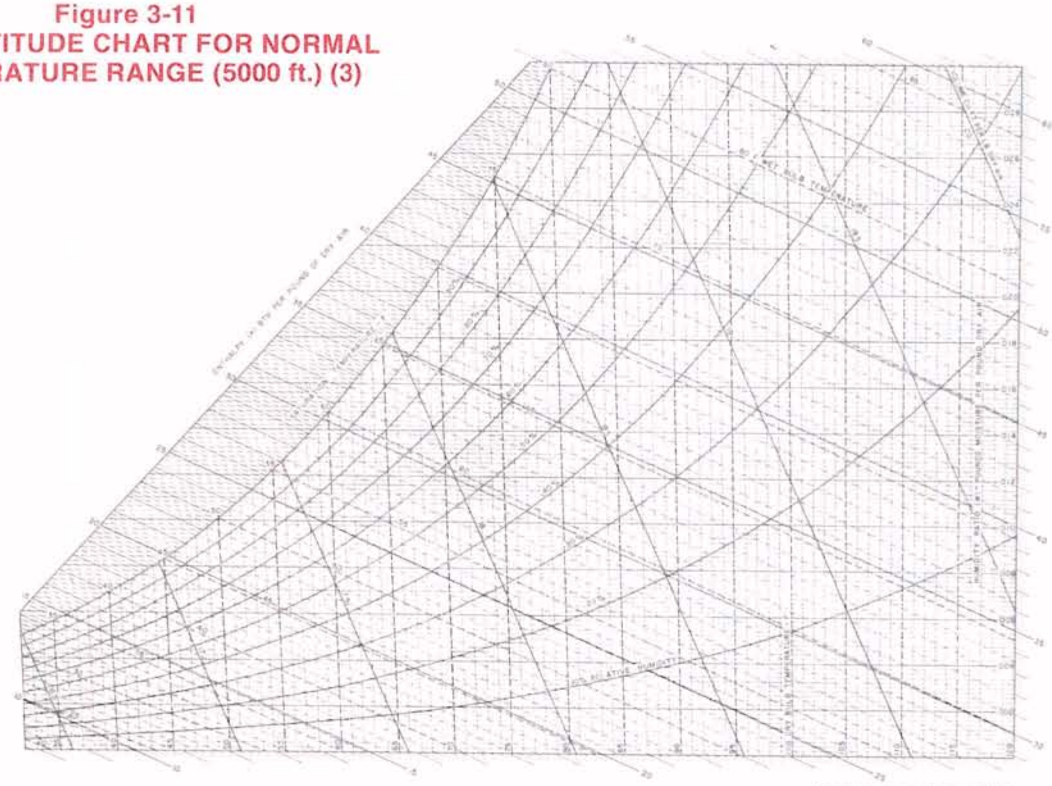
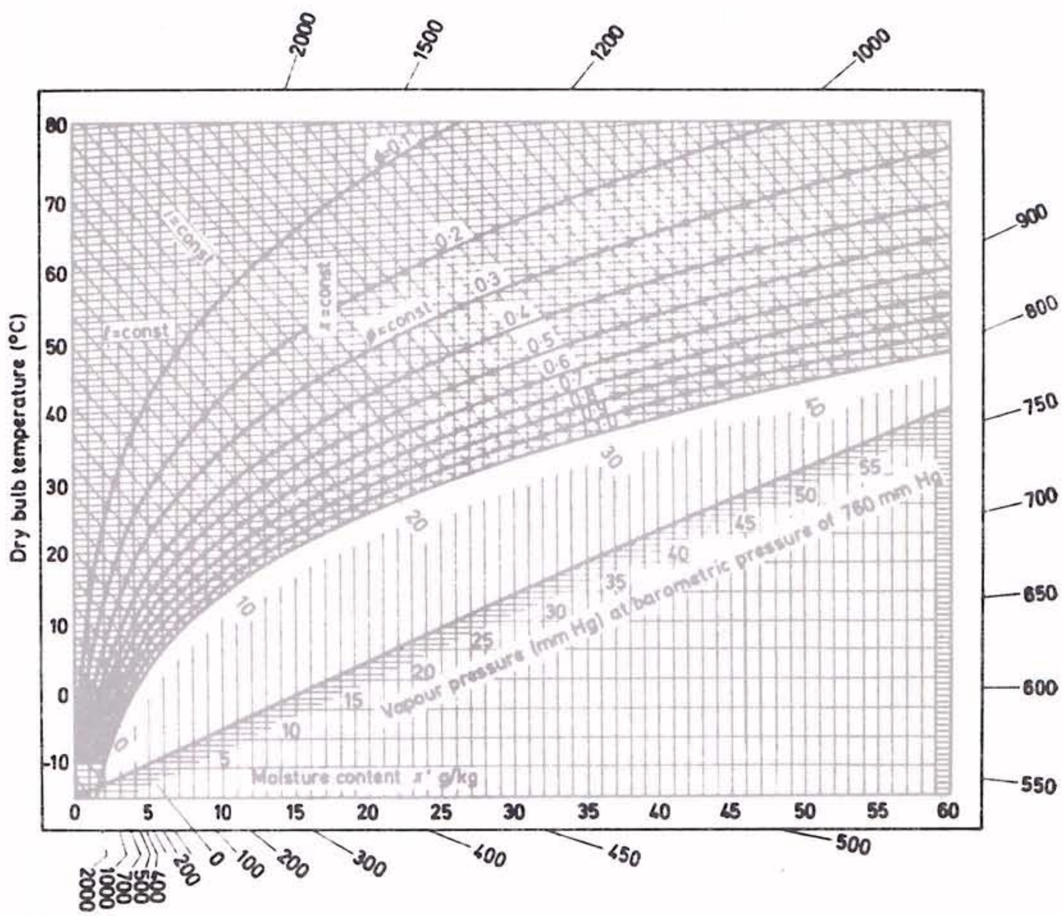


Figure 3-12 HIGH ALTITUDE CHART FOR NORMAL TEMPERATURE RANGE (7500 ft.) (3)



Mollier diagram

Figure 3-13 EUROPEAN PSYCHROMETRIC CHART

SECTION IV
THE ENERGY RECOVERY PROCESS

A FUNDAMENTALS OF HEAT RECOVERY

It is necessary to use the psychrometric chart to explain the energy recovery process. To illustrate, the enthalpy of the W = .0144 lb/lb humidity ratio moist air at 80°F (point A) is 35 Btu/lb and is 40.2 Btu/lb at 102°F (point B).

It takes 40.2 − 35 = 5.2 Btu to heat up the amount of moist air containing one pound of dry air from 80°F to 102°F. Conversely, 5.2 Btu will be released when this air is cooled from 102°F to 80°F.

It can be seen that heat is required to increase the moisture content of the air without changing the dry bulb temperature. Thus, it takes 50 − 40.2 = 9.8 Btu/lb to increase the humidity ratio from W_B = .0144 lb/lb (point B) to W_C = .0231 lb/lb (point C) without any change in dry bulb temperature. This is the amount of heat required to evaporate .0231 − .014 = .0087 pounds of water at 102°F per lb of dry air. This heat, which is used only to convert water at 102°F into vapor at 102°F, is called *latent heat*. Conversely, the same amount of heat will be released when .0087 pounds of vapor per pound of air is removed from moist air by condensation.

In most air-to-air energy recovery situations, both airstreams will have different dry bulb temperatures and different humidity ratios. Referring again to Figure 4-1, the moist air "C" has a higher D.B. temperature and higher moisture content (humidity ratio W) than air "A". The enthalpy (heat content) of air "C" is 50 Btu/lb (referenced to 0°F) and the enthalpy of air "A" is 35 Btu/lb. The total difference is 15 Btu/lb of which 5.2 Btu/lb is in the form of sensible heat and 9.8 Btu/lb in the form of latent heat. *This is then the potential maximum amount of heat which, given opportunity, may be transferred from air "C" to air "A"*

B MECHANICS OF HEAT FLOW

The basic use of energy recovery in air-to-air situations is to separate waste heat from an airstream and transfer this recovered heat to a location and airstream where it can be used in place of or as a supplement to a primarily heated airstream. It is easier to understand heat flow under these situations by considering flow of sensible heat and latent heat separately.

It was stated earlier that heat flows from a higher temperature area to a lower temperature area, with the temperature difference being the driving force. It takes additional energy in some form to bring a substance to a higher temperature level, such as compression, electrical energy, etc. In the example in Figure 4-1, heat will flow from air "C" to air "A". For the process to be effective, the resistance to the heat flow must be as low as possible. In actual recovery devices, the largest resistance is that of the boundary layers of the interface, be it a two-sided or one-sided partition.

The transfer of latent heat is associated directly with the addition (evaporation) or removal (condensation) of water vapor from moist air. Unless the temperature is maintained intentionally, evaporation will cause cooling of the ambient space, since the heat of evaporation has to come from some external source. Condensation, by releasing heat, will increase the temperature in the same manner.

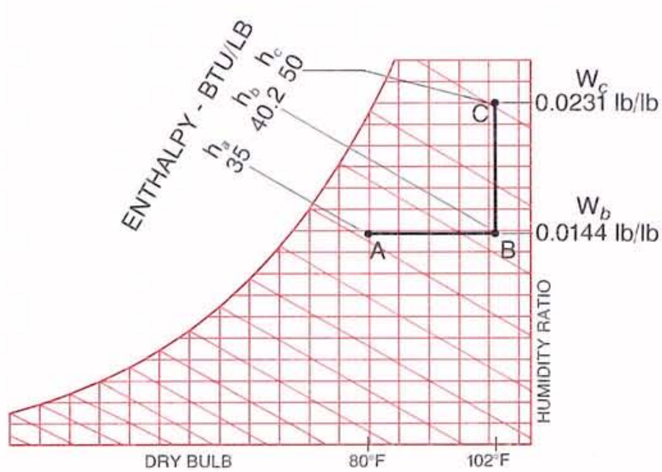


Figure 4-1 ENERGY RECOVERY PROCESS

The driving force for evaporation and condensation is the vapor pressure. If water, at 75°F with a vapor pressure of 11.7 in. w.g., is put into 75°F, 40% R H air having 4.75 in. w.g. vapor pressure, the water will evaporate until (if conditions allow, such as a vapor tight enclosed space) the saturation point is reached at which the vapor pressure of the moist air will increase to the vapor pressure of evaporation of 11.7 in. w.g., as shown in Figure 4-2

If the same 75°F water is placed into 97°F, 50% R H air which has 12.65 in. w.g. vapor pressure, the air moisture will condense indefinitely if the source of supply air is large. In an enclosed space (by ignoring released heat — for the sake of simplicity for the example), moisture would be condensed until the vapor pressure of the moist air would equal the evaporation pressure and the moist air would become drier (Figure 4-3).

Since the release of latent heat is associated with condensation, and an increase of enthalpy by addition of latent heat is associated with evaporation, the following events take place during the latent heat transfer process:

a. Latent heat flows from the higher vapor pressure area to the lower pressure area.

b. Latent heat is removed from the higher enthalpy airstream by removing moisture. This means that the condensed moisture can be held by some physical body that allows the condensate to be transferred to another area where this "heat" could be used.

c. By condensation, the airstream is dehumidified.

d. The released latent heat increases the temperature of the condensate and the condensate carrying media.

After transferral of the "removed" condensate, the following process occurs:

a. Since the vapor pressure at the new location is below the pressure of the condensate evaporation pressure, the condensate will evaporate.

b. The air at this location will be humidified (enthalpy is increased) by the amount of latent heat introduced by the addition of water vapor.

c. Converted into sensible heat, latent heat (in first half of the process) is used to evaporate the condensate. As this heat is used up, the amount of condensate diminishes and the condensate, as well as its transfer media, cools off for the next cycle.

Thus, the transfer of latent heat is possible only if moisture (condensate) is physically transferred from one airstream to another. The term "condensation" is used in its general term and includes, besides the commonly known process of converting vapor into liquid, other mechanisms, such as absorption (as for alumina, silica jel), adsorption (as for metal surface), water of crystallization, and chemical changes.

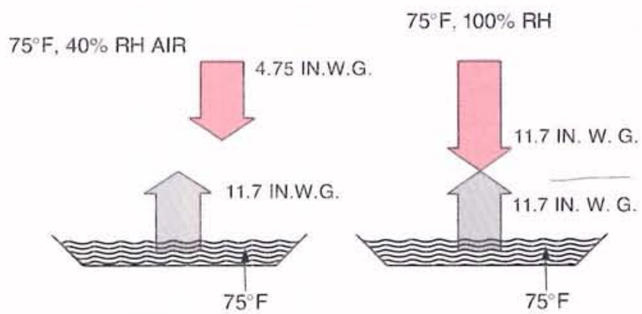


Figure 4-2 VAPOR PRESSURE EXAMPLE NO. 1

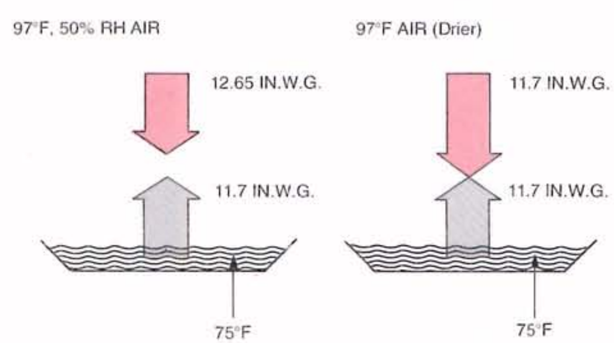


Figure 4-3 VAPOR PRESSURE EXAMPLE NO. 2

C EFFECTIVENESS

The performance of air-to-air heat exchangers is usually expressed in terms of their effectiveness in transferring: (1) sensible heat (dry bulb temperature); (2) latent heat (humidity ratio); or (3) total heat (enthalpy). The effectiveness "e" of a heat exchanger is defined as follows:

Equation 4-1

$$e = \frac{\text{actual transfer for the given device}}{\text{maximum possible transfer between the airstreams}}$$

Referring to Fig. 4-4:

$$e = \frac{W_s\,(X_1 - X_2)}{W_{min}\,(X_1 - X_3)} = \frac{W_e\,(X_4 - X_3)}{W_{min}\,(X_1 - X_3)},$$

where

e = sensible heat, latent heat, or total heat effectiveness

X = dry-bulb temperature, humidity ratio, or enthalpy at the locations indicated in Fig. 4-4.

W_s = mass flow rate of supply, pounds dry air per hour

W_e = mass flow rate of exhaust, pounds dry air per hour

W_{min} = smaller of W_s and W_e

The leaving supply air condition is then

$$X_2 = X_1 - e\,\frac{W_{min}}{W_s}\,(X_1 - X_3)$$

and the leaving exhaust air condition is

$$X_4 = X_3 + e\,\frac{W_{min}}{W_e}\,(X_1 - X_3)$$

The effectiveness of a particular air-to-air energy recovery device is a function of several variables, including the air mass flow, the airflow ratio between supply and exhaust streams, and the energy transfer characteristics of the device. Because of this, performance data must be established for each individual type of device. Manufacturers of each type may be contacted for detailed performance data.

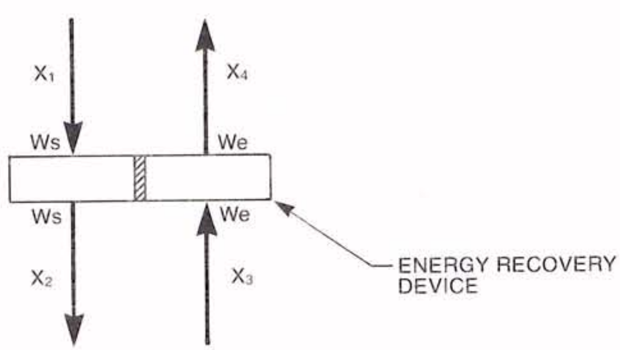


Figure 4-4 EFFECTIVENESS RATINGS (3)

1. Total Heat Devices

The ability of an energy recovery device to transfer heat from one airstream to another equal airstream is measured and compared by qualifying its efficiency. As was previously stated, the maximum potential of transferrable heat is the difference of enthalpies of both airstreams $h_c - h_a$ (Figure 4-1) which consists of the sensible enthalpy $h_b - h_a$ and latent enthalpy $h_c - h_b$:

Equation 4-2

$$h_{total} = h_{sensible} + h_{latent}$$
$$= (h_b - h_a) + (h_c - h_b)$$

The effectiveness of the energy recovery device is 100% if it transfers the maximum potential of transferrable heat between the two airstreams involved. In actual practice, the energy recovery device will transfer a portion of the maximum potential heat. This portion then determines the effectiveness of the device. If the difference of enthalpies is 15 Btu/lb and the device can transfer 12 Btu/lb, then the effectiveness of the device is said to be (using Equation 4-1):

$$e = \frac{12}{15} = 0.8;\ \text{or in percentage,}$$

$$e = \frac{12}{15} \times 100 = 80\%$$

This effectiveness value does not distinguish how much of the transferred energy or heat is sensible and how much of it is latent. Generally, all points of the two involved airstreams lay as straight lines on the psychrometric chart (as shown in Figures 3-6 and 3-7 in Section III) for devices which are capable of transferring both latent and sensible heat.

2. Sensible Heat Devices

Sensible heat energy recovery devices that do not permit transfer of moisture between airstreams are the fixed plate exchanger, thermosiphon (heat pipe) exchanger, and runaround coil exchangers. "Sensible only" rotary wheel devices may be distinguished from the above, since it is possible under certain conditions to transfer moisture between the airstreams with the wheel.

Sensible heat systems are justified primarily on their ability to transfer or exchange sensible heat. Sensible heat systems are not easily justified when the transfer of moisture between airstreams provides for more recoverable energy, and there is a need for the recovered energy. Total heat devices are more easily justified in applications in the comfort-to-comfort area where the ventilation air can represent a high latent heat gain during summer operation and/or a heating and humidification requirement during winter operation.

Since the sensible heat device does not transfer moisture, it cannot heat and humidify supply air during winter operation. However, dehumidification of the exhaust airstream in winter and supply airstream in summer is possible if the surface temperature of the heat exchanger is less than the dew point temperature of the exhaust or supply. Thus, a sensible device (other than the wheel) *can* transfer this heat of condensation to the other airstream, but cannot transfer the moisture or condensate.

The effectiveness of sensible heat recovery devices can be rated two ways:

a. The percentage ratio of the actual recovered portion of sensible heat to the total potentially transferable heat.

b. The percentage ratio of the actual recovered heat to the maximum transferable sensible heat.

The energy recovery device effectiveness numerical value, when computed with sensible heat transferred/total heat potential, will be smaller than when both values are sensible heat (sensible heat transferred/sensible heat potential).

3. Summary

Care should be taken when working with effectiveness percentages. Mathematically, sensible heat effectiveness is not the same thing as total heat effectiveness. High sensible heat effectiveness for a device does not mean it has high total heat effectiveness and vice versa.

None of the energy recovery devices are justified on effectiveness alone. Rather, they are justified on their ability to recover usable energy and be adaptable to the many other conditions required by the system installation. Generally speaking:

a. Sensible heat devices including sensible heat wheels are usually justified when most of the energy savings for an application are in the sensible heating of the supply airstream.

b. Total heat devices are usually justified when significant energy savings are obtained when it is possible to heat and humidify and/or cool and dehumidify supply airstreams.

D DEVICE CONFIGURATION

1. Counterflow (Air-to-air)

The configuration, design and performance of air-to-air energy recovery devices depends on the relationship of the two airstreams. In a counterflow pattern (Figure 4-5), the high heat content moist air (hotter of two and having a higher humidity ratio) and the low heat content air enter the device from opposite sides with air "B" giving up some energy and the other air, "A", gaining this energy, air "B" becomes colder and drier while air "A" becomes hotter and more humid in a "total heat" situation. At the exit end, each airstream will meet the opposite airstream at its original conditions. Therefore, given opportunity and time, *all* potentially transferable heat would be transferred. The theoretical maximum effectiveness of the counter-flow device is 100%.

2. Parallel Flow (Air-to-air)

In the parallel flow devices (Figure 4-6), both airstreams, upon entry, confront each other at maximum difference of temperature and humidity. Therefore, maximum heat transfer will take place at this end. With airstream "B" progressively losing heat and airstream "A" absorbing it, the differences in temperature and humidity will grow smaller, as was discussed earlier in this section. Given the opportunity and time, the two airstreams will become equal in temperature and humidity ratio. The maximum energy actually lost by airstream "B" and gained by

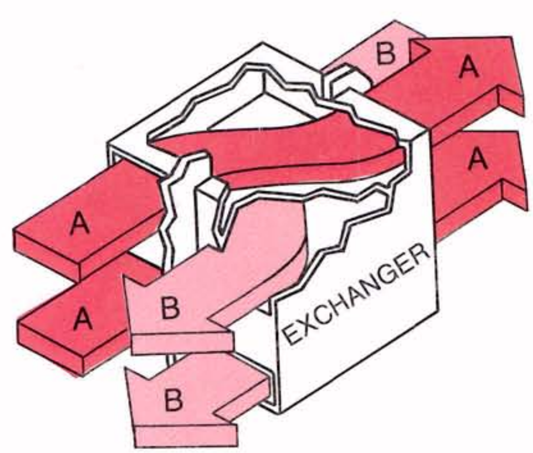


Figure 4-5 COUNTERFLOW AIRSTREAMS

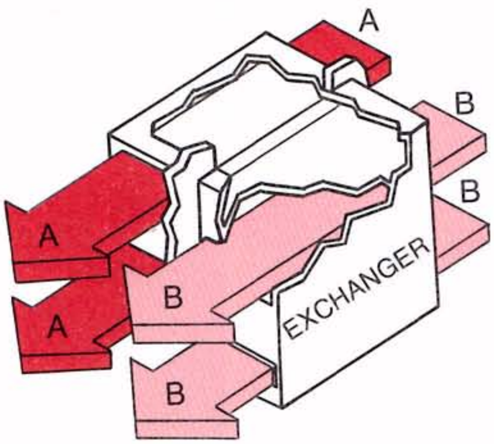


Figure 4-6 PARALLEL FLOW AIRSTREAMS

airstream "A" will be only half of the potential transferable energy. The theoretical maximum effectiveness of the parallel flow device is 50%.

3. Cross-flow (Air-to-air)

In the cross-flow configuration (Figure 4-7), the "A_1", portion of airstream "A" will cross the airflow of "C" at its maximum values and will have excellent heat transfer. The same can be said about the "C_1" portion of airstream "C" in relation to the airflow "A". But airstream "A_2" will just begin to transfer energy crossing airstream "C" after it is completely depleted and the same can be said about airstream "C_2". Still, giving opportunity and time, a 100% theoretical effectiveness is possible. Because of economic considerations, this maximum is never obtained and actual effectiveness falls short of that for counter-flow devices.

4. Closed Pipe Loop (Air-to-fluid-to-air)

With the closed pipe loop or the run-around coil exchanger, it becomes possible to separate the

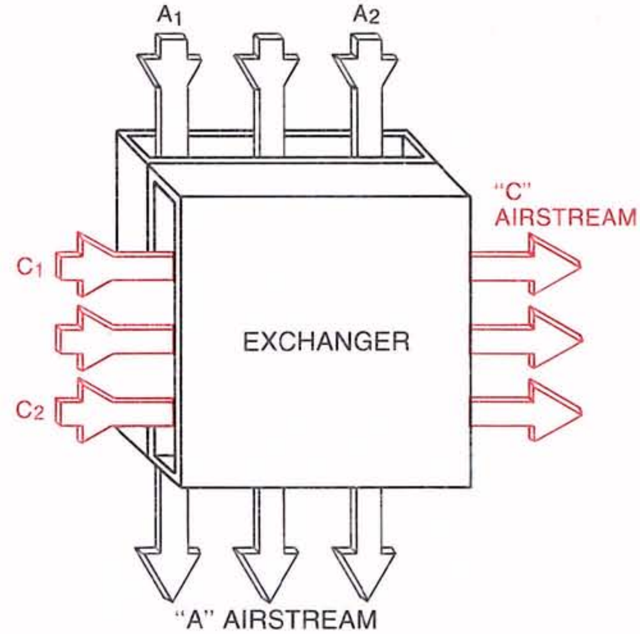


Figure 4-7 CROSS-FLOW AIRSTREAMS

airstreams for a considerable distance. This method requires *two* devices, one for transfer of energy from an airstream to the media (usually a liquid) of the closed pipe loop and a second device for transfer of the energy from the closed loop media to another airstream. If each one of the two devices had a theoretical 100% effectiveness (sensible heat only), the combined effectiveness could also be 100%. In actual practice, if individual effectivenesses are 80% (0.8) for example, the overall sensible heat effectiveness from one airstream to the other will 64% ($0.8 \times 0.8 = 0.64$).

When multiple tower exchangers are used with a sorbent liquid as the media in the circulating pipe loop, total heat (enthalpy) can be transferred. This process can also be seasonally reversible.

SECTION V
INTRODUCTION TO EQUIPMENT

Whenever air (a gas) is being exhausted from the confines of a space at a temperature different than that of the environment, it is very probable that energy is being wasted. Additional air must be brought back into the space to fill the void. When the space contains people, the replacement air should be distributed at a comfortable temperature. Often this requires the addition or removal of heat from the replacement air. When an industrial process exhaust system is used, the replacement gas (or air) is usually heated to the process temperature.

Air is exhausted from a space because of contaminants. The most obvious of these contaminants is carbon dioxide and moisture. Both are the result of the presence of people and combustion. There are many other contaminants that exist within an exhaust gas, the most common of which are dust, grease, sulfur dioxide and chlorine gas.

It is desirable to remove energy from exhaust gases and transfer this energy to the incoming airstream while simultaneously keeping the pollutants in the exhaust system. Therefore, it is necessary to place some form of barrier, one resistant to pollutants but not resistant to energy transfer, between the exhaust and replacement gas streams. To make this energy transfer, a large amount of surface area (approximately 0.5 square feet of material per cfm of gas) must be used. To obtain that amount of surface in a reasonable volume, the gases must exit and return through a series of relatively small passageways. This results in gas pressure losses and the possibility of contaminant entrapment within the confines of the energy recovery device. Many different heat exchanger designs and system have emerged in an attempt to satisfy all of the many different requirements.

Any energy recovery device installed in an exhaust duct may be susceptible to fouling to some degree, depending upon the nature of the contaminants in the exhaust stream, and upon the dewpoint of the flowing gas. If allowed to continue, fouling will decrease the heat transfer performance. Also, air flow may be impeded which, in the case of paint ovens, could result in increasing solvent concentrations above safe and acceptable levels. Therefore, filters are recommended unless other economical methods of cleaning the recovery device are used.

If filters are used, the filter placement should be upstream of the energy recovery unit and is recommended for both air streams. While low to medium efficiency filters will generally be sufficient, high efficiency filtering may be recommended downstream of the unit when outdoor air is being supplied to a critical area (such as an industrial white room, or hospital operating rooms). Manometers and alarm contacts may be furnished for dirty filter indication.

A TRANSFER EQUIPMENT IN GENERAL

The basic function of energy transfer equipment and materials is the transfer of energy from point-to-point in the system, while, at the same time, providing the means to isolate cycle from cycle and system from system. The differences among the various types of equipment, materials, and systems is determined by the functions required: the need to heat, cool, humidify, dehumidify, and/or isolate. The differences also become evident by the need for heat absorption or rejection in the basic physical properties of temperature and pressure. Because of the limited range of temperature and pressure variations in the natural environment and in the environmental system, the requirements of the energy transfer components are also limited. The limitations can be an advantage, establishing predictable ranges in the selection of equipment and components. Even the penalties imposed upon the liquid hydronic portions of systems by pressures resulting from elevation or height may be turned to advantage by locating equipment at or toward the top of the system, therefore making use of the predictably lower pressure found there. Rated working pressures of equipment, physical structure and weight, and overall system costs might be reduced.

However, because of the inability to provide a perfect energy transfer device, or stated differently, because of the inefficiencies of heat transfer by equipment and materials, some difficulties arise in the selection and operation of the equipment. The freezing point of water at 32°F also imposes some difficulties, however, proper design, selection and application of equipment and systems make it possible to overcome these deficiencies, sometimes to our advantage.

A point often overlooked, which can be an advantage or disadvantage, is the limitation imposed by the standard pieces of equipment and materials available from manufacturers. Special equipment may be designed for any application or capacity, but often at an unreasonable additional cost. For the most part, manufacturers provide a range of selections which cover all of the required conditions, but judgement must be used to select the most practical available size and type for the energy transfer requirements.

B TYPES OF EQUIPMENT

Air-to-air heat exchangers, for the purpose of this discussion, will consist of rotary wheels, plate types, tower, and finned tube systems. It should be recognized that there are other energy recovery systems, such as waste heat boilers, heat pumps, etc. However, it is essential that each air-to-air system and its equipment be thoroughly understood in order to assure the correct and most economical system selection for the job.

1. Rotary Wheel

Rotary wheel exchange devices are porous discs fabricated from some material having a high heat retention capacity which rotate through two side-by-side ducts. The axis of the disc is located parallel to and on the partition between the two ducts. As the disc is slowly rotated, sensible heat (and often moisture containing latent heat) is transferred to the disc by the hot air from the first airstream and then from the disc to the second cooler airstream. The pores of heat wheels carry a small amount of the exhaust stream into the supply side of the exchanger. Should this result in an undesirable contamination, a purge section may be added.

Rotary wheel exchangers are commercially available with a number of different types of discs. The original disc was a metal frame packed with a core of knitted stainless steel or aluminum wire mesh, not unlike common kitchen "steel wool." Other wheels are made of corrugated metal assembled to form many

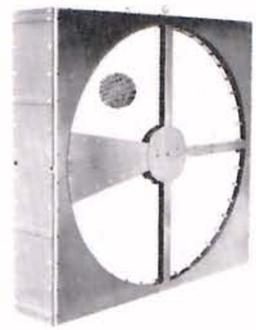


Figure 5-1 ROTARY WHEEL EXCHANGER (5)

parallel passages (laminar wheels); ceramic laminar wheels for higher temperatures; and metal or composition wheels coated, impregnated or treated with a hygroscopic material so that latent heat may be transferred.

2. Fixed Plate

These passive systems consist of alternate channels of exhaust and supply gas streams separated by a thin wall of metal or other materials. Sensible heat in the exhaust gas or airstream is transferred to the supply airstream. The gas or airstreams may pass in parallel flow, counter flow, or cross-flow arrangements, all without contamination. Most units are available in the modular design concept. Typical configurations for low to medium temperature exchangers are shown in Figures 5-2 and 5-3. High temperature exchangers are shown in Figures 5-4 and 5-5.

Figure 5-2 FIXED PLATE COUNTER FLOW EXCHANGER (7)

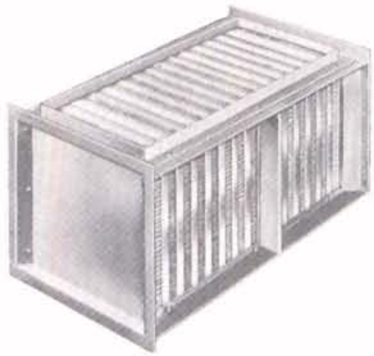


Figure 5-3 CROSS-FLOW PLATE EXCHANGER (14)

Figure 5-4 HIGH TEMPERATURE PLATE EXCHANGER (10)

Figure 5-5 HIGH TEMPERATURE TUBE EXCHANGER (14)

3. Multiple Tower

This energy recovery system permits wide separation of the supply and exhaust duct systems. The recovery system consists of separate multiple contactor towers for each of the exhaust and supply duct systems involved, with a sorbent liquid continuously being circulated between them. This liquid acts as the vehicle for transporting both sensible and latent heat, and is usually a halogen salt solution, such as lithium chloride and water.

4. Finned Tube Exchangers

Energy exchange systems using finned tubes are divided into two groups. The first type are conventional or modified water type coils that can be used for run-around systems, as illustrated in Figure 5-7 and 5-8. A coil in one or more duct systems transfers heat via some suitable liquid to other coils in another duct system(s). Ethylene glycol is commonly used as the transfer fluid, but there are many other commercial thermal fluids on the market as shown in Table 12-1 of Section XII— "Thermal Transfer Fluids." This energy exchange system permits wide separation of two or more duct systems.

The second type is the relatively new thermosiphon or heat pipe exchanger. This device, while appearing as one large coil, is face divided to provide exhaust and supply sides connected by sealed tubes containing an appropriate heat transfer fluid. The efficiency of this fluid depends on the anticipated temperatures involved, as evaporation of the fluid takes place on the hot side and condensation takes place on the cold side. The condensed liquid is usually returned to the hot side by a capillary action or gravity.

Figure 5-6 TOWER EXCHANGER (12)

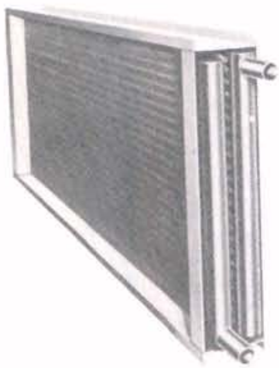


Figure 5-7 CONVENTIONAL WATER COIL (AIR-TO-LIQUID-TO-AIR) EXCHANGER (14)

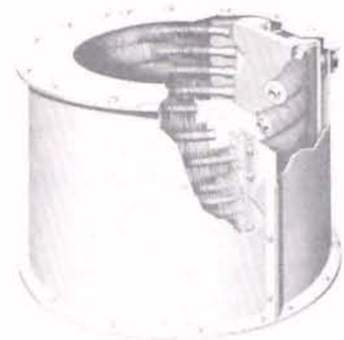


Figure 5-8 HIGH TEMPERATURE (GAS-TO-LIQUID-TO-GAS) EXCHANGER (14)

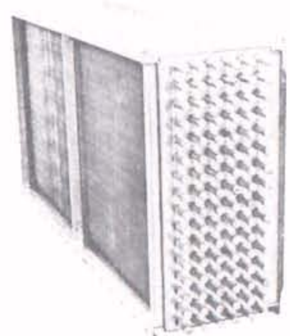


Figure 5-9 HEAT PIPE EXCHANGER (17)

C CHARACTERISTICS

1. Energy Transfer Process

In any heat exchange, energy may be transferred by one or more of the usual physical methods of convection, conductance, absorption, or radiation. Radiation is considered insignificant within the limits of this discussion. The rate of energy transfer is dependent on the velocities of the gasses or fluids, the temperature differences, and the cleanliness and configurations of the transfer surfaces.

In all heat exchangers, except the run-around types, the absorbed heat moves by conductance from the hotter gas stream through a metal, ceramic or hygroscopic material, into the cooler airstream. In the finned-tube units, the heat must be absorbed by a fluid which, in turn, returns this energy to a second surface for conductance into the cooler gas stream. In the multiple tower system, heat is transferred by absorption into the fluid and then desorption from the fluid.

2. Temperature Ranges

In selecting a suitable energy recovery device, the range of the two involved gas stream temperatures must be known. Naturally, temperatures below the freezing point and above the boiling point of water merit special consideration. Condensation, the spread of temperature ranges in the gas streams, and the differences between these variable ranges must also be considered.

Table 5-1 indicates a reasonable series of temperature ranges and other data for preliminary energy recovery system selection.

3. Transfer Efficiencies

In selecting a recovery system, the contractor must be aware of corrosion, contamination, pollutants, noxious gases, odors, etc., that might affect the heat transfer or be detrimental to the desired system. These conditions may affect heat transfer by particles blocking one of the equipment airstreams, by equipment failure due to total corrosion or breakdown of the materials of construction, or by contamination of one of the airstreams or transfer fluids. Filters are recommended for both airstreams unless other methods are provided, such as washdown equipment or access for manual cleaning.

Type Exchanger	HVAC Applications	Exhaust to 250°F Supply -20° to 110°F	Exhaust 250°-800°F Supply -40° to 500°F	Exhaust 800° to 1200°F Supply -40° to 500°F	Exhaust 1200° to 1800°F	Sensible Heat Recovery	Total Heat Recovery	Remote Airstreams	Possible Cross Contamination	Corrosive Gases Permitted (special construction)	Air-to-Air	Air-to-Liquid-to-Air
Rotary Wheel												
Metallic	X	X				X			(1)	(3)	X	
Hygroscopic	X	X				X	X		(1)		X	
Ceramic			X	X	X	X			(1)	(3)	X	
Fixed Plate	X	X	X	X	X	X				X	X	
Heat Pipe	X	X	X	X		X				X	X	
Run-around Coil	X	X	X			X		X		X		X
Multiple Tower	X	X				X	X	X	(2)			X
Thermosiphon	X	X	X	X		X		X		X		X
Dry Air Cooler	X					X	(4)				X	

Notes:
(1) Can be limited to less than 1% with a purge section.
(2) Gaseous cross-contamination can occur.
(3) When used in process-to-process applications where cross-contamination can be tolerated.
(4) Has evaporative cooling capabilities.

Table 5-1 CHARACTERISTICS OF ENERGY RECOVERY DEVICES

Most manufacturers will rate the recovery efficiency on new equipment. The designer must assure the owner or operator that the energy recovery unit be maintained in an efficient operating status, and he must provide the necessary filtration and/or access to do so.

4. Orientation

It was stated very briefly that the heat wheel, heat pipe, and plate-type exchangers require the two gas streams be in close proximity. Conversely, the multiple tower, coil run-around, and thermosiphon systems permit wide separation of the gas streams. In fact, these latter systems permit the transfer fluid to pass through several exhaust and/or supply gas streams in completing a single loop.

For maximum efficiency, particularly in lower temperature applications in the three former systems, the counter flow arrangement is used. In the plate-type unit, the cross-flow arrangement lends itself to more convenient duct connections.

Should condensation be anticipated on the exhaust side, the orientation and/or velocity of the gas flow should be taken into consideration to keep the surfaces from being plugged and to permit removal of the condensed liquid. Any condensation that would form a tar or solid buildup on the surfaces must be avoided, unless adequate and economical cleaning methods are employed.

D RATINGS

One of the basic problems in the air-to-air energy recovery equipment industry is the lack of rating standards. Some of the equipment, such as water coils, which have been used for years in the HVAC industry, have time-proved capacity ratings. However, the balance of the equipment is rated by the manufacturer and the degree of test and research data varies from theoretically-calculated data to extensive laboratory testing under varying conditions. These factors are always present in a young, growing industry. The air conditioning industry had the same problems (and sometimes over-rated equipment) until uniform standards were established. Other organizations are also concerned and are taking steps to alleviate these problems.

Industry practice on the rating of equipment airflow has been to size the energy recovery equipment based on the average of the two air volumes being handled when they are unequal, or to size the equipment based on the larger of the two airstreams. The total combined volume of the two airstreams should never be used.

1. American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE)

Under the guidance of the ASHRAE Air-to-Air Energy Recovery Committee T.C. 5.5, the ASHRAE Standard 84P— "Method of Testing for Rating Air-to-Air Heat Exchangers" —has been developed. The standard prescribes methods for rating the heat transfer capacities, the thermal effectiveness, and supply air contamination of air-to-air heat exchangers.

The purposes of Standard 84P are to:

a. Establish a uniform method of testing for obtaining rating data.

b. Specify types of test equipment for performing such tests.

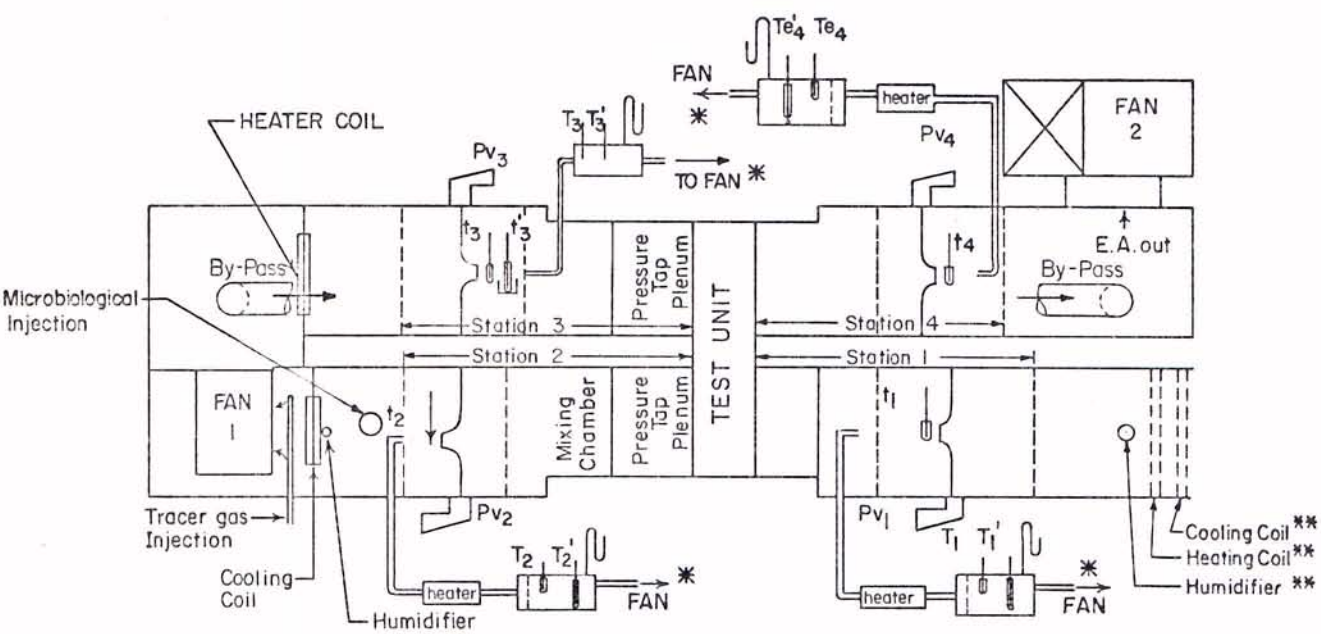
c. Specify data required and calculations to be used.

The ASHRAE T.C. 5.5 Committee has several research programs in various stages of completion on air-to-air energy recovery equipment or systems:

a. RP-140 "Development of Performance Characteristics of Two-phase Thermosiphon Loops."

b. RP-173 "Method of Evaluation Based on the Operating Characteristics of the Following Air-to-air Heat Exchangers: Plate, Heat Pipe, Run-around. . ."

c. RP-188 "Multiple-Tube Evaporation and Condenser Thermosiphon Loops."



✱ Return to chamber sampled
✱✱ Optional

Figure 5-10 SCHEMATIC DIAGRAM OF TYPICAL AIR TO AIR HEAT EXCHANGER TEST EQUIPMENT (ASHRAE STANDARD 84P) (3)

ASHRAE RP-133—"Standard Procedure for Determining Performance of Air-to-air Heat Recovery Equipment"—was completed in 1976 and a proposed research project would evaluate the effect of condensible contaminants to air-to-air exchangers.

2. Air Conditioning and Refrigeration Institute (ARI)

A Heat and Energy Recovery Systems Section, of ARI, was recently formed. They are developing an air-to-air heat exchanger standard for possible use in a certification program. Under ARI certification programs, each equipment manufacturer who participates must:

a. Certify that his equipment complies with all requirements of the standard.

b. Submit required data for verification by ARI that the proposed "standard equipment ratings" conform with the test results and within the tolerances prescribed.

c. Participate in the pre-certification check test program.

d. Agree to the requirements of the random test program at an independent testing laboratory under contract to ARI.

It will probably be several years before the ASHRAE standards referenced by ARI will be completed. Consequently, the ARI program will not become an actuality for several years.

SECTION VI
ROTARY WHEEL EXCHANGERS

A DESCRIPTION

The rotary air-to-air heat exchanger, popularly called a heat wheel, is a revolving cylinder made up of axially air-permeable media having a large internal surface for intimate contact with the air passing through it. The air duct connections (Figure 6-1) are arranged so that each of the airstreams flow axially through approximately one-half of the wheel in a counterflow pattern. The porous media that is heated from the warm duct airstream, rotates into the cold duct airstream where it releases the newly obtained energy. Some wheels are treated with a hygroscopic material to permit them to also transfer moisture from one airstream to another. Such wheels are called "total heat" wheels. Non-hygroscopic wheels are called "sensible heat" wheels.

1. Temperature Range

Heat wheels are available for temperatures from −70°F to +1500°F. By application and design, they can be broadly divided into two categories — "comfort" applications and "process" applications.

a. *Comfort.* Comfort applications are designed for transfer of either sensible heat or total heat at temperatures from −70°F to about +200°F maximum. The media is made of metal, mineral or man-made materials.

b. *Process.* Low temperature process applications are for sensible heat transfer at temperatures up to approximately 400°F. The media is usually made of aluminum. However, stainless steel, monel and other corrosion-resistant materials may be used for corrosive atmospheres. Medium temperature process applications are used for temperatures up to 800°F. The media is made of stainless steel, monel or other similar metals and ceramic materials. High temperature process applications can be used up to 1500°F. The media is made of either ceramic materials or high temperature-resistant steels.

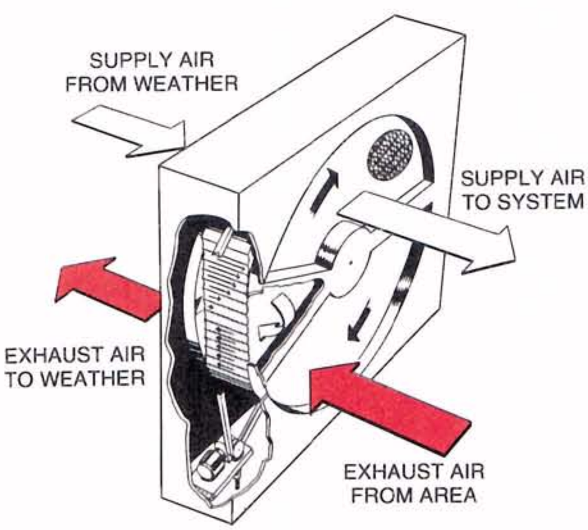


Figure 6-1 ROTARY WHEEL EXCHANGER (5)

2. Construction

For optimal performance, comfort wheels are constructed from 6 inches to 12 inches thick. High temperature wheels may be as thin as 4 inches or as thick as 3 feet, depending on the allowable pressure drop of each of the two airstreams. Shipping and installation problems limit comfort wheels to about 15 feet in diameter. Process wheels up to 50 feet in diameter are available. The larger diameter wheels are assembled at the job location. It is preferred to have the wheel mounted in the vertical plane (horizontal airflow). Horizontal mounting of the wheel requires special design considerations and can be more expensive to install.

The materials from which the heat transfer section, rotating cylinder, and parts of the supporting frame of the heat wheel are constructed, are influenced by the contaminants, the dew point, and temperature of the exhaust and supply airstreams. For normal comfort ventilating systems, the frame materials are structural steel and the rotating cylinder is constructed of either metallic or non-metallic materials. The heat transfer media is made of metal, inorganic fibers or man-made materials, and has either random or directionally-oriented air passages.

Random oriented media is made of foam, felt or wire mesh, with the wire mesh being the most common material for non-hygroscopic wheels. Aluminum wire is used for normal comfort ventilation systems, and stainless steel or monel wire mesh is used for high temperature and corrosive atmosphere applications.

Directionally oriented media consists of small tubular air passages in the axial direction of the wheel. The triangular shape is preferred because it gives the largest exposed to air surface per cross-sectional area, and it is mechanically strong and easily produced by interleaving flat and corrugated ribbons of material. The internal media surface exposed to the airflow is about 100 to 1000 sq. ft. per sq. ft. of face area, depending on the design and expected effectiveness.

Wheels are driven by a constant speed or a variable speed drive connected to the shaft with a sprocket wheel and chain, or by a belt running on the periphery of the wheel; by direct drive of the shaft; and by friction drive on the periphery.

Contact or non-contact (labyrinth) seals are used along the periphery and radially to minimize the leakage of air from one airstream to another and to the surrounding space. Only non-contact seals are used for medium and high temperature wheels.

B PRINCIPLES OF OPERATION

1. The Transfer Process

Figure 6-2 illustrates the typical processes that take place in the two airstreams when a sensible heat wheel is used to transfer heat from one airstream to the other. In this case, it is presumed that no condensation occurs within the wheel when in the warm duct (i.e., the cold air temperature is above the dew point temperature of the warm air). The cool air is heated from 1 to 2 while the warm air is cooled from 3 to 4.

Figure 6-3 illustrates the corresponding processes for the case where condensation does occur in the warm airstream followed by evaporation into the cool airstream.

Figure 6-4 illustrates the typical total heat wheel processes on a psychrometric chart for the special case where the effectiveness for the total heat and for the latent heat are equal.

