An Affordable Compressed Air Dryer for Home Use Spray Painting

by

RODNEY JEFFERS

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Signature of Author

Certified by

Accepted by

Mutha A.-Ubaidi, PhD, Department Head
Mechanical Engineering Technology
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Designing an affordable compressed air dryer for my senior design project was an incredible achievement. I could not have completed it without the help of several people. I would like to thank Ron Brinkerhoff, my advisor for guiding me throughout the entire process. His knowledge, hard work and dedication as a professor disciplined and challenged me to work even harder than I had ever imagined. I would like to thank the MET department Head Dr. Muthar Al-Ubaidi who helped me solve a great deal of problems in the design. I would like to thank Dr. Maria Kreppel for her time and dedication in preparing me for tech expo, the final project presentation and report, as well as for future endeavors. I would also like to thank the rest of the MET faculty for preparing me for this type of project. Without all of there hard work and dedication I would have surely drown.

I would like to thank Campbell Hausfeld for their generous donations of materials, testing equipment and lab facilities. Employees of Campbell that I would like to thank are Joe Abt, Bill Bryant, Steve Brown, A.J Viel, Todd Reger, Chris Klein and Rick Strack for all of their contributions and help along the way. I would like to thank TKF Conveyors INC. for donating materials for the reports as well as the use of their copiers, computers and printers. I would like to thank Dr. Gun of Sharpe Manufacturing for spending a great deal of time on the phone answering questions pertaining to my air dryer design. His time and help allowed me to gain a better understanding of how air dryers function.

I could not have completed this project without the support of my parents, family, friends and most of all my fiancée Andrea. They all tolerated the busy schedule that I led on a day-to-day basis and always listened to me complain about everything. Except for my dad who always said, “if it were easy, everyone would be doing it.” This was true which made me complain a little less and work a little harder. I thought perhaps somewhere in the middle of the project I was going to lose it, but hung in there and succeeded. Again for everyone that helped me in one-way or another throughout this project, as well as my college education I thank you.
This report discusses in detail the need for an affordable compressed air dryer for home-use spray painting and covers the entire design, fabrication and testing sequence involved for the completion of a working prototype model. The most common problems involved in the spray painting process are high levels of humidity in the compressed air and the high cost of equipment to eliminate it. In order to achieve a quality paint job these problems need to be addressed and solved.

The solution is to efficiently remove the bulk quantity of moisture from the compressed air by forcing the high temperature air from the air compressor to cold air at the outlet of the dryer therefore lowering its dew point temperature. The simplistic yet functional design incorporates only a few main components such as the heat exchanger tube, insulated container, water separator, and quick disconnect fittings. An ice and water combination bath supplies enough cooling capacity for two to three hours of operation depending on the outside climate conditions.

This report details the analysis at each stage of the design process from customer input, design, fabrication and assembly of the prototype and then to the final proof of design testing procedures. Customer surveys supplied the overall design with three main customer requirements or objectives: portability for easy storage, durability for long use, and low reasonable cost allowing the home user to easily afford. The final prototype design proved to be an efficient device for eliminating moisture content in compressed air therefore allowing a high quality paint job at a low cost when spray painting.
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PROJECT INTRODUCTION

The intended goal of this project was to design a prototype compressed air dryer that was both economical and adequate as compared to current equipment. Dry air is the leading factor in spray painting. The spray-painting process requires compressed air to be as dry as possible, but how dry? The air should be dry enough so no moisture will condense in the airlines. Whatever type of air dryer is selected the objective is the same – to prevent moisture from condensing in the airlines by removing large amounts of humidity from the air stream. The main concern is the pressure dew point temperature level. By lowering the dew point temperature of the compressed air it will efficiently lower the moisture content within the system.

An air compressor aids as a powerful mechanical system in producing a large quantity of “on-demand” air. However, in doing so, it consequently transfers oil deposits into the air stream as well as heating and cooling the air which causes high levels of condensation to form producing water. Raw air from the air compressor is hot, dirty and loaded with moisture. The extreme elevated temperatures cause the moisture content to become rather large (Appendix A, pg. 2). This buildup of water can lead to catastrophic defects when spray painting a car.

Most people are unfamiliar with the theory in which the spray painting process takes place. Compressed air that is hot and saturated with moisture enters the spray gun and flows through a design of passageways. The paint mixture is placed inside the canister and as the air travels through the guns passage ways the paint is siphoned out of the canister and mixed with the air stream. The air paint mixture is then expanded through a tiny orifice in the gun where the atomization process occurs. The atomization process, in theory takes a wet paint air stream and turns it into an extremely fine air mist of paint molecules. However, if the initial compressed air stream is loaded with moisture then it makes the atomization process difficult to complete as the moisture slows down the process that ultimately leaves a heavier and thicker air mist of molecules rather than a fine mist.

However, the most vital concern of the compressed air is the percent level of compressed air humidity (CAH). Levels exceeding 20% can often cause a list of several detrimental defects in the finished quality such as cloudy topcoats and discolored
base coats (See Appendix B). The goal is to dry the air therefore lowering the percent level of CAH in the system.

**SCOPE OF THE SOLUTION**

Ensuring your quality of work in spray painting can be quite costly and could however be more economical to better meet the typical consumer. The goal is dry air and the outcome is a higher level of paint quality for a significant lower over-all cost. This goal can easily be executed with a simple, but extremely effective idea of basic compressed air logic. When air is compressed it is heated to high temperatures, however when it is allowed to flow through the system and air-line hose it cools down the longer it travels (See Appendix A). As it cools down it causes condensation to form producing water in the airlines. However, this water can be efficiently eliminated from the system by forcing the hot air to cold air in the very beginning of the process therefore allowing cold to moderately warm dry air to flow through the rest of the system.

**SCOPE OF THE PROJECT:**

This project will be successfully carried out through a detailed technical overview of the entire prototype design process. First, a detailed project schedule was generated to allow for easy transition throughout each of the stages. See Appendix C for an excel spreadsheet revealing the schedule used in this project. A project budget was also created in order to record and keep an eye on how much money was being spent in certain areas. A detailed overview of the budget can be found in Appendix D. The getting started section will provide a general starting point in the design process. This will lead to the voice of customer in order to collect and organize the type and style of product the customers desire to obtain. The next section covers all of the technical analysis concerning the design of the prototype air dryer. Each component of the assembly is carefully analyzed and selected based on various assumptions as well as economical means. By analyzing and selecting each of the components the next section offers an overview of the detailed parts. Now that the system has been designed the fabrication and assembly process is executed. Once the prototype has become a working model a series of testing procedures and analysis is performed to determine the efficiency level of
the design. The final section organizes and details all of the accomplishments, achievements and recommendations concerning the prototypes overall performance. All of these sections combine to allow for a smooth, yet thorough transition throughout the project that will help in delivering the product to the customer a huge success as well as profitable.

Section 1: Getting Started

Assumptions:

To get started and successfully complete the prototype design of an air-drying device for the spray-painting process several assumptions must first be made in order to advance to the next section of the design process. A complete list of all assumptions is as follows:

- Inlet air from compressor is 120 degrees Fahrenheit
- Pressure at the inlet of the spray gun is 45-PSI gauge
- Operable painting conditions are above 50 degrees Fahrenheit and below 85% relative humidity
- Nominal size of heat exchanger tube is ½”
- Compressor size is a 5 horsepower with a 20 to 60-gallon tank capacity
- Paint spray gun ratings are not to exceed 5 CFM of flow rated at 40-PSI gauge

Influential Design Factors:

A list of several influential design factors was generated by customer observation in order to help focus on certain areas of the design. These factors were considered to be persuasive in the sense that they will aid in the selection of the proper materials, as well as the machining and manufacturing processes that will be used for the final product.

Weight  The weight of the system should be kept lightweight.
Ease of Use  The product should require little effort to understand and operate.
Cost  Cost should be kept reasonable with respect to quality and current equipment in the market.

Maintenance  System should require little to know maintenance.

Reliability  The system should perform in a consistent and repeated manner.

Safety  The system should be safe to use causing no harm.

Production  Part production and assembly must be able to be accomplished at a realistic cost.

Summary:

To get started on an in depth project such as this it is often easy to begin with key assumptions, which ultimately dictate the design by stating all of the conditions that must be applied. The influential design factors were listed to help focus in on some important aspects of the design. These factors were kept in mind throughout the entire design process.

SECTION 2: Responding to Customer Needs

This section involves and focuses heavily on several objectives, such as hearing the voice of the key customers, knowing the effect of all of the variables, prioritizing device features and criteria, and selecting the most economical and efficient of the three alternatives. A customer survey was designed consisting of several short questions that allowed us to collect specific information about the high concerns and importance of dry air. The key customer that was targeted was the homeowner and do-it-yourselfer. This survey can be found in Appendix E as well as a summary of all of the numerical results can be found in Appendix F.

All of the survey responses along with the Quality Functional Deployment method also known as the QFD analysis were utilized to build the house of quality. A detailed structure of the House of Quality can be found in Appendix G. Once the House of Quality was constructed the three alternative design configurations were determined and roughly designed. Using the Pugh selection process method the three configurations
were eliminated down to one alternative that was ultimately the most attractive and appealing design of the three.

**House of Quality:**
*The following sections are the WHATS, WHYS, and the HOWS that were collected, calculated and determined through the results of the customer survey (Appendix E).*

**Voice of the Customer (WHATS)**
The following Customer Requirements (CR’s) were rated on a scale of 1 through 5, 5 being the best (See Appendix G).

- **Reliability:** Must be able to perform properly with minimal to no malfunctions.
- **Quick Set-Up:** This product should allow for easy installation and disassemble.
- **Portability:** The product should be compact in size as well as light in weight in order to make this an item that can easily be transported and moved from one point to another.
- **Safety:** The product should be safe and meet all compliance standards to ensure the safety of the operator.
- **Storage:** This product should be designed so that it can easily be put away in storage when not in use.
- **Durability:** The air dryer will often be used in such environments as garages and local work areas. The product must be durable enough to withstand human abuse that may be caused during storing as well as assembling and disassembling.
- **Ease of Serviceability:** If perhaps there was a failure or malfunction concerning the air dryer, then the final design should allow for easy access to change out and replace parts easily and inexpensively. Also the loading of ice and water and the emptying of the condensate should both be accessible.

**Air Dryer Characteristics (HOWS)**
The following Engineering Requirements (EC’s) have been chosen to become critical variables in attaining customer satisfaction through the customer requirements. In the House of Quality, their importance is rated a 1 for weak, 3 for medium and 9 for high (See Appendix G).
- **Pipe Length (in):** The length of the pipe must be kept at a realistic value but long enough to properly cool the compressed air.

- **Pipe Diameter (in):** The pipe diameter must be near or close to the rest of the piping sizes to reduce pressure drop in the airlines.

- **Pipe Material (lbs/in\(^3\)):** The pipe material must meet standards for compressed air as well as serve as the best possible heat exchanger.

- **Container Material (lbs/in\(^3\)):** The container material must be durable enough to withstand human abuse but light enough to easily be moved.

- **Container Size (gal.):** The size must be kept to a minimum due to the relationship of size and weight. If it is larger in size then obviously it will weigh more. However, the air dryer must be large enough to hold enough ice to properly cool the air to a certain degree, but small enough to be portable enough to carry or move from one place to another.

- **Cooling Capacity (btu/hr):** This more or less refers to the amount of operation time with respect to amount of heat it is encountering. The cooling capacity must be enough to allow for sufficient time to do the desired work.

- **Installation Time (min.):** Should be easy to install and disassemble.

- **Standard Parts:** Standard pipe fittings with quick disconnect features should be included to allow for easy assembly and disassembly.

- **Weight (lbs):** The weight must be kept to a minimum in order to easily move the system from one point to another.

**Modified Importance (WHYS)**

In the House of Quality the following headings are the importance of each customer requirement, which were rated on a scale of 1 through 5, with 5 being the most important (See Appendix G).

- **Customer Importance:** How important is the CR to the customer.
• **Dryer in Market:** How do the CR’s of the dryer on the market rate to the customer?

• **Planned Dryer:** How would we like to rate with the customer?

• **Improvement Ratio:** Ratio of Planned Dryer to Dryer on Market.

• **Sales Point:** Extras. These are features about my dryer that can be used to sell customers my dryer and to show why it is adequate and economical than the competitors dryer.

• **Improved Ratio:** This is the product of Planned Air dryer, Improvement Ratio, and Sales Point.

• **Relative Weight:** Found by dividing the improvement ratio by the sum of all improvement ratios.

**Air Dryer Alternative Designs:**

Three alternative air dryer configurations were determined based primarily on data obtained from the House of Quality. However, before any ideas were generated a better understanding of the air-drying process and desired functions of the prototype design were needed. In order to achieve this aspect a detailed block diagram of the air drying process was developed (See Appendix H). This diagram allows a better understanding of what the prototype will be designed to achieve. The following sections include a brief description of each alternative design concept.

**Alternative #1:**

This concept entails an estimated 5-gallon round bucket container with coil tubing contained in it. Standard connections of ½” NPT with quick connects for both the inlet and exit are included. Ice is used to surround the coil of tubing, which cools the air as it passes through the container. As the air is cooled the water vapor condenses into water. A water separator (trap) is installed to trap the condensed water from exiting the system and flowing through the rest of the airlines. This concept would be designed to sit on the floor and would be portable enough to move from one place to another. Refer to Appendix I for a visual image of an AutoCAD drawing.
Alternative #2:
This concept would include a 4 to 8 inch diameter cylindrical container 6 feet in length. The system would be a freestanding unit. Inside the container would include a straight piece of tubing to serve as the “heat exchanger.” Both the inlet and outlet will include standard ½” NPT fittings with quick connect adaptors. A water separator (trap) will be included at the base of the container. Refer to Appendix J for a visual image of an AutoCAD drawing.

Alternative #3:
This concept is made up of a rectangular container in which a straight piece of tubing is ran through one end of the container, bends at 180 degrees and comes right back out the same end two times. Both the inlet and outlet have standard ½” connections with quick connects. A water separator will be included at the outlet of the air stream. Container dimensions range from 10 to 20 inches in height and width and 20 to 40 inches in length. Refer to Appendix K for a visual image of an AutoCAD drawing.

Selection Process:
To determine which alternative concept was the most appealing design the Pugh’s selection process chart was used to compare each of the three alternatives to the original or datum concept. The datum concept chosen for this application was a refrigerated air dryer similar in size. The specifics of this product can be found in Appendix L.

