DEHUMIDIFICATION

APPLICATIONS ENGINEERING MANUAL

a spectrum of solutions for

- Industrial Dehumidification
- Drying
- Complete Environmental Control

Inventor of the Rotary Silica Gel Dehumidifier

An ISO 9001:2008 & 14001:2004 company

Providing Complete Environmental Control Systems to the World

Dehumidifiers for all Applications

Including but not limited to:

- Ammunition Storage
- Archives
- Breweries
- Candy Manufacturing
- Chemicals
- Chocolates
- Clean Rooms
- Computer Rooms
- Corrosion Control
- Defence Equipments
- Electronics
- Fertilizers
- Films
- Food & Food Packaging
- Glass Laminating
- Hospitals
- Ice Rinks
- Indoor Arenas
- Laboratories
- Leather
- Libraries
- Lithium Chloride Batteries
- Machine Parts
- Marine
- Military Applications
- Military Equipment Storage
- Milk Powder
- Museums
- Optical (Polishing)
- Pharmaceuticals
- Photographic Materials
- Plant Lay-up
- Plastics
- Powder Storage
- Pumping Stations
- Razor Blades
- Safety Glass
- Seed Drying
- Semiconductor Components
- Ship Lay-up
- Snack Foods
- Steel-Belted Tyres
- Tea & Coffee
- Tyre & Rubber Manufacturing
- Typewriter Ribbons (Coating)
- Water & Waste Water
- Welding Electrodes
- Wood Drying

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INTRODUCTION: MEET BRY-AIR

Bry-Air (Asia) is an ISO 9001:2008 and ISO 14001:2004 certified company specializing in technologies and products with desiccants at the core in relation to air. Bry-Air is known world wide for its expertise as a "solutions" company for moisture and humidity control.

Today, Bry-Air (Asia) has a global presence with a wide range of "airgineering" equipment, strong brands and a world wide global network . Bry-Air (Asia) has wholly owned subsidiaries in Malaysia, China and Germany, a licensee in Brazil and is the only Indian HVAC& R Company to acquire an overseas dehumidification firm, A+H, Hamburg, Germany. The China operations have 4 offices in Shanghai, Beijing, Guangzhou and Wuhan. Bry-Air also has representative offices in Bangkok-Thailand, Johannesburg-South Africa, Perth-Australia, Muntinlupa-Philippines, Jakarta-Indonesia and Istanbul-Turkey and office and warehouse in Sharjah-UAE

HOW IT ALL BEGAN

....in USA

In 1963, Arthur G. Harms, a Bryant sales representative decided to buy the Bryant division of Carrier-Air Conditioning Corporation. Shortly after the acquisition, Bry-Air began manufacturing in Sunbury, Ohio.

...in India

In India, it all started almost 30 years ago with the setting up of Arctic India Sales in 1979 as a marketing company for environmental control products. Arctic India Sales started out by representing Bry-Air Inc. USA for its dehumidifier in India. In 1981, it promoted the first joint venture in a small scale sector, Bry-Air India (Now Bry-Air Asia) with Bry-Air Inc. USA to become the first dehumidifier manufacturing company, east of the Suez.

Research and Development

Bry-Air has always been in the forefront of technology and innovation through extensive research and R&D in moisture control, air treatment and optimisation of energy usage. A pioneer in the field of dehumidification, Bry-Air has maintained the initiative, searching for new ways to bring the benefits of research to satisfy customer needs. Bry-Air is the inventor of the Rotary Silica Gel Dehumidifier.

The latest innovation from Bry-Air is the Green Dehumidifier 'EcoDry' incorporating the innovative BRYSMART EMS (Energy Management System) (patent pending) which will allow energy savings up to 45%. This is the 9th patent to the credit of the Bry-Air Group.

Our Range

Operating within a broad framework of "Environment and Energy", Bry-Air today is a multi-product, multi-location company with plant locations in India, Malaysia, China, Brazil, Germany and USA. The company's plants are amongst the most modern ones, supported by computerized 3D designing, CNC fabricated and powder coating facilities and automated conveying system.

The range of products manufactured by Bry-Air are:

- Desiccant Dehumidifiers
- Air and Gas Purification Systems,
- Tray Dryers
- Drying, Conveying & Blending equipments
- Flexible Barrier Storage Systems and
- Complete Environmental Control Systems.

Our Customers

Our customers are from a plethora of industries and our dehumidifiers are effectively being employed in production, processing, storage and packaging in: pharmaceuticals, chemicals, defense, safety glass, food and drinks, seeds, fertilizer, paper and printing, plastics, wood , leather , cement etc. The products manufactured by Bry-Air are not only sold in the Indian Market through its network, own direct sales and service offices, but also exported to over 40 countries all over the world.

THE PURPOSE OF THIS MANUAL

The Bry-Air Dehumidifier Manual is designed to provide specific information on humidity, dehumidification processes, and steps necessary for selecting the appropriate dehumidifier for a specific application.

A series of discussions on the basic nature of wet and dry air, plus details on the use of dehumidifiers in numerous industrial environments are incorporated in Manual. Several real-life situations are also detailed. Calculations help illustrate how to choose the most suitably sized dehumidifier in each scenario. (For assistance in choosing the dehumidifier to meet your particular needs, please contact the dehumidifier experts at Bry-Air.)

We at Bry-Air hope that this Manual will be of significant assistance to all who consider dehumidified air as a necessity.

WHAT IS RELATIVE HUMIDITY?

Anyone who has suffered the discomforts of hot, humid summer weather understands that it is not just the heat, but also the humidity that makes the air feel so miserable. That "muggy" feeling comes from the relative humidity or saturation level– that is, the amount of water contained by a pound of air at a specific temperature and atmospheric pressure.

When air has 50% relative humidity (RH), we say it is 50% saturated (the terms are numerically so close that we use them interchangeably). The air contains about half the water it could hold at the same temperature and pressure. Obviously, as air approaches 100% saturation, it can take on less and less water until at 100% RH, the air cannot hold more water.

Relative humidity is determined by comparing the "wet-bulb" and "dry-bulb" readings of a humidity measuring device- a hygrometer (see the table below). Once known, these values identify a point on the psychometric chart (see Appendix *I,* page 35) where air vapor mixture properties can be read directly.

The following hygrometers can measure the humidity or hygrometric state of the air:

A **psychrometer** consists of two thermometers (matched in type, scale and range), one of which has a cloth wick–a "sock" applied to its bulb. To use, the wick is wetted with distilled water and ventilated with air moving at a recommended 900 to 1000 feet per minute (fpm) or more at right angles to the instrument.

Dew-point hygrometers visually note when humidity-that is, water in the air-condenses on a cooled metallic surface. The temperature at which this condensation or dew-point occurs can help determine other air properties via charts and tables. Several types of dew-point hygrometers are widely used.

These **hygrometers** may be mechanical, electrical, electrolytic or gravimetric in nature. However, no organic material consistently reproduces its action over an extended time, especially in extremes of humidity or temperature. So this category is of limited value.

MEASUREMENT OF HUMIDITY

HYGROMETER	APPLICATION	RANGE, °F	PRECISION, °F	LIMITATIONS
Psychrometer	Room of building, outside air, air moving in ducts	20° to 140°wb	±.4° wb	Should be used in air stream moving about 1000 fpm, small diameter wet bulbs may be used at lower velocities; difficult to use at subfreezing temperatures
Dew-point; Condensation type	Automated systems in industrial processes, meteorological observations, remote locations	–150° to 200° dp	0.2° to 2° dp	Expensive
Fog-type	Wide range, method for sampling	-80°to ambient dp	±2° dp	Manual, series of readings needed for measurement
Salt-phase transition	Meteorological measurements laboratory; simple to use	0° to 160° dp 56° dp DEP	±2° dp	Not usable below approximately 15% RH; susceptible to some atmospheric contaminants
Dimensional change: Mechanical	Control, measurement where air motion is slight	20 to 100% rh -40° to 125° dp	±3% RH	Frequent calibration required when used at extremes of range; hair has considerable lag, low sensitivity, and is adversely affected by temperature above 125°F and RH below 20%
Electrical conductivity	Measurement, control	-40° to 120° dp	±1.5 to 3% RH	Susceptible to damage by air contaminants-some to water; require frequent calibration checks.
Electrolytic	Measurement	-60° to –5° dp	5% of scale range	Ordinarily limited to low humidities
Gravimetric	Measurement, standard		±0.1 to 2%	Special equipment and extreme care required for high accuracy

PART ONE: USES OF DRY AIR

In many manufacturing processes, humidity control is necessary to completing a particular process successfully. Because failure of a process can be directly tied to humidity level control, it is vital to know:

- What equipment is available
- How to choose appropriately sized equipment
- How to effectively use the equipment to control moisture in the process area

Since dry air may be desired for many commercial applications, and specific problems encountered may be as complex as the atmosphere itself, three important steps are the focus of this booklet: how to select, size, and apply the correct Bry-Air Dehumidifier. Consider the following dozen typical situations.

FOOD PRODUCTION

When exposed to high relative humidity, such familiar foods as potato chips, dry breakfast cereals, and soda crackers exhibit an affinity for water. These and similar foodstuffs are manufactured using high temperature processes, so we expect that excess water has been driven out and the foods are dry. However, if these foodstuffs are allowed to remain exposed in a humid environment, even for a short time, they will absorb water from the surrounding air. Although the quality is not affected, these foods characteristically become soggy and rubbery and generally not very appetizing.

Controlled humidity is vital to successful storage of seeds and grain, and for food and pharmaceutical production

In processing such powdery foods as cocoa, gelatin, and dehydrated concentrates, the presence of moisture in the surrounding atmosphere can cause tiny particles of the powder to stick or cluster together, thus inhibiting their free flow in manufacturing or packaging processes.

Processing machinery can also be affected by moisture in the air, which can interfere with operation and obstruct the free and easy movement of the foodstuff.

Obviously, when moisture can have such a profound effect on food and/or machinery, the solution lies in conditioning the air surrounding the processing and manufacturing area.

PHARMACEUTICAL PRODUCTION

Rapid technological advances in the pharmaceutical industry have focused attention on a vast number of moisture control issues. For example, the presence or absence of a specific amount of moisture in the processing area may be required to grow certain organic cultures. Or, the presence of moisture may be absolutely necessary for the manufacture of a particular drug. Similarly, the absence of moisture may be equally imperative for the production of some other drug**. Strict control of moisture** is a key factor in the manufacture of most drugs and medicines.

As with foodstuffs, many materials used to produce pharmaceuticals have a physical affinity for moisture. This can cause lumping or caking of powdered material. Further, some powders that are bound into a capsule or formed into a tablet under high pressures will adhere only when in a dry state, Humidity can cause a tablet to crumble, and in some cases, it can cause the drug to decompose and diminish in its therapeutic value.

To assure consistently high quality drugs, the processing area and machinery must be surrounded by air whose dryness is accurately known and controlled.

INDUSTRIAL CHEMICAL PRODUCTION

The same sort of lumping and caking of powdered substances previously discussed is also a major problem in industrial chemical production. Some chemicals decompose in the presence of water vapor. In other situations, water vapor can actually cause a chemical reaction that changes the character of the product.

Atmospheric moisture is also a natural enemy to many grinding and pulverizing operations. Water vapor in contact with the product can make it resilient and difficult to grind, causing it to cling to the grinding machine and defy pneumatic conveyance from one process area to another.

HUMIDITY CONTROL OF WAREHOUSE STORAGE

Although mildew, rust, and corrosion are the enemies of goods in storage, they cease being threats when the moisture is substantially reduced in the storage area. Generally, an atmosphere of less than 40% RH keeps these degradative processes dormant.

Maintaining a dormant state is also important for seed storage. For example, if corn seeds are stored in ambient conditions, it's possible that as few as 7% will germinate the following season. Contrast that to seeds stored in a controlled environment; they usually germinate in the 90% range.

HUMIDITY CONTROL OF ROOMS WHERE EQUIPMENT IS OPERATING

In many instances, moisture is detrimental to functioning of electrical or mechanical devices. Thousands of electrical relays may be threatened by pitting due to excessive arcing under high humidity conditions. Also, the presence of water vapor may corrode the contact points of infrequently operated electrical conductors resulting in poor closure of electrical circuits. In extreme cases, there may be no closure at all.

In other such places as radar stations and industrial applications, electronic equipment is also subject to loss of efficiency and high maintenance costs when exposed to humid conditions. And for work environments containing computers and other data processing systems, humidity control is essential for assuring proper equipment function, Where practical, maintenance of dry enclosures may be necessary, Sometimes, it may be necessary to condition the equipment itself, since electrical operating devices are often enclosed in their own metal cases.

HUMIDITY CONTROL OF PACKAGING EQUIPMENT ROOMS

Frequently, the equipment used to package products will not function efficiently or properly if the surrounding air is humid. Candy wrapping machines and packaging machines for food powders or drug packets are typical examples. Something as simple as a cake mix becomes a double problem; not only does the powdery mix clump and not flow properly in humid conditions, but the packaging equipment is also hindered by the humidity.

Depending on the product, it may be necessary to dry the packaging room and even go to the extreme of providing a dry bath for storage bins or hoppers, especially where powders are used. In situations where packages are heat-sealed, a container's own moisture content may adversely affect the adhesion of the sealing material being used. Here neither the product nor the machinery is directly responsible for the problem; instead, the moisture content of the package is the culprit. Surrounding the area with dry air is the solution.

ORGANIC PRODUCT DEHYDRATION

Organic products are particularly challenging because of their high degree of affinity for water. Unfortunately, it is often impossible to use heat to release this water because heat can have a damaging effect.

Dry, relatively cool air can be used to dry organic materials, but it must be circulated under varying velocities, and this creates the problem of special handling that is required with finely divided particles, for example.

INORGANIC PRODUCT DEHYDRATION

Inorganic products are generally easier to dry than organic products because heat can be used as a drying agent. However, many inorganic compounds absorb large quantities of water. This is not water of crystallization- that is, it does not enter the lattice structure of the compound- but it is nonetheless tightly held by the compound. When water of crystallization is involved, even the use of heat can be impractical… or damaging.

But for most inorganic products, dry air may enhance operational efficiency and product quality.

THE EFFECTS OF CONDENSATION

Moisture sweating, particularly on moving parts, can be very detrimental. For example sweating occurs as equipment is being cooled in polymer injection molding operations. Because molds are artificially chilled, dehumidified air must be used to surround them, or condensation will form… and water is one item that must be avoided here.

Another example is the water pumping station whose numerous valves, fittings and other parts may become rusty and need periodic painting or replacement. In a large facility, a major effort of repainting, replacement, and mopping up may be necessary to deal with condensed water.

Insulating the pipes helps reduce the amount of dripping condensate. However, valves and other such fittings that remain uninsulated present a constant maintenance problem.

Dry air in the pumping station and pipe gallery provides a solution.

HYGROSCOPIC RAW MATERIALS STORAGE

When such hygroscopic raw materials as rubber and plastic are used, process difficulties can occur in a humid atmosphere. Moulded products made of these materials can develop "air" pockets caused by stress; other imperfections can result from moisture adsorbed by the raw materials. In automobile production, it may be virtual impossibility to vulcanize tyre cord to rubber when the cord contains moisture.

Dry Air used for storage and possibly in the production area can alleviate this situation.

MARINE AND LAND-BASED BLASTING

In marine and land-based applications, sandblasting removes surface damage and exposes the base material-often metal-that will receive a protective coating.

Inside ships, or in underground or land-based storage tanks, a flow of dehumidified air on the newly prepared surfaces prevents rust or mildew formation while clean-up occurs and the coating step is prepared. Usually the dry air is forced inside the structure via normal ventilation lines.

POWER PLANT AND MARINE LAY-UP

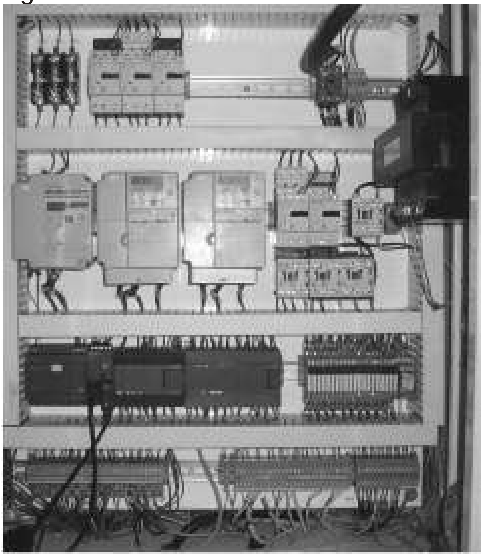
When a nuclear power plant is shut down for refueling-a process that can take a whole year-dehumidified air can keep such non-nuclear components as boilers, condensers, and turbines rust free.

For fossil fuel power stations, the laying-up process is usually part of putting power production on hold. Here the reason may be for furnace or boiler repair or the lay-up might be due to less expensive power becoming available from a nearby source. During these periods, a flow of dehumidified air in the facility is used to prevent rust or other harmful, moisture-related problems.

Ships can also be layed-up. Some are "mothballed" for indefinite storage. Many such vessels are later reactivated, cleaned-up, and set to sea. During the interim, dehumidified air keeps rust, mildew, and corrosion from ruining the engine room, cargo holds, and living or working quarters.

Other ships are part of the "ready fleet"-anchored at sea, fully equipped, and ready for a crew to come on board and set sail almost immediately. These, too, are protected by a steady flow of dehumidified air that is continuously pumped throughout the vessel.

Above: Successful production of moisture-sensitive products relies on dehumidification.



Right: Electrical control equipment is extremely sensitive to moisture which can cause short circuits. A dehumidifier system installation eliminates this problem.

PART TWO:
HOW TO PRODUCE DRY AIR

Because the amount of water that can be contained in air is a function of the temperature and pressure of that air, our next step is to look at ways to remove moisture by changing the temperature or pressure.

USING COMPRESSION TO DRY AIR

As air is compressed, the dew-point or temperature at which water will condense is raised. Therefore, to get dry air we need to find a way to cool the compressed air. But costs can be prohibitive because equipment, space, and auxiliary equipment are necessary for the process. However, if compressed air is already used in the primary operation and only very small amounts of dry air are needed for humidity control, compression may be a feasible route to dry air.

When air at extremely high pressure (over 200 lb/sq in) is needed, small quantities of high pressure air may be used to maintain small enclosures at the required moisture level. It is also possible to use small amounts of the high pressure air with a smaller air facility to control moisture on a limited scale.

USING REDUCED TEMPERATURES TO DRY AIR

Lowering air temperature decreases the air's ability to hold moisture. Thus, the air can be made drier by cooling it. However cooling air just to dry it is usually not practical. An exception might be when cool air is needed anyhow, that air's dryness satisfies the needed moisture conditions, and enough conditioned air is available. Normally, this method is reserved for applications where outdoor air is being dried to levels only slightly lower than the incoming ambient-that is, the system air.

To remove large amounts of water by cooling the air, over-cooling and subsequent reheating air required. But such procedures typically have problems with operation and maintenance, as well as cycle and control; the method is unsuitable for producing large quantities of dry air. Another limitation to this technique is the freezing point of water. When air is dried via refrigeration, the cooling surfaces of the coils may reach sub-freezing temperatures. This causes ice to form, which, in turn, reduces the efficiency of the cooling system. So anti-icing devices or duel systems and defrost cycles may be required.

To prevent such cooling coil icing, a brine spray is commonly used. The brine must be reconstituted periodically or continuously. This requires additional equipment, maintenance and operating costs. Although this strategy is workable and often satisfactory, the complexities associated with cycling and controlling are detracting factors.

A special case involves a brine spray that can pick up moisture from the air at normal temperatures. This brine must be cooled and regenerated or reconcentrated either continuously or periodically. To deliver air at very low moisture, such a system is necessarily complex. For example, the brine must be mechanically refrigerated, and at all levels of drying, cooling must be used during the moisture absorbing cycle and after the regenerating or reconstituting cycles.

USING DESICCANTS TO DRY AIR

The most simple, straightforward way to obtain dry air is to use desiccants-that is, adsorbants or materials that have a natural affinity for water. A desiccant is able to take up the additional moisture given up by the air without changing its size or shape. So as air stream can pass through a desiccant and become significantly drier without elaborate cooling, compression, cooling water, or other complex systems or controls. After the drying task is complete, the desiccant is regenerated via heat. Then the desiccant is ready to dry more air.

A Bry-Air Dehumidifier utilizes only a relatively small amount of desiccant at any one time and constantly regenerates it as part of a continuous cycle. This simple device is manufactured in two designs and many sizes, from very small to very large to meet various dry air requirements.

An added feature of the Bry-Air Dehumidifier is its ability to function equally well at extremely low to very high levels of humidity with no regeneration problems and no changes in cycle control. Its versatility in performing any type of application is unique among most methods of drying air.

PART THREE:
CONSTRUCTION OF CONTROLLED SPACE

To prepare any space for humidity control, certain precautions are necessary, regardless of the type of air drying equipment or the method used to do the drying.

Satisfactory moisture control-better known as customer satisfaction-depends on many variables. Some are listed below.

THE NATURE OF WATER VAPOR

Consider two closed rooms, adjacent to one another. If the partial pressure of the water vapor in room 1 is greater than the partial pressure of the water vapor in room 2, then the water vapor will travel through the wall into room 2 regardless of the composition of the wall.

Let's take the hypothetical example a step further. If the absolute humidity of the air in room 1 is greater than that of the air in room 2, then the water vapor pressure will be higher in room 1. Therefore, when drying room 2, the problem of new water coming through the wall from room 1 must be considered.