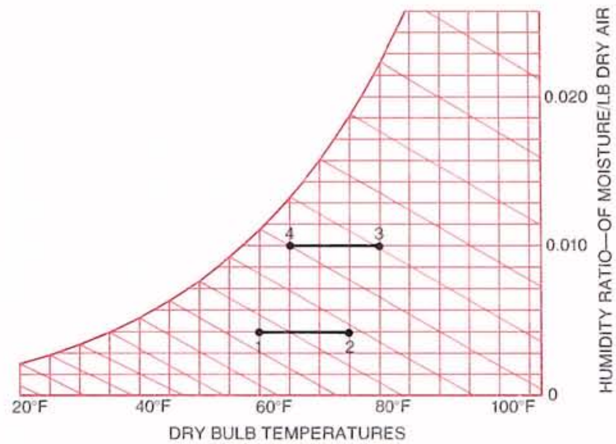


Figure 6-2 SENSIBLE HEAT TRANSFER PROCESS

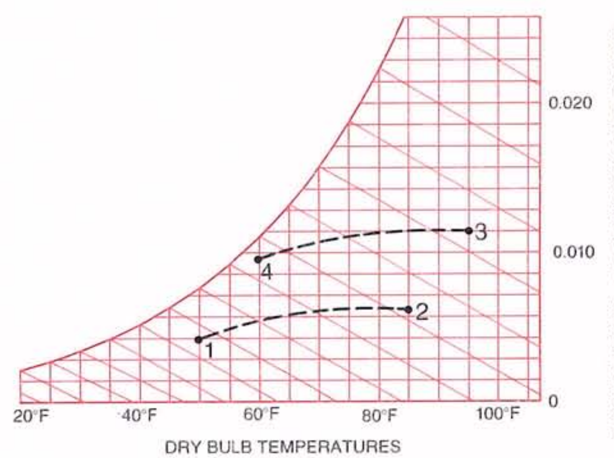


Figure 6-3 SENSIBLE HEAT TRANSFER UNIT TRANSFERRING LATENT HEAT

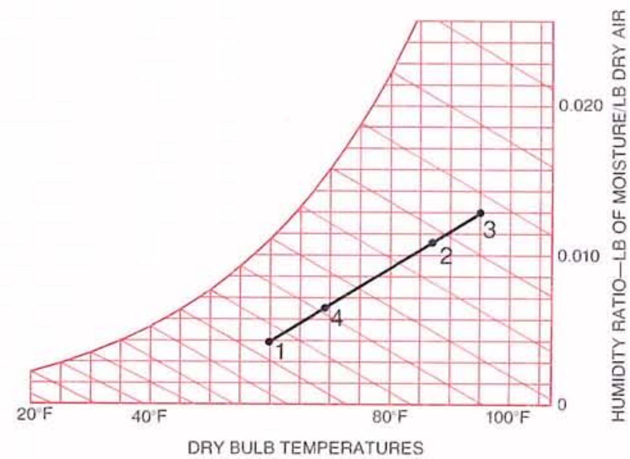


Figure 6-4 TOTAL HEAT TRANSFER PROCESS

2. Cross Contamination

The media of the rotary exchanger occupies about 5% of the wheel volume for metal mesh and 8% to 25% for oriented media. When the media is in the (sometimes contaminated) exhaust airstream, the remaining space surrounding the media is filled with exhaust air. During rotation, most of the entrained exhaust air being enclosed by partitions and air seals of the cut-off zone is carried to the supply air side. The supply air, by flowing in the opposite direction to the exhaust air, pushes the entrained exhaust air out of the media, where it is mixed with the supply air. The entrained exhaust air carryover is directly proportional to the speed of rotation and volume of the media. For example, a 78'' diameter, 8'' thick wheel rotating at 8 rpm, handling 10,000 cfm on each side, can entrain up to 1.8% of the exhaust air (180 cfm).

It is possible to reduce this carryover (if it is not allowed as in hospitals, animal shelters, and laboratories), by purging the entrained exhaust air with supply air into a segment of the wheel adjacent to the cut-off zone and returning the expelled exhaust air to the exhaust airstream (see Figure 6-5). For the purge to function properly, the exhaust airstream of the wheel should always be at a lower pressure than the supply airstream.

Tests show that a properly operating purge section will reduce carryover to less than 1% by volume for random oriented media, and to less than 0.2% for exchangers with directionally oriented media. Since purge volumes are small, (5% or less of the supply air volume), they are usually ignored in performance calculations, but must be considered in the selection of the system fans.

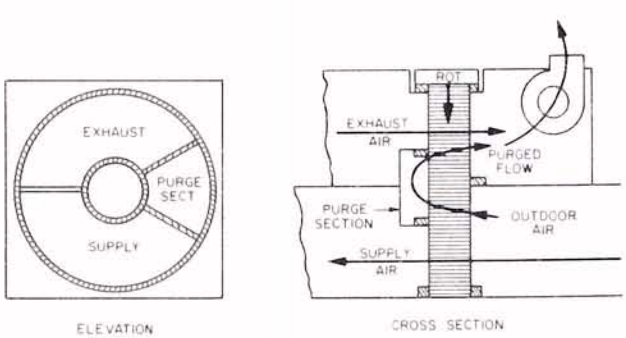


Figure 6-5 ROTARY WHEEL PURGE SECTION (DIRECTIONALLY ORIENTED MEDIA) (3)

C PERFORMANCE

1. Pressure Drops

The performance of an air-to-air rotary heat exchanger may be characterized by the pressure drops incurring in the airstreams and by the effectiveness of the unit. Both are functions of the face velocity which is based on the area exposed to the airflow. The total face area of the wheel is divided in half, with each half being used by one of the two airstreams.

Practical face velocities for comfort applications are in the 400 to 900 fpm range. Low velocities normally result in lower pressure drops, higher effectiveness and lower operating costs, but require larger size units and more space for installation. Higher airstream velocities offer lower initial costs and smaller space requirements, but operating costs are usually higher as a result of the increased pressure drops.

Figure 6-6 shows typical pressure drop versus face velocity curves for both directionally and random oriented media. The pressure drop is proportional to the velocity in the power of 1.7 for knitted metal mesh, 1.5 for felt-type material, and 1.05 to 1.1 for directionally oriented media.

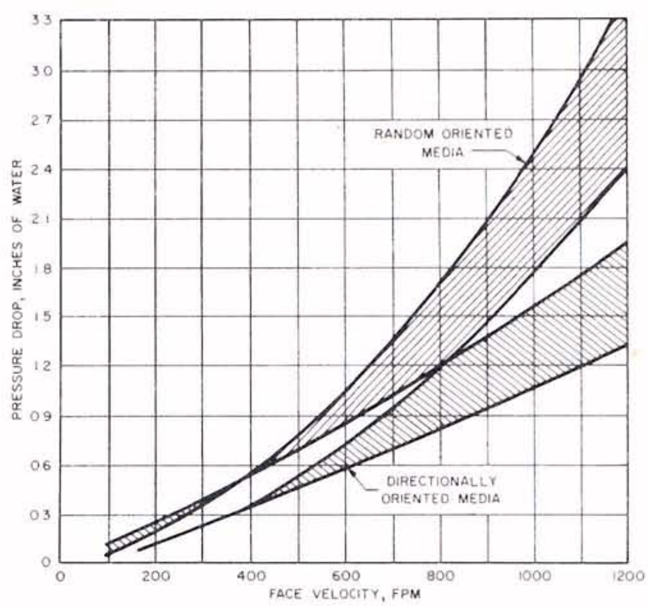


Figure 6-6 REPRESENTATIVE PRESSURE DROPS FOR ROTARY WHEEL EXCHANGERS (3)

2. Effectiveness

For equal supply and exhaust airflows, the effectiveness (e_T) of the total heat transfer wheels is the ratio of the actual enthalpy recovered to the maximum possible enthalpy rise. Typical effectiveness of the total heat transfer wheel at a 600 fpm face velocity is 75% to 85%. Figure 6-7 shows a typical performance of a hygroscopic wheel for summer cooling conditions at e_T = 75% and winter heating conditions at e_T = 80%.

In contrast, the effectiveness (e_S) of the sensible heat transfer wheels (non-hygroscopic) can be defined as the ratio of the enthalpy rise to the maximum possible sensible enthalpy rise, rather than the total enthalpy difference. Effectiveness values given for sensible heat transfer wheels cannot be compared directly with the effectiveness values given for the total heat transfer wheels. Typically, the sensible heat effectiveness (e_S) of these wheels, without condensation, is about 80%, while the latent heat effectiveness (e_T) may vary due to the vapor content of the two airstreams and proximity to the saturation. When the dry bulb temperature of one airstream is substantially below the dew point temperature of the other airstream with a higher vapor content, the media may cool below the dew point of the latter and condensation may occur. The heat recovery under these conditions will be increased and, while the sensible effectiveness will be reduced, the total effectiveness will be higher than if the condensation had not occured. An example of the performance of the sensible heat transfer wheel is plotted in Figure 6-8. For summer cooling, sensible effectiveness is 80% and total effectiveness is 29%. For winter heating, the sensible (e_S) is 62%, but the total (e_T) is 54%.

For other than 600 fpm face velocities, the effectiveness changes almost linearily, as shown by the typical curve in Figure 6-9. It increases a few points with the decrease of velocity to 400 fpm, and decreases a few points with the increase of velocity to 900 fpm. As a function of rotational speed, effectiveness increases rapidly with increased speed from just above no rotation and levels off at 4 to 20 fpm depending on design.

The effect of unequal airflows upon the effectiveness of the air-to-air heat transfer wheel is shown in Figure 6-10 as a function of the ration Q max./Q min. For calculations, the effectiveness for equal flows is found by using the smaller air volume (Q min). Then it is corrected for unequal flow by plotting a curve parallel to those shown.

Example

Q max/Q min = 5250 cfm/4200 cfm = 1.25

The effectiveness (e) is found for equal airflows to be 79%. As plotted in the chart of Figure 6-10, e = 85% for unequal airflows.

D APPLICATION

The rotary wheel exchanger usually can be economically installed where the two involved airstreams are

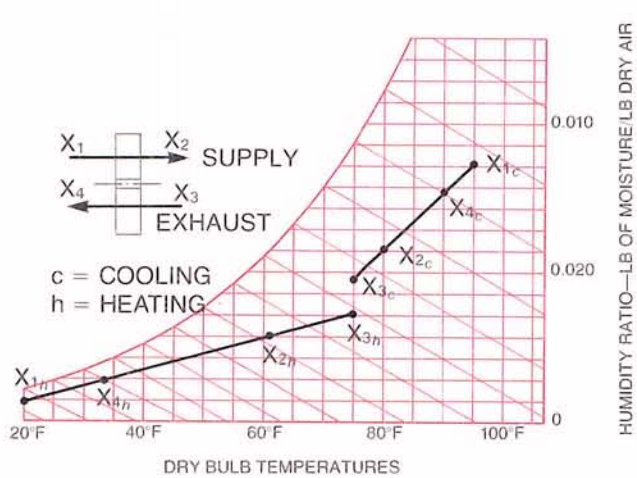


Figure 6-7 TYPICAL PERFORMANCE OF A HYGROSCOPIC ROTARY WHEEL (3)

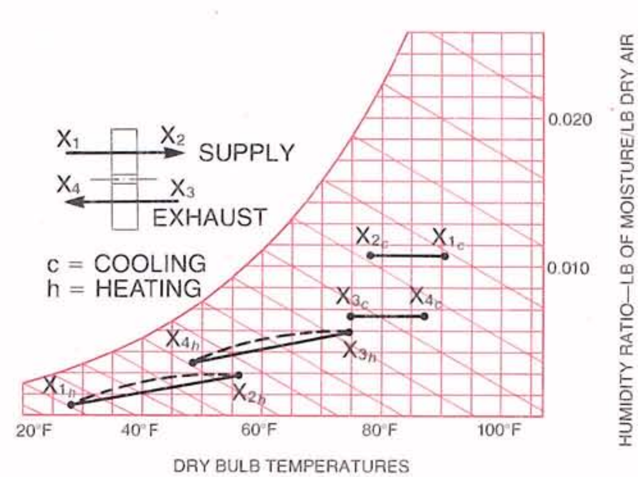


Figure 6-8 PERFORMANCE OF A NON HYGROSCOPIC ROTARY WHEEL (3)

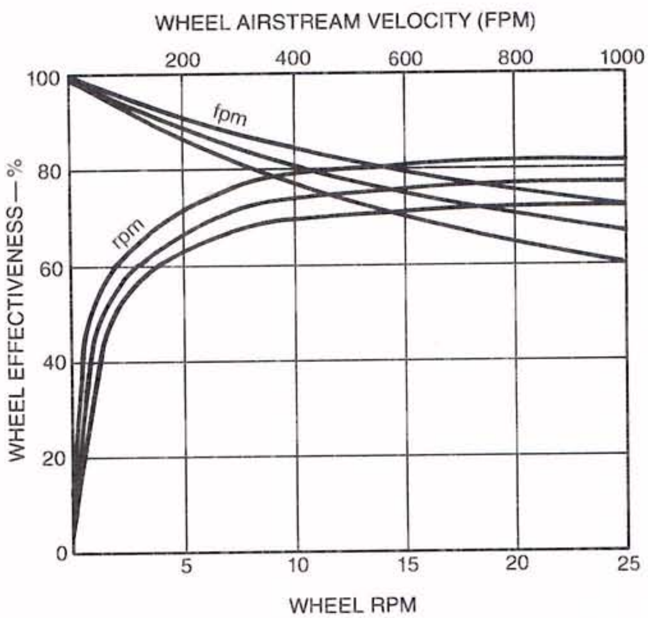


Figure 6-9 TYPICAL ROTARY WHEEL EFFECTIVENESS

in close proximity. These airstreams should be arranged to flow through the heat wheel in opposite directions. It is this counterflow arrangement that creates the high effectiveness of 65% to 85% at equal airflow rates. With parallel airflows, the effectiveness is reduced to approximately one-half of that obtained by counterflow arrangement.

There are conditions when full heat recovery is not required. Most ventilations systems are engineered to maintain specific indoor conditions at extreme outside design conditions. During most of the actual operating hours, outside conditions are less severe and over-recovery of heat may occur if the full capacity of heat transfer is utilized. Also, certain applications having high internal sensible heat gains require supply air temperatures significantly below the exhaust temperatures. To prevent overheating, it may be necessary to reduce the quantity of heat recovered by the heat exchanger. This is typically found in the cold duct of a multi-zone or double-duct system, or in the air supply for a reheat or induction system. To permit efficient temperature control of the airstream supplied to the hot duct, a separate cold-duct supply fan with a heat exchanger bypass may be desirable. Additional methods of controlling capacities will be covered later in this section.

1. Filters

Filters for both airstreams are recommended. If outdoor air is to be used for applications requiring special cleanliness, such as white rooms and hospital operating areas, additional filters of proper efficiency should be installed downstream from the unit in the supply airstream. If wet snow can present a problem at the air inlet, the designer should take steps to assure continuous airflow.

Fans should be located in the energy recovery system so that the supply airstream of the exchanger is at a greater pressure than the exhaust airstream. This is done to assure proper operation of the purge system and prevent contamination of the supply air by seal leakages. The fan capacities must be adequate to accommodate the extra air quantities needed for the purge section and cross-leakage.

2. Condensation and Freeze-up

In rotary, non-hygroscopic exchanger, condensation on media will occur when some portion of the media is cooled to a temperature below the dew point of the warm airstream. The media temperature on the cool side of the wheel is estimated to be the average of the cold air entering and the warm air leaving the media, as shown in Figure 6-11. (The average temperature of the media $= \dfrac{21°F + 40°F}{2} = 30.5°F$.) Small amounts of condesation may be evaporated into the other airstream. For greater amounts of condensation, provision must be made to collect the excess condensate which drips from the wheel.

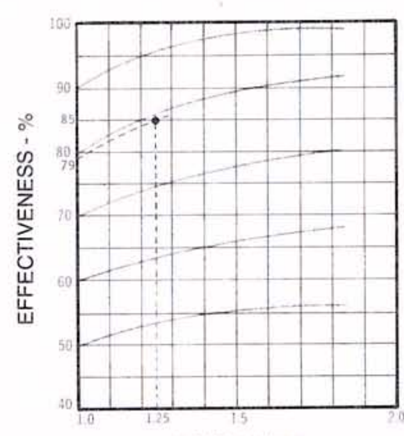


Figure 6-10 EFFECTIVENESS OF UNEQUAL FLOWS

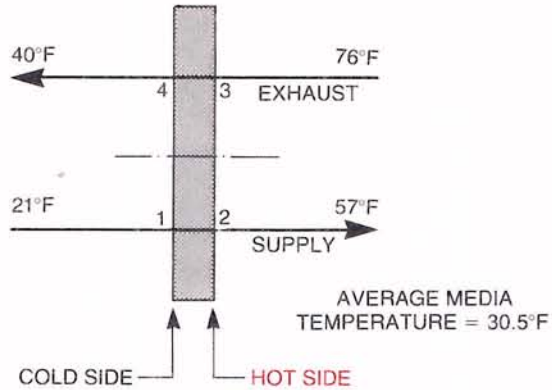


Figure 6-11 CONDENSATION AND FROSTING

Frosting (sublimation of water vapor) and icing (freezing of the sub-cooled condensate) will occur on non-hygroscopic wheels when the temperature of any portion of the media is below the frost point of the exhaust air. The rate at which the frost will accumulate depends on the temperature of the supply air, humidity ratio of the exhaust air, effectiveness of the unit, and the duration of these conditions. When the exhaust air contains a high moisture content, the moisture does not give up its latent heat of condensation, but rotates into the incoming cold airstream where it can quickly freeze. Generally, frost will form first on the discharge face of the exhaust airstream and will increase in thickness and depth of penetration into the media with the duration and intensity of the frosting conditions. In extreme cases, the airflow can be interrupted completely.

For wheels with hygroscopic media, frosting and freezing can occur at lower temperatures. The designer must analyze how the desicant of the selected rotary wheel exchanger could react under job conditions from data furnished by the wheel manufacturer.

E CAPACITY CONTROLS

Frosting of rotary wheel exchangers can be controlled by preheating the cold supply air, although preheat should be kept to a minimum. It is sufficient to bring the temperature of the media slightly above 32°F. It could be more economical to use an alternate control method which consists of reducing the effectiveness of the heat transfer either by use of a speed control or by bypassing a portion of the cold supply air. This method requires larger heating coils to compensate for the loss of recovered heat. Frost control by either method is actuated by an increase in the pressure drop across the wheel.

For conditions requiring variable heat recovery, the effectiveness of the heat wheel is modulated by varying the rotational speed from zero effectiveness at standstill to optimum energy recovery at speeds of about 6 rpm to 12 rpm for the hygroscopic and about 8 rpm to 20 rpm for the non-hygroscopic wheels.

The most frequently used variable speed drives are: the silicon rectifier, solid state controlled variable speed DC motors, and constant speed AC motors with the solid state controlled hysteresis coupling. Both types are controlled by a modulating sensor at the desired location (Figure 6-12). Care should be exercised in the selection of control devices where speeds close to zero are expected frequently, in order to maintain the control point without overshooting, cycling or hunting.

An alternate method of controlling the amount of recovered energy is by means of face and by-pass dampers (Figure 6-13) or with by-pass dampers only (Figure 6-14).

A deadband control may be necessary when no energy recovery is needed, as in the case when the outdoor air temperature is greater than the required supply air temperature, but is below the exhaust air temperature. The purpose of the deadband control is to completely unload the exchanger, either by stopping the wheel or by stopping the wheel and opening a bypass to reduce the pressure drop.

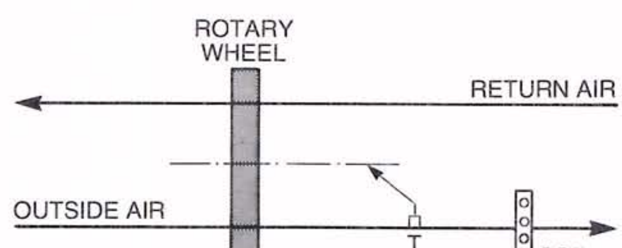


Figure 6-12 VARIABLE SPEED DRIVE

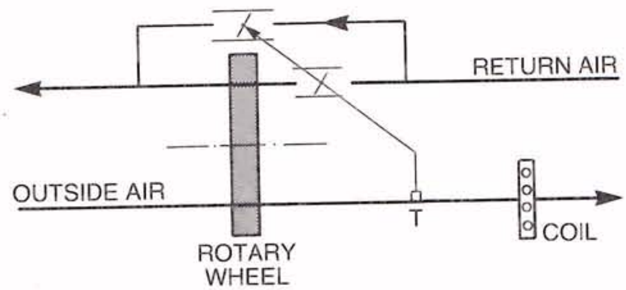


Figure 6-13 FACE AND BY-PASS DAMPERS

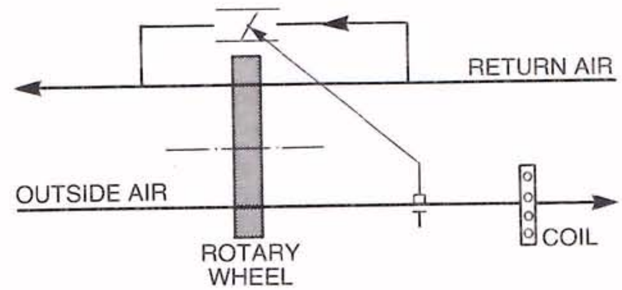


Figure 6-14 BY-PASS DAMPER

F MAINTENANCE

Air-to-air rotary wheel exchangers normally operate with a minimum amount of required maintenance. However, the following maintenance items must be considered for optimum performance: cleaning of the wheel media surface, inspection of seals, and regular servicing of drives, motors, and speed reducers.

In most installations, exchanger wheel media needs periodic cleaning. It is very important to note that cleaning methods suitable for one type of media are not necessarily suitable for other types. Non-hygroscopic wheels can be cleaned using steam, compressed air, hot water, or a suitable solvent. Hygroscopic wheels may be cleaned by vacuuming the face of the wheel and/or using dry compressed air to clean out the passages. Unless otherwise recommended by the manufacturer, hygroscopic wheels must not be washed or subjected to high humidities, as the desicant material might dissolve and be lost from the media.

The air seals on all types of wheels should be periododically inspected for proper clearance and signs of wear.

Motor and reduction gear maintenance should be in accordance with the manufacturer's recommendations. Special attention must be given to speed control motors that have commutators and brushes. These require more frequent inspection and maintenance than induction motors, as the brushes require periodic replacement and the commutator has to be turned. Wheels with belt or chain drives should be inspected for proper tension, following the manufacturer's recommendations. As "permanently" lubricated and sealed ball bearings are usually used in rotary wheel construction, servicing of bearings is normally not required, but regular inspections for wear are still part of a good maintenance program.

SECTION VII
FIXED PLATE EXCHANGERS

A DESCRIPTION

Fixed surface plate type energy recovery exchangers can be broadly classified into two categories: the pure-plate heat exchanger, consisting of only primary heat transfer surface, and the plate-fin heat exchanger, made up of alternate layers of separate plates and interconnecting fins. The pure plate exchanger is usually of a counterflow design, whereas the basic plate-fin exchanger is a cross-flow design with combinations sometimes being arranged to approach the counterflow configuration. Counterflow airstreams provide the greatest temperature difference for maximum heat transfer, but a cross flow design can sometimes give more convenient airstream duct connections. The typical plate exchanger transfers sensible heat only, except when the temperature of one airstream is sufficiently low to cause condensing of moist air in the other airstream. With the addition of moisture to the exhaust airstream, certain fixed plate exchangers can also cool the outside airstream (see Section XI — Dry Air Cooler Exchangers").

1. Temperature Range

The temperature range of the fixed plate exchanger is the broadest of all air-to-air energy recovery equipment. With the low point being established at any usable temperature below 32°F, the maximum operating temperature separates the equipment into three categories:

a. low temperature exchangers to 400°F

b. medium temperature exchangers to 900°F

c. high temperature exchangers to 1600°F

Because the high temperature fixed plate exchanger can operate in a wide range of temperatures, expansion and contraction can become a problem. Expansion joints in the connecting ductwork is a workable solution. Suspension type mounting is also used to help minimize the restraint of the systems.

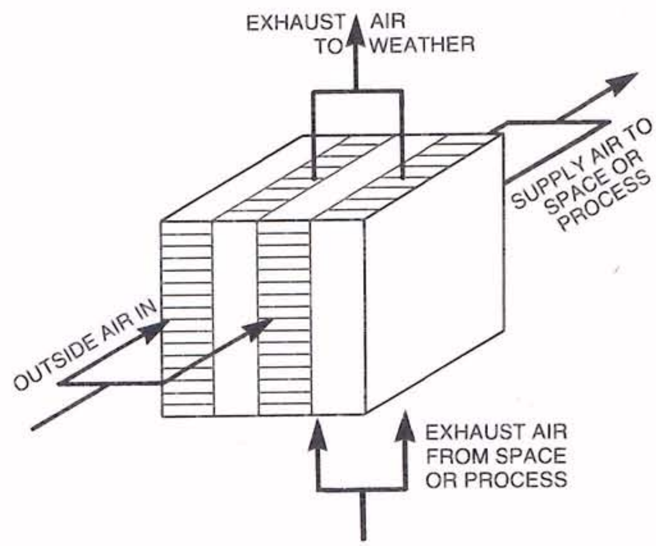


Figure 7-1 PLATE TYPE HEAT EXCHANGER

2. Construction

Plate exchangers are of many proprietary designs with weights, sizes and flow patterns being highly dependent upon the manufacturer. Many are built in modules, where by using various combinations of height and width, an exchanger can be provided for almost any airflow and pressure drop requirement. In general, the size can be estimated to be between 7 and 19 square feet per 1000 cfm with the pressure drop ranging between 0.5 and 1.50 inches of water. Plate or plate-fin exchangers are available with spacing from 2 to 11 plates or fins per inch. The most popular material of construction is aluminum, not because of its high thermal conductivity, but because of its corrosion resistance and relative ease of fabrication. The material's conductivity has a negligible effect on the effectiveness of fixed plate exchangers. Therefore fixed plate exchangers are fabricated from various steel alloys for applications up to 1600°F with excellent heat exchange between airstreams.

Capacities of the plate exchanger modules range from 1,000 to 10,000 cfm and can be arranged in the field for installations up to 100,000 cfm. The modules are welded or assembled with seals to eliminate cross contamination or to keep it at a minimum. Some units are offered that allow easy access to the heat transfer surfaces for cleaning after installation. Automatic wash systems for remote cleaning are also available.

B PRINCIPLES OF OPERATION

1. The Transfer Process

The process that takes place between the two airstreams is a sensible heat transfer, as illustrated in Figure 7-2. Under certain conditions, condensation can occur.

Moisture in exhaust air represents latent energy. As can be witnessed by examining the psychrometric chart in Figure 7-3, once the dew point of the exhaust gas is reached, the temperature of the exhaust decreases at a rate much lower than the rate of the incoming air for equal airflow. Consequently, the incoming air can be at a temperature considerably below 32°F before frosting will occur on the exhaust side.

2. Cross Contamination

One of the advantages of plate exchangers is that it is a static device which can be built with little or no leakage between airstreams, depending on temperatures, the type of construction, and the assembly method when modules are used. Some of the exchangers are built with a continuous sheet of metal formed to produce two divided air passages, some with sheets formed and welded, and others with tubes rolled into crown sheets similar to the construction of fire-tube steam boilers.

Modular units are assembled and sealed using gasketing, molded seals, welded seams, etc., depending on the application and the temperatures of the airstreams. When *zero* leakage between airstreams is required, the effects of corrosion, deterioration and routine maintenance on the type of sealing used in the manufacturing and/or assembly of the fixed plate recovery units should be evaluated.

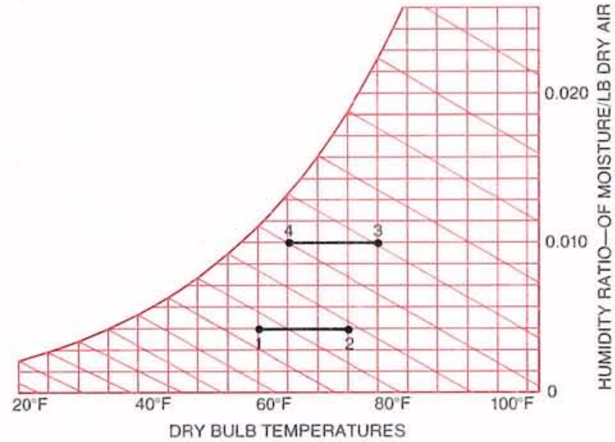


Figure 7-2 SENSIBLE HEAT TRANSFER PROCESS

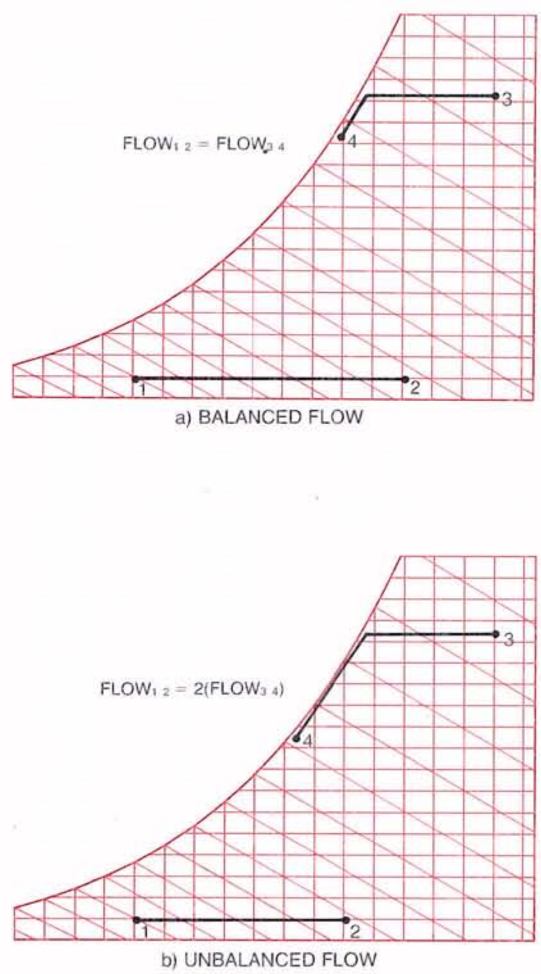


Figure 7-3 CONDENSING WITHIN A SENSIBLE PLATE HEAT EXCHANGER (3)

C PERFORMANCE

1. Pressure Drops

To attempt to develop a general range of airstream velocities and pressure drops for all types of fixed plate energy exchangers would be futile, as applications and construction methods are widely varied. Figures 7-4 and 7-5 are typical performance curves used for two different types of plate exchangers. Attention is again called to the large changes in air density, especially at the higher temperatures found in process energy recovery systems. A typical correction factor chart for pressure losses versus average temperature is shown in Figure 7-6.

2. Effectiveness

The fixed plate heat exchanger can achieve a high sensible heat effectiveness because it is usually a counterflow device with only primary surface area and is not dependent on a secondary fin surface to enhance its heat transfer capabilities. Energy recovery effectiveness (sensible) for the pure plate exchanger is in the range of 50% to 80%; and the effectiveness (sensible) is in the range of 40% to 60% for the plate-fin type. Effectiveness curves of a typical fixed plate exchanger are shown in Figure 7-7.

A high percentage of effectiveness is not always desirable in high temperature energy recovery systems. Exhaust airstream temperatures approaching 1600°F can heat the other airstream up to 1000°F when there is a 63% effectiveness. Unless all of this high temperature air can be utilized, recovery equipment with a lower effectiveness could supply the needed recovered heat to meet the job requirements. The less efficient devices could also have other advantages such as larger air passages (less pressure drop), easier maintenance, etc.

D APPLICATION

The fixed plate exchanger is similar to the rotary wheel exchanger in that it can be the most economically installed where the two airstreams are in close

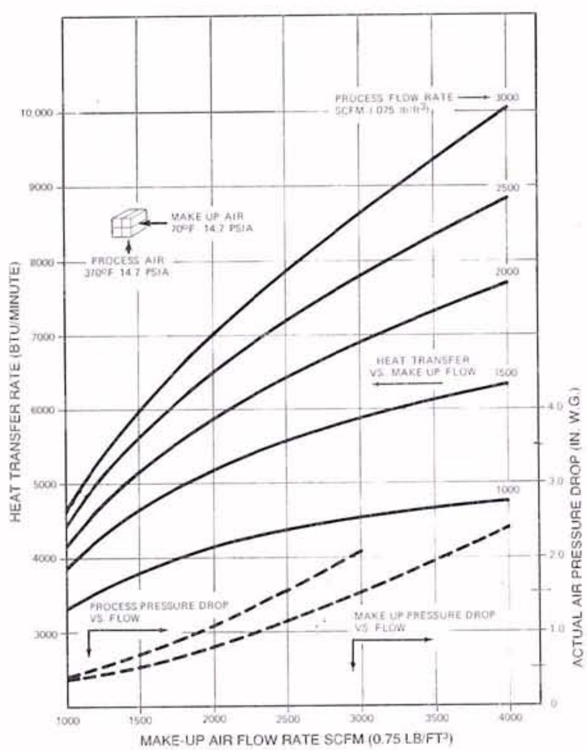


Figure 7-4 HEAT TRANSFER AND PRESSURE DROP VS. MAKE-UP AIR FLOW (PLATE-FIN RECOVERY UNIT) (10)

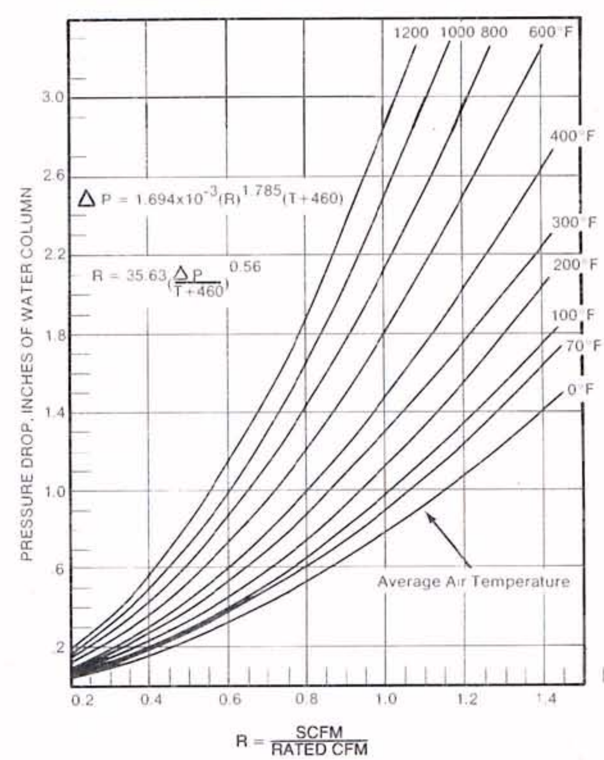


Figure 7-5 PRESSURE DROP VS. FLOW (FIXED PLATE RECOVERY UNIT) (7)

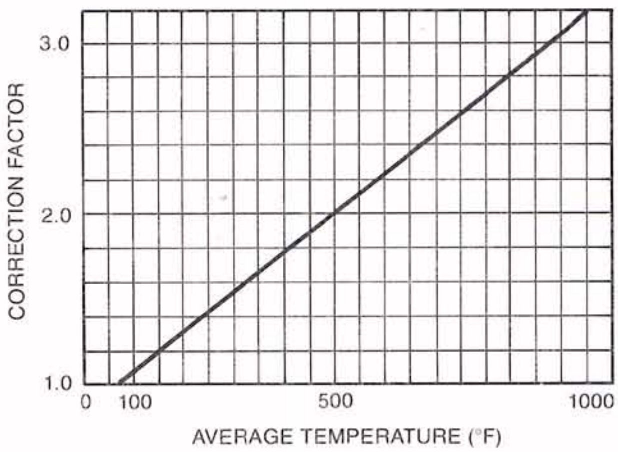


Figure 7-6 PRESSURE LOSS TEMPERATURE CORRECTION FACTOR

proximity. When an energy recovery system is installed in a new building, the duct systems, including the intake and exhaust openings, can be designed for economical installation and operational costs. In retrofit work each installation is peculiar to the application, and will vary from one site to another. All costs should be carefully calculated before selecting the type of recovery unit and system to be used.

1. Filters

If the exhaust airstream (as in a process application) is laden with heavy particulates, contaminants, etc., filters should be installed to reduce fouling. However, filters are not normally required for most plate exchangers. In fact, with some types of applications, it can be more economical to clean the exchangers than to maintain the filters. Some types of plate-fin exchangers usually require filters in both airstreams. Static pressure monitoring devices could be used for observation of the system static pressure drop to indicate when cleaning of the filters and/or recovery equipment is required.

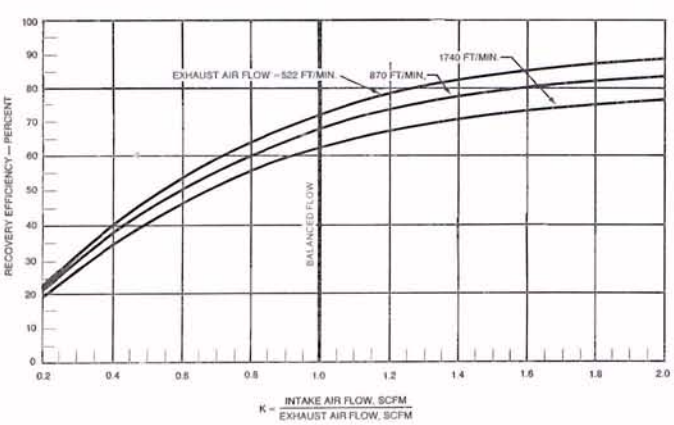


Figure 7-7 EFFICIENCY CURVES (Effectiveness) FOR UNBALANCED AIR FLOWS (Fixed Plate Exchanger) (7)

2. Condensation & Freeze-up

In the fixed plate exchanger, as well as in other sensible heat exchangers, the greater the moisture content of the exhaust gas stream, the less likely that freezing will occur on the surface of the exchanger (see Figure 7-8). Also, if the fixed plate exchanger is oriented in a manner so that the condensate run-off and the airflow are in the same direction, no noticeable increase in pressure drop will occur as condensing begins to take place. However, each situation that appears to pose a freeze problem should be evaluated and brought to the attention of the fixed plate device manufacturer for his recommendations. Several solutions to freeze problems exist, all of which reduce the amount of energy saved during the period that frosting might occur.

FROST THRESHOLD TEMPERATURE T_1 (°F) FOR ENTERING OUTSIDE AIR					
ENTERING EXHAUST AIR CONDITIONS		RATIO OF FRESH AIR FLOW TO EXHAUST AIR FLOW, K			
°F	RH%	0.5	0.7	1.0	2.0
60	30	2	15	23	32
60	40	2	15	23	32
60	50	−4	9	18	32
60	60	−9	4	13	22
70	30	−13	5	17	28
70	40	−21	−3	10	21
70	50	−27	−9	3	15
70	60	−32	−13	−1	10
80	30	−35	−11	−4	19
80	40	−44	−20	−5	10
80	50	−53	−30	−14	1
80	60	−62	−39	−23	−8
90	30	−58	−30	−11	5
90	40			−24	−8
90	50				−20

Figure 7-8 FROST THRESHOLD TEMPERATURES (7)

Many heat recovery applications have high moisture content exhaust gases from which heat is to be extracted, such as drying ovens, curing ovens, swimming pools and cooking areas. Fixed plate exchangers are suited to such applications because the air to be heated can benefit by the transfer of the latent heat of condensation. For every pound of moisture condensed on the exhaust side, approximately 1000 btu is transferred to the incoming air. As a result, the incoming airflow can be as much as two or three times the exhaust gas flow with no appreciable decrease in the incoming temperature effectiveness. This feature of the plate exchanger is of considerable importance when it is desired to use waste process heat for heating make-up air. Most units are factory equipped with condensate drains for removal of the condensate, as well as the waste water from a water-wash system when used.

E CAPACITY CONTROLS

Face and by-pass dampers in the supply airstream is an effective method of controlling the temperature of the make-up or outside airstream, while simultaneously giving a means to by-pass the supply air for cross-control of the exhaust air passage ways.

F MAINTENANCE

Fixed-plate exchangers normally operate with the least amount of maintenance of all of the air-to-air energy recovery units. However, all units and systems require periodic maintenance depending upon the application and amount of particulates and condensibles in the airstreams. Depending upon the application, plate exchangers may be cleaned in place by cleaning solutions, water, steam chemicals and compressed air. Modular units, after disassembly, can be cleaned by immersion in a tank containing a strong chemical solution or by decomposing the contaminants to an easily removable ash in high temperature drying ovens. See Section XVI—"Maintenance and Field Testing" for detailed information.

SECTION VIII
THERMOSIPHON (HEAT PIPE) EXCHANGERS

A DESCRIPTION

Thermosiphon heat exchangers are devices which utilize the natural gravity circulation of an intermediate fluid to transfer energy between exhaust and supply airstreams through a "boiling and condensing" process. They may be classified into two types; sealed tube thermosiphons (heat pipe exchangers) and coil loop thermosiphons.

The sealed-tube type exchanger illustrated in Figure 8-1, is essentially a series of heat pipes set into metal fins similar to the normal HVAC finned-tube coil. The only distinction that can be made between the two types of thermosiphon exchangers is that heat pipes are usually considered to utilize, if not solely rely on, capillary forces to cause the intermediate liquid to flow from the cold to the hot end of the tubes, whereas the pure thermosiphon tube relies only on gravity. While the heat pipe appears very similar to the standard HVAC coil, it differs in one major aspect. Rather than each tube being connected to another by a return bend or header, each individual tube is a heat pipe which operates independently from the others in the exchanger.

Warm air is passed through one side of the exchanger, and cool air through the other side in the opposite direction (a counterflow arrangement). Energy from the warm airstream is transferred by the heat pipes to the other side of the exchanger where it warms the cooler airstream. Although the heat pipes span the width of the unit, a sealed partition separates the two airstreams, preventing any cross-contamination between them.

The coil-loop type, illustrated in Figure 8-2, is similar in appearance to the "run-around coil exchanger" described in the next section. The most obvious difference is the absence of a circulating pump in the thermosiphon loop. This arrangement does not require the two airstreams to be adjacent nor does it require other forms of energy to drive a pump. However, the orientation of the two coils must be such that condensate can flow by gravity into the coil operating as the "evaporator". This places significant restrictions on the system layout, especially if it is a bidirectional installation.

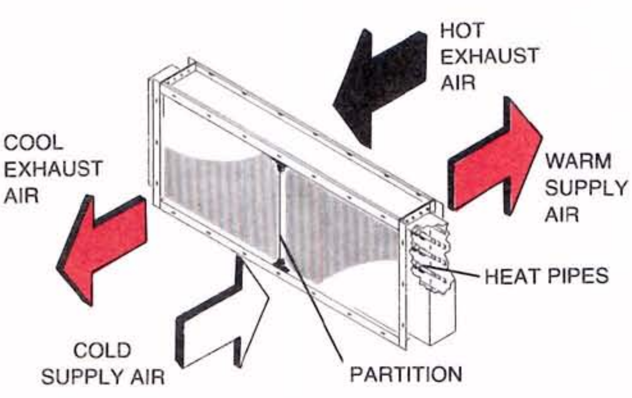
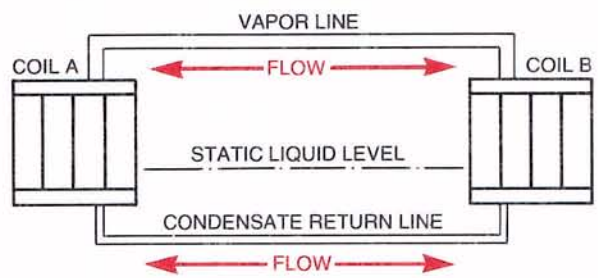
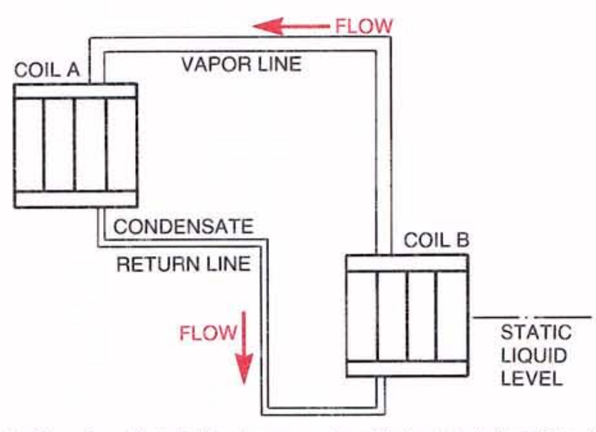


Figure 8-1 HEAT PIPE EXCHANGER (17)



Bidirectional installation. Loop can transfer heat in either Direction, A to B or B to A.



Unidirectional installation. Loop can transfer heat only from B to A.

Figure 8-2 ELEVATION VIEW OF COIL LOOP THERMOSIPHONS (3)

1. Temperature Range

Thermosiphon exchangers are available to operate in the range of temperatures from −60°F to 1325°F. Units that operate up to 525°F are manufactured from aluminum, copper and steel, while the higher temperature units are usually made of steel.

2. Construction

Thermosiphon exchangers for exhaust temperatures below 500°F are most often constructed with aluminum tubes and fins, or for HVAC use, with copper tubes and aluminum fins. Thermal cycling can cause a loosening of the fin-tube bond, resulting in a higher thermal impedance and a lower exchanger effectiveness. Devices designed for high exhaust temperatures have the greatest potential problem. Units with copper tubes and fins are also utilized where the aluminum units are unsuitable because of corrosion or cleaning solvents. Thermosiphon exchangers for use above 500°F are generally constructed with steel tubes and fins, which are often aluminized to prevent rusting.

Composite systems for special applications may be created by assembling units having different materials of construction and/or different working fluids; e.g., a steel exchanger which reduces the exhaust temperature from, say, 650°F to 450°F followed by an aluminum exchanger which reduces the exhaust temperature even further. Thermosiphon exchangers can also be given protective coatings for corrosion prevention.

B PRINCIPLES OF OPERATION

1. The Transfer Process

A thermosiphon exchanger is a sealed device or system containing a working fluid, part of which is in the vapor form and part in the liquid form. Because both liquid and vapor phases are present, the pressure in a thermosiphon is governed by the liquid temperature at the liquid-vapor interface. If the airstreams cause a temperature difference between two coils in a thermosiphon where liquid-vapor interfaces are present, then the resulting vapor pressure difference causes vapor to flow from the warm coil to the cool coil. The flow is sustained by condensation in the cool coil and evaporation in the warm coil. Thus, energy is transported from the warm airstream to the cool airstream as latent heat in the vapor. When the cool and warm coils of the thermosiphon are suitably oriented physically with respect to one another, the condensate will return to the evaporator coil by gravity or by action of a capillary wick, thus completing its cycle.

The natural circulation, "gravity" return feature of a thermosiphon has both advantages and disadvantages for the designer. The prime advantage is that units may be designed which will transfer energy between supply and exhaust airstreams in one direction only (unidirectional), or in either direction with equal or different effectivenesses in each direction, without the need for a control system.

The **heat pipe** is a tube which has been fabricated with a capillary wick structure, filled with a refrigerant, and sealed. Thermal energy applied to either end of the sealed pipe causes the refrigerant at that end to vaporize. The refrigerant then travels to the other end of the pipe where the removal of thermal energy causes the vapor to condense into liquid again, giving up the latent heat of condensation. The condensed liquid then flows back to the evaporator section (i.e., the hot end) by action of the capillary wick completing the cycle (See Figure 8-3). Factors which affect capacity are: wick design, tube diameter, working fluid, and tube orientation relative to horizontal.

Sealed tube thermosiphons are used only when the supply and exhaust airstreams are adjacent to one another. They have operating characteristics, applications, and limitations essentially identical to those described above. It should be pointed out, however, that for a bidirectional system where the tubes are mounted horizontally, the thermosiphon performance is much more sensitive to tube misalignment than is the heat pipe. For the latter, capillary forces ensure that liquid returns to the evaporator end of the tube whereas the thermosiphon relies only

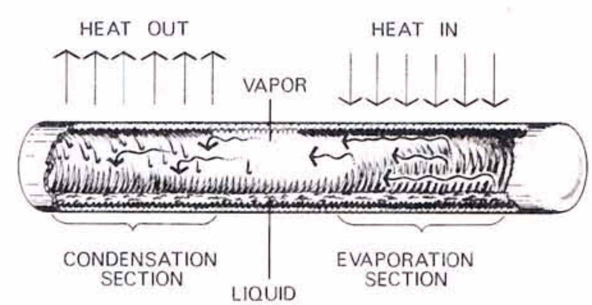


Figure 8-3 HEAT PIPE OPERATION (17)

on gravity. If all the liquid resides at the cold end of a tube, then no heat transfer can take place in that tube. This is not a problem for unidirectional heat flow installations where the evaporator (hot) end of the tube is always at a lower elevation than the condenser end.

Coil loop thermosiphons may be used when the supply and exhaust airstreams are not adjacent to one another. Although similar in appearance and application to the "Run-around coil exchanger" described in Section IX, there are a number of features which are unique to the coil loop thermosiphon.

As illustrated in Figure 8-2, a single coil loop thermosiphon consists of two coils interconnected by a vapor and a condensate line. The loop is charged with a quantity of working fluid at its saturation state so that part of the loop is filled with liquid and part with vapor. The pressure within the sealed loop is thus dependent upon the working fluid utilized and the fluid temperature at the liquid vapor interface.

In order for a coil loop thermosiphon to operate, the coil in the warmer air duct must have some liquid working fluid present at all times in order that vaporization can take place. If liquid is present at all times in both coils, the system can transfer heat in either direction. If it is present only in one coil, heat can flow only from that coil to the other. For this unidirectional system, if the supply and exhaust temperatures reverse, it simply stops operating because there is no liquid to vaporize. Since the condensation and evaporation processes are affected in different ways by the diameter, length, and orientation of the coil tubes,

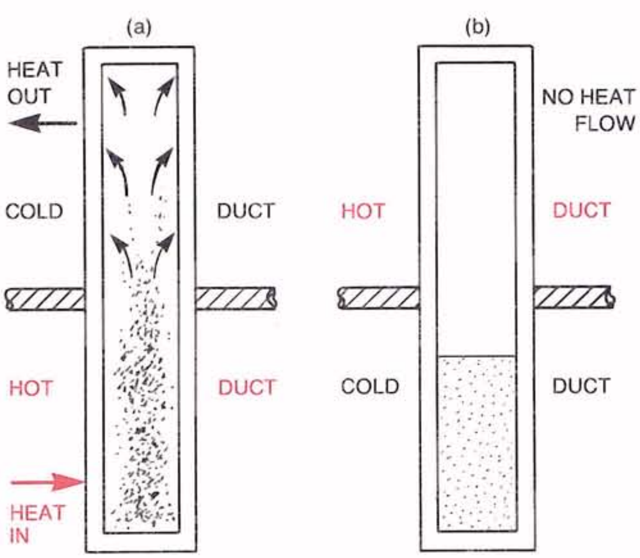


Figure 8-4 SCHEMATIC ELEVATION VIEW OF SEALED TUBE THERMOSIPHONS (3)

as well as by the amount of liquid charge present in each coil, systems may be designed to provide a different effectiveness for each energy flow direction without the use of external controls.