Pugh’s Selection Process Chart

<table>
<thead>
<tr>
<th>Criterion</th>
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<th>Alt. #2</th>
<th>Alt. #3</th>
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<tbody>
<tr>
<td>Height</td>
<td>S</td>
<td>-</td>
<td>S</td>
<td>DATUM</td>
</tr>
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<tr>
<td>Weight</td>
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<td>Ease of serviceability</td>
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<td>+</td>
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<tr>
<td>Easy storage</td>
<td>+</td>
<td>-</td>
<td>S</td>
<td>DATUM</td>
</tr>
<tr>
<td>Quick set-up</td>
<td>+</td>
<td>+</td>
<td>S</td>
<td>DATUM</td>
</tr>
<tr>
<td>Product Cost</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>DATUM</td>
</tr>
<tr>
<td>Σ+</td>
<td>5</td>
<td>5</td>
<td>3</td>
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<td>2</td>
<td>0</td>
<td>DATUM</td>
</tr>
<tr>
<td>ΣS</td>
<td>2</td>
<td>0</td>
<td>4</td>
<td>DATUM</td>
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From the results obtained on the previous page, alternative# 1 seemed to be the best choice for the design approach for this prototype air dryer. Alternative# 2 was close by having the same amount of pluses as alternative# 1, however it concluded with two negatives where as alternative# 1 had two of the sames. The new design would be better off with a couple of sames rather than a couple of negatives. For this design, alternative# 1 was the chosen design concept.

SECTION 3: Technical Analysis

Based on the assumptions made in the earlier section of this report several calculations were further developed to determine the details of each component design that make up the prototype layout. The series of calculations were based on a constant airflow of 5 standard cubic feet per minute (scfm) at a pressure of 55 pounds per square inch (psig). The 55 pounds per square inch specification came from a product information sheet for a particular base coat paint (refer to Appendix M). Most paints are rather similar concerning the rating of pressure to use when spray painting. The 5-scfm assumption was based on a wide range of non-professional spray guns. The analysis calculations have been split up into different sections relating to each of the main components of the design.

Heat Exchanger Tube Length:

To gain a better understanding or visual image of this calculation refer to Appendix N for a complete detailed analysis of the equations and calculations that were made concerning the tube length. To determine the length of the heat exchanger tube various equations from heat transfer methods were compiled and used to calculate the minimum length required to lower the temperature of the compressed air from 120° F to 40° F.

The system was assumed to be a forced convection application where at no time were there any signs of any temperature induced gradients in either the liquid used to cool the system or in the heat exchanger tube itself. The tube would be ultimately exposed to a uniform wall temperature of relatively 32°F. The properties of the air stream were obtained from a handbook using the inlet and outlet average temperature.
From this the amount of heat removed from the air could be determined. However, two additional types of heat needed to be determined. The change in enthalpy from the inlet temperature to the outlet temperature equals the heat removed from the water vapor. And the enthalpy of the outlet temperature multiplied by the theoretical amount of water that was collected equals the amount of heat removed from the water. These three heat rate values were then summed to get a final value of heat removed from the air stream. The log mean temperature difference was then calculated. The Reynolds #, friction factor and nusselt # were also calculated. Using the nusselt#, thermal conductivity of the air and the inner diameter of the pipe, the heat transfer coefficient (h) could be determined.

Then the basic heat transfer equation $Q = h \ A \ \Delta T$ was used in which all values were known except for the surface area which was solved for. Knowing the surface area of the pipe the length was determined. So in order to reduce the inlet temperature of 120°F to an outlet temperature of 40°F under the specified design parameters the length of the pipe should be at least 14.7 feet long. However this is not the final length. Since the coil tube has a specified starting and ending point a helical coil equation was used to determine the final length. Using this equation a total of 10 coils at a 6” pitch diameter was to be used. Performing the rest of the calculations led to a final length of 14.891 feet.

**Heat from atmosphere:**

The heat from the atmosphere that enters the container needed to be calculated next. The processes of both heat convection for the inside and outside container walls and heat conduction through the material were analyzed to calculate the amount of ice that is melted due to the outside temperature conditions. The outside conditions of the container were assumed to be at 90°F with an R-value of 0.25 for moving air in the summer. The inside conditions of the container were assumed to be relatively 32°F with a heat transfer coefficient of 25.78 BTU/hr ft² °F. The container material selected has an R-value of 7 and a thickness dimension of 1 inch. The three equations led to three unknowns in which were solved simultaneously to derive the inside and outside surface temperatures. These two values were then substituted back into any one of the three equations to determine the heat transfer rate through the container. It
was determined on a 90°F day that 66.5 BTU’s/HR was being transferred through the container. A detailed description of the analysis and equations used can be found in Appendix O.

**Required Amount of Ice for Operation:**

To determine the amount of ice required to do the job, the numerical results from the survey responses, which can be found in Appendix E, were used to set a specific time of operation. From the survey results, to apply one base coat of paint it took nearly anywhere from a half an hour to one hour to complete. From this information the intended goal of the prototype air dryer was to deliver dry air for at 2.5 hours, which allowed for multiple coats.

Certain values from the previous sections of calculations were used to determine the amount of ice needed, for example, the amount of heat produced by the compressed air through the heat exchanger tube and the amount of heat transferred through the container was added. The operational time was set at one hour and the enthalpy value of ice was obtained from a handbook. The mass of ice was calculated to be 7.843 pounds per hour. So in order to run the prototype for 2.5 hours approximately 20 pounds of ice was needed. Ice is sold in standard bags of 7 and 22 pounds, which can be found at most grocery stores as well as local gas stations and food marts. So, by using a standard 22-pound bag of ice the prototype will operate properly for 2.8 hours. For further details and information of this calculation refer to Appendix P.

**Container Size:**

To determine the container size in terms of gallons the amount of ice calculated in the previous section must be used. Using the 22 pounds of ice and the density value of ice a value in cubic feet can be calculated and then converted into gallons. However, the value given for the density of ice is in terms of a solid block and the ice being used will be in various sized cubes. A size factor of cube to solid ice was determined in order to relate the two in terms of displaced volume. An assumption of 50% was first used in the earlier calculations. However, an actual test of two 7-pound bags in a 5-gallon water
cooler determined the size factor to be precisely 46%. The cubed ice has a displaced volume of 46% more than solid ice.

Knowing the mass of the ice, density of ice, size coefficient and cubic foot to gallon conversion factor the size of the container was calculated to be 4.197 gallons. A final size of a 5-gallon container would be used in the design. For the prototype model this value was efficient because of its easily availability, however, for the final product the container size should be kept as close to the 4.197 gallons as possible. So perhaps using a 4-gallon container would make more sense in terms of performance and economical means. For more specific information of the container size calculations refer to Appendix Q for a complete analysis of all equations and values used.

**Water Separator Cup Size:**

The water separator cup size is referring to the capacity in fluid ounces of the cup that holds the condensate. To determine the cup size other calculations were first made for example, the amount of water being condensed and removed from the system in a one-hour operating period directly dictated the size of the cup. The prototype in theory was cooling 120 °F compressed air to 40 °F. From this assumption the water vapor content in the air stream was calculated there for deriving the amount of water that was condensed in the process. The calculations state the amount of water removed within a one-hour period was 8.647 ounces. For more specific information concerning these calculations refer to Appendix R for a complete analysis of all equations and values used to determine the cup size of the water separator.

**Expanded Air Properties:**

This calculation refers to the properties of the air just before and after the air is at the expansion nozzle of the spray gun. The compressed air expands rapidly causing atomization of the paint to occur. This atomization process is dependent upon the moisture content of the air as well as the temperature of the air. The desired temperature of the air is to be high enough so that when expanded it does not drop below the dew point therefore condensing water into the air stream. Calculations concerning this can be
found in Appendix S, which is a detailed analysis of the air properties at the expansion chamber.

SECTION 4: Detailing the Design

This portion of the project was based on using the previous calculated design values to create the final design. After the materials were selected, dimensioned detailed drawings were created for each of the component parts including various compression fittings, insulated container, heat exchanger tube, water separator, and sealing washers. A mechanical assembly drawing can be found in Appendix T, which refers each of the individual drawings. The following sections will discuss in detail the selection process of the key components.

Heat Exchanger Tube:

In the previous sections the length of the heat exchanger tube was calculated to be 14.891 feet long. This amount of tubing is a rather large amount of tubing that must somehow be arranged to minimize the space that it takes up. According to other previous sections such as the alternative selection section the arrangement design was determined to be a coil of tubing. This design allowed compacting 14.891 feet of tubing into a 6-inch diameter by 7-inch high volume space. The inlet side would be of course the higher point of the two openings, which in sense gravity is utilized in pulling water down the spiral of tubing and out the exit into a water separator. Refer to Appendix U for a detailed drawing of the heat exchanger tube.

Copper tubing was selected to be the material due to its excessively high heat transfer rate. Type ACR copper tubing was selected which stated by the copper tube handbook is often used in compressed air applications as well as in refrigeration processes. The tube contained a .44-inch inner diameter and a .5-inch outer diameter. This particular specification was chosen due to the size of the airline coming out of the air compressor. More times than often the average size outlet port on an air compressor will be 0.5 inches. The rule of thumb in compressed air line plumbing is to be consistent throughout the system with the same size lines to reduce pressure drop. For more
information concerning the specifications of the copper tube chosen for this design refer to Appendix V.

**Container:**

The size of the container was already calculated in previous sections in which a 5-gallon size was chosen. However, the style and type was primarily based on the influential design factors stated in the beginning of the design process. The container needed to be small, appealing, durable, safe and lightweight. In considering these characteristics the best place to begin the search was a particular style of water cooler. A water cooler serves as a great container that will insulate the ice on the inside to maximize the time of operation. It is also small in size, appealing to the eye, lightweight in structure and safely built to specific design standards.

From this information, a 400 series igloo cooler model number 451-5 gallon was chosen as the container to use in the design. It is a 5-gallon size capacity container with side handles and a pressure fit lid. A lid keeper chord is included to ensure it from ever being lost. The container is durable in construction with two layers of polyethylene material and a layer of polyurethane foam insulation between them. This provides adequate insulation for the prototype to work properly. The thermal conductivity value of polyurethane foam was determined to be 0.14285. This value is the inverse of the R-value of polyurethane foam, which is a 7 to 10 R. Refer to Appendix W for specifications on polyurethane foam. For more specific information on this particular container refer to Appendix X for products specifications or Appendix Y for a general AutoCAD drawing.

**Water Separator/Filter:**

Previous calculations such as the cup size dictated the selection of a water separator or filter suitable for this application. The previous calculations stated a cup size of roughly 8.647 fluid ounces. However, due to high cost a filter with a smaller cup size of 7 fluid ounces was chosen instead. A Campbell Hausfeld filter number PA2121 was selected. This component incorporates a 5-micron filter that removes dirt particles and condensed water. The filter is equipped with a see through bowl for easy viewing with a
metal guard around it to protect it from impact. The bowl can easily be taken off for discarding collected waste and is equipped with a quarter turn valve to drain the bowl frequently. For more information or specifications concerning this product refer to Appendix Z.

**Compression Fittings & Seals:**

The selection of compression fittings came from both product catalogues of Campbell Hausfeld and Parker Pneumatics. To implement a sufficient seal at the sidewalls of the container bulkhead fittings were selected. From the Parker Pneumatic company bulk head fittings include both external and internal National Pipe Threads with a self-locking washer and nut to ensure proper sealing. A .375-inch to .25-inch reducer adaptor and .375-inch by 2 inches long pipe nipple were also selected to allow for proper assembly. These fittings were assembled on the outer side of the container while the inside contains 45° flared fittings. These are .5 inch to .375-inch male fittings at a 45° angle. The 45° specifications refer to the angle of the flared end. A short flaring nut was used to tighten the tube to the fitting. More specific information and specifications on all of these fittings that were used can be found in Appendix AA.

From Campbell Hausfeld, two components were selected based upon the customers needs. From the House of Quality in previous sections it was noted that quick disconnect couplings used for easy assemble and disassemble were a customers high concern. To meet the request of the customers an industrial style .25 inch disconnect coupler and plug were selected. Specific information concerning these disconnect couplings can be found in Appendix AB.

Both the inlet and outlet container ports are potential areas of leaking water therefore needing special seals to provide a tight fit against the inner wall of the container. The seals for the design were selected from the McMaster-Carr catalogue. These seals consisted of a neoprene-bonded washer in which the neoprene rubber material is rigidly attached to a galvanized steel washer. This type of seal will provide a sufficient seal therefore avoiding leaks. More information about this style of bonded washer can be found in Appendix AC.
SECTION 5: Build & Test

Fabrication & Assembly:

The next step was to build a working prototype model. The coiled tube heat exchanger was first constructed. This was achieved by wrapping it around the appropriate size piece of round steel stock. Each of the ends then needed to be flared in order to mate up with the flare fittings. A flared fitting was chosen due to its high sealing capabilities in which the tube was to be immersed in ice water. This was achieved by using a small hand-flaring tool.

Next the container selected for the design needed to be adjusted. Both the inlet and outlet ports were drilled through the container's walls. A 1 ¾” inch hole was drilled through the outer wall and polyurethane insulation. A 1” hole was drilled through inner wall. This configuration allowed the brass bulkhead fitting to be rigidly attached.

Once fabrication was completed the prototype was assembled. The coiled tube heat exchanger and all necessary fittings were assembled outside the container and then placed into the container where a nut and lock washer were placed on the bulkhead fittings and gently tightened down. Next the water separator was attached completing the assembly. Refer to Appendix AD for an assembly schematic of the prototype.

Testing the Prototype Design:

Once the assembly was completed, the testing phase began. In order to test the prototype certain environmental conditions were first applied. For example, the prototype was to be tested precisely in a 90°F controlled environment with a level of 85 percent humidity. A greenhouse was used to replicate these extreme conditions. For clarification a detailed schematic showing the complete test set-up can be found in Appendix AE.

The use of both a 20 gallon capacity, 5 horsepower air compressor provided by Campbell Hausfeld was used as the air source. An airline hose ½ inch in nominal diameter and 6 feet in length was connected to the prototype and to the air compressor. A 25-foot air hose with ½ inch nominal diameter was connected to the outlet of the prototype. At the end of that air hose is where the paint gun was attached. Between the
coupler of the hose and the spray gun a flow valve was used. This valve helped to control the pressure at the desired level, in this case at the maximum of 45 psi. A dew point monitor was installed directly after the outlet of the prototype to carefully monitor the dew point range of the system.

Several thermal couples were used in various places to obtain useful data for experimental calculations. For instance, a thermal couple was placed on the inside and outside of the container to accurately measure the amount of heat being transferred through the container. Other thermal couples were placed on the wall surface of the heat exchanger to measure the wall temperature. Temperature data was also gathered using temperature probes in which were placed at both the inlet and outlet of the prototype as well as at the spray gun and flow meter.

All measurements were recorded every 5 minutes for one hour. All of the information and specifications concerning all of the equipment and instruments used in the testing procedure can be found in Appendix AF.