A vapor barrier can slow down the passage of vapor from wet to drier areas, but it cannot keep water out; it can only slow the rate of penetration.

The choice of vapor barrier is based on the degree of dryness required in the controlled space, the efficiency of the equipment being used for drying, and the cost of construction.

Commercial vapor barriers-moisture resistant construction material, paints, and other coatings-offer a variety of design alternatives. Manufacturers of vapor barrier materials can supply specific information on their products.

In addition to the vapor barrier, certain aspects of construction must be given careful attention.

CONSTRUCTION CONSIDERATIONS

Several techniques control the permeation of water vapor:

1. Any vapor barrier must be continuous, without breaks or tears.
2. All lap joining must be tightly closed (this is particularly critical when mechanical or caulked joints are used).
3. Insulation between vapor barriers can be a potential problem: if construction occurs in humid weather, water can be "sealed in" between the two vapor barriers.

   Sealed-in vapor will travel into the controlled space and impose an extra drying load on the drying equipment. This extra load lasts only until the insulation dries out, but meanwhile, humidity control is difficult.

   If a heat source is present (even heat from the sun), serious damage can be caused by the expanding trapped vapor. There have been cases when so-called "non-permeable" materials have split open at a joint because of vapor pressure. Examples include a floor or tiled wall that has literally lifted from its mounting surface because the surface was wet during application.
4. Final inside vapor barriers should be applied only after the enclosed area has been dried. Drying equipment should be used to withdraw as much moisture as possible before the final barrier is applied. Of course, without a barrier in place equipment cannot dry the air to design specifications, but a significant amount of moisture can and should be removed before all the vapor barrier material is in place.

   (Although this strategy runs counter to most industrial planning suggestions, the concept of drying the structure before applying the final vapor barrier is a precaution that is often overlooked and can help prevent customer dissatisfaction)
5. All doors-service or personnel-should be weather-stripped or air-locked through vestibules if the desired dryness warrants it. Any crack or opening around a door will admit vapor.

When conveyor openings or similar elements are used, a drop curtain, shroud, or tunnel can restrain the movement of water vapor.

PART FOUR:
CALCULATING THE VARIOUS MOISTURE LOADS

The following methods have been used successfully to calculate vapor loads; replacing the extensive calculations and laboratory tests that might otherwise be required when a designer considers a new space humidity problem or application.

Actual data from moisture loads entering a space through walls, floors, and ceiling are available for various moisture loads and classes of construction. A survey sheet, such as the sample in Appendix *II*, page 39 will help you gather data for the needed calculations.

TABLE I
RECOMMENDED DESIGN OUTSIDE MOISTURE LEVEL

DESIGN		DESIGN	
Outside Wet Bulb	Specific Himidity	Outside Wet Bulb	Specific Himidity
°F	gr/lb	°F	gr/lb
81	149	75	121
80	143	74	117
79	139	73	113
78	134	72	109
77	130	71	106
76	125	70	102

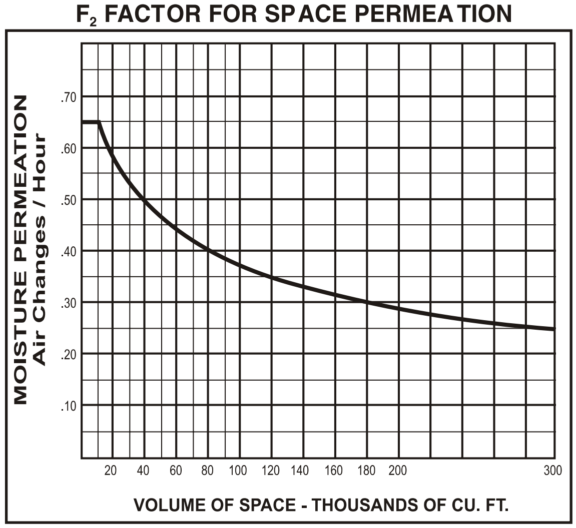
TABLE II
F_1 FACTOR FOR GRAIN DIFFERENCE

Gr/lb Difference	F_1 Factor
35	1.0
40	1.12
50	1.35
60	1.59
70	1.82
80	2.06
90	2.29
100	2.53
110	2.76
120	3.00

For standard types of construction, Bry-Air has determined values for calculating the moisture load entering a space at controlled humidity levels. Usually these calculations are relatively easy. The following tables are aids for load calculations.

Outside humidity levels shown in the Table 1 are deliberately higher than data for design specifications. This compensates for days when the design wet-bulb temperatures are reached and the design dry-bulb temperatures are lower than expected (thus creating higher total humidity). Use the area design wet-bulb and the specific humidity figures shown here to accurately rate the moisture control situation. Further information on design can be found in Appendix *I* and in the ASHRAE Fundamentals Handbook, "Weather Data and Design Considerations".

TABLE III
F_2 FACTOR FOR SPACE PERMEATION



Space moisture load is a combination of permeation and infiltration and both will be encountered in determining the load. Permeation is a straight line function of the difference in interior and exterior vapor pressure (determined by gr/lb). As shown in Table III, infiltration, represented in air changes per hour is not straight line because of the two factors involved:

1. Each pound of air entering the space will impose a moisture load determined by the difference in interior and exterior moisture content.
2. Since the vapor pressure differs as the moisture content, the vapor will move at a higher velocity than the air.

The combination of the two factors, results in the space moisture load increasing at an ever increasing rate as the difference between the interior and exterior moisture contents increase.

In view of the above, the F-1 factor is used to adjust for the increased vapor velocity. Therefore, the combination of the F-1, and F-2 factors represent the space moisture load anticipated from both permeation and infiltration.

TABLE IV

F_3 FACTORS FOR CONTRUCTION		F_4 FACTORS FOR VAPOR BARRIERS	
Masonry or Frame Construction	1.0	Laminated, mylar–metallic or polyethylene film	0.50
Sheet metal, steel welded	0.3	Two layers edge sealed moisture paper	0.67
Module panel, caulked and sealed	0.5	Two coats vapor proof paint	0.75

If the product of F3 x F4 is less than 0.5, use 0.5. If the room is completely vapor proofed, with continuous vapor barrier under the floor (or of all-metal, welded material) the factor may be reduced to 0.3.

CALCULATE THE PERMEATING LOAD THROUGH A STRUCTURE

To determine the grains of moisture penetrating the construction into a controlled space, use the following calculation.

$\frac{V}{C}$ x G x F_B x F_B x F_B x F_B = Grs/hr. (To determine grains/minute divide answer by 60).

= Amount of vapor able to permeate the closed space through construction and vapor barriers.

V = volume of controlled space in question – $ft.^B$

C = 14 = constant used to translate $ft.^B$ to pounds. This constant is used regardless of the density of the air.

G = difference between the grs/lb of outside air and the grs/lb desired in the controlled space.

F_B = moisture difference factor (Multiplier from Table II).

F_B = Permeation factor (Multiplier from Table III).

F_B = Construction factor - Table IV

F_B = Barrier Factor - Table IV

{ See note with Table IV.

The above equation can be used to solve a typical example as follows:

Problem – Find the amount of moisture that will permeate the room defined below.

Sample Calculation – Space to be controlled:

(1) Room with 12" masonry walls.
(2) Two coats of aluminum paint as vapor barrier.
(3) Volume of room – 22,000 $ft.^3$
(4) Outside Design: 95°F db 77°F wb (Table I Shows 130 gr/lb)
(5) Required – To hold in room – 40 gr/lb

$\frac{V}{C}$ x G x F_B x F_B x F_B x F_B = Grains per hour.

V	=	22,000	
C	=	14	
G	=	130 - 40 = 90.	Problem stipulates 40 gr/lb in the room; therefore, 130 - 40 = 90.
F_1	=	2.29 From Table II	(Factor for a moisture difference of 90 gr/lb).
F_2	=	0.58 From Table III	locate 22,000 on bottom line. Travel up and read curve at 0.58.
F_3	=	1.0 From Table IV.	
F_4	=	.75 From Table IV - (Factor for 2 coats of paint)	

$$\frac{22{,}000}{14} \times 90 \times 2.29 \times 0.58 \times 1.0 \times 0.75 = 140{,}884 \text{ grs/hr}$$

$$\frac{140{,}884}{60} = 2348 \text{ grs/min}$$

MOISTURE THROUGH INTERMITTENT OPENINGS

When such openings as service or personnel doors are opened periodically, moisture-laden air can enter the conditioned space. Also, vapor is constantly seeking drier space and will seep around and through doors, even when they are closed.

Obviously the first precaution is to assure that openings are adequately vapor-sealed. Then the drying equipment must deal with the moisture load that comes into a controlled space when the door is open. Assuming that the door is open only for short periods, calculate the moisture load as follows:

$O_{hr} = \frac{A}{C} \times G \times F_1 = \text{Grains}$

O_{hr} = number of times each hour the door is opened. (If unknown assume personnel door to be opened 2 times/hr for every occupant.)

A = area of the door opening in square feet.

C = 7 = constant

G = difference in specific humidity in grs/lb between controlled space and the adjacent space. See table I for outside wb to determine adjacent specific humidity.

F_1 = factor from Table II for moisture difference.

Example:

Door area – 3' x 7'
Door open - 6 times each hour
Moisture difference - 90 grs/lb

Solution:

$$O_{hr} \times \frac{A}{C} \times G \times F_1 = \text{grains per hour of additional load.}$$

$$6 \times \frac{21}{7} \times 90 \times 2.29 = 3710 \text{ grains per hour added to controlled space.}$$

Note that if the door is open for longer periods, use the calculation scheme below.

MOISTURE THROUGH FIXED OPENINGS (CONVEYORS, OPEN WINDOWS, ETC.)

Calculate the amount of moisture that travels through a fixed opening from a wet space to a drier space as follows:

$$\frac{A \times 300}{C \times D} \times G \times F_1 = \text{grains per hour of load through fixed opening.}$$

Where

A = Area of fixed opening in square feet,
300 = Experimental constant - velocity of vapor, ft/hr, at 35 gr difference,
C = 14 = constant factor to translate ft.^{3} to pounds,
D = feet = depth of opening,
G = grains = difference in grs/lb between wet space and drier space.
F_1 = Moisture Difference Factor from Table II.

Example:

Conveyor opening - 2 sq ft
Depth of opening - 1.5 sq ft
Moisture difference - 90 grains

Solution:

$$\frac{A \times 300}{C \times D} \times G \times F_1 = \text{grains per hour}$$

$$\frac{2 \times 300}{14 \times 1.5} \times 90 \times 2.29 = 5{,}889 \text{ grains per hour}$$

MOISTURE ORIGINATING IN THE CONTROLLED SPACE

Moisture or vapor originating in the controlled space comes from any of several sources, depending on the intended use for the space. Three basic sources of moisture are:

Population load, including people and animals

Product load, brought in by the product

Process load

Population Load

People working in an area add moisture to the air because of breathing and the evaporation of perspiration. When animals occupy the controlled space, moisture release is contributed by their excrement.

How much moisture do people or animals add to a controlled space? Such factors as the level of activity and the ambient temperature, atmospheric pressure, and humidity are well documented. (see Appendix III, page 40)

For animals, weigh the amount of water consumed during a given period and assume that much water will be eliminated.

Product Load

Any material manufactured in a controlled area can bring moisture with it and then release the moisture into the work area. Material brought into a warehouse tends to become drier; it gives up moisture over a period of time and loads the drying equipment accordingly.

All materials should be suspect. For example, most metals bring very little moisture, but nonmetals can carry surprisingly large amounts of water. The material's supplier should have information on its moisture carrying characteristics.

If such data are unavailable, a simple test should prevent an unexpected and substantial moisture load problem. Place a sample of the material in a small, dry container, or place some material in a tall hopper and blow air over it to dry it. Measure the moisture loss over an appropriate time interval to determine its dwell time, or how fast it gives up moisture. In some cases, a small pilot plant can be used to acquire definite data.

Process Load

The manufacturing process itself may expel moisture into the atmosphere of a controlled space. Open tanks or trays of liquid will add to the moisture load. (See Appendix *III*.)

Above: Successful production of moisture-sensitive products relies on dehumidification.

Other contributors include open stream exhausts, unvented combustion cycles, and aging or curing cycles.

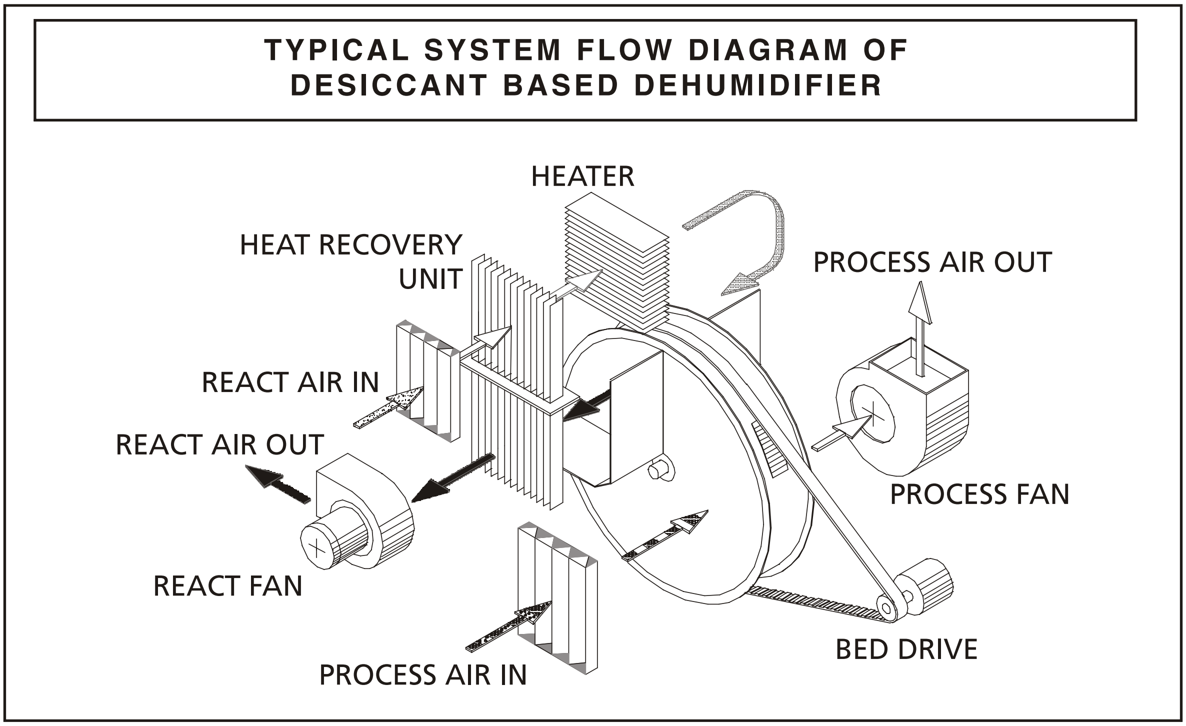
VENTILATING AIR-VAPOR LOAD

(VAPOR BROUGHT IN WITH OUTSIDE AIR)

Ventilating or make-up air from the outside contains moisture that must be removed. Some designers add this moisture load to the total calculated internal load to determine the required capacity of the drying equipment.

However, Bry-Air recommends this air not be considered part of the internal load. Rather, it should be considered at its point of entry. If this added, or make-up air from outside mixes with the return air and all go through the dehumidifier, then it is not added to the internal moisture load. But if only part of this outside/return air mixture passes through the dehumidifier, then the part bypassing the dehumidifier must be added to the internal load of the room. The added air is only part of the total air used in controlling space humidity. Since it rarely gets into the controlled space without first going through the dehumidifier, consider it at its point of entry-at the dehumidifier.

Moisture-laden air enters through the process inlet and moves through the Brysorb Plus™ desiccant media. The desiccant adsorbs the water vapor and the dehumidified air is then delivered through the process outlet directly into the controlled space or air stream. Then, as the desiccant media rotates into the reactivation airstream, the hot air entering through the reactivation inlet drives off the moisture and exhausts it into the atmosphere. After



reactivation the hot, dry desiccant rotates back into the process airstream where a small portion of the process air cools the desiccant so that it can begin the adsorption process all over again.

PART FIVE: SIZING THE DEHUMIDIFIER

When deciding what size dehumidifier to use, remember that controlled space requirements sometimes exceed the anticipated design peak load. Unusual and unforseen humidity loads-such as from abnormal weather conditions or the introduction of high-moisture content raw materials-can burden the drying equipment. Here we present a number of issues that must be considered in approaching and solving specific drying problems. Six typical humidity control example are presented:

Food and drug manufacturing, specifically raw materials and processing equipment (Production of hard candy)

Storage or equipment areas (Standby warehouse)

Product drying

Controlled humidity and temperature areas

Specific purposes for dry air production

Prevention of condensation (Water treatment plant)

Note: Dehumidifier performance used in these examples can be found in Appendix *IV*, page 44

EXAMPLE I: PRODUCTION OF HARD CANDY

During the candy and cough drop production, the material is in a plastic state. It must flow and be shaped by stamping machines. If the presence of excess moisture causes the material to become sticky, it will not flow freely and it will adhere to the stamping machine.

To eliminate this material and equipment problem, dry the surrounding air.

Physical Facts

1. Area to be conditioned – 60' x 42' x 16'
2. Outside design condition – 95°F db*; 75°F wb*
3. Controlled space requirement** - 75°F db; 35% RH
4. Physical openings – 1 door (6'7'); opened 6 times/hr.
5. Number of people working in area – 10
6. Construction – 8" masonry
7. Make-up air specified by owner – 350 cfm.

* db = dry bulb value; wb = wet bulb value
** See Appendix *V*, page 45.

Problem

To determine the size of dehumidifier necessary to maintain the desired controlled space conditions.

Assumptions

1. The door is adequately weather stripped and is of standard construction.
2. Ten workers in the area maintain a moderate pace; each requires ventilating air.
3. The interior of the control space is constructed with two coats of vapor barrier paint.
4. There are no other openings in and out of the controlled space.
5. All physical cracks are sealed.
6. A vapor barrier is provided in or under the concrete floor.

Space Moisture Loads to be Computed

1. Permeation load
2. Load through the door
3. Population load

PERMEATION LOAD

$$\frac{V}{C} \times G \times F_1 \times F_2 \times F_3 \times F_4 = \text{Grains per hour}$$

V = 60 x 42 x 16 = 40,320 $ft.^3$
C = 14 (Specific volume of dry air @ 95°F)
G = 75 grs/lb, outside design wet bulb of 75°F gives 121 gr/lb from Table I. Controlled space requirement of 75°F db, 35% RH yields 46 grains per pound from a standard Psychrometric chart. Therefore, 121-46 = 75 grs/lb.
F_1 = 1.94 From Table II – Factor for moisture difference of 75 gr/lb – interpolated
F_2 = .5 From Table III
F_3 = 1.0 From Table IV - Factor for 8" masonry
F_4 = .75 From Table IV – Factor for 2 coats of paint

$$\frac{40,320}{14} \times 75 \times 1.94 \times .5 \times 1.0 \times .75 = 157,140 \text{ grs/hour}$$

DOOR LOAD

$$O_{hr} \times \frac{A}{C} \times G \times F_1 = \text{grs/hr}$$

O_{hr} = 6
A = 6 x 7 = 42 sq ft
C = 7
G = 75 grs/lb
F_1 = 1.94

$$6 \times \frac{42}{7} \times 75 \times 1.94 = 5238 \text{ gr/hr}$$

POPULATION LOAD

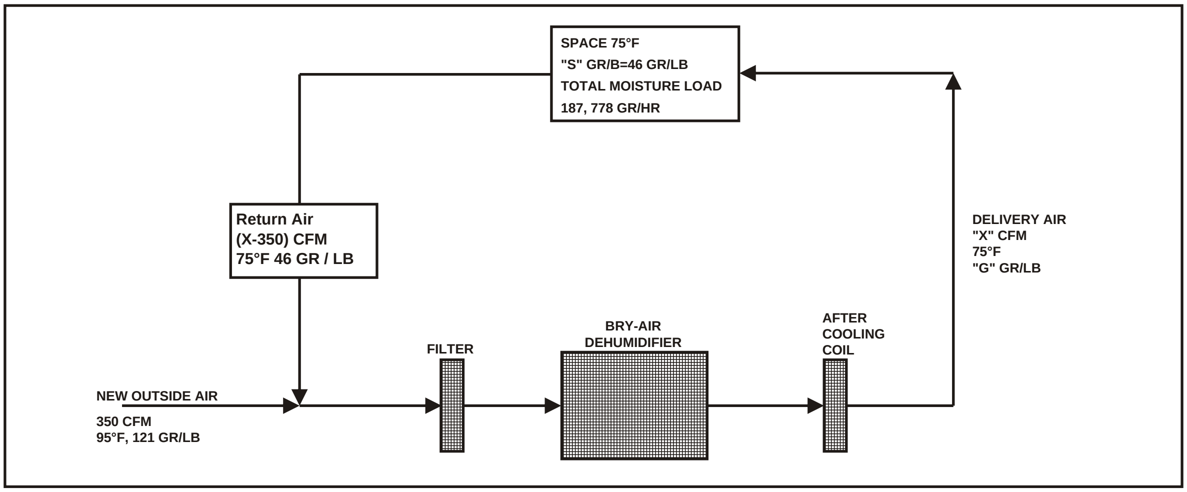
At a db of 75° F and working at a moderate rate, a person will expel 2,540 grains each hour. (See Appendix *III*)

Therefore, ten people will add
10 x 2,540 = 25,400 grains each hour

TOTAL LOAD

157,140 grs/hr – Permeation
5,238 grs/hr – Through door
25,400 grs/hr – Population load
187,778 grs/hr – Total

The drying system and load requirement are shown in the schematic below.