2. Cross Contamination

Heat pipe exchangers typically have zero cross contamination for pressure differentials between airstreams of up to 50 inches water column. A sealed partition effectively separates the two airstreams, thereby preventing any leakage from one airstream to the other. For additional insurance against cross contamination, it is simple to construct the unit with two separating partitions with a space between them of several inches. By attaching the supply and exhaust ducts to these partitions, any leakage that would occur would leak into the space between the two ducts rather than from one duct to the other.

Coil loop thermosiphon exchangers have no cross contamination, as the two airstreams are usually not in close proximity with each other when this type of system is used.

C PERFORMANCE

1. Pressure Drops

Design face velocities for heat pipe heat exchangers range from 400 to 800 fpm, with 450 to 550 fpm most common. Pressure drops at 60% effectiveness (sensible) range from 0.4 to 0.7 in. w.g. at 400 fpm up to 1.5 to 2.0 in. w.g. at 800 fpm. Recovery performance, or effectiveness, decreases with increasing velocity, but the effect is not as pronounced as with pressure drop. A typical set of curves is shown in Figure 8-5.

Available fin designs include continuous corrugated plate fin, continuous plain plate fin, and spiral fins. These various fin designs and different tube spacing cause the variation in pressure drop noted above at a given face velocity.

2. Effectiveness

The effectiveness of a heat pipe or thermosiphon exchanger is a function of several external variables: the heat transfer surface area offered by the combination of rows and fins, the ratio of the heat capacities of the two airflows, the rate at which the two airflows are passed through the heat exchanger, and the temperature range for which the heat pipe was designed.

As indicated earlier, thermosiphon exchangers have finite heat transfer capacity dependent on their design and orientation. As a result, a heat pipe exchanger will be able to obtain the design effectiveness determined by the heat exchanger variables as long as the heat pipes can collectively transfer the resulting heat load from one side of the exchanger to the other. A heat pipe heat exchanger can thus be conceptually visualized as two separate heat exchangers thermally connected by heat pipes.

Heat pipe heat exchangers should be operated with counter flow airstreams for maximum effectiveness. However, they can be operated with parallel airstreams at reduced effectiveness. For example, a heat pipe heat exchanger operating at 60% effectiveness with equal mass flows in a counter flow arrangement will operate at 48% effectiveness with equal mass flows if placed in a parallel flow arrangement. As a rule of thumb, parallel flow effectiveness can be estimated to be 80% of counter flow effectiveness. Figure 8-5 contains performance data and Figure 8-6 equates effectiveness to the number of rows of heat pipes in a coil. It can be seen that as the total number of rows of tubes increases above 6 or 7, the effectiveness increases at a lower rate. For example, with a fin spacing of 14 fpi, it takes 6 rows of tubes to produce 60% effectiveness; doubling the number of rows to 12 increases the effectiveness to 75%. It can be further noted that the effectiveness of a heat pipe exchanger is dependent on the total number of rows, not on the number of rows in a single unit. Thus, two

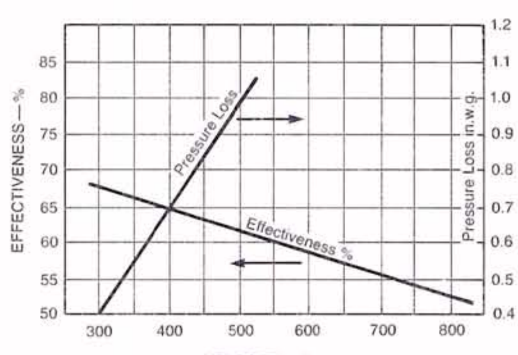


Figure 8-5 PERFORMANCE CURVES FOR A HEAT PIPE EXCHANGER (4)

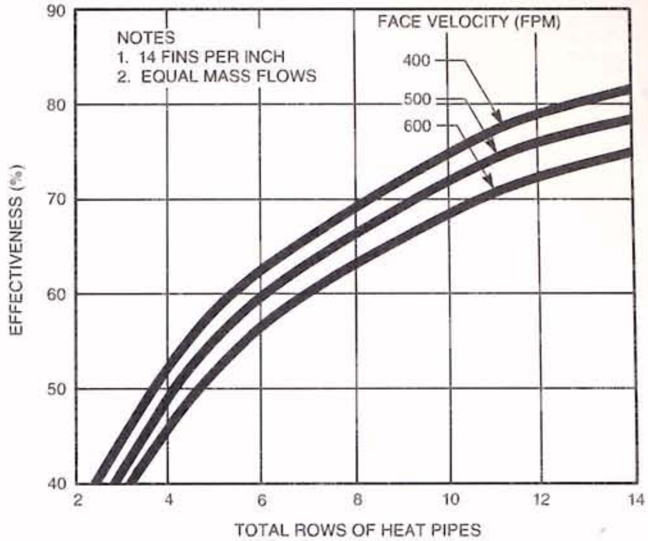


Figure 8-6 HEAT PIPE EXCHANGER EFFECTIVENESS

heat pipe units in series (as shown in Figure 8-9) yield the same effectiveness as a single unit of the same total number of rows. Units in series are often employed in industrial applications to facilitate handling, cleaning and maintenance.

D APPLICATION

Undirectional coil loop thermosiphons are inherently more efficient in operation than bidirectional heat pipe units operating under the same conditions since the coils and loop charge may be selected for optimum performance in carrying out only one function in the first case, rather than having to perform both functions for the bidirectional system. For either case, in order to recover a higher proportion of the available energy, several coil loop thermosiphons may be mounted like waffers along the supply and exhaust ducts, as shown in Figure 8-7, to create a counter flow heat exchanger having an overall effectiveness greater than that for a single loop. Any desired effectiveness can be achieved by utilizing multiple coil loop thermosiphons in this manner. In the example in Figure 8-7, the temperature distribution for a system having equal exhaust and supply flow rates and where each coil has an effectivenss of 50%, the effectiveness of each loop will then be approximately 25%. The effectiveness of two loops would be 40% and three thermosiphon loops would be 50%. If the effectiveness of each coil was 80%,

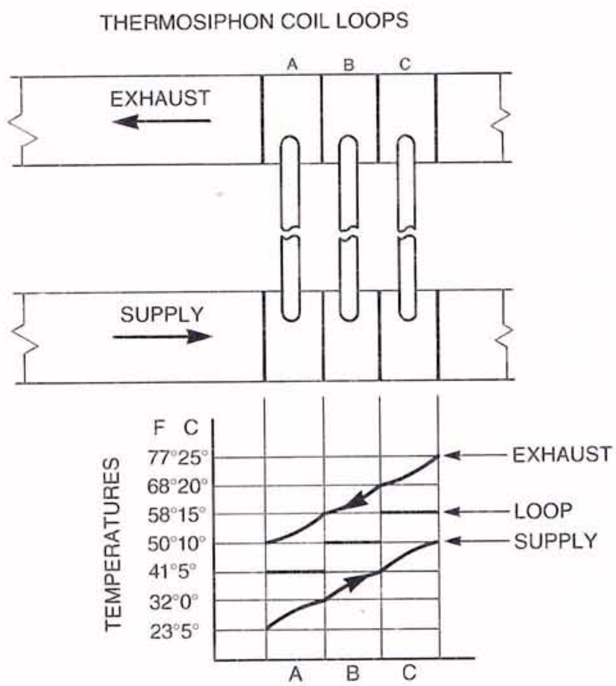


Figure 8-7 MULTIPLE THERMOSIPHON LOOP HEAT EXCHANGERS (3)

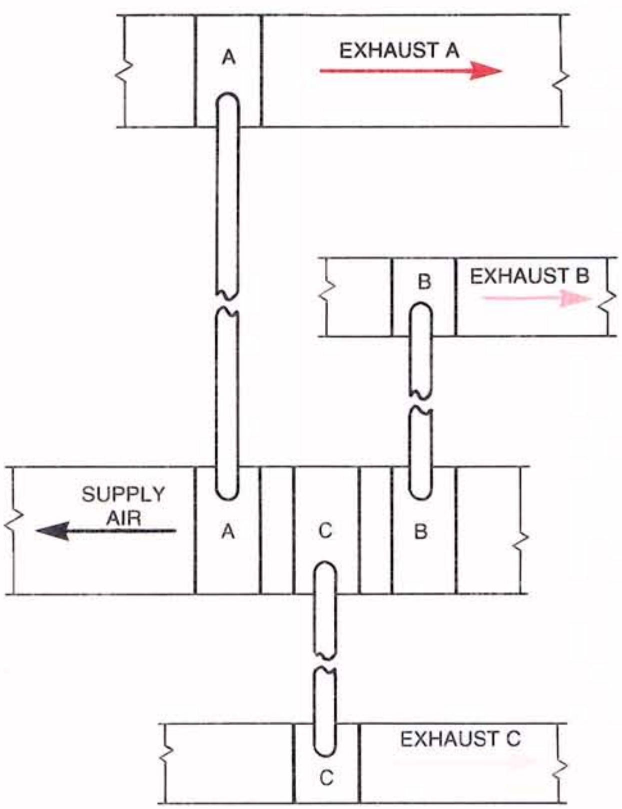


Figure 8-8 MULTIPLE EXHAUST SINGLE SUPPLY INSTALLATION (3)

then the effectiveness of the three thermosiphon loops would be 67%.

The most economic combination of coil sizes and number of loops depends upon the specified design criteria, the projected system lifetime, and the economic criteria used to optimize the design. Note, also, that since each loop is independent, different working fluids could be utilized in each loop if it is advantageous to do so.

Thermosiphon coil loops may also be used when multiple supply or exhaust ducts are present. Figure 8-8 illustrates schematically such a case where a common supply duct is being heated by three different exhaust ducts.

1. Filters

Heat pipe exchangers require the same filtration protection that normal HVAC system finned-tube coils require when operating under the same environment. Fin pitch, the number of rows, the size of the particulates in the airstreams, and the type of condensables are some of the items to consider.

2. Condensation and Freeze-up

As in all air-to-air energy exchangers, condensation will occur when some portion of the transfer surfaces or media attains a temperature below the dew point of the warm airstream. Provisions for condensate removal, such as drain pans or recessed ductwork, should be incorporated into the system so that the condensate will be prevented from flowing into the ductwork.

In cases of outside air temperatures below 32°F, this condensation may freeze. While a partial freeze-up may not cause any permanent damage to the heat exchanger, it will result in reduced heat recovery, reduction of exhaust airflow, and increased exhaust airstream pressure drop.

When condensation is expected during normal operations, the system designer might determine the approximate row in which this could occur. By separating the thermosiphon exchanger into two units at that point (as shown in Figure 8-9), the "wet portion," that may require special attention or cleaning, can be isolated.

Condensation of vapors from industrial process exhaust often occurs upon cooling. The amount of condensation depends on the initial concentration of the vapors and the amount of cooling that the exhaust undergoes. Freeze-up of this condensate seldom

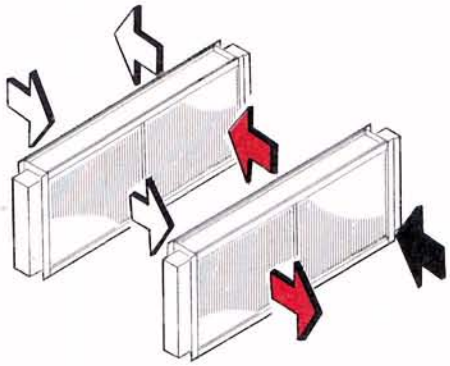


Figure 8-9 HEAT PIPE EXCHANGERS IN SERIES (17)

occurs because the exhaust temperature is usually higher than its freezing point. Unless it freezes, the condensation of either water or other vapors does improve heat recovery by increasing the exhaust side convection film coefficient, as long as it does not become sludge or a solid. An increase in the pressure drop of the exhaust airstream can occur however.

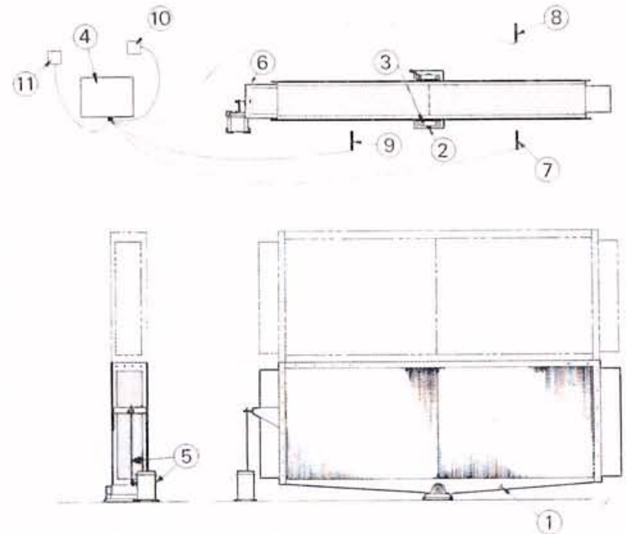
E CAPACITY CONTROLS

Gravity can be utilized to assist in returning the condensate within a bidirectional thermosiphon or heat pipe unit to the evaporator section by operating the heat pipe on a slope with the hot end below horizontal. Conversely, by placing the exchanger on a slope with the hot end above horizontal, gravity retards the condensate flow. This method of changing the slope or tilting the unit offers a means of controlling the amount of heat that the exchanger transfers and regulating its effectiveness.

In practice, tilt control is accomplished by pivoting the exchanger about the center of its base and attaching a temperature controlled actuator to one end of the exchanger (see Figure 8-10). Flexible duct connections allow freedom for the small tilting movement.

There are three conditions under which tilt control may be desired to regulate the recovery performance of a heat pipe exchanger:

1. To reverse the slope of the heat pipes to lower the warm side of the exchanger after the seasonal change-over, thus obtaining maximum energy transfer.



(1) special tilt base
(2) shaft
(3) bearings
(4) electronic controller
(5) electric actuator with linkage attachments
(6) follow arm bracket
(7) outside air temperature duct thermostat.
(8) supply leaving temperature sensor.
(9) exhaust leaving temperature sensor.
(10) supply leaving temperature remote set point control.
(11) exhaust leaving temperature remote set point control.

Figure 8-10 "TILT CONTROL" FOR A HEAT PIPE EXCHANGER. (17)

2. To regulate the supply air exit temperature leaving the unit to a desired maximum temperature.
3. To prevent frost formation on the weather face of the exhaust side of the exchanger at low outside air temperatures. By reducing the recovery of the unit, the exhaust air leaves the unit at a warmer temperature and stays above frost-forming conditions.

While tilt control is the only means of accomplishing all three of the above functions, there are other methods available to accomplish the individual functions. Regulation of the supply air temperature can be accomplished with the use of face and bypass dampers. Similarly, frost formation can also be prevented by the use of these devices. Preheat of the supply air duct upstream of the heat recovery unit provides still a third, but not the most desirable, approach to frost prevention.

F MAINTENANCE

Cleaning of thermosiphon or heat pipe exchangers is dependent upon the nature of the material to be cleaned from the units. Grease build-up from kitchen exhaust, for example, is often removed using an automatic water-wash system (Figure 8-11). Other applications utilize manual spray cleaning of the units (Figure 8-12), soaking of the units in a cleaning tank, or with compressed air or steam such as using soot blowers.

The decision for the cleaning method to be used should be made in the design stages, so that a proper selection of the heat exchanger can be made. An advantage over some other type coil systems, is that there is no piping to disconnect with the heat pipe banks. The frequency of cleaning is dependent on the quality of the exhaust airstream. HVAC systems generally require infrequent cleaning, while industrial systems usually require more frequent cleaning.

Since thermosiphon and heat pipe exchangers have no moving parts, they require a minimum of mechanical maintenance. However, the flexible connections, damper and tilt controls, and all other accessory items, such as automatic washer systems, should have regular inspections.

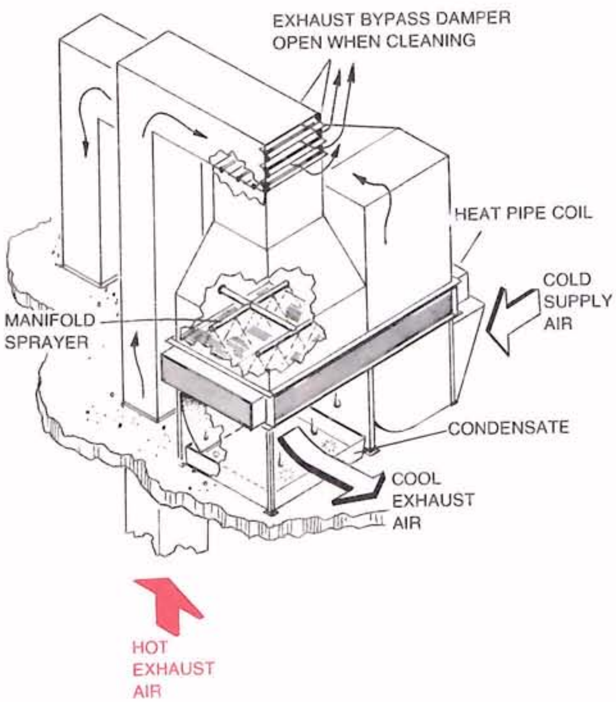


Figure 8-11 AUTOMATIC IN-DUCT SPRAY CLEANING SYSTEM (17)

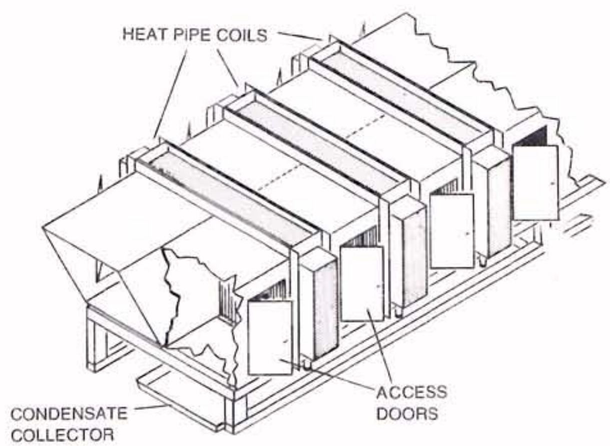


Figure 8-12 DESIGN FOR IN-DUCT MANUAL SPRAY CLEANING (17)

SECTION IX
RUN-AROUND COIL EXCHANGERS (COIL HEAT RECOVERY LOOP)

A DESCRIPTION

Standard extended surface, finned-tubed water coils used in normal HVAC applications are placed in the supply and exhaust airstreams of a building or process. The coils of the run-around system are connected via counterflow piping, and a pump circulates a water, glycol, or thermal fluid solution (Figure 9-1). Multiple coils can be simultaneously mounted in several different airstreams at both remote and local locations, and connected in series or parallel to best suit a given energy recovery process. The system is seasonably reversible. Specially designed coils are built for high temperature process heat recovery systems.

1. Temperature Range

The temperature range of run-around coil exchangers is limited by the thermal transfer fluid. At higher temperatures (above 400°F), special coil construction and control systems may be required to ensure a permanent bond of the coil fins to the tubes and to prevent the transfer fluid from reaching excessive temperatures.

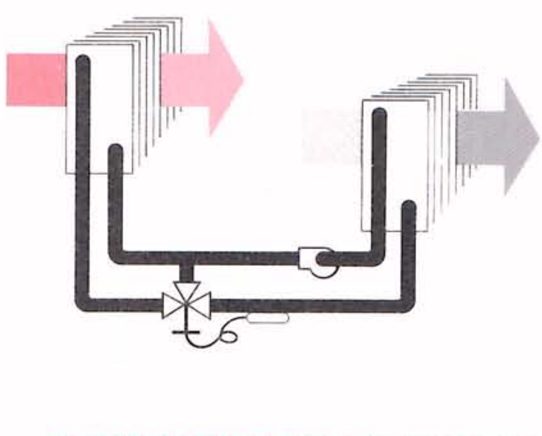


Figure 9-1 RUN-AROUND COIL EXCHANGERS (COIL HEAT RECOVERY LOOP) (20)

An inhibited glycol solution is usually used in the piping circuit in comfort range applications. An inhibitor should be maintained in the solution for corrosion protection. Glycol solutions (normally to 225°F in a 400°F maximum airstream), should not be used at temperatures greater than 300°F. Above this temperature, glycol has the potential to break down and form an acidic sludge. For temperatures above 300°F, thermal transfer fluids should be used (discussed later in this section).

2. Construction

Run-around coil systems incorporate coils constructed to suit the environment and operating conditions to which they are exposed. For typical comfort-to-comfort applications, the standard HVAC coil construction usually suffices. Process-to-process and process-to-comfort applications require consideration of the effect of high temperatures, condensibles, corrosives, and contaminants on the coil(s), and adequate cleaning requirements. The effects of the condensibles and corrosives may also require specialized coil construction materials and/or coatings.

A round spiral-finned tube core energy recovery coil has been designed for use with air or gas temperatures up to 800°F. These coils can be built with one or two passes (depending on allowable pressure drop) and varying fin spacing to accommodate airstreams with heavy contaminants. A coil of this type of construction is shown in Figure 9-2.

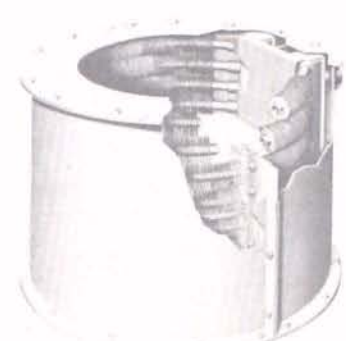


Figure 9-2 HIGH TEMPERATURE COIL EXCHANGER (14)

B PRINCIPLES OF OPERATION

1. The Transfer Process

The run-around coil system allows seasonal transfer of energy from a hot airstream to a cold airstream. When outdoor air is cooler than the exhaust air, primarily waste sensible heat is recovered. Similarly, when the exhaust air is cooler than the outside air, sensible heat is removed from the supply air. Thus, the supply air coil either preheats or or precools the outside air, being seasonally reversible.

2. Cross Contamination

Complete separation of the airstream eliminates the possibility of cross contamination between the outside airstream and the exhaust airstream(s).

C PERFORMANCE

1. Pressure Drops

When standard finned-tube water coils are used, coil engineering data should be used to determine air pressure drops for a specific design. Coil face velocities are typically in the 300-600 fpm range. Lower face velocities may result in first cost penalties. Higher face velocities may result in operating cost penalties because of higher airstream pressure drops.

Pressure drops of the high temperature round spiral-finned tube coils range from 0.02 in. w.g. to 4.0 in. w.g., depending on the fin spacing and the number of passes.

2. Effectiveness

The effectiveness of a heat recovery loop design should be carefully analyzed. When system designs are being evaluated on their economic merit, the best design is that which has the greatest net cost savings. Highest effectiveness is not necessarily synonymous with greatest net cost savings, although it is an indicator of the greatest energy savings. Net cost savings can most accurately be determined by applying the costs of heating and cooling, the operating costs of the system, the operating times of the system, the geographic location, and the schedule of system capacities (including effect of controls) versus outside air temperatures.

Applications with exhaust temperatures up to 400°F normally can achieve a sensible heat effectiveness of 60 to 65 percent. Applications in excess of 400°F usually reach a sensible heat effectiveness of up to 50 percent, which is slightly lower due to the reduced heat transfer capabilities of the thermal fluids and materials of construction.

Generally, the sensible heat recovery effectiveness is in the 40 to 60 percent range using reasonably deep coils. An attempt to further increase the system efficiency by adding additional rows to the coils is often offset by the higher fan horsepower required by the increased pressure drops of the airstreams. The effectiveness of the heat recovery loop is equal to the product obtained by multiplying the individual coil efficiencies (if each coil effectiveness is 80%, then $0.8 \times 0.8 = 0.64$ or 64% effectiveness for the loop).

D APPLICATION

The run-around coil energy recovery system affords a high degree of flexibility which makes it particularly well suited for renovation and industrial applications. The system accommodates remote location of supply and exhaust airstreams. It also allows the simultaneous recovery of heat from multiple supply and/or exhaust airstreams.

The use of an expansion tank (Figure 9-3), which allows a "closed" system, is required to: ensure that the system is filled whenever the mean fluid temperature is less than the fill temperature; provide room for volumetric expansion whenever the mean fluid temperature is greater than the fill temperature; prevent oxidation and subsequent corrosion when a glycol solution is used, as would happen in an "open" system.

It is recommended that the supply air fan be a draw-thru unit, and the exhaust air fan a blow-thru unit. This provides the maximum air temperature difference between the exhaust and supply air coils, hence the greatest system heat transfer capacity. Heat from the fan motors is also added.

The supply air temperature for most applications can typically be increased 60-65% of the temperature dif-

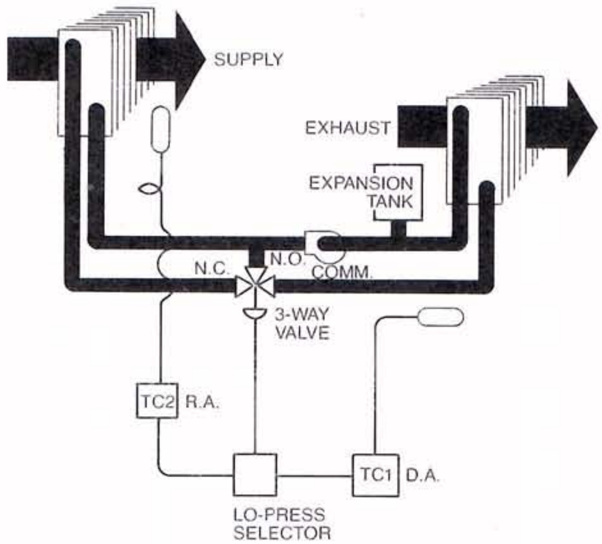


Figure 9-3 RUN-AROUND COIL SYSTEM (20)

ference between the two airstreams. In some applications in which the exhaust air quantity exceeds the supply air quantity and/or where the exhaust air has a high humidity level, the supply air temperature can be increased by up to 85% of the temperature difference between the two airstreams.

1. Filters

The coils used in the coil heat recovery loop system require the same filtration protection that normal HVAC system heating and cooling coils require when operating in the same environment. Fin pitch, the number of rows, the size of the particulates in the airstreams, and the type of condensables are some of the items to consider.

2. Condensation and Freeze-up

As in all air-to-air energy exchangers, condensation will occur when some portion of the transfer surfaces or media attains a temperature below the dew point of the warm airstream. Provisions for condensate removal, such as drain pans or recessed ductwork, should be incorporated into the system so that the condensate will be prevented from flowing into the ductwork.

In cases of outside air temperatures below 32°F, this condensation may freeze. While a partial freeze-up may not cause any permanent damage to the heat exchanger, it will result in reduced heat recovery, reduction of exhaust airflow, and increased exhaust airstream pressure drop.

E CAPACITY CONTROLS

As with other air-to-air energy recovery equipment, measures must be taken to prevent potential freezing of exhaust air condensate. A dual-purpose, 3-way temperature control valve is used to prevent exhaust coil freeze-up (see Figure 9-3). The valve ensures an entering thermal transfer solution temperature to the exhaust airstream of not less than 30°F. This is accomplished by bypassing the warm solution from the supply air coil. The valve can serve the additional purpose of ensuring that a prescribed supply air exit temperature is not exceeded. This applies for those applications where heat recovery must be varied or limited. This control can be accomplished by electric, pneumatic, or electronic control systems. The pump can be controlled by an outside air temperature thermostat.

In many installations, face and bypass dampers can be used around one of the coils (or sets of coils) to achieve an adequate control of the recovery system. Another method is to mix the airstreams when contaminants and other undesirable materials are not present in the exhaust airstream. Controls should also be used to protect the thermal transfer fluids from being exposed to temperatures in excess of their rated limits.

F MAINTENANCE

The only moving parts in the run-around coil system are the circulation pump, the 3-way control valve and damper operators. However, the following items must be considered to assure optimum operation: air filtration, cleaning of the coil surfaces, and periodic maintenance of the pump, control valve, dampers (if used) and damper motors and linkages. The system must be kept free of air, the fluid at the proper solution to prevent freezing and corrosion, and all parts of the piping system periodically inspected for leaks.

Coils can be cleaned using steam, compressed air, hot soapy water, or suitable solvents. If the exhaust air dictates frequent cleaning, automatic wash down cycles may be installed for certain contaminants. Drain holes and piping should also be checked for debris.

G THERMAL TRANSFER FLUIDS

Depending on the application, there are a host of intermediate thermal transfer fluids that can be used in run-around heat recovery systems. Before deciding on a fluid, consideration must be given to the following design variables:

- — need for freeze protection
- — need for burst protection
- — need for corrosion protection
- — heat transfer efficiency
- — fluid life and maintenance
- — fluid specifications versus design conditions

In many cases, demineralized water can serve as the heat transfer medium. This is preferred since pure water offers high heat-transfer efficiency. Water is limited in application because of the high 32°F freeze point. Consequently, there can be damage to the equipment at the lower ambient temperatures of many regional areas. Additionally, water could ionize and serve as an electrolyte in the presence of similar or dissimilar metals. Hence, galvanic corrosion of the equipment could occur.

The shortcomings of water have created the need for other heat transfer fluids. Specific recommendations for a given application should be sought from a fluid manufacturer or his representative.

1. Types

There are generally considered to be two basic categories for fluids exposed to the range of energy recovery system temperatures. They are either water-glycol mixtures or synthetic thermal transfer fluids.

a. Water-glycol Mixtures. The more commonly used mixtures are *ethylene glycol-water* and *propylene glycol-water.* They display similar characteristics at the same concentrations. Propylene glycol-water mixtures are of low toxicity. Consequently, they are used in those applications in close proximity to food or foodstuffs. Propylene glycol-water mixtures have higher viscosities than ethylene glycol-water mixtures. These mixtures allow freeze protection below 32°F. However, they can degrade at temperatures above 275°F to 300°F.

b. Synthetic Thermal Transfer Fluids. There are a number of fluids in this class (see Section XII — "Thermal Transfer Fluids" for more detailed information). Basically, they are aqueous or non-aqueous. The newer non-aqueous fluids are popular due to their low freeze points (as low as −40°F), wide temperature range, and good viscosity. Since these fluids have high boiling points, they are suitable for higher temperature applications (up to 700°F) where pressurized water systems are not feasible.

2. Freeze Protection

A run-around heat recovery system exposed to air temperatures below freezing should incorporate some form of freeze protection. Simple means of freeze protection such as draining and continuous circulation may not be appropriate for the run-around coil system.

Advantages of a freeze preventative solution are:

a. prevents equipment problems on start-up after a power failure or interruption.

b. eliminates the need for pipe heaters or steam tracing lines.

c. eliminates the need for draining; hence corrosion is not promoted.

d. eliminates the need for solution circulation during periods when energy recovery is not needed (pump energy is saved).

3. Burst Protection

A system that is protected from bursting during cold periods is not necessarily protected from freezing. A system that is protected from freezing is also protected from bursting. In burst protection, the fluid may crystallize, hence the equipment may be inoperable. However, failure due to expansion is avoided. Burst protection is achieved at lower concentrations than freeze protection.

In run-around coil applications, detailed review should be undertaken before deciding upon burst protection. Burst protection does not provide the advantages of freeze protection. Its advantages are:

a. lower cost due to lower concentration.

b. lower concentration means improved heat transfer efficiency.

Thermal transfer fluid manufacturers or their representatives should be contacted for specific recommendations for burst protection.

4. Corrosion Protection

Corrosion of equipment can lead to leaks, premature failure, and reduced heat transfer efficiency. Corrosion is enhanced by high temperatures and the presence of oxygen.

The glycol solutions break down gradually with use. The end products include organic acids which can corrode aluminum, copper, and steel. The corrosion is accelerated by higher temperatures. Glycols are available with various "inhibitors" which prevent the pH from dropping. Their effect is not permanent, and periodic fluid monitoring is desirable to insure timely inhibitor replenishment.

Many of the non-aqueous synthetic solutions are chemically stable at temperatures of 400°F and above and do not degrade to form corrosives. Consequently, these fluids can offer a long fluid life and ultimately, longer equipment life.

5. Thermal Transfer Efficiency

Thermal transfer fluids display less thermal conductivity than water. Typically, the synthetic fluids have lower thermal conductivities than the glycol mixtures. The effect on overall heat transfer must be determined. The increased energy savings of one fluid may justify its use over a more stable, more expensive fluid.

6. Fluid Life and Maintenance

It is difficult to pinpoint the expected life of the thermal transfer fluids. They depend on the installation and temperature extremes experienced. As a general rule, if fluid temperatures are above 100°F, systems should be closed (with an expansion tank) to minimize air contact and prevent oxidation. This will minimize corrosion and extend fluid life.

Fluid maintenance is primarily following the manufacturer's inspection procedures. Most manufacturers provide sample testing and will advise corrective action on request.

7. Fluid Specifications Versus Design Conditions

Table 12-1 in Section XII shows the operating ranges and other characteristics of the available thermal transfer fluids. With the overlap, the choice of a specific fluid is not immediately obvious.

The glycols are used for those applications within the temperature range of −40°F to 300°F. For temperatures higher than 300°F, the synthetic fluids are used. Synthetic or organic fluids are not normally used within the glycol range. The lower viscosities, lower cost, and better heat transfer characteristics of the glycols usually make the use of the synthetics in this range economically impractical.

SECTION X
MULTIPLE TOWER EXCHANGERS

A DESCRIPTION

The multiple tower system is an air-to-liquid, liquid-to-air enthalpy (total heat) recovery system. The system consists of multiple contactor towers handling the building or process supply air, and multiple contactor towers handling the building or process exhaust air. A liquid is continuously recirculated between the supply and exhaust air contactor towers and alternately directly contacts both supply and exhaust airstreams (Figure 10-1).

1. Temperature Range

Multiple tower enthalpy recovery systems are designed primarily for operating temperatures in the comfort-conditioning range. They are not generally suitable for high-temperature applications such as industrial oven exhaust airstreams. During summer operation, the system will operate with any building supply air temperatures as high as 115°F. Winter supply air temperatures as low as −40°F can generally be tolerated without freeze-up or frosting problems, since the sorbent solution is an effective antifreeze at all useful concentrations.

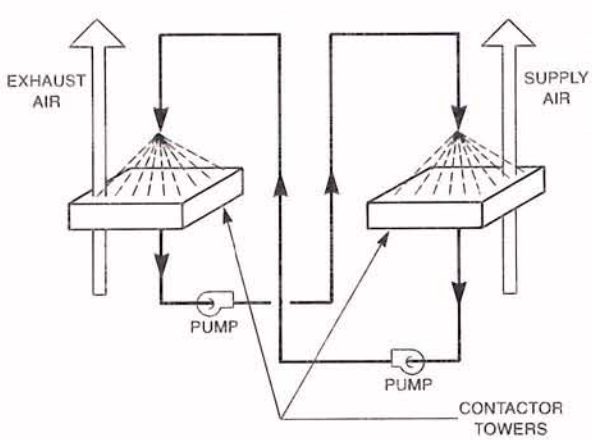


Figure 10-1 TWIN-TOWER ENTHALPY RECOVERY LOOP

2. Construction

Contactor towers are manufactured in both vertical and horizontal airflow configurations. In the vertical airflow configuration, the supply or exhaust airstream passes vertically through the contact surface in a counter flow relationship to the sorbent liquid in order to achieve the highest possible contact efficiency. In the horizontal airflow configuration, the supply or exhaust airstream passes horizontally through the contact surface in a cross-flow relationship to the sorbent liquid yielding a slightly lower contact efficiency. Vertical and horizontal contactor towers can be used in a common system. Contactor towers of both configurations are supplied with airflow capacities of up to 100,000 cfm.

Contact surfaces are usually made of non-metallic materials. The tower casing is usually made of steel with protective coatings. Air leaving the contactor tower passes through demister pads which remove any entrained droplets of the sorbent solution.

B PRINCIPLES OF OPERATION

1. The Transfer Process

One or more energy transfer towers through which the warmer airstream passes will extract both sensible heat and moisture by the process of the sorbent liquid being sprayed through the airstream. The "enriched" sorbent liquid is pumped to the tower(s) in the cooler airstream where the heat and moisture is transferred by spraying (Figure 10-2).

In this direct contact with the airstreams, the liquid acts as a vehicle for transporting humidity, as well as heat. The process is reversible, and in a typical comfort-conditioning application building supply air is cooled and dehumidified during summer operation, and heated and humidified during winter operation.

The sorbent solution is usually a halogen salt solution, such as lithium chloride and water. Pumps are

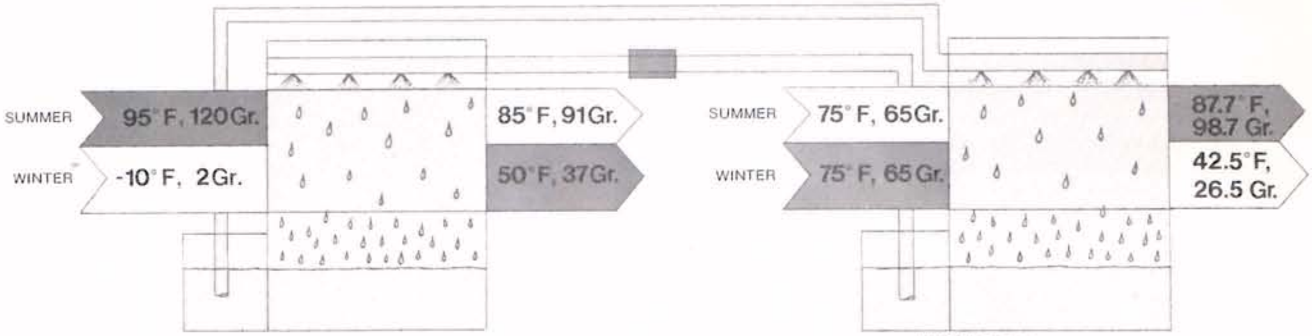


Figure 10-2 MULTIPLE TOWER EXCHANGE PROCESS (12)

used to circulate the solution between the supply airstream and exhaust airstream towers.

2. Cross Contamination

Particulate cross-contamination does not occur in multiple tower systems because the supply and exhaust contactors are independent units connected only by the sorbent liquid. Gaseous cross-contamination may occur to a limited degree, depending on the solubility of the gas in the sorbent solution. Tests of gaseous cross-contamination using the sulfur hexafluoride tracer gas technique of ASHRAE Standard 84, have shown typical gaseous cross-contamination rates of 0.025% for the multiple tower system.

If the building or process exhaust contains large amounts of gaseous contaminants, such as chemical fumes, hydrocarbons, etc., possible cross-contamination effects should be investigated as well as possible effects of the contaminants on the sorbent solution.

3. Bacteria Removal

Sorbent solutions (particularly the halide brines) are bacteriostatic (will not support bacterial life) at all useful concentrations. Micro-organism testing of the multiple tower system using the 6-plate Andersen Sampling Technique has shown that micro-organism transfer does not occur between the supply and exhaust airstreams. These tests have also shown that the exchanger towers are effective microbiological scrubbers, typically removing 94% of micro-organisms in both the supply and exhaust towers.

C PERFORMANCE

1. Pressure Drops

Figure 10-3 depicts a typical airside pressure drop for a multiple tower system. The towers are generally designed to operate between 350 and 525 fpm air face velocity. Airside pressure drops accordingly are generally in the range of 0.5 to 1.2 inches water column.

Since the supply and exhaust air towers are independent units connected only by piping, supply and

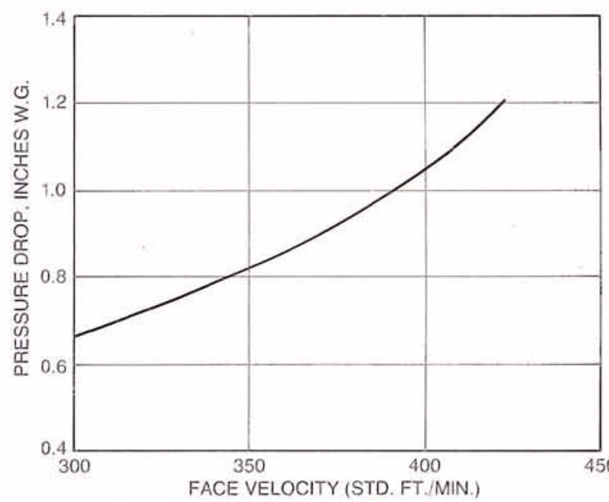


Figure 10-3 MULTIPLE TOWER AIRSIDE PRESSURE DROP (12)

exhaust air fans can be located wherever practical. The towers are generally designed to operate with any air inlet static pressure from −6 in. to +6 in. w.g. The exhaust tower may be operated at a higher internal static pressure than the supply tower without danger of exhaust-to-supply cross-contamination.

2. Effectiveness

Figure 10-4 depicts a typical enthalpy recovery effectiveness curve for a multiple tower enthalpy recovery loop. Effectiveness is shown as a function of the tower air face velocity. The effectiveness of the tower system will generally be about 70% at typical summer operating conditions, and about 60% at typical winter operating conditions.

D APPLICATION

Any number of supply air towers can be combined with a single exhaust air tower, or any number of exhaust air towers can be combined with a single supply air tower. If sufficient elevation difference exists between supply and exhaust towers, the gravity head can be used to return the sorbent solution from the upper tower or towers. Large vertical or horizontal distances between the supply and exhaust airstreams are limited only by the piping required between the multiple tower energy recovery system units.

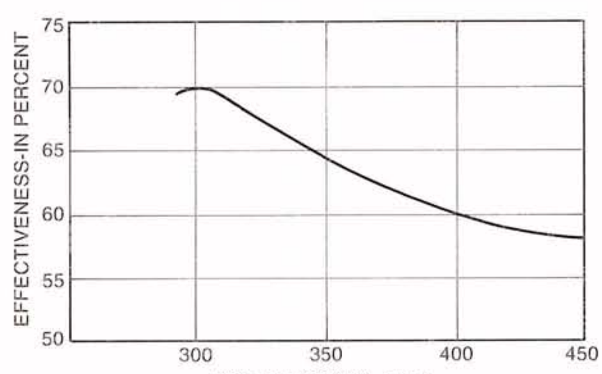


Figure 10-4 MULTIPLE TOWER SYSTEM EFFECTIVENESS (3)

1. Filters

The thermal transfer fluid is a bacteriostatic solution of water and lithium chloride salt. This, together with the efficient air scrubbing within the towers, makes the system an efficient decontaminator of both intake and exhaust airstreams. Dust removal tests show a dust removal efficiency of 35% as measured by the National Bureau of Standards Dust Spot Test Method on normal atmospheric dust.

If the building or process exhaust contains large amounts of lint, animal hair, or other solids, prefiltration should be provided upstream of the exhaust air tower.

2. Condensation and Freeze-up

When applying twin-tower systems on controlled-humidity buildings or areas located in the colder climates, consideration should be given to possible saturation effects. These saturation effects, which can cause condensation, frosting and icing in other types of equipment, may cause oversaturation of the sorbent solution in the multiple tower system. This phenomenon is prevented in the tower system by heating the sorbent liquid supplied to the supply air contactor tower, as shown in Figure 10-5. This elevates the discharge temperature and humidity of the supply air tower, allowing the system to attain humidity balance and preventing oversaturation.

E CAPACITY CONTROLS

Generally, the sorbent solution heater is controlled by a thermostat sensing the air temperature leaving the supply air tower. The tower system is capable of delivering air at a constant temperature all winter, regardless of the outside air temperature. Automatic addition of makeup water to maintain a fixed concentration of the sorbent solution enables the twin-tower system to deliver supply air at a fixed humidity during cold weather. Constant supply air temperature and humidity can thus be delivered by the multiple tower system without the use of preheat coils, reheat coils, or humidifiers in the supply airstream.

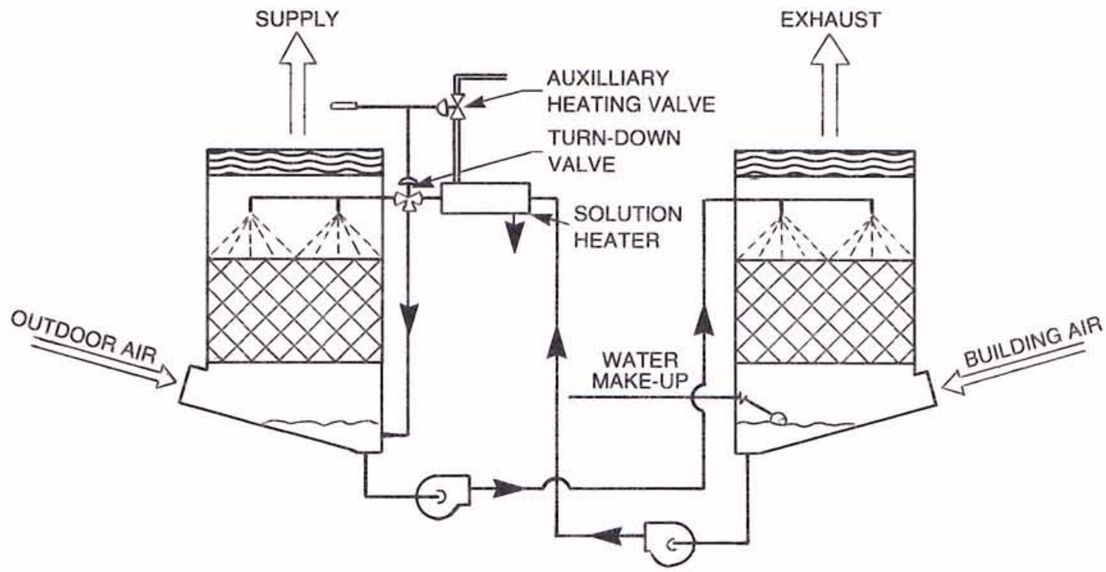


Figure 10-5 MULTIPLE TOWER OPERATION AND CONTROL (3)

A turndown cycle to limit the leaving temperature of the supply airstream tower during mild weather operation is achieved by automatically modulating the flow of the sorbent solution to the tower.

The multiple tower system offers a number of alternatives of capacity control depending upon the specific design. A steam heated shell and tube heat exchanger for the thermal transfer fluid (or solution) is used if over dilution of the solution during winter operation is a possibility. This heat exchanger can then be controlled to supplement heat recovery during winter extremes and provide a fixed temperature to the system during the winter cycle.

Also, by supplementing the heat exchanger with a controlled water make-up source, a fixed humidity can also be achieved. This combination of heat exchanger and water make-up can then eliminate the need for outside air preheat and reheat coils and a humidifier.

As the outdoor air approaches the desired winter base-line delivered air temperature, the tower system modulates its heat recovery potential. To reduce the amount of recovered heat, a thermostat on the intake tower leaving air temperature modulates a turndown valve to reduce the amount of solution sprayed over the intake tower contactor surface. This mode of control allows for exact winter delivered temperatures as the winter weather moderates to spring and as the fall weather approaches winter.

During spring and fall when the air temperature to the intake tower is the same as the desired delivered air temperature, then the turndown valve is in the full diverted position with no spray to the intake contactor surface and all pumps in the tower system are then de-energized.

For outside air temperatures between the desired base line winter delivered air temperature and the building exhaust temperature, the multiple tower system is then in a null cycle. During this cycle, air passes over the contactor surfaces but heat recovery is purposely stopped by de-energizing the circulating pumps.

As the outside temperature increases above the building exhaust air temperature, the pumps are energized and full heat recovery commences. During this cycle, normally the summer cycle, the outside air is then cooled by the heat recovery process.

F MAINTENANCE

Maintenance of the multiple tower system is primarily a matter of observation. On a monthly basis, the following items should be checked.

a. Spray pressure of tower units
b. Complete spray coverage of contactor surfaces

c. Solution concentrations

d. Level controller operation

On a bimonthly basis, the following items require inspection:

a. Solution color comparison

b. Mist eliminator inspection

On a six-month basis, the following should be attended:

a. Greasing of fan and pump bearings and fan and pump motor bearings.

b. Check of level controller and limit level controller function.

c. Washing and reinstallation of mist eliminators.

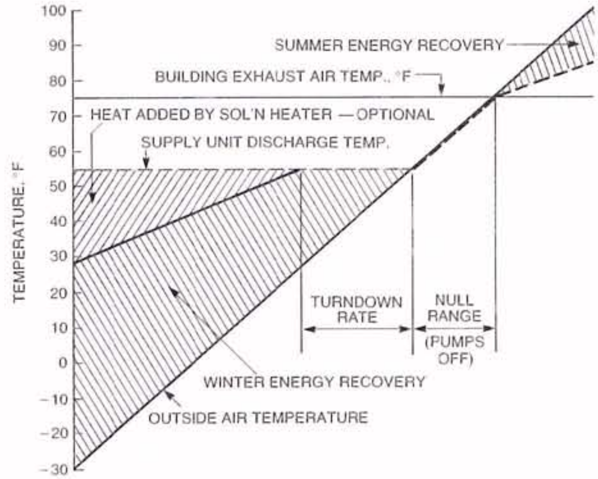


Figure 10-6 PERFORMANCE CHARACTERISTICS OF LIQUID SORBANT ENTHALPY SYSTEMS (3)

G THERMAL TRANSFER FLUIDS

The heat transfer fluid circulated between the multiple tower supply and exhaust units is usually a water and salt solution. Normally, lithium chloride and water or calcium chloride and water is the liquid used. These solutions are stable from −90°F to 240°F at the various concentrations. The solution concentration for a system is determined for the exact temperature extremes of an actual application.

SECTION XI
DRY AIR COOLER EXCHANGERS

A DESCRIPTION

Dry air evaporative cooling utilizes two well known engineering principles to provide *sensible cooling* with extremely low energy consumption. These two principles are evaporative cooling and heat transfer.