**Experimental Test Results:**

The proof of design testing was conducted in a green house where a high temperature and humidity range could be achieved. The air pressure through the air dryer was carefully maintained at 55 psig. An inlet temperature of 114°F was achieved and reduced down to 42°F. The air properties at the spray gun expansion valve were measured using a sling psychrometer, which obtained the dry and wet bulb temperatures of the air stream. A pressure dew point of 52°F was achieved. The airflow was calculated to be 6.708 actual cubic feet per minute, which was slightly higher than the design value of 5 scfm. This was due to the fact that in order to achieve the desired pressure of 45 psig at the spray gun the flow was unable to be restricted to the design value of 5 scfm. Temperature values received from the thermal couples on the heat exchanger exceeded the assumed constant wall temperature of 32°F. These values led to the source of error in which is discussed in the next section. For further detail the proof of design test results can be found in Appendix AG.
Section 6: Final Analysis

Material Cost:
- Insulated Container $32.00
- Water Separator $24.00
- Copper Tubing $16.00
- Various Fittings $24.00
- Sealing Washers $2.50

The total cost of the prototype design was calculated to be $98.50.

Efficiency Rating:

To determine the efficiency of the prototype, the results from the proof of design test were used to calculate the true heat transfer coefficient value and the design values were used to determine what the h-value needed to be in order for the prototype to meet up to the designed parameters. The efficiency of the prototype was calculated to be 93.92 percent. The reason being is that in the beginning stages of the project an assumption of the constant wall temperature of the heat exchanger was to be 32°F. From the experimental test data the wall temperature was found to be not constant.

Humidity Level:

The humidity level of the compressed air as it exits the spray gun expansion valve was calculated using two different methods. Method one uses the pressure dew point value achieved in the testing procedures along with the gauge pressure of the prototype to determine the atmospheric dew point of the air stream using a dew point conversion chart. Then the dry bulb temperature of the exiting air along with the atmospheric dew point was used to calculate the percent level of relative humidity in the compressed air. These calculations can be found in Appendix AH.

Method two used both the wet and dry bulb temperatures that were measured in the proof of design test. These values were used to graphically determine the percent level of humidity in the compressed air using a psychometric chart. These calculations can be found in Appendix AI.
**Recommendations:**

In order to take this prototype air dryer to the next stage and introduce it to the market a punch list of several recommendations was generated. Further testing and analysis needs to be performed in order to cover a wide range of air compressor types and sizes. Unnecessary fittings should be eliminated to reduce cost and increase profit. A new container design needs to be sought. High areas of concern are the holes for the inlet and outlet that are to be drilled through the sidewalls therefore having a large stress concentration in that particular area. These walls will perhaps take an every day abuse due to the location of the disconnect couplings. Moreover, the water separator will be hanging in mid air with only it being attached through the bulkhead fitting therefore causing high stress on the inner wall at that point. For the final product a special designed container would meet these specific requirements to allow for a better sealing and durability design. The lid is currently a pressure fit style in which brings up safety concerns if air were to leak inside the container which could blow the lid off causing operator injury. An alternative water separator could offer a larger cup size to collect the discarded water or perhaps even contain an automatic drain.

**Accomplishments and Conclusions:**

The intended goal of this project has been successfully met in designing a working prototype model of a compressed air drying system that is both economical and adequate to similar equipment. The prototype consisted of a 93.92 percent efficiency rating. It successfully reduced high levels of humidity in the compressed air to well below 20%. And it successfully met customer requirements.
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AIR DRYERS

Note: Figures and charts referred to in this section will be found at the end of the text.

Almost everyone is aware that our atmosphere has the capacity to hold and move water from one place to another, and all atmospheric air has some water vapor content. This is because of solar evaporation, and in the subsequent transfer and condensation, vast quantities of water are freighted from one part of the earth to another. As a matter of fact, there are, at any given moment, over 3,100 cubic miles of water held in the atmosphere in the form of vapor, ice, and liquid particles. The moderation of earthly temperatures and the life support resulting from the capacity of air to hold water are vital to our survival.

Air, when compressed, represents an industrial power source which grows in importance annually. Ever since widespread application of compressed air came into common industrial usage, the one single drawback has been the contamination common to all compressed air systems. These contaminants fall into 3 major categories: 1) solid dirt and pipe scale particles, 2) oily aerosols, and 3) liquid water. While porous oil and water can constitute a serious contamination in certain hypercritical air applications, air can generally be considered clean and dry when moist of the solids have been removed and the system is free of solid oil and water (see section on filters).

Here, it is well to take a moment and consider that any substance can exist in 3 states: solid, liquid, and gaseous, with water being the most common example. It is particularly important in discussing water content of air that you always differentiate between water vapor, liquid water, and ice. Keep in mind that air dryers are designed to remove only the water vapor. Liquid water can easily be separated from the airstream by prefilters, and to introduce liquid water into an air dryer simply puts unnecessary and detrimental loads on the dryer.

Oil-removing filters are available, and one can remove virtually all of the solid and oily contamination from the compressed air. However, water vapor which cannot be filtered from the air is an inherent part of the intake air, and the subsequent condensation in the system at a controlled point, centrally located, has great appeal to industry and is the impetus of an ever-increasing sales volume of compressed air dryers.

To more effectively sell these dryers it is necessary to understand some of the basic physics involved. Air can be dried in 2 principal ways: mechanically and chemically. We will take these in order.

MECHANICAL AIR DRYERS

When air is mechanically compressed, the mere process of elevating the pressure tends to condense a portion of the water vapor present in the air. This is not always the case (affecting factors being initial relative humidity, temperature, and the final elevated pressure); however, in most cases, a portion of the water vapor in the air will condense when elevated to normal in-plant pressure starting from average ambient conditions.

A discussion of this will help to clarify this point. Please look at Figure 1. This is a chart showing water vapor content of air per cubic foot at various temperatures and percentages of saturation. Note that the ability of air to hold water vapor approximately doubles or is reduced by half with each 20 degrees Fahrenheit increase or decrease in temperature.

Let's take a cubic foot of atmospheric air at 60% relative humidity and 70°F. This represents a fairly common ambient condition. Under these circumstances, each cubic foot of air contains 4.79 grains of water vapor. When air is compressed to 100 PSIG, you must "crowd" 7.8 cubic feet of atmospheric air into a one-cubic-foot volume. This ratio is called the compression ratio and is found by dividing the absolute pressure of the air by atmospheric pressure: \( \text{PSIA} \times \frac{\text{atmospheric pressure}}{\text{atmospheric pressure}} \). Absolute pressure is the sum of gauge pressure and atmospheric pressure. Therefore, the compression ratio for 100 PSIG at sea level is 114.69 + 14.69 = 7.8. See Figure 2. Along with all the other gases in the air, we've also ended up with 7.8 times the original amount of water gas (or vapor) into this cubic foot volume. This cubic foot volume now has 7.8 x 4.79 grains of water vapor content, or 37.36 grains.

A brief aside is necessary here. Whenever air is compressed, a significant amount of the energy for compression is converted to heat. This is the heat of compression. Also, with the typical compressor, varying amounts of heat are added to the air through friction. In any case, you cannot avoid generating heat when you compress the air. This heat, of course, affects the capacity of the air to hold water vapor, but does not bear on the amount of water vapor in the compressed
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air, because once compressed, the air is in a closed system with no source from which to acquire additional water.

Discharge temperatures out of the typical industrial compressor will be in the 200°F to 400°F range. Some type of heat exchanger is (or should be) incorporated in the design of most compressed air systems. Typically, the temperature of the compressed air at the storage tank is in the 100°F to 120°F range. The reason for this higher-than-ambient temperature is that aftercoolers are not designed to bring the air temperature back to ambient, and the air will not normally remain in the tank long enough to return to surrounding temperatures.

At the relatively low pressure typical of in-plant industrial compressed air systems, we can disregard the pressure factor when trying to determine how much water vapor a volume of air will hold and need only consider the temperature, this is to say that whether the gauge reads zero or 300 PSI, for example, the water vapor capacity for all practical purposes is shown in Figure 1. This is implicit in Dalton’s law of partial pressures.

Now let’s return to our example. We have 37.36 grains of water vapor in every cubic foot of air at 100 PSIG. Since the air will not hold any water beyond saturation (100% relative humidity) and since we need only consider temperature, look again at Figure 1 in the 100% relative humidity column and see how much water vapor a cubic foot will hold at 100°F. Answer: 19.77 grains. How much water vapor is the air trying to hold? 37.36 grains. Subtract 19.77 from 37.36 to find how many grains of water will condense: 17.59 grains will condense - this amount for every 7.8 cubic feet of free air taken into the compressor under the ambient conditions described. What we’ve done here, through pressure and the inevitable cooling of the air back to typical receiver tank temperatures, is force the air to, and beyond its saturation point. In the process, we have condensed approximately one half of the water vapor that was originally in the air. If this liquid water is drained from the system, the air will remain “drier” to this extent. I put the word “drier” in quotes, the reason being that it is only “drier” in relation to its ambient condition. Please note that, as long as it remains at the pressure (100 PSIG) and at the temperature (100°F) it will be saturated, or at 100% relative humidity. Whatever temperature represents the saturation point of the air, that is also the air’s dew point. In this case, the dew point is 100°F. This is a pressure dew point.

If we allow the air to expand back to atmospheric pressure (see Figure 3) and keep the air in a confined container so it can’t pick up additional water vapor, we find that the volume will increase 7.8 times. Now the 19.77 grains of water vapor must be distributed (diluted) equally throughout a volume 7.8 times as large, which means that each cubic foot of “expanded” air now contains only 2.53 grains of water vapor (19.77 + 7.8).

Look at Figure 1 and read down the 100% column. (This is the dew point column.) Find at what temperature 2.53 grains represents saturation or 100% R.H. You will find it to be about 57°F – this is the air’s atmospheric dew point. You will always find that atmospheric dew points will be lower than pressure dew points. Actually, one way to dry air is to reduce the pressure. Pressure reduction is an expansion process and given a fixed amount of water vapor as the air expands the vapor content is continually diluted per unit volume of air.

The air is now in the compressor storage tank at 100 PSIG and at 100°F. It is holding all the water vapor it can hold under these conditions and has a pressure dew point of 100°F. It also represents potential energy. To be transformed into useful kinetic energy, it must be piped to various more convenient points of use. Figure 4 shows a piping system, indicating more or less typical temperature differences; varying temperatures being common to all compressed air systems. There are unavoidable temperature drops through conduction and convection as the air flows from point to point. At Point A, the air has lost some of its heat content; its temperature is now 90°F, and some of the water vapor has condensed into liquid. The dew point is now 90°F.

The air at Point B has cooled to 70°F. More vapor has condensed. As a matter of fact, Figure 1 will show that it has only about half as much water vapor as it had at 90°F. The dew point is now 70°F.

Point C is the same as Point B. Keep in mind that the liquid water that has condensed between Points A, B, and C should be drained out of the system; otherwise, if for any reason the temperatures at Points B and C should increase back to 90°F, all the liquid water would be evaporated back into the compressed air.

At Point D, the air is piped over a furnace and is heated to 90°F. Here, by imparting heat energy into the air, we give the air a greater capacity for water vapor but, in fact, deny the air the actual water vapor. Therefore, the dew point remains the same (that is, its water vapor content remains the same), but the relative humidity drops from 100% to about 50%. In other words, the amount of water vapor in the air relative to its capacity is about 50%.

At Point E we cool the compressed air, bringing its temperature down to 55°F. It now has a pressure dew point of 55°F.

The air moves to Point F and is reheated back to
70°F. Again, we give the air a capacity for water vapor beyond its actual content; and, again, we lower its relative humidity to almost 61%; but the pressure dew point remains the same: 55°F.

To see how we arrive at these lower relative humidity figures, let's look at Figure 4. At Point E we have a dew point of 55°F, and Figure 1 shows saturation at 55°F to be 4.85 grains. At point F the water vapor content remains the same, but we reheat the air to 70°F. In Figure 1 see that at 70°F the air has a maximum capacity of 7.8 grains. Relative humidity is the ratio between the water vapor the air is actually holding (4.85 grains) and its total capacity (7.98 grains): 4.85 / 7.98 = 60.7% relative humidity.

Keep in mind that wherever there is a temperature drop from one point to another, condensation will occur and liquid water will appear. Most piping systems are extremely complex in pipe size, temperature variations, piping configurations and so forth. Water does, in fact, condense and collect in scores of places, requiring the installation of water separators, sump drains, drain legs, etc., and the need for a constant maintenance routine.

Take a look at Figure 5. Here, we have installed a compressed air dryer immediately after the storage tank. In this case, we have installed a mechanical refrigeration-type dryer. By a mechanical refrigeration process, we cool the air to a 35°F to 39°F pressure dew point. In this illustration, this is as cool or cooler than any subsequent ambient condition. No water can condense beyond the dryer, because the air is never subjected to temperatures below 55°F, and its relative humidity is always less than 100%. On a new system this can eliminate the need for individual separators, etc., because there is simply no liquid water condensing in the system beyond the dryer. This is the great appeal for central dryers; not only is the entire piping system dry, but maintenance has only one unit to maintain instead of many. It should be noted that on an existing system, filters, separators, drains, and drain legs should be left in place. While there should be no water to drain, there will be residual contamination which will take time to be purged from the system after the dryer is installed.

These then are the two common methods of mechanically condensing water vapor: mechanically pressurizing the air and mechanically cooling the air with refrigeration-type dryers.

**THE REFRIGERATION CIRCUIT**

There is nothing new about mechanical refrigeration. In the opening decades of this century, refrigeration was in wide use to manufacture ice. The principles of mechanical refrigeration were the same then as now, and whether the refrigerator is a 500 HP unit for cooling an office building or a small compressed air dryer, their operation is basically the same.

Refrigeration is based on the physical fact that it requires heat to evaporate a liquid. This same heat is given up when the vapor condenses. The controlled evaporation and subsequent condensation of any one of several commonly used fluids is the basis of all refrigeration systems. Most people, when they hear the word refrigeration, think of "cold" or "cooling." However, the practice of refrigeration is the transfer of heat; and, even though we may use these subjective terms, the technology of refrigeration is the moving of heat from one place to another.

Thermodynamics is that branch of science dealing with the mechanical action of heat: a study of the "flow" of heat from one place to another. Certain of the laws of thermodynamics is basic to refrigeration. Heat is a fundamental form of energy resulting from the transformation of other kinds of energy. The friction of moving parts in all kinds of machinery causes heat. The flow of electricity generates heat as well as all kinds of chemical reactions, which in turn often generate heat. The heat of compression as the air pressure is elevated by a compressor is one of the sources of heat with which an air dryer is involved.

Heat always flows from that thing or place which is warmer (higher temperature) to that which is colder (lower temperature). If we were to describe that which is warmer as the "top of a hill" and that which is colder as the "bottom of a hill", heat will always spontaneously flow downhill. The greater the difference between the hot and the cold, the "steeper" the hill and the "faster" (more efficient) will be the heat flow (or transfer). This temperature differential is abbreviated as A T.

