AN AIR CONDITIONING SYSTEM

FOR

DEMONSTRATION AND RESEARCH

by

Robert Stirling Gay

Thesis submitted to the Graduate Faculty of the

Virginia Polytechnic Institute

in candidacy for the degree of

MASTER OF SCIENCE

in

Mechanical Engineering

September 15, 1960

Blacksburg, Virginia

TABLE OF CONTENTS

LIST OF FIGURES AND PLATES

I. DEDICATION AND ACKNOWLEDGEMENTS

For the friendly encouragement which led me to graduate work, this thesis is dedicated to
and

A few of the many persons to whom the writer is indebted will be acknowledged with sincere gratitude for their assistance.

Members of the final Graduate Committee: Professor H. N. Jones, Chairman; Professor C. H. Long; and Professor L. A. Padis.

Members of the initial Graduate Committee: Professor C. H. Long, Chairman; Professor J. B. Jones; and Professor C. E. Trent.

Personal friends employed by industrial firms, in particular

of Westinghouse.

Industrial firms and their representatives, in particular Minneapolis-Honeywell Regulator Company.

Professor Oliver P. Strawn, Jr. for invaluable suggestions and constructive criticism.

II. INTRODUCTION AND SUMMARY

At the present time the Mechanical Engineering Laboratory at the Virginia Polytechnic Institute has no apparatus which can be used to demonstrate year-around air conditioning processes. There is a large well-insulated room in the present Heating and Ventilating section of the Laboratory, with dimensions of 17 ft. 5 in. deep x 21 ft. 7 in. wide x 13 ft. high. It is planned that this room will be used with the proposed built-up air conditioning unit. When it is desired that the entire room be held at constant temperature and constant humidity, then the built-up unit will supply conditioned air to the room and exhaust air from the room.

Another function also is contemplated for the test room. Models simulating modern buildings will be erected inside the test room to facilitate the study of heat transfer and vapor migration through building materials. For this purpose conditioned air from the proposed built-up unit will be circulated inside the model building, and the extremes of winter and summer conditions generated by heating and cooling coils will surround the model.

The purposes of this thesis were to investigate the contemporary field of air conditioning in industry and education; to determine what equipment should be recommended for installation in the Mechanical Engineering Laboratory;

and to design the components and controls for a unit which will be modern, practical, and educational. The components of the unit were designed or selected on the basis of the research summarized in this thesis in Chapter II, "Review of Literature."

The professional experience of the writer of this thesis included employment in four major fields of air conditioning: research and development for a major equipment manufacturer, air conditioning and mechanical contracting, consulting engineering, and teaching.

Background work for this project included a thorough investigation of the following:

(1) The present work and objectives of industrial air conditioning manufacturing firms.

(2) Methods of instruction and the air conditioning laboratory facilities of other colleges.

(3) Current practices in the air conditioning of homes, commercial buildings, and industrial plants.

The term "air conditioning" may be defined as the process by which simultaneously the temperature, radiant heat interchange, humidity, movement, and purity of the air in enclosed spaces are controlled within specified limits.

A central fan system was designed. Within the necessary design limitations, every effort was made to incorporate as many different methods as possible which demonstrate air conditioning processes, and every effort was made to enhance the effectiveness and attractiveness of the unit for its intended purposes of demonstration and research. The immediate objectives of the unit are:

(1) In conjunction with lecture courses, to provide means for acquainting undergraduate students in engineering with modern air conditioning processes and equipment by having them run standard performance tests on the unit.

(2) To provide an efficient facility for extensive graduate research in many phases of the air conditioning and refrigeration field.

(3) To contribute to the general research of the Engineering and Agricultural Experiment Stations the valuable asset of an automatically controlled constant temperature, constant humidity test chamber.

The main part of the conditioning apparatus will be supported in ductwork about four feet above the floor level so that weigh tanks, scales, condensate coolers, and drain lines may be placed underneath, thereby allowing the processes to be controlled and observed at normal eye level. In order to keep within the prescribed area, the air measurement chamber and fan will be mounted at a higher level. Ducts will carry air from the existing insulated test room to the unit and return conditioned air to the test room. A duct to extend from an existing roof opening will be large enough to carry 2000 cfm and will contain a freeze-resistant tempering coil.

Manually adjusted opposed blade dampers will control air quantities at the junction of the outside air duct and the return air from the test room. Air passing through the built-up unit will pass through the following components: disposable filter, electronic filter, activated carbon filter, steam preheater coil, direct expansion coil, hot or chilled water coil, opposed blade face and bypass dampers, washer, steam reheater coil, air temperature measuring chamber, air flow measuring chamber, and centrifugal fan with inlet vanes. Among the accessory components are a Refrigerant 12 refrigeration unit with water cooled condenser, water chiller, steam water heater, high-head circulating pump, and a wide assortment of valves and actuators for automatic control of any one of the many cycles being run.

On the suggestion of Professors J. B. Jones, C. E. Trent, and C. H. Long, each piece of conditioning equipment operating on the summer cycle was designed for a nominal cooling capacity of five tons when the system is circulating the equivalent of 2000 standard cfm of air. The winter cycle equipment was designed to supply humidified air to the test chamber at not less than 100°F when 9000 pounds per hour of outside air at 0°F enters the equipment. The proposed unit, with its automatic control, will be capable of maintaining within close limits any reasonable air temperature and humidity under all conditions of operation on both cooling and heating cycles.

This thesis will summarize the research which was combined with the experience and judgment of the writer to design a suitable unit for the institution. Many figures and drawings have been included. Authorities on laboratory design, including representatives from York Corporation, Armstrong Cork Corporation, Worthington Corporation, The University of Minnesota, Westinghouse Electric Corporation, and Purdue University were personally consulted. This thesis contains sufficient information so that when the unit is approved for erection, the components can be immediately procured and detailed shop drawings made.

The writer believes that the proposed unit would provide this school, at moderate cost, with facilities for demonstration and research which would be unsurpassed by any educational institution in the country.

III. REVIEW OF LITERATURE

This section will report the research which was necessary to design a suitable year-around air conditioning unit for the Mechanical Engineering Laboratory. A brief survey of industrial research was made in order to learn the problems mechanical engineering students entering that field would face, and should be preparing for in their professional training; and in order to become cognizant of the current problems, methods, facilities, and aims of industrial firms so that future research performed in the Mechanical Engineering Laboratory at V. P. I. will attract professional interest and merit recognition.

Other colleges were investigated, to study not only their laboratory facilities but also their methods of instruction in air conditioning, and their interpretation of the responsibility of colleges in air conditioning instruction.

Types of air conditioning systems were investigated to determine the most suitable unit for V. P. I., a unit which would embody all the objectives previously stated as design criteria. The various components in a complete system and the methods of automatic control for year-around equipment were investigated. An analytic study was made of undesirable atmospheric impurities; this study included the origins, the effects on health and comfort, the particle sizes, and the most modern methods for their removal.

Since the accuracy of experimental data can be only as reliable as that of the measuring instruments, an investigation was made to determine the most dependable means for measuring the wet-bulb and dry-bulb temperatures and the flow rate of air through the conditioning apparatus. Sufficient information was gathered to facilitate the design of an air temperature and air flow measuring chamber for the proposed unit.

Current textbooks, handbooks, and bulletins of the air conditioning industry were consulted for design and application data on all the components of a complete year-around air conditioning unit. This study covered steam heating coils, chilled and hot water coils, direct expansion cooling coils, fans, dampers, ductwork design, condensing units, water chillers, water heaters, circulating pumps, washer-dehumidifiers, and refrigerants.

It was found that the science of air purification is expected to become of increasing importance in the near future, and in the field of human comfort to become a necessity rather than a luxury.

A. RESEARCH PROJECTS AND LABORATORY FACILITIES IN INDUSTRY

In order personally to observe current research activity in industry, the writer visited York Corporation in York, Pennsylvania; the Armstrong Cork Company in Lancaster, Pennsylvania; and the Westinghouse Air Conditioning Division in Staunton, Virginia. In addition he was employed by the air conditioning division of the Worthington Corporation in East Orange, New Jersey.

Besides individual research, the major firms also engage in general, co-operative research. By membership in professional organizations such as the National Warm Air Heating and Air Conditioning Association, the American Society of Heating, Refrigerating, and Air-Conditioning Engineers, and the American Society of Mechanical Engineers, industrial concerns jointly sponsor research at independent laboratories and engineering experiment stations of universities. This research is of a general nature deemed beneficial to the industry as a whole, and the results are made available to the members in the professional publications. An example illustrates the method. For many years the University of Illinois has performed research under contract to the N.W.A.H.A.C.A. Typical projects were the optimum placing of registers and grilles in residences, and the performance and cost analysis of the operation of typical residential conditioning units.

YORK CORPORATION

The York Corporation has facilities to check the capacity ratings of all its products. Many methods of testing which were developed by York for its own purposes were later adopted by the A.S.R.E. or A.R.I. as industry standards. Preparatory to acceptance for installation, all large centrifugal compressors must pass a precise capacity rating test in a special calorimeter. Small package-type units are spot-checked in standard calorimeters to insure uniformity in manufacturing.

For the sake of simplicity and economy, the hot-gas bypass method is used to load compressors in breakdown tests. The procedure is to connect the compressor to a conventional condenser. Enough hot gas from the compressor discharge is mixed with liquid from the condenser to simulate normal temperature and pressure at the compressor suction. By this method the condenser need remove only the compressor heat of compression, and no external artificial load is required. The load on the compressor is at rated conditions.

Model rooms were constructed to test the air flow patterns and operating characteristics of such devices as high pressure induction units. The necessary built-up unit for supplying such models is always available, and can hold the supply conditions within close limits.

Extensive, recently developed facilities are available for testing package-type units. A test cell was specially built for window units in which the outside of the unit is exposed to conditions closely simulating summer design temperature and humidity, and the indoor part of the unit supplies conditioned air to a room in which the heat and moisture, supplied as load, for the tested unit are closely measured. In addition, there is a padded, sound-proof room mounted on coil springs to dampen out external vibrations. This room with complex equipment, is used to analyze and study the noises generated by machinery components, such as reciprocating compressors and propeller and centrifugal fans.

ARMSTRONG CORK COMPANY

The Armstrong Cork Company, well known manufacturer of duct and pipe insulation, has a modern laboratory in Lancaster, Pennsylvania. Conditioning equipment capable of maintaining close control of specified temperatures and humidities through a wide range supplies air to ducts which are covered with insulation to be tested, or supplies water to pipes covered with insulation. The ambient temperature and humidity of the test chamber containing the ducts and pipes can be controlled at precise, predetermined temperatures and humidities. The point of failure of an insulation, which is the condition at which dew will form on its outside surface, can be accurately determined. Based on these tests, application recommendations for the products in field use are highly accurate. A highly complex built-up type air conditioning unit used in the above tests to supply both conditioned air and conditioned water is automatically controlled from a central indicating and recording panel.

WESTINGHOUSE

The Westinghouse facilities in Staunton, Virginia include a research laboratory equipped with calorimeters to give exact capacity ratings of refrigeration compressors, and apparatus for operating machines until failure occurs. There are also soundproof and vibration-isolated rooms for analysis and study of the noises generated by compressors and fans. Research and development engineers constantly study the problems associated with performance of the products, such as efficiency of extended surface tube and fin coils. Such research is a combination of theoretical analysis and laboratory testing. In addition, methods of manufacture and testing are constantly appraised for improvement and lower cost.

The facilities for testing the performance of coils will be described in detail. A large central built-up air conditioning unit supplies air to a number of test cells in the building. The central unit supplies air at a specified condition regardless of whether the test cell adds heat or cools the air, which is constantly recirculated. The apparatus and conditions for a specific test on an experimental coil were as follows.[49] The test cell was located within the interior confines of a larger building so that the ceiling and walls were all interior walls. The outside temperature of the test cell walls was maintained in a range from 70° to 80°. The walls were constructed of a single layer of cinder blocks eight inches thick. Interior dimensions were 17 ft. 6 in. high x 25 ft. wide x 35 ft. 10 in. long. The ceiling and floor were of poured concrete with a glazed finish. The room was furnished with two air measuring sections. One had a nominal capacity of 18,000 cfm, and the other had a capacity of 4,500 cfm. The smaller was used for the test which is described.

The central built-up unit supplying conditioned air to the various test cells consisted of an air washer with circulating pump, a steam reheat coil, an air supply fan, and associated duct work. The duct work and anemostats, used to distribute the entering air, were mounted on the inner side of the test cell ceiling. If desired, air flowing through the test cell could be exhausted to the atmosphere while fresh outside air was drawn into the inlet of the central built-up unit. The chilled water used in the air washer was supplied from, and returned to, a central chilling tank. The hot water that was used in the air washer was supplied from, and returned to, a central water heating tank. The water was cooled by a central water chilling unit of 75 tons capacity. The hot water was heated with steam from the central power plant that was injected directly into the hot water tank. Steam for the reheat coil was supplied from the central power plant. The wet-bulb and dry-bulb temperatures of the air were maintained by a pneumatic control system, which indicated, recorded, and controlled the conditions within the test cell at any predetermined setting. The wet-bulb sensing element was located in the conditioned air supply duct so that the air leaving the air washer passed over it. The dry-bulb temperature sensing element was mounted in the conditioned test cell.

The steam and water valves were of the single seated, proportioning type. The air washer casing was constructed of 18 gage galvanized steel. Exterior dimensions were 96 in. long x 56 in. high x 99 in. wide. Two banks of washer nozzles were provided. Six banks of eliminators were provided on the downstream side of the washer. The eliminators were constructed on 22 gage galvanized steel and spaced on one and one-half inch centers. An entrance baffle made of galvanized steel 3/16 in. thickness containing 1-5/8 in. diameter holes placed on 2 in. staggered centers was utilized to prevent the stratification of the air passing through the washer.

Test Cell Equipment. The equipment and instrumentation used in conjunction with the evaporator coil to be tested consisted of the following:

(1) Semi-hermetic Refrigerant 12 compressor, nominal rating of three tons.

(2) Refrigerant condenser of the two-pass, shell and tube, water cooled type, 6-1/2 in. diameter and 31 in. long. The water circuit was through 5/8 in. nominal outside diameter copper tubing having 19 integral fins per inch; each fin was 1/16 in. high, giving an external surface 3-1/2 times the area of the internal surface.

(3) Manually operated refrigerant flow control needle valve in each of the three coil circuits.

(4) Sporlan type distributor.

(5) Sporlan filter-dryer in the liquid line between the refrigerant condenser and the refrigerant distributor.

(6) High-and low-pressure control switch to protect the compressor motor from overheating, and to protect the condenser against excessive pressure.

(7) Condenser water flow control valve to maintain a constant condensing temperature.

(8) Water-filled, inclined tube manometers to indicate the pressure drop across the air measuring nozzle section and to indicate the air pressure drop across the evaporator coil, having a range of 0 to 4 in. water gage in 0.01 in.

increments.

(9) An automatic calibrating and indicating recording potentiometer to obtain temperature readings with copper-constantan thermocouples.

(10) Mercury-in-glass thermometers calibrated by the Bureau of Standards with a range of 30° to 120° in 0.1° increments. Used to indicate the temperature of the air entering and leaving the evaporator coil, the water entering and leaving the condenser, and the temperature of the air sample from the special air measuring section. Thermometers measured both dry-bulb and wet-bulb temperatures.

(11) Platform, springless type beam scales used to weigh condenser water.

(12) Vertical U-tube type mercury manometer connected to the suction header of the evaporator coil to indicate the pressure at which boiling took place. Calibrated from 0 to 60 psig in 0.1 pound increments.

(13) Mercury-in-glass type barometer used to standardize the evaporating pressure and temperature since the difference in pressure between the coil and outside were measured.

(14) Bourdon tube type pressure gages at high and low sides of compressor.

(15) Copper-constantan thermocouples were used to measure temperature at the compressor suction and discharge, the refrigerant leaving the condenser, the refrigerant

entering the expansion valve, the coil suction header, and the suction header tube. The thermocouples were placed on the exterior surface of the copper tubes after the surface had been cleaned with sandpaper. Each thermocouple was wrapped with rubber waterproof tape and with two layers of felt Chromtite insulating tape.

(16) Temperature measuring section constructed according to A.S.R.E. standards. Air leaves the mixing section of vertical and horizontal vanes at a velocity of approximately 1000fpm and is accelerated to a velocity of approximately 3500 fpm at the throat of the Venturi section, from which the sampling fan draws air at approximately 1300 fpm across the wet-bulb and dry-bulb thermometers in an outside, parallel duct.

(17) Air flow rate measuring section constructed according to A.S.R.E. standards with two diffusers, a nozzle section, and two air pressure taps. The nozzle section contains four nozzles, each with a cross sectional area of 0.165 square feet. The air flow is obtained by determining the air pressure drop across the nozzle section. Two pressure taps are mounted in the bottom center section of the air measuring duct in such a way that the openings are flush with the internal wall surface. The aperture in each tap is 1/64 in. diameter.

SIGNIFICANCE

The air conditioning industry, in an age of accelerated engineering development in such fields as rocket propulsion, has been noted for its relative lack of change. There have been few radical developments in the past thirty years, and the changes were long foreseen and long awaited. Many improvements were primarily the applications to refrigeration of advances in other fields, as centrifugal and high speed reciprocating compressors. Two reasons for conservatism were the severe competition and rivalry in the industry and the lack of interest in air conditioning by the public.

Nevertheless, the products on the market today are highly dependable in service and very reasonable in cost. This is the result of applied research and development by the manufacturing firms. This research, though usually kept strictly confidential, is sometimes done under contract with independent laboratories or the engineering experiment stations of universities. This is the most economical procedure when a company does not desire to make a permanent addition of equipment and highly qualified research personnel to its existing facilities. An example is the research contracted by the Westinghouse Electric Corporation with the Virginia Polytechnic Institute for study of sensible cooling characteristics of extended surface coils, and analysis of an experimental model of a new air-to-air heat pump. The economy of this procedure and the inevitable wealth of fresh ideas at such reasonable cost make this plan seem of even greater significance for the future.