The dry air evaporative cooler exchanger (Figure 11-1) is designed with two air passages, a dry one through the interior of the heat exchange tubes and a wet one over the exterior of the heat exchange tubes. Water is circulated over the exterior of these heat exchange tubes while air is moved over them, creating an evaporative cooling effect that reduces the temperature of the tubes. At the same time, air is moved through the dry air passage in the interior of the cooled tubes, thus releasing its heat to the cooled tube surfaces, creating cool air that has not had moisture added to it. This is the cooled dry-side airstream.

Certain types of fixed plate exchangers can also be used as dry air evaporative coolers by spraying a mist of water vapor into the exhaust airstream prior to entering the heat exchanger. The mist deposits on the exhaust side surfaces and is evaporated, thereby lowering the temperature of the plates to approximately the exhaust airstream wet/bulb temperature.

1. Temperature Range

The major use of the dry air evaporative cooler is to pre-cool the outside airstream to reduce the load of the air conditioning system. The cooled, "dry-side" air may also be used as an exclusive cooling source in all applications where design conditions permit. The temperature range is in the normal comfort environmental range of approximately 35°F to a maximum of 125°F.

2. Construction

The dry air cooler casing is fabricated from galvanized steel. The lower water basin is additionally coated with an asphalt emulsion. Tube headers and tubes are made of non-corrosive material. The tubes are covered with sleeves of a knitted synthetic fabric which has high moisture retention and spreading capabilities. All of these materials are impervious to nearly all chemicals, acids, or corrosive atmospheres.

The upper water distribution pan is constructed of totally non-corrosive materials and provided with dimpled drip perforations which allows water to drip over each of the tubes in the core assemblies with maximum saturation of the knitted sleeve material. The water which is pumped from the lower basin, flows through a filter media which is a matted-glass fiber material specifically selected to provide excellent filtration with a minimum pressure drop.

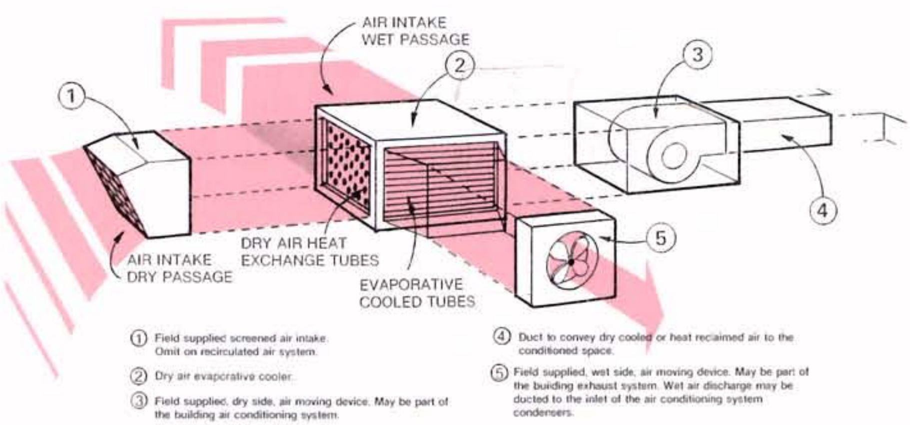


Figure 11-1 DRY AIR EVAPORATIVE COOLER EXCHANGER (22)

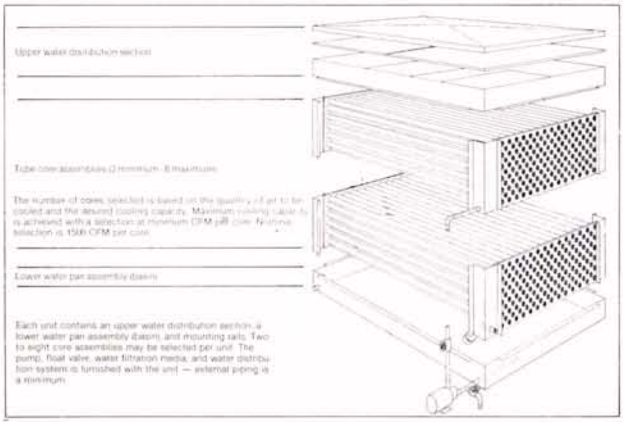


Figure 11-2 DRY AIR COOLER EXCHANGER (22)

B PRINCIPLES OF OPERATION

1. The Transfer Process

a. The Dry Air Cooler Exchanger

In order to obtain the maximum cooling capability of the dry air cooler, the system design should provide that the building exhaust air be ducted to supply the air source for the wet-side air passage. This pre-conditioned exhaust air can thereby serve a useful purpose prior to wasting its energy into the atmosphere. It provides a useable low wet bulb temperature in the cooling season, thus increasing the capacity of the unit for cooling. In lieu of building exhaust air, outside ambient air may be used as the wet-side air source as shown in Figure 11-1.

The wet-side air is cooled as it passes over the outside of the tubes and becomes relatively saturated. Further energy can be saved by using this air to pre-cool air conditioning system condensers when near-saturated air is satisfactory.

During the cooling operation of the unit, the dry bulb temperature leaving the wet-side will be depressed. The depression will vary somewhat depending on dry-side air temperature and the evaporation rate. This depression can be expressed as *approach to wet bulb temperature*. The unit, based on the minimum recommended wet-side air, will achieve an approximate 10°F DB approach. When building exhaust air is provided as the wet-side air source, the maximum cooling capacity is obtained.

During the heating season, the unit acts as a plate type heat exchanger as the outdoor air is pre-heated by recovering the heat of the exhausted building air. This heat recovery is accomplished by deenergizing the unit circulating pump and allowing the knitted tube sleeves to dry out. Automatic controls can energize and deenergize the water pump based on the entering air temperature of the outdoor airstream.

Well over 90% of most cooling system operating hours are at partial load conditions. It is important to evaluate the capability of dry air evaporative cooling units at these partial load conditions. The cooling work performed by the units increases as the wet-bulb temperature decreases. As the outside dry bulb temperature decreases, the outside wet bulb temperature usually decreases also. The factors contribute to a very favorable chain of events:

1. As outside temperatures fall, the total system load decreases.
2. Along with this, the wet bulb temperature has decreased.
3. As the wet bulb temperature falls, the unit capacity increases.

Hence, the units will provide a larger portion (percentage) of the total HVAC system load than during full load operation. As wet bulb temperatures decline, the percentage increases until a point is reached where the HVAC system cooling load and outside wet bulb temperatures can allow for the total cooling to be provided by the dry air evaporative cooler.

b. The Fixed Plate Exchanger

Fixed plate (sensible) energy recovery devices may also employ the unique concept known as indirect evaporative cooling or "dry evaporative cooling." Normal evaporative cooling processes reduce the supply air temperature while increasing the moisture content and are adiabatic. Indirect evaporative cooling reduces the enthalpy level of the supply air and under certain conditions can condense moisture out of the supply air stream. This is accomplished by spraying a mist of water vapor into the exhaust air stream prior to the fixed plate heat exchanger, where it is deposited on the exhaust side surface and is evaporated. The temperature of the plates is lowered to approximately the exhaust wet bulb temperature. The outside air entering the heat exchanger is exposed to the cooled plates and can approach to within approximately 25 percent of the difference between the outside dry bulb and exhaust side wet bulb temperatures. This process is shown schematically in Figure 11-3. The effectiveness of recovery can be in the 50 to 65 percent range as compared to 25 to 40 percent for straight sensible recovery.

2. Cross Contamination

Tube headers of the dry air cooler unit are fabricated of heavy-wall, high-impact polystyrene material. Solvent welding of the close tolerance tube header achieves an excellent water-tight modular core assembly so that cross contamination is virtually eliminated.

C PERFORMANCE

1. Pressure Drops

The wet side of the dry air cooler unit is sized for a minimum air quantity with a pressure drop of 0.15 in. w.g. The dry-side air quantity can vary, but performance tables recommend pressure drops between 0.22 in. w.g. and 0.85 in. w.g. The following example should be used with Figure 11-4:

Example No. 1

Establish the required air quantity to be cooled as well as the design conditions (entering dry-bulb temperature to the dry-side and entering wet-bulb temperature to the wet-side.)

a. Given:

Pre-cool outdoor air.
Wet-side air source = Building exhaust air.
Air quantity to be cooled = 26,500 Cfm
Outdoor air design temp. = 98°DB
Exhaust air design temp. = 62°WB

b. Solution:

Minimum core usage - $\dfrac{26{,}500 \text{ Cfm}}{2{,}000 \text{ Cfm/core}} = 13.25$

Use 14 cores - $\dfrac{26{,}500 \text{ Cfm}}{14 \text{ cores}} = 1890$ Cfm/core

Use 2,000 Cfm/core chart - interpolate if desired.

Step 1 .... 36°F ΔT (98° DB − 62° WB)
Step 2 .... 62°WB entering wet-side
Step 3 ... Lvg. dry-side temp. = 82° (ΔT = 16°F.)

(26,500 Cfm) (16°F ΔT) (1.08) = 457,900 Btu/Hr (38.2 tons)

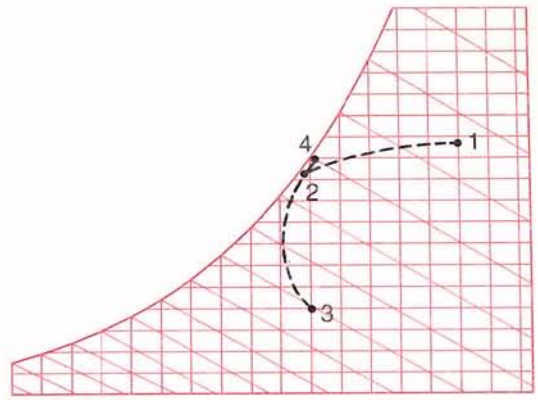


Figure 11-3 THERMODYNAMIC PROCESS FOR INDIRECT EVAPORATIVE COOLING (3)

DRY-SIDE 2000 CFM PER CORE (.85" S.P.)

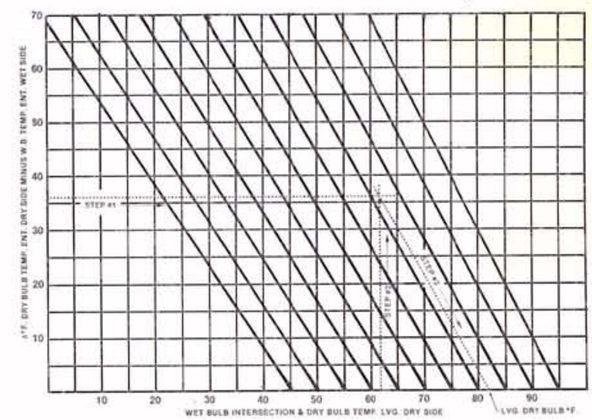


Figure 11-4 DRY AIR COOLER PRESSURE CHART (22)

c. Selection:

(2) 7 core units operating in parallel at 1890 Cfm per core
(1) Dry-side fan: select supply air fan apparatus to handle .85''SP loss units.
(1) Wet-side fan (which is common to 2 parallel units): select at a 900 Cfm per core. (14 cores @ 900 Cfm/core = 12,600 Cfm @ 0.15'' SP)

2. Effectiveness

The heat recovery capacities, per core, of the dry-air cooler with a minimum of 900 cfm of exhaust air to the "wet" airstream are given in the table below:

As with most heat recovery units, tempered (preheated or bypass) air must be provided when temperatures are below 32°F to avoid frost accumulation; hence, 35°F entering air temperature represents a reasonable heat recovery design temperature for the dry air cooler unit.

cfm	Exhaust Air Temp.	Ent. Air Temp. Dry Side	Lvg. Air Temp. Dry Side	Capacity Btu/Hr. Output	% Effectiveness
1000	80°F	35°F	47°F	12,960	38
1500	80°F	35°F	44°F	14,580	43
2000	80°F	35°F	43°F	17,280	51

D APPLICATION

1. Filters

Air filtration of the outside airstream (dry side) is normally required. In very dirty areas it may be advisable to consider filtration of the wet-side air. Polyester media filters are recommended, as they will not be damaged by dampness resulting from rain or fog. High capacity filters will provide long life prior to changeout. Select filters at a face velocity of approximately 400 fpm.

2. Condensation and Freeze-up

In areas where temperatures consistently drop to below freezing, it is advisable to consider freeze protection of the dry air cooler unit pump and basin on the same basis as would be given to air conditioning water towers or evaporative condensers.

Typical methods are an automatic or manual draining of the water pan, the use of electric pan heaters, steam or hot water coils. It is recommended that a 40°F basin water temperature be maintained.

E CAPACITY CONTROLS

Cooling is obtained by energizing the water circulating pump (and wet-side fan if used). A temperature-sensing control element is normally located in the entering (dry-side) outside airstream to control the pump (and wet-side fan). Heat recovery (building exhaust air across the wet side) is obtained by thermostatically turning off the water pump. When exhaust air temperatures are higher than entering outdoor air temperatures, automatic heat recovery operation is obtained by deenergizing the pump. Many other varieties of control operation may be applied, such as face and bypass dampers, etc.

F MAINTENANCE

The required maintenance of the dry air cooler is minimal. The only moving part to be serviced is the basin water pump. Periodically examine the water filter media and change when necessary. Check the operation and level of the float valve, and check all control valves and thermostats. Annually clean the flush basin.

Water treatment is normally not required, however, installation of the unit in some areas may require a periodic or continuous water treatment program to obtain maximum performance. Algae growth may be non-existent in some areas, and quite severe in other areas. Periodic treatment can control the worst of algae problems. In areas where the make-up water has a high percentage of dissolved solids, a program to prevent scaling should be considered. Mineral scale deposits in the unit are minimal due to the small temperature difference between the tube and water.

SECTION XII
THERMAL TRANSFER FLUIDS

Most of the discussion in this section was taken from a paper entitled "Design and Operational Consideration for High Temperature Organic Heat Transfer Systems" prepared by W.F. Seifert and Dr. L.L. Jackson of the Dow Chemical Company, and C.E. Sech of Charles E. Sech Associates, Inc. Consulting & Design Engineers, and presented at the 71st National AIChE Meeting in Dallas, Texas. The edited material is reprinted with the permission of the American Institute of Chemical Engineers.

Practically every designer will become involved in the selection of a heat transfer fluid at one time or another during his career. The most common heat transfer fluids are steam and water, and if the temperature is above the freezing point of water (32° F) and below about 350° F, the choice is usually between these two fluids. On the other hand, if the temperature of application is below the freezing point of water or above about 350° F, it is necessary, or at least desirable, to consider other fluids.

For temperatures below the freezing point of water the most common heat transfer fluids are air, refrigerants such as halogenated hydrocarbons, ammonia, brines and/or solutions of glycol and water.

As temperatures increase above 350° F, the vapor pressure of water increases rapidly, and the problems of structural strength for processing equipment becomes more and more severe. Thus with high temperature systems it becomes increasingly important to consider fluids with vapor pressures lower than water.

Some of the more frequently used high temperature organic heat transfer fluids available today are shown in Table 12-1.

In the design of a high temperature organic heat transfer system, the engineer has two key problem areas to evaluate. These are: the selection of the heat transfer media and the system design and selection of equipment.

A HEAT TRANSFER MEDIA EVALUATION

Once the decision has been made to use an organic high temperature heat transfer fluid, the designer needs to select a material that will perform satisfactorily and safely at the process temperatures required. To do this, one can draw on past experience or make relative comparisons of the existing fluids by compiling the data available from fluid manufacturers. The important factors that must be evaluated in selecting a high temperature heat transfer fluid can be categorized into the following areas.

1. Toxicity and Environmental Ecology

Toxicity and ecology are, of course, extremely important from both an operating standpoint and a process standpoint. There is always a chance that a heat transfer fluid may find its way through packing glands on valves, pumps, heat exchangers, etc., and if this happens operators, maintenance men, and the environment in general will be exposed to the fluid. More ecological information for evaluating this subject is being made available from many fluid manufacturers today.

2. Corrosiveness to Materials of Construction

In general, a heat transfer fluid should be noncorrosive to mild steel. Otherwise the first cost of the equipment can become prohibitively high unless there would be extenuating circumstances. It should be noted that all of the chlorinated compounds recognized as heat transfer fluids, are essentially noncorrosive to mild steel as long as all traces of water are kept out of the system and as long as the fluid is not overheated. If halogenated materials are overheated either by a bulk temperature higher than the recommended maximum limit or by localized hot spots in a furnace, hydrogen chloride gas will be evolved. The hydrogen chloride gas will remain relatively noncorrosive to mild steel as long as the system is kept absolutely dry, but if traces of water are present, the hydrochloric acid formed will be extremely corrosive, particularly at elevated temperatures. Chlorides can also cause stress corrosion of stainless steels if water is present.

Table 12-1 HEAT TRANSFER FLUIDS

Registered Tradename of Fluid	Composition	Producer	Usable Temperature Range	Flash Point °F	Fire Point °F	Specific Heat Btu/lb°F 200°F	Specific Heat Btu/lb°F 400°F	Viscosity Lb/ft hr 200°F	Viscosity Lb/ft hr 400°F	Thermal Conductivity Btu/hr ft² (°F/ft) 200°F	Thermal Conductivity Btu/hr ft² (°F/ft) 400°F	Density Lb/ft³ 200°F	Density Lb/ft³ 400°F
Dowtherm A	diphenyl-diphenyl oxide eutectic	Dow Chemical Co.	60°F to 750°F	255°F c.o.c.	275°F c.o.c.	0.43	0.50	2.57	0.90	0.077	0.068	62.5	56.4
Dowtherm G	di & tri aryl ethers	Dow Chemical Co.	20°F to 700°F	305°F c.o.c.	315°F c.o.c.	0.42	0.48	7.02	1.65	0.074	0.072	65.5	60.0
Dowtherm J	alkylated aromatic	Dow Chemical Co.	−100°F to 575°F	145°F c.o.c.	155°F c.o.c.	0.49	0.60	0.90	0.34	0.073	0.068	50.4	44.1
Dowtherm LF	aromatic blend	Dow Chemical Co.	−25°F to 600°F	260°F c.o.c.	280°F c.o.c.	0.43	0.51	2.66	0.92	0.077	0.067	61.2	55.9
Dowtherm SR-1	inhibited ethylene glycol	Dow Chemical Co.	−40°F to 300°F	250°F (100%) p.m.c.c.	250°F (100%) c.o.c.	0.88*	—	2.18*	—	0.243*	—	62.1	—
Dowfrost	inhibited propylene glycol	Dow Chemical Co.	−40°F to 300°F	214°F t.c.c.	220°F c.o.c.	0.92*	—	2.18*	—	0.220*	—	64.1	—
Mobiltherm 600	mineral oil	Mobil Oil Co.	10°F to 600°F	415°F c.o.c.	N/A	0.46	0.56	16.2	2.78	0.066	0.062	52.0	47.9
Mobiltherm 603	mineral oil	Mobil Oil Co.	20°F to 600°F	405°F c.o.c.	N/A	0.54	0.65	9.92	2.27	0.074	0.069	44.0	40.2
Mobiltherm Light	mineral oil	Mobil Oil Co.	−15°F to 400°F	250°F c.o.c.	N/A	0.44	0.53	3.82	1.38	0.065	0.061	55.6	52.0
Therminol 44	modified ester base	Monsanto	−50°F to 425°F	405°F c.o.c.	438°F c.o.c.	0.51	0.57	2.34	0.66	0.076	0.065	54.1	48.7
Therminol 55	synthetic hydrocarbon	Mosanto	0 to 600°F	355°F c.o.c.	410°F c.o.c.	0.52	0.62	10.3	2.18	0.075	0.069	52.4	47.8
Therminol 60	polyaromatic compound	Mosanto	−60°F to 600°F	310°F c.o.c.	320°F c.o.c.	0.44	0.54	4.0	1.3	0.073	0.068	59.1	54.6
Therminol 66	modified terphenyl	Mosanto	15°F to 650°F	355°F c.o.c.	382°F c.o.c.	0.44	0.53	10.0	2.1	0.067	0.061	59.8	55.0
Therminol 88	mixed terphenyl	Mosanto	300°F to 750°F	375°F c.o.c.	460°F c.o.c.	Solid	0.50	Solid	2.0	Solid	0.071	Solid	59.9
Therminol VP-1	diphenyl oxide eutectic	Mosanto	60°F to 750°F	255°F c.o.c.	275°F c.o.c.	0.42	0.49	2.66	0.87	0.075	0.065	62.7	56.7
Thermia A	mineral oil	Shell Oil Co.	−50°F to 350°F	300°F c.o.c.	N/A	0.51	—	N/A	—	0.072	—	52.4	—
Thermia C	mineral oil	Shell Oil Co.	15°F to 600°F	455°F c.o.c.	N/A	0.51	0.60	N/A	N/A	0.073	0.069	51.8	47.5

* — 50% solution by weight. N/A — Information not available.
Note: Consult manufacturer for detailed application recommendations.

3. Flammability

Lack of flammability is always vital whenever there is a chance that a fluid may not be completely separated from all sources of ignition. Some of the chlorinated compounds such as chlorinated biphenyls are fire resistant because they will not support combustion due to the chlorination. However, if they are heated to a sufficiently high temperature they exhibit a flash point and an explosive range. They will burn if subjected to the ignition conditions encountered in the fire box of a fired heater. Thus, organic fluids must not be exposed to a source of ignition.

While nonchlorinated heat transfer fluids will burn, this factor presents no problems if they are contained properly. If, due to some unusual occurrence, they leak from the system into a space other than the fire box of a furnace, they will almost invariably, if not always, be below their auto ignition temperatures before they come in contact with air. Thus there must be a source of ignition before a leak outside a fire box can be serious. Moreover, combustion requires a mixture of air and vapors having a concentration within the flammability limits of the fluid. For continued burning, the liquid must be at temperatures higher than its fire point.

B THERMAL STABILITY AND CHEMICAL STRUCTURE

Several generalizations can be made about thermal stability and degradation of organic heat transfer media.

1. In comparing classes of compounds, aromatic materials have thermal stability generally superior to aliphatic compounds.
2. For commercial products, the recommended maximum operating temperature is a rough measure of relative thermal stability.
3. Polymer formation is detrimental particularly if the polymerization is exothermic. Polymers increase the viscosity of a fluid and promote carbonization leading to inefficiency and the potential failure of the heater.
4. Fluid degradation should produce a minimum of volatile materials such as hydrogen, ethylene and other light hydrocarbons. These decomposition products will increase operating losses and they are a safety hazard (fire and toxicity) in a vented heating loop.
5. Degradation should not produce reactive or corrosive compounds. Acids, such as HCl, are corrosive, toxic, and they accelerate fluid breakdown at high temperatures. Cracking products such as olefins will polymerize under operating conditions.
6. Oxidative stability can be an important factor if air is present at high temperatures.

Thermal degradation of an organic heat transfer fluid is important to the extent that it influences the functionality of the fluid. Decomposition is manifested by the appearance of low boiling components and/or high boiling materials (including polymeric tars). Low boiling materials in a fluid may cause excessive venting and high make-up rates may be encountered. In contrast, high boiling materials and polymers in a fluid will result in higher viscosities. Increasing viscosity will accelerate the degradation process due to less efficient heat transfer and high film temperature. Excellent fluid stability is a critical ingredient in arriving at a reliable, efficient, and safe high temperature heat transfer system.

C EQUIPMENT DESIGN AND OPERATING CONSIDERATIONS

1. Fired or Electrical Heater Design

Heaters for organic heat transfer fluids generally are designed with lower heat fluxes than heaters for water or steam boilers. Fired steam boilers and water heaters are frequently designed with heated fluxes as high as 40,000 to 50,000 Btu/hr-ft^2. Fire heaters for organic heat transfer fluids are usually designed with average heat fluxes ranging from 5,000 to 12,000 Btu/hr-ft^2. The actual allowable heat flux is usually limited by a maximum film temperature, and this in turn is dependent upon factors such as maximum bulk temperatures, velocity of the fluid across the heat transfer surface, uniformity of heat distribution in the furnace, and heat transfer properties of the fluid in question. Precautions must also be taken to guard against excessive accumulations of high molecular weight decomposition products, corrosive gases, etc. In the case of fluids which can vaporize within their respective recommended operating temperature ranges, attention must also be paid to the percentage of liquid which will be vaporized. On one hand vaporization can be beneficial in that it tends to increase the film heat transfer coefficient and also can vaporize at a constant temperature which tends to limit the maximum film temperature for a given set of conditions. On the other hand, however, if too much fluid is vaporized so that the heat transfer surface is, for all practical purposes, blanketed with vapor rather than liquid, the film heat transfer coefficient will be reduced very rapidly and dangerously high surface temperatures can develop resulting in severe fluid degradation and mechanical failure.

All other things being equal, any organic heat transfer fluid degrades in proportion to the temperature.

When considering operating a fluid at temperatures higher than the manufacturer's recommended maximum temperatures, it is extremely important to guard against approaching the critical temperatures that may cause the fluid to carbonize and form hard carbon scale on the heat transfer surface. When this happens the hard carbon tends to insulate the surface and decreasing the rate of heat transfer at this point. This results in an increased metal temperature under and around the edges of the carbon, thus accelerating the rate of hard carbon scale formation. If this condition is not discovered and corrected, it will lead to overheating and ultimate destruction and rupture of the heat transfer surface.

The formation of soft or particulate carbon as well as high molecular weight polymers need not be serious as long as the condition is recognized and kept under control. It is important, however, that the fluid be sampled periodically in accordance with the recommendations of the fluid manufacturer and the heater manufacturer. With fluids having an atmospheric boiling point outside their recommended operating temperature ranges, the quantity of soluble degradation products is usually determined by measuring the viscosity of the sample at a given temperature.

In addition to these high molecular weight products which are soluble in the fluid, there will frequently be an accumulation of fine particulate organic insolubles as well as some particles of mill scale in the sample. The percentage of this sediment in the fluid must also be kept within limits recommended by the manufacturer of the fluid.

Whenever the decomposition products, the insolubles, or the acidity of the heat transfer media exceeds the established recommendations, it is important that the fluid in the system be replaced or reclaimed. Some of the manufacturers of high temperature organic heat transfer fluids offer a reclamation service for their products. On stream purification units are available to reclaim some of the fluids semicontinuously in the field without returning the fluid to the manufacturer. However, it is important that the cost of reclaiming the fluid be included in evaluating the overall operating cost. If any given fluid is overstressed severely above the recommended maximum temperature, the decomposition rate will become quite high. If this happens, the cost of removing the high molecular weight polymers, the cost of replacement fluid, and, especially the cost of lost production can be appreciable.

The most frequent causes of excessive fluid degradation operating difficulties in a fired heater are:

a. Flame Impingement

Flame impingement on a heat transfer surface will invariably cause trouble if not corrected. The results will be lower heat efficiencies, loss of capacity, and ultimately tube failures. Flame impingement can be caused by using improper burners, improper adjustment of a burner, or by poor furnace design.

b. Low Circulation Rate

1. Forced circulation heater

Poor circulation rates through the heater can be the result of a power failure, instrumentation failure, or pump cavitation due to either low liquid level or system contamination with a low vapor pressure material.

2. Natural circulation heater

Poor flow in a natural circulation heater can be attributed to low fluid level in the heater, insolubles restricting the flow in the tubes or uneven firing of the burners of the furnace.

c. High Heat Fluxes

When a heater is operated above its rated design capacity, the film temperatures that the fluid is exposed to will be in excess of the recommended design value. To insure that excessive degradation of the fluid does not take place, the heater manufacturer should be consulted to determine their recommendations for circulation rate and burner modifications.

d. Heat Transfer Fluid Contamination

Contamination can be caused by a process material leaking into or unintentionally being charged into the high temperature heating system. This situation can cause significant problems depending on the thermal stability and quantity of process fluid added.

Most any type of organic contaminants are potential hazards because of their lower thermal stability. If the contamination is excessive, hard carbon scale can deposit on the heat transfer surface of the heater; once this hard carbon deposit starts, degradation of the heat transfer fluid will be accelerated due to the excessive film temperatures. Even if a hard carbon scale is not formed on a heat transfer surface, organic contaminants can accelerate the decomposition rates for a heat transfer fluid.

If the organic contaminant is acid in nature, such as a fatty acid, severe corrosion of mild steel (in the presence of moisture) can result. Likewise, with inorganic acids or strong alkalies severe corrosion of mild steel can occur. The corrosion is usually most severe in the hottest part of the system, the tubes of the heater, unless the corrosive material is quite volatile. In the case of HCl, for example, regardless of whether the acid has leaked into the heat transfer fluid or whether it is formed by thermal decomposition, the corrosion always appears to be most severe in the vapor or gas space *above* the liquid. This is probably because these are the areas where traces of moisture are most likely to collect and the moisture causes the HCl to become very corrosive.

2. Heater and Vaporizer

As is the case with any refractory lined furnace, boiler, etc., heaters for organic heat transfer fluids should be started and operated at low heat input when they are started initially or whenever they have been shut down for an extended period. New refractory or refractory in a furnace which has been standing idle will always contain considerable quantities of absorbed water. Unless the refractory is heated slowly so the water can escape slowly as it is vaporized, internal pressure will develop causing the refractory to spall.

Additionally, fluid manufacturers recommend maximum heat up rates for cold systems to protect their fluids and heater. The viscosities of all of the organic fluids increase as their temperatures decrease until they become solid or so viscous that they are difficult to pump. The increase in viscosity causes a decrease in the film heat transfer coefficient in two different ways. First, it reduces the amount of flow through the heater for a given pump head, which decreases the film heat transfer coefficient because of a decrease in velocity. Second, the increased viscosity causes a reduction in the film heat transfer coefficient for a given flow rate or velocity. These two effects are additive and, unless care is exercised in starting up a cold system, some extremely high film temperature rises can be experienced resulting in damage to the fluid and/or equipment. For a given heat flux the film temperature rise is, of course, inversely proportional to the film heat transfer coefficient.

3. Piping and Pump Seals

It is common practice to use welded joints wherever possible and to keep flanged joints at a minimum. Prior to the development of the spiral wound stainless steel-asbestos gaskets, it was a standard practice to use series 30 steel raised face flanges at all flanged joints. The reason for the heavy flanges, in many cases, was not to withstand high pressures but to be able to achieve the gasket pressures required for a leak proof joint without warping the flanges. Although many users still prefer the series 30 flanges in all instances because of their ability to not yield under cycling thermal stresses and to therefore maintain tighter joints, the spiral wound gaskets have made it possible to maintain tight joints in many instances with series 15 flanges.

Pump glands can be sealed either with special pump packings for high temperature service or with mechanical seals. Mechanical seals for high temperature service are becoming more and more popular. When the system is clean, free of abrasive sediment and contaminants, mechanical seals have been known to function effectively without leakage and without attention for as long as five years. Some of the pump and pump seal manufacturers recommend that mechanical seals be flushed with a stream of clean cooled fluid taken from the pump discharge. Filters or small cyclone separators may be used to clean the fluid. In some cases, a small water cooled heat exchanger is used to cool the fluid before it is put into the flushing connection on the seal chamber. Until recently seal and pump manufacturers have recommended water jacketed seal chambers when pumping fluids at temperatures above about 450°F. The purpose of this was to protect the seal rings made of Teflon resin or Viton fluoro-elastomer which formed a seal between the pump shaft or shaft sleeve and the portion of the seal which was free to slide on the shaft. Many of the seal manufacturers are now offering a bellows type seal for use without cooling at temperatures considerably above 450°F. Seal manufacturers are also offering seals with graphite rings in place of Teflon rings to seal the sliding portion of the mechanical seal to the pump shaft. It is claimed that these seals can also be operated at temperatures above 350°F without external cooling. Most of the mechanical seals used in pumps for handling high temperature organic heat transfer fluids have a dense carbon or tungsten carbide stationary ring which mates with a rotating ring having either a Stellite or a tungsten carbide face, depending on the seal manufacture. Experience indicates there is a limiting temperature at which these materials can be used without external cooling or without flushing them with a cooled stream of fluid.

4. Expansion Tank

The use of an expansion tank, which allows a "closed" system, is required to: ensure that the system is filled whenever the mean fluid temperature is less than the fill temperature; provide room for volumetric expansion whenever the mean fluid temperature is greater than the fill temperature; prevent oxidation and subsequent corrosion when a glycol solution is used, as would happen in an "open" system.

With many of the organic fluids, it is necessary to charge or "blanket" the tank(s) with an inert gas such as nitrogen, or to use a "cold seal" type expansion system.

5. Summary

Regardless of the intended use of a thermal transfer fluid, contact the manufacturer of the selected fluid for detailed recommendations and information. There are many additional properties of the fluids that are not listed in Tabel 12-1 which can affect the performance of the system or possibly create serious maintenance problems. Every component in the piping system that comes in contact with the fluid (or its vapor) must be evaluated.

SECTION XIII
ECONOMICS OF ENERGY RECOVERY

A JUSTIFICATION OF ENERGY RECOVERY

1. Existing Buildings

The majority of the buildings now in use were designed and constructed when most forms of energy were readily available and inexpensive. The structures, the electrical and mechanical systems, and the landscaping were designed to minimize initial costs and to fit into the contemporary architectural mode. As a result, the majority of these buildings are now energy inefficient as most are over ventilated, overheated (in winter), overcooled (in summer) and overlighted (while both occupied and unoccupied).

In the industrial sector, waste heat from all types of processes is being exhausted to the atmosphere, while nearby office buildings and warehouses of the same plant location use other fuels as energy for the building environmental systems.

However, before energy recovery is attempted, all efforts should be concentrated on conservation of energy by using good maintenance procedures on the environmental systems and realistic use of the building systems as outlined in the SMACNA "Guidelines for Energy Conservation in Existing Buildings". Ventilation loads should be reduced to the minimum required by local codes unless there are unusual conditions. The remaining energy that is being discarded throughout the year can then be analyzed against the cost of the installation and operation of recovery systems.

In existing buildings, the proposed use of an energy recovery system would depend on many factors:

a. The physical space requirements and the distances between airstreams.

b. The temperature and latent heat differences between the airstreams.

c. The mass flow rates of the airstreams.

d. The efficiency of the recovery devices.

e. The additional energy required to operate the recovery system.

f. Modifications of the building environmental systems.

When all of the above data is used in a building energy analysis and there is a yearly net savings in energy consumption, it would seem that an energy recovery system would be feasible if the "payback period" was reasonable or additional energy was unattainable.

The following are some "rule-of-thumb" indicators that are being used for a quick (but not too accurate) analysis of when energy recovery might be considered:

a. Building exhaust systems when one or more of the following occur:
   1. The flow rate is over 1000 cfm.
   2. Heating degree days are over 3500.
   3. There are 8000 cooling degree hours above 78°F.
   4. There are 12,000 wet bulb degree hours above 66°F wet bulb temperature.

b. Incinerators consuming 1000 pounds of solid waste per day or greater.

2. New Buildings

ASHRAE Standard 90-75 "Energy Conservation in New Building Design", contains the following recommendation for use of energy recovery systems:

> "5.9 *Energy Recovery*
> It is *recommended* that consideration be given to the use of recovery systems which will conserve energy (provided the amount expended is less than the amount recovered) when the energy transfer potential and the operating hours are considered."

B BENEFITS OF ENERGY RECOVERY

In general, the motivation for building owners to invest in energy recovery is that they expect the resulting benefits to exceed investment costs. Factors that have recently made such investments attractive are rising fuel costs and curtailment of regular fuel sources which threaten production cutbacks and changeover to other energy sources. In addition, mandatory pollution controls and rising labor costs cut into profits and cause firms to look more closely for ways to control costs.

The kinds of potential benefits which may result from energy recovery are:

1. Fuel savings
2. Reduced size, hence lower capital cost, of heating/cooling equipment
3. Reduced maintenance costs for existing equipment
4. Reduced costs of production labor
5. Pollution abatement
6. Improved product
7. Revenue from sales of recovered heat or energy

These benefits were suggested by a preliminary look at existing applications. However, only "fuel savings" was found in every case examined. The other benefits, "savings in capital and maintenance costs on existing equipment, pollution abatement, labor savings, product improvement, and revenue from sales of recovered heat", appear limited to certain applications.

1. Fuel Savings

Fuel savings result when waste heat is recovered and used in substitution for newly generated heat or energy. For example, heat from stack flue gas may be recovered by an energy recovery device and used to preheat the input water, thereby reducing the amount of fuel needed for steam generation.

2. Lower Capital Costs

Savings in capital costs for certain items of existing equipment (i.e., regular equipment apart from that required for waste heat recovery) may be possible if recovered heat reduces the required capacity of the heating and/or cooling equipment. For example, installation of rooftop energy recovery equipment on buildings with high ventilation requirements can enable significant reductions in the size and cost of the building's heating and cooling system. This potential for savings is not limited to new construction when existing equipment needs to be replaced.

3. Reduced Maintenance

Reduced maintenance and repair on certain items of existing equipment may, in some instances, be a further benefit of investment in energy recovery. The principal impact on the maintenance of existing equipment is likely to result from the planning, engineering, and installation phases of investment in energy recovery, when the existing equipment and plant processes are often scrutinized. Existing faults may be identified and corrected; and improved maintenance practices may be extended to existing equipment.

4. Lower Labor Costs

Another kind of benefit which may result from investment in energy recovery systems, is savings in labor costs. Labor savings can result, for example, from a lowering of industrial furnace changeover time (i.e., the time needed to alter furnace temperatures required for a change in production use) by preheating combustion air with waste heat. Savings may also result from faster furnace start-ups, accomplished by similar means. By reducing the amount of labor "downtime", unit labor costs are reduced. (A tradeoff may exist between idling the furnace at higher temperatures during off-duty hours and incurring labor "downtime" during furnace start-ups. If the existing practice is to idle the furnace at high temperatures in order to avoid "downtime", the savings from using waste heat recovery to preheat air would be in terms of fuel reductions rather than lower labor costs.)

5. Pollution Abatement

Pollution abatement is a beneficial side effect which may result from recovery of waste heat. For example, the pollution abatement process in textile plants will often be facilitated by waste heat recovery. Pollutants (plasticizers) are usually collected by circulating air from the ovens (where fabrics are coated or backed with other materials) through electrostatic precipitators. The air must, however, be cooled to accomplish collection of pollutants. If it were not for heat recovery, it would be necessary to cool the air by other means, which would generally entail additional fuel consumption. Thus, there is a twofold impact on fuel use from this application of heat recovery. Another instance of pollution abatement, as a side effect of waste heat recovery, occurs if pollutants are reduced by the higher furnace temperatures resulting from preheating combustion air with waste heat.

The pollution abatement side effects represented by the two preceding examples are distinguishable from the use of systems to recover heat from a pollution abatement process, where recovery of heat does not in itself contribute to pollution abatement. For example, the recovery of waste heat from the incineration of polluting fumes is a method of reducing the cost of pollution abatement by producing a useful by-product from the abatement process. However, the waste heat recovery does not itself contribute to the pollution abatement process and therefore does not yield multiple benefits; the only benefit is the value of the fuel savings from using the recovered heat in other processes.

6. Product Improvement

Product improvement is a further potential side effect of energy recovery. For example, by achieving a more stable furnace temperature and a reduction in furnace aeration, use of a recuperator to preheat combustion air may reduce the undesirable scaling of metal products. In absence of preheating combustion air, it would be necessary to invest in improvements to furnace controls or in some other means of preventing scaling; to secure the same product quality.

7. Revenue-Generating

A final potential benefit from energy recovery, as suggested by existing applications, is the generation of revenue from sales of recovered waste heat or energy. In some cases, the recoverable waste heat cannot all be used by the plant itself. Recovery may still be advantageous if there are adjoining plants which are willing to purchase the recovered heat. In this case, the potential benefits are revenue-generating, rather than cost-reducing, and would be measurable in terms of dollars of revenue received.

8. Existing Performance Records

With information such as records of past operating levels and expenses, the efficiency of the proposed energy recovery equipment, the level of expected furnace operation, the demand for recycled heat, and the expected price of fuels, it should be possible to make a close estimate of the savings in fuel costs that would result from substituting waste heat for newly-generated heat or energy. Certain of other potential benefits, such as labor cost savings and product improvement, might be more difficult to estimate. **However do not use past performance data for a building furnished by others without verification, particularly when firm price contracts are involved.**

C COMPUTING FUEL SAVINGS

Before any economic analysis can be made on a heat recovery system, four basic steps must be completed:

1. Determine design and operating conditions for flows, temperatures, allowable pressure drops, hours of operation (including different seasons), type and cost of fuels used, and efficiency of fuel-using equipment.
2. Using design conditions, size and the energy recovery system, and determine the amount of recovered energy.
3. Calculate fuel saved for a given time period — usually one year.
4. Determine installed cost of system.

Energy recovery equipment usually accounts for a small portion of the total installation cost. The customer will eventually need accurate cost estimates which usually must be obtained from an installing contractor. Only after the equipment energy consumption rates, the installed cost of the recovery system, and the hours of operation are known, can the economics of the installation be determined.

Some energy recovery systems operate independent of ambient conditions. But often it is required that make-up air be returned to a plant or process. When this is the case, the heat recovered should be calculated using the degree-day method, which takes into account daily variations in the ambient temperature. The economic analysis in this section will be developed around the following two methods:

1. Heat recovery — independent of ambient temperature
2. Heat recovery — dependent on ambient temperature

The basic calculation in any economic analysis of an energy recovery system is the determination of fuel saved for a given time period. The equation for determining this value is:

Equation 13-1

$$F = \frac{Q \times N \times (ETD)_a}{E \times C \times (ETD)}$$

F = Units of fuel saved during time period.
C = Heat value of unit of fuel.
Q = Heat recovered — Btu/hr at design conditions.

N = Number of hours heat recovery system operates in the time period used in the analysis (e.g. for an Oct. 1 to May 1 heating season— 212 days × 24 hrs/day = 5,088 hours)

E = Efficiency of fuel using device or system over the time period (express as a decimal)

$(ETD)_a$ = Entering temperature difference between average ambient (t_{1a}) for the specified time period and the hot inlet temperature ($T_1 - t_{1a}$) – °F

(ETD) = Entering temperature difference at design conditions ($T_1 - t_1$) – °F

T_1 = Temperature of the hot gas or air entering the heat recovery unit at design conditions – °F

t_1 = Temperature of the cold fluid (air to liquid) entering the heat recovery unit at design conditions – °F

t_{1a} = Average ambient temperature for time period (N) – °F

For a one year period, t_{1a} is calculated as follows: Referring to Table 13-1, which gives the degree-day values for major cities throughout the United States, Canada, and Australia, select the appropriate

Table 13-1 AVERAGE DEGREE DAYS, CITIES IN UNITED STATES, CANADA, AND AUSTRALIA (Base 65°F)

State	Station	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	April	May	June	Yearly Total
Ala.	Birmingham	0	0	6	93	363	555	592	462	363	108	9	0	2551
Ariz.	Phoenix	0	0	0	22	234	415	474	328	217	75	0	0	1765
Ark.	Little Rock	0	0	9	127	465	716	756	577	434	126	9	0	3219
Calif.	Los Angeles	0	0	6	31	132	229	310	230	202	123	68	18	1349
	San Francisco ...	192	174	102	118	231	388	443	336	319	279	239	180	3001
Colo.	Denver	0	0	90	366	714	905	1004	851	800	492	254	48	5524
Conn.	Hartford	0	12	117	394	714	1101	1190	1042	908	519	205	33	6235
Del.	Wilmington	0	0	51	270	588	927	980	874	735	387	112	6	4930
D.C.	Washington	0	0	33	217	519	834	871	762	626	288	74	0	4224
Fla.	Miami Beach	0	0	0	0	0	40	56	36	9	0	0	0	141
	Pensacola	0	0	0	19	195	353	400	277	183	36	0	0	1463
Ga.	Atlanta	0	0	18	124	417	648	636	518	428	147	25	0	2961
	Savannah	0	0	0	47	246	437	437	353	254	45	0	0	1819
Idaho	Boise	0	0	132	415	792	1017	1113	854	722	438	245	81	5809
Ill.	Chicago	0	12	117	381	807	1166	1265	1086	939	534	260	72	6639
Ind.	Indianapolis	0	0	90	316	723	1051	1113	949	809	432	177	39	5699
Iowa	Des Moines	0	6	96	363	828	1225	1370	1137	915	438	180	30	6588
Kans.	Topeka	0	0	57	270	672	980	1122	893	722	330	124	12	5182
Ky.	Louisville	0	0	54	248	609	890	930	818	682	315	105	9	4660
La.	New Orleans	0	0	0	19	192	322	363	258	192	39	0	0	1385
Me.	Portland	12	53	195	508	807	1215	1339	1182	1042	675	372	111	7511
Md.	Baltimore	0	0	48	264	585	905	936	820	679	327	90	0	4654
Mass.	Boston	0	9	60	316	603	983	1088	972	846	513	208	36	5634
	Worcester	6	34	147	450	774	1172	1271	1123	998	612	304	78	6969
Mich.	Detroit	0	0	87	360	738	1088	1181	1058	936	522	220	42	6232
Minn.	Minneapolis	22	31	189	505	1014	1454	1631	1380	1166	621	288	81	8382
Miss.	Jackson	0	0	0	65	315	502	546	414	310	87	0	0	2239
Mo.	St. Louis	0	0	60	251	627	936	1026	848	704	312	121	15	4900
Mont.	Billings	6	15	186	487	897	1135	1296	1100	970	570	285	102	7049
	Helena	31	59	294	601	1002	1265	1438	1170	1042	651	381	195	8129
Neb.	Lincoln	0	6	75	301	726	1066	1237	1016	834	402	171	30	5864
	Omaha	0	12	105	357	828	1175	1355	1126	939	465	208	42	6612
Nev.	Las Vegas	0	0	0	78	387	617	688	487	335	111	6	0	2709
	Reno	43	87	204	490	801	1026	1073	823	729	510	357	189	6332
N.H.	Concord	6	50	177	505	822	1240	1358	1184	1032	636	298	75	7383
N.J.	Atlantic City	0	0	39	251	549	880	936	848	741	420	133	15	4812
N.M.	Albuquerque	0	0	12	229	642	868	930	703	595	288	81	0	4348
N.Y.	Albany	0	19	138	440	777	1194	1311	1156	992	564	239	45	6875
	Buffalo	19	37	141	440	777	1156	1256	1145	1039	645	329	78	7062
	New York	0	0	30	233	540	902	986	885	760	408	118	9	4871

degree-day value. For example, 7635 is the degree-day value for Milwaukee, Wisconsin. Divide the value for Milwaukee by the total days in a year. 7635/365 = 20.91°F which is the difference between the average yearly temperature in Milwaukee and 65°F. (Degree-day tables are based on a difference between average daily temperatures and a nominal 65°F indoor temperature.) The average ambient temperature is 65°F minus the calculated difference, or in Milwaukee, t_{1a} = 65° − 20.91° = 44.09°F.

The following examples illustrate how to calculate fuel cost savings that result from the installation of energy recovery systems.

Example No. 1

A paint drying oven which operates 24 hours/day, 365 days/year, is discharging 10,000 scfm of air at 200°F. This waste heat is to be used for comfort heating. Factory air enters the heat recovery unit at 60°. No make-up air is required. The system is used only during the heating season — October 1 to May 1. An energy recovery system is to be installed which is predicted to recover heat from the oven exhaust at a rate of 473,500 Btu/hr at design conditions. The customer pays $0.12/therm for natural gas to heat his plant, and the average efficiency of the heating sys-

Table 13-1 AVERAGE DEGREE DAYS, CITIES IN UNITED STATES, CANADA, AND AUSTRALIA (Base 65°F)

State	Station	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	April	May	June	Yearly Total
N.C.	Raleigh	0	0	21	164	450	716	725	616	487	180	34	0	3393
N.D.	Bismarck	34	28	222	577	1083	1463	1708	1442	1203	645	329	117	8851
Ohio	Cincinnati	0	0	39	208	558	862	915	790	642	294	96	6	4410
	Columbus	0	0	57	285	651	977	1032	902	760	396	136	15	5211
Okla.	Tulsa	0	0	18	158	522	787	893	683	539	213	47	0	3860
Ore.	Portland	25	28	114	335	597	735	825	644	586	396	245	105	4635
Pa.	Philadelphia	0	0	60	297	620	965	1016	889	747	392	118	40	5144
	Pittsburgh	0	9	105	375	726	1063	1119	1002	874	480	195	39	5987
R.I.	Providence	0	16	96	372	660	1023	1110	988	868	534	236	51	5954
S.C.	Columbia	0	0	0	84	345	577	570	470	357	81	0	0	2484
S.D.	Sioux Falls	19	25	168	462	972	1361	1544	1285	1082	573	270	78	7839
Tenn.	Nashville	0	0	30	158	495	732	778	644	512	189	40	0	3578
Tex.	Amarillo	0	0	18	205	570	797	877	664	546	252	56	0	3985
	Dallas	0	0	0	62	321	524	601	440	319	90	6	0	2363
	Houston	0	0	0	6	183	307	384	288	192	36	0	0	1396
Utah	Salt Lake City	0	0	81	419	849	1082	1172	910	763	459	233	84	6052
Vt.	Burlington	28	65	207	539	891	1349	1513	1333	1187	714	353	90	8269
Va.	Richmond	0	0	36	214	495	784	815	703	546	219	53	0	3865
Wash.	Seattle	50	47	129	329	543	657	738	599	577	396	242	117	4424
	Spokane	9	25	168	493	879	1082	1231	980	834	531	288	135	6655
W. Va.	Charleston	0	0	63	254	591	865	880	770	648	300	96	9	4476
Wisc.	Milwaukee	43	47	174	471	876	1252	1376	1193	1054	642	372	135	7635
Wyo.	Cheyenne	28	37	219	543	909	1085	1212	1042	1026	702	428	150	7381
Atla.	Calgary	109	186	402	719	1110	1389	1575	1379	1268	798	477	291	9703
B.C.	Vancouver	81	87	219	456	657	787	862	723	676	501	310	156	5515
Man.	Winnipeg	38	71	322	683	1251	1757	2008	1719	1465	813	405	147	10679
N.B.	St. John	109	102	246	527	807	1194	1370	1229	1097	756	490	249	8219
Nfld.	St. John's	186	180	342	651	831	1113	1262	1170	1187	927	710	432	8991
N.S.	Halifax	58	51	180	457	710	1074	1213	1122	1030	742	487	237	7361
Ont.	Toronto	7	18	151	439	760	1111	1233	1119	1013	616	298	62	6827
Que.	Montreal	9	43	165	521	882	1392	1566	1381	1175	684	316	69	8203
Sasks.	Regina	78	93	360	741	1284	1711	1965	1687	1473	804	409	201	10806
Adelaide, Australia														2302
Melbourne, Australia														2675
Perth, Australia														1393
Sidney, Australia														1318

tem is 80%. Find the total amount of gas saved during the heating season.