At this point, let's make a distinction between heat and temperature. In the final analysis, temperature is a measure of molecular or atomic movement — how fast the particles of a substance are traveling or vibrating. (Absolute zero is a theoretical condition where you have no molecular motion.) With any molecular motion, you have a measurable temperature and there is heat involved. Heat is the amount of this kind of energy in a system, while temperature is the intensity of the heat. For example, a tub of lukewarm water will contain a surprising amount of heat, but have a relatively low temperature. Conversely, a small ball bearing can be white-hot (be at a very high temperature) but have little total heat content, because there is very little substance in this small ball bearing to hold a great amount of heat. Nevertheless, regardless of total
heat content, if the high temperature ball is plunged
into the tub of water, the flow of heat will be from the
ball to the water. The water temperature would in-
crease an imperceptible amount, and the ball would be
cooled to the temperature of the water.

Devices for measuring temperature (not amounts of heat) are thermometers calibrated on two commonly
used scales: Fahrenheit and Celsius. The Fahrenheit
scale sets the melting point of pure ice at 32° above zero
degrees (at sea level) and the boiling point at 180
calibrations above 32°, or at 212° above zero.

The Celsius thermometer sets the melting point of
ice at 0° and the boiling point of water at 100°. Anders
Celsius, a Swedish astronomer, invented the celsius scale which until recently was called the Centigrade
(100 calibrations) thermometer. The Fahrenheit scale is
not used outside of America and Canada. The Celsius
scale is used by technical people in all walks of science
throughout the world. Another temperature scale wide-
ly used by scientists is the Kelvin scale. Lord Kelvin, a
19th century British physicist, established this abso-
lute scale and based his calibration on the Celsius scale,
but set 0° Kelvin at absolute zero.* The various laws
describing the behavior of air are based on absolute
temperatures -- and pressures. 0°K is equivalent
to -273.18°C.

Before we get into the refrigeration circuit, let's
become familiar with some terms and physics which
bear on the subject:

Heat Flow (Transfer): Heat flows in 3 basic
ways:
1. Conduction is the flow of heat through a substance.
   Physical contact is required for heat to flow by conduction.
2. Convection is the transfer of heat by a moving fluid
carrier, either gas or liquid.
3. Radiation is the transfer of heat across an empty
space or vacuum by waves similar to light waves or
radio waves.

Sensible Heat: That heat which can be sensed and
results in a change of temperature.

Latent Heat: "Latent" is derived from a Latin
word meaning hidden. Latent heat is that heat which
does not bring about a change of temperature. The
Latent Heat of Vaporization is the heat required to
boil a liquid. Assuming a given pressure, any liquid will
boil at some maximum constant temperature. For
example, at sea level liquid water boils at a constant
temperature of 212°F (100°C). One cannot get the
temperature of water any higher than this at sea level
pressure. There is also the Latent Heat of Fusion which
is the heat required to transform a solid to the liquid
state (melting). Again, this occurs at a constant tem-
perature -- in the case of water, at 32°F (0°C).

Super Heat: The heat over and above that required
to change a substance state from liquid to gaseous.
This is also sensible heat, because it shows as a change
of temperature.

BTU: These are initials for British Thermal Unit
which is the amount of heat required to raise the
information of one pound of water one degree
Fahrenheit. Figure 6 shows the BTU's involved in
bringing one pound of water from ice at -40°F to a
gaseous state at 212°F. Note: From -40°F to ice at 32°F
requires 36 BTU's; the latent heat of fusion is another
144 BTU's. The sensible heat required to bring the
water to boiling temperature is 180 BTU's. The amount
of latent heat for vaporization -- a full 970 BTU's! The
fact that it takes substantially more heat for the vapor-
ization cycle with any substance is important in the
refrigeration process: to wit, the relatively large am-
mounts of heat involved in the vaporization/condensa-
tion cycle.

An aside here: One often hears the term a "ton of
refrigeration." This is the amount of heat required to
melt a ton of ice: 288,000 BTU's. To melt a ton of ice in
24 hours is in turn 12,000 BTU's per hour (BTU/H).
Depending upon temperatures, heat transfer efficiencies
and other factors, a ton of refrigeration is approxi-
mately equal to 1 HP.

Figure 7 shows the BTU's required to evaporate a
pound of refrigerant 12 at 33 PSI-G which is more or
less a typical evaporating temperature of R12 when
used in a refrigerated air dryer. You will note that
capacity of R12 to absorb heat is substantially less than
that of water. However, compensating for this are
other advantageous chemical properties of R12 discussed
below.

General (Perfect) Gas Law: This describes the
relationship between the volume, pressure, and tem-
perature of a gas. This law assumes, among other
things, that the atoms and molecules making up the gas
are without mass or dimension. While the perfect gas
does not exist, this law is endlessly useful as a first
approximation to the real gas. For all practical pur-
poses, air and super-heated refrigerant gas at typical
pressures and temperatures may be considered perfect
gases, and their behavior can be expressed as: PV =
MRT where "P" is absolute pressure, "V" is volume,
"M" is mass, "T" is absolute temperature, and "R" is a gas constant.

**Vaporization/Condensation, Pressure/Temperature Relationships for Fluids:** It requires a fixed amount of heat to vaporize a given amount of liquid. This same fixed amount of heat is given up when the vapor condenses back to liquid. This exchange of heat, again, is the basis of mechanical refrigeration. As stated earlier, a mechanical refrigerator is a machine which controls the evaporation and condensation of a suitable fluid.

A vaporization/condensation curve can be plotted for any fluid, and the specific temperatures at which this occurs will vary with the chemical properties of the fluid and the pressure on the fluid. For example, we all know that water will boil or condense at 212°F at sea level pressure. Reduce the pressure and water will boil/condense at a lower temperature. In Denver this temperature is 202°F because of the lower atmospheric pressure. Home pressure cookers are used to get higher water temperatures for cooking of foods more quickly. Here a sealed lid, safety valve, etc., bring the pressure within the vessel to 15 PSIG (or about 30 PSIA) at which pressure the water temperature can be raised to 260°F.

Figure 8 shows these relationships for water as well as three commonly used refrigerants: R12, R22, and ammonia. At any particular point on these curves the fluid is either vaporizing or condensing, depending on whether heat is being added to or withdrawn from the fluid. A mechanical refrigerator, by manipulating the pressure of the refrigerant, can cause the fluid to absorb heat (evaporate) at a low temperature and then condense and give up that same heat at a higher temperature. The refrigerant compressor raises the pressure of the refrigerant, and the expansion valve lowers the pressure.

Refrigerant 12 and 22 are widely used in compressed air dryers. Their chemical names are Dichlorodifluoro Methane and Monochlorodifluoro Methane respectively, but are best known as Freon 12 and Freon 22. (Freon is a trade name of the DuPont Corporation.) These hydrocarbons meet all the requirements of a good refrigerant: They will vaporize and condense at pressures and temperatures that are economically achieved and, in addition, are safe (nontoxic, nonflammable), odorless, colorless, noncorrosive, and are relatively inexpensive.

Ammonia is a good refrigerant and continues to be used in large refrigeration systems because of its very low cost. Ammonia (NH3) is a pungent, toxic gas, but note the similarity of the ammonia and R22 curves.

Figure 9 shows a schematic of a typical refrigeration circuit common to compressed air dryers. The graphs above and below the schematics show the approximate temperature and pressure of the refrigerant as the fluid moves through the circuit from one component to another. There will, of course, be variation and differences depending on manufacturer, size of units, etc. For example, dryers using R12 do not normally have an accumulator in the circuit nor a fan control. Keep in mind that this is a closed circuit with the refrigerant moving round and round from the compressor back to the compressor. Because it is a closed circuit, one can start at any place in the circuit to describe its function.

Let us start at the Receiver. The name is derived from the fact the component receives liquid refrigerant from the condenser. Figure 10 shows a cutaway drawing of a typical receiver. A portion of the refrigerant is condensed in the condenser and flows into the receiver at a relatively high temperature and pressure, and enters the top of the receiver as a spray of liquid and gas. A dip tube assures that only liquid refrigerant flows out of the receiver.

**Filter/Dryer.** These are components, sometimes housed in the same canister, through which the liquid refrigerant flows, and their function is to filter extraneous solids and dirt from the refrigerant (such as residual solder flash) inadvertently left in the circuit during manufacture. The dryer section of the canister is a water vapor desiccant to capture and hold any water that might have been in the circuit prior to charging with the refrigerant. Incorporated in the larger dryers is a Sight Glass. This is a device for visually monitoring the flow of the refrigerant. (Chronic "bubbling" at the sight glass, for example, usually indicates an undercharge of refrigerant.)

**The T.E.V. (Thermal Expansion Valve).** Note in Figure 9 the sharp drop in pressure and temperature as the liquid refrigerant passes across the expansion valve. The expansion valve is primarily a temperature-sensing device with a modulating orifice across which the liquid refrigerant flows. At the orifice a portion of the liquid is flash vaporized. The heat for this vaporization is drawn from the remaining liquid. There is now a relatively cool liquid at a pressure of around 33 PSIG flowing into the evaporator. (Refer to Figure 8 and note that at this pressure R12 will vaporize at a constant temperature of about 33°F.) The Evaporator is one of three heat exchangers in the dryer. The compressed air gives up heat to evaporate the refrigerant and, assuming enough refrigerant and time, will continue to give up heat until the compressed air temperature approaches the vaporization temperature of the refrigerant. This heat transfer is through conduction and
convection. Virtually all the heat absorbed by the refrigerant from the compressed air is latent heat and does not show as a temperature rise. Remember, any liquid evaporates at a constant temperature. In the larger dryers this latent heat of evaporation drawn out of the compressed air can be in the hundreds of thousands of BTU's per hour.

To prevent liquid refrigerant from passing all the way through the evaporator into the refrigerant compressor is the primary function of the thermal expansion valve. It does this by maintaining about 10° of super heat at the evaporator discharge. It operates as follows (see Figure 11): A small I.D. tube connects the thermal bulb to the top of the diaphragm in the valve. The bulb is in firm contact with the metal at the discharge side of the evaporator. The bulb and tube and the area at the top of the diaphragm are factory charged and sealed by the valve manufacturer with the same refrigerant used in the circuit. The valve is selected and adjusted to work in conjunction with the specific design of the dryer. As the temperature of the refrigerant gas drops at the evaporator outlet, the refrigerant gas in the thermal bulb contracts allowing the orifice to close. With less refrigerant flowing into the evaporator, there is less cooling, and the refrigerant temperature rises; the gas in the thermal bulb expands causing the orifice to open; more refrigerant flows into the evaporator with more cooling effect, etc., etc. As stated, the expansion valve is selected and set to maintain about 10° super heat at the outlet of the evaporator. Keep in mind that, while the actual evaporating temperature could be adjusted up or down the graph (Figure 9), the valve's primary function is to maintain super heat, whatever the specific evaporating temperature may be.

Hot-Gas Bypass Valve. This is the component selected by most dryer manufacturers to control the specific temperatures desired. It is a pressure-sensitive device and modulates the flow of hot refrigerant gas to bypass the expansion valve. This hot-gas flow will vary from zero to virtually all the refrigerant, depending on the heat load in the evaporator. (“Heat load” in dryers is another expression for compressed airflow; the unit is a spring-loaded, diaphragm-type, pressure-regulating valve (not unlike a compressed air regulator). The valve is pilot controlled by the changing pressure on the low side of the refrigerant circuit and is in the circuit to keep the suction (low side) pressure fairly constant. The modulating suction pressure, of course, results in modulating the temperature as well as the flow of the hot gas.

Controlling the temperature in a compressed air dryer is extremely important. For this reason, a pressure-sensitive device is most often used, because pressure signals result in a much faster response than a temperature signal. Designers must avoid freezing the water that is condensing in the compressed air side of the evaporator. A freeze-up can rapidly damage the evaporator or cause a severe restriction of compressed airflow resulting in costly downtime and loss of production.

Figure 12 is a cutaway drawing of a typical bypass valve. If the pilot pressure (low side pressure) begins to fall, indicating a matching fall in temperature, the valve responds and allows hot gas to dump directly into the suction side of the refrigeration circuit. As the low side pressure rises, indicating a matching rise in temperature, the valve reduces the flow of hot gas. The designer will set the valve to be at zero bypass flow at the standard-rated conditions of the dryer. (Please remember that the evaporator is on the suction side of the circuit.)

Some manufacturers use a temperature-sensitive device to control specific evaporator temperature. This is simply a thermal bulb attached to a selected section of the evaporator interfacing with an electric switch to shut off the refrigerant compressor. While this is a much less expensive control, most manufacturers avoid this method because of the relatively wide temperature (dew point) fluctuations with this type of control.

Figure 13 shows a typical performance curve of the hot-gas bypass valve. The valve can be adjusted to any desired temperature range, but the curve will remain essentially the same. The solid line is the hot-gas bypass flow curve, and the dotted line is a typical temperature approach curve of the compressed air from zero to rated flow.

The effectiveness of any heat exchanger is expressed by the term “approach.” This is how close in temperature the compressed air approaches the evaporating temperature of the refrigerant. Because of unavoidable inefficiencies, the outer surface temperature of the refrigerant tubing will be higher by a few degrees to that of the actual evaporating temperature. There is also a cost as well as a size limit to the amount of cooling surface a designer is allowed to put into the evaporator. These physical as well as economical restrictions result in an optimum approach of around 7° under current competitively-rated specifications. That is, if the vaporization temperature of the refrigerant is, say, 33°F, the surface temperature of the tubes carrying the refrigerant will be 35°F with an approach of 7° resulting in the compressed air being chilled to 38° to 40°F as it leaves the evaporator. As you can see in the chart, the approach widens as the heat load increases. This increase of approach simply cannot be avoided.
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**ITEMIZED TASK**

- Design Research
- Preliminary Budget
- Project Schedule
- Survey created
- Survey info received
- QFD chart generated
- Proof of Design
- Proposal Due
- Design Research
- Design Concepts
- Weighted Objective Selection
- Detailed Design - Mechanical
- Detailed Design - Thermo
- Component Layout
- Plan for Design/Build
- Final Design Review
- Interim Design Report

*Design Freeze*

- Interim Design Report Due
- Order Final Parts
- Fabricate Parts
- Assemble Fixture
- Test of Product
- Write Report
- Oral Presentation
- Tech Expo
- Graduation

**APPENDIX C**
### Appendix D

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**Total Dollar Amount:** $195.00

### Up-To-Date Budget

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**Total Dollar Amount:** $94.59
Appendix E

User Survey – Spray Painting

The following survey is being conducted to collect customer data in aspects of the spray-painting process and air-drying equipment. This information will be used to design a prototype model of an air drying device in conjunction with a air compressed system.

1.) Have you ever used a paint sprayer with an air-compressed system?

If no, disregard the rest of the survey and please list any contact information, such as name, phone number and email of any friends and family that have had previous experience in spray painting.

