PSYCH - 3

MECHANICAL DESIGN MANUAL SUMMARY SHEET

SUBJECT: INDUSTRIAL COOLING AND DEHUMIDIFICATION

DESCRIPTION: Design of systems for cooling and dehumidification which require humidity levels lower than are normal for comfort cooling.

APPLICATION:
Industrial Processes
Candy Manufacturing
Environmental Chambers
Condensation Control for Electronic Devices
Corrosion Control

SPECIFICATION SECTIONS AFFECTED: File Under 15790 Air Coils

COST:

REFERENCES:
ASHRAE 1988 Equipment Handbook, Chapter 7
ASHRAE 1989 Fundamentals Handbook, Chapter 19
Kathabar Catalog
The Dehumidification Handbook, by Cargocaire Engineering Corporation

DATE UPDATED: October 15, 1991

INDUSTRIAL COOLING AND DEHUMIDIFICATION

by

Jeffrey G. Boldt, P.E.
KJWW Engineering Consultants
Rock Island, Illinois

SUMMARY

This paper compares systems for cooling and dehumidification of air with emphasis on applications which require low humidity. Typical uses for low humidity air conditioning are:

- Industrial Processes
- Candy Manufacturing
- Environmental Chambers
- Condensation Control for Electronic Devices
- Corrosion Control of Storage Areas

LOAD CALCULATION AND CONTROL

Calculation of heat loss and gain is a well established science and will not be discussed here. Calculation of moisture loads is a much less precise science. This paper gives only the basic principles of latent load calculation.

Latent load an be a major portion of the total load. Sources of moisture include permeation of the room envelope due to vapor pressure differences and infiltration, openings in the vapor barrier, personnel, products and make-up air. Vapor pressure differences can reach 0.6 inches of mercury (0.3 psi or 8.2 inches water column).

It is vital that a good vapor barrier be provided and that air leakage be minimized. This has a major affect on the size, cost and energy use of the system.

PSYCHROMETRIC CHARTS

In order to understand the processes of cooling and dehumidification it is first necessary to understand psychrometric charts. A "psych chart" is a graph which shows the properties of air at various temperatures and containing various amounts of moisture.

3/96

The most commonly used psych chart is ASHRAE PSYCHROMETRIC CHART NO. 1 which is based on the air pressure at sea level and covers a range of temperatures between 32 and 120 degrees F. See figure 1. Other charts are available for high and low temperature applications and other altitudes.

At the bottom of the chart the **dry bulb temperature** is listed. This is the temperature which a thermometer measures. It does not indicate the amount of moisture in the air. Along the right edge of the chart are **humidity ratio** values. These indicate the amount of moisture in each pound of dry air. The figures vary between zero and 0.030 pounds of moisture per pound of dry air. Air containing no moisture is called **dry air**.

Around the entire perimeter of the chart are **enthalpy** figures. These represent the amount of energy in each pound of dry air. The amount of enthalpy increases when either dry bulb temperature or humidity ratio increases. Enthalpy figures are arbitrarily based on a value of zero Btu per pound for dry air at zero degrees F.

Nearly parallel to the enthalpy lines are a lighter set of lines of **wet-bulb temperature**. These lines represent the dry bulb temperature which the air would be cooled to if it were cooled by evaporating water into it until **saturated** (able to accept no more moisture). An evaporative cooler cools air along the wet-bulb lines. The maximum wet-bulb temperatures experienced in the midwest are usually around 80 degrees. When your neighbor comments that the temperature was 100 degrees and 100% relative humidity that he is mistaken. This would represent a wet-bulb temperature of 100 degrees.

Dew point temperature or **dew point** is the temperature at which a sample of air when cooled without addition or removal of moisture will become saturated and condensation will begin to form.

Relative humidity (RH) is indicated by a set of curved lines. Relative humidity is 100% when air is saturated at a given temperature and is zero percent for dry air. Note that warm air can accept more moisture than cool air.

Standard air is defined as dry air at 59 degrees F and 29.921 inches barometric pressure (average sea level pressure). This is the air which is assumed in the formula Q=cfm*1.085*delta-T. Because of humidity in the supply air the actual figure is closer to 1.1 than 1.085 for most cooling systems. If designing at high altitudes this figure is not correct and compensation must be made for less dense air with less heating or cooling capacity per cubic foot.