Note that 350- cubic feet per minute (cfm) outside air is based on a requirement of 30 cfm for each of 10 workers is introduced at the dehumidifier. The effect of this air on the ultimate dehumidifier size will be handled below.

Proceed with the following calculation:

$$X = C \times \frac{gr/hr}{60} \div (S-G)$$

Where : X = Delivery air rate from dryer to space in cfm
gr/hr = Total moisture load in grain per hour in the space

C =14 = constant
S = 46 = Grs/lb moisture requirement of controlled space. In the absence of a ventilation requirement this would be the inlet condition at the dryer.
G = Grs/lb of air leaving dryer. Refer to Chart 1, Appendix *IV*, Enter curve at 46 grain "Inlet moisture condition." Intersect 75° Inlet air temp curve at 14 gr/lb.

$$14 \times \frac{187{,}778}{60} \div (46 - 14) = 1369 \text{ cfm}$$

From the above calculation the space moisture load is 187,778 gr/min. 1369 cfm air at 14 grs/lb will maintain the space design conditions.

At this stage in the procedure, it is necessary to resort to the method of approximation to select the correct dryer.

In addition to handling the space load the dryer must handle the moisture load contributed by the 350 cfm outside air requirement. So use a 2000 cfm Bry-Air Dryer (FL*i* - 4200).

If the dryer has a delivery rate of 2000 cfm, and if 350 cfm of outside air is to be introduced, there remains 1650 cfm of air from the conditioned space. Tabulate this air mixture

350 cfm x 121 gr/lb =	42,350
1650 cfm x 46 gr/lb =	75,900
2000 cfm	118,250

Then $\frac{118,250}{2000} = 59.1$ grs/lb

Refer again to Chart 1, Appendix *IV*, it shows that air entering the dryer at 59.1 grs/lb would leave the dryer at approximately 23 grs/lb. (NOTE: Interpolate between the 75° and 85° curves since the air is a mixture of 75°F and 95°F = 79°F.)

Total moisture pickup $\frac{X}{C}$ x (S – G) x 60 – Total Moisture Pickup

$\frac{2000}{14}$ x (46-23) x 60 = 197,143 grs/hr total removal capacity

The following work sheet is a demonstration of what the calculations will look like.

In the above calculations, moisture gain or air leakage in the process ductwork was not considered. If, however, the process and return ductwork did contribute to the moisture load, the total duct volume would be an additional space. Then the permeation calculation on page 9 would be used: V = duct volume; C = 14; F_1 from Table II, with moisture difference G measured from inside process air duct to surround ambient; F_3 for tight, good commercial ductwork = 0.6. Add the resultant moisture gain to the room total load. A nominal allowance for process air lost due to duct leakage = 5%.

Recommendation

Selecting an FL*i* - 4200 at 2000 cfm is the best choice for the hard candy manufacturing example. While it may seem to be an oversized selection, consider that all desiccants in all manufacturers' desiccant dryers will age, will possibly become physically and chemically contaminated with dirt, dust, or chemicals, and will gradually lose their effectiveness. Fortunately, with the FL*i* - 4200 higher levels of moisture in the leaving air (upto 24 grs/lb) dry air-could be tolerated prior to a desiccant change. So what appears to be an oversized selection would actually allow much longer use of a desiccant charge or rotor and provide the economies of longer use.

BRY-AIR DEHUMIDIFIER CALCULATION SHEET

PROJECT: EXAMPLE 1 – PRODUCTION OF HARD CANDY

Conditions				Room Size		
Surrounding	95 FDB	75 FWB	121gr	60'L 42'W 16'H	=	40,320 Ft^3
Design	$\frac{75\ FDB}{20\Delta T}$	58.5 FWB	35% RH	$\frac{40,320\ FT^3}{60}$	=	672

Permeation GR/HR

Volume $\frac{40,320\ FT^3}{14} \times \frac{75}{(\Delta GR)} \times \frac{1.94}{(F_1)} \times \frac{0.5}{(F_2)} \times \frac{1}{(F_3)} \times \frac{0.75}{(F_4)}$ = 157,140

Door Load

Openings/hr $6 \times \text{Area}\ \frac{42FT^2}{7_2} \times \frac{75}{(\Delta GR)} \times \frac{1.94}{(F_1)}$ = 5,238

Openings/hr $0 \times \text{Area}\ \frac{0FT^2}{7} \times \frac{75}{(\Delta GR)} \times \frac{1.94}{(F_1)}$ = 0

Fixed Opening $\frac{0}{(Area)} \times \frac{300}{14} \times \frac{1}{(dep)} \times \frac{75}{(\Delta GR)} \times \frac{1.94}{(F_1)}$ = 0

People Load $\frac{10}{(No.People)} \times \frac{2540}{(F_5)}$ = 25,400

Product Load 0 gr/hr removed = 0

Product Load 0 gr/hr added = 0

TOTAL ROOM LOAD = 187,778

CFM required $14 \times \frac{187,778}{60} \div (46 - 14)$ = 1369 CFM

Make-up Air $\frac{350}{14} \times \frac{75}{(\Delta GR)} \times 60$ = 112,500

TOTAL GR/HR = 300,278

1,650 CFM RETURN AIR @ 75 FDB, 46 GR/LB

350 CFM MAKE-UP AIR @ 95 FDB, 121 GR/LB

= DEHUMIDIFIER INLET CONDITION:

CFM	2,000
DB TEMP.	79
GR/LB	59

Dehumidifier Sizing

$14 \times \frac{300,278}{60} \div (59 - 23)$ = 1,946

CFM Required

Proof

$\frac{2000}{14} \times (46 - 23) \times 60$ = 197,143

System Capability

DEHUMIDIFIER REQUIRED FL*i* - 4200

PROCESS OUTLET TEMP. 130° F

EXAMPLE II: STANDBY WAREHOUSE

Moisture damage in a standby or storage warehouse can be avoided by surrounding the machinery, equipment, or material with dry air.

Physical Facts

1. Area to be conditioned - 210' x 176' x 45' = 1,663,200 cubic feet
2. Outside design condition – 95°F db; 77°F wb
3. Controlled space requirement* - 85°F db; 40% RH
4. No physical openings nor appreciable amount of door openings or closing specified
5. No people working in the area
6. Construction – 8" masonry.

* See Appendix *V*

Problem

To determine the size of the dehumidifier required to maintain standby conditions.

Assumptions

1. All physical cracks are sealed and the floor properly vapor-proofed.
2. If the room is completely vapor-proofed, use Table 4 on page 8.
3. Two coats of vapor barrier paint have been applied externally for metal clad construction.

* **External application is recommended because:**

- Outside walls are usually easier to access than inside walls for paint application.
- Coating the outside walls discourages water permeation into the wall and thus minimizes water accumulation in the wall structure itself.

Space Moisture Loads to be Computed

1. Permeation load
2. Moisture load.

The Permeation Load is the only moisture load involved in this example.

$$\frac{V}{C} \times \Delta G \times F_1 \times F_2 \times F_3 \times F_4 \quad = \quad \text{Grains per hour}$$

Where:

V = 210 x 176 x 45 =1,663,200 cubic feet
C = 14 = Constant
G = 58 grs/lb. Outside design wet bulb of 77°F gives 130 grs/lb from Table I, page 8. Controlled space requirement of 85°F db, 40% RH yields 72. grs/lb from a standard Psychrometric chart. Therefore, 130-72 = 58.
F_1 = 1.54 from Table II - Factor for moisture difference of 63 grs/lb
F_2 = 0.24 from Table III - extrapolated as straight line for a volume of 1,663,200 cubic feet.
F_3 = 1.0 from Table IV - Factor for 8" masonry.
F_4 = .75 from Table IV - Factor for 2 coats of paint.

$$\frac{1{,}663{,}200}{14} \times 58 \times 1.54 \times 0.24 \times 1.0 \times 0.75 = 1{,}910{,}019 \text{ grs/lb.}$$

Refer to schematic below which shows the load requirements and drying system.

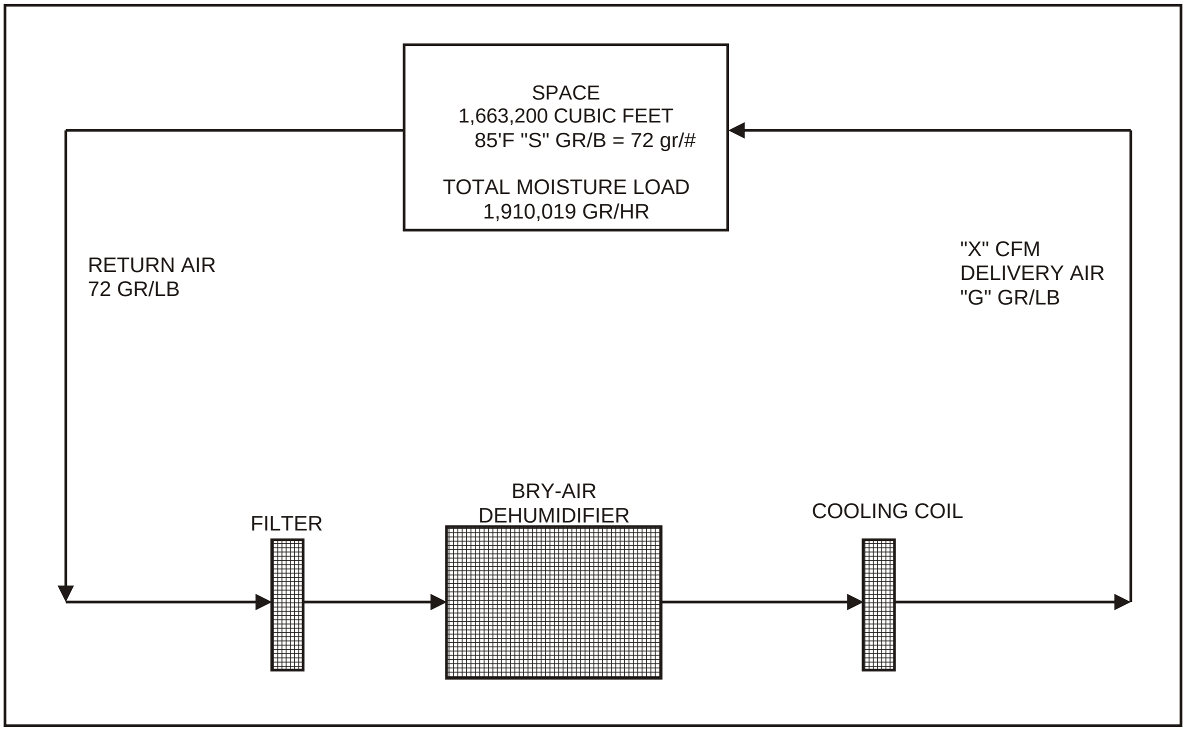
$$X = C \times \frac{gr/hr}{60} \div (S\text{-}G)$$

Where: X = cfm delivery air rate from dryer
C = 14 (constant)
S = 72 grs/lb moisture requirement of controlled space

G = grs/lb in air leaving the dryer. Refer to Chart 1, Appendix *IV*
Enter curve at 72° "Inlet Moisture condition". Interpolate "Inlet Air Temperature Curve" between 75° and 95° and find "leaving moisture" at 33 grs/lb.

$$X = 14 \times \frac{1{,}910{,}019}{60} \div (72 - 33)$$

X = 11,427 cfm



Therefore 11,427 cfm of air (33 gr/lb) from the dehumidifier is needed to maintain a grain level of 72 gr/lb.

In a building of this size and shape, air distribution ducts are practical for effectively spreading the air so it can return to a common point and re-enter the dehumidifier.

Recommendation

Use one Bry-Air FLB-25000, Dehumidifier at 12,500 CFM in this standby warehouse with a fan sized to handle the necessary static pressure of the duct system.

BRY-AIR DEHUMIDIFIER CALCULATION SHEET

PROJECT: EXAMPLE II – STANDBY WAREHOUSE

	Conditions				Room Size		
Surrounding	95 FDB	77 FWB	130gr		210'L 176'W 45'H	=	1,663,200 Ft^3
Design	$\frac{85\ FDB}{10\Delta T}$	0 FWB	$\frac{72gr}{58\Delta GR}$	40RH	$\frac{1{,}663{,}200\ FT^3}{60}$	=	27,720

Permeation — GR/HR

Volume $\frac{1{,}663{,}200\ FT^3}{14} \times \frac{58}{(\Delta GR)} \times \frac{1.54}{(F_1)} \times \frac{0.24}{(F_2)} \times \frac{1}{(F_3)} \times \frac{0.75}{(F_4)}$ = 1,910,019

Door Load

Openings/hr $0 \times \text{Area}\ \frac{0FT^2}{7} \times \frac{58}{(\Delta GR)} \times \frac{1.54}{(F_1)}$ = 0

Openings/hr $0 \times \text{Area}\ \frac{0FT^2}{7} \times \frac{58}{(\Delta GR)} \times \frac{1.54}{(F_1)}$ = 0

Fixed Opening

$\frac{0}{(Area)} \times \frac{300}{14} \times \frac{1}{(dep)} \times \frac{58}{(\Delta GR)} \times \frac{1.54}{(F_1)}$ = 0

People Load

$\frac{10}{(No.\ People)} \times \frac{2540}{(F_5)}$ = 25,400

Product Load

0 gr/hr removed = 0

Product Load

0 gr/hr added = 0

TOTAL ROOM LOAD = 1,910,019

CFM required

$14 \times \frac{1{,}910{,}019}{60} \div (72 - 33)$ = 11,427 CFM

Make-up Air

$\frac{0}{14}\ CFM \times \frac{58}{(\Delta GR)} \times 60$ = 0

TOTAL GR/HR = 1,910,019

12,500 CFM RETURN AIR @ 85 FDB, 72 GR/LB

0 CFM MAKE-UP AIR @ 95 FDB, 130 GR/LB

= DEHUMIDIFIER INLET CONDITION:

CFM	12,500
DB TEMP.	85
GR/LB	72

Dehumidifier Sizing

$14 \times \frac{1{,}910{,}019}{60} \div (72 - 33)$ = 11,427

CFM Required

Proof

$\frac{12{,}500}{14} \times (72 - 33) \times 60$ = 2,089,286

System Capability

DEHUMIDIFIER REQUIRED FLB - 25000

PROCESS OUTLET TEMP. 138° F

EXAMPLE III: PRODUCT DRYING

Here we have a room used to remove water vapor from such products as cattle feed mixes, nylon or rayon cord for tires, raw plastic material, granular chemicals, raw paper stock, cardboard stock for coatings, or other similar products.

In this example, the room is used for drying cattle feed mixes, which are contained on drying carts that stand in the room until the specified level of dryness is attained.

Space condition requirements and product movement rate are determined by the manufacturer.

Physical Facts

1. Drying room size – 40' x 65' x 16'
2. Outside design condition – 93°F; db; 73°F wb
3. Controlled space requirement – 95°F; 15% RH (36 gr/lb)
4. One double door; (a) 6' x 7' (b) Opens at 2 times/hr
5. There are no other openings
6. There are no workers in room except to bring mix in and out
7. Product movement rate – 1500 lb/hr (i.e. carts with trays of mix are moved into the drying room at the rate of 1500 lb/hr)
8. Product enters room at 8% moisture and leaves at 4% moisture
9. Drying room wall construction – 8" masonry
10. 350 cfm outside air required by manufacturer

Problem

To determine the size of the dehumidifier

Assumptions

1. All physical cracks are sealed
2. The double door is weather stripped
3. Two coats of vapor barrier paint have been applied to the wall and ceiling construction of the drying room; the floor is suitably protected against vapor permeation

Moisture Loads to be Computed

1. Product load
2. Permeation load
3. Door load

Product Load

Since the product will lose 4% moisture (by weight) and there are 1500 pounds of product each hour:

1500 lb/'hr x (8 % - 4 %) = 60 lb/hr water removal

Since one pound of water equals 7000 grains, then:

60 x 7000 = 420,000 gr/hr product load

Note that the time needed to reduce the material to a 4% moisture level would have to be given or experimentally determined. These data would determine the amount of material to process and the size of the drying chamber needed.

PERMEATION LOAD

$$\frac{V}{C} \times \Delta G \times F_1 \times F_2 \times F_3 \times F_4 = \text{Grains per hour}$$

V = 40' x 65' x 16' = 41,600 cu ft.
C = 14 constant

G = 77 gr/lb, outside design wb of 73°F gives 113 gr/lb. from Table-I. Drying room space requirement of 95°F, 15% RH yields 36 gr/lb from the Psychrometric Chart.

F_1 = 1.99 From Table II - Factor for moisture difference of 84 grains.

F_2 = 0.50 From Table III - Permeation factor

F_3 = 1.0 From Table IV - Factor for 8" masonry.

F_4 = 0.75 From Table IV - Factor for 2 coats paint.

$$\frac{41{,}600}{14} \times 77 \times 1.99 \times 0.50 \times 1.0 \times 0.75 = \text{grs/hr}$$
$$= 170{,}742 \text{ grs/hr}$$

DOOR LOAD

$O_{hr} = \frac{A}{C} \times \Delta G \times F_1$ = grs/hr

O_{hr} = 2

A = 6' × 7' = 42 sq. ft.

C = 7 (constant)

ΔG = 77 grs/lb

F_1 = 1.99 From Table II - Factor for moisture difference of 84 grains.

$$2 \times \frac{42}{7} \times 77 \times 1.99 = \text{grs/hr.}$$
$$= 1839 \text{ grains}$$

TOTAL MOISTURE LOAD

420,000 gr/hr - Product Load
170,742 gr/hr - Permeation Load
<u>1,839 gr/hr - Door Load</u>
592,581 gr/hr - Total Moisture Load

The 350 cfm outside air requirement will be considered at a later stage in the calculation.

Proceed with calculation as follows:

$$X = C \times \frac{gr/hr}{60} \div (S - G)$$

Where:

X = cfm - air rate from dryer

C = 14 (constant)

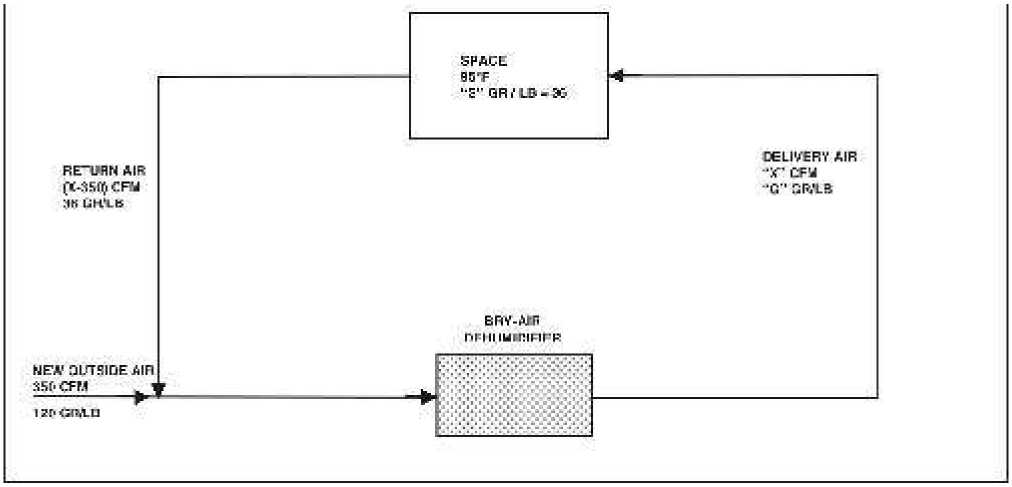
S = 36 = grs/lb drying room controlled space requirement. In the absence of an outside air requirement this would also be the inlet condition at dryer.

G = 15 gr/lb - equals condition of air leaving dryer. Refer to Chart 1, Appendix *IV*.

enter curve at 36 - intersect 95°F curve at 15 gr/lb.

$$X = 14 \times \frac{592{,}321}{60} \div (36 - 15)$$

$$X = 6584 \text{ cfm}$$



Recommendation

FLB 12500 Dehumidifier, rated at 7500 cfm, should be adequate. However, the first step should be to determine if this Dehumidifier has enough capacity to handle the 350 cfm outside air in addition to the moisture load in the drying room.