Solution:

Since recirculated air is used, this application is a heat recovery system which is independent of ambient conditions. In this case, $(ETD)_a$ and (ETD) are equal.

Therefore:

$$F = \frac{Q \times N}{E \times C} \quad \text{(from equation 13-1)}$$

$Q = 473{,}500$ Btu/hr

$N = 5{,}088$ hours (212 day heating season $\times$ 24 hrs/day)

$E = 0.80$

$C = 100{,}000$ Btu/therm (from Table 13-5)

$$F = \frac{473{,}500 \times 5{,}088}{0.80 \times 100{,}000} = 30{,}115 \text{ therms/year}$$

@ \$0.12/therm = \$3,613/year

Example No. 2

The plant in Example No. 1 is located in Milwaukee. The owner wants, as an alternate, to consider using the exhaust heat from his paint drying oven to preheat make-up air being drawn into the plant. On a 0° day it has been determined that the energy recovery system recovers 676,650 Btu/hr. Determine the fuel savings.

Solution:

$$F = \frac{Q \times N \times (ETD)_a}{E \times C \times (ETD)}$$

$Q = 676{,}500$ Btu/hr

$N = 5{,}088$ hours

$E = 0.80$

$C = 100{,}000$ Btu /therm

Degree days per month for Milwaukee from October 1 to May 1 are obtained from Table 13-1:

Month	*Degree Days*	*Days*
October	471	31
November	876	30
December	1252	31
January	1376	31
February	1193	28
March	1054	31
April	642	30
Total	6864	212

$t_{1_a} = 65°F - (6864/212) = 65°F - 32.38°F = 32.62°F$ for the heating season Oct. 1 to May 1.

$$(ETD) = (T_1 - t_1) = 200 - 0 = 200°F$$

$$F = \frac{676{,}650 \times 5{,}088 \times 167.38}{0.80 \times 100{,}000 \times 200}$$

$= 36{,}016$ therms/season

@ \$0.12/therm = \$4,322 per season

Table 13-2
EFFICIENCY OF FUEL UTILIZATION (14)

Type of Fuel-Burning Unit	Efficiency
Gas, designed unit	75-80%
Gas, conversion unit	60-80%
Oil, designed unit	65-80%
Oil, conversion unit	60-80%
Bituminous coal, stoker-fired	50-70%
Anthracite, stoker-fired	60-80%
Coke	60-80%

Table 13-3
PHYSICAL PROPERTIES OF FUEL OIL AT 60°F*

Grade No.	Sp gr	Lb per gal	Btu per lb	Net Btu per gal
6	1.0520	8.76	18,190	152,100
6	1.0366	8.63	18,290	149,400
6	1.0217	8.50	18,390	148,100
6	1.0071	8.39	18,490	146,900
6	0.9930	8.27	18,590	145,600
6, 5	0.9725	8.10	18,740	143,600
5	0.9465	7.89	18,930	140,900
4, 5	0.9218	7.68	19,110	138,300
4, 2	0.8984	7.49	19,270	135,800
2	0.8762	7.30	19,420	133,300
2	0.8550	7.12	19,560	130,900
1, 2	0.8348	6.96	19,680	128,500
1	0.8156	6.79	19,810	126,200

*140,000 Btu/gal is considered a good average.

Table 13-4
ANALYSES OF TYPICAL COALS (14)

Type	State	Heating Value-Btu per lb As Rec'd	Dry*
Anthracite	Colorado	14,099	14,490
Anthracite	Pennsylvania	13,298	13,682
Semianthracite	Colorado	13,468	14,155
Semianthracite	Pennsylvania	13,156	13,617
Semibituminous	Maryland	14,162	14,665
Semibituminous	Pennsylvania	14,279	14,800
Semibituminous	West Virginia	14,800	15,330
Bituminous	Alabama	14,141	14,602
Bituminous	Colorado	10,143	12,472
Bituminous	Illinois	11,741	12,746
Bituminous	Indiana	11,218	12,508
Bituminous	Iowa	10,019	11,678
Bituminous	Kansas	12,242	12,884
Bituminous	Kentucky	12,022	13,055
Bituminous	Missouri	10,622	12,619
Bituminous	Ohio	12,247	13,567
Bituminous	Oklahoma	11,695	11,945
Bituminous	Pennsylvania	13,699	14,178
Bituminous	Tennessee	13,048	13,788
Bituminous	Virginia	13,826	14,411
Bituminous	West Virginia	14,105	14,521
Sub-bituminous	Colorado	9,616	11,893
Sub-bituminous	Washington	9,331	11,312
Lignite	North Dakota	7,069	11,038
Lignite	Texas	7,348	11,084
Lignite	Wyoming	7,783	11,012

*14,000 Btu/lb is a good usable average.

Table 13-5
ANALYSES OF FUEL GASES (14)

Gas Type	Heating Value, Btu per cu feet	Heating Value, Btu per therm*
Natural:		
California	1,045	104,500
Kansas	992	99,200
Oklahoma	990	99,000
Pennsylvania .	1,210	121,000
Texas	975	97,500
Blast-furnace .	90	9,000
	92	9,200
Coke-oven ...	595	59,500
	536	53,600

*100,000 Btu/therm is considered a good average.

Example No. 3

Assume the 676,650 Btu/hr recovered by heating the make-up air in Example No. 2 can be returned to a 50% efficient paint drying oven which is heated electrically at \$0.03/kWhr and operates 16 hours/day, 5 days/week, 50 weeks/year. Find the dollar value of the energy saved.

Solution:

Q = 676,650 Btu/hr
N = 16 hrs/day × 5 days/wk × 50 wks/yr = 4000 hours
C = 3,413 Btu/kWhr
E = 0.50
T_{1_a} = 65°F − yearly average temperature
= 65°F − (7635/365) = 65°F − 20.91°F
= 44.09°F

$(ETD) = 200 - 0 = 200°F$
$(ETD)_a = 200 - 44.09 = 155.91°F$
$$F = \frac{676{,}650 \times 4{,}000 \times 155.91}{0.50 \times 3{,}413 \times 200}$$
$= 1{,}236{,}408$ kWhr/year
@ \$0.03/kWhr = \$37,092 per year

Example No. 4

A clinic in Chicago requires 20,000 scfm of make-up air. 15,000 scfm discharged at 70°F is available for preheating the make-up air. The clinic's heating system is 75% efficient and is fired with #2 distillate oil priced at \$0.42/gallon. The heating system operates 5,088 hours during the heating season. What will be saved by installing an energy recovery system which recovers 854,000 Btu/hr at 0°F ambient?

Solution:

$Q = 854{,}000$ Btu/hr
$N = 5{,}088$ hrs
$E = 0.75$
$C = 141{,}000$ Btu/gallon
t_{1_a} = From Table 13-1 for the Chicago area:

Month	*Degree Days*	*Days*
October	279	31
November	705	30
December	1051	31
January	1150	31
February	1000	28
March	868	31
April	489	30
Total	5542	212

$5542/212 = 26.14°F$
$65°F - 26.14°F = 38.86°F = t_{1_a}$ for the season
$(ETD) = 70° - 0 = 70°F$
$(ETD)_a = 70° - 38.86° = 31.14°F$
$$F = \frac{854{,}000 \times 5{,}088 \times 31.14}{0.75 \times 141{,}000 \times 70}$$
$= 18{,}279$ gals/year
@ \$0.42 gal = \$7,677 per year

Example No. 5

The flue gas from the boiler in Example No. 4 is 450°F. The clinic uses large volumes of hot water. In Chicago the average winter tap water temperature is 50°F. By installing an energy recovery device in the stack of the boiler, hot water can be heated during the heating season recovering an additional 168,000 Btu/hr. The hot water is used 16 hrs/day, 6 days/week. What will the additional fuel savings be by the installation of the energy recovery device?

Solution:

$Q = 168{,}000$ Btu/hr
$N = 16 \text{ hrs/day} \times 212 \text{ days} \times 6/7 = 2{,}907$ hours
$E = 0.75$
$C = 141{,}000$ Btu/gal

Since the heat is salvaged from the boiler flue gas which is independent of ambient temperature. $(ETD)_a$ and (ETD) are equal.

Therefore:

$$F = \frac{Q \times N}{E \times C}$$
$$= \frac{168{,}000 \times 2{,}907}{0.75 \times 141{,}000} = 4{,}618 \text{ gals/year}$$

@ \$0.42/gal = \$1,939 per year

When calculating the amount of fuel saved, caution should be exercised in obtaining the number of operating hours (N), as well as the selection of equipment efficiency (E). If salvaged heat is used for space heating only, yearly operating time can be as low as 600 hours for a one shift operation. Regarding efficiency, a well maintained unit heater system may have an efficiency approaching 80%, which means that for every 1 Btu saved by a heat recovery system, a savings of 1 Btu/.8 = 1.25 Btu of raw fuel can be saved. If, however, the system is old and not properly maintained and has an efficiency of 60%, the 1 Btu of recovered heat now becomes 1 Btu/.6 = 1.67 Btu of raw fuel saved.

The best return on an energy recovery system can be realized when the heat is obtained from a process. Most processes can operate on a year round basis and are independent of the heating season. If, for example, a process operates 24 hrs per day, 365 days per year, there is a total of 8,760 hours of potential recoverable energy.

D MAKING A SIMPLE CAPITAL INVESTMENT ANALYSIS

Once the engineer has determined the dollar value of fuel saved by the heat recovery system, he is ready to make an investment evaluation. As pointed out earlier, however, there is more to energy conservation than just dollars of fuel cost. Some other facts to be considered are:

1. By reducing the use of a lower cost fuel, it is often possible to eliminate the need to purchase

a more costly fuel. If so, the estimated savings should be based on the unit cost of the more expensive fuel.

2. If the recovered energy has allowed an increase in production, a portion, if not all of the profit increase can be presumed to have resulted from the investment in the heat recovery system. For example, if any energy recovery system reduces a customers scrap rate by $150 per day, then take credit for it.
3. Since fuel prices will probably continue to rise indefinitely, savings should increase by 6 to 8 per cent yearly or more.
4. If the installation of an energy recovery system reduces a possible loss of profit experienced during a fuel shortage, be sure to include that as a savings.
5. Some natural gas companies have contracts with their customers based on usage over the six summer months. In the winter, any volumes used above the contract amount is priced at a premium rate. Any gas saved beyond the contract base will save gas at a price equal to the sum of the contract price plus the premium. For example, a midwest gas company has a contract which charges a premium of 20¢/MCF for gas used beyond the contract base rate.

Certainly there are other factors that can and should be considered. These few have been suggested so that others might be thought of to be added to the list.

Two popular methods used to make a capital investment analysis are:

1. Simple pay back.
2. Simple rate of return.

In this section, these two methods will be discussed. Section XIV will discuss more complicated methods of making an energy recovery system investment analysis, which should be left to the accountants. It is essential, however, that all people involved in energy recovery systems have a knowledge of the simple and complicated methods for analyzing a project.

The following examples will illustrate the simple pay back and rate of return methods:

Example No. 6

An energy recovery system is to be installed in Milwaukee, Wisconsin, on a die casting operation where 13,000 scfm of 200°F air is being discharged to the atmosphere. On a 0° day, the system salvages 1,975,680 Btu/hr. The waste heat is to be used to heat make-up air. The plant's heating system, which uses natural gas priced at $0.12/therm, is 80% efficient. A sheet metal contractor has quoted an installed price of $16,000 for the system. Determine the simple return on investment for the system as well as the pay back time.

Solution:

a. Savings

$Q = 1{,}975{,}680$ Btu/hr

$N = 5{,}088$ hr

$E = 0.80$

$C = 100{,}000$ Btu/therm

$(ETD) = 200°F - 0° = 200°F$

$(ETD)_a = 200°F - 32.62°F = 167.38°F$

$$F = \frac{1{,}975{,}680 \times 5{,}088 \times 167.38}{0.8 \times 100{,}000 \times 200}$$

$= 105{,}159$ therms/year

@ $0.12/therm = $12,619./year

b. Investment Analysis

Capital Cost	$16,000.
Savings	$12,619. /year
Less property tax @ 1½% of capital cost	−240.
Less straight line depreciation @ 10% of Capital Cost	−1,600.
Net annual Savings before income taxes	$10,779.
Less income taxes @ 50% of savings	−5,389.
Net annual savings after income taxes	$5,390.

c. Return on Investment

$$\frac{\text{Net Annual Savings}}{\text{Capital Cost}} = \frac{5{,}390}{16{,}000} = 33.6\%$$

Pay Back =

$$\frac{\text{Capital Cost}}{\text{Net Annual Savings} + \text{Depreciation}} = \frac{16{,}000}{5{,}390 + 1{,}600} = 2.3 \text{ years}$$

Example No. 7

Determine the simple return on investment for the system in Example No. 6, except that #2 oil @ $0.35/gal is the fuel used in the boiler.

Solution:

a. Savings

$C = 141{,}000$ Btu/gal

$$F = \frac{1{,}975{,}680 \times 5{,}088 \times 167.38}{0.8 \times 141{,}000 \times 200}$$

= 74,581 gals/year @ \$0.35/gal
= \$26,103/year

b. Investment Analysis

Capital Cost	\$16,000.
Savings	\$26,103. /year
Less Property Tax @ 1½% of capital cost	−240.
Less Straight Line Depreciation @ 10% of Capital Cost	−1,600.
Net Annual Savings Before Income Taxes	\$24,263.
Less Income Taxes @ 50% of capital cost	−12,132.
Net Annual Savings After Income Taxes	\$12,131.

c. Return on Investment $= \frac{12,131}{16,000} = 75.8\%$

Pay Back $= \frac{16,000}{12,131 + 1,600} = 1.17$ years

Example No. 8

Two energy recovery units are to be installed in the exhaust stack of a wallboard manufacturing plant. The two units salvage 1,000,000 Btu/hr which is used to preheat the combustion air of the wallboard drier. The plant operates continuously, 24 hrs/day, 365 days/year. The dryer is 80% efficient and uses #6 oil at \$0.35/gal. Its heating value is 140,000 Btu/gal. The installed cost of the complete system is \$31,000. What is the return on investment and payback for the system?

Solution:

a. Savings

$Q = 1,000,000$ Btu/hr
$N = 24$ hrs/day $\times$ 365 days/yr
$= 8760$ hours
$E = 0.80$
$C = 140,000$ Btu/gal.

$$F = \frac{Q \times N}{E \times C}$$

$$= \frac{1,000,000 \times 8,760}{0.80 \times 140,000} = 78,214 \text{ gals/year}$$

@ \$0.35/gal = \$27,375/year

b. Investment Analysis

Capital Cost	\$31,000.
Savings	\$27,375. /year
Less Property Tax @ 1½% of capital cost	−465.
Less Straight Line Depreciation @ 6.6% of capital cost	−2,046.
Net Annual Savings Before Income Taxes	\$24,864.
Less Income Taxes @ 50% of Savings	−12,432.
Net Annual Savings After Income Taxes	\$12,432.

c. Return on Investment

$$\frac{\text{Net Annual Savings}}{\text{Capital Cost}} = \frac{\$12,432}{31,000} = 40.1\%$$

Pay Back =

$$\frac{\text{Capital Cost}}{\text{Net Annual Savings + Depreciation}} = \frac{31,000}{12,432 + 2,046} = 2.1 \text{ years}$$

E LIFE-CYCLE COSTING

All too often, first cost has preoccupied the minds of both the owner and designer, causing them to neglect giving proper consideration to system life and operating cost. A system that is inexpensive to buy may be expensive to operate and maintain.

With inflation, construction costs have escalated. The cost of money and energy continue to increase dramatically, but not always in the same proportion. These factors have created a more rational and factual approach to the real costs of a system, by analyzing both owning and operating costs over a fixed time period (life cycle costs).

Life cycle costing (LCC) is defined by the Federal Government to mean ". . . the total costs of owning, operating, and maintaining a building over its economic life, including its fuel and energy costs, determined on the basis of a systematic evaluation and comparison of alternative building systems."

An Executive Order directs federal agencies to consider in their building plans only those energy conservation improvements which are cost-effective based on a life-cycle cost approach, and further directs agencies to give the highest priority to the most cost-effective projects.

Life cycle costing is a generally accepted means, in both public and private arenas, of recognizing the sum total of all costs (and benefits) associated with a project during its estimated lifetime. During its more than 20 years of existence and application, LCC techniques have evolved from simple manual calculations to complex computerized operations requiring vast data bases.

1. Basic Procedures

Listed below are basic procedures for performing an LCC analysis:

- Identify the alternative approaches to achieve the objective.
- Establish a time frame for the analysis.
- Identify the cost parameters to be considered in the analysis.
- Convert costs and savings occurring at different times to a common time.
- Determine the cost-effectiveness of the alternatives.
- Analyze the results for sensitivity to the initial assumptions.

Section XIV—"Energy Recovery System Investment Analysis", contains all of the procedures needed to make a thorough LCC analysis. However, the basic elements are described as follows:

a. Annual Owning Cost

1. *Initial Costs* — The amortization period must be determined in which the initial costs are to be recovered and converted by use of a capital recovery factor (CRF) into an equivalent annual cost (see Table 13-6).

2. *Taxes*

   a. Property or real estate taxes.
   b. Personnel payroll taxes.
   c. Building management personal property taxes.
   d. Other building taxes.

3. *Insurance*

b. Annual Operating Cost

1. *Annual Energy Costs*

   a. Energy and fuel costs.
   b. Water charges.
   c. Sewer charges.

2. *Annual Maintenance Costs*

   a. Maintenance contracts.
   b. General housekeeping costs.
   c. Labor and material for replacing worn parts and filters.
   d. Chemicals and cleaning compounds.
   e. Costs of refrigerant, oil and grease.
   f. Cleaning & painting.

Table 13-6 CAPITAL RECOVERY FACTORS

	Rate of Return or Interest Rate, Percent									
Years	3½	4½	6	8	10	12	15	20	25	30
2	0.52640	0.53400	0.54544	0.56077	0.57619	0.59170	0.61512	0.65455	0.69444	0.73478
4	.27225	.27874	.28859	.30192	.31547	.32923	.35027	.38629	.42344	.46163
6	.18767	.19388	.20336	.21632	.22961	.24323	.26424	.30071	.33882	.37839
8	.14548	.15161	.16104	.17401	.18744	.20130	.22285	.26061	.30040	.34192
10	.12024	.12638	.13587	.14903	.16275	.17698	.19925	.23852	.28007	.32346
12	0.10348	0.10967	0.11928	0.13270	0.14676	0.16144	0.18448	0.22526	0.26845	0.31345
14	.09157	.09782	.10758	.12130	.13575	.15087	.17469	.21689	.26150	.30782
16	.08268	.08902	.09895	.11298	.12782	.14339	.16795	.21144	.25724	.30458
18	.07582	.08224	.09236	.10670	.12193	.13794	.16319	.20781	.25459	.30269
20	.07036	.07688	.08718	.10185	.11746	.13388	.15976	.20536	.25292	.30159
25	0.06067	0.06744	0.07823	0.09368	0.11017	0.12750	0.15470	0.20212	0.25095	0.30043
30	.05437	.06139	.07265	.08883	.10608	.12414	.15230	.20085	.25031	.30011
35	.05000	.05727	.06897	.08580	.10369	.12232	.15113	.20034	.25010	.30003
40	.04683	.05434	.06646	.08386	.10226	.12130	.15056	.20014	.25006	.30001

g. Testing.
h. Waste disposal.

3. *Operators* — The annual wages of building engineers and/or operators should not be included as part of maintenance, but entered as a separate cost item.

2. Initial System Costs

The first impact of an energy recovery system is the initial cost of the system. A careful evaluation of all cost variables entering into the system should be made if maximum economy is to be achieved. The designer has a great influence on these costs when he specifies the duct system material, system operating pressure, duct size and complexity, fan horsepowers, the type of energy recovery device, and determines the space requirements for ductwork and all apparatus. The SMACNA "HVAC Duct System Design" manual contains valuable information on these subjects.

The amortization period of useful life for energy recovery devices will vary widely with the type of application. The duct systems also could have a shorter life expectancy than HVAC duct systems which are usually amortized the same as the building.

In Table 13-6, data is given for capital recovery factors based on years of useful life and the rate of return or interest rate. The purpose of this table is to give a factor which, when multiplied by the initial cost of a system or component thereof, will result in an equivalent uniform annual cost for the period of years chosen. More detailed tables are found at the end of Section XIV.

3. Operating Costs

Since all energy recovery systems require two airstreams, attention should be directed to energy costs which are created by the systems. The important determining factor for fan size and horsepower, other than air quantity, is the system total pressure. Since fans normally operate continuously when a building is occupied or the process is underway, the energy demand of air fan distribution systems is one of the major contributors to the total system annual energy cost. Fan energy cost can be minimized by reducing duct velocities and using recovery devices with low static pressure losses; however, this has a direct bearing on the system first cost. Extra space might be required by the enlarged ductwork throughout the building and larger equipment rooms might also be required. It is extremely important for the designer to adequately investigate and calculate the impact of operating costs versus system first cost.

For example, it has been computed that a continuously-operating HVAC system costs 1 cent per cfm, per ¼ inch static pressure, based on 3 cents per kW cost of electrical energy. Therefore, a ¼ inch increase in static pressure for a 100,000 cfm system would add $1,000.00 to the cost of operation for one year at the 3 cents per kW rate. Increasing the system operating pressure may add to first costs by changing the class of duct construction required to contain a higher static pressure.

Because energy prices are widely expected to increase faster than general prices, provision for escalation of energy rates is to be included.

Labor and material costs, and other non-energy costs are assumed to increase in accordance with general price increases.

SECTION XIV
ENERGY RECOVERY SYSTEM INVESTMENT ANALYSIS

All of the text and tables in this section and some of the text in the preceeding section was taken from Chapter 3 of the National Bureau of Standards Handbook 121 entitled, "Waste Heat Management Guidebook" (with minor editing).

To evaluate the desirability of an investment, measures of costs are needed to compare with the benefits. Table 14-1 shows the type of costs which may arise in connection with energy recovery systems.

As may be seen, costs may begin before the energy recovery system is installed and extend throughout the period of continued plant operation. In most cases, the major cost item is likely to be the acquisition and installation of the heat exchanger, and should be relatively easy to estimate.

It is important that only those costs and benefits which are attributable to an investment be included in the analysis of that investment. For example, if a plant is required by mandate to add a pollution control apparatus, the decision to add an energy recovery system to the pollution control system should not be influenced by the costs of the pollution control system. As a further example, costs of equipment replacement or repair not necessitated by the addition of the energy recovery system should not be incorporated into the energy recovery evaluation, although it may be undertaken jointly for convenience.

A PARTIAL METHODS OF EVALUATION

The simplest procedures which are used by firms to try to evaluate alternative kinds and amounts of investments are visual inspection, payback period, and return on investment approaches which are termed "partial" here because they do not fully assess the economic desirability of alternatives. These partial methods may be contrasted with the more complete techniques, discussed later in the section which take into account factors such as timing of cash flows, risk, and taxation effects — factors which are required for full economic assessment of investments.

Despite their shortcomings, the partial techniques of analysis may serve a useful purpose. They can provide a first level measure of profitability which is, relatively speaking, quick, simple, and inexpensive to calculate. They may therefore be useful as initial screening devices for eliminating the more obviously uneconomical investments. These partial techniques (particularly the payback method) may also provide needed information concerning certain sensitive features of an investment. But where partial methods are used, the more comprehensive techniques may also be needed to verify the outcome of the evaluations, and to rank alternative projects as to their relative efficiency.

1. Payback Method

The *payback (also known as the payout or payoff) method* determines the number of years required for the invested capital to be offset by resulting benefits. The required number of years is termed the payback, recovery, or break-even period.

The measure is popularly calculated on a before-tax basis and without discounting, i.e., neglecting the opportunity costs of capital. Investment costs are usually defined as first costs, often neglecting salvage value. Benefits are usually defined as the resulting net change in incoming cash flow, or, in the case of a cost-reducing investment like energy recovery, as the reduction in net outgoing cash flow.

The payback period is usually calculated as follows:

Equation 14-1

$$\text{Payback Period (PP)} = \frac{\text{First Cost}}{\text{Yearly Benefits} - \text{Yearly Costs}}$$

Example No. 1

The payback period for a furnace recuperator which costs $10,000 to purchase and install, $300/yr on average to operate and maintain, and which is expected to save by preheating combustion air an average of 20,000 therms of gas per year at $0.07/therm (i.e., $1400/yr).

Solution:

$$PP = \frac{\$10{,}000}{\$1400 - \$300} = 9.1 \text{ yr.}$$

Table 14-1 POTENTIAL COSTS TO CONSIDER IN INVESTING IN ENERGY RECOVERY SYSTEMS (21)

Type of costs	Examples of costs
1. Pre-engineering and planning costs	Engineering consultant's fee; in house manpower and materials to determine type, size, and location of heat exchanger.
2. Acquisition costs of heat recovery equipment	Purchase and installation costs of a recuperator.
3. Acquisition costs of necessary additions to existing equipment	Purchase and installation costs of new controls, burners, stack dampers, and fans to protect the furnace and recuperator from higher temperatures entering the furnace due to preheating of combustion air.
4. Replacement costs	Cost of replacing the inner shell of the recuperator in *N* years, net of the salvage value of the existing shell.
5. Costs of modification and repair of existing equipment	Cost of repairing furnace doors to overcome greater heat loss resulting from increased pressure due to preheating of combustion air.
6. Space costs	Cost of useful floor space occupied by waste heat steam generator; cost of useful overhead space occupied by evaporator.
7. Costs of production downtime during installation	Loss of output for a week, net of the associated savings in operating costs.
8. Costs of adjustments (debugging)	Lower production; labor costs of debugging.
9. Maintenance costs of new equipment	Costs of servicing the heat exchanger and filters.
10. Property and/or equipment taxes of heat recovery equipment	Additional property tax incurred on capitalized value of recuperator.
11. Change in insurance or hazards costs	Higher insurance rates due to greater fire risks; increased cost of accidents due to more hot spots within a tighter space.

* In addition, attention should be given to the length of intended use, expected lives of related equipment, and the flexibility of alternative equipment of future modification and expansion.

a. Disadvantages

The disadvantages of the payback method which recommend against its use as a sole criterion for investment decisions may be summarized as follows:

(1) The method does not give consideration to cash flows beyond the payback period, and thus does not measure the efficiency of an investment over its entire life.

Consider, for example, the two alternative investments A and B, presented in Table 14-2. Using the undiscounted payback method, a firm would prefer Investment A, which has a payback period of 1.7 yr, to Investment B, which has a payback of 2.2 yr. Yet, depending upon the true opportunity cost of capital (i.e., the discount rate), Investment B, which continues to yield benefits beyond Investment A, may be a more profitable choice. (For example, with an opportunity cost of 10 percent, Investment A would yield $20,832 in total benefits, and Investment B, $22,383 in total benefits in present value terms).

(2) The neglect of the opportunity cost of capital, that is, failing to discount costs occurring at different times to a common base for comparison, results in the use of inaccurate measures of benefits and costs to calculate the payback period, and, hence, determination of an incorrect payback period. This problem is illustrated by the example of two alternative investments

Table 14-2 AN ILLUSTRATION THAT PAYBACK ANALYSIS DOES NOT TAKE INTO ACCOUNT CASH FLOWS BEYOND THE PAYBACK PERIOD (21)

Investment	First Cost	Yearly benefits			Payback period	Total present value benefits[a]
		Year 1	Year 2	Year 3		
Investment A	$20,000	$12,000	$12,000	$0	1.7 years	$20,832
Investment B	20,000	9,000	9,000	9,000	2.2 years	22,383

[a]Calculated for a discount rate of 10 percent, compounded annually.

shown in Table 14-3. Payback analysis using undiscounted values would result in indifference between Investments C and D. They both have a payback of 2 yr, and yield total benefits, undiscounted, of $25,000. But because Investment D yields more benefits toward the beginning than Investment C, and thereby allows the investor to realize a larger return on earnings, Investment D would be the preferred choice. In present value terms, with an opportunity cost of 10 percent, Investment C would yield total benefits of $20,697, and Investment D, $21,524.

In short, the payback method gives attention to only one attribute of an investment, i.e., the number of years to recover costs, and, as often calculated, does not even provide an accurate measure of this. It is a measure which many firms appear to overemphasize, tending toward shorter and shorter payback requirements. Firms' preference for very short payback to enable them to reinvest in other investment opportunities may in fact lead to a succession of less efficient, short-lived projects.

b. Advantages

Despite its limitations, the payback period has advantages in that it may provide useful information for evaluating an investment. There are several situations in which the payback method might be particularly appropriate:

1. A rapid payback may be a prime criterion for judging an investment when financial resources are available to the investor for only a short period of time.
2. The speculative investor who has a very limited time horizon will usually desire rapid recovery of the initial investment.
3. Where the expected life of the assets is highly uncertain, determination of the break-even life, i.e., payback period, is helpful in assessing the likelihood of achieving a successful investment. (This use of the payback method is discussed later in this section.)

Table 14-3 AN ILLUSTRATION THAT THE UNDISCOUNTED PAYBACK METHOD CAN RESULT IN INACCURATE MEASURES (21)

Investment	First Cost	Yearly benefits			Payback period	Total present value benefits[a]
		Year 1	Year 2	Year 3		
Investment C	$20,000	$5,000	$15,000	$5,000	2 years	$20,697
Investment D	20,000	15,000	5,000	5,000	2 years	21,524

[a]Calculated for a discount rate of 10 percent, compounded annually.

c. More Accurate Method

The shortcomings that result from failure to discount costs and the omission of important cost items can be overcome simply by using a more accurate calculation of payback. Essentially what is desired is to find the number of years, R, for which the value of the following expression is equal to zero:

$$C = \sum_{j=1}^{R} \frac{B_j - P_j}{(1+i)^j} \qquad \text{Equation 14-2}$$

where

C = Initial investment cost,

B_j = Benefits in year j,

P_j = Costs in year j,

R = Break-even number of years, and

i = Discount rate.

Where yearly net benefits are uneven, an iterative process can be used to determine the solution. If, on the other hand, yearly net benefits are expected to be about uniform, the following equation can be used to facilitate the calculation:

$$R = \frac{\log\left(1 + \frac{iC}{M}\right)}{\log(1+i)} \qquad \text{Equation 14-3}$$

where

R = Break-even number years,
M = Yearly net benefits,
C = Initial investment cost, and
i = Discount rate.

2. Return on Investment Method

The *return on investment (ROI) or return on assets method* calculates average annual benefits, net of yearly costs such as depreciation, as a percentage of the original book value of the investment.

The ROI is calculated as follows:

Equation 14-4

$$\text{Return on Investment (ROI)} = \frac{\text{Average Annual Net Benefits}}{\text{Original Book Value}} \times 100.$$

Example No. 2

Find the ROI for an investment in an energy recovery system:

Original Book Value = \$15,000
Expected Life = 10 yr
Annual Depreciation,
using a straight-line method $= \frac{\$15{,}000}{10} = \$1{,}500$

Yearly Operation, Maintenance
and Repair Cost = \$200
Expected Annual Fuel Oil Savings = \$5000

Solution:

$$\text{ROI} = \frac{\$5000 - (\$1500 + \$200)}{\$15{,}000} \times 100$$
$$= 0.22 \times 100 = 22 \text{ percent}$$

a. Disadvantages

The return on investment method is subject to the following principal disadvantages, and, therefore, is not recommended as a sole criterion for investment decisions:

1. Like the payback method, this method does not take into consideration the timing of cash flows, and thereby may incorrectly state the economic efficiency of projects.
2. The calculation is based on an accounting concept, original book value, which is subject to the peculiarities of the firm's accounting practices, and which generally does not include all costs. The method, therefore, results in only a rough approximation of an investment's value.

b. Advantages

The advantages of the return on investment method are that it is simple to compute and a familiar concept in the business community.

B COMPREHENSIVE METHODS FOR EVALUATING INVESTMENT ALTERNATIVES

There are additional methods of financial analysis which avoid the problems of the partial methods by taking into account total costs and benefits over the life of the investment and the time of cash flows by discounting. Methods of this type are the *present value of net benefits method*, the *annual net benefits method, the benefit/cost ratio method,* and the *in-*

ternal rate of return method. The *discounting of costs* is an element common to all of them.

1. Discounting of Costs

Investment in energy recovery systems, like many capital investments, will generally require a number of expenditures spread over a period of time and will result in cost savings (or revenue receipts) also spread over time. To evaluate correctly the profitability of such investments, it is necessary to convert the various expenditures and receipts to a common basis, because dollars spent or received at different times are not of equal value. If, for instance, the firm's earning opportunity is 8 percent per year compounded annually, $1.00 received today would grow to $1.47 in 5 years and would about double in 9 years. Deferral of the receipt of $1.00 for 5 years would mean that the earnings on it over the interim 5 years would not be realized, and it would be worth less than $1.00 received today. It would, in fact, be equivalent to receiving only $0.68 today. Therefore, other things equal, a firm will prefer early income and deferred expenditures. A real opportunity cost of capital exists even when there is no inflation or deflation and even when equity (nonborrowed) funds are used, as long as alternative productive investments are possible.

Cash flows occuring at different times can be converted to a common basis, i.e., discounted, by means of discounting equations (also known as interest equations). The six basic discounting equations are shown in Table 14-4, along with an abbreviated guide for their use.

The discounting of costs can be greatly simplified by the use of tables of discount factors, such as those in Tables 14-9 through 14-15. The discount factors are calculated from the discount equations for various time periods and opportunity costs (expressed as a rate of discount or interest). In the sample tables provided, factors are calculated for 50 to 100 interest periods (which could be years, months, or any other time interval desired), and for selected discount rates ranging from 6 to 25 percent per period. Factor tables covering other time periods and interest rates are available in most engineering economics textbooks.

Table 14-4 DISCOUNTING EQUATIONS (21)

Standard nomenclature	Use when	Standard notation	Algebraic form
Single compound amount equation	Given P; to find F	(SCA, i%, N)	$F = P(1+i)^N$
Single present worth equation	Given F; to find P	(SPW, i%, N)	$P = F\dfrac{1}{(1+i)^N}$
Uniform compound amount equation	Given A; to find F	(UCA, i%, N)	$F = A\dfrac{(1+i)^N - 1}{i}$
Uniform sinking fund equation	Given F; to find A	(USF, i%, N)	$A = F\dfrac{i}{(1+i)^N - 1}$
Uniform capital recovery equation	Given P; to find A	(UCR, i%, N)	$A = P\dfrac{i(1+i)^N}{(1+i)^N - 1}$
Uniform present worth equation	Given A; to find P	(UPW, i%, N)	$P = A\dfrac{(1+i)^N - 1}{i(1+i)^N}$

Where:

P = a present sum of money.
F = a future sum of money, equivalent to P at the end of N periods of time at an interest of i.
i = a discount or interest rate to reflect the opportunity cost of capital.
N = number of interest periods.
A = an end-of-period payment (or receipt) in a uniform series of payments (or receipts) over N periods at i interest rate, usually annually.

To find the comparable value of a given sum of money if it were received (or disbursed) at a different time, the appropriate equation from Table 14-4 may be used, or the corresponding discount factor may be multiplied by the given sum.

Example No. 3

If a firm's opportunity earning potential is 10 percent annually, what would be equivalent value today of \$500 received 5 years from now?

Solution:

Present value = \$500 (SPW), i = 10%, N = 5 years,

$$\$500 \frac{1}{(1 + 0.10)^5} = \$500\ (0.6209) = \$310 \text{ Equivalent}$$

value.

Example No. 4

Alternatively, receiving \$500, 5 years from now, would be equivalent to receiving equal payments of what amount in each of the 5 years.

Solution:

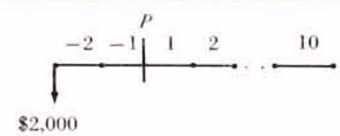
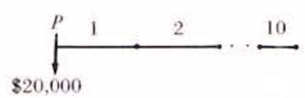
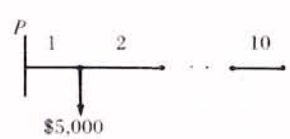
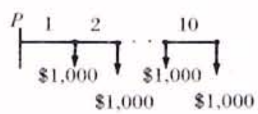
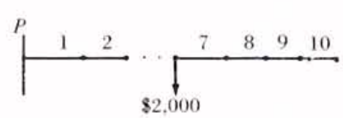
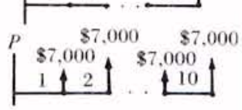
Annual Payment Equivalent = \$500 (USF) i = 10%, N = 5 years

$$\$500 \frac{0.10}{(1 + 0.10)^5 - 1} = \$500\ (0.1638) = \$82$$

Annual payment equivalent.

Table 14-5 illustrates the conversion to a present equivalent and an annual equivalent of various costs and benefits representative of an investment in energy recovery. The examples assume that the investing firm has an opportunity cost of 15 percent and a 10 year time horizon. The first column gives the kind of cost or benefit incurred. The second column uses a cash flow diagram to describe the pattern of cash outflow or inflow associated with the cost or benefit. The horizontal line of the cash-flow diagram is a time scale, where P indicates the present, the progression of time moves from left to right, and the numbers

Table 14-5 ILLUSTRATIVE DISCOUNTING OF REPRESENTATIVE COSTS AND BENEFITS (21)

Kind of cost or benefit	Cash flow pattern	Present equivalent	Annual equivalent
Planning for waste heat recovery		P=\$2,000 (SCA, 15%, 2 yr) =\$2,000 (1.322) =\$2,644	A=\$2,000 (SCA, 15%, 2 years) (UCR, 15%, 10 years) =\$2,000 (1.322) (0.1993) =\$527
Purchase and installation of recuperator		P=\$20,000 (1) =\$20,000	A=\$20,000 (UCR, 15%, 10 years) =\$20,000 (0.1993) =\$3,986
Cost of production downtime		P=\$5,000 (SPW, 15%, 1 yr) =\$5,000 (0.8696) =\$4,348	A=\$5,000 (SPW, 15%, 1 year) (UCR, 15%, 10 years) =\$5,000 (0.8696) (0.1993) =\$867
Net increase in annual operating and maintenance cost due to the recuperator		P=\$1,000 (UPW, 15%, 10 yr) =\$1,000 (5.019) =\$5,019	A=\$1,000 (1)
Replacement of parts		P=\$2,000 (SPW, 15%, 7 yr) =\$2,000 (0.3759) =\$752	P=\$2,000 (SPW, 15%, 7 years) (UCR, 15%, 10 years) =\$2,000 (0.3759)(0.1993) =\$150
Salvage value of heat recovery equipment, end of 10 years		P=\$7,000 (SPW, 15%, 10 yr) =\$7,000 (0.2472) =\$1,730	A=\$7,000 (USF, 15%, 10 years) =\$7,000 (0.0493) =\$345
Annual fuel savings, (assuming no change in future fuel prices) [a]		P=\$7,000 (UPW, 15%, 10 yr) =\$7,000 (5.019) =\$35,133	A=\$7,000 (1)

[a] Evaluation of costs and benefits when price changes are expected is discussed in the following major section.

between the points represent time intervals (in this case, years). The downward arrows represent expenditures (cash outflows), and the upward arrows represent receipts (cash inflows), as viewed from the standpoint of the firm.

Once all costs or benefits are discounted to the present, they may be combined in order to find the total. Thus, in this simplified example, total costs of the investment amount to $31,033 in present value terms, and total present value benefits amounts to $35,133 over the 10 years (found by aggregating all the costs and all the benefits in the Present Equivalent Column of Table 14-5). This is equivalent to incurring costs of $6185 annually and realizing benefits of $7000 annually for an annual benefit of $815 in each of the 10 years (found by aggregating the costs and the benefits in the Annual Equivalent Column).

2. Net Present Value (Net Benefits) Method

This method calculates the difference between the present value of the benefits and the costs resulting from an investment. The difference between benefits and costs is the net present value of the investment. A positive net present value means that the finance position of the investor will be improved by undertaking the investment; a negative net present value means that the investment will result in a financial loss.

To use this method in the evaluation of energy recovery investments the benefits would be defined as positive cash flows (as, for example, would result from sales of surplus, recovered waste heat), and/or reductions in cash outflows (as would result from substitution of recovered waste heat for newly-generated heat).

The equation for calculating the net present value, or net benefits, is as follows:

Equation 14-5

$$NPV = \sum_{j+1}^{n} \frac{(S_j+R_j) - (I_j - V_j + M_j)}{(1+i)^j}$$

where

NPV = Net present value benefits,
n = Number of time intervals over which the investment is analyzed,
S_j = Energy cost savings in year j,
R_j = Revenue from sale of excess energy received in year j,
I_j = Investment costs in year j,
V_j = Salvage value in year j,
M_j = Maintenance and repair costs in year j, and
$\frac{i}{(1+i)^j}$ = Single present value discount formula.

The acceptance criteria of a project, as evaluated with the net present value method, are that (1) only those investments having positive net benefits will be accepted (unless the project is mandatory), and (2) when selecting among mutually exclusive investments, the one with the highest positive net benefits will be chosen (or the one with the lowest negative net benefits if none of the alternatives has positive net benefits and the project is mandatory.)

a. Different Approaches

In using the net present value method to compare alternative investments, it is important to evaluate the costs and benefits of each alternative over an equal number of years. This may be done in any of the following ways, depending upon the nature of the investment.

(1) The costs and benefits can be measured over a time period that is a common multiple of the economic lives of the alternatives. For example, to compare heat exchanger A with a life of 5 years against heat exchanger B with a life of 10 years, alternative A could be evaluated on the basis of one replacement, and alternative B, on the basis of no replacements, such that benefits and costs of both systems would be computed for 10 years.

(2) Alternatively, benefits and costs alternative can be calculated in annual cost terms, based on 5 years for A and 10 years for B, and the annual benefits and costs can then be used to calculate the present value of benefits and costs for the desired number of years of service (for example, 12 years of service for both systems). This avoids the need to find a common multiple of system life.

(3) If either system will be used for only a limited period of time, which is less than a common multiple of the economic lives of the alternatives, the estimated cash flows associated with each system over the period of analysis can be discounted to present value, making sure to take into account the expected remaining value of each system at the time of terminated use. For example, the problem might be to choose between heat exchanger A, with its economic life of 5 years, and heat exchanger B, with its economic life of 10 years, where intended use of

either would be only 7 years due to expected closure of the plant at the beginning of the eighth year. System A would require one replacement because its expected life is only 5 years and it is needed 7 years. The remaining value of the replacement after 2 years of use would be discounted to present value and deducted from present value cost of the system. Similarly, an estimate would be required of the value of the remaining 3 years of life of system B. If removal costs are prohibitive, or if there is no good resale market for the equipment, the remaining value of both systems at the end of 7 years should be evaluated at zero dollars. This holds true even though the equipment could provide additional years of service if the existing operation were continued.

(4) An investment can also be evaluated on the basis of a perpetual, or indefinite period of use by "capitalizing" renewal costs and expected benefits. Present value benefits of an investment in perpetuity are calculated by dividing the expected annual benefits by the discount rate. Present value costs in perpetuity are calculated by converting all costs other than first costs to an annual equivalent, dividing by the discount rate, and adding the amount to the first cost. Thus, if first costs were \$5000; operation costs \$1000 yearly; and renewal costs \$4000 every 5 years, the present value capitalized cost of the above heat exchanger A in perpetual service would be equal to \$5000 + (\$1000 ÷ i) + (\$4000 (USF, i, 5 yr) ÷ i).

The choice among these four approaches to measuring present value is often not critical to the outcome; the particular nature of the investment will generally determine which approach is used. For example, in the case of a short-lived investment with unrecoverable salvage, the third approach explained above would be preferred.

b. Comparing Alternatives

The use of the net present value method to compare alternative investments in energy recovery equipment is illustrated in Table 14-6. It is assumed that the investment choice is between Plan X, the addition of a large energy recovery device designed to preheat combustion air to a high temperature, and Plan Y, the addition of a smaller device capable of a lesser amount of preheating but sized to avoid the replacement of fans, etc., on the existing equipment. It is further assumed that Plan Y has longer expected economic life due to lower temperatures.

Table 14-6 ILLUSTRATION OF NET PRESENT VALUE METHOD (21)

Investment alternatives	First cost	Expected economic life	Annual fuel savings	Net present value[a]
Plan X	\$20,000	6 years	\$7,000	\$12,847
Plan Y	12,000	10 years	5,000	16,256

[a]Based on a discount rate of 10 percent, an investment horizon of 8 years, and the assumption that the full value of any remaining life after 8 years can be salvaged.

Plan Y is found to provide \$3409 more in net benefits than Plan X. The additional fuel savings from the larger recovery device is, therefore, not sufficient to justify its additional costs.

In the case where alternative investments are expected to provide the same level of benefits, the net present value method becomes equivalent to a net present cost method, often referred to as *life-cycle costing,* or cost-effectiveness analysis. In this case, the most efficient alternative may be identified as the one with the least present value of costs alone. This approach is also often used when the benefit levels cannot be quantified.

c. Disadvantages

A feature of the net present value method which may be a disadvantage in some applications is that, in focusing only on net benefits, it does not distinguish between a project involving relatively large benefits and costs, and one involving much smaller benefits and costs, as long as the two projects result in equal net benefits. Thus a project requiring \$10,000 in present value costs, and resulting in \$11,000 in present value benefits (i.e., \$1000 in net benefits) will yield the same net benefits as a project costing \$1000 and resulting in \$2000 in total benefits (again \$1000 in net benefits). A way to avoid this problem is to compute benefit-to-cost ratios for further evaluation of the projects. Choices will be most efficient if independent projects are chosen in the order of their benefit-cost ratios, starting with the highest and working down until the budget is exhausted. (Later in this section under the heading of "Application of Evaluation Methods to Different Kinds of Decisions", there is a discussion of the appropriateness of each evaluation method to different decisions and suggests ways to avoid potential problems.)

Another possible disadvantage of the net present value method is that the results are quite sensitive to the discount rate, and failure to select the appropriate rate may alter or even reverse the efficiency ranking of the alternatives. For example, with too low a rate an alternative with benefits spread far into the future may unjustifiably appear more profitable than an alternative whose benefits are more quickly realized but of lower amount in undiscounted terms. Since changing the discount rate can change the outcome of the evaluation, the rate used should be considered carefully.

As was explained earlier, the discount rate which a firm should use to discount cash flows of an investment is the firm's opportunity cost of capital, expressed as an interest rate; i.e., the rate of return which will be foregone by using the funds (resources) for the investment under consideration instead of the next best investment opportunity available to it.

If the firm is uncertain as to the appropriate discount rate to use, it may wish to compute the net benefits of an investment based on several alternative discount rates to test for sensitivity of the outcome to the choice of rates.

d. Advantage

The net present value method has the advantage of measuring the net effect of an investment over its life, taking into account the opportunity cost of capital. The method is particularly useful for determining the efficient scale or size of an investment project.

3. Net Annual Value (Net Annual Benefits) Method

This method takes essentially the same form as the net present value method. The difference is that all costs and benefits of the net annual benefits method are converted to a uniform annual basis instead of to present value. The equation to be used is:

Equation 14-6

$$A = \sum_{j=1}^{n} \frac{(S_j + R_j) - (I_j - V_j + M_j)}{(1 + i)^j}$$
$$\frac{i(1 + i)^n}{(1 + i)^n - 1}$$

where

A = Annual value of net benefits,

n = Number of time intervals over which the invesment is analyzed,

S_j = Energy cost savings in year j,

R_j = Revenue received in year j,

I_j = Investment costs in year j,

V_j = Salvage value in year j,

M_j = Maintenance and repair costs in year j,

$\frac{1}{(l + i)_j}$ = Single present value discount formula,

and

$\frac{i(1 + i)^n}{(1 + i)^n - 1}$ = Uniform compound amount formula.

a. Different Approaches

If alternative investments have different life expectancies, either of two approaches may be taken to compare the alternatives.

(1) It may be assumed that whichever alternative is chosen will be needed for an indefinite period of time and, hence, will be renewed as needed. In this case, the annual cost of each system may be simply calculated for its expected economic life, regardless of the fact that the lives of the alternatives may be different.

(2) If the use of the investment alternatives is required for only a limited time, it is necessary to calculate costs based on the planned investment period, estimating the salvage value of each alternative at the planned time of terminated use. (This is similar to the third approach described under Net Present Value.)

With the cash flows of the preceding example (described in Table 14-6) and the assumption that either plan will be used for an indefinite period of time, the net annual value of Plan X is $2408, and of Plan Y is $3047. As we would expect, this method shows Plan Y to be the most efficient choice; as did the present value example.

b. Disadvantage

This method, like the most present value method, has the disadvantage of failing to distinguish between projects of unequal magnitudes which yield equal net benefits. However, analysis of benefit-cost ratios can be used to overcome this problem.

c. Advantage

A possible advantage of the net annual value method, as compared with the net present value method, is that the concept of an equivalent annual amount may be easier to understand than the concept of a present equivalent of all cash flows over the period of analysis.