Volume is shown by lines sloping more steeply than enthalpy lines. The figures range from 12.5 to 15.0 cubic feet per pound of dry air.

3/96

PSYCHROMETRIC CHART FOR COMFORT COOLING

Figure 2 represents a common psych chart for a comfort cooling application. Point 1 is the space condition of 75 degrees and 52% RH. Point 2 represents the outside air condition on a peak day of 95 degrees F and 78 degrees wet-bulb. Point 3 is the mixed-air condition assuming that the system uses 25% outside air. This air is cooled from point 3 to point 4 by a cooling coil. Initially the cooling coil has no effect on humidity ratio, but as the sir approaches saturation moisture is removed. The air does not ever quite touch the saturation line. A typical cooling coil will have a discharge of 55 degrees dry-bulb and 54.8 degrees wet-bulb.

The energy used by the cooling coil can be determined from the formula Q=4.5*cfm*delta-H, where H is enthalpy. If we assume a 1000 cfm system our energy use is 4.5*1000*(31.6-22.3) = 41,850 Btuh.

This is an efficient system for comfort cooling because the resulting humidity of 52% is acceptable and no additional energy needs to be expended to adjust the humidity. This is not necessarily true for process cooling.

LOW HUMIDITY MANUFACTURING AREA WITH COMFORT COOLING SYSTEM

The remainder of this paper will use a hypothetical Candy Manufacturing area to show the advantages and disadvantages of various types of HVAC systems. This is a common application with humidity requirements which are different from those for comfort cooling.

Our example is a 1000 square foot area with a **sensible** (dry-bulb heat) load of 20 Btuh per square foot and a **latent** (moisture) load of 5 Btuh per square foot. This area must be maintained at 70 degrees dry-bulb and 35% RH which is typical for a confectionery manufacturing area. The area requires 200 cfm of outside air for ventilation per ASHRAE Standard 62-1989 Ventilation for Acceptable Indoor Air Quality.

Let us examine the energy use of a typical comfort conditioning system for conditioning this space. We begin by constructing our psych chart. Point 1 is the space condition. Point 2 represents the peak load outside air condition. In order to find the mixed air temperature we must first determine the supply air temperature in order to calculate the supply air cfm. We can calculate our **sensible heat ratio** (the sensible load divided by the total sensible plus latent load) to be 80%. Unfortunately, following the sensible heat ratio line to the left yields a supply air

3/96

temperature of approximately 30 degrees F. This would require a direct expansion refrigeration coil with defrosting ability. This means that the fan will periodically stop and the coil will be defrosted using hot gas or electricity. The space humidity and temperature will rise while this cycle is completed. With an 80% sensible heat ratio the system would need to be greatly oversized as it would spend much time defrosting.

The normal solution is to add reheat in order to allow a higher supply air temperature. In order to perform any dehumidification the supply air temperature must be below 42 degrees F. If we assume that 40 degrees is the lowest supply air temperature which can be maintained without requiring defrost we find that the air must be reheated to 62 degrees in order to match the sensible heat ratio. A reheat coil heats the supply air from point 4 to point 5 which is the condition of the air entering the room.

The supply air cfm is found from cfm = Q/(1.1*delta-T) = 25,000 Btuh/[1.1*(70-62)] = 2,840 cfm. Mixed air consists of 200 cfm of outside air which is 7% of the total supply air. The mixed air temperature is, therefore, 71.75 degrees and the mixed air condition is represented by point 3.

The cooling coil energy is 2840*4.5*(24.1-15.1) = 115,020 Btuh.
The reheat coil energy is 2840*1.1*(62-40) = 68,728 Btuh.
Total heating plus cooling energy is 183,748 Btuh.

The load on the space was 25,000 Btuh, however, the conventional system uses 183,748 Btuh of energy to achieve the desired conditions. Is there a more efficient way?

DESICCANT SYSTEMS

Adsorption is the process of attracting and holding of one substance to another without causing a chemical change. This differs from **absorption** which does involve a chemical change. **Sorbents** are materials which attract and hold other gasses or liquids. **Desiccants** are sorbents that have a particular affinity for water. Most commercially used solid desiccants work by adsorption and most liquid desiccants work by absorption.