If the dryer has a delivery rate of 7500 cfm and 350 cfm of outside air is to be introduced, there remains 7150 cfm from the conditioned space. Tabulate this air mixture as follows:

350 cfm x 113 gr/lb =	39,550
7150 cfm x 36 gr/lb =	257,400
7500	296,950

$$\frac{296{,}950}{7500} = 39.6 \text{ gr/lb.}$$

Reference to Chart 1, Appendix *IV*, shows that air entering the dryer at 39.5 gr/lb would leave at 17 gr/lb

$$\frac{7500}{14} \times (39.6 - 17) \times 60 = 726{,}285 \text{ gr/hr}$$

The computed moisture load is 592,581 gr/hr. Therefore, the FLB - 12500 is adequate to handle the moisture load

BRY-AIR DEHUMIDIFIER CALCULATION SHEET

PROJECT: EXAMPLE III – PRODUCT DRYING

	Conditions				Room Size		
Surrounding	93 FDB	73 FWB	113gr		65'L 40'W 16'H	=	41,600 Ft^3
Design	$\frac{95\text{ FDB}}{-2\ \Delta T}$	0 FWB	$\frac{36\text{gr}}{77\Delta GR}$	15RH	$\frac{41600\text{ FT}^3}{60}$	=	693

Permeation — GR/HR

Volume $\frac{41600\text{ FT}^3}{60} \times \frac{77}{(\Delta GR)} \times \frac{1.99}{(F_1)} \times \frac{0.5}{(F_2)} \times \frac{1}{(F_3)} \times \frac{0.75}{(F_4)}$ = 170,742

Door Load

Openings/hr $2 \times \text{Area}\ \frac{42FT^2}{7} \times \frac{77}{(\Delta GR)} \times \frac{1.99}{(F_1)}$ = 1,839

Openings/hr $0 \times \text{Area}\ \frac{0FT^2}{7} \times \frac{77}{(\Delta GR)} \times \frac{1.99}{(F_1)}$ = 0

Fixed Opening

$\frac{0}{(\text{Area})} \times \frac{300}{14} \times \frac{1}{(\text{dep})} \times \frac{77}{(\Delta GR)} \times \frac{1.99}{(F_1)}$ = 0

People Load

$\frac{0}{(\text{No.People})} \times \frac{0}{(F_5)}$ = 0

Product Load

1500 lb/hr × (8% - 4%) × 7000 = 420,000

Product Load

0 gr/hr added = 0

TOTAL ROOM LOAD = 592,581

CFM required

$14 \times \frac{592{,}581}{60} \div (36 - 15)$ = 6,584 CFM

Make-up Air

$\frac{350}{14}\ \text{CFM} \times \frac{77}{(\Delta GR)} \times 60$ = 115,500

TOTAL GR/HR = 1,910,019

7,150 CFM RETURN AIR @ 95 FDB, 36 GR/LB

350 CFM MAKE-UP AIR @ 93 FDB, 113 GR/LB

= DEHUMIDIFIER INLET CONDITION:

CFM	7,500
DB TEMP.	95
GR/LB	40

Dehumidifier Sizing

$14 \times \frac{708{,}081}{60} \div (40 - 17)$ = 7,183

CFM Required

Proof

$\frac{7{,}500}{14} \times (36 - 17) \times 60$ = 610,714

System Capability

DEHUMIDIFIER REQUIRED FLB - 12500

PROCESS OUTLET TEMP. 135° F

EXAMPLE IV: CONTROLLED HUMIDITY AND TEMPERATURE AREAS

Many air conditioned manufacturing areas often have a required air flow to handle a sensible load in that space. This air quantity requirement and the accompanying dehumidifier size are usually greater` than those needed to handle a latent load.

By designing, a system for the sensible load situation and then determining the appropriate dehumidifier to handle the moisture load, the desired conditions for the space can be maintained.

Physical Facts

1. Area to be conditioned – 62.5' x 55' x 14'
2. Outside design conditions – 95°F db; 77°F wb
3. Controlled space requirement – 55°F db; 30 % RH; 20 gr/lb
4. Door – 1 (6' x 8'), 6 openings/hr; 1 (3' x 7'), 4 openings/hr
5. Other (fixed) openings – 2.8 sq. ft., w/tunnel 10' deep
6. Number of people working in area – 10
7. Air required for sensible temperature control – 24,715 cfm, 42°F
8. Construction – Block walls; drywall ceiling with vapor proofing; concrete floor on grade
9. Make-up air required – 2400 cfm
10. Air available for make-up – 50°F db/49°F wb; 50 gr/lb.

Problem

To determine the size of the dehumidifier needed in a controlled humidity and temperature area.

Moisture Load to be Computed

1. Permeation
2. Load through doors
3. Load through fixed openings
4. Population load

PERMEATION LOAD

$\frac{V}{C} \times \Delta G \times F_1 \times F_2 \times F_3 \times F_4$ = Grains per hour

V = 62.5’ × 55’ × 14’ = 48,125 cu. ft.
C = 14 = constant
ΔG = 110 (Ambient 130 gr/lb - room 20 gr/lb)
F_1 = 2.76 from Table II – Factor for moisture difference for 110 gr/lb
F_2 = 0.48 from Table III – for 48,125 cu.ft.
F_3 = 1.0 from Table IV – Frame masonry & frame construction
F_4 = 0.9 for vapor proof paint on walls & ceiling, untreated concrete floor

$$\frac{48{,}125}{14} \times 110 \times 2.76 \times 0.48 \times 1.0 \times 0.9 = 450{,}846 \text{ gr/hr}$$

DOOR LOAD

$O_{hr} \times \frac{A}{C} \times \Delta G \times F_1$ = grs/hr
O_{hr} = 6 openings/hr.
A = 6’ × 8’ = 48 sq. ft.
C = 7 = constant
ΔGr = 110 gr/lb
F_1 = 2.76

$$6 \times \frac{48}{7} \times 110 \times 2.76 = 12{,}491 \text{ gr/hr}$$

O_{hr} = 4 openings/hr.
A = 3’ × 7’ = 21 sq. ft.
C = 7 = constant
ΔG = 110 gr/lb
F_1 = 2.76

$$4 \times \frac{21}{7} \times 110 \times 2.76 = 3{,}643 \text{ gr/hr}$$

FIXED OPENINGS

$$\frac{A \times 300}{C \times D} \times \Delta G \times F_1 = \text{grs/hr}$$

A	=	area, 2.8 sq. ft.
300	=	Constant (vel. of vapor)
C	=	14 Constant
D	=	Depth of tunnel = 10'
DG	=	110 gr/lb
F_1	=	2.76

$$\frac{2.8 \times 300}{14 \times 10} \times 110 \times 2.76 = 1{,}822 \text{ gr/hr}$$

POPULATION LOAD

At a db of 55°F and working at a "light physical exertion" – 1100 gr/hr/person

10 people × 1100 gr = 11,000 gr/hr

TOTAL ROOM MOISTURE LOAD

450,846	gr/hr Permeation
12,491	gr/hr Door Load
3,643	gr/hr Door Load
1,822	gr/hr Fixed Opening Load
11,000	gr/hr Population Load
479,802	gr/hr TOTAL ROOM LOAD

The total room latent moisture load is 479,802 gr/hr, which is added into the calculation below to find the entering grain condition needed for the space.

$$\frac{\text{Total cfm}}{14} \times (S - G) \times 60 = \text{Room load (gr/hr)}$$

Total cfm = 24,715 cfm

14 = constant

S = 20 gr/lb. (condition of controlled space)

G = Unknown grain level needed entering space

60 = min/hr

$$\frac{24{,}715}{14} \times (20 - G) \times 60 = 479{,}802 \text{ gr/hr.}$$

G = 15.4 gr/lb

Thus the air to the room must be 15.4 gr/lb and the air mixture (return from the room plus the dehumidifier discharge) entering the main system fan should be 15 gr/lb. to allow for possible leakage into the system duct work. Here one must resort to trial and error techniques to select the dehumidifier size.

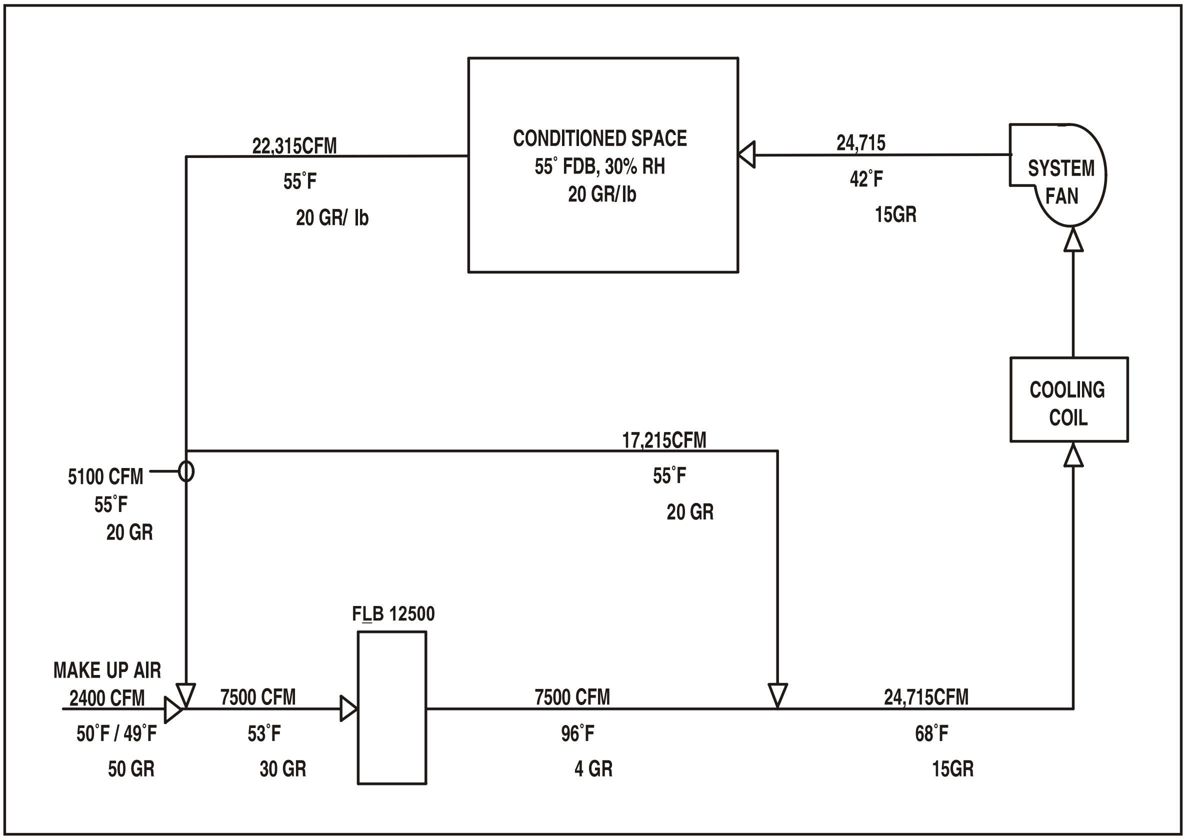
$$\frac{\text{cfm}}{14} \text{ x } (S – G) \text{ x } 60 = \text{X cfm – 7500 cfm – dehumidifier capacity (trial)}$$

14 = constant

S = 20 gr/lb. condition in the controlled space

G = 4 gr/lb. air leaving dehumidifier (Chart 1, Appendix *IV*) with entering air 53°F, 30 gr/lb.

$$\frac{7500}{14} \text{ x } (20\text{-}4) \text{ x } 60 = 514{,}285 \text{ gr/hr}$$



Note that the make-up air of 2400 cfm must mix with 5100 cfm of return air before entering the dehumidifier.

RECOMMENDATION

The FLB - 12500 Dehumidifier will satisfy the room load conditions when mixed with the remaining 17,215 cfm of return air and delivered into the conditioned space.

BRY-AIR DEHUMIDIFIER CALCULATION SHEET

PROJECT: EXAMPLE IV – CONTROLLED HUMIDITY AND TEMPERATURE AREAS

Conditions					Room Size	
Surrounding	95 FDB	77 FWB	130gr		62.5'L 55'W 14'H =	48,125 Ft^3
Design	$\frac{55\text{ FDB}}{40\ \Delta T}$	0 FWB	$\frac{20gr}{110\Delta GR}$	30RH	$\frac{48{,}125\ FT^3}{60}$ =	802

Permeation GR/HR

Volume	$\frac{48{,}125\ FT^3}{60} \times \frac{110}{(\Delta GR)} \times \frac{2.76}{(F_1)} \times \frac{0.48}{(F_2)} \times \frac{1}{(F_3)} \times \frac{0.9}{(F_4)}$	=	450,846
Door Load			
Openings/hr	$6 \times \text{Area } \frac{48FT^2}{7} \times \frac{110}{(\Delta GR)} \times \frac{2.76}{(F_1)}$	=	12,491
Openings/hr	$4 \times \text{Area } \frac{21FT^2}{7} \times \frac{110}{(\Delta GR)} \times \frac{2.76}{(F_1)}$	=	3,643
Fixed Opening			
	$\frac{2.8}{(Area)} \times \frac{300}{14} \times \frac{10}{(dep)} \times \frac{110}{(\Delta GR)} \times \frac{2.76}{(F_1)}$	=	1,822
People Load			
	$\frac{10}{(\text{No. People})} \times \frac{1100}{(F_5)}$	=	11,000
Product Load			
	0 gr/hr removed	=	0
Product Load			
	0 gr/hr added	=	0
	TOTAL ROOM LOAD	**=**	**479,802**
CFM required			
	$14 \times \frac{479{,}802}{60} \div (20 - 2)$	=	6,220 CFM
Make-up Air			
	$\frac{2400}{14}\text{ CFM} \times \frac{30}{(\Delta GR)} \times 60$	=	308,571
	TOTAL GR/HR	**=**	**788,373**

5,100 CFM RETURN AIR @ 55 FDB, 20 GR/LB

2,400 CFM MAKE-UP AIR @ 50 FDB, 50 GR/LB

= DEHUMIDIFIER INLET CONDITION:

CFM	7,500
DB TEMP.	53
GR/LB	30

Dehumidifier Sizing

$$14 \times \frac{788{,}373}{60} \div (30 - 4) = 7{,}075$$

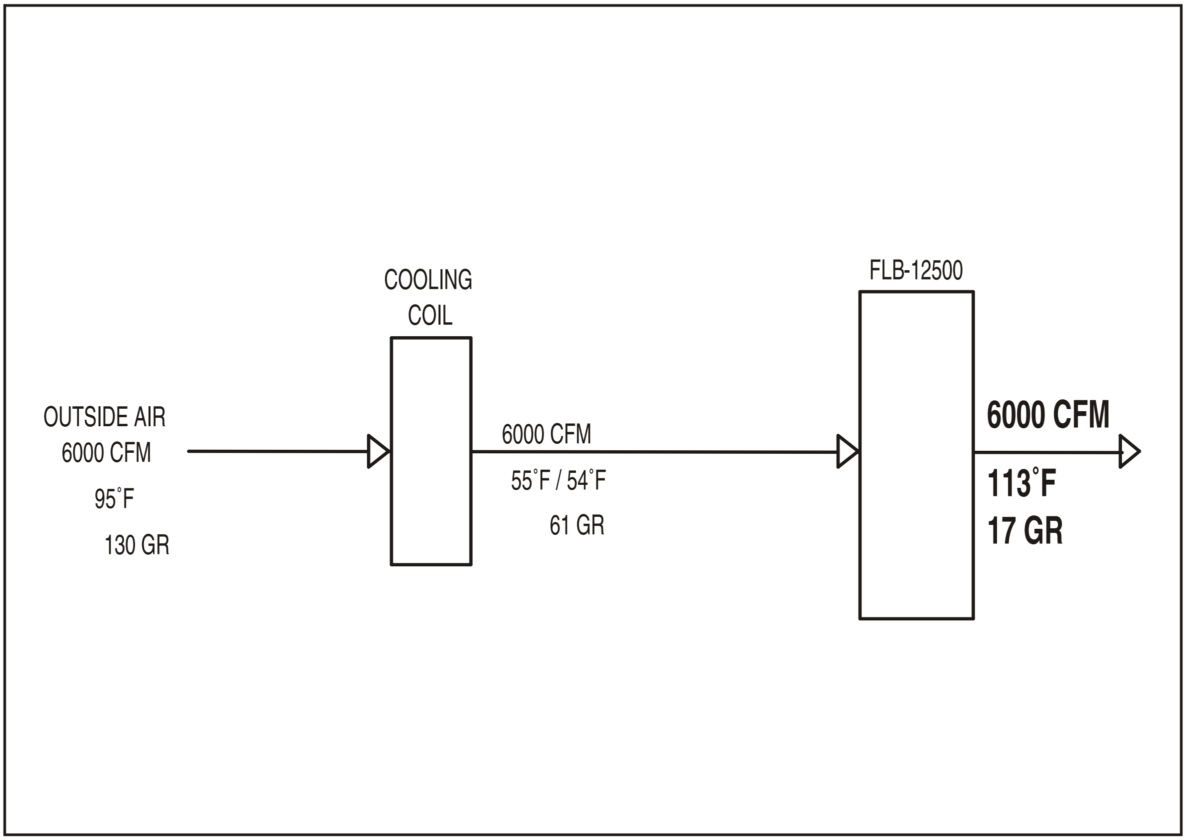
CFM Required

Proof

$$\frac{7{,}500}{14} \times (20 - 4) \times 60 = 514{,}285$$

System Capability

DEHUMIDIFIER REQUIRED	FLB - 12500
PROCESS OUTLET TEMP.	96° F



EXAMPLE V: PRODUCTION OF DRY AIR FOR A SPECIFIC PURPOSE

Many applications require a specific quantity of outside air to be delivered at a given moisture content and temperature. This requirement may be a need to make up air exhausted from a space or to supply air for a process such as a drying oven.

PHYSICAL FACTS

1. Maximum allowable moisture content – 17 gr/lb of dry air
2. Maximum allowable temperature – 115°F
3. Quantity of air required – 6,000 cfm
4. Maximum condition of outside air – 95°F, 130 gr/lb

From Chart 1, Appendix *IV*, it is obvious that 130 grain air cannot be reduced to 17 grains in a single pass through a dehumidifier, without other conditioning. Examination of Chart 1 shows that to produce 17 grains air leaving the dehumidifier, the inlet condition should be 64 grains or less at 60°F or less.

This is accomplished as shown above by installing a cooling coil upstream of the dehumidifier to reduce the temperature and moisture content of the outside air.

EXAMPLE VI: WATER TREATMENT PLANTS

In most water pumping stations, filteration plants, and waste water control plants, control of humidity in the pipe galleries, pump rooms, and control rooms is of prime importance. By reducing the dew-point temperature of the air below the temperature of the piping and walls, sweating and condensation can be eliminated. By circulating warm, dry air through the areas, water accumulation is avoided, maintenance for electrical controls, motors, and instruments is reduced, and paint lasts longer on the pipes, valves, and flanges.

A standard rule-of-thumb is used to approximate this type of application load:

$$\frac{\text{Volume of space to be conditioned}}{25} = \text{CFM dehumidifier}$$

(For each 25,000 cu.ft. space, supply 1,000 cfm of dry air.)

The use of an after-cooling coil for the dry air discharge from the dehumidifier can be omitted in most installations since the warm, dry air (low RH) will help heat the space during cool or winter conditions. Heat should not build up to an objectionable level because the large piping and wall areas are at the same temperature as the water in the system. Warm air also has the advantage of reducing the RH and increasing the air's capacity to carry away moisture.

ZERO LEAK SYSTEM FOR A LOW HUMIDITY SPACE

In a system where the ductwork and components are outside the controlled space, dry air leaving the system will induce the flow of humid air into the system. If the humid air is not dehumidified, each cfm will induce a load absorbing the capacity provided by 5 to 10 cfm of the dehumidified air. If all air that enters the system must pass through the dehumidifier, the additional load will be reduced by 50% or more. Typical air handling units (AHUs) are not built to be vapor tight. Standard sheet metal type ductwork has lapped seams that allow leakage. However, excellent silicone-based sealants are available; for applications requiring very dry spaces, the seams must be welded shut.

Having a "zero leak" condition means that all air leaving the controlled space is under positive pressure (to minimize infilteration), and all the entering air passes through the dehumidifier. The cost of dehumidification is high when moisture levels must be low. In these situations, even a small leak can double operating costs. Since ducts are a once-only expense, attention to ductwork is vital. The other approach to this situation-a higher capacity dehumidifier- means added costs, year after year.

It is unusual to find air handling units constructed for zero leak performance. Leaks are anticipated at removable panels, bearing flanges, drain pans, and through condensate drains with dry traps. Such units require additional sealing and check valves or positive water seals on the condensate lines. Cooling devices, especially the coils and fan, and the filter box need to be handled properly when cleaned, and they, too, must be tight. Obviously, there are many factors that ban restrict the zero leak principle.

One way to maximize the chances for a zero leak system is with air treatment equipment and dehumidifiers designed to fit and operate together effectively. Buying directly from the equipment manufacturer and not mixing sources of various equipment components focuses the accountability for moisture tightness.

Bry-Air constructs custom dehumidifiers and low dew-point AHUs for maximum efficiency. This extra care in design and construction means zero leak performance. Proper operating balance compensates for this situation.

When end users, consulting engineers, or other "specifiers" require a system to be substantially air tight, they means there is no leakage at any joint.

Assuring zero leak begins in the fabrication and construction stages. We recommend flanged or gasketed connections with welded seams and joints. As a minimum, all seams and joints should be caulked. Removable panels and access doors should be completely enclosed. (Coils will leak where return bends enter tube sheets.)

Prior to insulating, the system should be tested and deficiencies corrected. The best method is to use an open flame, if permissible. This requires candles or other sources of flame. In other situations, smoke devices or soap suds could be used.

When all these precautions in fabrication, construction, and testing are complete, the system will perform as designed. Periodic inspection during operation will allow leaks to be located and corrected.

PART SIX:
BRY-AIR DEHUMIDIFIERS FOR PRODUCT DRYING

Product drying applications include two general types: bulk drying and continuous drying. In bulk drying the material is loaded into a compartment and the entire load is dried as a batch. With continuous drying the wet material is continuously fed into a drying chamber and material continuously leaves the chamber, dried to the desired moisture level.

Drying potentials can be increased in two ways by:

Raising the product temperature by exposing it to heated air

Physically removing moisture from the surrounding air

The quantity of air needed for proper drying will vary widely with either type of drying system. But the drying characteristics and the approach to the problem are similar.

The Bry-Air Dehumidifier performs no miracles extracting moisture from the product into the surrounding air. But by maintaining the air at a lower moisture level, the Dehumidifier can increase the drying potential and the drying rate. More important, it can remove the variable of weather as a factor in a drying operation.