4. Benefit/Cost Ratio Method

The benefit/cost ratio method expresses benefits as a proportion of costs, where benefits and costs are discounted to either a present value or an annual value equivalent. The equation (with benefits and costs discounted to present value) to be used is:

Equation 14-7

$$B/C = \sum_{j=1}^{n} \left[\frac{S_j + R_j}{(1 + i)^j}\right] \Big/ \sum_{j=1}^{n} \left[\frac{I_j - V_j + M_j}{(1 + i)^j}\right]$$

where

B/C = Benefit / cost ratio,

n = Number of time intervals over which the investment is analyzed,

S_j = Energy cost savings in year j,

R_j = Revenue received in year j,

I_j = Investment costs in year j,

V_j = Salvage value in year j, and

M_j = Maintenance and repair costs in year j.

While the net present value and the net annual value methods require that discounted benefits minus discounted costs be positive in order for an investment to be worthwhile, the benefit/cost ratio method requires that the ratio of disounted benefits to costs be greater than 1.

a. Disadvantages

A disadvantage of the benefit/cost ratio method is that the ratio is influenced by the decision as to whether an item is classified as a cost or as a disbenefit, i.e., whether it appears in the numerator or denominator of the ratio. For many cost or benefit items, this is simply an arbitrary decision, but one which can lead to confusion as to the real efficiency of a project.

Another problem with the benefit/cost ratio method is that it is subject to be misapplied in determining the efficient scale of a given project. It pays to expand a project up to the point that the ratio for the last increment of the investment is equal to 1.0, assuming no alternative investment is available with a higher ratio. Because the benefit/cost ratio for the overall investment declines as the investment is expanded towards the most efficient level, a smaller less efficient size of a project may have a higher ratio for the total investment than a larger more efficiently sized project. This problem can be avoided by applying the benefit/cost ratio method to evaluate the efficiency of increments of an investment, rather than the total investment.

5. Internal Rate of Return Method

This method (not to be confused with the ROI method evaluated earlier) calculates the rate of return an investment is expected to yield. This may be contrasted with the net present value, the net annual value, and the benefit/cost ratio methods which calculate the net dollar value of the investment based on a predetermined required rate of return. The internal rate of return method expresses each investment alternative in terms of a rate of return (a compound interest rate). The expected rate of return is the interest rate for which total discounted benefits become just equal to total discounted costs, i.e., net present benefits or net annual benefits are equal to zero, or for which the benefit/cost ratio equals one. The criterion for selection among alternatives is to choose the investment with the highest rate of return.

The rate of return is usually calculated by a process of trial and error, whereby the net cash flow is computed for various discount rates until its value is reduced to zero.

Example No. 5

Calculate the internal rate of return for a heat exchanger which will cost $10,000 to install, will last 10 years, and will result in fuel savings of $3000 each year. Find which "i" will equate the following: $10,000= $3000 (UPW, i = ?, 10 yr).

Solution:

To do this, the net present value (NPV) for various i values (selected by visual inspection) is calculated:

$$\begin{aligned} NPV\ 25\% &= (\$3000)\ (3.571) - \$10{,}000 \\ &= \$10{,}713 - \$10{,}000 \\ &= \$713 \\ NPV\ 30\% &= (\$3000)\ (3.092) - \$10{,}000 \\ &= \$9276 - \$10{,}000 \\ &= -\$724 \end{aligned}$$

For i = 25 percent, the net present value is positive; for i = 30 percent, the net present value is negative. Thus, for some discount rate between 25 and 30 percent, present value benefits are equated to present value costs. To find the rate more exactly, without the benefit of a complete set of discount tables, one may interpolate between the two rates as follows:

$$\begin{aligned} i &= 0.25 + 0.05\ \frac{\$713}{\$713 - (-\$724)} \\ &= 0.275, \text{ or } 27.5 \text{ percent} \end{aligned}$$

To decide whether or not to undertake this investment, it would be necessary for the firm to compare the expected rate of return of 27.5 percent with its minimum attractive rate of return.

a. Advantage

Use of this method has the advantage of generally resulting in conclusions consistent with the three other comprehensive methods discussed in this section.

b. Disadvantages

There are several possible disadvantages which might arise. For one thing, under certain circumstances there may be either no rate of return solution or multiple solutions. Secondly, confusion may arise when this method is used to choose among mutually exclusive alternatives. For example, if the compared alternatives are different sizes of the same project (e.g., different capacity heat exchangers, the rate of return on the larger scale project may be lower than on the smaller scale project, causing the larger to appear less efficient than the smaller. However, additional investment in the larger project may none-theless yield a positive rate of return in excess of the minimum attractive rate of return. This problem, which is comparable to the problem which was described for the benefit/cost method, can be avoided by analyzing incremental changes in an investment. A third problem is that the rate of return may be somewhat more cumbersome to calculate than the other methods.

C SPECIAL FACTORS TO CONSIDER IN INVESTMENT ANALYSIS

1. Treatment of Income Taxes

Income taxes can have a profound impact on optimum investment decisions. By changing the effective values to the firm of the revenues and costs associated with an investment, taxes can reverse the relative profitability of alternative projects as evaluated apart from taxes, and they can alter the optimal size of investments. Taxes are, therefore, an important element in the economic evaluation of investment in energy recovery systems.

If, for example, an investment results in increased taxable revenue, the effective value of the additional revenue to the firm is reduced by taxation. Over the life of an investment, the effective present value of after-tax revenue (R) can be calculated as follows:

Equation 14-8

$$\overline{R} = \sum_{j=1}^{n} \frac{R_j (1 - t_j)}{(1 + i)^j},$$

where

n = the number of time intervals over which the investment is analyzed,
R_j = the taxable revenue resulting from the investment in year j, and
t_j = the firm's tax rate at the margin in the year j,
i = the discount rate.

Similarly, if an investment in benefits in the form of cost savings, the after-tax value of the benefits will be lessened by the loss of any related income tax deductions; i.e., the cost savings are generally not tax free. The present value of after-tax cost-savings (C) from an investment can be calculated as follows:

Equation 14-9

$$\overline{C} = \sum_{j=1}^{n} \frac{C_j (1 - a_j t_j)}{(1 + i)^j},$$

where

C_j = the cost savings from the investment in year j,
a_j = that percentage of the cost savings which were deductible from taxable revenue, and
t_j = the firm's tax rate at the margin in year j.

On the other hand, investment costs are also reduced somewhat by the value of corresponding tax deductions. Effective after-tax costs in present value terms can be calculated from equation 14-9 by redefining C, C_j, and a_j in terms of "costs" rather than "cost savings."

An economic analysis of investment in energy recovery systems might involve each of these three tax effects. For instance, system equipment costs may give rise to tax deductions in terms of depreciation allowances and/or investment tax credits; savings in fuel costs will reduce deductions from taxable income of current operating costs; and sale of excess heat or energy to other firms will result in additional taxes on the revenue received.

The following example compares the before-tax and after-tax annual net benefits from an investment in a recovery device, given the assumptions below.

Example No. 6

Investing Firm's Tax Rate, at the Margin: 40%
Opportunity Cost of Capital: 10%, after taxes
Source of Capital: Equity Funds
Method of Equip. Depreciation: Straight-line Method
First Cost of Recovery Device: $25,000
Annual Maintenance Cost: $500
Expected Salvage Value in 10 yr: $0
Expected Annual Fuel Savings: $5000

Solution:

a. *Benefits*

Annual Reductions in Fuel Costs, Before Taxes	= $5000
(minus)	
Annual Value of Tax Deductions Lost = 0.40 ($5000)	= 2000
(equals)	
Annual Reductions in Fuel Costs, After Taxes	= $3000

b. *Costs:*

Annual Capital Cost of Equipment = $25,000 (UCR; 10%; 10 yr) = $25,000 (0.1628)	= $4070
(plus)	
Annual Maintenance Cost	= 500
(equals)	
Total Annual Increases in Equipment and Maintenance Costs, Before Taxes	= $4570
(minus)	
Annual Value of Tax Deductions of Equipment and Maintenance Cost = 0.40 ($25,000/10 + $500)	= $1200
(equals)	
Annual Increase in Equipment and Maintenance Costs, After Taxes	= $3370

c. *Net Benefits: Before and After Taxes:*

Net Benefits of Investment, Before Taxes = ($5000 − $4570)	= $ 430
Net Benefits of Investment, After Taxes = ($3000 − $3370)	= $−370

In this case, before-tax evaluation of the investment in the recovery device indicates a positive annual net cost savings of $430. But the after-tax evaluation indicates a negative cost savings of $370, which means the investment would not be profitable.

Even though depreciation is not itself a cash flow, it affects the firm's cash flow through the deductions from taxable income. By its choice of depreciation methods (e.g., the declining balance method as opposed to the straight-line method of depreciation), an investing firm can affect the timing of the tax deduction and thereby influence the effective amount of income tax payments. The choice of a depreciation method which yields higher depreciation allocations in the early years will increase the present value of depreciation deductions, thereby decreasing the present value of income tax payments and raising the profitability of the investment to the firm.

2. Inflation

For simplicity, changes in price were not considered in the preceding illustrations of methods of evaluation and treatment of taxes. However, price changes may be very important in an investment analysis.

a. Price Changes

To take price changes into account, it is useful to distinguish between two kinds of price changes. One is a "nominal" price change which results from changes in the purchasing power of the dollar, i.e., inflation or deflation. The amount of this nominal price change is indicated by the change in the general level of prices as measured by price indices.

The other type of price change which can occur is a change in "real" terms. This means that the price of a given good or service rises proportionately more or less than the change in the general purchasing power of the dollar, such that its price changes over time relative to the prices of other goods and services. For example, given that the general price index rose over a 5 year period from 1.00 to 1.30, the price of a piece of equipment which also rose 30 percent over the same period, would in effect remain constant in terms of its price relative to prices in general; i.e., its price would change in nominal terms but would remain constant in real terms.

b. Constant Increases

An approach often followed in investment analysis is to assume that all costs and revenues inflate at the same general rate, and that they therefore remain constant in real terms. With this assumption, renewal costs and other future expenses and benefits are evaluated at present prices. If there is reason to believe that certain items of costs or benefits will not inflate at the general rate, then their future values are adjusted only for the estimated real change, i.e., the effective change after taking into account the change in the general price level. If, for example, oil prices were expected to rise at a rate of 6 percent a year, the imputed real rise in oil prices would be 3 percent, i.e., half the rise would be considered due to general inflation, and half due to changing demand and supply conditions for oil relative to other commodities. Alternatively, if the future price of a good or service were, say, fixed at current levels by lease arrangements (without a cost escalation clause) the price would decline in real terms, in the face of inflation. It would be necessary to adjust the future payments by a price index prior to discounting them to present value in order to express them in constant dollars. Employing this procedure, the appropriate discount rate is a real rate, that is, one from which the inflation factor has been removed.

Assuming constant future prices can greatly simplify an analysis and in many applications will result in reasonably accurate results. However, the success of the approach rests not only on the assumption that future receipts and expenditures will respond fully and evenly to inflation, but also on the assumptions that tax considerations and the source of investment funds do not importantly affect the outcome — two assumptions which may be quite unrealistic for private investments.

c. Taxes

The real after-tax return to the firm may be substantially changed by inflation, even if pre-tax investment receipts and costs are assumed fully and equally responsive to inflation. Other things equal, inflation will tend to have a detrimental effect on an investment financed principally by equity funds, i.e., nonborrowed funds. Among the reasons are the following:

(1) tax deductions for depreciation are unresponsive to inflation;

(2) terminal value of equipment is responsive to inflation and will be reflected in the capital gains tax;

(3) tax deductions for interest on the borrowed portion of capital are unresponsive to inflation, such that the present value of the deductions diminishes overtime;

(4) inflation in receipts tends to move a firm into a higher tax bracket.

d. Borrowed Funds

However, in the case of an investment funded by a large proportion of borrowed funds it is possible that the after-tax return to equity may be maintained, or even improved, other things equal, in the face of inflation due to the following factor: The real value of the debt charges (amortized principle and interest) to finance the investment may decline because of the fixed nature of these payments. This gain to the borrower may equal or exceed his loss due to the fixed nature of depreciation and interest deductions.

e. Simplification

From a practical standpoint, methods of simplifying the treatment of inflation are desirable whenever they can be implemented without significantly altering the results.

3. Uncertainties of an Evaluation

a. Key Factors

Evaluation results depend directly on both the data estimates and the assumptions employed in the analysis. Among the key factors affecting the outcome of evaluations of energy recovery systems are the following:

1. the cost estimates for planning, purchasing, installing, and operating and maintaining the energy recovery systems;
2. the additional costs imposed by the investment, such as labor downtime and production loss;
3. the future rate of real price escalation in energy sources;
4. the amount of usable energy recovered;
5. the economic lives of system components, the length of intended use of the system, the salvage values at termination;
6. the discount rate used to convert future costs and benefits to a common time.

b. Correct Values

There will often be uncertainity as to the correct values to use in evaluating an investment. Uncertainty, which can be defined broadly as disparity between the predicted and the actual, encompasses two specific concepts: "risk," an event whose probability of occurence can be predicted; and "uncertainty," an event whose expected chance of occurence can not be predicted.

c. General Approaches

Three general approaches are used to dealing with uncertainty (used in the broad sense): (1) probability analysis, (2) sensitivity analysis, and (3) break-even analysis.

Probability analysis is generally used for situations for which the probability of an expected occurrence can be estimated; i.e., for evaluating risk. By multiplying the probability that an event will occur by the resulting dollar value if it does occur, it is possible to express costs and benefits as "expected values," rather than simply as "point" estimates.

Example No. 7

Find a more accurate cost of production downtime during an installation which was roughly estimated at $2000.

Solution:

The following might be calculated as a more accurate statement of cost:

Possible Situation	Probabilities	Cost if Situation Develops	Expected Cost
No difficulties in installation	20%	$1500	$ 300
No serious difficulty	70%	3000	2100
Serious difficulty	10%	7500	750
TOTALS	100%		$3150

Probability or expected value analysis, therefore, provides a method of incorporating uncertainity into the investment evaluation in a quantitative way. It requires, however, determination of probabilities.

Sensitivity analysis can be used to assess the consequences of assuming alternative values for the significant variables in the analysis. By determining the effect on the outcome of potential variation in a factor, the analyst identifies the degree of importance of that estimate or assumption and can then seek more information about it if desired. For example, the profitability of a furnace recuperator might be tested for sensitivity to the expected utilization rate of the furnace.

Break-even analysis, a third technique for dealing with uncertainity, focuses on a single key variable which is regarded as uncertain, and calculates the minimum (or maximum) value of the variable which is required to achieve a given outcome. For example, one might solve for the rate of escalation in fuel prices required for an investment in a heat exchanger to break even, other things given. To find the break-even escalation rate, an equation is developed which equates capital and maintenance costs with fuel cost savings. The "uncertain variable," i.e., the fuel escalation rate, is entered as an unknown, and the equation is solved for the break-even rate.

D APPLICATION OF EVALUATION METHODS TO DIFFERENT KINDS OF DECISIONS

In Table 14-7, the manager of a plant may be confronted with different kinds of investment decisions. The nature of the decision will influence which method of evaluation is preferred.

a. Single Project

One kind of decision is whether to accept or reject a particular investment project — where acceptance may mean implementation without further analysis or designation of the project as potentially profitable and worthy of further evaluation against other profitable projects. A simple accept/reject decision might be made, for example, if there were only one kind and size of energy recovery system feasible for a particular plant application. Any of the comprehensive evaluation methods described in this section generally could be used to determine if the expected benefits (or cost savings) from the investment would exceed the expected costs.

b. Mutual Exclusive Projects

Another kind of decision is choosing among mutually exclusive projects, where any of them would be expected to yield positive net benefits. This is the kind of choice which arises when doing one project precludes doing the others. For example, it might be determined that energy recovery would be desirable in a particular furnace via either a heat wheel or a plate exchanger. Since only one method of energy recovery would be used, a choice must be made between them.

Determining the efficient scale or size of investment in a given project is a special case of mutual exclusions, in that choosing one size of a project usually means rejecting the other possible sizes. The choice of time for undertaking a single project, that is, whether to begin it now or to delay it, is another special case of mutual exclusion.

Table 14-7 KINDS OF INVESTMENT DECISION PROBLEMS[a] (21)

Decisions	Examples
To accept or reject a given project.	Project A is accepted if its level of profitability meets the minimal acceptable level.
To choose between mutually exclusive projects.	If Project A is accepted, Project B is rejected; if size 1 of Project A is accepted, size 2 of Project A is rejected.
To decide priority among independent projects.	Project A is preferred to Project B, but both are undertaken if funding is adequate.
To determine the desirability of a prerequisite project.	Project A yields a low return, but Projects B and C which yield high returns cannot be undertaken until A is done.
No decision needed — mandatory project.	Project A is undertaken regardless of its expected return in order to meet mandated requirement.

[a]Here for simplicity the kinds of investment decisions are classified in an absolute sense. In practice they are usually in relative degree rather than absolute. For example, projects may be imperfect substitutes for one another rather than perfectly mutually exclusive; projects are seldom perfectly independent since capital usually must be rationed and manpower may be overlapping; projects may be imperfect complements in that acceptance of one *encourages* acceptance of another, rather than complete prerequisites where acceptance of one project *requires* acceptance of another.

The appropriate approach to evaluating mutually exclusive projects is generally to determine the choice which maximizes net benefits by using either the present value or annual net benefit method of evaluation. Assuming that there is no budget constraint, the project which maximizes net benefits is the most efficient choice.

c. Independent Projects

Yet another kind of decision that the manager may face is the choice among a set of project alternatives which are independent, that is, projects where the undertaking of one bears no influence on the acceptance of the others. In face of budgetary and other constraints, it is usuallly impossible for a firm to carry out all of its potentially profitable investment opportunities; a ranking mechanism is needed to identify those projects which are most profitable. For example, investment in energy recovery equipment may compete for funding with other cost-saving or revenue generating investments.

When there is a budget constraint, the ranking of individual independent projects according to their net benefits cannot be relied upon to result in selection of that subset of projects from the group which will yield the overall maximum benefits. This is because a project that yields higher total net benefits than another project may yield lower benefits per last dollar of investment spent, such that combined benefits for the two projects could be increased by transferring funds from the first project to the second. Choosing among independent projects according to either the benefit/cost ratio or the internal rate of return criterion until the available budget is exhausted will yield the maximum total net benefits for that budget. For example, Table 14-8 shows a possible gain of $40,000 by using the benefit/cost ratio method rather than the net benefits criterion to rank Projects 1, 2, and 3, when the budget is fixed at $200,000.

Table 14-8 NET BENEFITS AND BENEFIT/COST RATIO RANKINGS FOR A SET OF INDEPENDENT PROJECTS (21)

			Net benefits		Benefit/Cost	
Project	Benefits(B) ($ thousands)	Costs(C) ($ thousands)	($thousands)	Project ranking	Ratio	Project ranking
1	300	200	100	1	1.50	3
2	200	120	80	2	1.67	2
3	140	80	60	3	1.75	1

d. Prerequisite Project

A fourth kind of decision is determining the desirability of a project when it is a prerequisite to the undertaking of other projects. This type of decision may be particularly relevant to the evaluation of energy recovery systems, in that investment in energy recovery may in some cases be necessary to preserve the overall operation of a plant. This situation might arise in the face of severe curtailment of fuel allocations. Without some means of stretching available fuel supplies, plants might have to reduce production levels and lay off workers, thereby causing substantial loss of revenue, without equal reductions in costs. Alternatively, investment in energy recovery might be necessary in order to enable desired expansion of production in the face of short supplies of fuel, or to avoid costly changeover to substitute sources of energy.

If the project in question is a prerequisite for other activities of the firm, the evaluation of the prerequisite project should take into account the overall return expected not only from it, but also from the other activities whose success depends upon it. Even if the project shows a very small or negative rate or return when evaluated in terms of enabling other profitable activities to be undertaken. If the overall return to investment on the group of interdependent projects is attractive, funds should be allocated first to the prerequisite project.

In most cases, any of the comprehensive evaluation methods would be adequate to assess the importance of a project that is prerequisite to other functions of the firm. The essential factor is that all relevant costs and benefits attributable to the investment be included in the evaluation.

e. Mandated Projects

The final entry in Table 14-7 does not involve a decision. It is the case in which there is no decision to be made because the investment project must be undertaken by mandate, regardless of its profitability to the firm. Legislated requirements for specified levels and/or types of pollution control investments are examples. An important point to remember in this case is that other investments which are being considered, perhaps in conjunction with the required investment, should not be saddled with its costs and/or benefits. For example, the decision to add an energy recovery system to a legally required pollution incinerator; the investment evaluation should not lump the costs and benefits of the two systems together.

E SUMMARY

Each of the investment evaluation methods has its particular advantages and disadvantages, and will be a useful decision criterion in certain cases. For most decision problems, the net present value or the net annual value method, supplemented by benefit/cost ratios or internal rates of return, will provide adequate measures for economically efficient investment decisions.

Table 14-9
6% COMPOUND INTEREST FACTORS[a] (21)

	Single payment			Uniform series			
n	Compound amount factor F/P	Present worth factor P/F	Sinking fund factor A/F	Capital recovery factor A/P	Compound amount factor F/A	Present worth factor P/A	n
1	1.0600	0.9434	1.000 00	1.060 00	1.000	0.943	1
2	1.1236	.8900	0.485 44	0.545 44	2.060	1.833	2
3	1.1910	.8396	.314 11	.374 11	3.184	2.673	3
4	1.2625	.7921	.228 59	.228 59	4.375	3.465	4
5	1.3382	.7473	.177 40	.273 40	5.637	4.212	5
6	1.4185	.7050	.143 36	.203 36	6.975	4.917	6
7	1.5036	.6651	.119 14	.179 14	8.394	5.582	7
8	1.5938	.6274	.101 04	.161 04	9.897	6.210	8
9	1.6895	.5919	.087 02	.147 02	11.491	6.802	9
10	1.7908	.5584	.075 87	.135 87	13.181	7.360	10
11	1.8983	.5268	.066 79	.126 79	14.972	7.887	11
12	2.0122	.4970	.059 28	.119 28	16.870	8.384	12
13	2.1329	.4688	.052 96	.112 96	18.882	8.853	13
14	2.2609	.4423	.047 58	.107 58	21.015	9.295	14
15	2.3966	.4173	.042 96	.102 96	23.276	9.712	15
16	2.5404	.3936	.038 95	.098 95	25.673	10.106	16
17	2.6928	.3714	.035 44	.095 44	28.213	10.477	17
18	2.8543	.3503	.032 36	.092 36	30.906	10.828	18
19	3.0256	.3305	.029 62	.089 62	33.760	11.158	19
20	3.2071	.3118	.027 18	.087 18	36.786	11.470	20
21	3.3996	.2942	.025 00	.085 00	39.993	11.764	21
22	3.6035	.2775	.023 05	.083 05	43.392	12.042	22
23	3.8197	.2618	.021 28	.081 28	46.996	12.303	23
24	4.0489	.2470	.019 68	.079 68	50.816	12.550	24
25	4.2919	.2330	.018 23	.078 23	54.865	12.783	25
26	4.5494	.2198	.016 90	.076 90	59.156	13.003	26
27	4.8223	.2074	.015 70	.075 70	63.706	13.211	27
[illegible]	5.1117	.1956	.014 59	.074 59	68.528	13.406	28
[illegible]	5.4184	.1846	.013 58	.073 58	73.640	13.591	29
[illegible]	5.7435	.1741	.012 65	.072 65	79.058	13.765	30
31	6.0881	.1643	.011 79	.071 79	84.802	13.929	31
32	6.4534	.1550	.011 00	.071 00	90.890	14.084	32
33	6.8406	.1462	.010 27	.070 27	97.343	14.230	33
34	7.2510	.1379	.009 60	.069 60	104.184	14.368	34
35	7.6861	.1301	.008 97	.068 97	111.435	14.498	35
40	10.2857	.0972	.006 46	.066 46	154.762	15.046	40
45	13.7646	.0727	.004 70	.064 70	212.744	15.456	45
50	18.4202	.0543	.003 44	.063 44	290.336	15.762	50
55	24.6503	.0406	.002 54	.062 54	394.172	15.991	55
60	32.9877	.0303	.001 88	.061 88	533.128	16.161	60
65	44.1450	.0227	.001 39	.061 39	719.083	16.289	65
70	59.0759	.0169	.001 03	.061 03	967.932	16.385	70
75	79.0569	.0126	.000 77	.060 77	1 300.949	16.456	75
80	105.7960	.0095	.000 57	.060 57	1 746.600	16.509	80
85	141.5789	.0071	.000 43	.060 43	2 342.982	16.549	85
90	189.4645	.0053	.000 32	.060 32	3 141.075	16.579	90
95	253.5463	.0039	.000 24	.060 24	4 209.104	16.601	95
100	339.3021	.0029	.000 18	.060 18	5 638.368	16.618	100

[a] The letter notations in the column headings, e.g., *F/P*, *P/F*, etc., indicate the operation to be performed with the factors in that column. For example, *F/P* indicates that the single compound amount factors are used to find the future value of a given present amount.

Table 14-10
8% COMPOUND INTEREST FACTORS (21)

	Single payment			Uniform series			
n	Compound amount factor F/P	Present worth factor P/F	Sinking fund factor A/F	Capital recovery factor A/P	Compound amount factor F/A	Present worth factor P/A	n
1	1.0800	0.9259	1.000 00	1.080 00	1.000	0.926	1
2	1.664	.8573	0.480 77	0.560 77	2.080	1.783	2
3	1.2597	.7938	.308 03	.388 03	3.246	2.577	3
4	1.3605	.7350	.221 92	.301 92	4.506	3.312	4
5	1.4693	.6806	.170 46	.250 46	5.867	3.993	5
6	1.5869	.6302	.136 32	.216 32	7.336	4.623	6
7	1.7138	.5835	.112 07	.192 07	8.923	5.206	7
8	1.8509	.5403	.094 01	.174 01	10.637	5.747	8
9	1.9990	.5002	.080 08	.160 08	12.488	6.247	9
10	2.1589	.4632	.069 03	.149 03	14.487	6.710	10
11	2.3316	.4289	.060 08	.140 08	16.645	7.139	11
12	2.5182	.3971	.052 70	.132 70	18.977	7.536	12
13	2.7196	.3677	.046 52	.126 52	21.495	7.904	13
14	2.9372	.3405	.041 30	.121 30	24.215	8.244	14
15	3.1722	.3152	.036 83	.116 83	27.152	8.559	15
16	3.4259	.2919	.032 98	.112 98	30.324	8.851	16
17	3.7000	.2703	.029 63	.109 63	33.750	9.122	17
18	3.9960	.2502	.026 70	.106 70	37.450	9.372	18
19	4.3157	.2317	.024 13	.104 13	41.446	9.604	19
20	4.6610	.2145	.021 85	.101 85	45.762	9.818	20
21	5.0338	.1987	.019 83	.099 83	50.423	10.017	21
22	5.4365	.1839	.018 03	.098 03	55.457	10.201	22
23	5.8715	.1703	.016 42	.096 42	60.893	10.371	23
24	6.3412	.1577	.014 98	.094 98	66.765	10.529	24
25	6.8485	.1460	.013 68	.093 68	73.106	10.675	25
26	7.3964	.1352	.012 51	.092 51	79.954	10.810	26
27	7.9881	.1252	.011 45	.091 45	87.351	10.935	27
28	8.6271	.1159	.010 49	.090 49	95.339	11.051	28
29	9.3173	.1073	.009 62	.089 62	103.966	11.158	29
30	10.0627	.0994	.008 83	.088 83	113.283	11.258	30
31	10.8677	.0920	.008 11	.088 11	123.346	11.350	31
32	11.7371	.0852	.007 45	.087 45	134.214	11.435	32
33	12.6760	.0789	.006 85	.086 85	145.951	11.514	33
34	13.6901	.0730	.006 30	.086 30	158.627	11.587	34
35	14.7853	.0676	.005 80	.085 80	172.317	11.655	35
40	21.7245	.0460	.003 86	.083 86	259.057	11.925	40
45	31.9204	.0313	.002 59	.082 59	386.506	12.108	45
50	46.9016	.0213	.001 74	.081 74	573.770	12.233	50
55	68.9139	.0145	.001 18	.081 18	848.923	12.319	55
60	101.2571	.0099	.000 80	.080 80	1253.213	12.377	60
65	148.7798	.0067	.000 54	.080 54	1847.248	12.416	65
70	218.6064	.0046	.000 37	.080 37	2720.080	12.443	70
75	321.2045	.0031	.000 25	.080 25	4002.557	12.461	75
80	471.9548	.0021	.000 17	.080 17	5886.935	12.474	80
85	693.4565	.0014	.000 12	.080 12	8655.706	12.482	85
90	1018.9151	.0010	.000 08	.080 08	12723.939	12.488	90
95	1497.1205	.0007	.000 05	.080 05	18701.507	12.492	95
100	2199.7613	.0005	.000 04	.080 04	27484.516	12.494	100

Table 14-11
10% COMPOUND INTEREST FACTORS (21)

	Single payment		Uniform series				
n	Compound amount factor F/P	Present worth factor P/F	Sinking fund factor A/F	Capital recovery factor A/P	Compound amount factor F/A	Present worth factor P/A	
1	1.1000	0.9091	1.000 00	1.100 00	1.000	0.909	1
2	1.2100	.8264	0.476 19	0.576 19	2.100	1.736	2
3	1.3310	.7513	.302 11	.402 11	3.310	2.487	3
4	1.4641	.6830	.215 47	.315 47	4.641	3.170	4
5	1.6105	.6209	.163 80	.263 80	6.105	3.791	5
6	1.7716	.5645	.129 61	.229 61	7.716	4.355	6
7	1.9487	.5132	.105 41	.205 41	9.487	4.868	7
8	2.1436	.4665	.087 44	.187 44	11.436	5.335	8
9	2.3579	.4241	.073 64	.173 64	13.579	5.759	9
10	2.5937	.3855	.062 75	.162 75	15.937	6.144	10
11	2.8531	.3505	.053 96	.153 96	18.531	6.495	11
12	3.1384	.3186	.046 76	.146 76	21.384	6.814	12
13	3.4523	.2897	.040 78	.140 78	24.523	7.103	13
14	3.7975	.2633	.035 75	.135 75	27.975	7.367	14
15	4.1772	.2394	.031 47	.131 47	31.772	7.606	15
16	4.5950	.2176	.027 82	.127 82	35.950	7.824	16
17	5.0545	.1978	.024 66	.124 66	40.545	8.022	17
18	5.5599	.1799	.021 93	.121 93	45.599	8.201	18
19	6.1159	.1635	.019 55	.119 55	51.159	8.365	19
20	6.7275	.1486	.017 46	.117 46	57.275	8.514	20
21	7.4002	.1351	.015 62	.115 62	64.002	8.649	21
22	8.1403	.1228	.014 01	.114 01	71.403	8.772	22
23	8.9543	.1117	.012 57	.112 57	79.543	8.883	23
24	9.8497	.1015	.011 30	.111 30	88.497	8.985	24
25	10.8347	.0923	.010 17	.110 17	98.347	9.077	25
26	11.9182	.0839	.009 16	.109 16	109.182	9.161	26
27	13.1100	.0763	.008 26	.108 26	121.100	9.237	27
28	14.4210	.0693	.007 45	.107 45	134.210	9.307	28
29	15.8631	.0630	.006 73	.106 73	148.631	9.370	29
30	17.4494	.0573	.006 08	.106 08	164.494	9.427	30
31	19.1943	.0521	.005 50	.105 50	181.943	9.479	31
32	21.1138	.0474	.004 97	.104 97	201.138	9.526	32
33	23.2252	.0431	.004 50	.104 50	222.252	9.569	33
34	25.5477	.0391	.004 07	.104 07	245.477	9.609	34
35	28.1024	.0356	.033 69	.103 69	271.024	9.644	35
40	45.2593	.0221	.002 26	.102 26	442.593	9.779	40
45	72.8905	.0137	.001 39	.101 39	718.905	9.863	45
50	117.3909	.0085	.000 86	.100 86	1 163.909	9.915	50
55	189.0591	.0053	.000 53	.100 53	1 880.591	9.947	55
60	304.4816	.0033	.000 33	.100 33	3 034.816	9.967	60
65	490.3707	.0020	.000 20	.100 20	4 893,707	9.980	65
70	789.7470	.0013	.000 13	.100 13	7 887.470	9.987	70
75	1271.8952	.0008	.000 08	.100 08	12 708.954	9.992	75
80	2048.4002	.0005	.000 05	.100 05	20 474.002	9.995	80
85	3298.9690	.0003	.000 03	.100 03	32 979.690	9.997	85
90	5313.0226	.0002	.000 02	.100 02	53 120.226	9.998	90
95	8556.6760	.0001	.000 01	.100 01	85 556.760	9.999	95
100	13780.6123	.0001	.000 01	.100 01	137 796.123	9.999	100

Table 14-12
12% COMPOUND INTEREST FACTORS (21)

	Single payment		Uniform series				
n	Compound amount factor F/P	Present worth factor P/F	Sinking fund factor A/F	Capital recovery factor A/P	Compound amount factor F/A	Present worth factor P/A	n
1	1.1200	0.8929	1.000 00	1.120 00	1.000	0.893	1
2	1.2544	.7972	0.471 70	0.591 70	2.120	1.690	2
3	1.4049	.7118	.296 35	.416 35	3.374	2.402	3
4	1.5735	.6355	.209 23	.329 23	4.779	3.037	4
5	1.7623	.5674	.157 41	.277 41	6.353	3.605	5
6	1.9738	.5066	.123 23	.243 23	8.115	4.111	6
7	2.2107	.4523	.099 12	.219 12	10.089	4.564	7
8	2.4760	.4039	.081 30	.201 30	12.300	4.968	8
9	2.7731	.3606	.067 68	.187 68	14.776	5.328	9
10	3.1058	.3220	.056 98	.176 98	17.549	5.650	10
11	3.4785	.2875	.048 42	.168 42	20.655	5.938	11
12	3.8960	.2567	.041 44	.161 44	24.133	6.194	12
13	4.3635	.2292	.035 68	.155 68	28.029	6.424	13
14	4.8871	.2046	.030 87	.150 87	32.393	6.628	14
15	5.4736	.1827	.026 82	.146 82	37.280	6.811	15
16	6.1304	.1631	.023 39	.143 39	42.753	6.974	16
17	6.8660	.1456	.020 46	.140 46	48.884	7.120	17
18	7.6900	.1300	.017 94	.137 94	55.750	7.250	18
19	8.6128	.1161	.015 76	.135 76	63.440	7.366	19
20	9.6463	.1037	.013 88	.133 88	72.052	7.469	20
21	10.8038	.0926	.012 24	.132 24	81.699	7.562	21
22	12.1003	.0826	.010 81	.130 81	92.503	7.645	22
23	13.5523	.0738	.009 56	.129 56	104.603	7.718	23
24	15.1786	.0659	.008 46	.128 46	118.155	7.784	24
25	17.0001	.0588	.007 50	.127 50	133.334	7.843	25
26	19.0401	.0525	.006 65	.126 65	150.334	7.896	26
27	21.3249	.0469	.005 90	.125 90	169.374	7.943	27
28	23.8839	.0419	.005 24	.125 24	190.699	7.984	28
29	26.7499	.0374	.004 66	.124 66	214.583	8.022	29
30	29.9599	.0334	.004 14	.124 14	241.333	8.055	30
31	33.5551	.0298	.003 69	.123 69	271.292	8.085	31
32	37.5817	.0266	.003 28	.123 28	304.847	8.112	32
33	42.0915	.0238	.002 92	.122 92	342.429	8.135	33
34	47.1425	.0212	.002 60	.122 60	384.520	8.157	34
35	52.7996	.0189	.002 32	.122 32	431.663	8.176	35
40	93.0510	.0107	.001 30	.121 30	767.091	8.244	40
45	163.9876	.0061	.000 74	.120 74	1 358.230	8.283	45
50	289.0022	.0035	.000 42	.120 42	2 400.018	8.305	50
∞				.120 00		8.333	∞

Table 14-13
15% COMPOUND INTEREST FACTORS (21)

	Single payment		Uniform series				
n	Compound amount factor F/P	Present worth factor P/F	Sinking fund factor A/F	Capital recovery factor A/P	Compound amount factor F/A	Present worth factor P/A	n
1	1.1500	0.8696	1.000 00	1.150 00	1.000	0.870	1
2	1.3225	.7561	.465 12	.615 12	2.150	1.626	2
3	1.5209	.6575	.287 98	.437 98	3.472	2.283	3
4	1.7490	.5718	.200 26	.350 27	4.993	2.855	4
5	2.0114	.4972	.148 32	.298 32	6.742	3.352	5
6	2.3131	.4323	.114 24	.264 24	8.754	3.784	6
7	2.6600	.3759	.090 36	.240 36	11.067	4.160	7
8	3.0590	.3269	.072 85	.222 85	13.727	4.487	8
9	3.5179	.2843	.059 57	.290 57	16.786	4.772	9
10	4.0456	.2472	.049 25	.199 25	20.304	5.019	10
11	4.6524	.2149	.041 07	.191 07	24.349	5.234	11
12	5.3503	.1869	.034 48	.184 48	29.002	5.421	12
13	6.1528	.1625	.029 11	.179 11	34.352	5.583	13
14	7.0757	.1413	.024 69	.174 69	40.505	5.724	14
15	8.1371	.1229	.021 02	.171 02	47.580	5.847	15
16	9.3576	.1069	.017 95	.167 95	55.717	5.954	16
17	10.7613	.0929	.015 37	.165 37	65.075	6.047	17
18	12.3755	.0808	.013 19	.163 19	75.836	6.128	18
19	14.2318	.0703	.011 34	.161 34	88.212	6.198	19
20	16.3665	.0611	.009 76	.159 76	102.444	6.259	20
21	18.8215	.0531	.008 42	.158 42	118.810	6.312	21
22	21.6447	.0462	.007 27	.157 27	137.632	6.359	22
23	24.8915	.0402	.006 28	.156 28	159.276	6.399	23
24	28.6252	.0349	.005 43	.155 43	184.168	6.434	24
25	32.9190	.0304	.004 70	.154 70	212.793	6.464	25
26	37.8568	.0264	.004 07	.154 07	245.712	6.491	26
27	45.5353	.0230	.003 53	.153 53	283.569	6.514	27
28	50.0656	.0200	.003 06	.153 06	327.104	6.534	28
29	57.5755	.0174	.002 65	.152 65	377.170	6.551	29
30	66.2118	.0151	.002 30	.152 30	434.745	6.566	30
31	76.1435	.0131	.002 00	.152 00	500.957	6.579	31
32	87.5651	.0114	.001 73	.151 73	577.100	6.591	32
33	100.6998	.0099	.001 50	.151 50	664.666	6.600	33
34	115.8048	.0086	.001 31	.151 31	765.365	6.609	34
35	133.1755	.0075	.001 13	.151 13	881.170	6.617	35
40	267.8635	.0037	.00 56	.150 56	1 779.090	6.642	40
45	538.7693	.0019	.000 28	.150 28	3 585.128	6.654	45
50	1 083.6574	.0009	.000 14	.150 14	7 217.716	6.661	50
∞				.150 00		6.667	∞

Table 14-14
20% COMPOUND INTEREST FACTORS (21)

	Single payment		Uniform series				
n	Compound amount factor F/P	Present worth factor P/F	Sinking fund factor A/F	Capital recovery factor A/P	Compound amount factor F/A	Present worth factor P/A	n
1	1.2000	0.8333	1.000 00	1.200 00	1.000	0.833	1
2	1.4400	.6944	0.454 55	0.654 55	2.200	1.528	2
3	1.7280	.5787	.274 73	.474 73	3.640	2.106	3
4	2.0736	.4823	.186 29	.386 29	5.368	2.589	4
5	2.4883	.4019	.134 38	.334 38	7.442	2.991	5
6	2.9860	.3349	.100 71	.300 71	9.930	3.326	6
7	3.5832	.2791	.077 42	.277 42	12.916	3.605	7
8	4.2998	.2326	.060 61	.260 61	16.499	3.837	8
9	5.1598	.1938	.048 08	.248 08	20.799	4.031	9
10	6.1917	.1615	.038 52	.238 52	25.959	4.192	10
11	7.4301	.1346	.031 10	.231 10	32.150	4.327	11
12	8.9161	.1122	.025 26	.225 26	39.581	4.439	12
13	10.6993	.0935	.020 62	.220 62	48.497	4.533	13
14	12.8392	.0779	.016 89	.216 89	59.196	4.611	14
15	15.4070	.0649	.013 88	.213 88	72.035	4.675	15
16	18.4884	.0541	.011 44	.211 44	87.442	4.730	16
17	22.1861	.0451	.009 44	.209 44	105.931	4.775	17
18	26.6233	.0376	.007 81	.207 81	128.117	4.812	18
19	31.9480	.0313	.006 46	.206 46	154.740	4.844	19
20	38.3376	.0261	.005 36	.205 36	186.688	4.870	20
21	46.0051	.0217	.004 44	.204 44	225.026	4.891	21
22	55.2061	.0181	.003 69	.203 69	271.031	4.909	22
23	66.2474	.0151	.003 07	.203 07	326.237	4.925	23
24	79.4968	.0126	.002 55	.202 55	392.484	4.937	24
25	95.3962	.0105	.002 12	.202 12	471.981	4.948	25
26	114.4755	.0087	.001 76	.201 76	567.377	4.956	26
27	137.3706	.0073	.001 47	.201 47	681.853	4.964	27
28	164.8447	.0061	.001 22	.201 22	819.223	4.970	28
29	197.8136	.0051	.001 02	.201 02	984.068	4.975	29
30	237.3763	.0042	.000 85	.200 85	1 181.882	4.979	30
31	284.8516	.0035	.000 70	.200 70	1 419.258	4.982	31
32	341.8219	.0029	.000 59	.200 59	1 704.109	4.985	32
33	410.1863	.0024	.000 49	.200 49	2 045.931	4.988	33
34	492.2235	.0020	.000 41	.200 41	2 456.118	4.990	34
35	590.6682	.0017	.000 34	.200 34	2 948.341	4.992	35
40	1 469.7716	.0007	.000 14	.200 14	7 343.858	4.997	40
45	3 657.2620	.0003	.000 05	.200 05	18 281.310	4.999	45
50	9 100.4382	.0001	.000 02	.200 02	45 497.191	4.999	50
∞				.200 00		5.000	∞

Table 14-15
25% COMPOUND INTEREST FACTORS (21)

	Single payment		Uniform series				
n	Compound amount factor F/P	Present worth factor P/F	Sinking fund factor A/F	Capital recovery factor A/P	Compound amount factor F/A	Present worth factor P/A	n
1	1.2500	0.8000	1.000 00	1.250 00	1.000	0.800	1
2	1.5625	.6400	.444 44	.694 44	2.250	1.440	2
3	1.9531	.5120	.262 30	.512 30	3.813	1.952	3
4	2.4414	.4096	.173 44	.423 44	5.766	2.362	4
5	3.0518	.3277	.121 85	.371 85	8.207	2.689	5
6	3.8147	.2621	.088 82	.338 82	11.259	2.951	6
7	4.7684	.2097	.066 34	.316 34	15.073	3.161	7
8	5.9605	.1678	.050 40	.300 40	19.842	3.329	8
9	7.4506	.1342	.038 76	.288 76	25.802	2.463	9
10	9.3132	.1074	.030 07	.280 07	33.253	3.571	10
11	11.6415	.0859	.023 49	.273 49	42.566	3.656	11
12	14.5519	.0687	.018 45	.268 45	54.208	3.725	12
13	18.1899	.0550	.014 54	.264 54	68.760	3.780	13
14	22.7374	.0440	.011 50	.261 50	86.949	3.824	14
15	28.4217	.0352	.009 12	.259 12	109.687	3.859	15
16	35.5271	.0281	.007 24	.257 24	138.109	3.887	16
17	44.4089	.0225	.005 76	.255 76	173.636	3,910	17
18	55.5112	.0180	.004 59	.254 59	218.045	3.928	18
19	69.3889	.0144	.003 66	.253 66	273.556	3.942	19
20	86.7362	.0115	.002 92	.252 92	342.945	3.954	20
21	108.4202	.0092	.002 33	.252 33	429.681	3.963	21
22	135.5253	.0074	.001 86	.251 86	538.101	3.970	22
23	169.4066	.0059	.001 48	.251 48	673.626	3.976	23
24	211.7582	.0047	.001 19	.251 19	843.033	3.981	24
25	264.6978	.0038	.000 95	.250 95	1 054.791	3.985	25
26	330.8722	.0030	.000 76	.250 76	1 319.489	3.988	26
27	413.5903	.0024	.000 61	.250 61	1 650.361	3.990	27
28	516.9879	.0019	.000 48	.250 48	2 063.952	3.992	28
29	646.2349	.0015	.000 39	.250 39	2 580.939	3.994	29
30	807.7936	.0012	.000 31	.250 31	3 227.174	3.995	30
31	1 009.7420	.0010	.000 25	.250 25	4 034.968	3.996	31
32	1 262.1774	.0008	.000 20	.250 20	5 044.710	3.997	32
33	1 577.7218	.0006	.000 16	.250 16	6 306.887	3.997	33
34	1 972.1523	.0005	.000 13	.250 13	7 884.609	3.998	34
35	2 465.1903	.0004	.000 10	.250 10	9 856.761	3.998	35
40	7 523.1638	.0001	.000 03	.250 03	30 088.655	3.999	40
45	22 958.8740	.0001	.000 01	.250 01	91 831.496	4.000	45
50	70 064.9232	.0000	.000 00	.250 00	280 255.693	4.000	50
∞				250 00		4.000	∞

SECTION XV
AREAS OF APPLICATION

This section will present some ideas on where to look and how to look for air-to-air energy recovery opportunities. To do this, one must be thoroughly familiar with energy recovery devices and their application. There is a potential application for air-to-air energy recovery anytime there are two or more airstreams at different temperatures, and/or enthalpy levels. There must be energy to recover and a simultaneous use for the recovered energy. These general requirements create an immense opportunity for the application of air-to-energy devices, and they exist in the commercial, institutional, industrial and residential sectors. These opportunities can be either in new construction or in the replacement/retrofit areas.

None of the energy recovery devices can be used for every application. One must be familiar with the strengths and weaknesses of each device in regard to temperature, construction, reaction to effluents (both exhaust and supply), frost control, capacity control, location of airstreams, sensible or latent capability, and maintenance requirements. The "economics" of an application must be understood. In spite of life-cycle savings, the system first cost "pressure" is still a primary consideration. On new installations, the first cost of the heat recovery system may be offset by the savings in first cost of cooling and/or heating equipment. In other situations, the main interest may be in saving as much energy as possible because of a fuel availability problem. In these cases, first cost "pressure" should not be as much of a factor.

A MARKET POTENTIAL

It is difficult to precisely assess the dollar potential of the air-to-air recovery market. There are important variables which directly influence the rate of growth of this market. The variables are fuel costs, fuel availability, and Federal legislation. If these variables are in the right proportions, the market becomes virtually limitless.

Opportunities for air-to-air energy recovery occur in every sector. The discussion to follow categorizes the sectors by commercial, institutional, industrial and residential. In each sector are found several other sub-categories by:

1. Temperature level (general)

Low	*Medium*	*High*
Comfort Range	to 400°F	400°F+

2. Nature of effluents
   a. Is the air basically "dirty" or "clean"?
   b. Are contaminants present which are:
      1) corrosive?
      2) toxic?
      3) flammable?
3. Latent or sensible recovery.
4. Process-to-Process, Process-to-Comfort, Comfort-to-Comfort.
5. System to have controlled capacity or "run wild."

It might be assumed that the dollar potential of the market is nearly uniform across the commercial, institutional and industrial sectors, although the greatest amount of exhausted high temperature gases is in the industrial sector processes. It is also safe to say that the residential sector represents an insignificant percent of the total. Overall market potential is hard to determine. Estimates range from $20 million to $450 billion. A realistic range is probably $200 million to $1 billion per year. If the variables mentioned earlier were to have their full effect, then $450 billion per year might become a reality.

B COMMERCIAL

Applications in this area are primarily confined to the ventilation air requirements of the structure. Temperatures are in the low range. It is usually a comfort-to-comfort application and latent (total) heat recovery is desirable. These applications are more easily justified when the ventilation load is a large share of the building load.

Virtually every HVAC application presents an opportunity. Prime examples are found in the restaurant/kitchen areas and also laundry areas of commercial structures.

C INSTITUTIONAL

This area also provides excellent application opportunities. This is because of the high ventilation air requirements found due to the handling or processing of food and/or chemicals. Typically, the applications are low temperature, comfort-to-comfort, or process-to-comfort. There are often "dirty" air conditions on the exhaust.

Common applications are laboratories (including animal), hospitals, auditoriums, swimming pools and gymnasiums. Nursing homes can provide fertile ground because of the kitchen and laundry facilities.

The retrofit potential in this area is immense. There is great pressure to reduce health care and educational costs. Air-to-air energy recovery devices can contribute more to energy savings than any other type of energy conservation measure in the institutional sector.

D INDUSTRIAL

The industrial area offers the largest potential for growth of energy recovery. There is a fundamental explanation for this. If either legislation or natural actions occur that can curtail industrial fuel consumption, the only way that industry can sustain growth is by improving energy efficiency. The amount of waste energy to be recovered is phenomenal. In the industrial sector, it is estimated that 45-55% of the energy consumed is wasted.

Applications in this sector are characterized as mid to high temperature, dirty, and process-to-process, or process-to-comfort. Energy recovery devices may require capacity control, although in many cases, are allowed to run wild. To avoid overheating of the recovery systems, automatic bypass systems can be incorporated. This insures that the process can continue uninterrupted in the event it is necessary to shut down the recovery system.

The industrial area provides many opportunities for other types of energy recovery. The discussion thus far has concerned air-to-air devices. However, there are several air-to-fluid, or fluid-to-air applications. Fin tube exchangers are used for the heat transfer surface. A common air-to-fluid application is the preheating of boiler feedwater using combustion exhaust. Lower temperature applications would include the tempering or preheating of make-up water from the waste heat in the airstreams from electrical transformer and mechanical equipment rooms.