The primary problem with the conventional system is that it can not dehumidify the air sufficiently because of frosting. This is not a limitation for desiccant systems.

A second problem with the conventional system is that the cooling must be direct expansion as a chilled water system could not produce the required air temperatures. Chilled water systems commonly supply water at 45 degrees and can cool air to 50-55 degrees depending on the coil, velocity, etc. Many organizations

3/96

desire to eliminate small DX systems and would, therefore, prefer systems which can operate with 50-55 degree air.

All desiccant systems work on the same principle. The sorbent has a high affinity for moisture. It attracts moisture from the airstream. This releases heat and the airstream is heated and dehumidified. The moisture is removed from the desiccant by passing a **regeneration** airstream through it. This airstream is heated sufficiently to drive moisture from the desiccant. The reactivation airstream is cooled and humidified. Energy is input to heat the reactivation air and to cool the supply air (unless temperature is not important).

A "rule of thumb" is that desiccant systems should be considered whenever the supply air dewpoint is below 45 degrees and reheat would be needed for a conventional system. Desiccant systems are also economical to operate if a source of inexpensive heat is available as they have a higher ratio of heat-to-cooling energy than conventional systems.

SOLID DESICCANT SYSTEM

Solid sorption systems pass air through a rotary granular desiccant bed (Figure 4) or a rotary HoneyCombe bed (Figure 5). This heats and dehumidifies the air. The bed is typically circular and rotates. One side is adsorbing moisture from the supply air and the other is releasing moisture to a reactivation air stream. The reactivation air is usually outside air which has been heated to produce a low relative humidity which releases the moisture from the bed. The desiccant can be silica gel, lithium chloride or activated alumina, although other materials are also used. The analysis which follows is based on manufacturer's data for a lithium chloride rotary HoneyCombe system.

The desiccant could be placed in the return air, outside air or supply air portion of our system. Energy use is minimized if the sorption occurs in the outside air only. The total amount of moisture removal required is 0.08 pounds per minute which is slightly more than 200 cfm of air can provide. The next most efficient option is to precool the outside air, mix it with a small amount of return air and dehumidify that air before mixing it with the remainder of the mixed air. This is the option that will be analyzed.

The condition of the outside air after being cooled to 52 degrees is indicated by point 3. The minimum air quantity which must pass through the desiccant to provide the full dehumidification requirement is 300 cfm. The mixed air condition for this air is point 4. The desiccant moves the air to point 5. Point 7 is the

3/96

supply air condition. Point 6 is the mixture of dehumidified air and return air with enters the main cooling coil.

The supply air must be along the established sensible heat ratio line and can not be below 52 degrees if we wish to use chilled water for cooling. This establishes point 4.

The supply flow rate needed is 20,000/[1.1*(70-52)] = 1,010 cfm.

Precooling coil energy used is 200*4.5*(41.3-21.4)	= 17,910 Btuh.
Cooling coil energy used is 1,010*1.1*(78-52)	= 28,886 Btuh.
Total cooling energy	= 46,796 Btuh.
Reactivation air flow is 300*39.7/(250-120)	= 92 cfm.
The heating energy for the bed is 92*1.1*(250-95)	= 15,686 Btuh.

Cooling energy = 41% of the conventional system.
Heating energy = 23% of the conventional system.
Supply air flow = 36% of the conventional system.

LIQUID DESICCANT SYSTEM

Figure 6 shows a liquid desiccant system. Liquid desiccant systems primarily use lithium chloride or triethylene glycol. The following analysis is based on a lithium chloride system.

The pumping unit delivers absorbent to the dehumidification chamber where it dries and heats the air exactly as the solid desiccant. The air then passes over a cooling coil for sensible cooling. Approximately 15% of the pumped absorbent solution is diverted to the regeneration chamber. There regeneration air and absorbent are heated and the moisture is driven out of the absorbent.

Liquid desiccant systems are usually factory assembled systems complete with fans and coils. Because liquid absorbent systems crystalize below 15% relative humidity a higher portion of the system air flow must pass through the dehumidifier. For these reasons the liquid system will have somewhat higher energy consumption.

REFERENCES

ASHRAE 1988 Equipment Handbook, Chapter 7

ASHRAE 1989 Fundamentals Handbook, Chapter 19

The Dehumidification Handbook, by Cargocaire Engineering Corporation

3/96

ASHRAE PSYCHROMETRIC CHART NO. 1

NORMAL TEMPERATURE

BAROMETRIC PRESSURE 29.921 INCHES OF MERCURY

COPYRIGHT 1963

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.