Heating is less expensive than drying, so the obvious questions is: Where do Bry-Air Dehumidifiers apply?

In most drying processes, the released moisture goes into the air and must be physically removed or diluted with outside air. However, without a desiccant dehumidifier, the lowest possible moisturelevel in the chamber will equal that of the outside make-up air. But in practical terms, the moisture content of the air in the chamber will generally be somewhat higher than that of the outside air.

When heat is used alone, the drying potential is limited by the specific humidity of the outside air plus the safe temperature to which the product can be raised. Generally, a proper drying potential can be established with heat and outside air if the temperature can be raised to 140°F or above. If the temperature can not be raised over 120°F, then a Bry-Air Dehumidifier is the best solution. For temperatures in the 120°F to 140°F range, the decision depends on the product characteristics and the desired degree of dryness.

Drying operations involve the removal of free moisture, hygroscopic moisture, or a combination of both. Free moisture is water held on the surface or between molecules of a substance. Free moisture occurs when actual liquid water is used to mix or wash the product prior to drying. Hygroscopic moisture is held within the material's cells. Hygroscopic moisture will take up or dispel water in relation to the relative humidity of the air mixture to which it is exposed. When in equilibrium with air at 100% RH, the material will be hygroscopically saturated. Any hygroscopic material containing free moisture must be hygroscopically saturated.

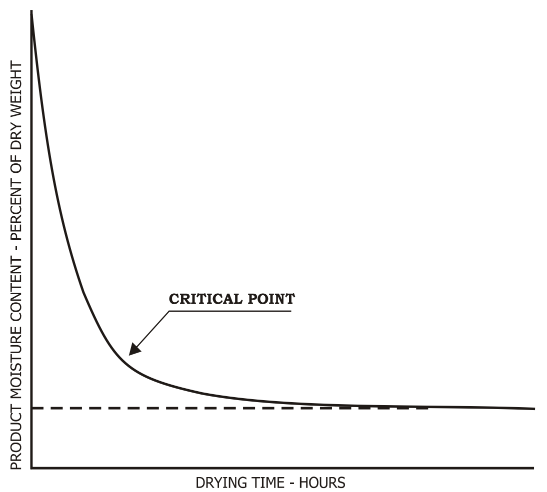
The removal of free water is a surface evaporation function and follows the calculation shown in Appendix *III* of this manual. The surface water temperature should be assumed to be the wet-bulb temperature of the surrounding air mixture. Note that air velocity is critical the drying speed.

The removal of hygroscopic moisture depends on the relative humidity difference between that of the products' equilibrium condition and that of the surrounding air. Velocity of the air over the product has little or no bearing on the drying speed.

The figure Fig 1 shows a typical drying curve. The sudden change in drying rate (at the critical point) denotes where the initial drying via removal of free moisture ends and hygroscopic drying takes over. In other words, the product has lost its free moisture, but is still hygroscopically saturated.

Each material has a different physical form that determines how it holds or gives up moisture. Since many of the newer materials lack published data on their drying rates, selecting appropriate air drying equipment must be done experimentally. The net effective drying surface and the hygroscopic properties cannot be determined in any other way.

Most drying problems are really a request for improving the speed or quality of an existing



Typical Drying Curve
Figure 1

drying operation. For example, before today's advanced dehumidifying equipment was available, candy manufacturers could make their product only in winter. In summer, attempts to manufacture candy might often end with a moldy product. Now, to meet production demands, the use of cooling equipment and a desiccant dehumidifier can imitate winter conditions all year.

Solving a drying problem usually involves a rather simple analysis of the drying cycle. If the analysis (that is, the test run) can occur during weather conditions that consistently give the desired drying result, the problem is simplified. Regardless, any test run will show the product's characteristics and give clues for solving the problem.

The test run should be made under actual production operation to secure information in either of the two following categories.

Bulk type drying system. Several trays in different locations in the compartment should be weighted and identified before being placed in the drying cabinet. They should be weighed at the start and at predetermined intervals (usually hourly), subtracting the tray weight, and quickly returning the tray to its original position after weighing. At the same time a wet-and dry-bulb reading (average throughout the cabinet) and air velocity reading over the product should be taken. Continue these procedures until the product in satisfactorily dried; weight should be noted at this point. The purpose is to establish a totally dry weight. Temperature should be high enough to keep the RH in the surrounding air at 5% or less.

Continuous type drying system. Here one must remove material samples at the start, finish, and at regular intervals along the drying tunnel. Such test points should be accurately marked and related to the drying time. Each sample should be weighed as soon as removed, then thoroughly dried at elevated temperature and reweighed. The dry-bulb temperature, wet-bulb temperature, and air velocity over the product should be determined at each point of product supply as well as at the start and end of the drying tunnel.

From this information the weight readings can be converted into% of moisture and plotted against drying time. Moisture content should be expressed as a percentage of the product's bone dry weight, not as a percentage weight of the test sample. If both free and hygroscopic water are removed from the sample, a characteristic curve will resemble that shown in the figure 1 on page 31.

SIZING THE DESICCANT DEHUMIDIFIER

Bulk type drying. On the characteristic curve, indicate the wet-bulb and dew-point temperatures equivalent to the reading taken during the test up to the critical point. From the critical point to the curve's end, show the dry-bulb temperature and the RH. The hygroscopic drying phase should be considered in making the first analysis (that is, the drying curve).

Some hygroscopic moisture (near the product's surface) is removed at the critical point. So make two assumptions:

The product is hygroscopically saturated at this point.

The product is substantially in equilibrium with the final RH at the end of the test (when it reaches the desired moisture content).

Thus, the average drying potential for this part of the test is

$$\left[\frac{90 + \text{final rh}}{2}\right] - \text{average RH observed during the hygroscopic drying portion of the test.}$$

If our test took 12 hours and we want it to be complete in 8 hours, or two-thirds the amount of time, then the hygroscopic portion of the test, which took 9 hours, needs to be completed in 6 hours. Further, the product's moisture level at the critical point minus the moisture remaining after complete drying equals the total weight of water to be removed in 6 hours.

This amount can be converted into grains per minute:

$$\frac{\text{Total hygroscopic water removed in lbs}}{\text{Time in hrs}} \times 116.6$$

To accomplish this dryness faster, the drying potential must be increased proportionately to the rate of test time vs. the desired time. But the product's average moisture level will be unchanged. Therefore, the average RH for drying is found by:

Average product RH – Required RH potential.

The average product RH combined with the average dry-bulb temperature dictates the specific humidity that must be maintained and defines the operating conditions for the Desiccant Dehumidifier.

The drying temperature should be as high as practical (usually 10°F below the maximum allowable product temperature). 95°F entering air is the highest recommended level (See Chart 1, Appendix *IV*). Thus if temperatures greater than 95°F are needed in the drying chamber, the recirculating air should be cooled to 95°F or below. Here the cost of the cooling coil, booster fan, and water used will be offset by the gain in moisture removal capacity. (Reduced ratings for inlet temperatures up to 115°F can be calculated. See your Bry-Air representative for details.)

The Bry-Air Dehumidifier will handle a mixture of recirculated air (at the average specific humidity already determined) and a minimum of 5% outside air (refer to Typical Flow Diagram). This establishes the level at which the dehumidifier must operate. From Chart I Appendix *IV*, the leaving moisture is determined. The difference between the gr/lb. moisture level maintained in the dehumidifier and the same parameter in the air leaving the dehumidifier is the pick-up factor. This figure divided into the average required moisture removal in gr/min determines the dehumidifier size in lb./min air capacity.

This unit capacity must be checked against the "free moisture" requirement this way:

$$\frac{\text{Total weight of free moisture (gr/min)}}{\text{Dehumidifier air capacity (lb/min)}}$$

$$= \text{moisture pickup factor (gr/lb)}$$

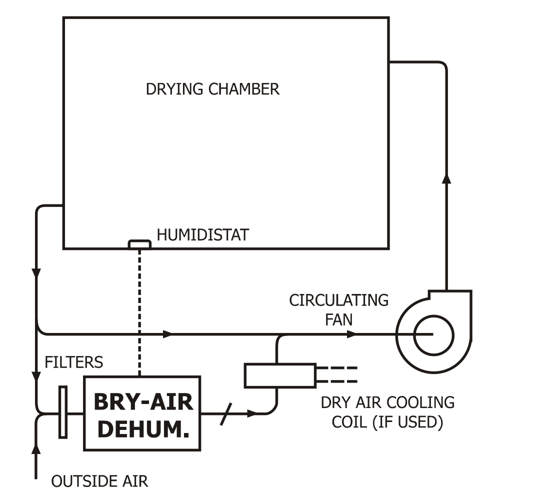
Refer to Chart 1, Appendix *IV*. Knowing the desired drying temperature helps pinpoint the inlet where the necessary moisture removal will occur. For example, if 42 gr/lb. must be removed (the drying temperature is 95°F), then follow the 95°F curve to the point where the difference between the inlet and outlet moisture is 42 gr/lb. Here the result is nearly 60 gr/lb.; that is where the leaving moisture is approximately 18 gr/lb.

To be safe, use a condition approximately 5 gr/lb. above that shown on the curve and allow for 5% outside air. Then the needed dew-point and wet-bulb temperature values can be established. Determine the vapor pressure equivalent for the temperature using Appendix *III*. The difference between these items establishes the drying potential. Then determine the average vapor pressure difference for the test run from the same table using the test dew-point and wet-bulb readings. The ratio of vapor pressure difference with the Bry-Air Desiccant Dehumidifier over that measured during the test should equal or be greater than the ratio of the drying time. (test vs. desired.)

Since air velocity also affects free moisture evaporation, drying can be some what controlled by changing air velocity to as high a level as possible without disturbing the product. Use a by-pass or fans within the chamber to increase the total circulation in the drying air circuit above the Bry-Air Dehumidifier's capacity.

For the test run, establish a velocity factor:

$$\frac{\text{(1 + test velocity in ft/min)}}{230^*}$$



Typical Flow Diagram for Bulk Dryer
Figure 2

* An established constant.

Also establish a velocity factor for the actual design:

$$\frac{\text{(1 + actual velocity in ft/min)}}{230}$$

At any given vapor pressure difference, the evaporation will vary directly according to the above factors.

Continuous drying. Since continuous drying systems characteristically have open ends, they usually require a great deal of additional outside or make-up air to make up for all the openings. For efficiency, keep such openings as small as practically possible. A minimum leakage equivalent to a 200 fpm velocity through the area should be positively introduced into the system.

The typical flow pattern for a continuous drying operation, has a separate circulating system for free moisture removal; the dehumidifier discharge is directed through the hygroscopic moisture phase. This configuration takes advantage of rapid circulation in the first space without carrying the wetter air into the final drying space. Note the separate circulating system for the free moisture removal stage. Dehumidifier discharge is directed through the hygroscopic moisture phase. This arrangement allows rapid circulation in the first space without carrying over moisture into the final drying space.

Use the bulk drying method to establish the vapor pressure difference to allow drying to proceed satisfactorily in the free moisture stage. Keep the velocity and temperature as high as practical. Design specifications will help establish the total circulation. The temperature plus necessary vapor pressure difference will establish the specific humidity (in gr/lb), that must be maintained.

Express the total product moisture removal in gr/min and add the moisture load introduced by make-up air. This latter load derives from the difference in specific humidity between the maximum design outdoor level and that maintained in the compartment multiplied by the quantity of outdoor air (in lb./min). The proportionate quantity of recirculated and outside air also determines the specific humidity of the mixture- which typifies air entering the Dehumidifier.

If the pre-cooling Bry-Air Desiccant Dehumidifier is used, the air temperature leaving the coil determines the Dehumidifier operating level. Refer to Appendix *IV*, Chart 1 to calculate the moisture level leaving the Dehumidifier and determine the removal per pound value;

$$\frac{\text{Total moisture removal load, (gr/min)}}{\text{Moisture removed by Dehumidifier (gr/lb)}}$$

= Dehumidifier size lb/min air capacity

The next step is to check the performance of the Dehumidifier in the hygroscopic drying section, measured in gr/min.

$$\frac{\text{Dehumidifier performance (lb/min)}}{2}$$

+ moisture level of air leaving Dehumidifier

= average moisture content of air in this portion of the drying chamber

Use the curve in Appendix *IV*, Chart 2 to determine the temperature of air leaving the Dehumidifier. The departing moisture will have a cooling effect; to find the average temperature subtract 0.625°F for each gr/lb. pickup.

Now that moisture content and temperature are known, the average RH needed is easily determined from a psychrometric chart. Compare this figure with the necessary RH to insure proper drying within the bulk type dehumidifier. This comparison will reveal whether or not the dehumidifier has the capacity to produce the desired drying rate.

Maintaining drying temperature. As already noted, water evaporation is a cooling process. Approximately 1052 Btu are needed to evaporate one pound of water. In other words, 6.65 gr/lb. represent 1 Btu or 1 gr/lb. represents 0.625°F.

As a product is dried, it releases moisture. Without proper control, this moisture can cool down the environment and result in an equilibrium condition where the drying practically ceases. So to maintain drying temperatures, heat must be supplied in an amount represented by the evaporation rate. Also, heat can be lost by conduction through cabinet walls. Thus it may be necessary to control the product to drying temperature (heat or cool it), and heat the make-up air to maintain the optimal drying temperature.

The process of adsorption is an exchange of heat in a like amount in the reverse direction. Thus, air heats as it passes through the desiccant dehumidifier. Approximately 30% additional heat builds up in the desiccant from the previous reactivation period, so the dehumidifier supplies all the required heat for evaporation and an additional 30% for other purposes. In some, instances that additional heat is required; in other cases, cooling may be needed.

DEHUMIDIFIER CAPACITY CONTROL

Several methods provide dehumidifier control.

On/Off control of the dehumidifier. Humidistat or dew-point control monitoring of space or return air is a method used where continuous process air is not needed. Often the dehumidifier is installed as an independent unit and is not tied into the make-up or outside air circulation system.

On/Off control of reactivation heaters and blower. This control method applies to continuous process air flow situations. However, the process air will have more variation in humidity than with other control methods.

Modulation of reactivation inlet temperature.

This strategy yields reduced energy consumption and supplies the minimum energy needed to maintain the process condition.

Modulation of reactivation inlet temperature and air volume. By modulating the reactivation air volume and temperature at specific values, the reactivation capability is increased and can be used over a wide range of operating conditions. This method also compensates for reductions in adsorption capacity.

Process face and bypass damper control. Here the moisture control of leaving air is due to varying the volume of air that bypasses the dehumidifier. However, a constant supply air volume must be maintained. This is the best scenario for tight humidity control.

CONCLUSION

Information in this manual was prepared to help customers choose the most effective hand efficient dehumidifiers. Please contact Bry-Air's dehumidifier experts for additional assistance and for more detailed information about physical characteristics and performance data relating to Bry-Air Dehumidifiers.

Bry-Air procuced systems that are built to meet specific customer needs.
Many customers require heating, cooling, air filtration, etc, in addition to air drying.

APPENDIX I: PROCESSES AND PROPERTIES OF AIR

Dry-bulb, wet-bulb, dew-point temperatures, relative humidity-these terms are so closely related that if two properties are known, all others shown in the figure 3 may be read from the chart. When air is saturated, dry-bulb, wet-bulb, and dew-point temperatures are identical. (See Example 2.)

Enthalpy of air for any given condition is the enthalpy at saturation corrected by the enthalpy deviation due to the air not being in a saturated state. The enthalpy (h) in Btu per pound of dry air is the enthalpy at the saturation h_{wb} plus the enthalpy deviation h_d See Example 2.

$H + h_{wb} + h_d$

If the air's moisture content increases or decreases in a psychometric process, the heat added (q) or removed (-q) is the difference between the enthalpy of the final or leaving air h_{la} and the initial or entering air h_{ea} minus the enthalpy of the moisture (water in liquid or ice state) added h_w or rejected h_w.

$q = h_{la} - h_{ea} - h_w$

See Examples 4 and 5.

The enthalpy of added or rejected moisture is shown in the small graphs at the top of the chart.

Enthalpy of added or rejected moisture and enthalpy deviation are usually omitted in applications not requiring precise results- for example, comfort air conditioning. Errors due to omissions for wet-bulb temperatures below 32°F is much larger than for omissions above 32°F.

Sensible heat factor. This is part of certain calculations for installing air conditioning equipment. A scale along the right side of the figure in Example 4 used with an origin at 80°F dry-bulb temperature and 50% RH provide a reasonable heat factor value. See Example 4.

Barometric pressures. In comfort air conditioning, a mercury reading of one inch or less either above or below the standard 29.92 inches of mercury is considered a standard reading.

When dry-bulb and dew-point temperatures are known for air at non-standard barometric pressures, values of % RH and grains of moisture per cubic foot are correct on a standard chart. But for given dry-bulb and wet-bulb readings at non-standard barometric pressures, all properties must be corrected.

INTERPRETING THE AIR CONDITIONING CHARTS

Generally, in graphic presentations, humidifying is shown by an upward line and dehumidifying is shown by a downward line.

Heating and cooling air without changes in moisture content involve only a change in sensible heat and appear as a horizontal line, to right or left respectively. Changes occur in dry-bulb, wet-bulb, RH, and enthalpy. Specific humidity and dew-point temperature remain constant.

In *heating and humidifying*, both sensible heat and specific humidity increase-shown as a line sloping upward and to the right. Changes occur in dry-bulb, wet-bulb, dew-point temperatures, and enthalpy. A difference in RH depends on the slope of the line.

For *cooling and dehumidifying*, both sensible heat and specific humidity decrease, so the line slopes downward and to the left. Dry-bulb, wet-bulb, dew-point temperatures, and enthalpy all change. Changes in RH are dependent on the slope of the line.

Evaporative cooling refers to air brought in contact with spray water at a temperature equal to the wet-bulb temperature of the air. The process takes place upward along the wet-bulb line. As sensible heat of the initial air vaporizes the water, the air's dry-bulb temperature falls. The sensible heat used to vaporize the water enters the air as latent heat in added vapor; thus no heat is added or removed. Wet-bulb temperature remains constant. Dew-point temperature, RH, specific humidity, and enthalpy increase. (In most evaporative cooling installations, heat may be added or removed during the process due to outside sources, this amount is usually negligible.)

In chemical dehydration, the air that contacts the chemical either adsorbs or absorbs moisture from the air. Thus in this energy constant process, heat is liberated and added to the air-

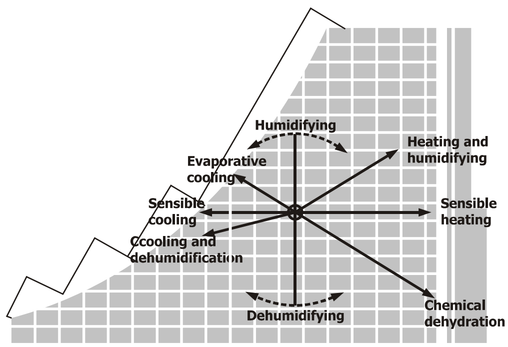


Fig. 1. Air Conditioining Processes

and this amount is basically equal to the latent heat of vaporization of the moisture removed. Indicated by a downward sloping line approximating the wet-bulb line, the slope of the chemical dehydration line may be either slightly greater or less than the wet-bulb line, depending on if heat is stored, liberated, or absorbed.

AIR CONDITIONING PROCESSES such as heating, cooling, humidifying and dehumidifying may be shown graphically on the chart. See Figure 1.

EXAMPLE 1. Reading Properties of Air

Given { DB = 70°F; WB = 60° F }

Find: { % RH; DP; Volume; GR of moisture per lb dry air; GR of moisture per cu ft }

Locate point of intersection on the chart of vertical line representing 70°DB and oblique line representing 60°WB. All values are read from this point of intersection.

Interpolate between relative humidity lines on 70°DB line, read RH = 56%.

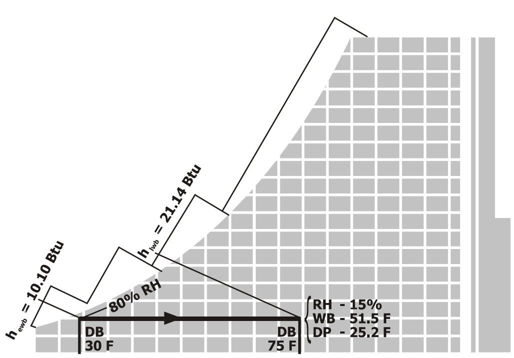


Fig. 2. Heating Process

Follow horizontal line left to saturation curve, read DP = 53.6°F.

Interpolate between lines representing cubic feet per pound of dry air, read v = 13.53 cu ft.

Follow horizontal line to right, read grains of moisture per pound of dry air, W=61.4 gr.

Grains of moisture per pound of dry air (61.4) divided by cubic feet per pound of dry air (13.53) = 4.54 gr per cu ft

EXAMPLE 2. Reading properties of Air

Given: { RH = 50%; WB = 60°F }

Find: { DB; DP; Gr. of moisture per lb. dry air; Enthalpy }

Locate point of intersection on the chart of 50% RH line and oblique line representing 60°WB. All values are read from this point.

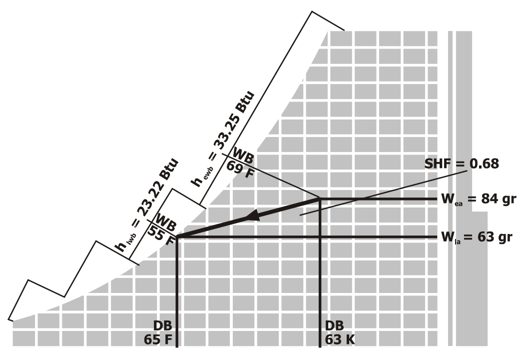


Fig. 3. Cooling and Dehumidifying

Follow vertical line downward to dry-bulb temperature scale, read DB = 71.9°F

Follow horizontal line left to saturation curve, read DP = 52.3°F.