2.) How often do you spray paint? (e.g. once a year, twice a month)

3.) What type, model and/or style of air compressor do you use?

4.) Do you use a water catcher/separator to collect the moisture and condensation buildup?

5.) Do you use any additional equipment or air drying devices such as coalescing and desiccant air drying filters?

6.) About how much did the equipment and items from questions 4 and 5 cost total?

7.) Do you think the dollar amount in question 6 is relatively cheap or expensive?

8.) Are these devices easy to use and control?

9.) Have you ever encountered troubles with water inside the air lines when spray painting?

10.) How often does water become an issue? (Every time you spray paint, or every other time)
Please rate the following characteristics in terms of an air-drying device.
Rating scale: NI – Not Important, I – Important, EI – Extremely Important

Quick Set-Up
Portability
Storage
Durability
Ease of Serviceability

12.) In your own opinion, how long in terms of minutes or hours does it take to apply one coat of paint considering the surface has been prepared prior to painting.

13.) Please List any other characteristics that you feel are important qualities concerning an air-drying device.
Appendix F

Numerical Results from Survey

The following results were obtained from the survey and summarized in the following columns.
A total of 10 different people were surveyed. 5 average consumers & 5 body shop employees

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<td>Ease of Serviceability</td>
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# Appendix G

## HOUSE OF QUALITY

9 = Strong  
3 = Moderate  
1 = Weak

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<th>Servicability - Ease of Use</th>
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<td>3</td>
<td>9</td>
<td>9</td>
<td>3</td>
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<td>3</td>
<td>9</td>
<td>9</td>
<td>3</td>
<td>1.7</td>
<td>0.12</td>
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<td>2</td>
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</tr>
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<td>9</td>
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<td>9</td>
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<td>1.7</td>
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<td>9</td>
<td>3</td>
<td>3</td>
<td>3</td>
<td>1.7</td>
<td>0.12</td>
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<td>5</td>
<td>5</td>
<td>5</td>
<td>5.5</td>
<td>0.21</td>
<td>5</td>
<td>2</td>
<td>5</td>
<td>lbs</td>
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</table>

**UNITS**

- Inches
- Inches
- Lbs/cubic in.
- Lbs/cubic in.
- Gallons
- Btu/hr
- Minutes
- Lbs
- Qty. & size

**TARGET VALUE**

- 20
- 0.5
- Copper
- Plastic
- 30
- 0.5
Appendix H

Air Dryer Flow Chart/Block Diagram

Compressed warm to hot air saturated with water

Flows from tank & thru system

Ice

Surrounds system & cools air

Condensation

Forms within the system

Water

Develops from condensation

Compressed cold air combined with water

Exits the system

Air/Water mixture

Enters water separator

Water

Collected & discarded

Compressed air

Exits the separator

Dry air

Delivered to end of system
Appendix L
# Appendix M

## PRODUCT INFORMATION

### MBC Acrylic Basecoat

**Background**
MBC is a fast drying, acrylic basecoat designed for today's automotive collision centers. MBC must be clearcoated with one of the MC clears.

**MBC Acrylic Basecoat**
- **Color**: MBC Acrylic Basecoat
- **Reducer**: MR185 Fast, MR186 Medium, MR187 Slow, MR189 Very Slow

**Compatible Substrates**
- Cured, cleaned and sanded OEM and refinish enamels
- MP170 Epoxy Primer
- MP176 Etch Primer - Should be primed before topcoating
- MP178 Plastic Primer
- MP180 2K Sealer
- MP181 1K Primer Surfacer
- MP182 2K Urethane Surfacer
- MP183 Compliant Surfacer
- MP184 Compliant Surfacer
- MX198 Polyester Primer
- MP210 1K High Solids Primer
- MP211 2K High Solids Primer
- MP213 GP Sealer

**Preparation**
- Surface cleaning: MX190 Cleaner, MX191 Low VOC Cleaner, MX192 Plastic Cleaner
- Sanding: 400 grit (machine or dry hand) or 500 grit (wet) on old finishes and primer surfacers

**Mixing**
- Ratios: MBC : MR Reducer = 1 : 1
- Tinting: MBC may be tinted up to 10% with OMNI™ AU mixing bases
- Additives: None

**Application**
- **Costs**: 2 coats or until hiding
- **Air pressure**: HVLP
  - 7 - 10 psi at the air cap
  - Conventional: 45 - 55 psi at the gun
- **Gun setup**: 1.3 - 1.6 mm or equivalent

**Dry Times**
- Between coats: 5 - 10 minutes at 70°F (21°C)
- Air dry: Tape: 45 minutes
  - Clear: 20 minutes minimum, 24 hours maximum at 70°F (21°C) to clearcoat

**Clean Up**
- Clean spray guns, gun caps, storage pots, etc., thoroughly with MR Reducer, MS 100 General Purpose Solvent or other appropriate clean up solvent after each use.
  - Follow EPA guidelines for proper storage and disposal of solvent-borne waste paint.

**Properties**
- **VOC**
  - Package: 5.7 lb./gal. max.
  - Applied: 6.3 lb./gal. max.
- **Film build per coat**
  - Applied (1:1): 0.5 - 0.6 mil
  - Applied (1:1): 176 - 198 sq. ft / gal., no loss

**Limitations**
OMNI™ AU is a new generation of paints and CANNOT BE MIXED WITH ANY COMPONENT OF THE OMNIcoat™ FACTORY PACKAGE PRODUCT LINE.

**Important**
The contents of this package may have to be blended with other components before the product can be used. Before opening the package, be sure you understand the warning messages on the labels of all components, since the mixture will have the hazards of all its parts. Spray equipment must be handled with due care and in accordance with manufacturer’s recommendations. Follow label directions for respirator use. Wear eye and skin protection. Observe all applicable precautions. See Material Safety Data Sheet and Labels for additional safety information and handling instructions.

**EMERGENCY MEDICAL OR SPILL CONTROL INFORMATION**
- In the U.S. (304) 843-1300; in Canada (514) 645-1320.
METHOD # 1:

This problem is solved by calculating the heat generated by the air only using the mass flow rate of the 5 scfm. Then the heat for the water vapor is calculated using the change in its enthalpy values. The sum of the two is the total amount of heat required to change the inlet air mixture conditions to the outlet air mixture conditions.

Objective:
Air at 55 PSI (3.062 atm) flowing at a rate of 5 scfm (.002818 kg/s) is to be cooled from 120 °F (322 °K) to 40 °F (277.59 °K) in a .375 inch (.9525 cm) inner diameter tube with uniform wall temperature of 32 °F. The length needs to be determined.

Assumptions/Conditions
- Forced Convection
- Negligible entrance effects on h bar
- Moderate property variation
- Standard conditions

Properties: Air at the average inlet and outlet temperature of 297.59 °K and 3.062 atm.

The following properties were obtained from air saturation tables.
\[ \mu = 1.947 \times 10^{-5} \text{ kg/m}*s \]
\[ C_p = 1.0072 \text{ kJ/(kg °C)} \]
\[ k = 0.0279 \text{ W/(m °C)} \]
\[ Pr = 0.7032 \]

**Analysis:** Noting that the specific heat is essentially uniform over the temperature range from 322 to 273.15 the following equation can be used to determine the heat generated.

\[ q_c = \text{mass flow rate} \times C_p \times (T_2 - T_1) = 0.002818 \text{ kg/s} \times 1.0072 \text{ kJ/kg K} \times (273.15 - 322) = -0.12615 \text{ kW} \]

Convert to English units to obtain, \(-430.4865 \text{ BTU/hr}\)

The heat \(q_c\) generated above is just the amount of heat taken out by the air only. Now calculations need to be made to determine the heat required for the water vapor as well as the liquid water produced. To do this certain values were used from the water separator cup size calculations which can be found in Appendix R.

The change in enthalpy of the water vapor from the inlet to the outlet is the heat required.

\[ \text{@ 40 F h = 1078.9 BTU/lb , @ 120 F h = 1113.5 BTU/lb} \]

The enthalpy value at each of the inlet and outlet multiplied by the amount of water vapor in pounds at the inlet and outlet.

\[ 1078.9 \text{ BTU/lb} \times 0.001091656 \text{ Lbs w / Lbs da} = 1.177787658 \text{ BTU/Lb} \]
\[ 1113.5 \text{ BTU/lb} \times 0.026268568 \text{ Lbs w / Lbs da} = 29.25005047 \text{ BTU/Lb} \]

The difference between the two is \(28.0726281 \text{ BTU/Lb}\). Now the inlet stream of dry air is calculated.

\[ 5 \text{ scfm} \times 0.07455 \text{ LBS/FT}^3 = 0.3725 \text{ LBS/Min} \times 60 \text{ Min} = 22.365 \text{ Lbs of dry air / Hr} \]
22.365 Lbs of dry air / Hr * 28.07226281 BTU/Lb = -627.8361578 BTU/HR

Enthalpy of the water produced which is relatively close to the temperature of the air stream of 40 F is \( h = 8.02 \) BTU/Lb. If 8.6476 ounces of water is produced every hour = 

\[
.563562293 \text{ Lbs/HR}
\]

\( 8.02 \text{ BTU/Lb} \times .563562293 \text{ Lbs/HR} = 4.5197 \text{ BTU/HR} \) for the liquid water

This heat values generated for the air, water vapor and liquid water are added together to reach a final total of heat required to change the inlet conditions to the outlet conditions.

Since the both the inlet and outlet temperatures are known the log rhythmic mean temperature difference LMTD relationship can be used.

\[
q_c = h_{bar} \times A_s \times \text{LMTD}
\]

\[
\text{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{(32-120)-(32-40)}{\ln\left((32-120)/(32-40)\right)} = -33.3626^\circ F
\]

To obtain \( h_{bar} \) the Reynolds number \( Re \) is calculated using the following equation.

\[
G = \text{mass flow rate} / A = .002818 \text{ kg/s} / \pi(.009525 \text{ m})^2 / 4 = 39.5529 \text{ kg/(m}^2 \text{s})
\]

\[
Re = \frac{GD/\mu}{39.5529 \text{ kg/(m}^2 \text{s}) \times (.009525 \text{ m})} / 1.947 \times 10^{-5} \text{ kg/m*s} = 19,350.48
\]

The flow is greater than 10,000 so it is considered to be turbulent. The next equations are used to determine Nusselt number \( Nu \) for uniform property conditions.

\[
f_{cp} = (1.58 \ln Re - 3.28)^2 = (1.58 \ln 19,350.48 - 3.28)^2 = .00659
\]

\[
Nu_{cp} = \frac{f/2}{1.07} Re Pr / 12.7 (\sqrt{f/2}) (Pr^{2/3} - 1) = 48.8939
\]
h bar = Nu * k/D = 48.8939 * (.0279 W/(m °C)/.009525 m) = 143.27 W/(m² °C) = 25.233 BTU/Hr² °F
As = qc / h bar * LMTD = -1062.842 BTU/hr / (25.233 BTU/Hr² °F * -33.3626 °F) = 1.2625 ft²
L = As / π(D) = 1.2625 / π(.375/12) = **14.71 feet**

*Refer to the following page for an Excel Spreadsheet Progra*
Appendix N
Air Dryer Calculation Program
Heat Transfer - Forced Convection Method

METHOD # 1:

<table>
<thead>
<tr>
<th>Variables</th>
<th>Assumptions</th>
<th>Units</th>
<th>Assumptions</th>
<th>Units</th>
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<td>F</td>
<td>322.0389</td>
<td>K</td>
</tr>
<tr>
<td>Outlet Temp.</td>
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<td>F</td>
<td>277.5944</td>
<td>K</td>
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<td>K</td>
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<td>0.002359782 cubic meter/sec</td>
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</tr>
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<td>Operable Pressure</td>
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<td>psi</td>
<td>3.742528</td>
<td>ATM</td>
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</table>

These values below correspond with the average inlet and outlet temperatures that were calculated in degrees Kelvin in the above columns:

U: 1.94694E-05
Cp: 1.007154567
K: 0.027910549
Pr: 0.703151442

CALCULATIONS:

Air Density: 1.194337 kg/cubic meter

Mass Flow Rate: 0.002818375 kg/sec

\[ Q_c = -0.126157286 \]

\[ Q_c = h \times A_s \times LMTD \]

Sum of All

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<tr>
<th></th>
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<tr>
<td>Water vapor</td>
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<td>only</td>
</tr>
<tr>
<td>Liquid Water</td>
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<td></td>
<td>-1062.842433 BTU/Hr</td>
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</table>

\[ LMTD = -33.3626 \]

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<td>h</td>
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<tr>
<td>As</td>
<td>1.661773428 Sq. Feet</td>
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<tr>
<td>Length=</td>
<td>14.42615212 Feet</td>
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8
Appendix O

Heat from atmosphere:

To determine the amount of heat that is going into the container from the atmosphere both concepts of heat conduction and heat convection were used. Heat convection occurs from the atmosphere to the container then is conducted through the material and again heat convection takes place to the surrounding temperature inside the container. The following equations were used to determine the amount of heat going into the system.

\[
\frac{Q_o}{A_o} = h_o (T_\infty - T_{\text{Two}}) \\
Q_o = \text{amount of heat convected outside of the container} = \text{unknown} \\
A_o = \text{surface area of outside of the container} = 4.95 \text{ Sq. Ft.} \\
h_o = \text{heat transfer coefficient of outside air film} = 4 \text{ BTU/hr ft}^2 \ ^\circ F \\
T_{\text{Two}} = \text{Temperature of outside wall} = \text{unknown} \\
T_\infty = \text{outside atmosphere temperature} = 90 ^\circ F
\]

\[
\frac{Q_i}{A_i} = h_i (T_{\text{W}i} - T_\infty) \\
Q_i = \text{amount of heat convected inside of the container} = \text{unknown} \\
A_i = \text{surface area of inside of the container} = 4.95 \text{ Sq. Ft.} \\
h_i = \text{heat transfer coefficient of inside fluid} = 25.78 \text{ BTU/hr ft}^2 \ ^\circ F \\
T_{\text{W}i} = \text{Temperature of inside wall} = \text{unknown} \\
T_\infty = \text{inside fluid temperature} = 32 ^\circ F
\]

\[
\frac{Q_m}{A_m} = k_m/X_m (T_{\text{Two}} - T_{\text{W}i}) \\
Q_m = \text{amount of heat conducted through container} = \text{unknown} \\
A_m = \text{surface area of conductive material} = 4.95 \text{ Sq. Ft.} \\
k_m = \text{thermal conductivity of the material}=1/R=1/7=.143 \text{ BTU/hr ft}^2 \ ^\circ F \\
X_m = \text{thickness of material} = 1 \text{ inch of polyurethane} \\
T_{\text{Two}} = \text{Temperature of outside wall} = \text{unknown} \\
T_{\text{W}i} = \text{Temperature of inside wall} = \text{unknown}
\]