SEA LEVEL

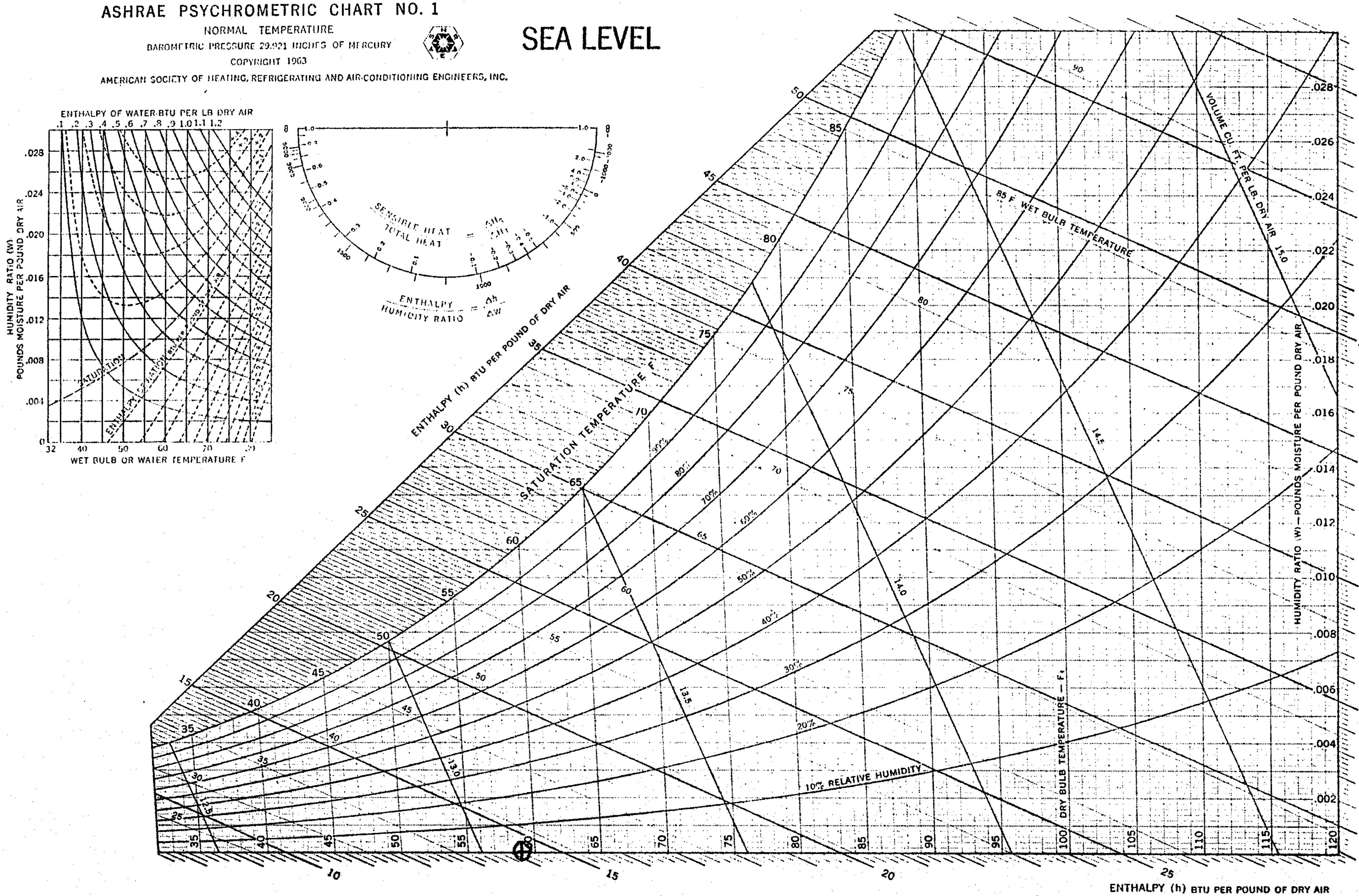