Follow horizontal line to right, read grains of moisture per pound of dry air, W = 58.4 gr.

Follow wet-bulb line to "Enthalpy at saturation" scale and read H_{wb} = 26.46 Btu. Read enthalpy deviation for point of intersection h_d = -.08 Btu. Enthalpy of air at given condition h= h_{wb} + h_d = 26.46 + (-.08) = 26.38 Btu per lb of dry air.

EXAMPLE 3. Heating process

(no change in moisture content)

Given { Initial Air { DB = 30°F; RH = 80% }; Air heated to 75° DB }

Find { Final Air { %RH; WB; DB }; Heat Added }

Locate the condition initial air on the chart.

Follow horizontal line to 75° DB.

Read:RH=15% : WB=51.5°F : DB=25.2°F.

Exact Solution – Head added:

Read enthalpy at saturation initial air h_{cwb} = 10.10 Btu

Read enthalpy deviation initial air h_{cd} = 0.06 Btu

Enthalpy of initial air $h_{ca} = h_{cwb} + h_{cd} = 10.10 + .06 = 10.16$ Btu

Read enthalpy at saturation of final air h_{lwb} = 21.14 Btu

Read enthalpy deviation of final air h_{ld} = 0.10 Btu

Enthalpy of final air $h_{la} = h_{lwb} + h_{ld} = 21.14 + (-0.10) = 21.04$ Btu

Heat added $q = h_{la} - h_{ca} = 21.04 – 10.16 = 10.88$ Btu per lb of dry air

Approximate Solution – Head added:

$q = h_{lwb} - h_{cwb} = 21.14 – 10.10 = 11.04$ Btu per lb of dry air.

The approximate solution is 1.5% higher than exact solution.

EXAMPLE 4. Cooling and Dehumidifying Process

(a) Moisture rejected as water condensate

Given:
- Initial Air: DB = 83°F, WB = 69°F
- Final Air: DB = 56°F, WB = 55°F
- Condensate rejected at 55°F

Find: Heat Removed; Sensible Heat Factor

Locate initial and final conditions of air on chart.

Read:
h_{cwb} = 33.25 Btu
h_{lwb} = 23.22Btu
h_{cd} = -0.12 Btu
h_{ld} = -0.01 Btu
$h_{ca} = 33.25 + (-0.12) = 33.13$ Btu
$h_{la} = 23.22 + (-0.01) = 23.21$ Btu

Read grains of moisture in initial air W_{ca} = 84

Read grains of moisture in final air W_{la} = 63 w = $W_{la} – W_{ca} = 63 – 84 = 21$ (moisture rejected)

Read enthalpy of rejected moisture (h_w) from diagrams at top of chart for 21 gr grains and 55 F = -0.8 Btu.

Exact Solution – Heat removed:

$q = h_{la} – h_{ca} – h_w = 23.21 – 33.13 – (0.08) = -9.84$ Btu per lb dry air.

Approximate Solution – Heat removed:

$q = h_{lwb} – h_{cwb} = 23.22 – 33.25 = -10.03$ Btu per lb dry air.

Approximate solution is 1.9% higher than exact solution.

To determine Sensible Heat Factor, draw a line between initial and final conditions. Draw a line parallel to this line from reference point (80 DB, 50RH) to Sensible Heat Factor scale, read SHF = 0.68.

EXAMPLE 5. Mixture of Air

Given:
- Inside Air : 3 Parts by weight — DB = 75°F, WB = 62°F
- Entering Air : 1 Part by weight — DB = 95°F, WB = 75°F

Find: Properties of Mixture

Locate on chart conditions of inside and entering air. Draw line connecting two points. Measure off distance equal to ¼ of line, starting from inside air condition. Point thus established represents condition of mixture of inside and entering air.

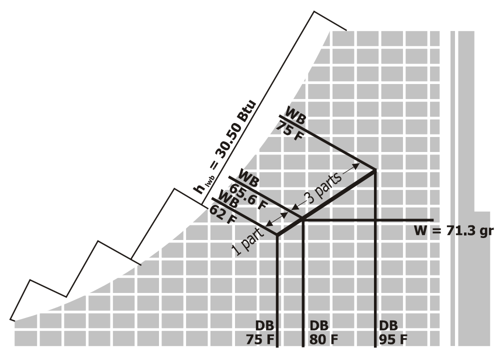
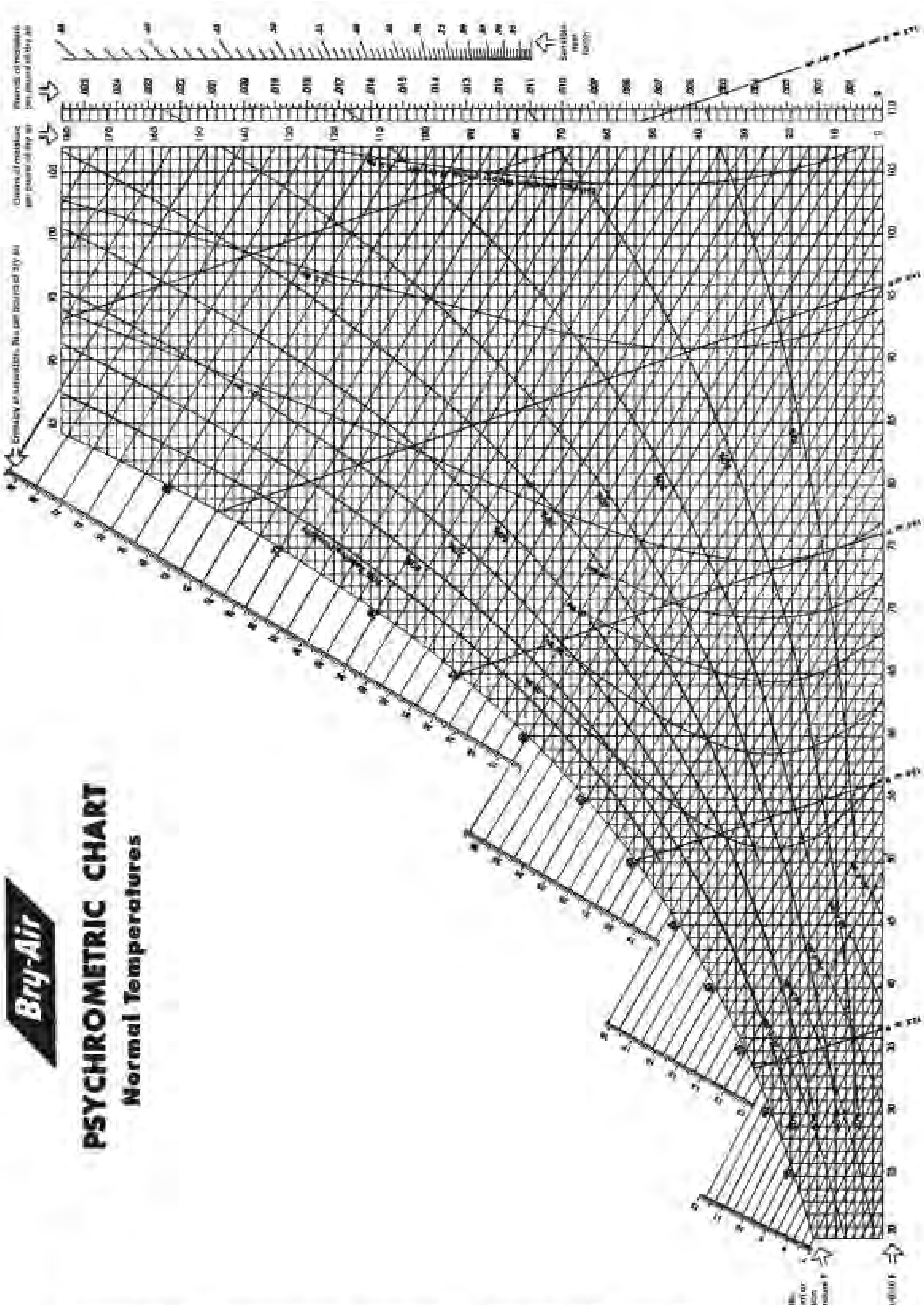


Fig. 4. Mixture of Air

Read properties of mixture:

DB = 80°F, WB = 65.6°F, h = 30.50 + (-0.11) = 30.39 Btu Moisture content (W) = 71.3 gr per lb of dry air.

When air quantities being mixed are at widely different temperatures, the above method is slightly in error. For exact solution calculate properties of mixture on basis of specific humidity and enthalpy.


Bry-Air
PSYCHROMETRIC CHART
Normal Temperatures

APPENDIX II
DEHUMIDIFIER SURVEY SHEET

Bry-Air **DEHUMIDIFIER SURVEY SHEET**

DATE ____________________
PRO. NO. ____________________

CUSTOMER ____ PHONE ______ / __________
ADDRESS ____ CONTACT ____________________
APPLICATION ____
DATE PROPOSAL REQ'D ____________________ DATE EQUIPMENT REQ'D ____________________
INSTRUCTIONS ____
____
____

CONDITIONS

	FDB	FWB	GR/LB	%RH
AMBIENT				
SURROUNDING				
DESIGN				

ROOM SIZE

____ L _____ W _____ H __________ FT'

CONSTRUCTION

	MATERIAL		TYPE V P
WALLS			
WALLS			
CEILING			
FLOOR			

INSULATION

	TYPE	THICKNESS	U FACTOR
WALLS			
CEILING			
FLOOR			

DOORS

NO.	TYPE	WIDTH	HEIGHT	OPENINGS / HR

FIXED OPENINGS (CONVEYORS, WINDOWS, & ETC.)

NO.	WIDTH	HEIGHT	DEPTH	USAGE

PEOPLE

NO.	TYPE WORK (LT. MOD. HEAVY)

PRODUCT LOAD

lbs / hr OR gr/hr REMOVED ____________________

EXHAUST AIR

CFM	CONTAMINATES

OTHER MOISTURE LOADS

TYPE	SQ. FT.	F°	BTUH

INTERNAL SENSIBLE LOADS

LIGHTS	WATTS
MOTORS	DRIVEN IN OUT H.P.
OVENS	BTUH
MATERIALS	F° S.H. LBS/HR
OTHER	

SERVICES AVAILABLE

ELECTRIC	________ VOLTS ___ PH _____ Hz
STEAM	________ PSIG
HOT WATER	_______ F° ____ GPM _______ PSIG
GAS	________ CFH ________ BTUH
WATER	_______ F° ____ GPM _______ PSIG
REFRIGERATION	________ TYPE ____________ BTUH

MAKE - UP AIR

CFM	FDB	FWB	gr / lb

FILTERS REQUIRED

TYPE	TYPE	%
RETURN AIR		
PROCESS AIR		
REACT AIR		

STATIC PRESSURES

__________ OUTLET __________ INLET ______ EXT

APPENDIX III:
DETERMINING MOISTURE OR LATENT LOADS

Moisture load can come from many sources, which provide the data needed to calculate the total latent load on any air conditioning or drying system. The total latent load equals the sum of applicable individual loads.

Outside design level. Bry-Air Dehumidifier performance characteristics are expressed in terms of specific humidity or grains per pound of air. To determine the outside design moisture level, use the standard design dry-bulb and wet-bulb conditions because this value measures the design total heat (wet-bulb) occurring with the highest practical dry-bulb. The design moisture level will exist when a lower dry-bulb occurs with the design wet-bulb. This condition represents the same total heat, but a higher specific humidity. The table below lists the recommended design specific humidity for various design wet-bulb temperatures. Use the standard accepted design wet-bulb for your locality.

EXAMPLE: If the accepted design level for your city is 95°F db (dry-bulb temperature) and 76°F wb (wet-bulb temperature), this condition equals 104 gr/lb. But there will be many days when 76°F wb. will occur at a lower dry-bulb temperature. From the table below, the proper design specific humidity for comfort would be 115 gr/lb; for industrial work it would be 125 gr/lb. Figures below assume that these levels will be reached or exceeded on 30% of summer days for comfort work and 10% of days process work.

Ventilation latent load. Determining the latent load equivalent to the outside air by subtracting the indoor or maintained specific humidity and multiplying that amount by the pounds of outside air brought into the system.

EXAMPLE: If 1,000 cfm ventilation air is at 125 gr/lb. design and the design inside condition is 70 gr/lb., what is the ventilation latent load?

$$\frac{1000}{14} \times (125 - 70) = 3930 \text{ gr/min or } 235{,}800 \text{ gr/hr.}$$

The average density of air is given as 14 cu.ft. per pound of air and is used regardless of the actual density at design conditions.

RECOMMENDED DESIGN OUTSIDE MOISTURE LEVEL

Design Outside Wet Bulb	Design Specific Humidity	
	Comfort Work	Process Work
F	(gr/lb)	(gr/lb)
81°	139	149
80°	130	143
79°	125	139
78°	120	134
77°	118	130
76°	115	125
75°	112	121
74°	108	117
73°	105	113
72°	100	109
71°	95	106
70°	90	102

LATENT HEAT DISSIPATED BY ADULT OCCUPANTS

Dry Bulb Temperature	Occupants At Rest	Occupant Doing Light Physical Exertion*	Occupant Doing Heavy Physical Exertion**
F	(gr/hr)	(gr/hr)	(gr/hr)
60°	400	1300	1960
65°	530	1630	2400
70°	670	2060	2920
75°	900	2540	3450
80°	1180	3040	3950
85°	1525	3550	4450
90°	1870	4000	5000

* Examples – Waiters, dinner dancing, light factory assembly work

** Examples – Factory machine operator, continuous dancing

Evaportion from a wetted surface. Determine the amount of moisture evaportion from a pan, tank or other wetted surface into a space using the following calcutions:

Where:

Gr. = moisture evaporated in grs/hr.

Vel = air velocity in F.P.M.

V_L = vapor pressure equivalent to temperature of surface water- inches of mercury.

V_A = vapor pressure equivalent to dewpoint temperature of air over surface – inches of mercury.

If air is moving across surface: $Gr. = 650 \times (1+\frac{Vel}{230}) \times (V_L - V_A) \times (\text{sq ft of surface}) \times \text{Activity factor}$

If air is impacting surface: $Gr. = 1350 \times (1+\frac{Vel}{250}) \times (V_L - V_A) \times (\text{sq ft of surface}) \times \text{Activity factor}$

TYPE OF POOL	ACTIVITY FACTOR	
Residential	0.5	
Condominium	0.65	
Therapy	0.65	
Hotel	0.80 (Maximum)	
Public Schools	1.0 (Maximum)	
Whirlpools Spa	1	
Wave Pools	1.50 (Minimum)	
DESIGN CONDITIONS		
TYPE OF POOL	AIR TEMP °F	WATER TEMP °F
Recreational	75 to 85	75 to 85
Competition	78 to 85	76 to 82
Therapeutic	80 to 85	85 to 95
Diving	80 to 85	80 to 90
Whirlpools Spa	80 to 85	97 to 104

VAPOR PRESSURES OVER WATER

Temperature Degrees F	Vapor Pressure Inches Mercury	Temperature Degrees F	Vapor Pressure Inches Mercury	Temperature Degrees F	Vapor Pressure Inches Mercury
30	0.1663	65	0.6222	100	1.933
35	0.2035	70	0.7392	110	2.60
40	0.2478	75	0.8750	120	3.45
45	0.3004	80	1.032	130	4.53
50	0.3626	85	1.213	140	5.88
55	0.4359	90	1.422	150	7.57
60	0.5218	95	1.660	160	9.65

MISCELLANEOUS MOISTURE LOADS

Description	Load Grs/Hr
Food (Per meal)	200
Steam Table (per sq ft top)	2,000
Coffee Urn, 3 gal. steam or electric	10,000
Coffee Urn, 3 gal. gas	15,000
Coffee Urn, 5 gal. steam or electric	17,000
Coffee Urn, 5 gal. gas	26,000
Hair Dryer, electric, per helmet	2,700
Hair Dryer, gas, per helmet	4,000
Unvented gas burners (nat. or mfg. gas) per 1000 Btu. input	650

Moisture permeation. This is discussed in detail in Part four.

Moisture loads in the table above represent unvented appliances. Although personal judgement is used to determine vent or hood efficiency, the hood efficiency should never be higher than 50%.

Drying hygroscopic materials. The calculations shown above apply only to evaporation of free water from a surface. When hygroscopic materials are in the first stages of drying-when the surface is actually wet-then the above relationship may exist. But after surface drying is complete, further drying will occur at a rate that depends on the rate of diffusion within the material; the rate varies with the degree of dryness within the material and is based on expected structural changes that occur during the drying process.

Establish the drying rate of hygroscopic materials in order to establish the hourly moisture load. Unfortunately these rates must be determined experimentally in each situation.

Usually, the desired outcome with hygroscopic drying is to improve drying rate or degree of dryness in the final product within an existing set up or with the addition of a Dehumidifier. In doing so, the desired drying period is generally included with the total weight of material to be handled.

Wt. of material entering – Average drying rate

$$= \frac{\text{Wt. of material leaving}}{\text{Drying time (hrs)}}$$

One caution here; the drying period cannot be arbitrarily assumed; it must be realistic. For example, if dry air circulates in a dehumidifier and cannot dry a material totally within 2 hours, then 2 hours will be neither a possible nor a realistic desired drying time.

Storage of hydroscopic materials. When hygroscopic materials enter a dry storage space, even for a short time, they contribute a moisture load that must be absorbed by the dehumidifier. The table (Page 43) lists the moisture holding capacity of various materials in equilibrium with air at the relative humidities shown. The % compare the moisture to the substance's totally dry weight.

If the incoming material has an unknown moisture content, assume that it is in equilibrium with 60% RH air. In winter, the materials will likely come into a room in equilibrium with 90% RH air. However, in winter most other sources of RH are lower, so the summer figure (60%) can be used all year, unless the product loads makes up most of the entire total and the permeation load is minor by comparison.

MOISTURE REGAIN OF VARIOUS HYGROSCOPIC MATERIALS

Moisture Content Expressed in Per Cent to Dry Weight of the Substance at Various Relative Humidities - Temperature, 75°F (24°C)

Classification	Material	Description	RELATIVE HUMIDITY - PER CENT 10	20	30	40	50	60	70	80	90	Authority
Natural Textile Fibers	Cotton	Sea island-roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorn
	Cotton	American-cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian merino-skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorn
	Silk	Raw cheyennes-skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry Spun-yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Avg. of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila & sisal-rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayon	Viscous Nitrocellulose Cupramonium	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cellulose Acetate	Fiber	0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3	Robertson
Paper	M.F.Newsprint	Wood pulp-24%ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	U.S.B. of S.
	H.M.F. Writing	Wood pulp-3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2	U.S.B. of S.
	White Bond	Rag - 1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U.S.B. of S.
	Com. Ledger	75% rag - 1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U.S.B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U.S.B. of S.
Misc. Organic Materials	Leather	Sole oak-tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5	Fuwa
	Rubber	Solid Tire	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99	Fuwa
	Wood	Timber (average)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0	Forest P. Lab
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0	Ford
Food Stuffs	White Bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9	Atkinson
	Macaroni		5.1	7.4	8.8	10.2	11.7	13.7	16.2	19.0	22.1	Atkinson
	Flour		2.6	4.1	5.3	6.5	8.0	9.9	12.4	15.4	19.1	Bailey
	Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
	Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
Misc. Inorganic Materials	Asbestos Fiber	Finely Divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84	Fuwa
	Silica Gel		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic Coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated Charcoal	Steam activated	7.1	14.3	22.6	26.2	128.3	29.3	30.0	30.1	32.7	Fuwa
	Sulfuric Acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

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APPENDIX *IV*
TYPICAL PERFORMANCE CURVES