The purpose of including a brief survey of industrial research was to learn the problems mechanical engineering students entering that field would face, and should be preparing for in their professional training; and to become cognizant of the current problems, methods, facilities, and aims of industrial firms so that future research performed at V.P.I. will be significant and merit recognition.

The new laboratory built by the Worthington Corporation at East Orange, New Jersey was not described in detail since its facilities duplicate those at York and Westinghouse.

It was found that the most valuable research is not necessarily the most expensive research. For example, there is little reason for V.P.I. to try to undertake the expensive project of developing better design in centrifugal compressors. On the other hand, it would be extremely worthwhile to develop imaginative and more practical methods of laboratory testing, making use of current developments in instrumentation; or to study such problems as the physiological and psychological effects of airborne impurities on human beings; or by research to modify present electronic filtration devices, with the purpose of reducing the high cost and operating hazards.

B. FACILITIES IN OTHER ENGINEERING COLLEGES

Though there do exist modern, large, and well-equipped experimental laboratories, it would not be desirable to copy any of them exactly. On the contrary, the laboratory design recommended for V.P.I. will be similar to laboratories in other schools only insofar as the needs and philosophies of instruction and research at V.P.I. are similar to those of other schools.

A search through national publications going back twenty-three years produced articles describing in detail the air conditioning units at the University of Illinois and Syracuse University. The Illinois unit is nationally known.

A number of schools were contacted by correspondence and their replies are discussed. The writer personally visited the new laboratories at Purdue University.

UNIVERSITY OF ILLINOIS AIR CONDITIONING UNIT

General. The University of Illinois is well known for the many scientific contributions made possible by research with its year-around built-up air conditioning unit, and with the research residences sponsored by the National Warm Air Heating and Air Conditioning Association. The results of over twenty years of continous research have been published from time to time in professional journals and in booklets available at moderate cost.

The overall length of the unit is 42 ft. 6-1/2 in.[58] At a nominal capacity of 3000 cfm the unit has a cooling capacity of 12 tons and a total heating capacity slightly more than 300,000 Btu per hr. Since no attempt was made to condition the air of a particular space, the unit discharges into the general laboratory space, which measures 120 ft. x 135 ft.

The return air duct joins the outside air duct, which conveys air from a weatherproof inlet on the roof of the laboratory. Part of the return air can be used for reheating purposes when manual dampers allow it to bypass the conditioning equipment. All dampers are manually operated. The mixture of return and outside air passes successively through disposable type dry filters, a steam preheater coil with bypass dampers, water humidifying nozzles, hot and chilled water coil, direct expansion cooling coil, washer, steam reheat coil, and centrifugal fan. Airtight windows are located at frequent intervals along the equipment to allow visual inspection of the processes, and marine-type lights illuminate the enclosures behind the windows. Space is provided between each component of the built-up conditioning unit for sufficient instrumentation to study the performance of any one part of the unit. Component parts of the unit will be described in detail; then some of the problems for which the apparatus is suitable will be listed.

<u>Filters</u>. The air filters are of the disposable dry type having ten pockets fitted with sheets of cellulose material. The surface area totals 70.4 sq. ft.

<u>Atomizing Mist Nozzles</u>. In order to regulate the humidity of the air entering the apparatus at any season, six atomizing mist nozzles are located between the air preheater and the chilled water cooling coil. The nozzles are in a single bank and their discharge is directed against the air stream. A single bank of eliminators and a fine mesh copper wire screen between the mist nozzle chamber and the cooling coil remove entrained droplets of water from the air. The nozzles are 1/32 in. x 1/2 in.

<u>Water Coil</u>. The water coil, which can be used with either hot or cold water, consists of four sections arranged so that the water may be circulated through one, two,

three, or four sections in series. This allows the use of two, four, six, or eight vertical rows of tubes when studying heat transfer in such coils with forced circulation of hot or cold water. Counterflow of the water and air occurs. Water is fed into each coil section at the bottom and is taken out at the top on the opposite side. An expansion tank insures that the four coil sections are always completely filled with water. Each vertical row of the coil has thirty-one 1/2 in. O.D. horizontal copper tubes.

Direct Expansion Cooling Coil. The direct expansion Refrigerant 12 cooling coil has seventy-two 3/4 in. O.D. horizontal copper tubes arranged in six vertical rows with twelve tubes in each row. The tubes are staggered in the vertical rows. The liquid refrigerant is supplied through a thermostatic expansion valve to the first vertical row of tubes by means of two vertical headers which have metering orifices to insure equal distribution of refrigerant to the tubes. Each air-cooling coil is placed over a drip pan having a drain connection to an insulated condensate receiver. The vapor condensed from the air passing over the coil surfaces may be caught in a weigh tank and then drained to the sewer. The specified capacities of the direct expansion coil and the chilled water coil are the equivalent of 12 tons of refrigeration each. This rating is based on 50 gpm of chilled water at 45° entering the tubes, with air at 85° dry-bulb temperature and 40 per cent relative humidity across the coils. This capacity is based on the plant handling the equivalent of 3000 cfm.

Air Washer. The washer has 20 spray nozzles arranged in two banks of 10 each with the spray water discharging in the direction of the air flow. Deflecting louvers are situated in the air stream at the inlet end of the washer. The air outlet end of the washer has 24 vertical eliminator plates which are kept flooded by 9 additional spray nozzles. The water supply pump delivers 50 gpm to the washer nozzles at a pressure of 25 psig. The piping is arranged for the use of either recirculated, heated, or chilled water. The maximum air velocity in the spray chamber does not exceed 450 fpm when the plant is operating at full capacity.

Preheater Coil. Based on the circulation of 13,500 pounds of dry air per hour, and a steam pressure in the coils of 5 psig, the preheater coil has a capacity of 162,000 Btu per hour with entering air temperature of 0°. The coil is of the extended surface type with sixteen 1/2 in. O.D. horizontal copper tubes placed in a single row. There are 85 thin aluminum fins per linear foot of tube length. Tube ends are fastened to vertical headers at each side of the coil section. The face area measures 24 in. x 36 in. so that the air velocity over the face area will not exceed 500 fpm when the unit is handling 3000 cfm.

Reheater Coil. Based on the circulation of 13,500 pounds of dry air per hour, and a steam pressure of 5 psig, the reheater coil has a capacity of 140,000 Btu per hour with entering air temperature at 50°. There are thirty-one 1/2 in. O. D. horizontal copper tubes arranged in two vertical, staggered rows. Coil construction and size are the same as the preheater coil.

<u>Fan</u>. The fan is a backward-curved blade, single-inlet, single-width unit which delivers 3000 cfm standard air at 1150 rpm. The fan motor has sufficient capacity to drive the fan when delivering 3000 standard cfm against a static pressure of 1.5 in. water gage greater than the total resistance of the laboratory installation. This is to anticipate the future installation of ducts to a remote auditorium. By manual control of the direct-current motor, the fan may be operated at eight different speeds.

<u>Compressor</u>. The Refrigerant 12 compressor has two 5-3/4 in. by 4 in. single-acting air-cooled cylinders which are rated at 13.1 tons of refrigeration when operated at a speed of 450 rpm with 38° suction temperature and a condensing pressure of 110 psig. The compressor motor is arranged for automatic starting and stopping and facilities are included for five different operating speeds by manual adjustment of the starter rheostat.

<u>Condenser</u>. The condenser is of the shell-and-tube type with seven water passes and a total of fifty-four

1-1/4 in. tubes. The diameter of the shell is 16 in. and the overall length is 9 ft. 9 in. The condenser has a pot welded to the underside of the shell which serves as a liquid receiver, thereby reducing the amount of refrigerant required in the system. The water supply to the condenser is under the control of a regulating valve in the water supply line which is actuated by the refrigerant pressure within the condenser shell. The condenser is also fitted with a water-pressure failure switch which stops the compressor motor whenever the water supply is shut off.

Water Chiller. The chiller has three consecutive horizontal water passes arranged one above the other. Each water pass consists of two 2-1/8 in. O.D. copper tubes 18 ft. 6 in. long. The tubes of each pass are connected in parallel by means of headers at each end. Within each 2-1/8 in. copper tube are five 5/8 in. O.D. copper tubes through which the water flows. Each water pass is fitted with a thermostatic expansion valve which supplies liquid refrigerant to one end of each of the 2-1/8 in. O.D. copper tubes. The expanded refrigerant flows in a single direction around the small water tubes and is removed at three connections to a common suction header. Suitable header arrangements at each end of the 2-1/8 in. O.D. copper tubes allow the proper flow of refrigerant and water. The refrigerant pressure in the water cooler is controlled by an adjustable back-pressure valve in the suction line leading to the compressor. The back-pressure valve functions to maintain a predetermined pressure of the refrigerant in the evaporator sections of the water cooler and is a protective device; in addition there is a remote-bulb thermostat in the line leading away from the water chiller outlet which guards against freezing of the chiller.

Water Heater. The water heater is a shell-and-tube steam type heat exchanger having two water passes. The capacity of the heater is 600,000 Btu per hour using 5 psig steam.

Circulating Pump. The centrifugal pump is a single stage, single inlet unit direct connected to a three horsepower motor. The pump has capacity of 50 gpm against a discharge pressure of 25 psig. Like the fan and compressor motors, it is operated on 230 volt direct current.

Air Friction Pressure-Loss Measurements. Measurements of air friction pressure losses are obtainable as they occur across the air filters, preheater coil, water coil, direct expansion cooling coil, washer, and reheater coil. For such measurements total-pressure Pitot tubes, constructed according to official specifications, are placed in representative positions on each side of the part across which the friction losses are to be ascertained. A differential-pressure gage properly attached to each pair of tubes gives the true friction-pressure losses without any effects due to change of the velocity pressure which may result from the change of the cross-sectional area of the duct.

Temperature Measurement. Copper constantan thermocouples are located at several places in the duct system for the measurement of both dry- and wet-bulb air temperatures. The couples are placed in pairs, dry and wet, and are distributed across the duct sections at the following locations: (1) nine pairs before the air filter inlet, (2) eight pairs between the preheater coil and the water coil, (3) eight pairs between the water coil and the direct expansion cooling coil, (4) eight pairs between the direct expansion coil and the washer inlet, (5) eight pairs between the washer outlet and the reheater coil, (6) five pairs between the reheater coil and the fan inlet, (7) at the return air inlet, and (8) at the outside air inlet. All thermocouples lead to individual switches and a common cold junction located at a switchboard. A precision potentiometer is used to measure either the electromotive force existent in an individual couple or the average electromotive force of any single group of either dry- or wet-bulb couples when connected in parallel in the potentiometer circuit. Studies were made in the placement of the individual couples to obtain the most representative temperatures of the air stream as it flows through the duct. Dry- and wet-bulb thermometers may be placed, with a limited stem insertion, into the air stream at all thermocouple bank locations except at the fan inlet.

Air Flow Measurement. The ducts are heavy gage galvanized sheet steel, circular in shape except at the conditioning equipment, and designed for a maximum velocity of 1000 fpm when the air is measured at standard conditions. Air flow measuring stations are located in the return air duct, outdoor air duct, at the fan inlet, and at the fan outlet. Except at the fan inlet, the air flow measuring stations have reduced cross-sectional areas simulating venturi tubes, which do not increase fan work greatly. In the return air duct there is an additional air flow measuring station having straightening vanes and a Pitot traverse. At the fan inlet is a simple Pitot traverse.

Air and Water Temperature Controls. All thermostats placed in either the air stream or in the water heater outlet have elements filled with a volatile liquid. A change of volume of the contained liquid in each case produces a movement of some part of the thermostat or regulator. Compressed air is available in service lines at 40 psig, which supplies the 15 psig pressure necessary to give motive power at the actuators. The steam reheat coil is under the control of either of two extended-disk type thermostats, one of which is located just beyond the fan outlet and one of which is placed in the return air duct. A three-way cock placed in the compressed air piping serving the two thermostats allows proper air temperature control from either thermostat location. An extended-disk type thermostat is placed ahead of the air preheater coil to insure protection to the apparatus whenever the entering air falls below a temperature of 35°. On the discharge side of the preheater is located a remote bulb thermostat which normally functions to regulate the temperature of the air leaving the preheater coil. The two preheater thermostats are connected into the system through a pilot valve which allows either thermostat to operate independently of the other. All of these controls are of the modulating type. All steam valves in supply lines to the water heater and the air heaters are of the direct acting type. The temperature regulator at the water outlet of the water heater has a remote bulb element. A thermostat with a finned thermal element is located between the washer outlet and the reheat coil. This unit functions in connection with a direct-acting three-way mixing valve to regulate the temperatures of either warmed or chilled water supplied to the washer. This control also operates the pressure switch connected to the solenoid valve in the refrigerant piping to the direct expansion coil. In this way, year-around dewpoint control of the air leaving the apparatus is effected.

Compressor, Condenser, and Water Chiller Controls. In connection with the condenser and the compressor suction line are three pressure-actuated electrical switches: a high-pressure cutout, which opens when the condenser pressure becomes excessive; a low-pressure cutout, which opens when the pressure in the suction line drops below a preset value; and a water-pressure failure switch which opens when the water supply to the condenser is stopped. The three electrical switches are wired in series and the compressor motor cannot be automatically started until all three switches are closed. Any action which causes any one of the three switches to open will cause the compressor motor to stop. The water chiller is protected against freezing by a thermostatically-operated switch in an electrical circuit to a solenoid valve in the liquid refrigerant line between the condenser and the water chiller. When the water temperature falls to that for which the thermostat is adjusted the electrical circuit is broken, the solenoid valve closes, and the compressor pumps down the suction line until the low-pressure cutout causes the compressor motor to stop. The thermostatic bulb of the low-temperature control is placed in the refrigerant return line from the water chiller outlet.

The solenoid valve in the liquid refrigerant line is also under control of a hand-operated switch at the fan-motor control panel, and a pressure switch actuated by a thermostat placed in the main air duct just ahead of the reheater coil. In order to secure continuous operation of the compressor these latter two switches and the thermostatically-operated switch at the water chiller outlet must be closed. The opening of any one of the switches will cause the solenoid valve in the liquid line to close, and the low-pressure cutout in the suction line will stop the compressor.

Versatility of the Unit. The apparatus is suitable for use in the study of many problems, including:[58] (1) air cleaning by use of either washer or mechanical filters; (2) air humidification; (3) air cooling and dehumidification by three different pieces of apparatus; (4) hot blast heating using either steam or hot water as a heating medium; (5) heat transfer of finned coils using either steam or hot water as a heating medium; (6) heat transfer of finned tube cooling coils, using either chilled water or direct expansion of the refrigerant, with dry and wet surfaces; (7) problems involving the reheating of cooled and dehumidified air by use of either a steam reheating coil or by bypassing recirculated air; (8) year-around air conditioning of spaces in which typical load conditions may be maintained either in summer or winter; (9) centrifugal fan performance under different conditions of load; (10) accurate measurements of both dry- and wet-bulb temperatures; (11) air distribution by means of nozzles, grilles, and diffusers; and (12) the measurement of air flow and friction losses in ducts and other components such as filters and coils.

Comments and Criticism. The University of Illinois has long been the recognized leader among colleges and universities in air conditioning instruction and research. For this reason its facilities were discussed in detail. The built-up conditioning unit spans over 42 feet and has a nominal cooling capacity of 12 tons. The large size permits less percentage of error in data and calculations from experimental tests.

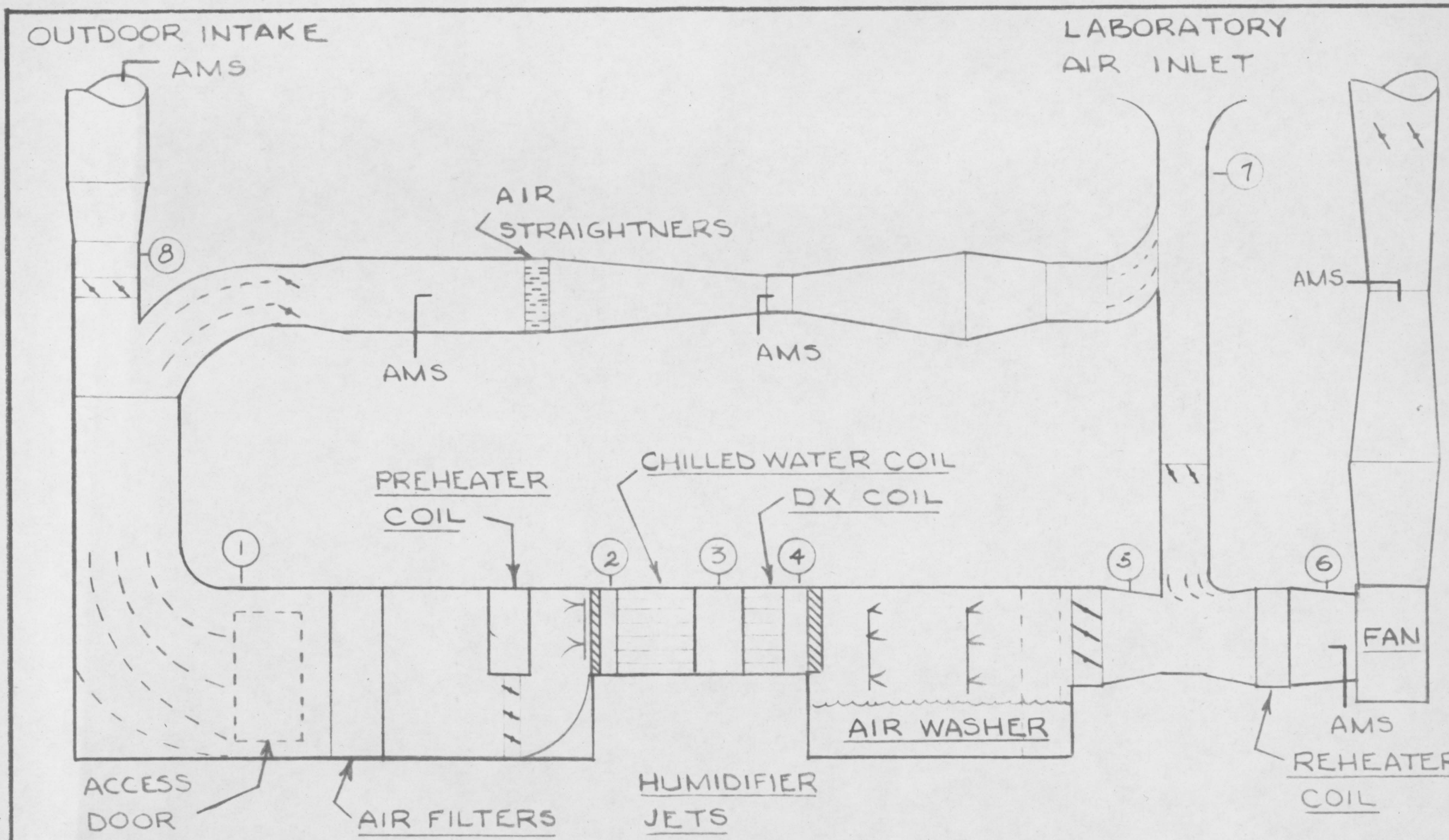
In recent years a number of developments and trends in the industry have occurred which make the Illinois laboratory obsolete. The electronic filter has come into common commercial and industrial use. Some of the benefits which it provides are super-filtered air for commercial comfort, and for the protection of delicate industrial controls; and savings both in initial and operating costs of cooling equipment because of less requirement for outside ventilating air.