Now there are three equations and three unknowns. The values for Two and Twi can be calculated by equating all three equations. Once these values are determined they can simply be substituted back into each of the three equations and they should all equal one another therefore obtaining the amount of heat going into the container.
Performing the calculations above determines the following:

\[
\begin{align*}
\text{Two} &= 73.3742 \, ^\circ\text{F} \\
\text{Twi} &= 34.5796 \, ^\circ\text{F} \\
Q &= 66.5 \text{ BTU/hr}
\end{align*}
\]

\textbf{Appendix P}

\textbf{Required Amount of Ice for Operation Calculations}

To determine the required amount of ice needed for operation the total heat inside the container needed to be determined. This value was determined by adding the heat produced by the airflow plus the heat going into the container from the atmosphere. Refer to Appendix ___ for the heat quantity from the atmosphere and Appendix ___ shows the calculations concerning the heat removed from the airflow. With these values and the assumption of a 1 hour operating time the required amount of ice can be determined. The following equation was used to determine the amount of ice needed.

\[
\text{Qt} \times T = mh \\
\text{Qt} = \text{heat removed from airflow} + \text{heat going into the container} \\
T = \text{Operational Time in hours} \\
m = \text{mass of the ice in pounds} \\
h = \text{Enthalpy of ice} = 144 \text{ BTU/lb}
\]

From previous calculations the heat removed from airflow was determined to be ___ and the heat going into the container determined to be ___. Furthermore, the total heat inside the container calculates to be

\[
(1062.842 \text{ BTU/hr} + 66.5 \text{ BTU/hr}) \times 1 \text{ hour} = m \times 144 \text{ BTU/lb}
\]

\[
m = 7.84265 \text{ pounds of ice required to cool the air properly for at least one hour}
\]

A standard 7 pound bag of ice will give an operation time of .89255 hours. As for 14 pounds of ice the operation time is 1.78511 hours.
Appendix Q

Container Size Calculations

To determine the size of the container a couple of different factors were used. For example, the solid to cubed ice size factor was determined through physical tests to be approximately 46%. This meaning that the cubed ice takes up precisely 46% more volume than the solid ice. An assumption of two 7-pound bags of ice was used to determine the volume displaced. From this assumption and the size coefficient the container size could be calculated in terms of gallon capacity. The following equation was used to determine the size.

\[
\frac{(\text{Mass of the ice} \div \text{density of ice}) \times \text{size coefficient}}{.1336802 \text{ ft}^3/\text{gallon}}
\]

For one 7 pound bag of ice

\[
7 \text{ Lbs} / 57.24644 \text{ Lbs/ ft}^3 \times 1.46 / .1336802 \text{ ft}^3/\text{gallon} = 1.33547 \text{ gallons}
\]

For two 7 pound bags of ice

\[
14 \text{ Lbs} / 57.24644 \text{ Lbs/ ft}^3 \times 1.46 / .1336802 \text{ ft}^3/\text{gallon} = 2.6709 \text{ gallons}
\]

For 22 pound bag of ice

\[
22 \text{ Lbs} / 57.24644 \text{ Lbs/ ft}^3 \times 1.46 / .1336802 \text{ ft}^3/\text{gallon} = 4.197 \text{ gallons}
\]
Appendix R

Water Separator Cup Size Calculations

The following equations were used to determine the amount of water that would be condensed out of the air in a given one hour of operation. Assumptions made concerning this calculation are that the system is initially in an 85% relative humidity environment with an air temperature of 90 °F.

First before any moisture calculations are made the 5 standard cfm must be converted to pounds of dry air.

Standard air is defined as air at 36% relative humidity, 68 F, and 14.696 psia.

\[ Pv @ 68 = 0.33893 \]
\[ Pv @ 36\% \text{ R.H.} = 0.12201 \]

\[ W = \frac{PV}{RT} = \frac{(14.696 - 0.12201)(144)(1)}{(53.34)(68 + 459.67)} = 0.07456 \text{ pound per cubic foot} \]

This value is then multiplied by 5 scfm to give a total delivery rate of 0.372816864 pounds per minute.

\[ Pv @ 90 \degree F = 0.69816 \text{ psi} \]
\[ Pv @ \text{R.H.} = 85\% = 0.593436 \]

**Inlet water vapor per pound of dry air:**

\[ Wp @ 0 \text{ psi} = \frac{(Pv \text{ at } \% \text{ humidity} \times 0.62443)}{(55 + 14.7 - Pv \text{ at } \% \text{ humidity})} \]
\[ Wp @ 0 \text{ psi} = \frac{(0.593436 \times 0.62443)}{(55 + 14.7 - 0.593436)} = 0.026268568 \text{ Lbs w / Lbs da} \]

**Water content after compressing and cooling:**

\[ Wp @ 55 \text{ psi} = \frac{(Pv \text{ at } \% \text{ humidity} \times 0.62443)}{(55 + 14.7 - 0.12164)} \]
\[ Wp @ 55 \text{ psi} = \frac{(0.12164 \times 0.62443)}{(55 + 14.7 - 0.12164)} = 0.001091656 \text{ Lbs w / Lbs da} \]

Ounces of water removed = (Wp 0 psi - Wp 55 psi) * mass flow rate of 5 scfm * 60 min * 15.355

\[ = (0.026268568 - 0.001091656) \times 0.372816864 \times 60 \times 15.355 \]

\[ = 8.647669477 \text{ ounces of water removed per hour.} \]
Appendix S

Air Properties at the Expansion Nozzle:

If saturated air at 50°F, 55 psig, and local barometric pressure of 14.7 psia were allowed to expand such as in a paint spray gun nozzle until its total pressure were the same as the local barometric pressure, what would be the vapor pressure?

Solution: The vapor pressure can be determined by using the following equation.

\[ P_{v2} = P_{v1} \times \left( \frac{P_2}{P_1} \right) \]

\[ P_{v1} = \text{saturated water vapor pressure at 50°F (from air saturation tables)} \]
\[ P_{v2} = \text{vapor pressure when the air has expanded to a total pressure of 14.7 psia.} \]

\[ P_1 = (55 + 14.7) = 69.7 \text{ psia} \]
\[ P_2 = (0 + 14.7) = 14.7 \text{ psia} \]
\[ P_{v2} = (.17798) \times (14.7/69.7) \]
\[ P_{v2} = .037536 \text{ psi} \]

Now, what would the temperature have to be in order for the dew point to be reached?

Solution: The vapor pressure has been calculated above which corresponds directly with a temperature in the air saturation tables. The temperature at which the vapor pressure of .037536 psi is located is at about 12°F.
Appendix V

Copper Tubing

Phillips & Johnston, Inc can supply copper tubing in the sizes you require, whether it is light or heavy wall. We can supply in coils, straight lengths or cut pieces to your requirements provide tubing to your exact specification, including but not to:

- Soft temper to full hard temper
- Capillary/restrictor tubing
- Level wound coils
- Standard/non-standard sizes and shapes

Copper Refrigeration Coils

Phillips & Johnston, Inc stocks copper refrigeration coils in manufactured to ASTM B260. The sizes we stock are listed:

<table>
<thead>
<tr>
<th>Part Number</th>
<th>Outside Diameter</th>
<th>Wall Thickness</th>
<th>Weight per 50' Coil</th>
</tr>
</thead>
<tbody>
<tr>
<td>810301</td>
<td>1/8&quot;</td>
<td>.030&quot;</td>
<td>1.74 lbs.</td>
</tr>
<tr>
<td>810302</td>
<td>3/16&quot;</td>
<td>.030&quot;</td>
<td>2.88 lbs.</td>
</tr>
<tr>
<td>810303</td>
<td>1/4&quot;</td>
<td>.030&quot;</td>
<td>4.02 lbs.</td>
</tr>
<tr>
<td>810304</td>
<td>5/16&quot;</td>
<td>.032&quot;</td>
<td>5.45 lbs.</td>
</tr>
<tr>
<td>810305</td>
<td>3/8&quot;</td>
<td>.032&quot;</td>
<td>6.70 lbs.</td>
</tr>
</tbody>
</table>

http://www.pjtube.com/prod_copper.html

2/18/2002
Tin-Lined Copper Coils

Where corrosion resistance counts

Key Benefits

- Made from seamless CDA-122 DHP copper
- All sizes are in stock and available for immediate shipment in 50' coils
- All stock material is tin-lined
- Material can be produced tin coated OD, ID or both
- Coils are furnished in dead soft temper for easy working
- Each coil individually packaged in a sturdy container

Typical Applications

- Outdoor Gas Lighting
- Fuel Lines
- Outdoor Gas Grills
- Oil Heat
- Beverage Dispensing Equipment
- Mobile Homes

Available Sizes

<table>
<thead>
<tr>
<th>Part Number</th>
<th>Outside Diameter</th>
<th>Wall Thickness</th>
<th>Weight per 50' Coil</th>
</tr>
</thead>
<tbody>
<tr>
<td>810306</td>
<td>1/2&quot;</td>
<td>.032&quot;</td>
<td>9.10 lbs.</td>
</tr>
<tr>
<td>810307</td>
<td>5/8&quot;</td>
<td>.035&quot;</td>
<td>12.55 lbs.</td>
</tr>
<tr>
<td>810308</td>
<td>3/4&quot;</td>
<td>.035&quot;</td>
<td>15.25 lbs.</td>
</tr>
<tr>
<td>810309</td>
<td>7/8&quot;</td>
<td>.045&quot;</td>
<td>22.75 lbs.</td>
</tr>
</tbody>
</table>

Other sizes are available on request

Please contact us and we will promptly process your inquiry.

http://www.pjtube.com/prod_copper.html

2/18/2002
### Technical Data: Tables

#### Table 1. Copper tube – Types, standards, applications, tempers, lengths

<table>
<thead>
<tr>
<th>Tube type</th>
<th>Color</th>
<th>Standard</th>
<th>Application 1</th>
<th>Commercially available lengths²</th>
<th>Drawn</th>
<th>Annealed</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Nominal or standard sizes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Type K</td>
<td>Green</td>
<td>ASTM B 88</td>
<td>Domestic water, Service and distribution, Fire protection, Solar, Fuel/fuel oil, HVAC, Snow melting</td>
<td>Straight lengths:</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>⅛ inch to 8 inch</td>
<td>20 ft.</td>
<td>20 ft.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>10 inch</td>
<td>18 ft.</td>
<td>18 ft.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>12 inch</td>
<td>12 ft.</td>
<td>12 ft.</td>
</tr>
<tr>
<td>Type L</td>
<td>Blue</td>
<td>ASTM B 88</td>
<td>Domestic water, Service and distribution, Fire protection, Solar, Fuel/fuel oil, Liquified petroleum (LP) gas, HVAC, Snow melting</td>
<td>Straight lengths:</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>¼ inch to 8 inch</td>
<td>20 ft.</td>
<td>20 ft.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>12 inch</td>
<td>18 ft.</td>
<td>18 ft.</td>
</tr>
<tr>
<td>Type M</td>
<td>Red</td>
<td>ASTM B 88</td>
<td>Domestic water, Service and distribution, Fire protection, Solar, Fuel/fuel oil, HVAC, Snow melting</td>
<td>Straight lengths:</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>¼ inch to 12 inch</td>
<td>20 ft.</td>
<td>N/A</td>
</tr>
</tbody>
</table>

² Commercially available lengths may vary by region and supplier. Application 1 includes various uses for copper tubing, but specific applications are not listed in this table.
<table>
<thead>
<tr>
<th>DMV</th>
<th>Yellow</th>
<th>ASTM B</th>
<th>Straight lengths:</th>
<th>Coils:</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACR</td>
<td>Blue</td>
<td>280</td>
<td>1¼ inch to 8 inch</td>
<td>3/8 inch to 4 ft</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>20 ft</td>
<td>1/8 inch</td>
</tr>
<tr>
<td>OXY, MED</td>
<td>Green</td>
<td>ASTM B</td>
<td>1/8 inch 1 5/8 inch</td>
<td>50</td>
</tr>
<tr>
<td>OXY/ACR/MED</td>
<td>Blue</td>
<td>819</td>
<td>20 ft</td>
<td>N/A</td>
</tr>
<tr>
<td>Type G</td>
<td>Yellow</td>
<td>ASTM B</td>
<td>3/8 inch to 1 2/8 inch</td>
<td>20 ft</td>
</tr>
<tr>
<td></td>
<td></td>
<td>837</td>
<td>20 ft</td>
<td>20 ft</td>
</tr>
</tbody>
</table>

1. There are many other copper and copper alloy tubes and pipes available for specialized applications. For information on these products, contact the Copper Development Association, Inc.

2. Individual manufacturers may have commercially available lengths in addition to those shown in this table.

3. Tube made to other ASTM standards is also intended for plumbing applications, although ASTM B 88 is by far the most widely used. ASTM Standard Classifications B 698 lists six plumbing tube standards including B 88.

4. Available as special order.

Appendix W

Polyurethane Foam Applications
SPRAY-ON FOAM & COATINGS INC.
Vancouver, Washington, U.S.A.
Phone (360) 573-3131  Fax (360) 574-1052

History of foam
Rigid, closed cell polyurethane foams, as we know them today, weren't
developed until the early 1960's; though first discovered in World War II while
attempting to find a replacement and an additive for natural rubber, which was in
short supply. Flexible polyurethane foams became an alternative to rubber and
other flexible foams in the 1950's, which led to the development of our present
day rigid foams.

Common uses
Polyurethane foam is one of the best insulating materials at an R-7 per inch it is
commonly used in: Building construction, ceilings, walls, floors, Tank insulation,
Boats, Marine floatation, Oil drilling rigs and camps, Earthquake protection,
Roofing, duct work, Cold storages, Controlled atmosphere buildings,
Condensation control, Art work, displays, etc.

Typical Properties:
Density: 1.8# - 8# per cubic ft. (1/2# - 50# available)
Compressive Strength: 28 - 120 psi.
Closed Cell: Greater than 90%
R-value: R - 7 per inch @ 2# density

http://www.sprayonfoam.com/polyurethane.html

3/4/2002
R-Value

A material's R-value is the measure of its resistance to heat flow. It is important to know the R-value because many states or regions require that a roof system have a minimum amount of thermal resistance on commercial, industrial, and/or institutional buildings. The way it works is simple: the higher the R-value, the more the material insulates.

Some common roofing materials and their corresponding values for Thermal Conductance (C) and Thermal Resistance (R) are shown in the following table.