Fluid-to-air applications occur everywhere water is used in cooling a process. An example is the hot oil used in heat treating that must be cooled with well water or cooling tower water. The hot water that is usually discarded could be used for satisfying a make-up air heating requirement.

Make-up air requirements provide another prime opportunity for air-to-air energy recovery. These applications are low temperature, comfort-to-comfort, tend to be "dirty," and usually require capacity and frost control. This sector offers both new construction and retrofit opportunities.

E RESIDENTIAL

This area provides some air-to-air energy recovery opportunities. The more common device is a packaged plate-fin exchanger with a small circulating fan on the cold air stream. Devices are usually counterflow and mount in the furnace exhaust pipe.

Price is a big factor. In retrofit situations the homeowner may be able to save more by making the same investment in improved or additional insulation, more sophisticated controls, storm windows, or caulking of cracks.

In new home construction, the potential buyer is usually under great first cost pressure. He may not be willing to invest in something that is justified on payback. There are many home "extras" that are more appealing (e.g., new appliances, finishing the basement, adding an extra room, landscaping, etc.).

F GENERAL

Some of the more common areas where air-to-air heat recovery can be applied are listed in Table 15-1. However, the first cost economics of the potential application of a recovery system is still the greatest factor in the determination of whether the installation will be made or not. Tax credits (if available) and fuel shortages will certainly cause many installations to be made. Mandatory regulations such as new construction building "energy budgets" which are part of the mandated performance standards in the Federal "Energy Conservation Standards for New Buildings Act of 1976" (Title III of PL 94-385) will cause energy recovery systems to be designed into the more energy efficient new buildings. Similar to the resistance met by the air-conditioning industry in the 1930's because people were not familiar with its benefits, the energy recovery industry must educate the public to where "energy recovery" (and its benefits) becomes a household word. Then the first cost economics factor will become less important, as the on-going cost benefits (savings) will accelerate the sales of systems.

TABLE 15-1 APPLICATIONS OF ENERGY RECOVERY

Comfort-to-Comfort	*Process-to-Comfort*	*Process-to-Process*
Hospitals	Kitchens	Heat Treating
Hotels	Laundries	General Drying
Laboratories	Food Drying	Milk Drying
Schools	Bake Ovens	Grain Drying
Plants	Gen. Drying Ovens	Foundries
Restaurants	Print Presses	Paint Baking
Recreational Facilities (swimming pools)	Annealing Furnaces	Coffee Drying
Animal Care Facilities		Print Presses
Paint Booths		Malt Processing
		Curing Operations
		Pulp & Paper Industry

SECTION XVI
MAINTENANCE AND FIELD TESTING

The performance of any energy recovery device, once it is selected and the total system is properly designed for a particular application, is based on a good installation following accepted practices, testing and adjusting for proper operation, and good maintenance procedures. This section will basically deal with energy recovery systems after they have been installed.

A SYSTEM START-UP

Prior to the start of the testing and adjusting procedures for the total energy recovery system, including the related process or HVAC systems, all equipment and related components should be checked as listed below.

1. Fans and Rotary Wheels
   a) Confirm that the equipment has been checked for:
      Rotation
      Lubrication
      Belt tension
      Wheel-housing clearance
      Motor fastening
      Absence of foreign objects
      Duct-flexible connector-fan alignment
      Alignment of drive
      Keyway and set screw tightness
      Clean condition
      Vibration isolation adjustment
      Drain outlets for moisture
      Static pressure control, if any
   b) Investigate and locate all start-stop, disconnect and circuit interruption devices.
   c) Inspect duct inlet and outlet conditions which might deter performance.
2. Air Conditioning Units
   a) Follow the fan check list generally.
   b) In addition inspect the air flow pattern from the outside air intake to the fan discharge.
   c) Investigate air intake provisions to insure an adequate air supply on fan start.
   d) Confirm installation of filter media.
3. Duct System
   a) Verify that all outside air intake, return air and exhaust air dampers are operable.
   b) Confirm that all system volume dampers and fire dampers are installed and in the full open position.
   c) Inspect access doors and hardware for tightness and leakage.
   d) Verify that all air terminals have been installed and that terminal dampers are fully open.
   e) Inspect coils, energy recovery devices, duct heaters, and terminals for leakage at duct connections and piping penetrations.
   f) Confirm locations for Pitot tube traverse measurements and accessibility for measurements in general.
   g) Confirm openings in walls above ceilings for air passage.
   h) Confirm that all architectural features such as doors and ceilings are installed and are functional with regard to air circuits.
4. Pumps
   a) Confirm alignment, coupling, grouting and fastening of pump installations.
   b) Bleed air from pump casings.
   c) Inspect vibration isolation systems at pumps and inspect alignment and restraint of flexible connections.
   d) Confirm pump and motor lubrication.
   e) Verify identification and rotation of pumps.
5. Spray Towers
   a) Spray tower fan sections should be checked as outlined under fans. Towers should be checked for proper fluid level, float setting, spray pattern, and make-up water availability.
   b) Verify the adjustment of controls as required.
   c) Inspect vibration isolation system.
6. Coils and Heat Exchangers
   a) Inspect face areas of all coils for fin damage that could affect air flow.
   b) Confirm provisions in piping for flow and temperature measurements.
   c) Confirm open position of water valves. Vent as necessary.
   d) Inspect for excess air leakage from tube sheet at tubes.

e) Inspect return bends and tubes for damage.
f) Confirm that piping is connected properly for counter or parallel flow heat transfer.

7. Piping System
   a) Confirm that the system has been hydrostatically tested, filled, flushed, refilled and vented as required.
   b) Confirm that strainers have been cleaned.
   c) Inspect pressure reducing valve operation for both system values and makeup valves.
   d) Confirm that all manual valves are in the open position, and all automatic valves in the proper position.
   e) Inspect the water level in the expansion tank.
   f) Confirm accessibility into ceilings and walls for adjustment of balancing valves and access to flow meters.

B PERFORMANCE TESTING

The following guide on performance testing is based on a total heat rotary wheel exchanger, which is probably the most complex to test. The principles would apply to almost all of the other heat exchangers.

In all cases, manufacturer's installation and operating manuals should be consulted prior to system start-up, but in general, there are *four* key factors to consider before a heat wheel is installed:

a. *Drive Access:* Take care to install the wheel such that service personnel have access to the drive motor. Motor location varies widely by manufacturer; the majority have drive motors located in the wheel casing but access can be through the face of the unit or through the side panels.

b. *Wheel Access:* Good design practice should allow for the possibility of wheel replacement. Since designs vary widely, consult individual manufacturers for access recommendations, but side access of ½ wheel diameter is a good rule to follow. Additionally, access to both wheel faces will allow the exchanger to be periodically cleaned in accordance with the manufacturer's recommendations.

c. *Fan Replacement:* Fans can be located on either the entering or leaving sides of the wheel, but optimum design would place both the supply and exhaust fans drawing air through the wheel.

d. *Mounting:* Most wheels can be mounted in either the horizontal or vertical plane. However, when the exchanger is mounted horizontally, insure that the manufacturer is aware of the fact so that thrust bearings and casing support are properly located.

The performance of any heat exchanger is measured by its effectiveness, the ratio of the actual rate of heat transfer to the maximum possible rate. The total heat rotary wheel exchanger transfers enthalpy, the sum of sensible and latent heat, rather than sensible heat alone. Sensible heat exchanged may be determined from measurement of flow rates and the difference between entering and leaving dry bulb temperatures, whereas, the enthalpy, or total heat, exchanged requires measurement of flow rates and the entering and leaving dry bulb temperatures and moisture contents. The differences of these psychrometric values allows the determination of the effectiveness of an enthalpy exchanger on both the sensible and latent heat basis.

1. General Considerations

a) Prior to any testing, the system should be checked as outlined above. The faces of the wheel must be clean and undamaged. The gap between the wheel and air seals should be as recommended by the manufacturer.

b) If the unit is equipped with an adjustable speed drive system and is at a reduced speed phase of operation, the controls should be reprogrammed to bring the exchanger wheel to full speed.

c) Stable inlet and exhaust conditions must be maintained throughout the test.

d) Psychrometric measurements of both the supply and exhaust air streams are to be taken at the entering and leaving sides of the exchanger.

e.) The volumetric flow rates of both air streams must be accurately determined.

f) Measurements should be taken at points sufficiently remote from the exchanger to preclude the possibility of temperature, moisture and flow stratification. If stratification is suspected, additional measurements should be made at alternate locations.

g) Duct work between the test sites and the unit must be airtight to prevent leakage from altering the composition of the measured stream.

h) There must be no air modification devices between the points where readings are taken and the exchanger to assure that the psychrometric conditions measured are identical to those at the unit.

2. Psychrometric Readings

Procedures for measurement of air temperature and moisture content have been described in the "Procedural Standards for Testing, Adjusting, Balancing of Environmental Systems" published by the National Environmental Balancing Bureau (NEBB). The basic provisions of good practice are outlined here, but established standards may be consulted for a more detailed description.

a. Dry and Wet Bulb Temperatures Above Freezing

1. Dry and wet bulb thermometers may be used to determine moisture content when the wet bulb temperature of the air stream is above the freezing point of water.
2. The thermometers used should be a laboratory grade partial immersion type with scale divisions not greater than 0.2°F. The thermometer length and range must be such that the indicated temperature can be read without disturbing the position of bulb.
3. Each dry and wet bulb thermometer pair must be matched such that, when both are dry, identical readings are observed at the same ambient condition.
4. In most operating situations, using a mercury in glass thermometer, velocity corrections need not be applied to indicated wet bulb temperatures when the air flow is between 700 and 2000 Fpm.
5. The wet bulb thermometer should be located 4 inches downstream from the dry bulb thermometer in line with air flow to minimize measurement induced errors.
6. There must be no stratification of temperature or moisture in the measured air stream. The point of measurement should be as remote as possible from any duct irregularities.
7. Thermometers must be sealed at the duct to prevent any air leakage which may alter the condition of the measured air.
8. Before each reading, the wet bulb thermometer wick should be wetted with distilled water whose temperature is at or slightly above the wet bulb temperature.
9. Dry and wet bulb thermometer readings should be taken simultaneously when the wet bulb temperature reaches its minimum value.
10. If possible, simultaneous psychrometric readings should be taken at the inlet and outlet to the dehumidifier of each air stream.

b. Wet Bulb Temperature Below Freezing

It is recommended that, at wet bulb temperatures below freezing, a thermometer having an ice coated bulb and without a wick be used to determine wet bulb temperature as water may be cooled below its freezing point without change of state and the vapor pressure characteristics of super cooled water differ from those of ice.

The ice film is best formed by dipping the chilled thermometer into distilled water at approximately 32°F. The thermometer is then removed from the water and the film allowed to freeze. The process may be repeated several times if necessary to build up a suitable film thickness. As a result of the reduced vapor pressure at low temperatures, a longer time is necessary to reach equilibrium than at higher temperatues. This condition is offset, however, by the ice remaining on the bulb for a much longer period of time. Readings of the wet-bulb thermometer must be continued over a sufficiently long period to insure that equilibrium has been reached.

c. Low Dew Point Measurements

1. A dew point cup is used to determine the moisture content of air whose dew point is between 0°F and minus 100°F.
2. The polished inner container of the dew point cup must be free of any surface contaminants which may act as insulation.
3. All tubing from the measured air source to the dew point cup must be vaportight.
4. The rate of flow shall be 5 cubic feet per hour. Adequate time shall be allowed to permit the sample to completely purge the cup, usually 5 minutes.
5. Fill the dew point cup with acetone sufficient to cover the thermometer to its rated immersion depth. The thermometer must be accurate to within 1°F.

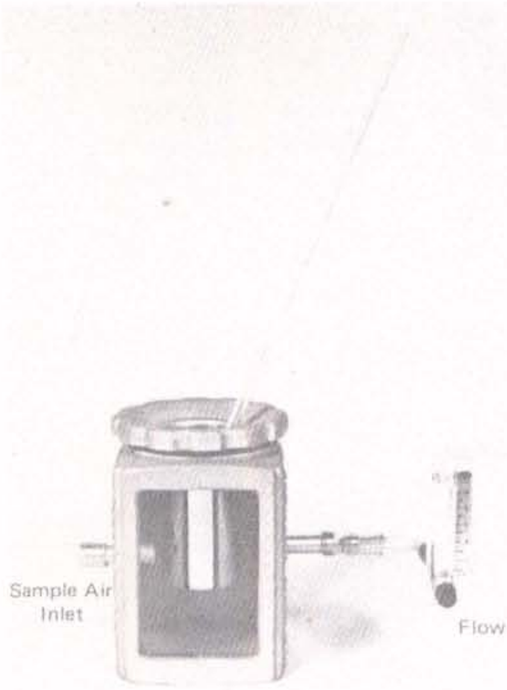


Figure 16-1 DEW POINT CUP (5)

6. Chill the cup slowly by adding small amounts of crushed dry ice while stirring the acetone vigorously with the thermometer. The rate of cooling shall be decreased for low dew points.
7. The polished surface of cup must be well lighted. The temperature at which the first sign of dew or mist appears is the dew point temperature.
8. A positive dewpoint determination is made when dew or frost can be made to disappear and reappear with slight fluctuations in the temperature of the cup.
9. There must be assurance that the source of the dew or frost induced on the dew cup is moisture. Other condensible vapors in an air stream may have dewpoints higher than that of moisture and their dew points must be differentiated from the moisture dewpoint.

3. Airflow Measurements

The static pressure loss across the exchanger wheel may be used to reliably measure the volumetric airflow in the supply and exhaust circuits. Likewise, the purge flow may be determined from the static pressure differential between the supply inlet and the exhaust outlet. The sample report form shown in Figure 16-2 can be used for entering the necessary data taken from field measurements.

An alternate method of airflow measurement should be undertaken if the wheel appears to be clogged with dirt or the surfaces are damaged. The supply flow must be measured downstream of the unit and the exhaust upstream of the unit. Determination of the purge flow requires an additional measurement of either the supply or exhaust flow at the opposite side of the unit of the previous measurement. The purge flow is the difference between the upstream and downstream flow rates, in standard units, of either circuit. The determination of air volume flow requires accurate and precise equipment, practiced technique and strict adherence to established procedures. Standard primary systems provide accuracy only when air flow is uniform, and within given velocity ranges. Only measurements determined within the framework of an established test code can be used with confidence. Values gained from secondary instruments when the characteristics of the airsteam are unknown, non-uniform, or inconsistent with the range of the instrument can be used only on a comparative basis.

The static pressure of each air circuit is to be determined at the entering and leaving sides of the recovery unit. The static pressure drop across the wheel is the difference in static pressures at the entering and leaving sides. This difference can be both measured and calculated. Some units are shipped with differential pressure gauges. These measured values should match the differences in individual absolute readings. Using the same methods, the differential between the supply air entering the unit and the exhaust air leaving must be determined.

4. Wheel Rotation

The wheel rotation is to be observed and timed. The rotational speed is recorded in units of revolutions per minute.

5. Transfer Ratio and Effectiveness

The Transfer Ratio is the ratio of the measured heat or moisture exchange of the supply circuit to the measured heat or moisture exchange of the exhaust circuit. Satisfactory measurement of the system's air properties may be assumed when this value is between 0.9 and 1.1.

If the results of this performance test are unsatisfactory, contact the device manufacturer for analysis of data and recommendations.

SMACNA

ENERGY RECOVERY EQUIPMENT TEST REPORT (AIR TO AIR)

PROJECT ______________________ SYSTEM/UNIT ______________________

LOCATION ______________________ MODE OF OPERATION ______________________

UNIT DATA	
Make/Model Number	
Type/Size	
Serial Number	
Make Sheave	
Sheave Diam./Bore	
No. Belts/make/size	
Media Type	
Face Area/Rows	

MOTOR DATA	
Make/Frame	
H.P./RPM	
Volts/Phase/Hertz	
F.L. Amps/S.F.	
Make Sheave	
Sheave Diam./Bore	
Sheave ℄ Distance	

TEST DATA SUPPLY SIDE	DESIGN	ACTUAL
CFM/FPM		
Outdoor Temp. DB/WB		
Inlet Air Temp. DB/WB		
Outlet Air Temp. DB/WB		
Inlet Air-Grains/lb.		
Outlet Air-Grains/lb.		
Ent./Lvg. Air Press.		
Air Pressure ΔP		
Air Temperature ΔT		
Effectiveness – %		
Transfer Ratio		

TEST DATA EXHAUST SIDE	DESIGN	ACTUAL
CFM/FPM		
Inlet Air Temp. DB/WB		
Outlet Air Temp. DB/WB		
Inlet Air-Grains/lb.		
Outlet Air-Grains/lb.		
Ent./Lvg. Air Press.		
Air Pressure ΔP		
Air Temperature ΔT		
Wheel RPM		
Coil Tilt Angle		
Purge CFM		
Damper Position		

REMARKS:

TEST DATE ______________ READINGS BY ______________________

FORM ER-1-78

SHEET METAL AND AIR CONDITIONING CONTRACTORS NATIONAL ASSOCIATION

Figure 16-2 SAMPLE REPORT FORM (16)

The effectiveness (sensible/sensible, enthalpy/enthalpy, or sensible/enthalpy) is then computed as outlined in earlier sections of this manual. The value obtained is entered on the report form with the rest of the data to be submitted, or used.

C MAINTENANCE

Any heat exchanger installed in an exhaust duct may be susceptible to fouling to some degree, depending upon the nature of the contaminants in the exhaust stream, and upon the dewpoint of the flowing gas. If allowed to continue, fouling will decrease the heat transfer performance. Also, air flow may be impeded which, in the case of paint ovens, could result in increasing solvent concentrations above safe and acceptable levels.

The following instructions are taken from a typical maintenance manual and provide general dismantling and disassembly procedures for cleaning modules of waste heat recuperators (fixed plate exchangers). The principles would also apply to many other types of energy recovery devices.

Installation will vary from one location to another, depending on its application. Although it is impractical to cover all installations, this sub-section will attempt to provide sufficient information so that most installations can be properly maintained. A typical Waste Heat Recuperator installation is shown below.

1. General Information

a. Monitoring Device

It is recommended that a differential static pressure gage or some other type of monitoring device connected to a light or alarm, be installed in the exhaust

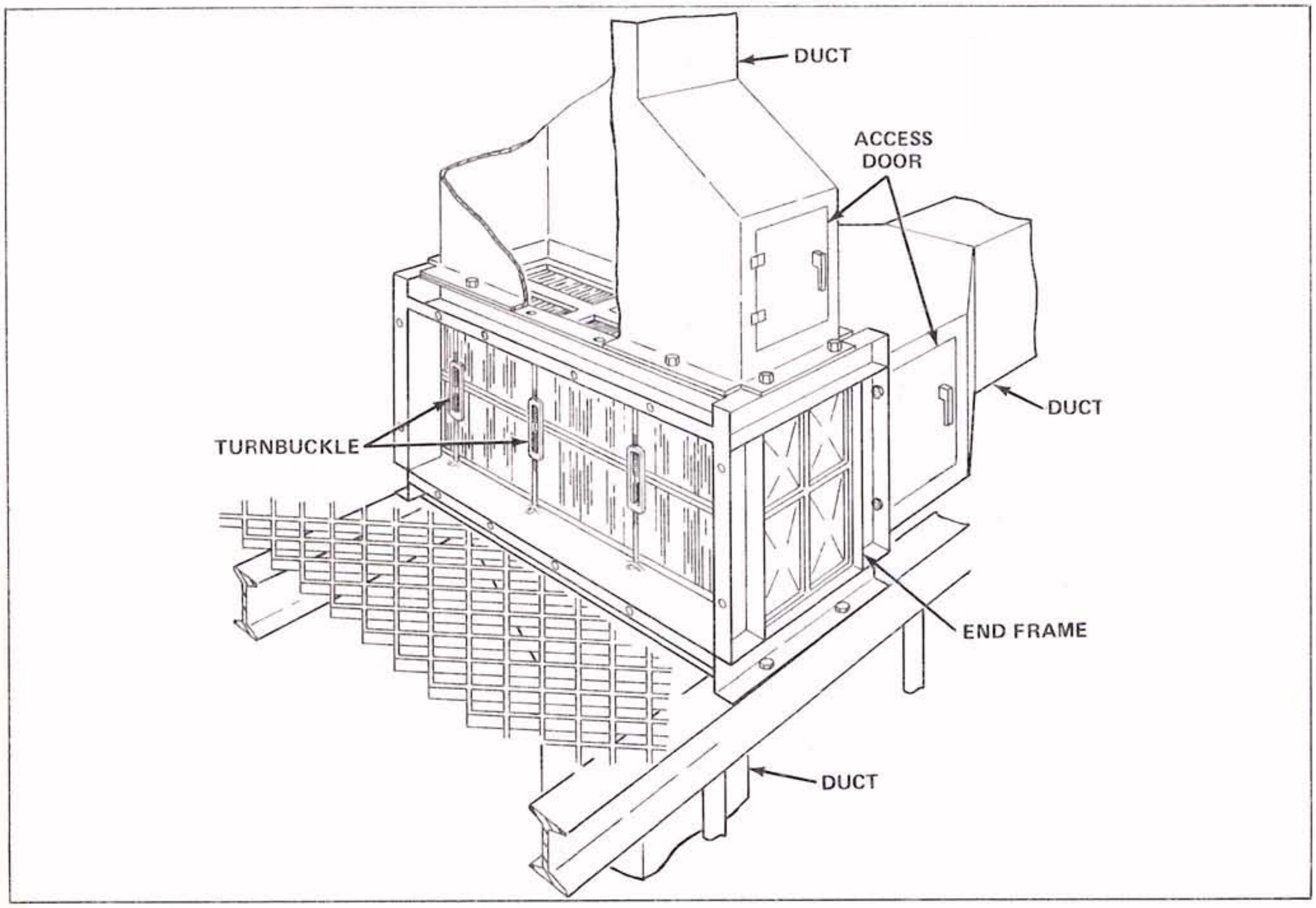


Figure 16-3 TYPICAL INSTALLATION, RECUPERATOR ASSEMBLY (FIXED PLATE TYPE EXCHANGER) (10)

air flow. This device should be calibrated so that when the pressure differential across the recuperator increases to 50% more than its design value, a warning will be given. For instance, if the recuperator operates with a 1.0'' w.g. pressure drop, a 1.5'' w.g. pressure drop should trigger the warning. A 50% increase in pressure drop occurs when approximately 20% of the air passages are plugged. At this point the modules should be inspected and cleaned. This is only a general rule; individual systems may not tolerate the loss of air flow resulting from a 50% increase in pressure drop. Other systems may operate below design flow rates and, thus, have lower pressure drops. Warning devices on these systems must be set at lower pressures to detect fouling.

b. Cleaning

Figure 16-4 shows a typical example of the types of fouling that may be encountered in process applications. The manner in which the recuperator may be cleaned, depends largely on the nature of the contaminant. Steam cleaning, detergent wash, solvent soak and wash, and high temperature baking, may all be effective cleaning procedures. The exact procedure and frequency is best determined by experience — Refer to the Process Chart in Table 16-2 for recommended cleaning procedures. Table 16-1 contains a list of *Safety Precautions* which *must be observed.*

c. Disassembly

The manufacturers' disassembly instructions provide complete step-by-step procedures for dismantling and removing modules. In many cases, removal of all modules is neither necessary nor recommended. As most contaminants generally accumulate on modules facing the process air (exhaust) inlet, if possible, it is recommended that the Recuperator be dismantled only to the extent necessary, starting with modules nearest the process air inlet.

2. Inspection

a. Visually inspect modules for fouling of air passages and structural damage. With the aid of a flashlight or other suitable means, sight through the air passages to determine the extent of contamination build-up.

b. Check seals for deterioration and leakage. Insure that swelling or hardening of seals is not evident; hardened or swollen seals may not provide adequate protection against cross-contamination.

c. Check for secure mounting of frame and attachment of ducts and sheet metal.

d. Check turnbuckles to insure they are drawn up tight. Upper frame should provide sufficient clamping pressure on modules to prevent them from shifting or moving without crushing seals or modules.

Table 16-1
Safety Precautions (10)

The following is a summary of the WARNINGS (personnel) and CAUTIONS (equipment) which must be observed when servicing Waste Heat Recuperators.

WARNINGS

- Organic solvents, including kerosene, varnoline, etc. have very low flash points. These solvents present dangerous fire hazards when used for cleaning.
- When handling chemical cleaners in dry form, a respirator must be used.
- In any of the cleaning procedures, due to the caustic nature of the cleaners, cleaning personnel must wear protective clothing and face gear. Always work in a well ventilated area.
- Due to the eye hazard from flying particles and dust when using compressed air for cleaning, all personnel in the near vicinity should wear eye protection and a respirator.
- Handle modules carefully when dipping in a tank for immersion cleaning to avoid burns caused by splashing hot caustic chemicals.
- The burn-off oven should be installed in a well ventilated area. Never open the furnace during burn-off. The entrance of a large volume of air could cause combustion. Handle modules with care to prevent burns.
- Due to the unpredictable or unknown characteristics of the contaminants, before using the burn-off procedure a careful analysis should be made to protect the operator and/or avoid damage to the module. Examples: liquid entrapment which could cause an explosion and lint, oil, etc. which could catch on fire.

CAUTIONS

- Insure that turnbuckles are not over-tightened
- High temperatures used in burning-off contaminants can result in remelting the brazing material. Do not exceed 1000°F.

Check for secure mounting of frame and attachment of ducts and sheet metal.

Check turnbuckles to insure they are drawn up tight. Upper frame should provide sufficient clamping pressure on modules to prevent them from shifting or moving without crushing seals or modules.

3. Repair

Repair of modules is limited to straightening bent fins. Fins should be straightened with needle nose pliers if they impede air flow through the air passages.

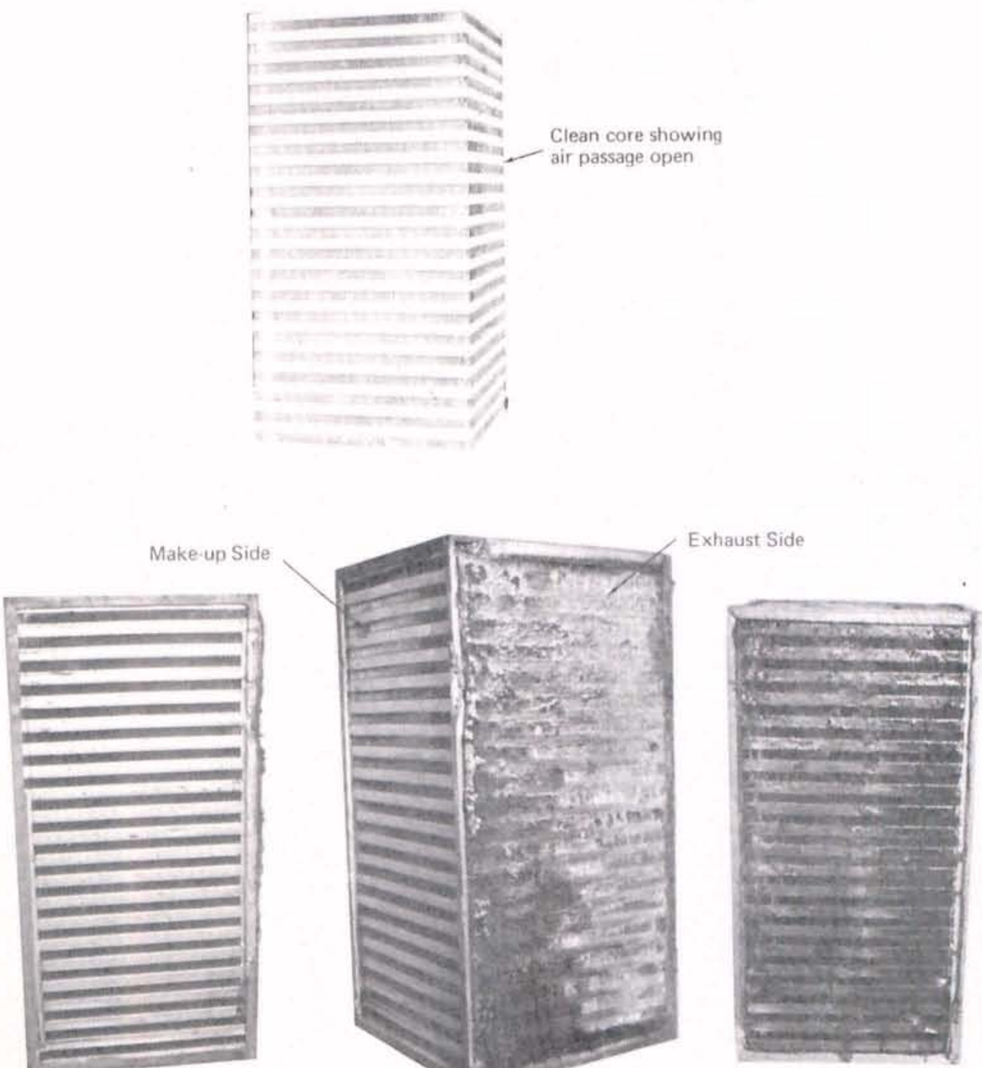
4. Replacement

a. Modules

Modules will function with approximately 20% of the air passages blocked. If structural damage is more severe than this, causing loss of performance and increase in pressure drop, the module should be replaced.

b. Seals

Seals should be replaced if there is any visual sign of breaks, cracks, deterioration, or other damage such



Make-up Side

Core from ELPO/ paint oven showing effects of condensation of solvents on the exhaust side, make-up side clean.

Exhaust Side

Figure 16-4 EXAMPLES OF FOULING (10)

as swelling, lengthening, etc. Seals should remain soft and flexible.

5. Reassembly

Reassembly is basically the reverse of disassembly. When installing the cleaned and/or repaired modules, the following guidelines should be followed:

a) Prior to installing seals, spray or coat with a silicone lubricant or, where silicone is unacceptable, a graphite lubricant. This will keep them from sticking when disassembling.

b) Seals must be installed on the correct face of the module as illustrated in Figure 16-5 to avoid air flow restriction.

c) Insure that modules are properly mated so that air flow is not impeded. The air channels of one module should mate with those of the other. (See Figure 16-6).

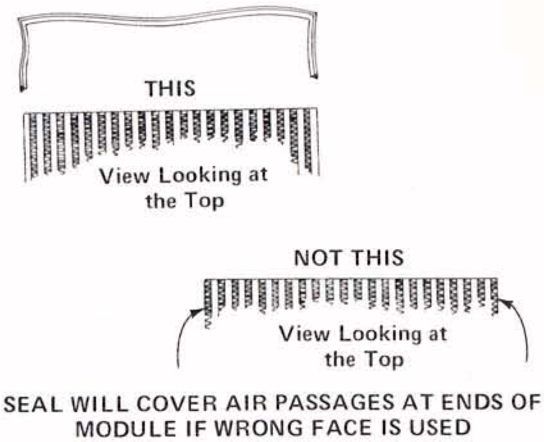


Figure 16-5 SEAL WILL COVER AIR PASSAGES AT ENDS OF MODULE IF WRONG FACE IS USED (10)

D CLEANING

When the air passages of the modules are blocked sufficiently to impede air flow (approximately a 50% pressure drop increase), the modules should be cleaned. Select the cleaning procedure best suited for the process. Seals should normally be removed during cleaning since chemical cleaners can attack them.

1. Cleaning Recommendations

The following are recommendations for cleaning recuperator modules. The selection of the cleaning method employed should be dependent on space, time, and equipment available and the degree and nature of contamination.

The selection of the best cleaner for a set of operating conditions is based on many factors. Sometimes one of these factors is so important that the other factors are secondary. Selection of the cleaner may be dictated by one of the following factors:

a) *Type of Contaminant* — most commercial alkaline cleaners are blends of a number of alkaline salts with one or more wetting agents, therefore, dependent on the blend, some cleaners will be more successful in removing one type of contaminant, while others will be successful in removing other types.

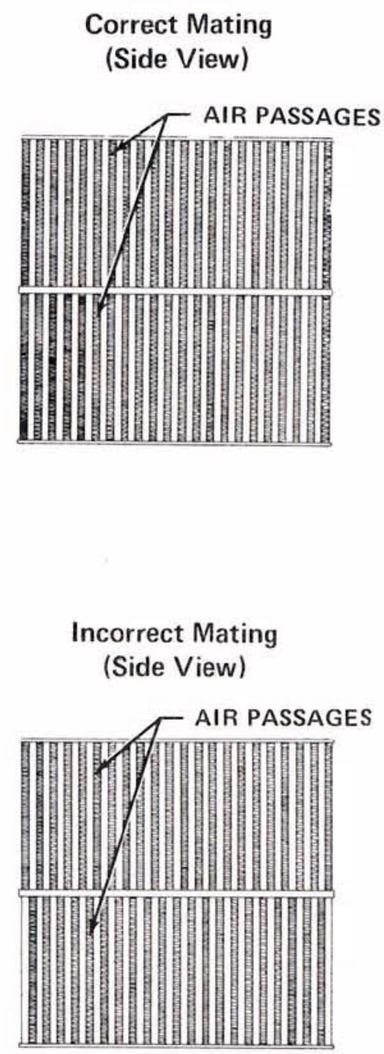


Figure 16-6 MODULE MATING (10)

b) *Attack on Aluminum* — the sensitivity of aluminum prohibits the use of some cleaners even though superior performance in soil removal could be obtained with them.

WARNING: ORGANIC SOLVENTS, INCLUDING KEROSENE, VARNOLINE, ETC. HAVE VERY LOW FLASH POINTS. THESE SOLVENTS PRESENT DANGEROUS FIRE HAZARDS WHEN USED FOR CLEANING.

NOTE: Refer to the Process Chart for recommendations of cleaners, etc.

In general, cleaning is accomplished by one of the following three methods, e.g. pressure cleaning (liquid, steam or air), immersion, or burn off.

These three cleaning methods described above are not mutually exclusive and can be used in combination. For example, a heavily contaminated recuperator can be partially immersed in an immersion tank to remove the heavy contaminants and then steam sprayed to remove the rest of the contamination. It must be stressed at this point that a final rinse with water should always be used after any of the cleaning methods. The reason for this is that a prolonged exposure of the metal, to alkaline cleaners, can cause etching. A compressed air blow-off is also recommended after this rinse to remove the loose contamination still remaining before it has a chance to dry.

2. Cleaning Procedures

A description of the different cleaning methods and cleaning procedures for the modules are described in the following paragraphs.

After the modules are removed from the frame assembly, the frame and attaching sheet metal should be checked for contaminants. These contaminants should be removed before reinstalling the modules. Removal of these contaminants can generally be accomplished by scrubbing, spray cleaning, or steam cleaning. Follow the usual precautions depending on method employed.

IN ANY OF THE FOLLOWING PROCEDURES, DUE TO THE CAUSTIC NATURE OF THE CLEANERS, CLEANING PERSONNEL MUST WEAR PROTECTIVE CLOTHING AND FACE GEAR. ALWAYS WORK IN A WELL VENTILATED AREA.

Caustic chemicals and flying particles represent personnel eye hazards. Consequently, it is recommended that eye wash facilities be available.

a. Pressure Cleaning

This method employs the use of a chemical spray, detergent mixed with steam, or compressed air, and is generally good for light to moderate contaminants. Use the procedure best suited for the particular process.

1. Spray Cleaning

Use a chemical spray, working the spray against the module(s). The high level of agitation caused by the spray will loosen the contaminants and remove them. Temperature of the mix should be between 125° and 180°F.

2. Steam Cleaning

Use a hot detergent solution mixed with steam and apply in a coarse high volume spray. Direct the spray back and forth over the module until the contaminants are flushed away. With this procedure, modules can be taken out of the frame assembly and steamed by an operator, on the floor, with the steam gun, or can be left in the housing unit and steamed through access doors. Other means of steaming could be done automatically with spray nozzles directly in the housing unit and turned on daily or weekly.

Steam cleaning may not always be successful in removing highly tenacious type contaminants or in cores contaminated to such a degree that the fins are totally plugged. A more costly immersion or burn-out method might have to be used. With a properly planned steam maintenance program these costly methods can be avoided. CLEAN FREQUENTLY TO AVOID HEAVY BUILD-UP OF CONTAMINANTS.

3. Compressed Air

Compressed air is generally used to remove flax, lint, dust, etc. from the air passages. Use compressed air to remove loose particles from the module passages.

DUE TO THE EYE HAZARD FROM FLYING PARTICLES AND DUST WHEN USING COMPRESSED AIR FOR CLEANING, ALL PERSONNEL IN THE NEAR VICINITY SHOULD WEAR EYE PROTECTION AND A RESPIRATOR.

b. Immersion Cleaning

This method employs the use of a tank which contains a strong chemical mix. The module(s) is immersed in the tank and left to soak for a period of time. This method is generally good for moderate to heavy contaminants.

1. Using a sling and basket or tray, carefully lower the recuperator in the tank.

HANDLE MODULES CAREFULLY WHEN DIPPING IN TANK TO AVOID BURNS CAUSED BY SPLASHING HOT CAUSTIC CHEMICALS.

2. Let the module(s) soak for a period of time. When it appears that the contaminats have been removed, remove the module from the tank, rinse with copious amounts of water, and blow dry with compressed air.

c. Burn-Off

This method uses a drying oven operating with a controlled atmosphere at very high temperatures (800°–1000°F) to burn and decompose the contaminants to an easily removable ash. This method is good for heavy contaminants, providing quick cleaning results.

THE BURN-OFF OVEN SHOULD BE INSTALLED IN A WELL VENTILATED AREA. NEVER OPEN THE FURNACE DURING BURN-OFF. THE ENTRANCE OF A LARGE VOLUME OF AIR COULD CAUSE COMBUSTION. HANDLE MODULES WITH CARE TO PREVENT BURNS.

DUE TO THE UNPREDICTABLE OR UNKNOWN CHARACTERISTICS OF THE CONTAMINANTS, BEFORE USING THE BURN-OFF PROCEDURE A CAREFUL ANALYSIS SHOULD BE MADE TO PROTECT THE OPERATOR AND/OR AVOID DAMAGE TO THE MODULE. EXAMPLES: LIQUID ENTRAPMENT WHICH COULD CAUSE AN EXPLOSION AND LINT, OIL, ETC. WHICH COULD CATCH ON FIRE.

High temperatures can result in remelting the brazing material. Do not exceed 1000°F.

1. Place the module in the oven (see figure).
2. Close and secure oven door.
3. Turn drying oven on.
4. After a period of 45-60 minutes, turn oven off, allow to cool, then open door and remove module.
5. If necessary, use compressed air to remove loose ash from the module air passages.

Table 16-2 PROCESSES CHART FOR CLEANING (10)

Problem	Procedure	Cleaning Agent
	WATER DRY CONDENSER OVENS	
Moderately Contaminated Cores	Immersion, 2 oz./gal., 150°F	Amchem Ridoline No. 357
	Immersion, pure solvent, 70°F	DuBois Aero-Carb**
	Steam, 4 lbs./50 gal.	DuBois Sprex A.C.
Lightly Contaminated Cores	Steam, 2 oz./gal.	Pennwalt MC-7
	Steam, 1 part Echelon/10 parts H_2O	Pennwalt Echelon
	LINT AND DUST REMOVAL	
Loose Particles	Clean thoroughly with compressed air at 100 psi	Compressed Air
	DRYING OVENS FOR WAX	
Heavily Contaminated Cores	1. Immersion, boil, one part arrest/6 parts H_2O 2. Immersion, boil, 6 oz./gal. arrest, immersion time twice that of step 1	Pennwalt Arrest and Strategy*
	Immersion, boil, 6 oz./gal.	DuBois Sprex A.C.
	Immersion, 160°F, 4 oz./gal.	Heatbath Unikleen 49—D
	Steam, 6 oz./gal.	Pennwalt Strategy*
Moderately Contaminated Cores	Immersion, 160°F, 2 oz./gal	Amchem Ridoline No. 57
	PAINT OVENS* **	
Heavily Contaminated Cores	Immersion, 6 oz./gal., rolling boil	DuBois Sprex A.C.
	Burn-off 800°F – 1000°F	Heat
Heavily to Moderately Contaminated Cores	Immersion, 6 oz./gal., 180°F	Pennwalt Strategy**
Moderately Contaminated Cores	Immersion, 6 oz./gal., rolling boil	DuBois Prime
	Immersion, 2 oz./gal., 140°F – 160°F	Amchem Ridoline No. 57
	Immersion, 2 oz./gal., 150°F	Amchem Ridoline No. 357
	Immersion, pure solvent, 70°F	DuBois Aero-Carb*
Moderately to Lightly Contaminated Cores	Steam, 4 lb./50 gal.	DuBois Sprex A.C.
Moderately to Lightly Contaminated Cores where heated tanks are not available.	Immersion, 6 oz./gal., room temperature	Oakite A1 Cleaner 166

*Contains phenols. (These chemicals may present a disposal problem and are toxic if used under improper conditions.)
**Contains cresols, chromates, methylene chloride. (These chemicals may present a disposal problem and are toxic if used under improper conditions.)
***For removal of lacquers, enamels, urethanes, and ELPO.

A HVAC EQUATIONS IN U.S. UNITS (BRITISH SYSTEM) (16)

1. AIR EQUATIONS

a) $V = 1096 \sqrt{\frac{V_p}{d}}$

or for standard air:

$V = 4005 \sqrt{V_p}$

$\left(d = 1.325 \frac{P_b}{T}\right)$

V = Velocity (fpm)

V_p = Velocity pressure (in w.g.)

d = Density (lb/cu ft)

P_b = Absolute static pressure (in. Hg) (Barometric pressure + static pressure)

T = Absolute Temp. (460° + °F)

b) $Q = 60 \times C_p \times d \times cfm \times \Delta t$

or for standard air:

Q (sens.) $= 1.08 \times cfm \times \Delta t$

c) Q (Lat.) $= 4840 \times cfm \times \Delta W$

d) Q (Total) $= 4.5 \times cfm \times \Delta h$

e) $Q = A \times U \times \Delta t$

f) $R = \frac{1}{U}$

Q = Heat Flow (btu/hr)

C_p = Specific Heat (Btu/lb/°F)

Δt = Temperature Difference (°F)

ΔW = Humidity Ratio (lb H_2O/lb dry air)

Δh = Enthalpy Diff. (Btu/lb dry air)

A = Area or Surface (sq ft)

U = Heat transfer coefficient

R = Thermal resistance

g) $\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = R$

P = Absolute pressure (lb/sq ft)

V = Total Volume (cu ft)

R = Gas constant

T = Absolute temp. (460° + °F)

h) $TP = V_p + SP$

i) $V_p = \left(\frac{V}{4005}\right)^2$

j) $V = V_m \sqrt{\frac{0.075}{d}}$ (other than standard)

k) $cfm = A \times fpm$

TP = Total pressure (in w.g.)

V_p = Velocity pressure (in w.g.)

SP = Static pressure (in w.g.)

V = Velocity (fpm)

V_m = Measured velocity (fpm)

d = Density (lb/cu ft)

A = Area or Surface (sq ft)

B HVAC EQUATIONS IN METRIC UNITS (16)

1. AIR EQUATIONS

a) $V = 1.288\sqrt{V_p\left(\frac{1.2}{d}\right)}$

or for standard air:

($d = 1.2$ kg/m^3)

$V = 1.3\sqrt{V_p}$

V = Velocity (m/s)

V_p = Velocity Pressure (pascals)

d = Density (kg/m^3)

b) $Q = 3.6 \times C_p \times d \times l/s \times \Delta t$

or for standard air:

($c_p = 0.28$ W/kg/°C)

Q (sens.) $= 1.21 \times l/s \times \Delta t$

c) Q (Lat.) $= 3.0 \times l/s \times \Delta W$

d) Q (Total Heat) $= 1.27 \times l/s \times \Delta h$

e) $Q = A \times U \times \Delta t$

f) $R = \frac{1}{U}$

Q = Heat Flow (Btu/hr)

C_p = Specific Heat (W/kg/°C)

Δt = Temperature Difference (°C)

ΔW = Humidity Ratio (g H_2O/kg dry air)

Δh = Enthalpy Diff. (kJ/kg dry air)

A = Area of Surface (m^2)

U = Heat transfer coefficient

R = Thermal Resistance

g) $\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = R$

P = Absolute Pressure (pascals)

V = Total Volume (m^3)

R = Gas Constant

T = Absolute Temperature (°K)

h) $TP = V_p + SP$

i) $V_p = \left(\frac{V}{1.3}\right)^2 = \frac{d}{2} \times V^2$

j) $V = 1.4\sqrt{\frac{V_p}{d}}$

k) m^3/s = A x m/s

TP = Total Pressure (pascals)

V_p = Velocity Pressure (Pa)

SP = Static Pressure (Pa)

= Velocity (m/s)

d = Density (kg/m^3)

A = Area (m^2)

U.S. UNITS

2. FAN EQUATIONS

a) $\frac{cfm_2}{cfm_1} = \frac{rpm_2}{rpm_1}$

b) $\frac{SP_2}{SP_1} = \left(\frac{rpm_2}{rpm_1}\right)^2$

c) $\frac{bhp_2}{bhp_1} = \left(\frac{rpm_2}{rpm_1}\right)^3$

d) $\frac{\text{rpm (fan)}}{\text{rpm (motor)}} = \frac{\text{Pitch diam. motor pulley}}{\text{Pitch diam. fan pulley}}$

cfm = cu ft/min

rpm = revolutions/min

SP = Static pressure (in w.g.)

bhp = brake horsepower

3. HYDRONIC EQUATIONS

a) $Q = 500 \times gpm \times \Delta t$

b) $\frac{\Delta P_2}{\Delta P_1} = \left(\frac{gpm_2}{gpm_1}\right)^2$

c) $\Delta P = \left(\frac{gpm}{C_v}\right)^2$

d) $whp = \frac{gpm \times H \times Sp.\ Gr.}{3960}$

e) $bph = \frac{gpm \times H \times Sp.\ Gr.}{3960 \times E_p}$

f) $NPSHA = P_a \pm P_s + \frac{V^2}{2_g} - P_{vp}$

g) $h = \frac{fLv^2}{2gD}$

gpm = Gallons per minute

Q = Heat Flow (Btu/hr)

Δt = Temperature diff. (°F)

ΔP = Pressure diff. (psi)

C_v = Valve constant

whp = Water horsepower

bhp = Brake horsepower

H = Head (ft w.g.)

Sp. Gr. = Specific gravity (use 1.0 for water)

E_p = Efficiency of pump (%/100)

NPSHA = Net positive suction head available

P_a = Atm. press. (use 34 ft w.g.)

P_s = Pressure at pump centerline (ft w.g.)