The charcoal filter has also come into use as a partner of the electronic filter. This combination can remove every impurity known to occur in air, and has begun to replace the air washer, which is bulky, noisy, and expensive to operate and maintain, particularly in such critical applications as hospitals.

The Illinois unit has a set of bypass dampers which allow return air to bypass the conditioning apparatus. In testing specific components of the built-up unit, it is important that all circulated air pass through the tested component. Unfortunately, it is almost impossible to prevent some leakage through dampers, and though the leakage may not be enough to measure, it is enough to affect the test data and calculations.

The safety devices of the Illinois unit are standard equipment, and the air temperature measuring devices are good.

The cooling coils are rated at 85° dry-bulb entering air temperature. Though this is not the design temperature which would be used in a comfort installation, it is the rating temperature prescribed in A.R.I. Standard 411-56.



SYMBOLS: AMS ~ AIR MEASURING STATION, PITOT TRAVERSE

○— ~ LOCATION OF THERMOCOUPLES IN DUCTS

FIGURE 1. SCHEMATIC DIAGRAM OF EXPERIMENTAL AIR CONDITIONING EQUIPMENT AT THE UNIVERSITY OF ILLINOIS

SYRACUSE UNIVERSITY AIR CONDITIONING UNIT

General. The Department of Mechanical Engineering at Syracuse University completed an exceptionally well-designed built-up air conditioning unit in 1943.[26] The unit has an outside air intake with a tempering coil in the duct. The outside air duct makes a junction with the return air duct. The conditioning components in order are the preheater coil, disposable type filters, washer, direct expansion cooling coil, reheater coil, and centrifugal fan. Space was provided between the various elements to permit measurement of velocity and temperature of the air. A flange type joint, secured by bolts, was employed to facilitate the removal of any part of the system, such as cooling or heating coils, for repairs or replacement. The fan may be operated to discharge air through a spun aluminum calibrated nozzle for determining total air quantity handled, or by means of a selective damper the air may be conveyed to the classroom above the laboratory where methods of air distribution can be demonstrated and tested. An access door is provided near the fan inlet. To determine the air throw and distribution from various types of outlets, a eucalyptus leaf smoke pot may be placed in the duct which provides a visible, non-toxic pattern.

To simulate summer conditions at any time of the year, tempering and preheater coils and a compressed air operated water atomizing nozzle are located on the inlet side of the cooling coil and washer.

All steam piping is insulated with 85 per cent magnesia covering and all fittings and valves exclusive of unions are covered with asbestos cement. The length of steam piping and the efficiency of the insulation are such that 2-3° of the superheat developed as a result of the pressure reduction at the reducing valve is available in the steam at the points of entry into the steam operated devices. This condition eliminates the necessity of installing calorimeters for determining steam quality. Thermometer wells and mercury manometer tube connections are provided at the inlet of each item of equipment supplied with steam, in order that the degree of superheat at operating pressure may be determined. Having only 1-2° superheat is not enough when using a laboratory grade thermometer in an oil filled brass thermometer well. Replacing the wells with stuffing boxes permits exposing the thermometer bulbs directly to the steam; packing similar to that used on gage glasses is employed.

Air Washer. Since no standard commercial washer was available for the small size required, a nominal capacity of 700 cfm, it was necessary to fabricate the washer at the site. The washer has two banks with 12 spray nozzles per bank; operating at 10 psig, the washer has a capacity of 7 gpm, which is equal to 10 gpm per 1000 cfm. Water is circulated by a single stage, close coupled pump. No flooding nozzles are used. The eliminator plates are spaced on 1-1/8 in. centers. Each plate is formed from a single sheet of metal and has four bends or changes of air driection. The washer admittedly does not have the efficiency of commercial washers but is valuable in demonstrating evaporative cooling and humidification.

Cooling Coil. The cooling coil is of the extended fin type, three rows deep in the direction of air flow, with row control by means of an expansion valve for each row. With a variable speed motor driving the compressor and a cooling coil that may be operated with one-third, two-thirds, or all of its surface active, it is possible to demonstrate many of the situations encountered in the field. For example, it is possible to show the effect on the leaving wet-and dry-bulb temperatures when the coil is operated at constant refrigerant temperature and air quantity, but the amount of coil surface is varied. This demonstrates coil bypass factor. Manual shutoff valves are installed on the liquid and suction sides of each row of the coil. An air bypass is provided over the top of the coil. This, combined with the reheater, affords an opportunity to demonstrate both means of reheat for partial load temperature and humidity control.

Automatic Control. To illustrate the common methods of controlling blast heaters, face and bypass dampers, air washer dewpoint, and modulating expansion valves, a pneumatically actuated system of automatic control was installed. By means of manual selective valves it is possible to change readily from the heating cycle to the cooling cycle. An electro-pneumatic switch installed in the fan motor circuit permits air to be supplied to the control instruments only when the fan motor circuit is energized. A multi-louver damper, operated by a static pressure regulator, is installed in the built-up unit just ahead of the preheater coil. Static pressure control is used in demonstrating air flow with several types and makes of outlets.

Comments and Criticism. The Syracuse unit was well designed and is capable of effective student demonstration of several air conditioning cycles and effects. Its conspicuous fault is its limited amount of equipment; for example, it lacks an electrostatic air filter, an activated carbon filter, a chilled and hot water coil, a means for chilling the washer circulating water, and a means for accurate measurement of air temperature and quantity.

The small size of the unit in itself is not a handicap to precision experimentation, as is generally assumed. This important fact was borne out recently by Westinghouse engineers in Staunton, Virginia.[(50)] The Westinghouse engineers found that a coil with face area of only one-half square foot and having the condensing unit, air-flow measurement chamber, and fan on the same miniature scale gave the same test accuracy, percentage wise, as the earlier apparatus several times larger.

A noteworthy item is the fact that steam supplied for heating is slightly superheated to eliminate the necessity for calorimeters. This can be accomplished by insuring that the steam entering the reducing valve has low moisture content so that after throttling it will be superheated. Placing bare mercury-in-glass thermometers directly in steam lines with gage glass stuffing boxes as a steam seal would not seem a safe laboratory procedure, however. Steam at 5 psig pressure is approximately 227°; any temperature higher than about 140° will burn the human skin. Using thermocouples instead would provide both accuracy and safety.

The washer was not constructed with sufficient attention to details to obtain as high an efficiency as commercial models attain. Again, the size was not the handicap.

There are several commendable features about the Syracuse unit. Compressor speed may be varied by changing the speed of the direct-current motor drive. By placing an expansion valve at each of the rows in the direct expansion coil, bypass factor can be demonstrated either by varying the number of active rows with a constant air quantity, or by varying the air quantity with a constant number of active rows, or by changing the air quantity and number of active rows at the same time. In addition, it is possible to change the refrigerant temperature by changing the compressor speed. All these demonstrations are highly practical.

PURDUE UNIVERSITY CLIMATE CHAMBERS

Purdue University has ambitious plans to become a national center for basic and applied research in climate control at its Ray W. Herrick Laboratories, named for a major financial contributor. The Purdue Center for Research in Refrigeration and Climate Control estimates that $1,153,400 is needed for the first three years of operation, which includes much capital investment. The major objectives are environmental research and thermodynamic mathematical investigations which may reduce heat transfer and mass transfer problems to workable formulas. The Center plans basic research in heat transfer, mass transfer, thermodynamics, and fluid mechanics, all directed toward solution of problems in refrigeration and air conditioning. In addition, the Center will attempt to bring together through research the basic subject matter areas of mechanical engineering and environmental physiology for the purpose of enlarging the scope of climate control and to increase productivity in agriculture and industry. The four environmental conditions to be studied are temperature, humidity, light intensity, and air movement.

A brick barn was converted to form the first unit of the Center, to which two modern wings were added. In one section of the first unit it is planned that a typical, full-size modern house will be erected inside a conditioned chamber so that the reaction of a normal family to varying weather conditions can be studied. Another wing has an active laboratory for general research; its facilities include a high altitude chamber.

Climate Chambers. In the second new wing are two 23 ft. x 18 ft. controlled climate chambers, with ante-chambers having flush-mounted platform scales for weighing research livestock. The scales are accurate within one-half pound. The chambers are mirror images with control panel room between. Each room is fitted with both fluorescent and incandescent lighting at the ceiling, sealed from the room itself. Marine type electrical outlets and hot and cold water service outlets are available. Large drain troughs are installed in the floors. Automatically controlled devices empty both the supply and drain lines and traps when room temperatures approach freezing. An intercom connects each room to the control panel room. Bahnson compressed air and fresh water humidifiers add humidity directly to each room when indicated. The walls are glazed tile, the floor concrete.

After circulating through 30 in. wide passages outside the walls, 10,000 cfm of conditioned air is supplied to each chamber through the perforated ceilings. Air is exhausted through four 34 in. x 28 in. grilles with volume dampers.

Conditioning Equipment. The refrigeration system consists of three 50 hp booster compressors and three 50 hp high stage compressors working independently on a three-circuit calcium chloride brine system. Frick, Carrier, and Worthington compressors are represented. Refrigerant 12 liquid leaves each receiver through a Sporlan dryer and "See-All" flow and moisture indicator, passes through the liquid to suction heat exchanger, then through the liquid subcooler to the solenoid and expansion valves at the brine chiller. One set of solenoid and expansion valves is used for high temperature work and the other set for low temperature work. Each high stage compressor discharge line is equipped with an oil trap.

One brine pump recirculates brine between the chiller and the cold brine storage tank. A second brine pump recirculates brine from the cold brine tank to coils located in the test chamber slabs and to brine coils located in the Kathabar air washer. A third brine pump recirculates brine from the hot brine tank through the brine heater to the coils on the leaving side of the air washers. Three-way mixing valves are located at each coil. Either hot or cold brine is sent to the three-way control valves on the floor coils by additional three-way valves actuated from the central panel.

Outside fresh air is drawn in through a mechanical filter, steam preheater coil, and dryer; a booster fan discharges it into the intake plenum of the Kathabar air washer, where it is mixed with filtered return air from one of the test rooms. The mixed air proceeds through the Kathene and brine coil dehumidifier and cooler, brine reheater coil, steam grid humidifier, and centrifugal fan with inlet vanes. The Kathabar air washer and the outside air dryer have a steam regenerator which boils off moisture from the Kathene solution as it gathers moisture from the air. The test room floor coils are 1 in. pipe embedded 3 in. below the surface.

Controls. Adjacent to the test rooms is the central pneumatic control panel. A schematic arrangement of the major components of the conditioning equipment is graphically presented on the panel. Fans and motors can be started at the panel, and lights show which are running. Temperatures can be found by pressing a button for any location pictured on the panel, which includes important points in the air and brine cycles, as well as six locations in the concrete floor of each test room. The panel was presented as a gift by Minneapolis-Honeywell. There is one temperature indicator for all thermocouple readings, and two indicator-recorder-controllers for each test chamber: dry-bulb and wet-bulb temperatures for the air in each chamber, and temperature control of the slab. The climate range for each chamber varies from 0° to 110° and 40 to 95 per cent relative humidity. If desired, it is possible to have the floor at 110° and the room air at 0°, or to have the floor at 0° and the room air at 110°, or any intermediate combination.

Comments. Approximate cost of the project was $435,000; it will be ready for use about October, 1960. The test chambers are designed primarily to study environmental effects on animals, which may lead to a new and profitable market for equipment manufacturers. The Center is located among the agricultural buildings of the campus. The equipment is not designed for demonstration or instruction purposes; in fact, no work of the Center includes the undergraduate student. The main purpose of the climate chambers is simply that of providing a facility for graduate research.

MISCELLANEOUS COLLEGES

A number of other schools were contacted and replies were received from Cornell University, The University of Florida, The Pennsylvania State University, North Carolina State College, The University of Tennessee, and The University of Alabama.

Only the letter[45] from Richard B. Knight, Professor-in-Charge of the Heating and Air Conditioning Curriculum at North Carolina State College, contained significant information, which is quoted:

"The air conditioning unit in our laboratory consists of the following items arranged in series; 1) an electrostatic air filter, 2) a Freon-12 direct expansion coil, 3) a chilled water cooling coil, 4) face and bypass dampers, 5) an air washer, 6) a steam heated reheat coil, and 7) a centrifugal fan with volume control. Air can be fed to the unit from any or all of three sources; outside air, laboratory air, or air from a silica gel dehumidifying unit. At the present time students perform experiments showing 1) the effect of air velocity on the performance of a direct-expansion coil, 2) the effect of varied water flow rates on the performance of a chilled water coil, 3) the effect of face and bypass dampers on coil leaving air conditions, and 4) the effect of varying water temperatures on the performance of an air washer. In our laboratory there is also a steam-jet refrigeration system which is studied by students and, incidentally, furnishes chilled water to the cooling coil and air washer."

C. TYPES OF AIR CONDITIONING SYSTEMS

To be a complete air conditioning system, the apparatus must be able to control not only the temperature, humidity, purity, and motion of the air, but also the interchange of radiant energy. The most effective system developed for commercial use which can accomplish this is a combination of a panel type system to control radiant heat interchange and a central fan circulating system to pressurize the conditioned space and prevent infiltration. The central system also continuously supplies clean air, gives a "live" feeling to the space by moving the air, prevents stratification, and removes stale air. Cooling systems have been installed which remove as much as sixty per cent of the total heat by aluminum panels in which water well above the dewpoint temperature is circulated. The use of panels is particularly desirable because of the corresponding reduction in the duct sizes which carry conditioned air.

Variations of the simple central fan system have been developed for multi-zone buildings and include the dual duct system and the induction unit system. The dual duct system has central conditioning apparatus with two ducts leading to all conditioned spaces. One duct contains conditioned air according to the season and the other duct contains unconditioned air. For example, in winter one duct would contain warm air and the other duct unconditioned, or cold, air. Air-mixing room units under control of the room thermostat draw warm and cold air from the ducts in accordance with the room requirements, affording individual room control without zoning.[39]

The induction unit provides hot or cold water to the unit coil in each conditioned space according to the season. In addition, conditioned air is continuously supplied to nozzles in each unit which induce circulation of air through the conditioning coil and into the conditioned space. Air entering at the nozzles is called primary air and is always fresh, never recirculated. When occupied, commercial buildings have an internal cooling load throughout the year due to lights and people, which would allow the primary air to be cool throughout the year while the water supplied to the unit coil varies according to the season and the zone, and has as its function the counteracting of heat gain or heat loss due to wall and window transmission.

CONTROL OF CENTRAL FAN SYSTEMS

In this section will be given a brief description and appraisal of the most successful, modern methods for controlling the individual pieces of equipment which are recommended for installation in the proposed built-up unit. Based on this review, a later section of the thesis will indicate which method of control is recommended for each component.

Tempering Coil Control. The purpose of the tempering coil is to insure that air entering the remainder of the unit will at all times be above the freezing point. The steam-distributing coil is the best for this service. Freezing of the coil is prevented, even at low load conditions, since counterflow of the steam and condensate takes place inside the coil, and live steam is distributed equally to all parts of the coil. Condensate collects in headers outside the air duct. Nevertheless, the experience of control engineers has been that it is possible for such coils to freeze. The recommendation of the Minneapolis-Honeywell Regulator Company[63] is that the steam supply valve be modulated to the full open position when the entering air temperature drops to 35°. A duct thermostat upstream from the coil actuates the steam supply valve of the coil. As the coil operates at full capacity when the entering air temperature is 35° or lower, the coil capacity must be limited to a rise of about 30° to avoid losing control of the air by overheating.

Preheater Coil. In some rather expensive large systems, each room will have heating panels or convectors controlled by a room thermostat. Fresh air at moderate temperature to each room is supplied from a central duct system. In the above case, the central preheater coil is controlled by a discharge duct thermostat. A method which gives less comfort but more economy is to let the preheater coil supply all the heat for the building. The coil may be controlled either by a space thermostat mounted at a representative location or a duct thermostat in the return air stream. The space thermostat gives less representative sampling but is preferred because of its easy accessibility by the occupants.