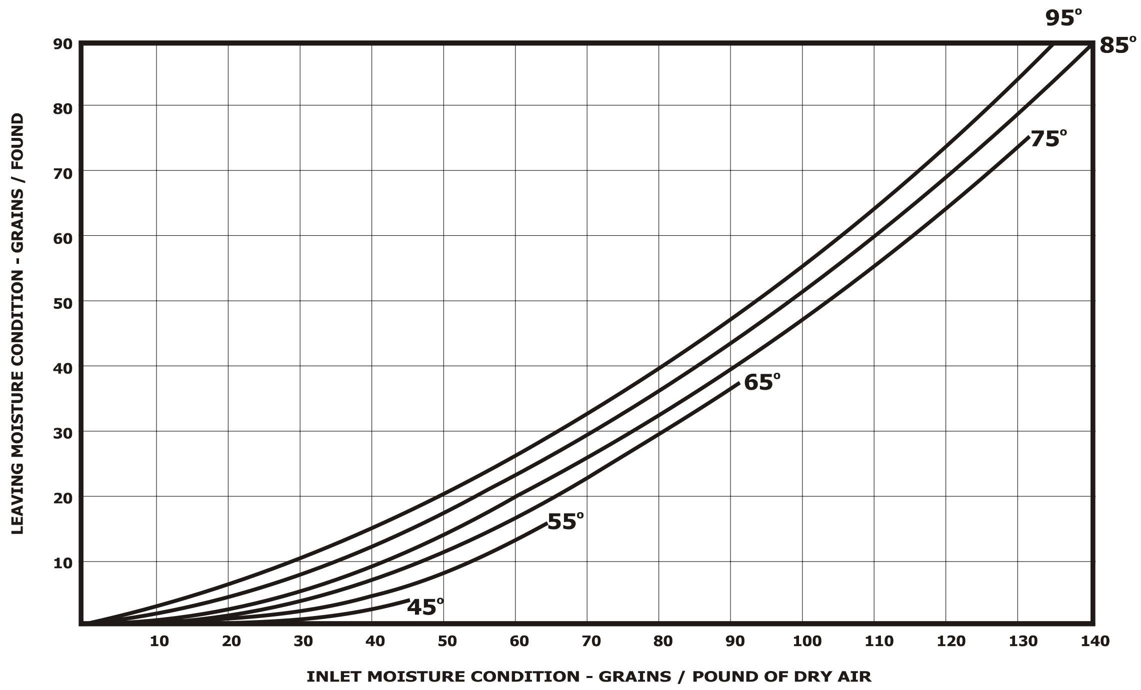


CHART - 1

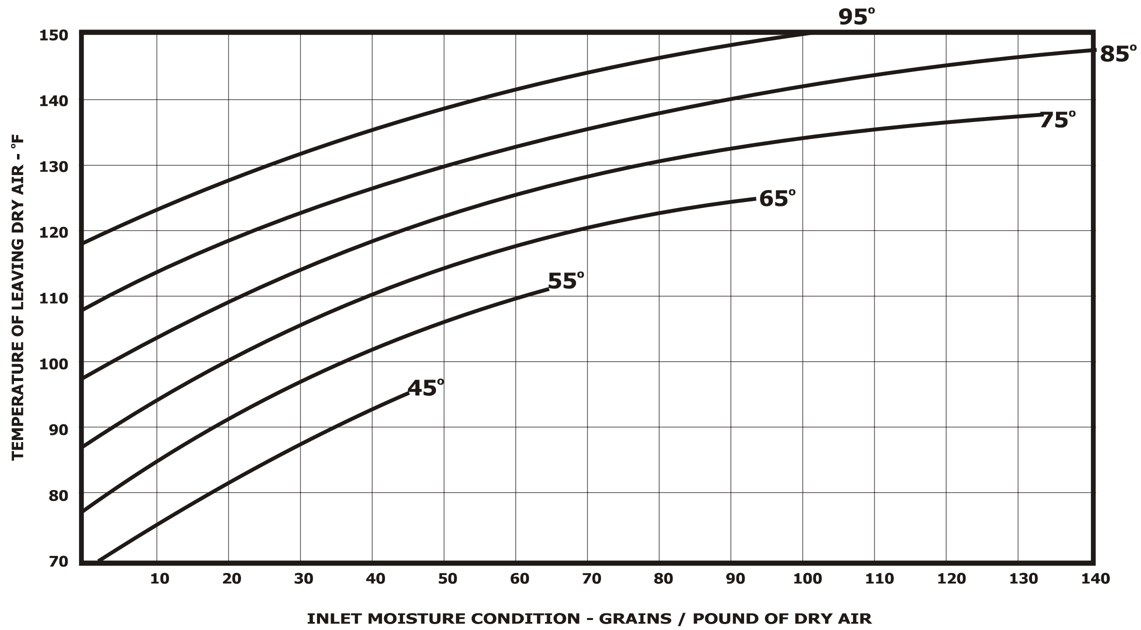


CHART - 2

- BRYSORB-100 DESICCANT
- STANDARD AIR VELOCITIES
- FOR ESTIMATING REFERENCE ONLY

APPENDIX V
TYPICAL APPLICATIONS STANDARDS

TYPICAL APPLICATION STANDARDS

	TYPICAL CONDITIONS		
APPLICATION	TEMP°F	HUMIDITY	NOTES
Aerosols	80	2 gr/lb dry air	Use hood over Freon charging nozzle and purge hood with very dry air
Brewing and Distilling			
Hops Storage	35	60% RH	
Fermentation Rms		45% RH	To reduce mold growth on walls
Filter Rooms		45% RH	To reduce mold growth on walls
Grain Storage	60	40% RH	
Baking			
Sugar Storage	80	35% RH	
Icing and Glazing	80	35% RH	
Cookie Drying	65	20% RH	
Filled Cookie Setting	20-30		Dry air to minimize defrost requirement
Potato Chips	75-80	20% RH	
Candy			Care should be exercised in the system design to ensure against recirculating air-borne sugar dust in the system.
Caramel Cooling	60	40% RH	
Bar Cooling	0	40% -50% RH	
Chocolate	90	13% RH	
Hard Candy Making	75	35% RH	
Hard Candy Packing	75	35% RH	
Sugar Storage	80	35% RH	
Concentrates			
Molasses Grinding		Less than 25%RH	Do not recirculate - Precool and dry to 20-25 gr/lb
Honey Grinding		Less than 25%RH	
Instant Coffee Packing	80	20% RH	
Citrus Crystal Packing	80	15% RH	
Ferrous Cupola Dry Blast		40 gr/lb dry air	
Crystal Growing		Varies	Dependent on particular salt
Crystal Cutting		Varies	Dependent on particular salt
Coils-Electronic		Lowest possible 1-3 gr/lb dry air	Prevent moisture absorption by insulation
Coils - TV and X-Ray	72	15% RH	
Transformer Manufacturing and Rebuilding	80	5% RH	
Elec. Appliances	72	15% RH	In blankets and pads with Nylon sheathing or insulation, moisture causes shorting and change in electrical characteristics and moisture absorbed in sealed sheath heating elements can cause failure and even explosion.
Lightning Arrestor (Assembly)	60	20% RH	

TYPICAL APPLICATION STANDARDS

APPLICATION	TYPICAL CONDITIONS TEMP°F	HUMIDITY	NOTES
Switch Gear Assembly	68	20-24% RH	
Lamination	80	20% RH max.	Paper, cloth or wood, glass fibers, etc. bonded with plastic, shellac, varnish or glue. Store material at 20% rh to insure good bond. Store finished goods at less than 80°F at 20% rh to prevent adhesion.
Chinchilla Raising	32-35	18-25% RH	High humidity is a health hazard. Low temperature and humidity provide denser and better pelts.
Matches	70	40% RH	For drying
Paper Coating	80	20% RH	For paper storage prior to coating with wax, plastic, adhesive etc. To insure bond (See lamination) (Limit paper to 4% moisture)
Paint Spraying and Lacquering	80	50% RH or less	Control spray booth or cabinet to prevent blushing caused by moisture condensing on evaporative cooled surface (evaporation of paint or lacquer solvent). Do not recirculate.
Paper and Fiber Molding	80	20% RH	Finished good storage to prevent sticky condition and loss of shape
Printing and Binding	90	30% RH	Paper storage to prevent excessive moisture content (4-6% maximum) and resultant high static electric charge. Changes in moisture content between impressions causes poor register
Pharmaceuticals			
Ampule Mfg. 80 30% rh			
Penicillin Packaging	80	5-15% RH	Hygroscopic Product
Capsule Storage	75	35-40% RH	Gelatin softens and distorts at higher temperature and humidity
Colloids	70	35% RH	
Cough Syrups	80	40% RH	
Cough Drops	70	30% RH	Similar to hard candy
Effervescents	90	15% RH	Powders or Tablets
Extracts -			
Glandular	70-80	5-10% RH	
Liver	70-80	5-10% RH	
Tablet Coating	80	5-30% RH	
Tablet Compressing	70-80	10-30% RH	
Powder Storage	70-80	15-30% RH	
Powder Mfgr	70-80	15-30% RH	
Powder Milling	80	35% RH	
Plastics, Nylon, etc.	80	3-15% RH	For drying storage prior to molding
Plastics, Manufacturing Area for Thermo Setting and Moulding Compounds	80	25-30% RH	
TV-Radio Broadcasting	70	20% RH	Tape head life extension

TYPICAL APPLICATION STANDARDS

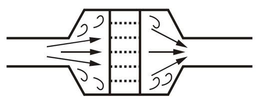
	TYPICAL CONDITIONS		
APPLICATION	TEMP°F	HUMIDITY	NOTES
Plastic, Laminated			
Built-up areas	70	20% RH	High efficiency filters required to eliminate dust and fumes
Parchment storage	70	35% RH	
Plywood	90	15-25% RH	Cold pressing
Rubber Products			
Dipped Goods	75-90	25-30% RH	Dew point of air must be below evaporation temperature of solvent.
Cementing	80	25-30% RH	
Tire Cord	125	7% RH	For storage to prevent moisture absorption and resulting loss and adhesion with rubber.
Glass Laminating	68-70	15-20% RH	To insure good long life bond
Explosives			
Gun Powder, Solid Fuels	35-75	10-50% RH	Range is great dependent on material, but storage is critical – eg - solid fuel power is calculated with consideration of time stored at given condition
Freezer Tunnel Vestibules	75	10-20% RH	Introduce air as curtain in front of tunnel opening
Pipe Galleries	40-80	40-50% RH	Maintain space dew-point below piped fluid temperature to eliminate condensation. Maintain temperature to eliminate corrosion.
Telephone & Electrical Cable Wrapping	80	5-20% RH	Extra high voltage cable wrapping has no lower limit on moisture. Upper limit is about 5% rh.
Seed Drying & Storage	35-80	10-25% RH	Low temperature drying and low rh storage preserves germination potential.
Food Drying Bins	95-105	2 gr/lb dry air	Spices - condiments
Transformer (Electric)	90	10-25% RH	
Transistors	72-75	25-40% RH	
Tubes - Radio - TV - Lighting	90	15-25% RH	Drying inside coating - There is no minimum moisture level - The dryer the air - the better
Missiles	35	35% RH	Purging and cooling
	80	25% RH	Assembly
Fertilizer Storage	AMB	40-50% RH	Deliver air, 1-1/2 air changes/hour at 25% rh when corrected to ambient dry bulb.

The above applications represent a very small% of the needs and uses for dry air; the conditions cited are typical of past uses. Specific needs change, as do technology and manufacturing process. Your Bry-Air representative can determine appropriate values for a particular application.

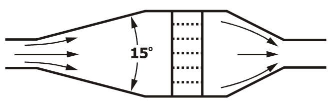
APPENDIX *VI*

METHODS FOR DUCTWORK DESIGN

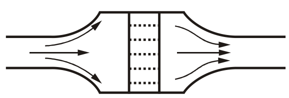
AIR FLOW THROUGH ENLARGED DUCT SECTIONS



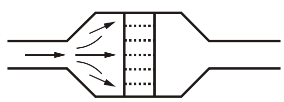
(a) With sudden enlargement air continues to flow mainly through center area.



(b) Gradual enlargement allows air to spread uniformly.

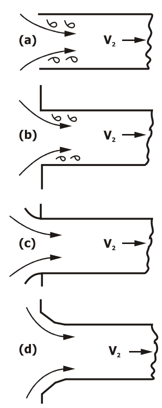


(c) Curved duct sides encourage the air to spread.



(d) Baffles are a practical solution for poor distribution in installed ductwork.

ENTRANCE LOSSES AT DUCT OPENINGS



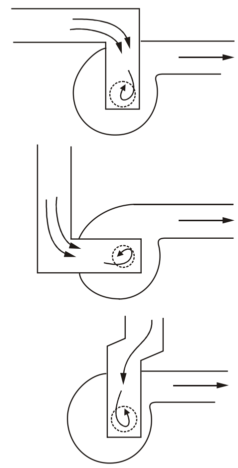
(a) $\text{Loss} = 1.0\left(\frac{V_2}{1096}\right)^2\rho$

(b) $\text{Loss} = 0.5\left(\frac{V_2}{1096}\right)^2\rho$

(c) $\text{Loss} = 0.1\left(\frac{V_2}{1096}\right)^2\rho$

(d) $\text{Loss} = \text{approx. } 0.2\left(\frac{V_2}{1096}\right)^2\rho$

CONDITOINS LIKELY TO CAUSE SPIN AT FAN INLET.

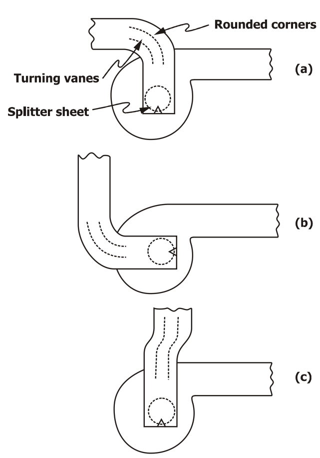


(a) Forward spin

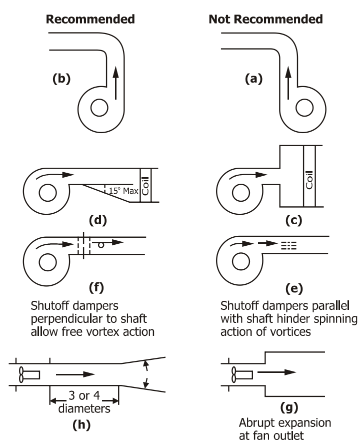
(b) Reverse spin

(c) Reverse spin

IMPROVED DESIGN TO DISCOURAGE INLET SPIN

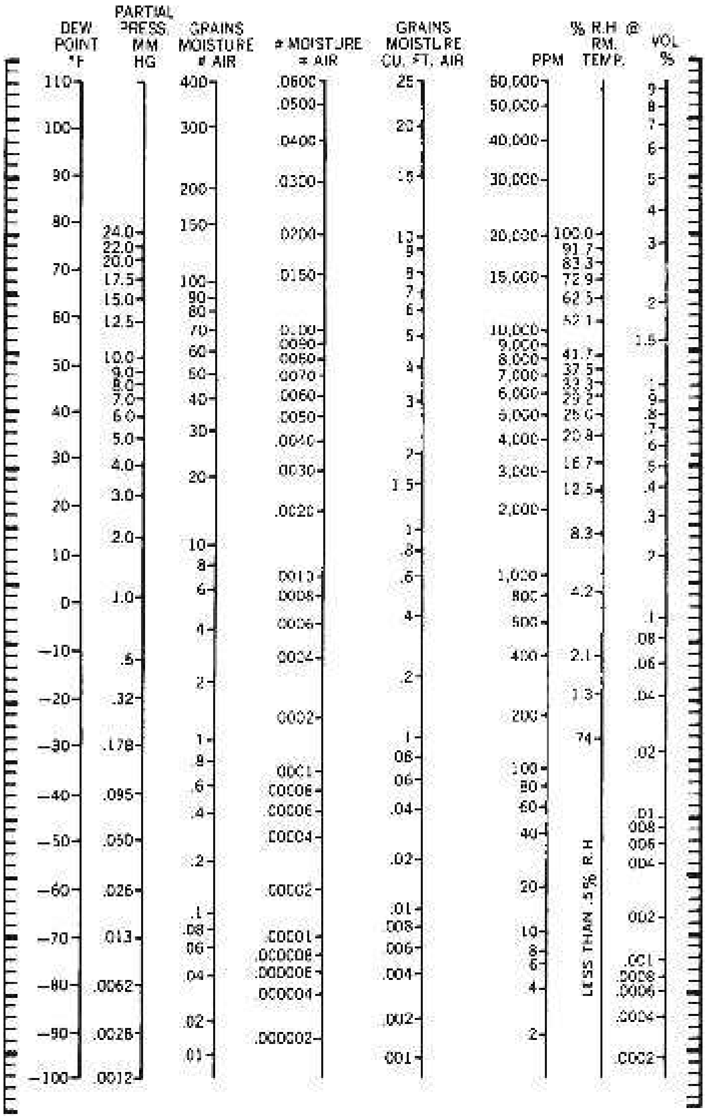


FAN DISCHARGE DUCT DESIGNS.



EQUIPMENT	RECOMMENDED VELOCITY RANGE FPM
Outside Air Intakes	300-1500
Filters	300-1000
Heating Coils	300-1000
Pre-Cooling Coils	300- 500
Fan Outlets	1300-2800
Main Ducts	1000-2400

MOISTURE CONTENT OF AIR AT ONE ATMOSPHERE



AIR PRESSURE CONVERSIONS

1 Atmosphere	=	29.92”	Hg
1 Atmosphere	=	14.7	lb / sq. in.
1 Atmosphere	=	407.0”	W.C.
1 Atmosphere	=	33.9	Ft. W.C.
1” Hg	=	0.49	lb / sq. in.
1” Hg	=	13.6”	W.C.
1” W.C.	=	0.074	Hg
1” W.C.	=	0.036	lb / Sq. in.
1” W.C.	=	25.4	mm W.C.
1” W.C.	=	0.081”	Hg
1 lb Water	=	1 pint	= 7000 grains
1 Grain	=	15.43	grams
1 Grain / lb	=	0.14	g / kg
1 g / kg	=	7.14	gr / lb

MOISTURE REMOVAL

LBS/HR $= \frac{4.5 \times CFM \times \Delta GR}{7000}$

Pints/Day $= \frac{24 \times CFM \times 4.5 \times \Delta GR}{7000}$

Grains/Min. $= \frac{(LBS/HR \times 7000)}{(4.5 \times \Delta GR)}$

(7000 = grains per lb of H2O)
(14 = cu. ft. of 1 lb of air)

Energy Formulas :

KW $= \frac{CFM \times 1.08 \times \Delta T}{3412}$

$= \frac{BTUH}{3412}$

CFM $= \frac{KW \times (3412)}{(\Delta T \times 1.08)}$

Temp. Rise $= \frac{BTUH}{CFM \times 1.08}$

BTUH = KW × 3412
= HP × 2545
= CFM × 1.08 × ΔT

USEFUL CONVERSION FORMULAS – AIR & MOISTURE

Actual CFM to Standard CFM :

$\frac{13.34 \text{ Cu. ft. / lb.}}{\text{Actual Cu. ft. / lb.}} \times \text{Actual CFM} = \text{SCFM}$

Feet per minute to CFM :

$\frac{\text{Duct size sq. in.}}{144} \times \text{FPM} = \text{CFM}$

Actual CFM to lbs. of air per hour :

$\frac{\text{Actual CFM} \times 60 \text{ (Min./Hr.)}}{\text{Cubic ft. per lb.}} = \text{lbs./hour}$

Standard CFM to lbs. of air per hour :

SCFM × 4.5 = lbs./hr.

To compute air changes per hour in a space :

$\frac{\text{CFM} \times 60 \text{ (min./hr.)}}{\text{Cu. ft. (Volume)}} = \text{AC/hour}$

DEW-POINT
DEGREES F°/C° TO GRAINS

DEWPOINT Degrees °F	DEWPOINT Degrees °C	Grains/ Pound
100	37.78	302.3
99	37.22	292.7
98	36.67	283.4
97	36.11	274.4
96	35.55	265.6
95	35.00	257.1
94	34.44	248.9
93	33.89	240.9
92	33.33	233.1
91	32.78	225.6
90	32.22	218.3
89	31.67	211.2
88	31.11	204.3
87	30.55	197.7
86	30.00	191.2
85	29.44	184.9
84	28.89	178.8
83	28.33	173.0
82	27.78	167.2
81	27.22	161.7
80	26.67	156.3
79	26.11	151.1
78	25.55	146.0
77	25.00	141.1
76	24.44	136.4
75	23.89	131.7
74	23.33	127.3
73	22.78	123.0
72	22.22	118.8
71	21.67	114.7
70	21.11	110.7
69	20.56	107.0
68	20.00	103.2
67	19.44	99.7
66	18.89	96.2
65	18.33	92.8
64	17.78	89.6
63	17.22	86.5
62	16.67	83.4
61	16.11	80.4
60	15.56	77.6
59	15.00	74.8
58	14.44	72.1
57	13.89	69.5
56	13.33	67.0
55	12.78	64.0
54	12.22	62.3
53	11.67	60.0
52	11.11	57.8
51	10.56	55.7

DEWPOINT Degrees °F	DEWPOINT Degrees °C	Grains/ Pound
50	10.00	53.6
49	9.44	51.6
48	8.89	49.7
47	8.33	47.8
46	7.78	46.1
45	7.22	44.3
44	6.67	42.6
43	6.11	41.0
42	5.55	39.5
41	5.00	38.0
40	4.44	36.5
39	3.89	35.1
38	3.33	33.7
37	2.78	32.4
36	2.22	31.2
35	1.67	29.9
34	1.11	28.8
33	0.56	27.6
32	0.00	26.5
31	-0.56	25.3
30	-1.11	24.2
29	-1.67	23.1
28	-2.22	22.0
27	-2.78	21.0
26	-3.33	20.1
25	-3.89	19.1
24	-4.44	18.2
23	-5.00	17.4
22	-5.56	16.6
21	-6.11	15.8
20	-6.67	15.1
19	-7.22	14.4
18	-7.78	13.7
17	-8.33	13.0
16	-8.89	12.4
15	-9.44	11.8
14	-10.00	11.2
13	-10.56	10.7
12	-11.11	10.2
11	-11.67	9.7
10	-12.22	9.2
9	-12.78	8.8
8	-13.33	8.3
7	-13.89	7.9
6	-14.44	7.5
5	-15.00	7.1
4	-15.56	6.8
3	-16.11	6.4
2	-16.67	6.1
1	-17.22	5.8
0	-17.78	5.5