<table>
<thead>
<tr>
<th>Material</th>
<th>Thickness In Inches</th>
<th>C-Value</th>
<th>R-Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Metal</td>
<td>N/A</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>Concrete</td>
<td>1.0</td>
<td>3.333</td>
<td>0.30</td>
</tr>
<tr>
<td>Gypsum</td>
<td>1.0</td>
<td>1.667</td>
<td>0.60</td>
</tr>
<tr>
<td>Wood</td>
<td>1.0</td>
<td>1.099</td>
<td>0.91</td>
</tr>
<tr>
<td>Tectum</td>
<td>1.0</td>
<td>0.500</td>
<td>2.00</td>
</tr>
<tr>
<td>Inside Air Film</td>
<td>N/A</td>
<td>1.087</td>
<td>0.92</td>
</tr>
<tr>
<td>Outside Air Film - Summer</td>
<td>N/A</td>
<td>4.000</td>
<td>0.25</td>
</tr>
<tr>
<td>Outside Air Film - Winter</td>
<td>N/A</td>
<td>5.882</td>
<td>0.17</td>
</tr>
<tr>
<td>Vapor Retarders</td>
<td>N/A</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>Material</td>
<td>R-Value</td>
<td>Thickness</td>
<td>Density</td>
</tr>
<tr>
<td>--------------------------</td>
<td>---------</td>
<td>-----------</td>
<td>---------</td>
</tr>
<tr>
<td>BUR Gravel</td>
<td>N/A</td>
<td>0.2941</td>
<td>0.34</td>
</tr>
<tr>
<td>BUR Smooth</td>
<td>N/A</td>
<td>4.167</td>
<td>0.24</td>
</tr>
<tr>
<td>Fiberboard</td>
<td>1.0</td>
<td>0.360</td>
<td>2.78</td>
</tr>
<tr>
<td>Perlite</td>
<td>1.0</td>
<td>0.360</td>
<td>2.78</td>
</tr>
<tr>
<td>Phenolic Foam*</td>
<td>1.0</td>
<td>0.120</td>
<td>8.30</td>
</tr>
<tr>
<td>Fiber Glass</td>
<td>1.0</td>
<td>0.256</td>
<td>3.90</td>
</tr>
<tr>
<td>Polyisocyanurate</td>
<td>1.0</td>
<td>0.180</td>
<td>5.56</td>
</tr>
<tr>
<td>Polyisocyanurate Composite</td>
<td>1.5</td>
<td>0.240</td>
<td>4.17</td>
</tr>
<tr>
<td>Polystyrene Bead Board</td>
<td>1.0</td>
<td>0.280</td>
<td>3.57</td>
</tr>
<tr>
<td>Polystyrene Composite Board</td>
<td>1.5</td>
<td>0.301</td>
<td>3.32</td>
</tr>
<tr>
<td>Polystyrene - Expanded (EPS)**</td>
<td>1.0</td>
<td>0.260</td>
<td>3.85</td>
</tr>
<tr>
<td>Polystyrene - Extruded (XEPS) ***</td>
<td>1.0</td>
<td>0.200</td>
<td>5.00</td>
</tr>
<tr>
<td>Sprayed Polyurethane Foam****</td>
<td>1.0</td>
<td>0.150</td>
<td>6.88</td>
</tr>
<tr>
<td>Cork</td>
<td>1.0</td>
<td>0.280</td>
<td>3.57</td>
</tr>
</tbody>
</table>

* Problems have been reported with regard to the use of Phenolic Foam roof insulation. Incidents of deck corrosion have been reported in cases where the insulation is in direct contact with steel roof decks and there is moisture present.

** Molded, Expanded Polystyrene Insulation, also referred to as MEPS, can have an R-value that will vary from less than 4.00 to slightly more than 4.00. The amount shown is an average amount used for roof system R-value calculations.

http://www.roothehelp.com/Rvalue.htm

3/4/2002
CONSTRUCTION

400 Series Plastic Water Coolers


400 Series Plastic Water Coolers:
Igloo delivers cold, clean water.
Rugged, durable Igloo 400 Series Industrial Strength Water Coolers keep water cold, clean and available. Built for the real world™ the 400 Series takes the abuse of the roughest job site and the most punishing work crews.

- Red and Yellow high visibility safety colors
- HDPE hide won’t chip, peel, buckle or rust
- UV stabilizers prevent fading, cracking in all exposures
- White FDA Grade inner liners are easy to clean and resist stains and odors
- Reinforced handles and cup dispenser bracket
- "Drinking water" imprint meets OSHA requirements

All parts are replaceable - for extended life
Specialty colors and imprints quoted upon request
# CONSTRUCTION

## 400 Series Plastic Water Coolers

<table>
<thead>
<tr>
<th>Model #</th>
<th>Product Capacity</th>
<th>Exterior Dimensions (DxH)</th>
<th>Interior Dimensions (DxH)</th>
</tr>
</thead>
<tbody>
<tr>
<td>421</td>
<td>2 Gallon - 7.6L</td>
<td>11.1&quot; x 14.8&quot; 28.3 x 37.5 cm</td>
<td>5.5&quot; x 9.5&quot; 13.9 x 24.1 cm</td>
</tr>
<tr>
<td>431</td>
<td>3 Gallon - 11.4L</td>
<td>13.8&quot; x 13.8&quot; 35.1 x 34.9 cm</td>
<td>9.5&quot; x 10.4&quot; 24.1 x 26.4 cm</td>
</tr>
<tr>
<td>451</td>
<td>5 Gallon - 18.9L</td>
<td>13.1&quot; x 20.4&quot; 33.2 x 51.8 cm</td>
<td>9.3&quot; x 16.6&quot; 23.5 x 42.1 cm</td>
</tr>
<tr>
<td>4101</td>
<td>10 Gallon - 37.9L</td>
<td>15.8&quot; x 23.3&quot; 40.2 x 59.1 cm</td>
<td>12.5&quot; x 19.1&quot; 31.8 x 48.6 cm</td>
</tr>
</tbody>
</table>

For more information, please call (800) 324-2653 or email us.

---

400 Series Plastic Water Coolers  
300 Series & 600 Series Metal Water Coolers  
300 Series & 600 Series Metal Water Coolers Dimension Chart  
Stainless Water Coolers  
Accessories

---

http://www.igloocommercial.com/cs4.htm
Appendix Z

General Purpose Filter
Model: PA2121

- 5-micron filter removes unwanted dirt particles and condensed water to extend tool life
- See-through bowl with metal guard allows easy monitoring of fluid level while protecting unit against impact
- Quick-release bowl and quarter-turn drain allows easy maintenance
- 57 SCFM flow capacity at 90 PSI. 150 PSI maximum pressure. 3/8-inch NPT(P) ports

- Warranty: 1 Year
- New Unit: $23.00

The PA2121, general purpose filter protects tools and equipment by removing most solid and liquid contaminants. This may include dust, dirt, pipe scale, rust, liquid water and bulk oil. A 5-micron filter element removes the smallest of particles. See-through bowl with metal guard allows easy monitoring of fluid level while protecting unit against impact. Features a quick-release bowl and quarter-turn drain for easier use. Standard series provides up to 57 SCFM flow capacity at 90 PSI. 150 PSI maximum pressure. 3/8-inch female NPT ports.
SAE 45° Flared Fittings

Economical brass fittings that resist mechanical pullout. Can be assembled and disassembled repeatedly. For use with copper, brass, aluminum and welded steel hydraulic tubing that can be flared, or as an SAE hose adapter.

Working Pressure
450 to 2800 PSI (tubing dependent)

Temperature Range
-65° to +250°F

More Information

Back

http://www.parker.com/brassprod/sae45ffbb.htm

3/8/02
### SAE 45° Flared Fittings

#### Male Run Tee 151F

<table>
<thead>
<tr>
<th>PART NO.</th>
<th>TUBE SIZE</th>
<th>PIPE THREAD</th>
<th>STRAIGHT THREAD</th>
<th>M</th>
<th>H</th>
<th>FLOW DIA. D</th>
<th>M/D</th>
<th>H/D</th>
</tr>
</thead>
<tbody>
<tr>
<td>151F-4-2</td>
<td>1/4</td>
<td>1/8</td>
<td>7/16-20</td>
<td>.86</td>
<td>.75</td>
<td>.189</td>
<td>.189</td>
<td></td>
</tr>
<tr>
<td>151F-4-4</td>
<td>1/4</td>
<td>1/4</td>
<td>7/16-20</td>
<td>.92</td>
<td>.89</td>
<td>.189</td>
<td>.189</td>
<td></td>
</tr>
<tr>
<td>151F-5-4</td>
<td>5/16</td>
<td>1/4</td>
<td>1/2-20</td>
<td>.92</td>
<td>.95</td>
<td>.220</td>
<td>.220</td>
<td></td>
</tr>
<tr>
<td>151F-5-6</td>
<td>3/8</td>
<td>3/8</td>
<td>5/8-18</td>
<td>1.04</td>
<td>1.04</td>
<td>.253</td>
<td>.253</td>
<td></td>
</tr>
<tr>
<td>151F-6-6</td>
<td>3/8</td>
<td>3/8</td>
<td>5/8-18</td>
<td>.98</td>
<td>1.00</td>
<td>.283</td>
<td>.283</td>
<td></td>
</tr>
<tr>
<td>151F-8-6</td>
<td>3/4</td>
<td>1/2</td>
<td>5/8-18</td>
<td>1.25</td>
<td>1.19</td>
<td>.350</td>
<td>.350</td>
<td></td>
</tr>
<tr>
<td>151F-8-8</td>
<td>1/2</td>
<td>3/4</td>
<td>3/4-16</td>
<td>1.35</td>
<td>1.23</td>
<td>.406</td>
<td>.406</td>
<td></td>
</tr>
<tr>
<td>151F-10-8</td>
<td>5/8</td>
<td>1/2</td>
<td>7/8-14</td>
<td>1.38</td>
<td>1.39</td>
<td>.509</td>
<td>.509</td>
<td></td>
</tr>
</tbody>
</table>

#### Union Elbow 155F

Ref: SAE 010201a *Thread Protectors*

<table>
<thead>
<tr>
<th>PART NO.</th>
<th>TUBE SIZE</th>
<th>STRAIGHT THREAD</th>
<th>M</th>
<th>H</th>
<th>FLOW DIA. D</th>
<th>M/D</th>
<th>H/D</th>
</tr>
</thead>
<tbody>
<tr>
<td>155F-2</td>
<td>1/8</td>
<td>5/16-24</td>
<td>.94</td>
<td>.79</td>
<td>.079</td>
<td></td>
<td></td>
</tr>
<tr>
<td>155F-3</td>
<td>3/16</td>
<td>3/8-24</td>
<td>.73</td>
<td>.62</td>
<td>.126</td>
<td></td>
<td></td>
</tr>
<tr>
<td>155F-4</td>
<td>1/4</td>
<td>7/16-20</td>
<td>.90</td>
<td>.89</td>
<td>.189</td>
<td>.189</td>
<td></td>
</tr>
<tr>
<td>155F-5</td>
<td>5/16</td>
<td>1/2-20</td>
<td>.92</td>
<td>.95</td>
<td>.220</td>
<td>.220</td>
<td></td>
</tr>
<tr>
<td>155F-6</td>
<td>3/8</td>
<td>5/8-18</td>
<td>1.04</td>
<td>1.04</td>
<td>.253</td>
<td>.253</td>
<td></td>
</tr>
<tr>
<td>155F-8</td>
<td>1/2</td>
<td>3/4-16</td>
<td>1.31</td>
<td>1.23</td>
<td>.406</td>
<td>.406</td>
<td></td>
</tr>
<tr>
<td>155F-10</td>
<td>5/8</td>
<td>7/8-14</td>
<td>1.39</td>
<td>1.39</td>
<td>.502</td>
<td>.502</td>
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</tr>
<tr>
<td>155F-12*</td>
<td>3/4</td>
<td>1-1/16-14</td>
<td>1.63</td>
<td>1.63</td>
<td>.635</td>
<td>.635</td>
<td></td>
</tr>
</tbody>
</table>

### 45° Elbow 159F-259F

Ref: SAE 010302 *Thread Protectors*

<table>
<thead>
<tr>
<th>PART NO.</th>
<th>TUBE SIZE</th>
<th>STRAIGHT THREAD</th>
<th>M</th>
<th>H</th>
<th>FLOW DIA. D</th>
<th>M/D</th>
<th>H/D</th>
</tr>
</thead>
<tbody>
<tr>
<td>159F-4-2</td>
<td>1/4</td>
<td>1/8</td>
<td>.78</td>
<td>.58</td>
<td>.188</td>
<td>.188</td>
<td></td>
</tr>
<tr>
<td>259F-4-2</td>
<td>1/4</td>
<td>1/8</td>
<td>.78</td>
<td>.58</td>
<td>.188</td>
<td>.188</td>
<td></td>
</tr>
<tr>
<td>159F-4-4</td>
<td>1/4</td>
<td>1/4</td>
<td>.75</td>
<td>.65</td>
<td>.188</td>
<td>.188</td>
<td></td>
</tr>
<tr>
<td>259F-4-4</td>
<td>1/4</td>
<td>1/4</td>
<td>.75</td>
<td>.65</td>
<td>.188</td>
<td>.188</td>
<td></td>
</tr>
<tr>
<td>159F-5-4</td>
<td>5/16</td>
<td>1/4</td>
<td>.75</td>
<td>.65</td>
<td>.220</td>
<td>.220</td>
<td></td>
</tr>
<tr>
<td>259F-5-4</td>
<td>5/16</td>
<td>1/4</td>
<td>.75</td>
<td>.65</td>
<td>.220</td>
<td>.220</td>
<td></td>
</tr>
<tr>
<td>159F-6-4</td>
<td>3/8</td>
<td>3/8</td>
<td>.89</td>
<td>.86</td>
<td>.283</td>
<td>.283</td>
<td></td>
</tr>
<tr>
<td>259F-6-4</td>
<td>3/8</td>
<td>3/8</td>
<td>.89</td>
<td>.86</td>
<td>.283</td>
<td>.283</td>
<td></td>
</tr>
<tr>
<td>159F-6-6</td>
<td>3/8</td>
<td>3/8</td>
<td>.91</td>
<td>.93</td>
<td>.283</td>
<td>.283</td>
<td></td>
</tr>
<tr>
<td>259F-6-6</td>
<td>3/8</td>
<td>3/8</td>
<td>.91</td>
<td>.93</td>
<td>.283</td>
<td>.283</td>
<td></td>
</tr>
<tr>
<td>159F-8-6</td>
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### 45° Flare Elbow to Metric Adaptor 159F-X-MIX

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<th>N</th>
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Note: Viton o-ring is standard

---

Parker Hannifin Corporation
Parker Brass Products Division
Otsego, Michigan
### SAE 45° Flared Fittings

#### Male Run Tee 151F

<table>
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#### 45° Elbow 159F-259F

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#### 45° Flare Elbow to Metric Adaptor 159F-X-MIX

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Note: Viton o-ring is standard
Parker Pipe Fittings

Anchor coupling 207ACBH

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Coupling 207P

Ref. SAE 130138

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Reducer coupling 208P

Ref. SAE 130138

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Bushing 209P

Ref. SAE 130140

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Parker Brass Products Division
Parker Hannifin Corporation
Otsego, Michigan

The World Standard
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**NOTES:**

1. EAGLE EQUIPMENT CORPORATION
   CINN. OH 45241
2. CINCINNATI HOSE & FITTINGS
   CINN. OH 45215
3. COMMERCIAL PRACTICES & TOLERANCES APPLY.
Appendix AB

Campbell Hausfeld - Built to Last - Accessories

Air Compressors - Quick Connect Couplers & Plugs
Details for Part Number MP3234

1/4" Male Industrial Couplers - 1/4" I.D. Body

- The MP3234, Industrial Coupler has a 1/4" Inner Diameter and uses a 1/4" Female NPT connection to provide a quick, convenient and reliable method for changing between air tools. For Maximum performance and power we recommend 3/8" inner diameter fittings. These fittings provide 100% more airflow for more power to the tool. Teflon Tape must be used with all fittings to achieve maximum performance (MP5136).
- Model MP3234

PRICE: $6.56

Air Compressors - Quick Connect Couplers & Plugs

Details for Part Number MP2468

1/4" Male Industrial Plugs - 1/4" I.D. Body

- The MP2468, Industrial Plug has a 1/4" Inner Diameter and uses a 1/4" Female NPT connection to provide a quick, convenient and reliable method for changing between air tools. For Maximum performance and power we recommend 3/8" inner diameter fittings. These fittings provide 100% more airflow for more power to the tool. Teflon Tape must be used with all fittings to achieve maximum performance (MP2136).
- Model MP2468

PRICE: $2.71

Appendix AC

Cushioning & Sealing Washers

This item matches all of your specifications.