In the control of dewpoint temperature on the winter cycle, the heating coil is controlled by a dewpoint thermostat with remote bulb at the washer outlet. Regardless of the temperature of air entering the coil, the coil must have sufficient heating capacity so that after passing through the washer, the air will be at the desired dewpoint. In industrial applications, where the internal sensible and latent heat loads are generally constant, the dewpoint thermostat may be set at a predetermined value. For commercial and residential applications, where the sensible and latent heat loads are constantly changing, it is common practice to reset the control point of the dewpoint thermostat with a compensating space humidity controller.

The preheater coil does not require protection against freezing since the air is maintained always above the freezing point by the tempering coil.

Reheater Coil Control. On the winter cycle in a central fan system, a reheater coil is necessary to control space sensible temperature. The steam supply to the coil is generally controlled by a space thermostat. Accurate control of space temperature will be obtained regardless of whether there is a compensating humidity controller to reset the control point of the dewpoint thermostat. On the summer cycle, when dehumidification is usually accomplished by chilling the air, a reheater coil controlled by the space thermostat is required to provide enough heating to counteract the excessive cooling to effect necessary dehumidification. The capacity of the coil is predicated on the lowest sensible heat ratio anticipated, the lowest apparatus dewpoint available, and the quantity of air being circulated.

Direct Expansion Coil. In general, for direct expansion coils a space thermostat may be used to perform the following functions: open and close a solenoid refrigerant valve on the coil inlet; control the starting and stopping of one or more compressors, or otherwise control the compressor capacity of the system; position face and bypass dampers, in combination with the opening and closing of the solenoid valve or the starting and stopping of the compressor; control the sequence opening and closing of solenoid refrigerant valves arranged in parallel at the coil inlet; control a back pressure regulating valve with proportional action; or control a liquid refrigerant valve on the inlet to the coil with proportional action. On-off cycling of the refrigerant valve or compressor will give a wide variation, or "hunting" characteristic, to discharge temperature and humidity. To prevent this it is desirable to employ proportional damper controls or proportional control of refrigerant flow in conjunction with refrigeration equipment capacity controls. In small systems where the space thermostat controls positive shut-off of refrigerant flow to the cooling coil, a non-restarting relay is placed in the power supply circuit of the compressor to forestall unnecessary short-cycling of the compressor while the coil is off cycle.

In addition to providing cooling, a direct expansion coil may be used for dehumidification. On moderately cool but very wet summer days the direct expansion coil may be used for dehumidification if a reheater is provided to maintain the space temperature. For this service, the direct expansion coil may be controlled by the humidity controller, or the thermostat, each acting through the diverting relay.

Chilled water coil. The space thermostat may be used to control a throttling valve on either the inlet or outlet of a chilled water cooling coil; control a three-way valve mixing chilled and recirculated water; position face and bypass dampers at the coil; or position a damper which bypasses recirculated air around the cooling coil. It is imperative that no outside air be allowed to enter the conditioned space without passing through the conditioning coil. Otherwise, it would be impossible to control space humidity at a comfortable level. In laboratory apparatus the duct bypassing air around the conditioning coil is not considered good design practice since the impossibility of effecting tight shut-off precludes precise air flow measurement across the coil.

Although a chilled water system generally is best for large, rambling installations, the direct expansion type system has certain advantages. First, there is no change in temperature of the coil refrigerant along the path of air flow since the cooling is effected by a substance changing phase. Second, there is not such a marked change in coil surface temperature at low load conditions. Third, there is more flexibility in design because the direct expansion coil can operate at lower temperatures; consulting a psychrometric chart shows that having the required low apparatus dewpoint temperature will in many cases eliminate the need for reheating, or at least reduce the reheater load.

Water Chiller. As with cooling coils, the current practice is to use "dry" rather than "flooded" water chillers. There are two advantages. First, the return of entrained lubricating oil to the compressor is assured; second, the system does not require such a large quantity of refrigerant for operation. Control of flow rate generally is provided by a thermostatic expansion valve. The solenoid valve is controlled either by the temperature of water returning to the chiller or water leaving the chiller. A safety feature is an immersion thermostat which provides positive shut-off in case of malfunctioning to prevent freeze-up. Nevertheless, it has been found best not to rely exclusively on the freeze-up protection, but to provide full flow of water through the chiller at all times. Such a procedure is possible when the chilled water is being supplied to cooling coils through a three-way valve used to divert chilled water around the coil at partial load conditions. However, this is not possible when the chiller is used to condition water for a washer, since the washer requires full pressure for proper atomization at the spray nozzles. In this instance, it is necessary to watch the operation of the chiller controls closely during the period of low dehumidifying load at the washer.

Air Washer. Since the air leaving an efficient washer is practically saturated, the dewpoint temperature of the air leaving the washer is approximately equal to the temperature of the spray water in the washer tank near the end of the washer. A thermostat with sensing bulb located in the air leaving the washer can be used to maintain accurate dewpoint control, which will provide accurate control of relative humidity as long as the sensible heat ratio in the space does not change. On the winter cycle, the dewpoint thermostat controls either a heating coil in the air stream or a heater in the circulating water circuit. On the summer cycle, the dewpoint thermostat controls a direct expansion coil, the diverting valve for a chilled water coil, or the mixing valve in the washer water circuit. To compensate for changes in the sensible heat ratio in commercial and residential applications, a space humidity controller may be used as a master controller to readjust the dewpoint thermostat setting. The circulating water pump operates at all times.

Humidifier. The purpose of the humidifier is to provide humidification of the air on the winter cycle. Single-bank sprays are usually controlled by a two-position room or return-air humidity controller, or a wet-bulb thermostat. Any of these may be used to actuate a water line solenoid valve or else the water pump. Modulating pneumatic humidity controllers are able to operate in a two-position manner by means of a pneumatic-electric relay. Humidification on the summer cycle is a rare and special requirement, but can easily be accomplished by the controllers and apparatus usually installed for the winter cycle.

Outdoor Air Dampers. In central fan systems it is customary to have an outdoor damper open whenever the fan is running to admit enough air to meet minimum ventilation requirements. For economical operation it is possible on the winter cycle to use outdoor air in a separate duct system for cooling the interior zones of a large builing which show a net heat gain throughout the entire year. On the summer cycle, economies can be effected in mechanical cooling when the outside temperature is mild by having a combination of outdoor-air and return-air thermostats operate to select the cooler source of air. Besides providing the necessary controls, it is only necessary to have the outside air duct large enough to admit all outside air to the system, and to be fitted with suitable tight-closing dampers. If possible, the return-air and outside-air dampers should be interlocked to insure against maladjustment of the controllers.

Fan. In central fan systems it is generally considered the best practice to operate the fan continuously. Continuous fan operation prevents stratification of air in the conditioned spaces and provides for continous cleaning of the air. The control system should be so arranged that all controls assume a safe, or "normal," position in case of power failure at the fan motor. This is accomplished by an interlocking relay which shuts down power to the control system when the fan stops.

METHODS OF AUTOMATIC CONTROL

Applications of automatic control systems range from simple residential temperature regulation to precision control of complex industrial manufacturing processes. The air conditioning engineer has three kinds of control systems from which to choose, each of which has definite advantages. They are electric, electronic, and pneumatic. In many cases one would perform the services as well as the others, and the choice is arbitrary. With any of the systems it is possible to have two-position control, floating control, or proportional control. Each will be described briefly and its dominant advantages listed.

Electric. With an electric control system, the reaction of the sensing element, or controller, to a change in conditions is transmitted electrically to the actuator. For this purpose, electricity has the following advantages:[33]

(1) It is available wherever power lines run.

(2) Electric wiring is usually easy to install.

(3) Electric power readily amplifies the relatively feeble impulse received from the sensing element of the controller, making it possible to control systems which ordinarily could not be controlled.

(4) The impulse received from the sensing element can be applied directly to produce one or several combinations or sequences in electric output.

Thus, one actuator can be made to perform several desired functions.

(5) It readily permits controlling from a remote point.

Electronic. Electronic control systems are applied where an extremely rapid rate of response must be made to a small incremental change in the controlled variable. Electronic control systems are different from other electric control systems.[33] The sensing function is performed by a length of temperature-sensitive wire wound on a bobbin. The electrical resistance of the wire for a given temperature, and the rate of change with respect to temperature changes are known. By exposing the wire to the temperature to be measured and determining the electrical resistance of the wire, a quick and accurate measure of temperature is obtained. Resistance is measured by unbalance of a bridge circuit. The electronic relay detects the unbalance, and amplifies the minute signals produced by the sensing element to energize a circuit which can control a conventional electric operator. Thus, electronic control systems consist of an electronic sensing device, an electronic relay, and conventional electric actuators.

Pneumatic. Pneumatic systems are commonly used in commercial and industrial buildings, though, like electric and electronic, they are applicable to all systems regardless of size. Some of the advantages are:[37]

(1) Pneumatic equipment is inherently adapted to modulating operation, but positive or two-position operation may readily be provided.

(2) A great variety of control sequences and combinations can be provided with relatively simple equipment.

(3) Pneumatic equipment is trouble-free and safe in operation, and any operating difficulties are simple to trace and correct.

(4) Costs of equipment, installation, operation, and service are relatively low. This is especially true on large jobs, and in existing buildings.

At the present time, most commercial and industrial control systems are pneumatic.[63] Most residential systems are electric. Though not conspicuously more expensive than the other two systems, electronic control systems are applied only where an extremely rapid rate of response is needed, as in industrial processing and in precision laboratories. In comfort conditioning, the electric and pneumatic systems are still able to sense a change in the controlled variable and correct it before a person in the space can feel a change in conditions. Pneumatic systems have the longest estimated life and the best reputation for trouble-free operation and economy in maintenance.

D. AIRBORNE IMPURITIES

In the Code of Minimum Requirements for Comfort Air Conditioning,[16] air conditioning is defined as "the process by which simultaneously the temperature, moisture content, movement, and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. If an installation cannot perform all these functions, it shall be designated by a name that describes only the function or functions performed."

With an increase every year in the number of factories, machines, and other sources of atmospheric pollution, the necessity of better and cheaper air purification equipment becomes apparent. Motor control rooms of factories have conditioning systems with electronic filters which remove all impurities to protect expensive, delicate equipment. Investigations have shown that good purification apparatus in the return air cuts down greatly on the initial and operating costs in comfort installations.[60] A review of current literature will show the concentration and interest of research facilities currently developing and improving air purification methods.

This section will first describe those things which are regarded as impurities, their origins, and their effects on human comfort and health. Second, the impurities will be classified according to size and other properties which point toward the most efficient methods of removal from the air. Third, the modern common methods and apparatus employed for air purification will be described and evaluated.

IMPURITIES AND THEIR ORIGINS

The normal constituents of the earth's atmosphere are oxygen, nitrogen, carbon dioxide, water vapor, argon, small or negligible amounts of other inert gases, hydrogen, variable traces of ozone, and small quantities of microscopic and submicroscopic solid matter, sometimes called permanent atmospheric impurities.[38]

Solid Particulate Air Contaminants. Solid particulate air contaminants may be classified as dusts, fumes, and smokes.[38] Dusts occur as the disintegrated form of solid matter. Dusts may be of the mineral type, such as rock, ore, metal, sand, and rubber; vegetable, such as grain, flour, wood, cotton, and pollen; or animal, such as wool, hair, silk, feathers, and leather. Dust particles are smaller than 100 microns diameter. (One million microns equals one meter; 24,500 microns equal one inch.) During its life a single automobile tire, by friction with the surface on which it travels, will produce about 750 billion rubber dust particles.[48]

Fumes are solid particles commonly formed by the condensation of vapors from normally solid materials such as molten metals. Metallic fumes generally occur as oxides in air because of the highly reactive nature of finely divided matter. They are predominantly below the one micron size.

Smokes are the extremely small solid particles produced by incomplete combustion of organic substances such as tobacco, wood, coal, and oil. The products of incomplete combustion such as carbon particles, fly-ash, cinders, tarry matter, and unburned gases as a group are commonly referred to as smoke, though technically the term smoke refers only to solid particulate matter. The finest particulate constituents are in the range of 0.1 to 0.3 micron in size.

Liquid Particulate Air Contaminants. Mists and fogs are liquid particulate air contaminants. Mists are very small airborne droplets of materials that are ordinarily liquid at normal temperatures and pressures. They are commonly evolved from industrial processing equipment. Also, very small droplets expelled or atomized into the air by sneezing constitute mists containing microorganisms that become air contaminants.

Fogs are airborne droplets formed by condensation from the vapor state. An example will illustrate the size of the droplets: The water vapor condenses on nuclei consisting of dust particles suspended in the air. In a rectangle of fog 100 feet long, six feet high, and six feet wide there are about 120 billion droplets which comprise less than half a glass of water.[36]

Non-Particulate Air Contaminants. Vapors and gases are non-particulate air contaminants; they are the superheated phase of substances that are either liquid or solid in their commonly known state, such as pyridine, chlorides, and benzene. Vapors may be changed to the solid or liquid phase by increasing the pressure, decreasing the temperature, or both.

Various methods of eliminating air-entrained gaseous and vaporous contaminants have been employed.[61] One approach has been the introduction of chemical agents either directly into the conditioned space or else into the recirculated air stream to react with the impurities in a manner tending to destroy or otherwise alter their objectionable character. One obstacle to this method is that the chemical agents used are often toxic. Another obstacle is that if weaker chemical agents are used which are not toxic and do not decompose or react with the impurities, they are generally merely a screen for masking the disagreeable odor with a stronger one of presumably more agreeable characteristics. Some deodorizing substances contain ingredients such as formaldehyde which deaden or anesthetize the olfactory nerves, thus preventing the detection of either the masking or the offending odors.

EFFECTS ON HEALTH AND COMFORT

Solid Particulate Air Contaminants. Mist dusts are not harmful until heavy concentrations are reached. The general study of dust with relation to health is called pneumoconiosis (meaning dust-lung). Of the more common industrial dusts -- silica, coal, stone, and iron -- silica is the most injurious to health. Particles small enough to enter the lung cells (about 10 microns and less) in small concentration are either oxidized by the blood stream or removed by the lymphatic system to lymph nodes where they are deposited. When the rate of dust entrance becomes excessive, the lung eventually becomes pathologically incapacitated and death results.

Fumes are generally the product of industrial processes and vary from slightly toxic, as acetone or toluene, to lethal, as nitric and hydrofluoric acids.

Smoke is an old problem. Because it was a nuisance and believed harmful to health, the use of coal was banned by royal proclamation in England in 1306. One man, reported to have been caught burning coal, was condemned to death and executed.(48) The effect of the solid particles in smoke is not well determined. It is known that people living in metropolitan areas such as New York City have dark, almost black, lungs, while people living in rural areas characteristically have natural pink lungs.(25) Absorption of solar ultraviolet light by smoke and soot is recognized as a health problem in many industrial cities.[38] Smoke causes a loss of 50 per cent of ultraviolet light intensity in Baltimore, and a loss sometimes as high as 50 per cent of total light intensity in New York City.

Liquid Particulate Air Contaminants. Mists and fogs cause gaseous contaminants to accumulate; combined with moisture the contaminants may cause irritation to the eyes, nose, and respiratory passages, resulting many times in illness and death.[38]

Non-Particulate Air Contaminants. Vapors and gases are harmful according to the nature of the substance and the concentration. As constituents of smoke, vapors and gases are not removed even when electrostatic arrestors eliminate the nuisance of effluent solid particulate matter; a conspicuous example of the damage is the barren landscape, devoid of vegetation, surrounding paper mills. The damage to human health can be inferred. Pollution in industrial districts is caused by industrial processes, trash burning, fuel burning, and automotive vehicle operation. When these vapors and dusts combine with water vapor causing fog, the result is called smog. Normally the atmosphere does not become sufficiently polluted to cause any distress, but in foggy weather, during periods of low air velocity, or during periods of temperature inversion in the atmosphere, the accumulation may become sufficiently great to cause eye irritation, irritation of the respiratory passages, and even death to those afflicted with respiratory or heart ailments.[60] In a study conducted by the Stanford Research Institute, it was found that no single contaminant in the concentration found in Los Angeles smog caused distress. Irritation was caused by specific combinations of vapors and particulate matter.[60] No simple remedy is apparent at the present time.

Internal pollutants presenting problems in comfort air conditioning are the expired air from human beings and body odors. Extensive research on the nature of expired air from human beings was done by Drs. J. S. Billings and S. W. Mitchell in 1893-1895 for the Smithsonian Institute.[60] Since then many less inclusive studies have been made which validify the results of the original investigation in every detail. The conclusions of the investigation may be summarized as follows:

(1) Only in coughing or sneezing is particulate matter discharged from the lungs. No volatile poisonous matter is expired other than carbon dioxide.

(2) Air expired from the lungs has no odor. Bad breath is generally due to an unclean mouth and teeth condition.

(3) The concentrations of carbon dioxide and diminu-

tion of oxygen which is found in poorly ventilated schools, theaters, and barracks is not sufficiently great to account for the discomfort that such conditions produce in many persons. The main causes are excessive temperature, excessive humidity, and unpleasant odors.

In 1936 the A.S.H.V.E., in cooperation with the Harvard School of Public Health, undertook a study of ventilation requirements for control of body odors.(71) Several conclusions of the study are:

(1) Wide individual variation exists in the amount of odor emmitted by various groups of people, according to age, sex, bathing habits, and cleanliness of clothing. Even healthy clean adults, just after a bath gave off appreciable odor, which required from 15 to 18 cfm of unconditioned outdoor air per person in order to dilute the indoor air odor to a level that was not objectionable to persons entering the room from relatively clean air.

(2) The olfactory organs are quickly and easily fatigued, though they can easily perceive the sudden appearance of new odors. The occupants of a crowded and poorly ventilated room are not aware of the body odors which may even be intolerable to a newcomer. Breathing odor-free air quickly restores the sensitivity.