V = Velocity head at point P_s (fpm)

P_{vp} = Absolute vapor pressure (ft w.g.)

g = Gravity acceleration (32.2 ft/sec^2)

h = Head loss (ft)

f = Friction factor (Moody)

L = Length of pipe (ft)

D = Internal diameter (ft)

v = Velocity (ft/sec)

METRIC UNITS

2. FAN EQUATIONS

a) $\frac{m^3/s_2}{m^3/s_1} = \frac{rad/s_2}{rad/s_1}$

b) $\frac{SP_2}{SP_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^2$

c) $\frac{kW_2}{kW_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^3$

d) $\frac{\text{rad/s (fan)}}{\text{rad/s (motor)}} = \frac{\text{Pitch diam. (mm) motor pulley}}{\text{Pitch diam. (mm) fan pulley}}$

m^3/s = cubic metres/second
rad/s = radians/second

SP = Static pressure (Pa)

kW = kilowatts

3. HYDRONIC EQUATIONS

a) $Q = 4190 \times m^3/s \times \Delta t$

b) $\frac{\Delta P_2}{\Delta P_1} = \left(\frac{m^3{}_2}{m^3{}_1}\right)^2$

c) $\Delta P = \left(\frac{m^3}{C_v}\right)^2$

d) $WP = 0.1020 \times m^3/s \times H \times d$

e) $BP = \frac{WP}{E_p}$

f) $NPSHA = P_a \pm P_s + \frac{V^2}{2g} - P_{vp}$

g) $h = \frac{fLv^2}{2gD}$

Q = Heat Flow (kilowatts)

Δt = Temperature Difference (°C)

m^3 (used for large volumes)

ΔP = Pressure Diff. (Pa or kPa)

C_v = Valve constant

WP = Water power (W or kW)

H = Head (k P_a)

d = Density (kg/m^3)

BP = Brake power (W or kW)

E_p = Efficiency of Pump (%/100)

NPSHA = Net positive suction head available

P_a = Atm. press. (pascals)
(Std. Atm. press. = 101 325Pa)

P_s = Pressure at pump centerline (Pa)

$\frac{V^2}{2g}$ = Velocity pressure at point P_s (Pa)

P_{vp} = Absolute vapor pressure (Pa)

g = Gravity acceleration (9.807 m/s^2)

h = Head loss (m)

f = Friction factor (Moody, metric)

L = Length of pipe (m)

D = Internal diameter (m)

v = Velocity (m/s)

U.S. UNITS

4. ELECTRIC EQUATIONS

a) $\text{Bhp} = \dfrac{\text{I x E x P.F. x Eff.}}{746}$ (Single Phase)

b) $\text{Bhp} = \dfrac{\text{I x E x P.F. x Eff. x 1.73}}{746}$ (Three Phase)

c) $E = IR$

d) $P = EI$

e) $\dfrac{\text{F.L. Amps* x Voltage*}}{\text{Actual Voltage}} = \text{Actual F.L. Amps}$

*Nameplate ratings

I = Amps (A)

E = Volts (V)

P.F. = Power factor

R = ohms (Ω)

P. = watts (W)

Table 17-1 AIR DENSITY CORRECTION FACTORS (U.S. Units) (16)

Altitude (ft)	–	Sea Level	1000	2000	3000	4000	5000	6000	7000	8000	9000	10,000
Barometer	in Hg –	29.92	28.86	27.82	26.82	25.84	24.90	23.98	23.09	22.22	21.39	20.58
	in w.g. –	407.5	392.8	378.6	365.0	351.7	338.9	326.4	314.3	302.1	291.1	280.1
Air Temp.	−40°	1.26	1.22	1.17	1.13	1.09	1.05	1.01	0.97	0.93	0.90	0.87
°F	0°	1.15	1.11	1.07	1.03	0.99	0.95	0.91	0.89	0.85	0.82	0.79
	40°	1.06	1.02	0.99	0.95	0.92	0.88	0.85	0.82	0.79	0.76	0.73
	70°	1.00	0.96	0.93	0.89	0.86	0.83	0.80	0.77	0.74	0.71	0.69
	100°	0.95	0.92	0.88	0.85	0.81	0.78	0.75	0.73	0.70	0.68	0.65
	150°	0.87	0.84	0.81	0.78	0.75	0.72	0.69	0.67	0.65	0.62	0.60
	200°	0.80	0.77	0.74	0.71	0.69	0.66	0.64	0.62	0.60	0.57	0.55
	250°	0.75	0.72	0.70	0.67	0.64	0.62	0.60	0.58	0.56	0.58	0.51
	300°	0.70	0.67	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.50	0.48
	350°	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.51	0.49	0.47	0.45
	400°	0.62	0.60	0.57	0.55	0.53	0.51	0.49	0.48	0.46	0.44	0.42
	450°	0.58	0.56	0.54	0.52	0.50	0.48	0.46	0.45	0.43	0.42	0.40
	500°	0.55	0.53	0.51	0.49	0.47	0.45	0.44	0.43	0.41	0.39	0.38
	550°	0.53	0.51	0.49	0.47	0.45	0.44	0.42	0.41	0.39	0.38	0.36
	600°	0.50	0.48	0.46	0.45	0.43	0.41	0.40	0.39	0.37	0.35	0.34
	700°	0.46	0.44	0.43	0.41	0.39	0.38	0.37	0.35	0.34	0.33	0.32
	800°	0.42	0.40	0.39	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29
	900°	0.39	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29	0.28	0.27
	1000°	0.36	0.35	0.33	0.32	0.31	0.30	0.29	0.28	0.27	0.26	0.25

Standard Air Density, Sea Level, 70°F = 0.075 lb/cu ft

METRIC UNITS

4. ELECTRIC EQUATIONS

a) $\text{Bhp} = \dfrac{I \times E \times P.F. \times Eff.}{746}$ (Single Phase)

b) $\text{Bhp} = \dfrac{I \times E \times P.F. \times Eff. \times 1.73}{746}$ (Three Phase)

c) $E = IR$

d) $P = EI$

e) $\dfrac{\text{F.L. Amps}^{*} \times \text{Voltage}^{*}}{\text{Actual Voltage}} = \text{Actual F.L. Amps}$

*Nameplate ratings

I = Amps (A)
E = Volts (V)
P.F. = Power factor
R = ohms (Ω)
P. = watts (W)

Table 17-2 AIR DENSITY CORRECTION FACTORS (Metric Units) (16)

Altitude (m)	Sea Level	250	500	750	1000	1250	1500	1750	2000	2500	3000
Barometer (kPa)	101.3	98.3	96.3	93.2	90.2	88.2	85.1	83.1	80.0	76.0	71.9
Air Temp. °C 0°	1.08	1.05	1.02	0.99	0.96	0.93	0.91	0.88	0.86	0.81	0.76
21°	1.00	0.97	0.95	0.92	0.89	0.87	0.84	0.82	0.79	0.75	0.71
50°	0.91	0.89	0.86	0.84	0.81	0.79	0.77	0.75	0.72	0.68	0.64
75°	0.85	0.82	0.80	0.78	0.75	0.73	0.71	0.69	0.67	0.63	0.60
100°	0.79	0.77	0.75	0.72	0.70	0.68	0.66	0.65	0.63	0.59	0.56
125°	0.74	0.72	0.70	0.68	0.66	0.64	0.62	0.60	0.59	0.55	0.52
150°	0.70	0.68	0.66	0.64	0.62	0.60	0.59	0.57	0.55	0.52	0.49
175°	0.66	0.64	0.62	0.62	0.59	0.57	0.55	0.54	0.52	0.49	0.46
200°	0.62	0.61	0.59	0.57	0.56	0.54	0.52	0.51	0.49	0.47	0.44
225°	0.59	0.58	0.56	0.54	0.53	0.51	0.50	0.48	0.47	0.44	0.42
250°	0.56	0.55	0.53	0.52	0.50	0.49	0.47	0.46	0.45	0.42	0.40
275°	0.54	0.52	0.51	0.49	0.48	0.47	0.45	0.44	0.43	0.40	0.38
300°	0.51	0.50	0.49	0.47	0.46	0.45	0.43	0.42	0.41	0.38	0.36
325°	0.49	0.48	0.47	0.45	0.44	0.43	0.41	0.40	0.39	0.37	0.35
350°	0.47	0.46	0.45	0.43	0.42	0.41	0.40	0.39	0.38	0.35	0.33
375°	0.46	0.44	0.43	0.42	0.41	0.39	0.38	0.37	0.36	0.34	0.32
400°	0.44	0.43	0.41	0.40	0.39	0.38	0.37	0.36	0.35	0.33	0.31
425°	0.42	0.41	0.40	0.39	0.38	0.37	0.35	0.34	0.33	0.32	0.30
450°	0.41	0.40	0.38	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.29
475°	0.39	0.38	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.29	0.28
500°	0.38	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.30	0.28	0.27
525°	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.30	0.29	0.27	0.26

Dry air weight at Sea Level, 21°C = 1.205 kg/m³

C METRIC UNITS AND EQUIVALENTS (16)

Table 17-3 METRIC UNITS (Basic & Derived)

Unit	Symbol	Quantity	Equivalent or Relationship
ampere	A	Electric current	Same as U.S.
candela	cd	Luminous intensity	1 cd/m^2 = 0.292 ft lamberts
Celsius	°C	Temperature	°F = 1.8 °C + 32°
coulomb	C	Electric charge	Same as U.S.
farad	F	Electric capacitance	Same as U.S.
henry	H	Electric inductance	Same as U.S.
hertz	Hz	Frequency	Same as cycles/second
joule	J	Energy, work, heat	1 J = 0.7376 ft-lb = 0.000948 Btu
kelvin	K	Thermodynamic temperature	°K = °C + 273.15° $= \frac{°F + 459.67}{1.8}$
kilogram	kg	Mass	1 kg = 2.2046 lb
litre	l	Liquid volume	1 l = 1.056qt = 0.264 gal
lumens	lm	Luminous flux	1 lm/m^2 = 0.0929 ft candles
lux	lx	Illuminance	1 lx = 0.0929 ft candles
metre	m	Length	1 m = 3.281 ft
mole	mol	Amount of substance	—
newton	N	Force	1 N = kg · m/s^2 = 0.2248 lb (force)
ohm	Ω	Electrical resistance	Same as U.S.
pascal	Pa	Pressure, stress	1 Pa = N/m^2 = 0.000145 psi
radian	rad	Plane angle	1 rad = 57.29°
second	s	Time	Same as U.S.
seimens	S	Electric conductance	—
steradian	sr	Solid angle	—
volt	V	Electric potential	Same as U.S.
watt	W	Power, heat flow	1 W = J/s (metric) = 3.4122 Btu/hr (or 1 W = 0.000284 tons of refrig.)

Table 17-4 METRIC EQUIVALENTS

Quantity	Symbol	Unit	U. S. Relationship
acceleration	m/s^2	metres/second squared	$1\ m/s^2 = 3.281\ ft/sec^2$
angular velocity	rad/s	radians/second	1 rad/sec = 9.549 rpm
area	m^2	square metre	$1\ m^2 = 10.76$ sq ft
atmospheric pressure	—	101.325 kPa	29.92 in Hg = 14.696 psi
density	kg/m^3	kilograms/cubic metre	$1\ kg/m^3 = 0.0624$ lb/cu ft
density, air	—	$1.2\ kg/m^3$	0.075 lb/cu ft
density, water	—	$1000\ kg/m^3$	62.4 lb/cu ft
enthalpy	kJ/kg	kilojoule/kilogram	1 kJ/kg = 0.4299 Btu/lb dry air
gravity		$9.8067\ m/s^2$	$32.2\ ft/sec^2$
heat flow	W	watt	1 W = 3.412 Btu/hr
length (normal)	m	metre	1 m = 3.281 ft
length (large)	km	kilometre	1 km = 0.6214 miles
linear velocity	m/s	metres/second	1 m/s = 196.9 fpm = 2.237 mph
mass flow rate	kg/s	kilograms/second	1 kg/s = 7936.6 lb/hr
moment of inertia	$kg \cdot m^2$	kilograms x square metre	$1\ kg. \cdot m^2 = 23.73\ lb \cdot$ sq ft
power	W	watt	1 W = 0.00134 hp'
pressure	kPa	kilopascal	1 kPa = 0.296 in Hg = 4.022 in w.g.
specific heat-air (C_p)		$1000\ J/kg \cdot K$	$1000 J/kg \cdot K = kJ/kg \cdot K$ $= 0.2388\ Btu/lb°F$
specific heat-air (C_v)		$712\ J/kg \cdot K$	0.17 Btu/lb° F
specific heat-water		$4190\ J/kg \cdot K$	1.0 Btu/lb° F
specific volume	m^3/kg	cubic metres/kilogram	$1\ m^3/kg = 16.019$ cu ft/lb
thermal conductivity	w/m . °K	watts/metre x °K	1 w/m · °K = 6.933 Btu · in/hr · cu ft · °F
volume flow rate	m^3/s l/s	cubic metres/second litres/second	$1\ m^3/s$ = 2118.88 cfm (air). = 15,850 gpm (water) $1\ m^3/s$ = 1000 l/s

SECTION XVIII
GLOSSARY

A TECHNICAL TERMS

Absorbent: A material which, due to an affinity for certain substances, extracts one or more such substances from a liquid or gaseous medium with which it contacts and which changes physically or chemically, or both, during the process. Calcium chloride is an example of a solid absorbent, while solutions of lithium chloride, lithium bromide, and ethylene glycols are liquid absorbents.

Absorption: A process whereby a material extracts one or more substances present in an atmosphere or mixture of gases or liquids accompanied by the material's physical and/or chemical changes.

Acceleration: The time rate of change of velocity; i.e., the derivative of velocity with respect to time.

Acceleration Due to Gravity: The rate of increase in velocity of a body falling freely in a vaccum. Its value varies with latitude and elevation. The International Standard is 32.174 ft. per second per second.

Adiabatic Process: A thermodynamic process during which no heat is added to, or taken from, a substance or system.

Adsorbent: A material which has the ability to cause molecules of gases, liquids, or solids to adhere to its internal surfaces without changing the adsorbent physically or chemically. Certain solid materials, such as silica gel and activated alumina, have this property.

Adsorption: The action, associated with the surface adherence, of a material in extracting one or more substances present in an atmosphere or mixture of gases and liquids, unaccompanied by physical or chemical change. Commercial adsorbent materials have enormous internal surfaces.

Air, Ambient: Generally speaking, the air surrounding an object.

Air, Dry: Air without contained water vapor; air only.

Air, Outdoor: Air taken from outdoors and, therefore, not previously circulated through the system.

Air, Outside: External air; atmosphere exterior to refrigerated or conditioned space; ambient (surrounding) air.

Air, Recirculated: Return air passed through the conditioner before being again supplied to the conditioned space.

Air, Reheating of: In an air-conditioning system, the final step in treatment, in the event the temperature is too low.

Air, Return: Air returned from conditioned or refrigerated space.

Air, Saturated: Moist air in which the partial pressure of the water vapor is equal to the vapor pressure of water at the existing temperature. This occurs when dry air and saturated water vapor coexist at the same dry-bulb temperature.

Air, Standard: Dry air at a pressure of 760 mm (29.92 in.) Hg at 21°C (69.8°F) temperature and with a specific volume of 0.833 m^3/kg (13.33 ft^3/lb).

Air Changes: A method of expressing the amount of air leakage into or out of a building or room in terms of the number of building volumes or room volumes exchanged.

Air Conditioning, Comfort: The process of treating air so as to control simultaneously its temperature, humidity, cleanliness and distribution to meet the comfort requirements of the occupants of the conditioned space.

Air-conditioning Unit: An assembly of equipment for the treatment of air so as to control, simultaneously, its temperature, humidity, cleanliness and distribution to meet the requirements of a conditioned space.

Air Cooler: A factory-encased assembly of elements whereby the temperature of air passing through the device is reduced.

Air Diffuser: A circular, square, or rectangular air distribution outlet, generally located in the ceiling and comprised of deflecting members discharging supply air in various directions and planes, and arranged to promote mixing of primary air with secondary room air.

Air Washer: A water spray system or device for cleaning, humidifying, or dehumidifying the air.

Algae: A minute fresh water plant growth which forms a scum on the surfaces of recirculated water apparatus, interfering with fluid flow and heat transfer.

Anemometer: An instrument for measuring the velocity of a fluid.

Approach: In an evaporative cooling device, the difference between the average temperature of the circulating water leaving the device and the average wet-bulb temperature of the entering air. In a conduction heat exchanger device, the temperature difference between the leaving treated fluid and the entering working fluid.

Aspect Ratio: In air distribution outlets, the ratio of the length of the core opening of a grille, face, or register to the width. In rectangular ducts, the ratio of the width to the depth.

Aspiration: Production of movement in a fluid by suction created by fluid velocity.

Attenuation: The sound reduction process in which sound energy is absorbed or diminished in intensity as the result of energy conversion from sound to motion or heat.

Barometer: Instrument for measuring atmospheric pressure.

Boiling point: The temperature at which the vapor pressure of a liquid equals the absolute external pressure at the liquid-vapor interface.

British Thermal Unit (Btu): The Btu is defined as the heat required to raise the temperature of a pound of water from 59° F to 60° F.

Bypass: A pipe or duct, usually controlled by valve or damper, for conveying a fluid around an element of a system.

Calibration: Process of dividing and numbering the scale of an instrument; also of correcting or determining the error of an existing scale, or of evaluating one quantity in terms of readings of another.

Capillarity: The action by which the surface of a liquid, where it contacts a solid (as in a slender tube), is raised or lowered.

Celsius (Formerly Centigrade): A thermometric scale in which the freezing point of water is called 0 deg. and its boiling point 100 deg. at normal atmospheric pressure (14.696 psi).

Change of State: Change from one phase, such as solid, liquid or gas, to another.

Coefficient of Discharge: For an air diffuser, the ratio of net area or effective area at vena contracta or an orificed airstream to the free area of the opening.

Coefficient of Expansion: The change in length per unit length or the change in volume per unit volume, per deg. change in temperature.

Coefficient of Performance, (COP), Heat Pump: The ratio of the compressor heating effect (heat pump) to the rate of energy input to the shaft of the compressor, in consistent units, in a complete heat pump, under designated operating conditions.

Coil: A cooling or heating element made of pipe or tubing.

Comfort Chart: A chart showing effective temperatures with dry-bulb temperatures and humidities (and sometimes air motion) by which the effects of various air conditions on human comfort may be compared.

Comfort Cooling: Refrigeration for comfort as opposed to refrigeration for storage or manufacture.

Comfort Zone: (Average) the range of effective temperatures over which the majority (50 percent or more) of adults feels comfortable; (extreme) the range of effective temperatures over which one or more adults feels comfortable.

Condensate: The liquid formed by condensation of a vapor. In steam heating, water condensed from steam; in air conditioning, water extracted from air, as by condensation on the cooling coil of a refrigeration machine.

Condensation: Process of changing a vapor into liquid by extracting heat. Condensation of steam or water vapor is effected in either steam condensers or dehumidifying coils, and the resulting water is called condensate.

Condenser (refrigerant): A heat exchanger in which the refrigerant, compressed to a suitable pressure, is condensed by rejection of heat to an appropriate external cooling medium.

Conditions, Standard: A set of physical, chemical, or other parameters of a substance or system which defines an accepted reference state or forms a basis for comparison.

Conductance, Surface film: Time rate of heat flow per unit area under steady conditions between a surface and a fluid for unit temperature difference between the surface and fluid.

Conductance, Thermal: Time rate of heat flow through a body (frequently per unit area) from one of its bounding surfaces to the other for a unit temperature difference between the two surfaces, under steady conditions.

Conductivity, Thermal: The time rate of heat flow through unit area and unit thickness of a homogeneous material under steady conditions when a unit temperature gradient is maintained in the direction perpendicular to area. Materials are considered homogeneous when the value of the thermal conductivity is not affected by variation in thickness or in size of sample within the range normally used in construction.

Conductor, Thermal: A material which readily transmits heat by means of conduction.

Connection in Parallel: System whereby flow is divided among two or more channels from a common starting point or header.

Connection in Series: System whereby flow through two or more channels is in a single path entering each succeeding channel only after leaving the first or previous channel.

Control: A device for regulation of a system or component in normal operation, manual or automatic. In automatic, the implication is that it is responsive to changes of pressure, temperature or other property whose magnitude is to be regulated.

Convection: Transfer of heat by movement of fluid.

Convection, Forced: Convection resulting from forced circulation of a fluid, as by a fan, jet or pump.

Convection, Natural: Circulation of gas or liquid (usually air or water) due to differences in density resulting from temperature changes.

Cooling, Evaporative: Involves the adiabatic exchange of heat between air and a water spray or wetted surface. The water assumes the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger.

Cooling, Regenerative: Process of utilizing heat which must be rejected or absorbed in one part of the cycle to function usefully in another part of the cycle by heat transfer.

Cooling Coil: An arrangement of pipe or tubing which transfers heat from air to a refrigerant or brine.

Cooling Effect, Sensible: Difference between the total cooling effect and the dehumidifying effect, usually in watts (Btuh).

Cooling Effect, Total: Difference between the total enthalpy of the dry air and water vapor mixture entering the cooler per hour and the total enthalpy of the dry air and water vapor mixture leaving the cooler per hour, expressed in watts (Btuh).

Cooling Range: In a water cooling device, the difference between the average temperatures of the water entering and leaving the device.

Core Area (Face Area): The total plane area of the portion of a grille, register, or coil bounded by a line tangent to the outer edges of the openings through which air can pass.

Corresponding Values: Simultaneous values of various properties of a fluid, such as pressure, volume, temperature, etc., for a given condition of fluid.

Corrosive: Having chemically destructive effect on metals (occasionally on other materials).

Counterflow: In heat exchange between two fluids, opposite direction of flow, coldest portion of one meeting coldest portion of the other.

Critical Velocity: The velocity above which fluid flow is turbulent.

Crystal Formation, Zone of Maximum: Temperature range in freezing in which most freezing takes place, i.e., about $-3.9°$ to $-1.1°C$ (25° to 30°F) for water.

Cycle: A complete course of operation of working fluid back to a starting point, measured in a thermodynamic terms (functions). Also in general for any repeated process on any system.

Cycle, Reversible: Theoretical thermodynamic cycle, composed of a series of reversible processes, which can be completely reversed.

Dalton's Law of Partial Pressure: Each constituent of a mixture of gases behaves thermodynamically as if it alone occupied the space. The sum of the individual pressures of the constituents equals the total pressure of the mixture.

Damper: A device used to vary the volume of air passing through an air outlet, inlet, or duct.

Degree Day: A unit, based upon temperature difference and time, used in estimating fuel consumption and specifying nominal heating load of a building in winter. For any one day, when the mean temperature is less than 65° F, there exist as many degree days as there are Fahrenheit degrees difference in temperature between the mean temperature for the day and 65° F.

Dehumidification: The condensation of water vapor from air by cooling below the dewpoint or removal of water vapor from air by chemical or physical methods.

Dehumidifier: (1) An air cooler or washer used for lowering the moisture content of the air passing through it; (2) An absorption or adsorption device for removing moisture from air.

Dehydration: (1) removal of water vapor from air by the use of absorbing or adsorbing materials; (2) removal of water from stored goods.

Density: The ratio of the mass of a specimen of a substance to the volume of the specimen. The mass of a unit volume of a substance. When weight can be used without confusion, as synonymous with mass, density is the weight per unit volume. The reciprocal of specific volume.

Desiccant: Any absorbent or adsorbent, liquid or solid, that will remove water or water vapor from a material. In a refrigeration circuit, the desiccant should be insoluble in the refrigerant.

Design Working Pressure: The maximum allowable working pressure for which a specific part of a system is designed.

Dewpoint, Apparatus: That temperature which would result if the psychrometric process occurring in a dehumidifier, humidifier or surface-cooler were carried to the saturation condition of the leaving air while maintaining the same ratio of sensible to total heat load in the process.

Dew Point Depression: The different between dry bulb and dew point temperatures (°F DB — °F DP).

Differential: Of a control, the difference between cut-in and cut-out temperatures or pressures.

Diffuser, Air: A circular, square, or rectangular air distribution outlet, generally located in the ceiling and comprised of deflecting members discharging supply air in various directions and planes, and arranged to promote mixing or primary air with secondary room air.

Draft: A current of air, when referring to the pressure difference which causes a current of air or gases to flow through a flue, chimney, heater, or space; or to a localized effect caused by one or more factors of high air velocity, low ambient temperature, or direction of air flow, whereby more heat is withdrawn from a person's skin than is normally dissipated.

Drier: A manufactured device containing a desiccant placed in the refrigerant circuit. Its primary purpose is to collect and hold within the desiccant, all water in the system in excess of the amount which can be tolerated in the circulating refrigerant.

Drift: In a water spray device, the entrained unevaporated water carried from the device by air movement through it.

Drip: A pipe, or a steam trap and a pipe considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

Dry: To separate or remove a liquid or vapor from another substance. The liquid may be water, but the term is also used for removal of liquid or vapor forms of other substances.

Dry-bulb, Room: The dry-bulb (dewpoint, etc.) temperature of the conditioned room or space.

Duct: A passageway made of sheet metal or other suitable material, not necessarily leaktight, used for conveying air or other gas at low pressures.

Dust: An air suspension (aerosol) or particles of any solid material, usually with particle size less than 100 microns.

Effect, Humidifying: Latent heat of water vaporization at the average evaporating temperature times the number of pounds of water evaporated per hour in Btuh.

Effect, Total Cooling: The difference between the total enthalpy of the dry air and water vapor mixture entering a unit per hour and the total enthalpy of the dry air and water vapor (and water) mixture leaving the unit per hour, expressed in Btu per hour.

Effect, Sun: Solar energy transmitted into space through windows and building materials.

Effective Area: The net area of an outlet or inlet device through which air can pass; it is equal to the free area of the device times the coefficient of discharge.

Effectiveness (Efficiency): The ratio of the actual amount of heat transferred by a heat recovery device to the maximum heat transfer possible between the airstreams (sensible heat/sensible heat, sensible heat/total heat, or total heat/total heat).

Equal Friction Method: A method of duct sizing wherein the selected duct friction loss value is used constantly throughout the design of a low pressure duct system.

Enthalpy: Thermodynamic property of a substance defined as the sum of its internal energy plus the quantity Pv/J, where P = pressure of the substance, v = its volume, and J = the mechanical equivalent of heat. Also called "total heat" and "heat content."

Enthalpy, Specific: A term sometimes applied to enthalpy per unit weight.

Entrainment: The capture of part of the surrounding air by the air stream discharged from an outlet (sometimes called secondary air motion).

Entropy: The ratio of the heat added to a substance to the absolute temperature at which it is added.

Entropy, Specific: A term sometimes applied to entropy per unit weight.

Equivalent Duct Diameter: The equivalent duct diameter for a rectangular duct with sides of dimensions a and b is $\sqrt{4ab/\pi}$.

Evaporation: Change of state from liquid to vapor.

Evaporative Cooling: The adiabatic exchange of heat between air and a water spray or wetted surface. The water approaches the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger.

Evaporator: That part of a refrigerating system in which refrigerant is vaporized to produce refrigeration.

Extended Surface: Heat transfer surface, one or both sides of which are increased in area by the addition of fins, discs, or other means.

Face Area: The total plane area of the portion of a grille, coil, or other items bounded by a line tangent to the outer edges of the openings through which air can pass.

Face Velocity: The velocity obtained by dividing the air quantity by the component face area.

Fahrenheit: A thermometric scale in which 32 degrees denotes freezing and 212 degrees the boiling point of water under normal pressure at sea level (14.696 psi).

Fan, Centrifugal: A fan rotor or wheel within a scroll type housing and including driving mechanism supports for either belt drive or direct connection.

Fan Performance Curve: Fan performance curve refers to the constant speed performance curve. This is a graphical presentation of static or total pressure and power input over a range of air volume flow rate at a stated inlet density and fan speed. It may include static and mechanical efficiency curves. The range of air volume flow rate which is covered generally extends from shutoff (zero air volume flow rate) to free delivery (zero fan static pressure). The pressure curves are generally referred to as the pressure-volume curves.

Fan, Propeller: A propeller or disc type wheel within a mounting ring or plate and including driving mechanism supports for either belt drive or direct connection.

Fan, Tubeaxial: A propeller or disc type wheel within a cylinder and including driving mechanism supports for either belt drive or direct connection.

Fan, Vaneaxial: A disc type wheel within a cylinder, a set of air guide vanes located either before or after the wheel and including driving mechanism supports for either belt drive or direct connection.

Filter: A device to remove solid material from a fluid.

Fin: An extended surface to increase the heat transfer area, as metal sheets attached to tubes.

Flow, Laminar or Streamline: Fluid flow in which each fluid particle moves in a smooth path substantially parallel to the paths followed by all other particles.

Flow, Turbulent: Fluid flow in which the fluid moves transversely as well as in the direction of the tube or pipe axis, as opposed to streamline or viscous flow.

Fluid: Gas, vapor, or liquid

Fluid, Heat Transfer: Any gas, vapor, or liquid used to absorb heat from a source at a high temperature and reject it to a lower temperature substance.

Force: The action on a body which tends to change its relative condition as to rest or motion.

Free Area: The total minimum area of the openings in an air inlet or outlet (or the net area of the opening in system components) through which air can pass.

Free Delivery-type Unit: A device which takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.

Freezing Point: Temperature at which a given liquid substance will solidify or freeze on removal of heat. Freezing point for water is 0°C (32°F).

Fumes: Solid particles commonly formed by the condensation of vapors from normally solid materials such as molten metals. Fumes may also be formed by sublimation, distillation, calcination, or chemical reaction wherever such processes create airborne particles predominantly below one micron in size. Such solid particles sometimes serve as condensation nuclei for water vapor to form smog.

Gas: Usually a highly superheated vapor which, within acceptable limits of accuracy, satisfies the perfect gas laws.

Gas, Inert: A gas that neither experiences nor causes chemical reaction nor undergoes a change of state in a system or process; e.g. nitrogen or helium mixed with a volatile refrigerant.

Gas Constant: The coefficient "R" in the perfect gas equation: $PV = RT$.

Grains of Moisture: The unit of measurement of actual moisture contained in a sample of air. (7000 grains = one pound of water).

Gravity, Specific: Density compared to density of standard material; reference usually to water or to air.

Grille: A louvered or perforated covering for an air passage opening which can be located in the sidewall, ceiling, or floor.

Head, Dynamic or Total: In flowing fluid, the sum of the static and velocity heads at the point of measurement.

Head, Static: The static pressure of fluid expressed in terms of the height of a column of the fluid, or of some manometric fluid, which it would support.

Head, Velocity: In a flowing fluid, the height of the fluid or of some manometric fluid equivalent to its velocity pressure.

Heat: The form of energy that is transferred by virtue of a temperature difference.

Heat, Latent: Change of enthalpy during a change of state, usually expressed in Btu per lb. With pure substances, latent heat is absorbed or rejected at constant pressure. Heat added or subtracted which is associated with the change of state of a substance (liquid becomes gas, solid becomes liquid, etc.) but does not change the dry bulb temperature of the substance. In psychrometry, a change of wet bulb and dew point temperatures does occur.

Heat, Sensible: Heat which is associated with a change in temperature; specific heat exchange of temperature; in contrast to a heat interchange in which a change of state (latent heat) occurs.

Heat, Specific: The ratio of the quantity of heat required to raise the temperature of a given mass of any substance one degree to the quantity required to raise the temperature of an equal mass of a standard substance (usually water at 59°F) one degree.

Heat, Total (Enthalpy): The sum of sensible heat and latent heat between an arbitrary datum point and the temperature and state under consideration.

Heat Capacity: The amount of heat necessary to raise the temperature of a given mass one degree. Numerically, the mass multiplied by the specific heat.

Heat Conductor: A material capable of readily conducting heat. The opposite of an insulator or insulation.

Heat Exchanger: A device specifically designed to transfer heat between two physically separated fluids.

Heat of Fusion: Latent heat involved in changing between the solid and the liquid states.

Heat of Vaporization: Latent heat involved in the change between liquid and vapor states.

Heat Pump: A refrigerating system employed to transfer heat into a space or substance. The condenser provides the heat while the evaporator is arranged to pick up heat from air, water, etc. By shifting the flow of air or other fluid, a heat pump system may also be used to cool the space.

Heat Transmission: Any time-rate of heat flow; usually refers to conduction, convection and radiation combined.

Heat Transmission Coefficient: Any one of a number of coefficients used in the calculation of heat transmission by conduction, convection, and radiation, through various materials and structures.

Heating, Regenerative (or Cooling): process of utilizing heat, which must be rejected or absorbed in one part of the cycle, to perform a useful function in another part of the cycle by heat transfer.

Humidifier: A device to add moisture to air.

Humidifying Effect: The latent heat of vaporization of water at the average evaporating temperature times the weight of water evaporated per unit of time.

Humidistat: A regulatory device, actuated by changes in humidity, used for the automatic control of relative humidity.

Humidity: Water vapor within a given space.

Humidity, Absolute: The weight of water vapor per unit volume.

Humidity, Percentage: The ratio of the specific humidity of humid air to that of saturated air at the same temperature and pressure, usually expressed as a percentage (degree of saturation; saturation ratio).

Humidity Ratio: The ratio of the mass of the water vapor to the mass of dry air contained in the sample.

Humidity, Relative: The ratio of the mol fraction of water vapor present in the air, to the mol fraction of water vapor present in saturated air at the same temperature and barometric pressure; approximately, it equals the ratio of the partial pressure or density of the water vapor in the air, to the saturation pressure or density, respectively, of water vapor at the same temperature.

Humidity, Specific: Weight of water vapor (steam) associated with 1 lb weight of dry air, also called humidity ratio.

Hygroscopic: Absorptive of moisture, readily absorbing and retaining moisture.

Inch of Water (in. w.g.): A unit of pressure equal to the pressure exerted by a column of liquid water 1 inch high at a temperature of 4° C or 39.2° F.

Induction: The capture of part of the ambient air by the jet action of the primary air stream discharging from an air outlet.

Infiltration: Air flowing inward as through a wall, crack, etc.

Insulation, Thermal: A material having a relatively high resistance to heat flow and used principally to retard heat flow.

Isentropic: An adjective describing a reversible adiabatic process; a change taking place at constant entropy.

Isobaric: An adjective used to indicate a change taking place at constant pressure.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

Law of Partial Pressure, Dalton's: Each constituent of a mixture of gases behaves thermodynamically as if it alone occupied the space. The sum of the individual pressures of the constituents equals the total pressure of the mixture.

Liquefaction: A change of state to liquid; generally used instead of condensation in case of substances ordinarily gaseous.

Load: The amount of heat per unit time imposed on or required from a system.

Louver: An assembly of sloping vanes intended to permit air to pass through and to inhibit transfer of water droplets.

Manometer: An instrument for measuring pressures: essentially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so constructed that the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: The quantity of matter in a body as measured by the ratio of the force required to produce given acceleration, to the acceleration.

Media: The heat transfer material used in rotary heat exchangers, also referred to as *matrix*

Melting point: For a given pressure, the temperature at which the solid and liquid phases of the substance are in equilibrium.

Micron: A unit of length, the thousandth part of 1 mm or the millionth of a meter.

Modulating: Of a control, tending to adjust by increments and decrements.

Outlet, Ceiling: A round, square, rectangular, or linear air diffuser located in the ceiling, which provides a horizontal distribution pattern of primary and secondary air over the occupied zone and induces low velocity secondary air motion through the occupied zone.

Outlet, Slotted: A long, narrow air distribution outlet, comprised of deflecting members, located in the ceiling, sidewall, or sill, with an aspect ratio greater than 10, designed to distribute supply air in varying directions and planes, and arranged to promote mixing of primary air and secondary room air.

Outlet, Vaned: A register or grille equipped with vertical and/or horizontal adjustable vanes.

Outlet Velocity: The average velocity of air emerging from an opening, fan, or outlet, measured in the plane of the opening.

Output: Capacity; duty; performance; net refrigeration produced by system.

Outside Air Opening: Any opening used as an entry for air from outdoors.

Overall Coefficient of Heat Transfer (thermal transmittance): The time rate of heat flow through a body per unit area, under steady conditions, for a unit temperature difference between the fluids on the two sides of the body.

Performance Factor: Ratio of the useful output capacity of a system to the input required to obtain it. Units of capacity and input need not be consistent.

Point, Critical: Of a substance, state point at which liquid and vapor have identical properties; critical temperature, critical pressure, and critical volume are the terms given to the temperature, pressure, and volume at the critical point. Above the critical temperature or pressure, there is no demarcation line between liquid and gaseous phases.

Point of Duty: Point of duty is a statement of air volume flow rate and static or total pressure at a stated density and is used to specify the point on the system curve at which a fan is to operate.

Point of Operation: Used to designate the single set fan performance values which correspond to the point of intersection of the system curve and the fan pressure-volume curve.

Point of Rating: A statement of fan performance values which correspond to one specific point on the fan pressure-volume curve.

Preheating: In air conditioning, to heat the air ahead of other processes.

Pressure: The normal force exerted by a homogeneous liquid or gas, per unit of area, on the wall of its container.

Pressure, Absolute: Pressure referred to that of a perfect vacuum. It is the sum of gauge pressure and atmospheric pressure.

Pressure, Atmospheric: It is the pressure indicated by a barometer. Standard Atmosphere is the pressure equivalent to 14.696 psi or 29.291 in. of mercury at 32° F.

Pressure, Critical: Vapor pressure corresponding to the substance's critical state at which the liquid and vapor have identical properties.

Pressure, Gauge: Pressure above atmospheric.

Pressure, Hydrostatic: The normal force per unit area that would be exerted by a moving fluid on an infinitesimally small body immersed in it if the body were carried along with the fluid.

Pressure, Partial: Portion of total gas pressure of a mixture attributable to one component.

Pressure, Saturation: The saturation pressure for a pure substance for any given temperature is that pressure at which vapor and liquid, or vapor and solid, can coexist in stable equilibrium.

Pressure, Static: The normal force per unit area that would be exerted by a moving fluid on a small body immersed in it if the body were carried along with the fluid. Practically, it is the normal force per unit area at a small hole in a wall of the duct through which the fluid flows (piezometer) or on the surface of a stationary tube at a point where the disturbances, created by inserting the tube, cancel. It is supposed that the thermodynamic properties of a moving fluid depend on static pressure in exactly the same manner as those of the same fluid at rest depend upon its uniform hydrostatic pressure.

Pressure, Total: In the theory of the flow of fluids, the sum of the static pressure and the velocity pressure at the point of measurement. Also called dynamic pressure.

Pressure, Vapor: The partial pressure exerted by the water vapor contained in air.

Pressure, Velocity: In moving fluid, the pressure capable of causing an equivalent velocity, if applied to move the same fluid through an orifice such that all pressure energy expended is converted into kinetic energy.

Pressure Drop: Pressure loss in fluid pressure, as from one end of a duct to the other, due to friction, dynamic losses, and changes in velocity pressure.

Primary Air: The initial airstream discharged by an air outlet (the air being supplied by a fan or supply duct) prior to any entrainment of the ambient air.

Properties, Thermodynamic: Basic qualities used in defining the condition of a substance, such as temperature, pressure, volume, enthalpy, entropy.

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere.

Psychrometric Chart: A graphical representation of the thermodynamic properties of moist air.

Pyrometer: An instrument for measuring high temperatures.

Radiation, Thermal: The transmission of heat through space by wave motion; the passage of heat from one object to another without warming the space between.

Radius of Diffusion: The horizontal axial distance an airstream travels after leaving an air outlet before the maximum stream velocity is reduced to a specified terminal level.

Refrigerant: The fluid used for heat transfer in a refrigerating system, which absorbs heat at a low temperature and a low pressure of the fluid and rejects heat at a higher temperature and a higher pressure of the fluid, usually involving changes of state of the fluid.

Register: A grille equipped with an integral damper or control valve.

Resistance, Thermal: The reciprocal of thermal conductance.

Resistivity, Thermal: The reciprocal of thermal conductivity.

Return Air: Air returned from conditioned or refrigerated space.

Room Dry Bulb (dewpoint, etc.) The dry-bulb (dewpoint, etc.) temperature of the conditioned room or space.

Saturation, Degree of: The ratio of the weight of water vapor associated with a pouind of dry air to the weight of water vapor associated with a pound of dry air saturated at the same temperature.

Secondary Air: The air surrounding an outlet that is captured or entrained by the initial outlet discharge airstream (furnished by a supply duct or fan).

Semi-Extended Plenum: A trunk duct that is extended as a plenum from a fan or HVAC unit to serve multiple outlets and/or branch ducts.

Sensible Heat Factor: The ratio of sensible heat to total heat.

Sensible Heat Ratio, Air Cooler: The ratio of sensible cooling effect to total cooling effect of an air cooler.

Sorbent: See absorbent.

Standard Air Density: Standard air density has been set at 0.075 lb/ft^3. This corresponds approximately to dry air at 70°F and 29.92 in. Hg.

Standard Rating: A standard rating is a rating based on tests performed at Standard Rating Conditions.

Static Regain Method: A method of duct sizing wherein the duct velocities are systematically reduced, allowing a portion of the velocity pressure to convert to static pressure offsetting the duct friction losses.

Subcooling: The difference between the temperature of a pure condensable fluid below saturation and the temperature at the liquid saturated state, at the same pressure.

Subcooling, Specific: The difference between specific enthalpies of a pure condensable fluid between liquid at a given temperature below saturation and liquid at saturation, at the same pressure.

Sublimation: A change of state directly from solid to gas without appearance of liquid.

Superheat, Specific: The difference between specific enthalpies of a pure condensable fluid between vapor at a given temperature above saturation and vapor at dry saturation, at the same pressure.

Superheat The difference between the temperature of a pure condensable fluid above saturation and the temperature at the dry saturated state, at the same pressure.

Surface, Heating: The exterior surface of a heating unit. *Extended heating surface* (or *extended surface*), consisting of fins, pins, or ribs which receive heat by conduction from the prime surface. *Prime surface:* heating surface having the heating medium on one side and air (or extended surface) on the other.

System: Central Fan A mechanical, indirect system of heating, ventilaitng, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and conveyed to and from the rooms by means of a fan and a system of distributing ducts.

System, Closed: A heating or refrigerating piping system in which circulating water or brine is completely enclosed, under pressure above atmospheric, and shut off from the atmosphere except for an expansion tank.

System, Duct: A series of ducts, conduits, elbows, branch piping, etc. designed to guide the flow of air, gas or vapor to and from one or more locations. A fan provides the necessary energy to overcome the resistance to flow of the system and causes air or gas flow through the system. Some components of a typical system are louvers, grilles, diffusers, filters, heating and cooling coils, energy recovery devices, burner assemblies, volume dampers, mixing boxes, sound attenuators, the ductwork and related fittings.

System, Flooded: A system in which only part of the refrigerant passing over the heat transfer surface is evaporated, and the portion not evaporated is separated from the vapor and recirculated.

System, Gravity Circulation: A heating or refrigerating system in which the heating or cooling fluid circulation is effected by the motive head due to difference in densities of cooler and warmer fluids in the two sides of the system.

System, Run-around: A regenerative-type, closed, secondary system in which continuously circulated fluid abstracts heat from the primary system fluid at one place, returning this heat to the primary system fluid at another place.

System, Unitary A complete, factory-assembled and factory-tested refrigerating system comprising one or more assemblies which may be shipped as one unit or separately but which are designed to be used together.

System Curve: A graphic presentation of the pressure vs. volume flow rate characteristics of a particular system.

System Effect Factor: A pressure loss factor which recognizes the effect of fan inlet restrictions, fan outlet restrictions, or other conditions influencing fan performance when installed in the system.

Temperature, Absolute Zero: The zero point on the absolute temperature scale, 459.69 degrees below the zero of the Fahrenheit scale, 273.16 degrees below the zero of the Celsius scale. (The zero point on the kelvin temperature scale.)

Temperature, Critical: The saturation temperature corresponding to the critical state of the substance at which the properties of the liquid and vapor are identical.

Temperature, Dewpoint: The temperature at which the condensation of water vapor in a space begins for a given state of humidity and pressure as the temperature of the vapor is reduced. The temperature corresponding to saturation (100 percent relative humidity) for a given absolute humidity at constant pressure.

Temperature, Dry-bulb: The temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.

Temperature, Effective: An arbitrary index which combines into a single value the effect of temperature, humidity, and air movement on the sensation of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation.

Temperature, Mean Radiant (MRT): The temperature of a uniform black enclosure in which a solid body or occupant would exchange the same amount of radiant heat as in the existing non-uniform environment.

Temperature, Saturation: The temperature at which no further moisture can be added to the air-water vapor mixture. Equals dew point temperature.

Temperature, Wet-bulb: Thermodynamic wet-bulb temperature is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. Wet-bulb temperature (without qualification) is the temperature indicated by a wet-bulb psychrometer constructed and used according to specifications.

Temperature, Wet-bulb Depression: Difference between dry-bulb and wet-bulb temperatures.

Temperature Difference, Mean: Mean of difference between temperatures of a fluid receiving and a fluid yielding heat.

Thermocouple: Device for measuring temperature utilizing the fact that an electromotive force is generated whenever two junctions of two dissimilar metals in an electric circuit are at different temperature levels.

Thermodynamics: The science of heat energy and its transformations to and from other forms of energy.

Thermodynamics, Laws of: Two laws upon which rest the classical theory of thermodynamics. These laws have been stated in many different, but equivalent ways.

The First Law

(1) When work is expended in generating heat, the quantity of heat produced is proportional to the work expended; and, conversely, when heat is employed in the performance of work, the quantity of heat which disappears is proportional to the work done (Joule);

(2) If a system is caused to change from an initial state to a final state by adiabatic means only, the work done is the same for all adiabatic paths connecting the two states (Zemansky);

(3) In any power cycle or refrigeration cycle, the net heat absorbed by the working substance is exactly equal to the net work done.

The Second Law

(1) It is impossible for a self-acting machine, unaided by any external agency, to convey heat from a body of lower temperature to one of higher temperature (Clausius);

(2) It is impossible to derive mechanical work from heat taken from a body unless there is available a body of lower temperature into which the residue not so used may be discharged (Kelvin);

(3) It is impossible to construct an engine that, operating in a cycle, will produce no effect other than the extraction of heat from a reservoir and the performance of an equivalent amount of work (Zemansky).

Throw: The horizontal or vertical axial distance an airstream travels after leaving an air outlet before the maximum stream velocity is reduced to a specified terminal level; e.g., 200, 150, 100, or 50 fpm.

Total Heat: Thermodynamic property of a substance defined as the sum of its internal energy plus the quantity Pv/J, where P = pressure of the substance, v = its volume, and J = the mechanical equivalent of heat. Also called "enthalpy" and "heat content."

Total Pressure Method: A method of duct sizing which allows the designer to determine all friction and dynamic losses in each section of a duct system (including the total system).

Transmission: In thermodynamics, a general term for heat travel; properly, heat transferred per unit of time.

Transmission, Coefficient of Heat: Any one of a number of coefficients used in the calculation of heat transmission by conduction, convection, and radiation, through various materials and structures.

Transmittance, Thermal (U factor): The time rate of heat flow per unit area under steady conditions from the fluid on the warm side of a barrier to the fluid on the cold side, per unit temperature difference between the two fluids.

Vane Ratio: In air distributing devices, the ratio of depth of vane to shortest opening width between two adjacent grille bars.

Vapor: A gas, particularly one near to equilibrium with the liquid phase of the substance and which does not follow the gas laws. Usually used instead of gas for a refrigerant, and, in general, for any gas below the critical temperature.

Vapor, Saturated: Vapor in equilibrium with its liquid: i.e., when the numbers per unit time of molecules passing in two directions through the surface dividing the two phases are equal.

Vapor, Superheated: Vapor at a temperature which is higher than the saturation temperature (i.e., boiling point) at the existing pressure.

Vapor, Water: Used commonly in air conditioning parlance to refer to steam in the atmosphere.

Vapor Barrier: A moisture-impervious layer applied to the surfaces enclosing a humid space to prevent moisture travel to a point where it may condense due to lower temperature.

Velocity: A vector quantity which denotes, at once, the time rate and the direction of a linear motion.

Velocity, Outlet: The average discharge velocity of primary air being discharged from the outlet, normally measured in the plane of the opening.

Velocity Reduction Method: A method of duct sizing where arbitrary reductions are made in velocity after each branch or outlet.

Velocity, Room: The average, sustained, residual air velocity level in the occupied zone of the conditioned space; e.g., 65, 50, 35 fpm.

Velocity, Terminal: The highest sustained airstream velocity existing in the mixed air path at the end of the throw.

Ventilation: The process of supplying or removing air, by natural or mechanical means, to or from any space. Such air may or may not have been conditioned.

Viscosity: That property of semifluids, fluids, and gases by virtue of which they resist an instantaneous change of shape or arrangement of parts. It is the cause of fluid friction whenever adjacent layers of fluid move with relation to each other.

Volume: Cubic feet per pound of dry air in the air-water vapor mixture as used in psychrometrics.

Volume, Specific: The volume of a substance per unit mass; the reciprocal of density.

Wet-bulb Depression: Difference between dry-bulb and wet-bulb temperatures.

B ECONOMIC TERMS

Base Year: The year to which all future and past costs are converted.

Compound Interest: Interest paid on the principal and on any accumulated interest.

Constant Dollars: Values expressed in terms of the general purchasing power of the dollar in the base year. Constant dollars do not reflect price inflation.

Construction Cost: The "as-built" cost of construction based on bid price, including cost escalation.

Cost Effective: Estimated benefits (savings) from an energy conservation investment project are equal to or exceed the costs of the investment where both are assessed over the life of the project.

Current Dollars: Values expressed in terms of actual prices of each year; i.e., current dollars reflect inflation.

Differential Cost: The difference in total cost between two alternatives.

Differential Energy Escalation Rate: The expected difference between a general rate of inflation and the rate of cost increases assumed for energy.

Discount Rate: The rate of interest reflecting the time value of money that is used to convert benefits and costs occurring at different times to a common time. OMB Circular A-94 specifies that the discount rate for evaluating government projects be 10 percent. This 10 percent represents the rate of interest after inflation is removed.

Discount Factor: A multiplicative number for converting costs and benefits occurring at different times to a common basis. Discount factors are obtained by solving a discount formula based upon one dollar of costs or benefits and the assumed discount rate.

Discounted Payback Period: The time required for the annual net benefits derived from an investment to pay back the investment, considering the time value of money.

Discounting: A technique for converting costs and benefits occurring over time to equivalent amounts at a common point in time.

Economic Life: That period over which an investment is considered to be the lowest cost alternative for satisfying a particular need.

Effective Interest Rate: The actual annual rate determined from the rate for a shorter period. If we use i as the interest rate for the short period and n as the number of periods (in this case in each year), then

$$\text{Effective rate} = (1 + i)^n - 1$$

Interest: Payment for the use of money.

Interest Period: The time period for which the interest rate is applied.

Interest Rate: The ratio, usually expressed as a percentage, of the amount paid at the end of a period of time and the amount of money owed at the start of the period.

Investment or Initial Cost: The sum of the planning, design, and construction costs necessary to provide a finished building ready for use.

Life-cycle Costing (LCC): A method of economic evaluation of alternatives which considers all relevant costs associated with each alternative activity or project during the time it is in use. For buildings, life-cycle costs include all costs of owning, operating, and maintaining a building over its economic life, including its energy costs.

Maintenance and Repair Cost: The total of labor, material, transportation, and other related costs incurred in conducting corrective and preventative maintenance and repair on a building and its systems, components, and equipment.

Net Present Value of Savings: The present value of energy savings minus (or plus) the present value of the increase (or decrease) in all future non-energy costs.

Nominal Interest Rate: An annual rate expressed as a simple product of the rate for a smaller period multiplied by the number of these periods in a year. Thus, with a 1% interest rate for three months, the nominal annual rate would be 4%.

Operating Cost: The expenses incurred during the normal operation of a building or a building system, component, or equipment, including costs of manpower, fuel, power, water, etc.

Payback Period: The length of time required for the stream of net cash proceeds or cost savings produced by an investment to equal the original cash outlay required by the investment; see "Discounted Payback Period."

Present Value: Past and future costs or benefits expressed as a time — equivalent amount as of the present time, taking into account the time value of money.

Present Value Factor: The number by which a future value may be multiplied to find its value in today's dollars, based on a given discount rate.

Principal: The amount of money outstanding at the start of an interest period.

Recurring Costs: Those costs which recur on a periodic basis throughout the life of a project.

Residual (Salvage) Value: The net sum to be realized from disposal of an asset at the end of its economic life or at the end of the study period.

Sensitivity Analysis: Testing the outcome of an evaluation by altering one or more system parameters from the initially assumed values.

Simple Interest: Where money is borrowed for a number of periods, interest is paid only on the amount of the original principal. Interest is not paid on accumulated interest. This is rarely used.

Study Period: The length of time over which an investment is analyzed.

Sunk Cost: A cost which has already been made and should not be considered in measuring the economic performance of an investment alternative.

Time Value of Money: The difference between the value of a dollar today and its value at some future time if invested today at a stated rate of interest.

SECTION XIX

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Sheet Metal and Air Conditioning
Contractors National Association, Inc.
8224 Old Courthouse Road,
Tysons Corner, Vienna, Virginia 22180