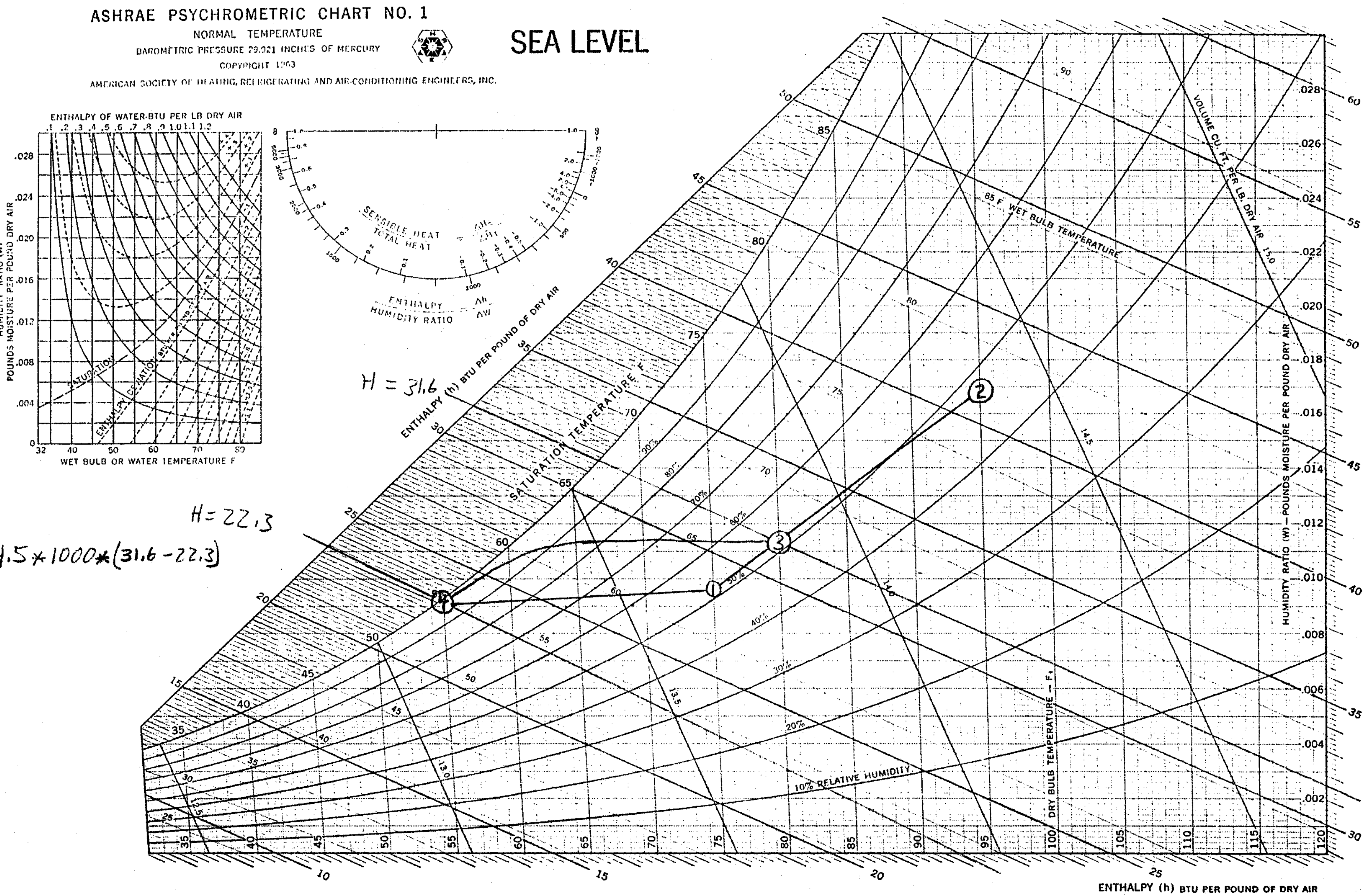