(3) The sensation of odor follows the Weber-Fechner law of psychological reactions:

Sensation = K x log of stimulus

In terms of outdoor ventilating air supply for dilution:

Odor intensity = k x log(1/outdoor air supply)

According to this formula, it takes a considerable quantity of outdoor air to change the odor intensity appreciably.

The effect on human health and comfort, then, of body odor is actually nil; but until this fact is universally known there will be a great deal of psychological suffering which, after all, is as uncomfortable and sometimes as harmful to health as exposure to poisonous substances.

SIZES OF PARTICULATE MATTER

The small particles which contaminate the air are generally invisible to the naked eye. Over a smoky industrial city, a single cubic foot of air may contain from 200,000 to 4,000,000 such particles. The smallest particle the unaided eye can distinguish is about 10 microns in diameter.[48] A micron-sized particle magnified to the size of a fine pencil dot is in the same proportion as a penny magnified to the size of a football stadium.

Solid particulate matter may be classified according to four zones in order of increasing size: Brownian, Cunningham, Stokes, and Newton.[60, 35]

Brownian Zone. The Brownian zone includes particles of 0.1 micron diameter and less in size. These particles do not settle; their movement is based solely on impact by gas moleoules in air which are in constant motion at extremely high velocities. Brownian particles are too small to have any effect on human comfort; they have only minor effect on visibility; and they will not sustain bacterial life for any appreciable length of time. They are not removed by any known filtering device, so it is fortunate that they are unimportant.

Cunningham Zone. The Cunningham zone includes particles from 0.1 to 1.0 micron diameter. Any slight turbulence of the atmosphere will keep them in suspension.

Like particles in the Brownian zone, they are considered to be a part of the permanent atmosphere and are removed primarily by rain or mist. Particles of this size comprise smokes and industrial dusts. They adversely affect visibility and sustain bacterial life for a short period of time. The electrostatic precipitator and washer are effective in removal of these particles.

Stokes Zone. The Stokes zone includes particles from 1.0 to 200 microns diameter, and is considered the most important zone. It comprises dusts, fly ash, pollens, tarry matter, etc. Bacteria is carried and sustained on these particles. In still air these particles settle at a rate which can be calculated from Stokes' laws.[35] As an example, a particle of 20 micronsdiameter and unit density leaving a 200 foot chimney, in a 15 mile wind, will be carried 21 miles before touching ground. A good dry type filter will remove particles of 1 micron diameter while a good viscous impingement filter will remove particles not smaller than 5 microns.

Newton Zone. Particles in the Newton zone are more than 200 microns in diameter. They settle with increasing velocity by gravitational pull. After a terminal velocity is reached, the rate of fall varies as the square root of the particle diameter and is substantially independent of viscosity.

E. METHODS OF AIR PURIFICATION

After the size and nature of the contaminating substances have been determined, it is possible to install efficient equipment which will remove all of those pollutants reviewed which cause human discomfort or industrial annoyance. A general classification would include viscous impingement filters, dry filters, electronic filters, washers and scrubbers, and adsorption filters.

VISCOUS IMPINGEMENT FILTERS

Two general types of viscous impingement filters are available: the unit type and the automatic type. The unit type is a replaceable filter made in standardized convenient face dimensions, and one to four inches deep. The most common material is a fine gage glass fiber for the disposable type and expanded aluminum for those which are washed and re-oiled when dirty -- the so-called permanent type. Filtering action takes place by impingement of impurities against an adhesive liquid. The amount of impurities removed by a viscous type filter is a function of the number of impingements. Each impingement removes approximately sixty per cent of the impurities passing it.[29] To be satisfactory, the filtering adhesive liquid must have good wetting power to retain the impurities; stable viscosity; sufficient fluidity to spread a thin, even coating; stability against evaporation; absence of odor; germicidal action to prevent growth of bacteria; and fire resistance.[29] An example is "Viscosine," a water-soluble proprietary oil sold by the Ameriacn Air Filter Company.

The density of the packing determines the air resistance through the filter, controls the efficiency for a given fiber, and limits the dust-holding capacity. Many commercial filters are designed to get the best combination of these factors by packing the fiber progressively, increasing the density toward the air-leaving side.

Automatic filters are more expensive and require far more space than the unit type. The filter medium, consisting of metal screens or plates, is a chain-and-sprocket-driven double curtain which continuously cleans and renews the oil coating in an oil bath at the lower end of travel. Plates forming the filtering medium or curtain overlap each other, and due to their special shape, many small air passages are formed between them. These air passages turn the air abruptly one or more times in order to give an impingement effect. The resistance of automatic filters is approximately constant; typically it is 3/8 in. water gage at a face velocity of 500 fpm. For unit filters resistance ranges from 0.1 to 0.2 in. water gage at a face velocity of 300 fpm. Sometimes a manometer indicator is installed across the filter bank.[38, 35]

DRY FILTERS

Dry filters, like viscous impingement type unit filters, may be of the cleanable or disposable types. They may be of similar construction and material to the viscous type, or they may be of paper or cloth sheets. Those similar in construction to viscous unit filters are inferior and seldom used now because of the greater efficiency of the viscous impingement principle. Paper used in the dry type filter resembles cloth in its fine porous structure. It is usually mounted in metal frames with deep folded construction to permit low velocity through the material. Cloth for such purposes must collect most of the dirt upon a hairy surface to prevent the dirt from plugging the holes. That is, impurities are caught by the "tentacles" protruding from the main surface. Some dry filters depend simply on straining out the impurities. Efficiency is a function of fiber diameter, packing density, and the thickness of the filtering medium. The smaller the mean fiber diameter, the smaller the dirt particles which will be strained out. A recent important advance is the availability of glass fibers in a wide choice of controlled diameters, permitting improvements in the performance of dry filters to equal and in some cases exceed electrostatic precipitators. An outstanding example is the "absolute filter," which is used for the filtration of radioactive dust and removes, by number, 99.98 per cent of all particles down to 0.3 micron.[29]

One of the features of the dry-type filter is that it can be vibrated by hand intermittently to remove most of the accumulated soil. Also, it can be located close to heating coils.

The recommended method for testing viscous and dry filters is that prescribed in the "A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work."[15] This involves measuring the percentage removal, by weight, of a standard dust. The disadvantage of any weight method is that the effectiveness with fine particles is masked, since (assuming equal density) one 10-micron particle weighs 1,000 times as much as a 1-micron particle. Particles of 1-micron size, incidentally, deposit the worst stains. Therefore, a viscous filter with an arrestance, or weight removal, of 80 per cent may have a discoloration efficiency, or total removal, of only 15 to 20 per cent.[29]

Filters operate with equal effectiveness whether at the fan suction or discharge. For compact installations it is standard practice to install them at the suction, which gives more even distribution of the air across the filter face.

ELECTRONIC FILTERS

The electronic filter, also called the electrostatic precipitator and the electrostatic air cleaner, is commonly regarded as the most efficient device for removing solid particulate matter from air. It can remove all particles over about 0.1 micron in diameter. This insures the removal of air bacteria, which is carried by such particles. It will not remove vapors and gases, the usual sources of odors since they will not accept the electric charge.

The present form of the electronic filter, which evolved from industrial 75,000-volt electrostatic precipitaters, was patented by Dr. G. W. Penney in 1938. At the reduced voltages used in air conditioning applications, the electronic filter does not generate ozone in intolerable amounts. In the Penney filter, the air stream was drawn through an ionizing field. The ionizer consisted of tungsten wires and grounded tubular electrodes, installed vertically. The electrodes were spaced approximately 2-1/2 in. apart, with the tungsten wires between. Suspended on insulated supports and maintained at a positive potential of 12,000 volts, the ionizer wires set up a powerful electrostatic field. This charge, flowing from the wires to grounded electrodes, hit the dust particles and caused them to lose a (negative) electron and become positively charged. Immediately beyond the ionizer was a series of vertical plates spaced about

3/8 in. apart. Alternate plates were charged positive and negative with about 6,000 volts. The charged particles were attracted to the negative plates and lost their charge.

In the present commercial precipitator a "power pack" is furnished as part of the unit, attached directly or remotely, which takes 115 volt alternating current, transforms it to the required high voltages, and rectifies it to direct current. The energy requirement for operation is about 15 watts per 1000 cfm. The resistance to air flow is negligible. Care is necessary in arranging the duct approaches on the entering and leaving sides to assure that the air is uniformly distributed over the cross-sectional area. Perforated metal baffles help prevent this stratification.[5]

There are two ways to prevent re-entry of the neutral dust particles resting on the collector plates into the air stream. One is by placing a mat of fine-gage glass fiber, coated with a viscous liquid, downstream from the collector plates. When the build-up of impurities on the plates becomes excessive, the accumulation is carried by the air stream onto the automatically renewing media; here the dirt and used media are tightly wound into a compact roll for disposal about twice a year.[12] (Example: American Air Filter "Rollotron.") The second method is to coat the collector plates with an adhesive liquid which traps the impurities. It may be automatic in operation (Example:

American Air Filter "Electro-Matic.")[11] or it may require manual cleaning. In the non-automatic type, the plates may either be removed and cleaned, or cleaned in place. The collector plates are cleaned with hot water and re-oiled with a viscous liquid at a frequency varying from two to eight weeks.[13] (Example: American Air Filter "Electro-Cell".) All electrostatic precipitators should be preceded by a mechanical filter to remove the bulk of impurities, and followed by a mechanical filter, as a safety precaution should the electrostatic unit be neglected and excessive build-up occur.

It should be pointed out that any purification apparatus will only clean the air that actually flows through it. In critical applications such as hospitals, a corollary to efficient air purification is the use of oiled floor mops and water soluble oils in laundering drapes and blankets, which help keep down particulate matter on which bacteria are carried and sustained.

Electrostatic precipitators are tested by the U. S. Bureau of Standards blackness test.[32] Air samples from upstream and downstream of the electrostatic filter are drawn through filter papers. The ratios of the areas of filter papers and the ratio of the amount of air drawn through the filter papers are adjusted to yield spots of equal blackness. A photometer is used for comparing the spots. The efficiency is computed from the ratio of areas, the volumes of air samples, and the photometer reading. The efficiency of each filter is rated at 500 fpm (standard) or at 400 fpm face area velocity.

AIR WASHERS

Air washers will remove a wider variety of air pollutants than any other single type of air cleaning device; their sole limitation is that they will not remove greasy particles; that is, particles not soluble in water. For complete cleansing of the air, a detergent may be added to the water, or a viscous unit filter and an electrostatic filter may be used in conjunction with the washer to remove tarry particulate matter. The major occurrence of tarry particulate matter is in tobacco smoke. Other troublesome components of tobacco smoke are the vapors of pyridine and nicotine. Pyridine is a colorless liquid with a pungent odor which is extremely irritating to the eyes and mucous membranes of the nose and throat. Nicotine is a colorless, bitter liquid which is exceedingly poisonous. Both can be effectively removed by an air washer if the water is kept fresh by frequent changing.[60]

An air washer consists essentially of a chamber or casing enclosing a spray nozzle system. A tank at the bottom of the chamber collects the spray water as it falls, and an eliminator section at the leaving end of the chamber removes drops of entrained moisture from leaving air. A washer is always placed at the suction side of the fan; on the discharge it would be almost impossible to prevent water leaks. Air is drawn through the inlet of the washer, where it comes into intimate contact with the spray water. A heat transfer takes place between the air and water, resulting in either humidification and cooling, humidification and heating, or dehumidification and cooling, depending on the relative temperatures of air and spray water.

Essential requirements for proper air washer operation are: uniform distribution of entering air across the chamber inlet; moderate air velocity in the washer chamber, ranging from 250 to 600 fpm; good atomization of the spray water in the path of the air; sufficient length of travel through the water spray and scrubber surfaces; and the elimination of entrained moisture from the outlet air.[38]

Where nozzles spray against the flow of air, vertical eliminator baffles should be provided at the washer entrance. An inlet diffusion plate should be placed at the washer entrance to prevent stratification of air flow. At the outlet end of the washer suitable eliminator plates are required.[38] These eliminators are made up of a series of corrugated plates, spaced about one inch apart and standing vertically. Each eliminator plate is made of a single galvanized sheet. The entrained moisture is removed by projecting lips or gutters in the corrugations.[23]

Except where the dust removal problem is severe, the scrubber plates are integral with the eliminator plates. A scrubber is a set of plates similar in construction to the eliminator plates, but not requiring the projecting lips.

The plates are kept wet by a set of flooding nozzles with course spray at a low pressure. The cleaning action of the washer is accomplished by the vapors and odors going into solution with the water spray. The solid particulate matter is wetted by the water and trapped on the scrubber surface.

In conditioning the air, the efficiency of an air washer is defined as the ratio of the actual drop in dry-bulb temperature to the maximum theoretical drop in dry-bulb temperature that could take place if the air emerged saturated. In a perfect washer, the final dry-bulb and dewpoint temperatures of the air would be equal to each other, and equal to the initial wet-bulb temperature of the air.[67]

Washer Limitations. In the past, air washers have been a standard part of the conditioning systems of most large commercial buildings, factories, and laboratories. But washers have limitations. First, they are bulky, noisy, and expensive to build and maintain. The effectiveness of washers in eliminating gaseous and vaporous impurities from air is confined to the relatively few gases and vapors which are water soluble. Though this limitation affects in no way a washer's effectiveness in applications such as textile mills, most airborne odorants are organic substances which are insoluble in water and hence cannot be removed by water scrubbing. These insolubles include saturated and unsaturated hydrocarbons, sulfur and nitrogen compounds, esters, and most of the odorous acids, aldehydes, and ketones. It is sometimes possible to treat the water to obtain solubility for some gases; for example, an eight per cent caustic water solution in a spray type washer effectively extracts airborne sulfur dioxide.[61] However, this makes necessary controlled maintenance of the strength of the water solution and the constant change of water to prevent accumulation and concentration of dissolved impurities. Even with the closest control, it is seldom possible to prevent some re-evaporation and escape of dissolved impurities.

ADSORPTION FILTERS

Any gas or vapor will, to some degree, adhere to any solid surface at ordinary or low temperatures. This phenomenon is called adsorption. Adsorption may occur either as physical or chemical adsorption. In physical adsorption, the attraction of the sorbent for the sorbate does not involve chemical reaction, and the heat liberated by the adsorption of the gas or vapor is approximately that which would be liberated by the simple condensation of the same kind and quantity of gas or vapor.

In physical adsorption gases and vapors, the adsorbates, are collected in a condensed state upon the surfaces of a solid, the adsorbent. Only certain specific solids whose structures include highly convoluted surfaces and a vast network of sub-microscopic pores and channels are practical adsorbents. It has been estimated that the aggregate area of the surfaces in one pound of granular activated carbon approximates 140 acres.(61)

This adsorptive characteristic is possessed not alone by carbons, charcoals, and chars, but also by such materials as zeolite, silica gel, alumina, and mica. All of these differ widely in the number and kinds of substances they will adsorb, as well as in the amount of sorbed substances they will retain. In general, the siliceous, metallic oxide and active earth types of adsorbents are electrically polar;

that is, their molecular structure contains an unsymmetrical electron distribution. Since polar substances have strong attraction for one another and since water is highly polar, the polar adsorbents mentioned above retain water in preference to most other fluids and are, therefore, incapable of adsorbing nonaqueous gases efficiently from a humid atmosphere. At the present time, physical adsorption is not well understood as a physical phenomenon. Chemical adsorption, or chemisorption, is the chemical reaction between a sorbent and a sorbate. The bond is usually very strong and the reaction is difficult to reverse.

In physical adsorption the degree of attraction is influenced by the molecular characteristics of the sorbate which, in turn, are related to its critical temperature and boiling point. These factors, therefore, provide a means for predicting the degree of adsorption to be expected. True gases having critical temperatures well below minus 50°C and boiling points below minus 150°C are virtually non-adsorbable at ordinary temperatures. The extreme mobility of their relatively light molecules creates an escaping tendency greatly exceeding the attractive force of the carbon. Such true gases include hydrogen, nitrogen, oxygen, carbon monoxide and methane. The non-adsorbability of these gases is fortunate in comfort conditioning because it assures that the chemical composition of the air will not be disturbed. Low boiling vapors having critical temperatures between approximately 0°C and 150°C and boiling points between minus 100°C 0°C have a lesser escaping tendency and are, therefore, moderately adsorbable. These vapors include ammonia, ethylene, formaldehyde, hydrogen chloride, and hydrogen sulfide.

Carbon has a very high adsorptive capacity for high boiling vapors (above 0°C) because the molecular escaping tendencies are low. This makes air recovery extremely effective in comfort conditioning work because the types of contaminants common to air pollution, particularly in inhabited areas or enclosures, are almost entirely vapors in this category. They include most organic compounds, as hydrocarbons, alcohols, ester, aldehydes, ketones, organic acids, nitrogen and sulfur compounds, and many inorganic vapors.

Charcoal made from coconut shell has proven to have a finer structure and a greater adsorbing surface than charcoal of any other type.[60]

Another approximate criterion for predicting the adsorbability of gaseous or vaporous substances is molecular structure. The adsorptivity of gaseous substances increases progressively with their molecular size. Thus, with vapors comprising a regular organic series, such as the hydrocarbon or the alcohol series, each member of the series is sorbed more readily than the next lower or lighter member. As a practical rule for air purification, any gas or vapor whose molecule contains fewer than three atoms independent of hydrogen is not practically adsorbable.

Two terms are particularly important in connection with adsorption filters.[61] "Activity" is the capacity of a sorbent for a particular gas or vapor, and is taken as the ratio of the weight of sorbed substance to the weight of the carbon when the carbon ceases to increase in weight when exposed to a stream of dry air saturated with the vapor or gas in question. "Retentivity" is the retentive capacity of a sorbent, and represents the practical saturation limit of the sorbent under conditions common to ventilation. It is measured by passing clean dry air at constant pressure and temperature continuously through a bed of granular carbon previously saturated with a specific gas or vapor and continuing the air flow until the carbon ceases to decrease in weight. This method determines the weight of the respective gas or vapor that the carbon will retain when exposed to clean air at specified conditions. The "retentivity" is the ratio of the weight of the retained substance to the weight of the carbon.