Part Number: 94708AS40

Type: Cushioning & Sealing Bonded Washers
Material: Neoprene
Screw Size: 1"
Inside Diameter: 1.050"
Outside Diameter: 2"
Thickness: 7/64"
Back: Galvanized Steel

←To Order this item, enter a quantity and click on the "Add to Order" button.

View this item on its catalog page.

To change any other features, please start your search again.
Appendix AF

Dew Point Transmitter
Series XDT
100°C to +20°C Dewpoint

http://www.cosa-instrument.com
COSA Digital Dewpoint Transmitters - XDT Series

General
The COSA digital dewpoint transmitters are designed as compact, simple and reliable instruments, which will continually monitor air dryer performance, compressed air quality and dry gas moisture. From ambient dewpoint levels to as low as -14°F (-100°C).

Applications
COSA dewpoint transmitters are used wherever the dew point in a gas is critical. Applications include: monitoring and control of air dryers, plastic dryers, welding and laser gases, petrochemical feedstock gases, natural gas, clean rooms, glove boxes, transformer and switch gear insulation gases, cryogenic gases, heat treating furnaces, industrial specialty gases and many more.

Electronics
The transmitter electronics take full advantage of state of the art microprocessor technology and offer many advanced intelligent features. With optional dual alarms, analog and digital outputs, the COSA dewpoint transmitters can be used as indicators, alarm units or controllers.

Programmable Alarm Relays Option
The two optional alarm relays can be independently programmed to switch at any dewpoint with variable hysteresis, which makes the transmitter ideally suited as an energy saving controller for desiccant dryers in “dew point demand” mode or safety cutoff in process control, high powered laser, etc. The status of the relays is shown on the display with flashing HI or LO indicators while displaying the dewpoint.

Analog and Digital Outputs Option
Analog and digital outputs are isolated from the sensor. The analog current or voltage output can be programmed to span the full or a portion of the range and is linear to the selected engineering units. The RS-232 interfaces into the serial port of any PC or Mac, for a simple operation with any standard communications program.

User friendly interface
The instrument is operated through a menu driven user interface consisting of a custom LCD display with optional backlight, and four push buttons.

Engineering Units
The user can select from the following engineering units: Dewpoint in °C or °F, ppmw, g H2O/m3, lbs H2O/ million scf.

Pressure Correct Function
Results are displayed at sensor pressure (atmospheric) or by pushing the Pressure Correct key at a user selectable alternate pressure, such as the line pressure.

SpanCheck in the Field
This field calibration procedure is fully automated and the user is prompted through a simple one minute procedure, which requires no additional equipment.

Error Indication
The instrument has indication for sensor open, short or electronic system failure, which can activate any of the alarm relays.

NIST / NPL Traceability Option
Certificates for NIST and NPL traceability are available upon request. Instruments can be recertified periodically at COSA’s laboratories.

Same Transmitter - Multiple Configurations

Model XDT-OEM
Model XDT-PM/C
Model XDT-NEMA

Model XDT-OEM is a stand alone board suitable for mounting in existing enclosures. Connections are made through a pluggable screw terminal block.

The electronic board can be broken into two parts and sandwiched to accommodate space constraints. (See Insert.)

Model XDT-PM houses the transmitter in a panel mount industry standard DIN 45700 box, 3" (7.6cm) deep. Connections are made through a pluggable screw terminal block.

The panel mount model is available with the push buttons on the outside of the front panel (shown above) or with the push buttons hidden behind the front panel.

Model XDT-NEMA houses the transmitter in a NEMA 4 (IP65) enclosure. Ideally suited for industrial environments.
Xentaur Hyper-Thin-Film (HTF) Al2O3™ Moisture Sensor Technology

The Xentaur HTF Aluminum Oxide™ sensor is the product of years of intensive research and has been thoroughly field proven. Xentaur HTF™ technology offers significant performance advantages over all other aluminum oxide moisture sensors.

High Capacitance Response:
Due to a hyper-thin film and a special activation process, Xentaur sensors have a capacitance change over their full range, several orders of magnitude larger than that of conventional aluminum oxide sensors.

Additionally, this change is quasi-linear and its sensitivity to temperature is negligible. The advantages of a linear high capacitance response are: better sensitivity, better repeatability, and faster response times. Also, the measurement system is less prone to noise and drift, and signal conditioning is kept to a minimum.

Interchangeable Sensors
HTF Al2O3 high capacitance sensors can be manufactured with high uniformity. Consequently, sensors are freely interchangeable without factory calibration or changing of EPROMS, as is required with conventional aluminum oxide sensors.

SpanCheck without reference standards:
Xentaur HTF™ high capacitance sensors have a very low residual capacitance when dry, and saturate at a pre-designed level of humidity above +68°F (+20°C). This allows a span check of the sensor by cupping the sensor in the palm of one’s hand for one minute, and adjusting the instrument to its upper range limit. The advantages of this span check system are obvious: Xentaur sensors can be field calibrated anywhere, anytime without using expensive and cumbersome calibration standards. Sensors do not have to be returned to the factory for recalibration, which also eliminates the need for a second stand-by sensor.

Waterproof Sensors
Xentaur HTF™ sensors are available in waterproof versions. Waterproof sensors can be fully immersed in water and will go back into operation after drying down.

Advanced Mechanical Sensor Design
Xentaur HTF™ sensors have been designed for the requirements of tough industrial environments. The sensor element is encapsulated in a 100 micron stainless steel sintered filter.

The filter housing is screwed onto a stainless steel sensor fitting which is available for pressures of up to 5000 PSI (FM tested). When sensor elements have to be replaced the sensor fitting can be saved. The sensor fitting has two mounting threads, which make it easy to use existing sample cells. The cable is connected through a BNC connector.

Technical Specifications of Xentaur HTF™ Sensors

<table>
<thead>
<tr>
<th>Sensor element:</th>
<th>High Thin Film High capacitance Al2O3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dew point range:</td>
<td>XTR-100: -148°F to +68°F (-100°C to +20°C)</td>
</tr>
<tr>
<td></td>
<td>XTR-95: -28°F to +68°F (-30°C to +20°C)</td>
</tr>
<tr>
<td>Capacitance:</td>
<td>15 nF to 200 nF</td>
</tr>
<tr>
<td>Accuracy:</td>
<td>±0.5°F (+2°F)</td>
</tr>
<tr>
<td>Repeatability:</td>
<td>±0.9°F (+2°F)</td>
</tr>
<tr>
<td>Response time:</td>
<td>For a step change from -40°C to +60°C:</td>
</tr>
<tr>
<td></td>
<td>95% in 90 seconds</td>
</tr>
<tr>
<td></td>
<td>99% in 450 seconds</td>
</tr>
<tr>
<td>Temperature range:</td>
<td>-22°F to +104°F (-30°C to +50°C)</td>
</tr>
<tr>
<td>Sample flow range:</td>
<td>Linear velocity at 1.5 mm/s</td>
</tr>
<tr>
<td></td>
<td>Static to 100 m/s</td>
</tr>
<tr>
<td>Storage temperature:</td>
<td>-40°F to +120°F (-40°C to +50°C)</td>
</tr>
<tr>
<td>SpanCheck:</td>
<td>Sensor saturates at dew point above +68°F (+20°C)</td>
</tr>
</tbody>
</table>

Fitting:

| Pressure operating range: | Standard 500 PSI (34 bar) |
|                          | Optional 5000 PSI (340 bar) |
| Mechanical connections:   | 1/4"NPT x 1.25 mm threads, and 3/4"NPT x 16 threads |
| Electrical connections:    | Female BNC connector |
| Sensor cable:             | Coaxial cable (75Ω with maximum capacitance of 60 pF/m) |
| Maximum cable length:     | 3,600 ft |

* These response times are not directly comparable to competitors data because of differences in measurement methodology and data presentation. Please inquire for detailed comparisons between Xentaur and other major sensor manufacturers.

Visit:
http://www.xentaur.com

HTF Al2O3™ Advantages:

- up to 600x more sensitive than conventional sensors
- Sensors are freely interchangeable
- Field calibration without reference standards
- Faster response
- Better repeatability
- Longer sensor life
- Less drift
- Negligible temperature coefficient
Technical Specifications XDT Series:

Transmitter:
- High capacitance Al2O3

Sensor type:
- LKG with optional backlight, 3.5 digits and custom legends for units and mode, audible alert

Input Resolution:
- 0.1°C dew point

Indicators:
- Dew point in °F and °C

Engineering units:
- Dew point in °F and °C,

ppmv, g H2O/m3, lbm H2O/mm

Contrails:
- 5 push buttons, all settings stored in EEPROM

Output options:
- 4-20mA or 0-24mA outputs, linear to selected engineering units, programmable span and range, 0.1°C dew point resolution

RS-232, baud rate 9600, resolution 0.1°C dewpoint

Isolation:
- Sensor is isolated from the power supply, analog output and RS-232

Alarm relays option:
- Two programmable alarm relays with programmable variable hysteresis, rated at 10A 90/240V

Failure indication programmable to trigger alarm relays

Power requirements:
- 100 - 240 VAC, 50 or 60 Hz, autotransforming

Connections:
- Pluggable screw terminal block

Enclosure:
- Polycarbonate, Nema 4/4x Dimensions: 4.7" x 8.3" x 3.5" (w x h x d); DIN 43700 - 3” deep

Warranty: 1 year for full workmanship and defective parts.

Dimensions of Element and Fitting:

Dimensions XDT-OEM

Temperature range of electronics: -14°F to 122°F (-26°C to 50°C)

Dimensions XDT-NEMA

Dimensions XDT-PM

Represented by:

Cosa Instrument Corporation
56 Oak Street, Norwood, NJ 07648
Tel: (201) 767-5600 • Fax: (201) 767-8604
e-mail: cosa@cosa.com
1404 E. North Belt Drive, Suite 160, Houston, TX 77032
Tel: (281) 538-0961 • Fax: (281) 538-0944

http://www.cosa-instrument.com
### Appendix AG

<table>
<thead>
<tr>
<th>Atm. Humidity</th>
<th>%</th>
<th>Atm. Pressure</th>
<th>Inches of Mercury</th>
<th>PSIA</th>
</tr>
</thead>
<tbody>
<tr>
<td>87%</td>
<td></td>
<td>29.15</td>
<td>14.31714</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Trial #1</th>
<th>Trial #2</th>
<th>Trial #3</th>
<th>Trial #4</th>
<th>Trial #5</th>
<th>Trial #6</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>10 min</td>
<td>20 min</td>
<td>30 min</td>
<td>40 min</td>
<td>50 min</td>
<td>60 min</td>
<td>1 hour</td>
</tr>
</tbody>
</table>

| T1 Room       | 92 | 92 | 94 | 95 | 95 | 96 | 94 |
| T2 Flowmeter  | 99 | 102 | 105 | 110 | 113 | 121 | 108.33333 |
| T3 Inlet      | 94 | 96 | 99 | 103 | 108 | 114 | 102.33333 |
| T4 Out cooler inlet | 85 | 85 | 86 | 86 | 87 | 87 | 86 |
| T5 In cooler inlet | 32 | 32 | 32 | 32 | 32 | 32 | 32 |
| T6 Top of Coil | 63 | 66 | 69 | 71 | 73 | 75 | 69.5 |
| T7 Middle of Coil | 32 | 32 | 36 | 37 | 37 | 38 | 35.33333 |
| T8 Bottom of Coil | 32 | 32 | 32 | 32 | 32 | 32 | 32 |
| T9 In cooler outlet | 39 | 38 | 37 | 37 | 36 | 36 | 37.16667 |
| T10 Out cooler outlet | 84 | 84 | 85 | 85 | 86 | 88 | 85 |
| T11 Spray Gun | 88 | 86 | 85 | 84 | 84 | 82 | 84.83333 |
| T12 Outlet   | 46 | 45 | 45 | 43 | 42 | 42 | 43.83333 |
| T13 Ice bath | 32 | 32 | 32 | 32 | 32 | 32 | 32 |

| P1 Regulator | 58 | 58 | 58 | 58 | 58 | 58 | 58 |
| P2 Flow meter | 55 | 55 | 55 | 55 | 55 | 55 | 55 |
| P3 Spray gun | 45 | 45 | 45 | 45 | 45 | 45 | 45 |
| P4 Cooler outlet | 54 | 54 | 54 | 54 | 54 | 54 | 54 |

| Dewpoint | 52 | 52 | 52 | 52 | 52 | 52 | 52 |

| Flow (cfm) | 3.2 | 3.2 | 3.2 | 3.2 | 3.2 | 3.2 | 3.2 |


| Air Density | 0.3350034 | 0.3332138 | 0.3314433 | 0.3285339 | 0.3268126 | 0.3223095 | 0.3295527 |

*Std. Air = .075*
Appendix AH
Appendix AH

From Experimental Data

70 °F Dry Bulb
15 °F Atmospheric Dew Point

From saturation temperature tables.
70 °F = Pws = .36297, 15 °F = Pw = .043384

Relative Humidity(RH) = Pw/Pws*100 = (.043384 / .36297) * 100

= 11.95%
Appendix AI

From Experimental Data

70 °F Dry Bulb
47 °F Wet Bulb

Match up the corresponding lines to determine % level of Humidity.

Relative Humidity (RH) = 12%