DEWPOINT Degrees °F	DEWPOINT Degrees °C	Grains/ Pound
-1	-18.33	5.2
-2	-18.89	5.0
-3	-19.44	4.7
-4	-20.00	4.5
-5	-20.56	4.2
-6	-21.11	4.0
-7	-21.67	3.8
-8	-22.22	3.6
-9	-22.78	3.4
-10	-23.33	3.2
-11	-23.89	3.1
-12	-24.44	2.9
-13	-25.00	2.7
-14	-25.55	2.6
-15	-26.17	2.4
-16	-26.67	2.3
-17	-27.22	2.2
-18	-27.78	2.1
-19	-28.33	2.0
-20	-28.89	1.8
-21	-29.44	1.7
-22	-30.00	1.6
-23	-30.56	1.6
-24	-31.11	1.5
-25	-31.67	1.4
-26	-32.22	1.3
-27	-32.78	1.2
-28	-33.33	1.1
-29	-33.89	1.1
-30	-34.44	1.0
-31	-35.00	1.0
-32	-35.56	.9
-33	-36.11	.9
-34	-36.67	.8
-35	-37.22	.8
-36	-37.78	.7
-37	-38.33	.7
-38	-38.89	.6
-39	-39.44	.6
-40	-40.00	.6
-41	-40.56	.5
-42	-41.11	.5
-43	-41.67	.5
-44	-42.22	.4
-45	-42.78	.4
-46	-43.33	.4
-47	-44.89	.3
-48	-44.44	.3
-49	-45.00	.3
-50	-45.56	.3

DEWPOINT Degrees °F	DEWPOINT Degrees °C	Grains/ Pound
-51	-46.11	.3
-52	-46.67	.3
-53	-47.22	.2
-54	-47.78	.2
-55	-48.33	.2
-56	-48.89	.2
-57	-49.44	.2
-58	-50.00	.2
-59	-50.56	.2
-60	-51.11	.1
-61	-51.67	.1
-62	-52.22	.1
-63	-52.78	.1
-64	-53.33	.1
-65	-53.89	.1
-66	-54.44	.1
-67	-55.00	.1
-68	55.56	.1
-69	-56.11	.1
-70	-56.67	.1
-71	-57.22	.1
-72	-57.78	.1
-73	-58.33	.1
-74	-58.89	.1
-75	-59.44	.1
-76	-60.00	.0
-77	-60.56	.0
-78	-61.11	.0
-79	-61.67	.0
-80	-62.22	.0
-81	-62.77	.0
-82	-63.33	.0
-83	-63.89	.0
-84	-64.44	.0
-85	-65.00	.0
-86	-65.56	.0
-87	-66.11	.0
-88	-66.67	.0
-89	-67.22	.0
-90	-67.78	.0
-91	-68.33	.0
-92	-68.89	.0
-93	-69.44	.0
-94	-70.00	.0
-95	-70.56	.0
-96	-71.11	.0
-97	-71.67	.0
-98	-72.22	.0
-99	-72.78	.0
-100	-73.33	.0

DEW POINT/FROST POINT CONVERSION TABLES

°F FROST POINT	°F DEW POINT	°F FROST POINT	°F DEW POINT
32	32	-12	-16.7
31	30.8	-13	-17.8
30	29.7	-14	-18.9
29	28.6	-15	-20.0
28	27.5	-16	-21.1
27	26.4	-17	-22.2
26	25.2	-18	-23.3
25	24.1	-19	-24.3
24	22.9	-20	-25.4
23	21.8	-21	-26.4
22	20.7	-22	-27.5
21	19.6	-23	-28.6
20	18.5	-24	-29.6
19	17.4	-25	-30.6
18	16.2	-26	-31.7
17	15.1	-27	-32.8
16	14.0	-28	-33.9
15	12.9	-29	-35.0
14	11.8	-30	-36.1
13	10.7	-31	-37.2
12	9.6	-32	-38.2
11	8.5	-33	-39.3
10	7.4	-34	-40.3
9	6.3	-35	-41.4
8	5.2	-36	-42.4
7	4.1	-37	-43.5
6	2.9	-38	-44.5
5	1.8	-39	-45.6
4	0.7	-40	-46.6
3	-0.4	-41	-47.7
2	-1.5	-42	-48.7
1	-2.6	-43	-49.8
0	-3.7	-44	-50.8
-1	-4.8	-45	-51.9
-2	-5.8	-46	-52.9
-3	-6.9	-47	-54.0
-4	-8.0	-48	-55.0
-5	-9.1	-49	-56.1
-6	-10.2	-50	-57.1
-7	-11.3	-51	-58.2
-8	-12.4	-52	-59.2
-9	-13.5	-53	-60.3
-10	-14.6		
-11	-15.6		

TEMPERATURE CONVERSION TABLE

TO °C	TO CONVERT °F or °C	TO °F	TO °C	TO CONVERT °F or °C	TO °F	TO °C	TO CONVERT °F or °C	TO °F
- 45.66	- 50	- 58.0	- 6.11	21	69.8	22.22	72	161.6
- 42.77	- 45	- 49.0	**- 5.56**	**22**	**71.6**	22.78	73	163.4
- 40.00	- 40	- 40.0	- 5.00	23	73.4	23.33	74	165.2
- 37.23	- 35	- 31.0	- 4.44	24	75.2	23.89	75	167.0
- 34.44	- 30	- 22.0	- 3.89	25	77.0	24.44	76	168.8
- 31.66	- 25	- 13.0	- 3.33	26	78.8	25.00	77	170.6
- 31.11	- 24	- 11.2	- 2.78	27	80.6	25.56	78	172.4
- 30.55	- 23	- 9.4	- 2.22	28	82.4	26.11	79	174.2
- 30.00	- 22	- 7.6	- 1.67	29	84.2	26.67	80	176.0
- 29.45	- 21	- 5.8	- 1.11	30	86.0	27.22	81	177.8
- 28.89	- 20	- 4.0	- 0.56	31	87.8	27.78	82	179.6
- 28.34	- 19	- 2.2	0	32	89.6	28.33	83	181.4
- 27.78	- 18	- 0.4	0.56	33	91.4	28.89	84	183.2
- 27.23	- 17	1.4	1.11	34	93.2	29.44	85	185.0
- 26.67	- 16	3.2	1.67	35	95.0	30.00	86	186.8
- 26.12	- 15	5.0	2.22	36	96.8	30.56	87	188.6
- 25.56	- 14	6.8	2.78	37	98.6	31.11	88	190.4
- 25.00	- 13	8.6	3.33	38	100.4	31.67	89	192.2
- 24.44	- 12	10.4	3.89	39	102.2	32.22	90	194.0
- 23.88	- 11	12.2	4.44	40	104.0	32.78	91	195.8
- 23.33	- 10	14.0	5.00	41	105.8	33.33	92	197.6
- 22.77	- 9	15.8	5.56	42	107.6	33.89	93	199.4
- 22.22	- 8	17.6	6.11	43	109.4	34.44	94	201.2
- 21.67	- 7	19.4	6.67	44	111.2	35.00	95	203.0
- 21.11	- 6	21.2	7.22	45	113.0	35.56	96	204.8
- 20.56	- 5	23.0	7.78	46	114.8	36.11	97	206.6
- 20.00	- 4	24.8	8.33	47	116.6	36.67	98	208.4
- 19.44	- 3	26.6	8.89	48	118.4	37.22	99	210.2
- 18.89	- 2	28.4	9.44	49	120.2	37.78	100	212.0
- 18.33	- 1	30.2	10.00	50	122.0	43.33	110	230.0
- 17.78	0	32.0	10.56	51	123.8	48.89	120	248.0
- 17.22	1	33.8	11.11	52	125.6	54.44	130	266.0
- 16.67	2	35.6	11.67	53	127.4	60.00	140	284.0
- 16.11	3	37.4	12.22	54	129.2	65.56	150	302.0
- 15.56	4	39.2	12.78	55	131.0	71.11	160	320.0
- 15.00	5	41.0	13.33	56	132.8	76.67	170	338.0
- 14.44	6	42.8	13.89	57	134.6	82.22	180	356.0
- 13.89	7	44.6	14.44	58	136.4	87.78	190	374.0
- 13.33	8	46.4	15.00	59	138.2	93.33	200	392.0
- 12.78	9	48.2	15.56	60	140.0	98.89	210	410.0
- 12.22	10	50.0	16.11	61	141.8	104.44	220	428.0
- 11.67	11	51.8	16.67	62	143.6	110.00	230	446.0
- 11.11	12	53.6	17.22	63	145.4	115.56	240	464.0
- 10.56	13	55.4	17.78	64	147.2	121.11	250	482.0
- 10.00	14	57.2	18.33	65	149.0	126.67	260	500.0
- 9.44	15	59.0	18.89	66	150.8	132.22	270	518.0
- 8.89	16	60.8	19.44	67	152.6	137.78	280	536.0
- 8.33	17	62.6	20.00	68	154.4	143.33	290	554.0
- 7.78	18	64.4	20.56	69	156.2	148.89	300	572.0
- 7.22	19	66.2	20.11	70	158.0			
- 6.67	20	68.0	21.67	71	159.8			

USEFUL CONVERSION FACTORS

TO CONVERT	INTO	MULTIPLY BY
ATMOSPHERES	KILOPASCAL (kP_a)	101.1
Btu	JOULES	1,054
Btu	KILOWATT-HRS	2.928×10^{-4}
Btu/hr	HORSEPOWER	3.929×10^{-4}
Btu/hr	WATTS	0.293
Btu/hr-sq ft	WATTS/SQ METER	3.15
Btu/lb	KILOJOULES/KILOGRAM	2.33
CENTIGRADE	FAHRENHEIT	(9/5C + 32)
COST, $/LB	$/KILOGRAM	2.205
COST, $/TON (REFRIG)	$/KILOWATT	0.284
CUBIC FEET	CU METERS	0.02832
CUBIC FEET	LITERS	28.32
CUBIC FEET/MIN, CFM	LITERS/SEC	0.4720
CUBIC FEET/MIN (CFM)	CUBIC METER/HR	1.6992
CUBIC INCHES	MILLILITERS	16.4
CUBIC METERS	CU FEET	35.31
CUBIC METERS	LITERS	1,000
CUBIC METERS/HR	CUBIC FEET/MIN (CFM)	.5885
FAHRENHEIT	CENTIGRADE	(°F-32) × 5/9
FEET	METERS	0.3048
FEET/MIN, FPM	METERS/SEC	5.08×10^{-3}
FEET MIN, FPM	MILES/HR	0.01136
GALLONS	CU FEET	0.1337
GALLONS	LITERS	3.785
GALLONS/HR	MILLILITERS/SEC	1.05
GALLONS/MIN	LITERS/SEC	0.06308
GRAINS (1/7000 LB)	GRAMS	0.06480
GRAINS/LB	GRAMS/KILOGRAM	0.143
GRAMS	GRAINS	15.43
GRAMS	POUNDS	2.204×10^{-3}
HORSEPOWER	Btu/hr	2,547
HORSEPOWER	KILOWATTS	0.7457
INCHES	MILLIMETERS	25.40
INCHES OF MERCURY	INCHES OF WATER	13.596
INCHES OF MERCURY	KILOPASCAL (kP_a)	3.38
INCHES OF WATER (AT 4°C)	INCHES OF MERCURY	0.07355
INCHES OF WATER (AT 4°C)	POUNDS/SQ IN	0.03613
INCHES OF WATER (AT 4°C)	PASCAL (P_a)	249.
JOULES	Btu	9.48×10^{-4}
JOULES	WATT-HRS	2.778×10^{-4}

USEFUL CONVERSION FACTORS

TO CONVERT	INTO	MULTIPLY BY
KILOGRAMS	POUNDS	2.205
KILOGRAMS/CU METER	POUNDS/CT FT	0.06243
KILOMETERS	FEET	3,281
KILOMETERS	MILES	0.6214
KILOMETERS/HR	MILES/HR	0.6214
KILOWATTS	Btu /HR	3,414
LITERS	CU FEET	0.03531
LITERS	GALLONS (U. S. LIQ)	0.2642
LITERS/SEC	GALLONS/MIN	15.852
METERS	FEET	3,281
METERS/SEC	FEET/MIN	196.8
METERS/SEC	KILOMETERS/HR	3.6
METERS/SEC	MILES/HR	2.237
MILES (statute)	METERS	1,609
MILES/HR	FEET/MIN	88
MILES/HR	KILOMETERS/HR	1.609
MILLIMETERS	INCHES	0.03937
NEWTON	POUNDS (force)	0.225
NEWTON/SQ METER	PASCAL (P_a)	1.0
PASCAL (P_a)	INCHES OF MERCURY	2.959×10^{-4}
PASCAL (P_a)	INCHES OF WATER	4.016×10^{-3}
PASCAL (P_a)	POUNDS/SQ IN	1.451×10^{-4}
POUNDS (force)	NEWTONS	4,448
POUNDS (mass)	GRAINS	7,000
POUNDS (mass)	KILOGRAMS	0.4536
POUNDS OF WATER	CU FEET	0.01603
POUNDS OF WATER	GALLONS	0.1199
POUNDS OF WATER	PINT	.9584
POUNDS OF WATER	LITERS	.0316
POUNDS/CT FT	KGS/CU METER	16.02
POUNDS SQ IN	INCHES OF MERCURY	2.036
POUNDS/SQ IN	KILOPASCAL (kP_a)	6.89
POUNDS/SQ IN	INCHES OF WATER	27.68
TONS, REFRIGERATION	Btu/HR	12,000
TONS, REFRIGERATION	KILOWATTS	3.52
WATTS	Btu/HR	3.414
WATTS	HORSEPOWER (ELEC.)	1.341×10^{-3}
WATT-HOURS	Btu	3.414
WATT-HOURS	JOULES	3.60×10^{3}

CONVERSION FACTORS

Quantity	SI Unit	c.g.s. Unit	f.p.s. or other
Length	1 m	100 cm	3.281 ft=39.37 in
Time	1 s	1 s	1 s
Mass	1 kg	1000g	2.205 ib
Area	1 m^2	104 cm^2	10.764ft^2=1550.0 in^2
Volume	1 m^3	106 cm^3	35.31 ft^3=6.102x104 in^3
Frequency	1 Hz	1 c/s	1 c.p.s.
Rotational frequency	1 S^{-1}	1 rev/s	60 r.p.m.
Density	1 Mg/m^3	1 Kg/m^3	62.43 lb/ft^3=0.036131lb/in^3
Velocity	1 m/s	100 cm/s	3.281 ft/s=39.37 in/s
Fluid velocity	1 m^3/s	6x10^4 lts./min.=$10^6$$cm^2$/s	2119 ft^3/min=6.1 x 104 in^3/s
Force	1 N	0.102 kgf=10^5 dynes	0.2248 lbf=7.233 pdl
Surface Tension	1/m	1 dyne/cm	
Pressure	1 Mpa	10.20 kgf/cm^2=107dyne/cm^2	145 p.s.i.
Stress	1 kPa	10.20 gf/cm^2=104dyne/cm^2	0.1450 p.s.i.
Strength	1 Pa	7.5x10^{-3}mmHg	
Modulus	1 mN/m^2	7.5x10^{-6}mmHg	
Torque	1 N/m	10.20 kgf cm	141.6 oz in
Energy	1 J	1.0197x10^4gf cm= 0.239cal=10^7ergs	0.738 ft lbf=9.48x10^{-4}Btu
Power 1 W	1 MJ	0.278kWh=239 kcal	
	0.86 kcal/h	0.00134 HP=3.41 Btu/h	948 Btu
Momentum	1 kgmS^{-1}	105 g cms^{-1}	7.23071b ft S^{-1}
Moment of Inertia	1 kg/m^2	107 g cm^2	23.730 lb ft^2
Thermal Conductivity	1 W/mK	0.00239 cal/cm°c	6.93 Btu In/ft^2°F
Viscosity - dynamic	1 Ns/m^2	10 poise	0.02089 lbf s ft^2
Viscosity - Kinematic	1 m^2/s	10^4stokes	
Permeability	1 m^4/Ns	1.013x$10^9$$cm^2$	
Electrical Resistivity	1 ohm	100 ohm	

* Pressure may also be expressed In bars (b)
1 bar = 105 N/m^2 = 14.504 p.s.i.
+ Electrical, mechanical or heat energy

Commonly used Metric Conversions

AREA

1 $inch^2$	in^2	=	645.6 mm^2
1 $inch^2$	in^2	=	0.000 645 m^2
1 $foot^2$	ft^2	=	0.0929 m^2
1 $yard^2$	yd^2	=	0.836 m^2

DENSITY

1 gram per cc	g/cc	=	0.036 1 lb/in^3
1 pound per cu ft	lb/ft^3	=	16.02 kglm^3
1 pound per cu ft	lb/ft^3	=	62.37 g/cc
1 pound per cu in	lb/in^3	=	0.027 7 kg/m^3

ELECTRICAL

1 volt per mil V/mil = 25.4 kV/mm

ENERGY - WORK

1 erg	dyn.cm	=	1 x 10^{-7} J
1 Newton-meter	N.m	=	1 J

1 watt-second	W.s	=	1 J
1 foot-pound	ft.lbf	=	1.356 J
1 horsepower-hour	hph	=	2.685 x 106 J
1 kilowatt-hour	kWh	=	3.6 106 J
1 calorie	cal	=	4.19 J
1 British thermal unit	Btu	=	1.055 J
1 British thermal unit	Btu	=	0.000 11 cal
1 inch-pound	in.lbf	=	0.133 J

FORCE

1 dyne	dyn	=	1 x 10^{-5} N
1 kilogram-force	kgf	=	9.81 N
1 pound-force	lbf	=	4.448 N
1 ton-force	tf	=	9.810 N

IMPACT STRENGTH

1 foot pound per inch^2	ft.lb/in^2	=	0.475 5 kJ/m^2
1 foot pound per Inch of notch (Izod test)	ft.lb/in	=	0.018 73 J/m
1 kgcm/cm	kgcm/cm	=	10 J/m

LENGTH

1 kilometer	km	=	1,000 m
1 centimeter	cm	=	0.01 m
1 millimeter	mm	=	0.001 m
1 micron	um	=	1 x 10^{-6} m
1 nanometer	nm	=	1 x 10^{-9} m
1 Angstrom unit	A	=	1 x 10^{-10} m
1 inch	in	=	0.025 4 m
1 foot	ft	=	0.304 8 m
1 yard	yd	=	0.9144 m
1 mile	mile	=	1.609 Km
1 inch/inch per °F	in./in./°F	=	0.5567 mm/mm/°C

MASS

1 gram	g	=	0.001 kg
1 ounce	oz	=	28.4 g
1 pound	lb	=	0.453 6 kg
1 ton	t	=	1,000 kg

POWER

1 kilowatt	kW	=	1,000 W
1 horsepower	hp	=	746 W
1 foot pound per second	ft.lbf/s	=	1.356 W

SPECIFIC ENERGY

1 calorie per gram	cal/gm	=	4,190 J/kg
1 Btu per pound	Btu/lb	=	2,326 J/kg
1 hph per pound	hph/lb	=	5.92 x 106 J/kg
1 kWh per kg	kWh/kg	=	3.6 x 106 J/kg
1 Kwh per kg	kWh/hg	=	1.644 hph/lb

STRESS

1 dyne per cm^2	dyn/cm^2	=	0.1 Pa
1 Newton per meter^2	N/m^2	=	1 Pa
1 Joule per meter^3	J/m^3	=	1 Pa
1 atmosphere	atm	=	1.013 x 105 Pa
1 mm mercury	mm Hg	=	133.3 Pa
1 mm water	mm H^{2}O	=	9.81 Pa

1 bar	bar	=	1 x 105 Pa
1 pound per inch^2	psi	=	6,890 Pa
1 megapascal	MPa	=	145 psi

TEMPERATURE

1 degree Farenheit	°F	=	1.8°C + 32
1 degree Celcius	°C	=	0.5555 (°F-32)
1 degree Kelvin	°K	=	°C + 273.16
1 degree Rankine	°R	=	°F + 459.69

THERMAL CONDUCTIVITY

1 gram		=	419 J/ms.K
1 Kcal/m.h.°C		=	1.163 J/ms.K
1 Btu/ft.hr. °F		=	1.76 J/ms.K
1 Btu/in.ft^2.h.°F		=	0.144 J/ms.K
1 Btu/ft.s. °F		=	6,230 J/ms.K
1 Wm.k		=	1 J/ms.K

VISCOSITY

1 poise	poise	=	0.1 Pa.s
1 poise	poise	=	14.5 x 10^{-6} psi.s
1 poise	poise	=	1 dyne/cm^2
1 poise-second/in^2	psi.s	=	6,897 Pa.s

VOLUME

1 cubic centimeter	cc	=	1 x 10^{-6} m^3
1 liter	L	=	0.001 m^3
1 milliliter	mL	=	1 x 10^{-6} m^3
1 cubic inch	in^3	=	1.639 x 10^{-5} m^3
1 cubic foot	ft^3	=	0.028 32 m^3
1 gallon US	gal US	=	0.003 785 m^3
1 gallon UK	gal UK	=	0.004 546 m^3

ABBREVIATION

A	=	Angstrom Unit	kg	=	Kilogram
atm	=	Atmosphere	L	=	Liter
BTU	=	British Thermal Unit	lb	=	Pound
C	=	Degrees Celsius	m	=	Meter
cal	=	Calorie	ml	=	Milliliter
cc	=	Cubic Centimeter	mm	=	Millimeter
cm	=	Centimeter	Mpa	=	Megapascal
cu	=	Cubic	N	=	Newton
f	=	Force	oz	=	Ounce
dyn	=	Dyne	Pa	=	Pascal
F	=	Degrees Farenheit	psi	=	Pounds per Square Inch
ft	=	Foot	R	=	Degrees Rankline
gal	=	Gallon	s	=	Second
g	=	Gram	t	=	Ton
h	=	Hour	tn	=	Ton US
Hg	=	Mercury	ton	=	Ton UK
Hp	=	Horsep	V	=	Volts
in	=	Inch	w	=	Watt
J	=	Joule	yd	=	Yard
K	=	Degrees Kelvin			

NOTES

NOTES

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