Water vapor is one substance for which activated carbon has special sorptive characteristics. While its activity for water vapor is high, its retentivity is practically nil.[61] In ventilation, therefore, the amount of water sorbed, or remaining sorbed, varies with the relative humidity of the air stream. Further, sorbed vapors and gases cause the sorbed water to leave the carbon and progressively reduce the sorptive capacity for water vapor. It is for this reason that activated carbon air purification applied to ventilation is equally effective under humid conditions.

When the carbon has reached its practical saturation in service, it must be removed for reactivation. As the retentive capacity of the carbon is an inverse function of the temperature to which it is exposed, reactivation consists essentially of heating the carbon in a controlled atmosphere until all the adsorbed gases and vapors are removed by desorption, displacement, or oxidation. The reactivating atmosphere may be an inert gas, superheated steam, or combustion gas of carefully controlled composition. The ultimate temperature necessary will depend upon the boiling points and critical temperatures of the sorbed substances and upon the nature of the reactivation atmosphere. In general, a temperature of 1000° or higher is required to completely release all of the gases and vapors accumulated in comfort conditioning.

In commercial practice it is necessary only to reduce the concentration of obnoxious gases to the established level, not to accomplish complete removal. Generally, adsorption filters should be preceded by a good dry filter and an electrostatic precipitator. Viscous filters should be avoided since the released oil vapors will shorten the service life of the adsorber. With the proper selection of dry filter, precipitator, and adsorber, the outdoor air quantity may be reduced to that amount required for pressurization to overcome crack losses of the structure.

SPECIALIZED AIR TREATMENT

Considerable research has been done in attempting to duplicate the stimulating quality of outdoor country air by ionization of the air handled by the air-conditioning system. In 1958, John C. Beckett, Chief Engineer of Wesix Electric Heater Company, reported on the role of ions in air conditioning.[22] Ions are generated from radioactive material in the soil and from cosmic rays, and do influence the human respiratory system as had long been supposed. In air cleaning equipment the inefficient collection of positive ions may depress the negative ion density enough to cause nose and throat irritation. Consequently, the most common situation in an air conditioned space is an inadequate supply of negative ions. A portable type negative ion generator has been developed which is mounted at the air supply outlet to the conditioned space, which avoids absorption of the ions by the cleaning apparatus.

In 1959, Rudolph A. Nagy, manager of Westinghouse ultraviolet development, reported[55] that ultraviolet lamps generate negative air ions, which are believed to have an exhilarating effect on the human organism. Positive ions are believed to have a depressing effect.

Ultraviolet sterilizing lamps have a demonstrated bactericidal action[29] both when mounted in a supply duct and when mounted in the conditioned spaces. Since an electrostatic precipitator is effective in removing all particulate matter on which bacteria may be carried, and hence in removing the bacteria as well, the principal application for ultraviolet lamps is in killing germs arising directly from occupants in the conditioned spaces.

Glycol vapor, particularly triethylene glycol, in minute concentrations in the air, exerts a powerful sterilizing action, provided the glycol dewpoint and air relative humidity are kept within certain limits.[29]

Ultrasonic vibrations are used in some cases to agglomerate very fine particles of matter into larger sized particles which may then be removed from the air stream by one of the previously mentioned methods.[35] Vibrations may be caused by electrical or pneumatic means and are usually in the range of three to five kilocycles per second. The ability to agglomerate depends to some extent on the nature of the material, size of particle, and power used.

F. STEAM AND WATER AIR CONDITIONING COILS

General Theory. The type of coil used in modern central fan systems is the light weight, extended surface, header-and-tube type. For heating service, coils are used for tempering, preheating, heating, reheating, or booster heating. The usual heating media are steam and hot water, though an economical method of summer reheating is by means of the hot discharge gas from a compressor.

Coils used for air cooling may cool and dehumidify or may remove sensible heat only. An example of cooling without dehumidification is a precooling coil using well water to reduce the load on the mechanical refrigerating equipment, or used in conjunction with a chemical dehumidifier. Formerly, the use of sprays with dehumidifying coils was popular since the arrangement was compact and obtained a good degree of air cleaning and odor absorption. The nuisance of frequent maintenance to remove scale deposits from the coil has brought about the development of efficient dry type filtering media as described earlier to replace the spray.

Heat transmission in extended surface type coils is impeded by three resistances: the outside air-film, the metal wall, and the inside fluid-film. Both the resistance of the metal wall to conduction and the inside fluid-film resistance are small compared to the outside air-film.[38] Design practice is to decrease the external surface resistance to approach that of the tube wall, and that of the inside fluid-film. This is accomplished by increasing the outside surface area by fins. The former practice of spraying the coil with water did not increase the overall heat transfer greatly. In fin or extended surface coils the external surface of the tubes is known as the primary surface, and the fin surface is known as the secondary surface. The primary surface consists of seamless round tubes of a metal suitable for the service; the most common sizes are 1/2, 5/8, 3/4, and 1 in. outside diameter. When more than one row is used, the pipes are staggered to give more air turbulence and consequently a higher heat transfer rate. The secondary surface commonly consists of spiral, flat, or corrugated fins. The design is evolving rapidly with the present need for more efficient heat transfer elements. Until several years ago a popular method for bonding the fin to the tube was by dipping the coil to coat the entire surface with solder. However, heat transmission through the solder is not as efficient as through the fin and tube metals, nor was the solder flexible enough to stand excessive strains. Modern practice is to bond the fin and tube using one of the several current patented methods of a mechanical press fit.

Common materials used in fin construction are phosphorized copper tubes and aluminum fins. Copper fins are required where the coils are wet by sprays or by excessive condensation of moisture. If immersed in a conductor, dissimilar metals cause galvanic action and electrolysis follows. Otherwise, aluminum is used because of its price advantage. The number of fins per inch of tube length varies from three to fourteen. Frost accumulation, dust, and airborne lint limit the spacing in difficult applications. Fourteen fins per inch is the most common spacing in comfort air conditioning. A typical height for fins is 13/32 inch, and tube spacing varies from 1-1/8 to 2-1/2 inches on centers.[7]

Free area through coils is usually sufficient, even in coils with fin spacing as dense as 14 per inch, so that condensed moisture will not be carried off the coil if the velocity of air, based on the entire face area, is under 500 fpm. Otherwise, eliminator baffles at the coil exit should be installed to prevent entrained moisture from entering the duct system beyond the coil. With tall dehumidifying coils, usually consisting of stacked separate coils, a collecting trough with drain to waste is placed at the bottom of each coil section.

Performance. The performance of heating and dry cooling coils depends in general upon:[38]

(1) The overall coefficient of heat transfer from the

fluid within the coil to the air it heats or cools.

(2) The mean temperature difference between the fluid within the coil and the air flowing over the coil.

(3) The physical dimensions of the coil.

Thus, for any one definite operating condition, the heating or cooling capacity of a given coil is expressed by the following basic formula:

$$q_t = U \times \Delta t_m \times A \times N$$

where:

q_t = total heat transferred by the coil in Btu/(hr)(sq.ft. coil face area)

U = overall coefficient of heat transfer in Btu/(hr)(sq.ft. external coil surface) ($^{\circ}F$ temperature difference between the fluid within the coil and the air flowing over the coil)

Δt_m = mean temperature difference, $^{\circ}F$, between the fluid within the coil and the air passing over it; commonly taken as logarithmic mean temp. difference

A = external surface area of the given coil, sq.ft./(sq.ft. of coil face area)(row of coil depth)

N = number of rows of coil depth

Of the above factors, "U" is the most difficult to determine. For finned coils it is calculated as follows:[38]

$$U = \frac{1}{\frac{R}{f_i} + \frac{1}{nf_o}}$$

where:

n = fin efficiency

R = ratio of total external surface to internal surface

f_i = fluid-film coefficient of heat transfer on inside surface of coil, Btu/(hr)(sq.ft.)(°F)

f_o = air-film coefficient of heat transfer on outside surface of coil, Btu/(hr)(sq.ft.)(°F)

The value of "R" varies from 10 to 30. It is put in the formula to place the internal surface coefficient of heat transfer on a basis of external surface.

In dehumidifying coils, the overall coefficient of heat transfer cannot be determined by the same method used for coils which transfer sensible heat only. The difficult problem is to determine the temperature of the wet coil surface. In the past, empirical approximations were used, based on experience with laboratory testing. In 1956, Dr. Tamami Kusuda of Worthington Corporation developed a precise graphical method for accurately determining the wet surface temperature.(46)

Cooling Coils. Cooling and dehumidifying coils are usually rated within the following limits:

Entering Air Dry-Bulb: 60° to 100°
Entering Air Wet-Bulb: 50° to 80°
Air Face Velocities: 300 to 800 fpm, 500 being common
Volatile Refrigerant Temperatures: 25° to 55° at coil suction outlet
Water temperatures: 40° to 65°
Water Quantities: 2 to 6 gpm per ton, or equivalent to a water temperature rise of 4° to 12°
Water Velocity: 2 to 6 fps

<u>Heating Coils</u>. The usual heating installation practice for steam and hot water coils is as follows: [39]

Air Face Velocity: 500 to 800 fpm, 500 being common
Delivered Air Temperature: varies from about 72° for ventilation only to about 150° for complete heating
Steam Pressure: 2 to 10 psig, 5 psig being common
Hot Water Temperature: 150° to 225°
Water Velocity: 2 to 6 fps
Water Quantity: based on about 20 degree temperature drop
Air Resistance: about 5/8 in. of water gage for public buildings and 1 in. w. g. for factories

In a year-around system, the air quantity through heating coils generally is made the same as that required for the summer load, which almost always is greater. The amount of resistance through the air circuit influences the fan horsepower and required fan speed.[1] In a water coil, the friction through the water circuit may be dictated by the head available from a given size of pump and pump motor.

<u>Coil Selection</u>. In the selection of a coil it is necessary to consider a number of factors:[38] duty required, whether heating, cooling, or dehumidifying; temperature of entering air; available heating and cooling media; space and dimensional limitations; air quantity limitations; allowable resistances in air circuit and through tubes; type of automatic control; and reputation of the manufacturer. In transfer of sensible heat only, the usual method is to select a coil that will effect the re-

quired temperature change of the nominal air flow quantity when a given amount of medium is available in the tubes at a given temperature. In dehumidifying coils the selection is made on the basis of entering and leaving wet-bulb temperatures.

In ordering coils for a specific installation the size is determined from the performance data and rating charts of a reputable manufacturer.

Chilled and Hot Water Coils. An increasingly popular type of coil both for central systems and for packaged units is the finned coil which uses chilled water in the summer and hot water in the winter; the same pump operates year-around. Coils from four to twelve rows are common, with a counterflow arrangement. In counterflow, the cold water enters the coil at the opposite end from which the air enters. The amount of cooling surface required depends upon the temperature difference between the air and water at various points in the coil. Consequently, the mean temperature difference in counterflow is always greater than in parallel flow for a given set of conditions, and a smaller amount of surface can be used to transfer a given quantity of heat. Parallel flow is seldom used. True counterflow is not obtained with finned coils, as it is in concentric tube condensers. The air flows across the tubes rather than along the tubes, and the water enters at the lower part of the downstream end of the coil, traverses the tube circuit, and leaves at the upper part of the front of the coil. Nevertheless, the value obtained for the mean temperature difference is close to the value for true counterflow with the number of rows usually encountered.

In actual coils there are generally from three to five degrees difference in temperature between the air temperature leaving the coil and the water temperature entering the coil. Although the required initial temperature of the water must be below the final dewpoint temperature to which the air is to be cooled, the amount of coil surface required for a given heat load is determined, in part, by the difference between the final wet-bulb temperature of the air and the initial temperature of the water. The smaller this difference, the larger will be the amount of coil surface needed to remove a given amount of heat from the air. This final difference can be made a s small as desired by using sufficient coil surface. However, for a moderate amount of coil surface, this difference is usually held to about five degrees.

The final temperature to which the water should be allowed to rise in a coil depends on how much of the coil surface is to be used for dehumidification. Usually, the initial portion of the coil is dry, removing sensible heat only, and the remainder of the coil is wet, removing both sensible and latent heat. If the maximum amount of dehumidification is desired, the temperature of the water throughout the coil must be lower than the dewpoint temperature of the entering air, and the amount of coil surface must be great enough to give minimum bypass factor.

The temperature rise of the water is generally limited to from seven to twelve degrees, fifteen degrees being considered the maximum. Proper design will balance the requirements of coil surface, water entering and leaving temperatures, and quantity of water circulated by the pump.

The overall face area of a coil is generally based on an air velocity of 500 fpm. Higher velocities not only necessitate greater fan power because of higher air friction drop, but also require eliminators at the exit to prevent the air from carrying entrained moisture into the conditioned spaces. Dehumidifying coils always require drip pans.

Compared with an air washer, a chilled water coil will control the dewpoint of the leaving air on normal summer cycles just as efficiently, and require much less space. Typically, a coil 20 in. deep will provide the same capacity as a washer 9 ft. long. The coil, however, will not purify the air nor provide control of dewpoint on the winter cycle. On the winter cycle, the washer humidifies and heats. Both the coil and washer require an external water chiller, water heater, and circulating pump.

G. DIRECT EXPANSION COOLING COILS

Coils for volatile refrigerants present more complex problems of fluid distribution than do water or steam coils. Properly designed and applied, however, they function well and find almost universal application in central air duct conditioning systems and in self-contained units of all sizes.

There are two types: flooded systems and thermal expansion valve systems.[38] In a flooded coil, the circulation is similar to that in a water tube boiler. The liquid is maintained at the proper level by the action of a float regulator. Since the flooded coil requires a large charge of refrigerant and inherently tends to collect the lubricating oil from the compressor which is entrained by the refrigerant, it has generally been replaced by the "dry coil," or the direct expansion coil with thermal expansion valve.

With the thermal valve system, there are two factors to consider.[38] There must be, generally, more than one refrigerant feed for each thermal expansion valve to prevent excessive pressure drop, which would raise the coil evaporating temperature correspondingly. Increasing the number of parallel circuits decreases the refrigerant velocity and friction of flow. If there is a large pressure drop through the evaporator coil, most of the coil will operate at a considerably higher pressure than the predetermined operating pressure at the compressor suction. The second factor is in arranging the coil so that the required suction superheat can be attained with a minimum sacrifice in the performance of the coil as a whole. That part of the coil containing vapor only will have much less cooling capacity than an equivalent length of coil containing liquid refrigerant.

Expansion Valves. Since the development of the first mechanical refrigeration system, the foremost control problem has been the method of expanding a high pressure liquid refrigerant into a low pressure wet gas mixture in direct proportion to the evaporating rate of the coil. Years ago, a constant pressure (automatic) expansion valve was designed to operate from the pressure in the evaporator and keep it constant, since the refrigerant pressure in the evaporator determined the evaporator temperature. It has two principal objections for general application.[21] First, when the refrigeration demand drops off, causing a low evaporator pressure, the valve pin will move to the full open position, attempting to bring the evaporator pressure up to the pressure setting of the valve. This results in the liquid refrigerant flooding back to the compressor and causing serious damage to the compressor. Second, if the load increases and raises the evaporator pressure above the pressure above the pressure setting of the valve, the valve pin will move in a closing direction until the evaporator pressure is reduced to the pressure setting of the valve, thus starving the evaporator when the load demand is highest. This valve, then,is suitable only for constant load applications; it has been largely replaced by the thermostatic expansion valve.

The thermostatic expansion valve is a precision device designed to regulate the rate of liquid refrigerant flow into an evaporator in exact proportion to the rate of evaporation of the liquid refrigerant in the evaporator.[21] The amount of refrigerant gas leaving the evaporator can be regulated since the valve responds both to the temperature of the refrigerant gas leaving the evaporator and to the pressure in the evaporator. This controlled flow prevents the return of liquid refrigerant to the compressor. In coils with long circuits, and in every case where a refrigerant distributor is used, an external equalizer connection is required. Otherwise, an unbalance at the valve diaphragm caused by the large pressure drop would require a higher superheat at the coil outlet than that designed for. The external equalizer usually connects the lower part of the valve diaphragm with the coil outlet at approximately the same location as the remote bulb, which is filled with the same refrigerant as that used in the system and is connected to the upper part of the valve diaphragm.

Solenoid Liquid Valves. Though the thermostatic expansion valve is manufactured as a tight seating device, it cannot be depended upon for positive shutoff because the seating surfaces are exposed to erosion and its design tends to trap dirt and scale. In addition, on the off cycle the remote bulb will be heated by the surroundings enough to cause the valve to open.[21] For these reasons, a solenoid liquid valve ahead of the thermostatic valve is always recommended. To protect the operation of both these valves, an 80-mesh screen strainer should be located immediately upstream, in addition to any other strainers in the system.[21]

Distributors. With thermostatic expansion valves it is advantageous to keep the pressure drop through the refrigerant feeds as low as possible.[38] The feeds are laid out to expose each to the same mean temperature difference so that each will handle the same refrigerating load. A distributor is placed between the valve and coil inlets to proportion the liquid-vapor mixture equally among the feeds.

There are several patented distributors in use. One type[65] consists of a cylindrical housing having an interchangeable nozzle at the inlet which increases the velocity of the liquid-vapor mixture by means of a pressure drop, thereby homogeneously mixing the liquid and vapor and eliminating the effect of gravity. The nozzle orifice centers the flow of refrigerant so that it impinges squarely on the center of a conical button. The outlet passage holes are spaced around the base of the conical button. Depending on the load condition, the pressure drop through the distributor and connecting tubes varies from about 10 to 25 psi. Tubing lengths to all circuits from the distributor should be the same to maintain equal pressure drop in all circuits.