Fig 1

3/96



$Q = 4.5 * 1000 * (31.6 - 22.3)$

=

Figure 2

3/96

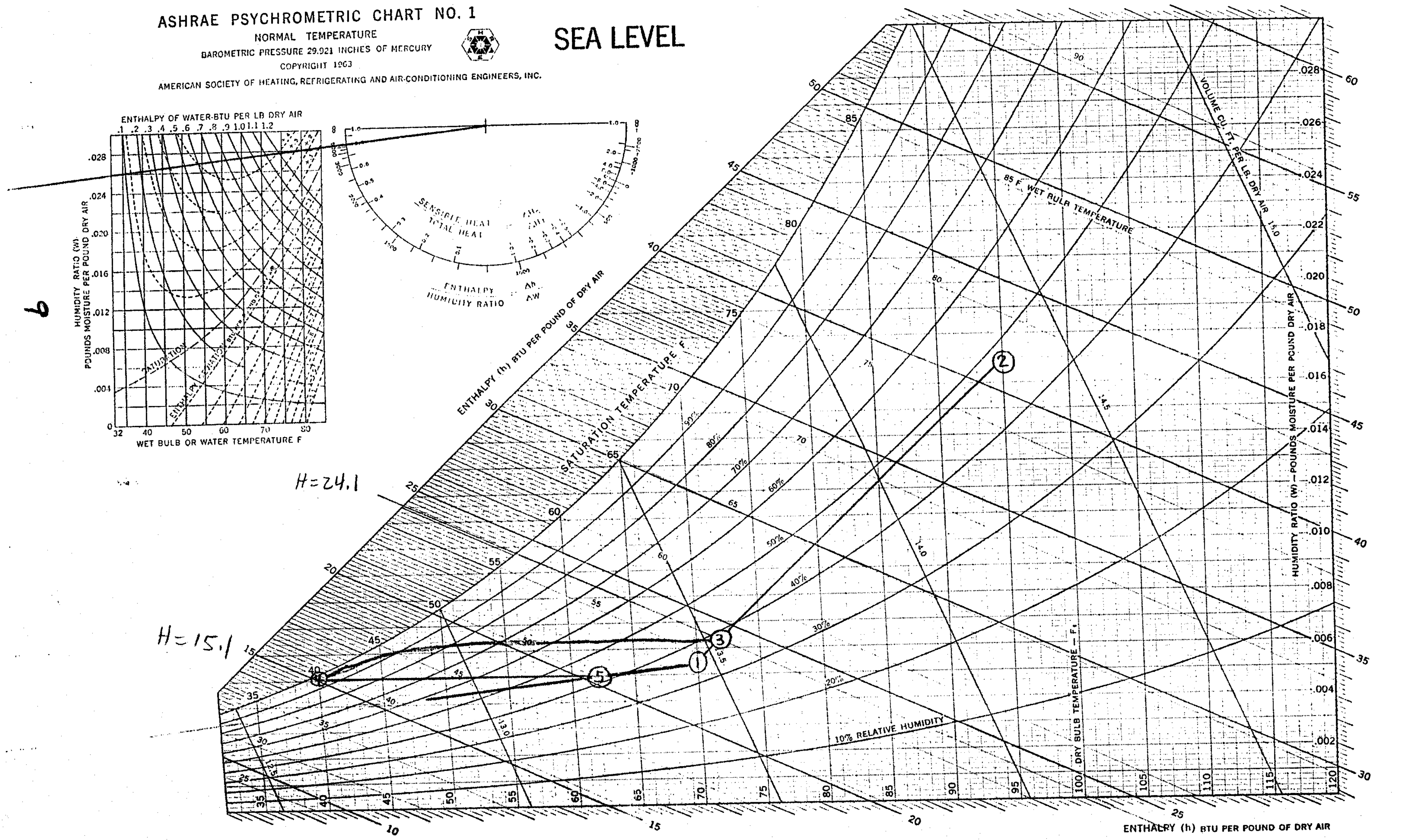
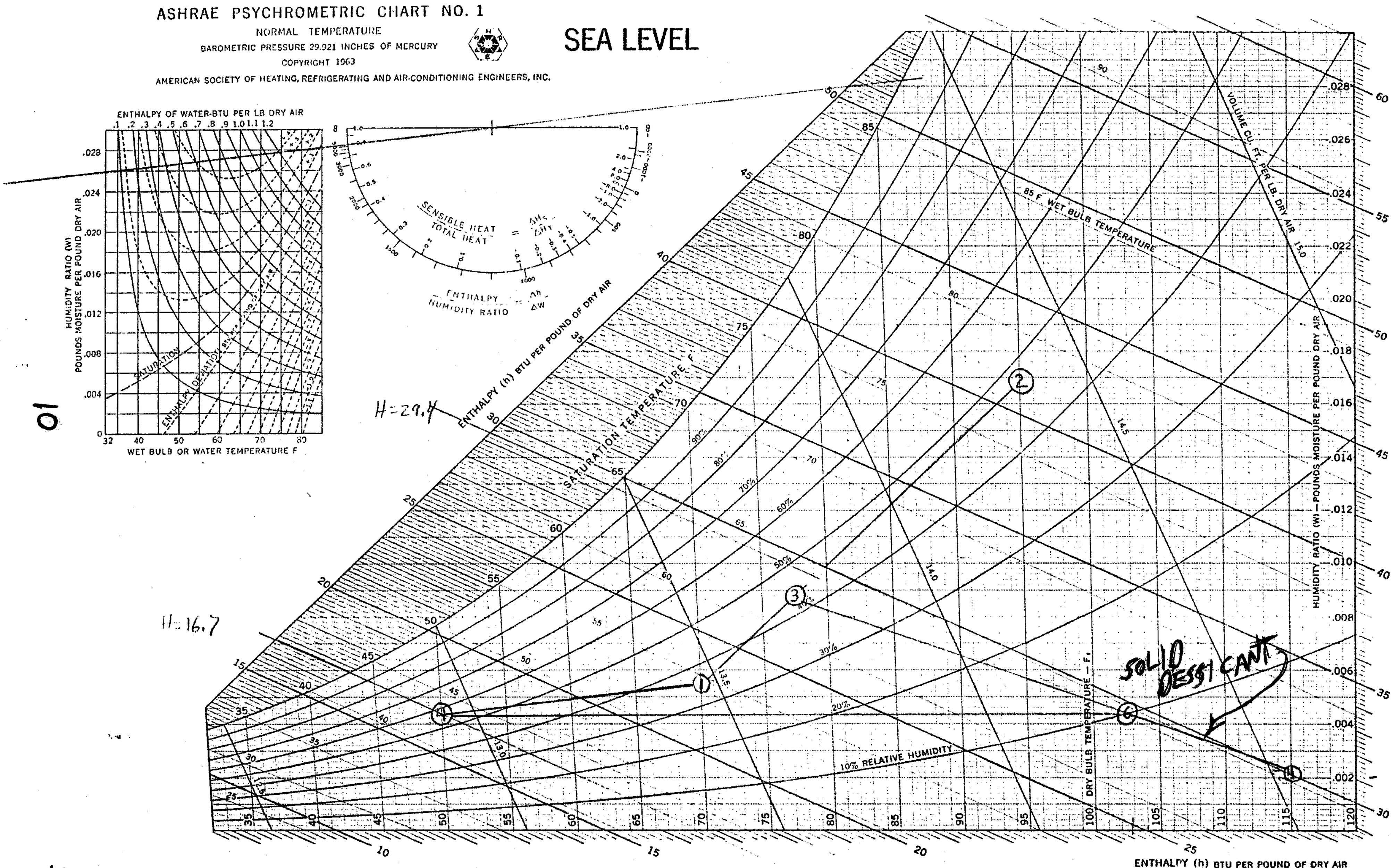


Figure 3

9

3/96



ASHRAE PSYCHROMETRIC CHART NO. 1
NORMAL TEMPERATURE
BAROMETRIC PRESSURE 29.921 INCHES OF MERCURY
COPYRIGHT 1963
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
SEA LEVEL
ENTHALPY OF WATER-BTU PER LB DRY AIR
HUMIDITY RATIO (W) POUNDS MOISTURE PER POUND DRY AIR
WET BULB OR WATER TEMPERATURE F
SENSIBLE HEAT / TOTAL HEAT
ENTHALPY / HUMIDITY RATIO
ENTHALPY (h) BTU PER POUND OF DRY AIR
SATURATION TEMPERATURE F
85 F. WET BULB TEMPERATURE
VOLUME CU. FT. PER LB. DRY AIR
HUMIDITY RATIO (W) — POUNDS MOISTURE PER POUND DRY AIR
DRY BULB TEMPERATURE — F.
10% RELATIVE HUMIDITY
SOLID DESSICANT
H=29.4
H=16.7

Figure 4

3/96