Another manufacturer(21) substitutes a smooth, contoured, "Venturi-Flo" approach to the conical button for the orifice.

Another type distributor has a weir type header for use on small coils, and a centrifugal type header for coils having tube lengths longer than 2 ft. 6 in.(5) The weir type separates the vapor from the liquid so that the liquid can build up a head and be metered through the graduated orifices on the sides of the tubes. The gases pass upward, while reversing their direction, into the ends of the large tube orifices and rejoin the liquid in each coil circuit. In the centrifugal header type a nozzle in the inlet accelerates the mixture of vapor and liquid flowing from the expansion valve through tangential slots to the whirling chamber. Centrifugal separation causes the liquid to pass through peripheral holes to each of the refrigerant circuits. The vapor at the center of the vortex moves into the header by a large hole and mixes with the liquid as it flows to the various circuits.

Air Velocity. The air velocity across the coil face area recommended for most dehumidifying coils in use is 500 fpm. This is a general figure because the actual velocity will vary according to the free area. Low air velocities result in inefficient use of the extended surface. When the air velocity approaches about 650 fpm, the danger of water being blown from the fins becomes imminent.[(49)]

H. AIR FLOW AND AIR TEMPERATURE MEASUREMENT

A.S.R.E. STANDARDS[20]

The official procedure for the air conditioning and refrigeration industry in designing air and temperature measuring apparatus is described and illustrated in A.S.R.E. Standard 16-56, "Methods of Rating and Testing Air Conditioners." The apparatus is sometimes referred to as a "code tester."

When nozzles are constructed in accordance with specifications, they may be used without calibration. If the throat diameter is five inches or larger, the coefficient may be assumed to be 0.99. For nozzles smaller than five inches in diameter, or where a more precise value of the coefficient is desired, the nozzle should be calibrated.[35] Formulas for calculation of air flow through nozzles are given in Section 6.3 of Standard 16-56.

AIR FLOW MEASUREMENT IN THE CARRIER LABORATORY

The June, 1947 issue of Refrigerating Engineering magazine contained the article, "Air Flow Measurement in the Laboratory," by D. D. Wile of Carrier Corporation, Syracuse, New York.[69] The aspects of the article which have current significance in air flow and air temperature measurement in the laboratory will be paraphrased.

Pitot tubes, anemometers, and other velocity measuring instruments are no longer acceptable for work of a high order of accuracy. After careful consideration of the merits of nozzles and orifices, nozzles were selected as being the more reliable under various conditions of operation. The principal objection to the use of the orifice is its sensitivity to conditions in the approach stream, which cause excessive variation of the flow coefficient. A nozzle coefficient, in the range of use encountered in air conditioning practice, should not fall below 0.984 and it never becomes greater than 0.995, a range of only 1 per cent. The recommended dimensions of code testers are the same as those adopted by the A.S.R.E.

The principal application for the code tester is coils. The tester is installed in a room which acts as the return air circuit for a large capacity air conditioner for automatically controlling test conditions. Regardless of the heating or cooling effect of the coil being tested, the conditioned air supplied to the room and at the face of the coil is maintained at a constant, predetermined condition.

The fan drawing air through the code tester must be selected for ample static head to overcome the resistance of the code tester interior in addition to the resistance of coils or other equipment that may be tested. If heating equipment is involved, the fan selection must allow for the higher temperature. Carrier practice has been to use uncommonly large heads across the nozzles and the air mixers with the result that accuracy, convenience, and compactness have been gained at the expense of fan power. The maximum static head has ranged from a 4 in. water gage head for a 300 cfm tester to a 7 in. head for a 4,000 cfm and a 16,000 cfm tester. In the latter case the reduction of air density, if not considered, would cause an error in flow calculation of nearly 1 per cent. Regulation of fan capacity can be accomplished by variable speed or by adjustable inlet vanes. Ordinary dampers have a tendency to produce unstable flow conditions.

The use of perforated plates as diffusing baffles, both upstream and downstream from the nozzles, make it possible to confine the nozzles in a comparatively short duct length. Excellent results were reported with plates having 1/4 in. holes staggered on 3/8 in. centers, giving a free area of 40 per cent of the face area.

Air discharged from coils or other air conditioning equipment may have nonuniform temperature distribution along with nonuniform velocity distribution, and one of the most difficult laboratory problems is to determine the true mean temperature. The procedure arrived at is to employ air mixers to make the velocity pattern uniform, and to average the temperatures from many points across the duct. Sampling tubes permit the drawing of air from numerous points across the duct area and conveying it to a convenient location where the average temperature may be determined by a single pair of wet-and dry-bulb thermometers.

The air mixers consist of a series of vanes arranged first to divide the air flow into a number of small streams and then divert these streams across each other. The vanes extend only about halfway across the duct with the remaining area blocked off to allow space into which the deflected air streams can flow. Two mixer assemblies are always used, one to mix vertically and one to mix horizontally. The vanes are positioned at 45 degrees, and their size depends on the size of the duct. A 4,000 cfm tester used 4 in. x 1-1/4 in. vanes spaced 1-1/2 in. apart.

The venturi throat should be as small as possible without developing excessive head loss, or about 1.5 times the total nozzle area in excess of the area occupied by the sampling tubes.

The nozzle design is of the Bureau of Standards elliptical type. Derivations and correction factors by which the exact amount of flow may be determined from the experimental data taken, are given in the body of the article.

AIR FLOW MEASUREMENT IN THE WESTINGHOUSE LABORATORY

Recent research in Westinghouse coil design was reported by A. L. Lee, Jr. in a Master's thesis.(69) The coil testing apparatus is located in a room which is maintained at constant, predetermined wet- and dry-bulb conditions by a large capacity air conditioning unit outside the room itself. The air supplied to the coil being tested is, therefore, constant. Calibrated mercury-in-glass thermometers are used to measure all air temperatures, and copper-constantan thermocouples are used to measure temperatures in the refrigeration circuit. A 24 in. round rubber flexible duct connects the tested coil to the temperature measuring section of the code tester downstream. A static pressure tap on the duct downstream from the coil measures the air friction drop across the coil being tested. Both vertical and horizontal mixing vanes are located in the entrance to the temperature measuring section. The dimensions and construction of the air flow and temperature measuring sections are in accordance with the specifications developed by D. D. Wile of Carrier Corporation.

After leaving the mixing vanes, the air is forced into a venturi section which contains a number of small sampling tubes at the throat of the unit. The air leaves the mixing section at a velocity of approximately 1,000 fpm and is accelerated to a velocity of approximately 3,500 fpm at the throat of the venturi section. The high air velocity minimizes the velocity variation of the air stream and allows the use of small sampling tubes. A small centrifugal fan is used to draw the air through the sampling tubes and over the wet- and dry-bulb thermometers located in the discharge air temperature sampling section, in which the velocity is about 1,300 fpm. The entire apparatus is insulated with two one-half inch layers of glass fiber covered on both sides with aluminum foil.

In the air measuring section a perforated baffle was placed on each side of the nozzle section. The baffles contain 1/4 in. holes on 3/8 in. staggered centers. The nozzle section contains four nozzles, each with a cross sectional area of 0.165 sq. ft. The 1/64 in. static pressure taps on each side of the nozzles were mounted in the bottom center section of the air measuring duct in such a way that the openings are flush with the internal wall surface.

The fan is driven by a direct current motor, with a rheostat for variable speed control. For more precision of control over air flow rate, there is a hand-adjustable duct damper at the fan inlet which bypasses the code tester. For the same rheostat setting at the fan motor, opening the damper will diminish the air quantity through the code tester slightly.

AIR TEMPERATURE MEASUREMENT WITH THERMOCOUPLES

Where the air velocity is uniform across a given section of duct, a grid of wet-bulb and dry-bulb thermocouples gives the temperature accurately enough for most experimental work. The thermocouples should be mounted on a simple wire grid to get complete coverage of the area, and should be connected in parallel to give an average of all the temperatures sensed. In this way, only one reading of the potentiometer is necessary.

With the exception of mercury-in-glass thermometers, thermocouples are the most universally used means of measuring temperatures. This acceptance is the result of numerous features, such as freedom from breakage, ease of application, reliability, remote reading, rapid response, ease of calibration, low cost, and small size.[70]

I. CONTROL OF AIR MOVEMENT

FANS

The centrifugal type fan is best adapted to moving air against considerable frictional resistance.[29] Centrifugal fans produce pressure from two independent sources:[38] (1) from the centrifugal force created by rotating the enclosed air column, and (2) from the kinetic energy imparted to the air by virtue of its velocity leaving the impeller. This velocity in turn is a combination of the rotative velocity of the impeller and of the air speed relative to the impeller. A fan with forward-curved blades depends less on centrifugal force for its pressure, and more on velocity pressure conversion in the scroll, with the result that it may run at relatively low speed. Conversely, a backward-curved blade builds up more of its pressure by centrifugal force, a more efficient form of energy transfer, and less by velocity conversion; it must therefore run at a higher speed.

Whether a forward- or backward-curved blade should be used depends upon space conditions, quietness required, efficiency at the specified load conditions, and desired performance characteristics. The maximum mechanical efficiencies obtainable for the various types of fans are not widely different, although there are marked differences in the rate at which the horsepower and pressure vary with changes of volume rate. The forward-curved blade is more compact. For a given air delivery and static pressure it operates at a lower tip speed and is quieter. Because of this, the forward-curved blade fan is more widely used in ordinary air conditioning work than any other type.[67]

The forward-curved blade fan produces maximum efficiency near the point of maximum static pressure, and the power curve rises rapidly with an increase in the rate of delivery. The latter characteristic demands careful calculation of the static pressure loss of a system to avoid overloading the motor. Characteristics of the backward-curved blade fan are: a greater range of stable operation due to a pressure curve which rises continuously as volume is decreased; a non-overloading ("limit-load") horsepower characteristic; a higher operating speed for a given fan flow rate and wheel diameter; in many cases, a higher efficiency; and, in some cases, a higher noise level.[29]

Centrifugal fans usually have a minimum noise level near the point of maximum efficiency. A noisy fan is generally found to be running at excessive peripheral speed because of excessive resistance, or because it is undersized for the application. Tip speed and outlet velocity are used as an index to proper selection, but it should be noted that the tip speed varies with the type of blading used and that a tip speed considered excessive for a forward-curved blade may be well within the operating range of a backward-curved blade.

Performance Curves. Fan performance curves show the relationship between the quantity of air that a fan will deliver and the pressure, either static or total, against which it can discharge the various quantities. The curves also indicate the horsepower required for the corresponding quantity flow. Constant speed performance curves are presented for a specific fan operating at a stated speed and handling air of a stated density. Performance curves are obtained from a series of laboratory tests on a fan with dampering at the end of the test duct. The pressure-volume curves for both backward- and forward-curved blades have a peak point at which the pressure is a maximum. To the right of this peak pressure point, both types of fans have a steadily falling, steep pressure-volume performance curve. This steeply falling performance curve is desirable in fans. If a fan with a steep performance curve is selected, a slight change in pressure from the one at which the fan is operating will not greatly affect the air delivery.

The forward-curved blade has a peak static pressure which corresponds to the region of maximum efficiency, whereas with the backward-curved blade, this maximum pressure occurs somewhat to the left of the region of maximum efficiency.

The horsepower for both types of fans is a minimum at no delivery. The horsepower for the forward blade increases continuously with increasing volume flow, with maximum horsepower occurring at free delivery conditions. The horsepower for the backward blade increases with increasing air flow only up to a point to the right of maximum efficiency and then gradually decreases; this is often referred to as a non-overloading horsepower characteristic because the maximum power which can be absorbed is usually not more than ten per cent above the power required at a normal selection point.[67]

Inlet Vanes. Fixed or movable inlet vanes that serve to control the direction of entering air are standard equipment on many types of centrifugal fans. Inlet vanes improve performance and avoid excessive noise where the fan entrance conditions are unfavorable because of congested or poorly designed duct connections. Inlet vanes create air spin which affects fan performance as well as dampering the air flow. This air spin is known as vortex control.[35] The vanes are so constructed as to give the air a spin in the direction of wheel rotation. The characteristic curves are changed and less horsepower is required for a given air flow than by straight dampering. The vanes are usually grouped together in a radial manner about a central ring and are pivoted and linked together so that each vane turns on its own axis to form a variable opening shutter.

Variable Speed Control. Besides air inlet vanes and dampers, there are several methods for varying the volume of air handled by a fan.[38] Where the change is made infrequently, the pulley or sheave on the driving motor or fan may be adjusted or changed to vary the speed. Variable speed pulleys or transmissions, such as fan belt change boxes, or electric or hydraulic couplings, may be used to vary the fan speed. Variable speed motors, either alternating or direct current, may be used.

Alternating current is preferred for general application, since a-c motors are less expensive and require less maintenance. A wound-rotor induction motor is generally used; the torque and the starting current can be regulated by varying the external resistance, which is in series with the rotor windings. Speed reduction to about 50 per cent of full speed is possible, but the efficiency is reduced at all speeds below normal by the amount of resistance in the circuit. At full speed the efficiency is slightly lower than for the usual squirrel cage motor.[29]

In direct-current motors, the series-wound motor gives a flat power characteristic, whereas for a centrifugal fan a rising power characteristic is necessary. Shunt-wound motors are the most suitable for fan drive, being normally constant speed and having a starting torque of about 150 per cent of full-load torque. They are made variable speed by inserting external resistance in series with the shunt field or the armature circuit. Compound-wound motors have high-starting-torque characteristics that are not necessary for fan drive.

In conclusion, from a power consumption consideration, a reduction in fan speed is the most efficient way to obtain variable-speed control. Next in order are inlet vanes, while dampers are the least economical.(29)

DAMPERS

The simple louver damper and the opposed-leaf damper are the commonest forms of the multiple-blade damper. The opposed is superior to the louver in minimizing turbulence, since the air leaves in about the same direction for any position. It has better performance characteristics, since the flow is well restricted at small opening angles, whereas a small opening of the louver damper lets through a large percentage of air and the last 45 degrees of travel has relatively little effect on air quantity. The linkage is somewhat more complicated for oppositely rotating blades, but is well justified where good control of air volume or of supply-air temperature is desired.

The design of multiple-leaf dampers should embody the following features:(29)

(1) Mechanical strength and rigidity.

(2) Static and dynamic balance of the assembly as a

whole in all angular positions.

(3) Approximate proportion between linkage movement and air flow, obtained by arranging the relative angular motions of the damper operating arm and the damper motor arm to give larger increments of leaf motion near the closed position than at the open position.

No advantage is gained by streamlining the blades.[35]

DUCTS

To insure desired air flow without excessive frictional and dynamic losses, design standards are essential to govern the fabrication of shapes, fittings, vanes, and connections to equipment. Nearly all of these standards are based on the following fundamentals of air flow.[29]

(1) Air flowing from a chamber or conveyor of smaller section area into one of larger area tends to continue in a straight line. Air will not diverge, unless changed by vanes, at an included angle greater than about 20 degrees. Where obtainable, the best practice dictates a divergence not greater than one inch in seven.

(2) Air flowing from a chamber or conveyor of larger section area into one of smaller area tends to converge uniformly and follows the laws of entrance to orifices in fluid flow.

Experiments reported by the Trane Company[67] have shown that the minimum friction loss is obtained when the divergent angle is about three degrees. Though such small angles can rarely be used, an effort should be made to keep the angle less than ten degrees.

Leakage and Vibration. Actual tests on typical supply systems have shown leakages from 5 to 30 per cent. The largest usual source is at transverse seams located against the wall or ceiling in such manner that tight joints are almost impossible. For supply systems with static pressures in excess of one inch water gage, calking, felting, or soldering is recommended.(29)

Systems requiring quietness in operation should be provided with asbestos cloth or other fireproof types of flexible connection to ducts or casings at the fan inlet and discharge. To prevent transmission of vibration to the building structure, a fan and motor base or foundation employing vibration-isolating material should be installed.(29)

J. SECONDARY CIRCUIT COMPONENTS

CONDENSING UNIT

The "condensing unit" of an air conditioning system consists in general of the compressor, compressor drive, condenser, and such auxiliary equipment as a refrigerant receiver and controls. These components are generally considered as a single unit for the purpose of rating capacity.

Compressor. Present day practice is to use high-speed reciprocating machines with one of the fluorinated hydrocarbons, such as Refrigerants 12 or 22. Cylinders may number from 1 to 16 and may be arranged in line, in single or multiple V or W, or radially. All are single-acting. Both suction and discharge valves are pressure-actuated rather than mechanically operated, and are located in the cylinder head. Ring plate or reed valves are used. Open compressors require a shaft seal to prevent leakage. Sleeve bearings and piston rings follow automotive design practice. An oil return check valve permits oil carried by the suction gas to be returned to the crankcase.[29]

Most modern motor-driven reciprocating compressors have been adapted to hermetic design to save in first cost, noise, and maintenance. This eliminates the shaft seal, shortens the shaft, simplifies bearing arragnement, and permits vapor cooling of the motor. Capacities extend above 100 horsepower.

Two types of compressor assemblies are available: the semi-hermetic, or bolted, serviceable type; and the welded hermetic type.

The compressor is protected against return of slugs of liquid by spring loaded cylinder heads, which lift under pressures greater than occur in normal operation. The cylinder walls and head are cooled by the surrounding air. In hermetic compressors, the motor windings as well as the lower part of the cylinder are cooled by the cold refrigerant vapor. For this reason, motors can safely be rated for higher output in hermetic equipment than in open drives.

The displacement of the compressor cylinders and the speed of rotation are based upon the volume of refrigerant to be removed from the evaporator. The cooling capacity of an evaporator depends upon the weight of refrigerant circulated, and the compressor must handle that weight at the specific volume which corresponds to the saturation pressure at the suction of the compressor. When plotted against increasing saturated suction temperatures, the refrigerating capacity of a compressor will consistently increase, showing that a compressor will have a much higher tonnage rating for air conditioning applications than for the lower temperatures of refrigeration service. At higher condensing temperatures, the capacity curves for a given compressor are parallel but correspondingly lower in tonnage rating. The lower capacity is due to the drop in volumetric efficiency caused by the re-expansion of the compressed vapor on the suction stroke.

System Capacity Control. There are a number of methods of accomplishing economical reduction of capacity when the load inevitably varies from design conditions:[29]

(1) Multispeed compressor motors. When a single reciprocating compressor is to be installed, a multispeed driving motor may be used. Common practice is to use a two-speed motor, which at low speed reduces the capacity either one third or one half. The lowest speed must not be below the range of proper operation of the compressor lubricating system. The speed may be controlled through electrical relays by a pressurestat in the suction line or by a room thermostat.

(2) Variable speed compressor motors. Variable-speed alternating-current induction motors offer no advantage for driving reciprocating compressors.

(3) Multiple units. Multiple compressors bring the following advantages: single-speed motors may be selected at best efficiency and operated continuously at this efficiency; stand-by equipment is available; and the compressors may be started in sequence to limit the current inrush.

(4) Clearance pockets. Used primarily on older equipment, the cylinder clearance is increased sufficiently to limit the amount of discharged vapor, and the work of compression is recovered by re-expansion.

(5) Cylinder cutouts. One or more cylinders may be made ineffective by by-passing the vapor from the discharge to the intake of the compressor. A solenoid valve is usually installed in the bypass connection to allow automatic control by either thermostat or pressurestat. The power does not decrease in proportion to the capacity because of the hydraulic loss incurred by flow through valves, cylinders, and connections.

(6) Suction-valve-lift unloaders. This is inherently the most efficient means of unloading thus far developed, since passage of the refrigerant vapor in and out of the cylinder through the suction valve, without compression, involves less loss than any other method. It also permits the unloading of individual cylinders with no serious space penalty and at moderate cost.

Capacity of the evaporator may be varied if it is not practicable to vary the capacity of the compressor. The compressor may be loaded artificially by transferring heat to the suction vapor. This heat may come from an external source or from the hot discharge vapor leaving the compressor. The system capacity is reduced because the weight of refrigerant entering the compressor and circulated through the system is reduced. The two devices which have attained commercial importance both use hot gas.[29] One type, balance loaders, transfer the heat indirectly through metal walls, boiling off the refrigerant and forming more vapor on the suction side for the compressor to handle. They are usually controlled by a constant-pressure expansion valve, opening on falling pressure to maintain a minimum evaporator pressure. The other type employs various arrangements of hot-gas bypass, which transfer the discharge gas directly into the low side. They are usually controlled by a constant-pressure vapor valve, which acts to admit vapor to the low side as the evaporator pressure tends to drop, thereby keeping the pressure in the low side substantially constant.

Evaporator unloading devices do not save power since no load is removed from the compressor. Their most useful purposes are to prevent the evaporator surface from frosting up and to avoid operation of the compressor at an excessively low back pressure.

Condensers. Shell-and-tube condensers comprise the large majority of water-cooled condensers now manufactured. The cooling water almost always flows inside the tubes, with refrigerant condensing on the outside. Tubes are circuited to give the desired design water velocity and pressure drop. Finned surface and high gas velocities are advantageous to increase transfer with the fluorinated refrigerants, which have low film coefficients; copper tubes are used. Except for small units, shell-and-tube condensers are built with removable heads to facilitate cleaning and scale removal on the water side. Current practice is to omit tubes near the bottom of the shell to provide liquid storage volume and avoid the need for a separate receiver. Adequate distribution space is required at the hot-gas inlet connection to permit the gas to be distributed laterally to all parts of the surface without excessive pressure drop. A purge connection is necessary since noncondensables in the system collect at the condenser.

Shell-and-coil condensers employ spiral tube bundles, and must be chemically cleaned. They are used in small systems because of low cost. Cleaning is accomplished by circulating a chemical solution through the water side.

Shell-and tube condensers are constructed with fixed tube sheets. Easy tube replacement is provided by using tube ends which are slightly larger than the outside diameter of the fins. Tube holes in the tube sheets are serrated. Shells are heavy wall, seamless steel pipe with no longitudinal joint. The tube is composed of one piece of metal; the fins are an integral part of the tube wall body. The spacing and height of the fins can give a total tube outside surface three hundred per cent greater than obtainable with bare tubes, which helps to compensate for the great difference in film coefficients between the water on the inside and the condensing refrigerant on the outside of the tube. The heat transfer value of the tube in increased in nearly direct proportion to the increase in outside surface area.[2]

REFRIGERANTS

No refrigerant in use at the present time has all the qualifications necessary for universal application. For each specific application the engineer must decide which would be the best refrigerant. Some of the factors involved in judging suitability are:[56]

(1) thermodynamic, physical, and chemical properties

(2) reaction with moisture

(3) leak detection

(4) action on components of the system

(5) safety considerations and governing codes

(6) cost

(7) type of compressor

(8) dielectric strength of the vapor and liquid

The refrigerants which most nearly meet the requirements dictated by comfort air conditioning installations are in the family of fluorinated hydrocarbons. Two of these, Refrigerants 12 and 22, have particularly favorable properties for use in reciprocating compressors operating in the air conditioning range. In the review of literature conducted by the writer, and in his experience in the field of air conditioning, these refrigerants were the only ones found in current application for air conditioning reciprocating compressors.

In comparing the properties of R-12 and R-22, it will be noted that there are more similarities than differences.[49]

Most comparisons are made with the refrigerant evaporator temperature at 5°F and the condensing temperature at 86°F. Both refrigerants are clear, almost colorless liquids at temperatures below the boiling points. They are considered non-toxic, non-irritant, non-explosive, and for all practical purposes, non-inflammable. Under extraordinary conditions when R-12 is exposed to an open flame, decomposition occurs and traces of the lethal phosgene gas are formed.[51]

R-12 has several advantages over R-22. It has been in use longer and is cheaper. The lower operating pressure differential reduces strain on the compressor parts and increases compressor life. A greater quantity of flow is required for the same refrigerating load, which permits a larger expansion valve orifice, and for small units makes accurate control of flow less difficult. Water is less soluble in R-12 than in R-22. This characteristic makes freeze-up of the expansion valve more ominous; however, it is believed that water in solution would be extremely corrosive, a fault more serious than the possibility of freezing the expansion valve. Water is soluble in R-22; at the present time it has not been determined what the direct consequences are.

R-12 is miscible in the liquid state with oil in all proportions. R-22 in oil separates into two phases, according to temperature and pressure and base stock of the oil.[56] Separation of the oil, and return of the oil from the evaporator, is accomplished by evaporating the refrigerant from the mixture. When a trap in the return piping occurs, the velocity of vapor must be rapid enough to entrain and return the oil back to the compressor. Having some oil passing through the pistons and discharge valves is desirable for lubrication, and to seal the valve surfaces, yet oil serves no useful purpose in the rest of the system.

Dielectric strength of vapor is important only in hermetic units, in which the refrigerant comes into direct contact with the field coils of the motor driving the compressor. If the dielectric strength of refrigerant vapor at suction conditions is low, possibility of short circuits between electrical conductors is increased; although in most designs no bare conductors are exposed to the refrigerant, all being fully insulated. R-12 has good dielectric strength as a liquid; R-22 is extremely poor. As a further precaution, care should be taken in selecting a lubricating oil having high dielectric strength to obviate trouble from that source.

WATER CHILLER

Water chillers are used for those many applications where direct expansion coils would not be suitable, and for process cooling of such liquids as calcium or sodium chloride, alcohols, glycols, solvents, machine tool cooling emulsions, and petroleum refinery liquids. In commercial and industrial buildings, and an increasing number of residences, the indirect system is frequently used when a number of air cooling units are installed in many different locations of the building. In this way, the possibility of losing liquid refrigerant from an extensive piping system is eliminated.[67]

When water is to be cooled by a mechanical refrigerating system, a heat exchanger is used to transfer the heat from the water to the vaporizing refrigerant. The water to be chilled is circulated through the heat exchanger by means of a pump, and is then delivered to the air conditioning units where the temperature of the water rises as it absorbs heat from the air being cooled and dehumidified. After leaving the air conditioning unit, the water flows to the pump and is again forced through the heat exchanger.

The shell-and-tube type is the only one in common use today.[67] It may be either of the "dry" or "flooded" type. The dry type admits refrigerant to the inside of the tubes with a thermostatic expansion valve in sufficient quantity that the vapor leaving will always be slightly superheated. This type is comparatively new, having been first introduced in 1937.[66] The flooded type generally has water inside the tubes, and the shell is kept full of liquid refrigerant. As a general rule small capacity chillers are dry and large capacity chillers are flooded. Nevertheless, dry chillers may be obtained in sizes to 250 tons, and flooded chillers may be obtained as small as 25 tons.

The flooded type requires special devices which return lubricating oil from the chiller to the compressor; in addition, heat exchangers are required in case slugs of liquid are returned with the oil. These devices, though expensive, are only a small fraction of the total cost of a large unit, which is able to save much in size and space because of the greater capacity inherent in being flooded. The dry type is used in small systems because no special difficulty is occasioned in returning the oil to the crankcase, and a large charge of refrigerant is not required to operate the system.

In the flooded type the tubes usually have an extended surface integral with the fin construction. The water or other liquid to be chilled makes from four to eight passes, being re-directed by the end headers. The level is controlled by a float feed valve. Sometimes the refrigerant is confined to a well in the bottom from which a small pump delivers it to flooding nozzles at the top of the shell from which it sprays over the tubes. In the dry type chiller the refrigerant is fed into the inside of tubes from thermostatic expansion valves at the inlet header. The number of valves and circuits increases as the size of chiller increases, and each circuit generally makes several passes. Star-shaped metal inserts are sometimes placed in the tubes to increase the coefficient of transmission on the refrigerant side. Over-and-under baffles make the water follow a tortuous circuit, giving more cooling capacity for a given rate of flow.

There is always danger of freezing the water in a chiller, particularly on light loads. A thermostat bulb should be placed in the leaving water to stop the compressor if the temperature of the water falls to about 36°. An additional precaution is the use of a back-pressure control valve in the suction line; regardless of the load, the evaporating pressure and temperature in the chiller can be maintained above freezing.[67]

WATER HEATER

The water heater is an indirect way of adding heat to the air to be conditioned, and is usually a steam-to-water heat exchanger. It is popular because a year-around system needs only one set of piping, one pump, and one set of conditioning units when chilled water is used in the summer and hot water is used in the winter. It is also a common adjunct to systems employing a washer for year-around service.

The construction is very simple. Commonly there is a header on only one end of the steel cylindrical shell, in which the water connections are made. The water tubes are closed loops to avoid the necessity of a header on the other end of the shell. The water may make from two to four passes. Generally the tubes have an extended surface on the outside to improve the coefficient of heat transfer of the condensing steam. Header and water boxes should be of a type to permit periodic cleaning of tubes.[29] The steam inlet connection is made on top near the leaving water connection. The condensate drain connection is at the opposite end from the steam inlet, and should have a float and thermostatic trap in the line.[66]

Using hot water requires an expansion tank because of the significant change in volume as water is heated. An open expansion tank must be located at a level high enough to prevent boiling of the water at the maximum temperature at which the system will operate. A closed expansion tank, known as a compression tank, may be located anywhere in the system.[29] It is important to remember that the compression tank is the "point of no pressure change" in the pump circuit; that is, it is the only point in the system at which the pressure will not change appreciably whether the pump is operating or not. Expansion of the heated water is taken up by compression of air in the tank. The usual practice is to size the tank for about 200 per cent of the maximum expansion that will occur between the lowest and highest temperatures to which the system will be subjected. If the temperature range is from 40° to 180°, the tank should contain about 6 per cent of the water-holding capacity of the system.[29]

A pressure relief valve should be located at the heater; no valves should separate the relief valve from the tank itself.

Occasionally steam is injected directly into the water tank of the washer, or water from conditioning coils is circulated to a tank where the water is heated by direct injection of live steam.[49] In each case the steam injected represents a loss to the boiler and must be replaced by an equivalent amount of distilled make-up water. When used to heat washer water, boiler compound will carry over with the s steam and lend its characteristic odor to the conditioned air.

CIRCULATING PUMP

For usual pumping requirements in heating and air conditioning work, the motor-driven centrifugal pump is particularly well-suited because of its favorable characteristics, such as instantaneous adjustment to varying loads and steady discharge, which reciprocating pumps cannot give.[29] Unlike the piston pump, which will deliver the same quantity of water regardless of the external head, the quantity of water that a centrifugal pump delivers varies with the external head. For this reason, it is necessary to know the head against which a centrifugal pump must work before the quantity of water that it will deliver can be predicted. Free passages exist at all times through the impeller between the discharge and the inlet side of the pump. The pressure differences in the pump must be produced by mass forces of the fluid and, consequently, are a function of the speed of rotation and of the rate of flow through the pump. For a constant speed of rotation, characteristic curves must be determined experimentally. Then it is possible to calculate the characteristic curves for any other speeds by means of the pump laws.

When plotted against increasing capacity in gallons per minute, the head is greatest at no-delivery conditions, from which it decreases; the efficiency rises to a peak at nominal capacity and falls off; and the power required rises from a finite value at no-delivery conditions and does not peak.(52) If the viscosity of the fluid is neglected, the pressure increase produced by the pump will vary with the square of the fluid velocities, and the flow conditions will be similar if all average fluid velocities and all peripheral velocities of the rotating parts are changed according to one constant ratio. This condition is satisfied if the capacity is changed proportionately with the speed of rotation. The power input is proportional to the capacity times the head produced, and varies as the cube of the speed.

The pump laws may be stated as follows:[67]

(1) The quantity of water delivered varies directly with the pump speed.

(2) The head developed by the pump varies as the square of the speed.

(3) The horsepower required by a pump varies as the cube of the speed.

For general service, a steep characteristic curve of head drop with increasing flow rate is preferable. It is obtained by using impellers with backward curved blades. With a steep characteristic, small variations in head will not appreciably affect the discharge of the pump.

To predict flow rate and head developed in a given piping system with a pump operating at constant speed, the system resistance and pump head may be plotted against rate of flow. The intersection of the two lines is the operating condition. If the system resistance is increased by partially closing a valve, the system resistance will increase for all rates of flow, and the resulting point of equilibrium, or intersection of the two curves, will indicate a lower flow rate at a slightly higher head.[67]

IV. DESIGN OF THE PROPOSED UNIT

The proposed unit was designed to demonstrate those cycles of operation which occur in residential, commercial, and industrial practice, and which represent accepted, good engineering practice. A central fan type year-around built-up unit was designed. The unit is versatile and has almost unlimited potentialities for demonstration and research. For example, a high pressure induction unit could be set up in the test room and supplied with hot or chilled water and high pressure conditioned primary air from the components of the proposed unit.

The basic cycles which the research reported in the Review of Literature showed to be of prime importance will be described and illustrated in the following section. The individual components were selected with sufficient capacity to assure balanced operation in all cycles. Special devices of design were planned to permit observation and study of performance throughout the entire capacity range as well as at design conditions of load. It was possible to design the system so that all the proposed cycles of operation may be set up and controlled from the central graphic panel. A control drawing is included in this thesis by which the overall control system may be installed and operated, and in which the schematic inter-relation of all parts is clearly shown.

The unit will be arranged and placed so that it can be observed by a fairly large group. Plastic windows and internal marine-type lights will allow visual observation of the various conditioning processes.

An outline of the features of the proposed built-up unit follows. A duct carrying air away from the test room will pass through the wall at the location of an existing inspection door and make junction with a duct bringing outside air from a roof intake. The outside air duct will contain a tempering coil. The mixture of outside and return air will pass in succession through a disposable type filter, electronic filter, activated carbon filter, steam preheater coil, direct expansion cooling coil, chilled and hot water coil, face and bypass dampers, washer-dehumidifier, steam reheater coil, air flow and air temperature measuring chamber, and centrifugal fan with inlet vanes. Ductwork will carry the conditioned air through the opening at the inspection door back into the test room. Completely automatic pneumatic controls will maintain the predetermined temperature and humidity of the test room within close limits. Instrumentation will be provided to measure operating temperature change, air friction, and air flow across each component in the air duct.

A. BASIC DESIGN CRITERIA

The following paragraphs describe the design criteria on which were based the selection of components and design of the proposed unit.

Design Temperature and Humidity. Outside design weather conditions for the winter cycles were taken as 0° temperature and five grains of moisture per pound of dry air. Since no official figures were available for Blacksburg, the official figures for Roanoke, Virginia[30] were modified to take account of the slightly higher altitude and colder weather in Blacksburg. Summer design conditions for the summer cycles were taken as 95° dry-bulb and 78° wet-bulb, the same as Roanoke. The reason for this is that the roof location of outside air intake will compensate for the cooler Blacksburg climate.

Both summer and winter inside design conditions were taken as 75°dry-bulb and 50 per cent relative humidity. Though it was realized that the inside design conditions might preferably be 76° and 50 per cent for the summer cycles and 75° and 40 per cent for the winter cycles according to present standards of Effective Temperature,[29] the proposed built-up unit is for experimental use, and if the equipment is designed to hold 75° and 50 per cent on both summer and winter cycles, continuous tampering with the controllers will be eliminated. This condition is 70° Effective Temperature,[9] the optimum year-around average for men and women in the middle part of the United States.[29]

<u>Control System</u>. The majority of control systems now being installed in medium and large size commercial and industrial buildings are pneumatic. Besides the advantages of pneumatic controls outlined earlier which assure a long life of trouble-free operation for the proposed unit, pneumatic controls were judged to be of greatest potential value from the educational standpoint.

<u>Fan</u>. The type of fan recommended is a backward-curved blade centrifugal type, with a direct-current motor drive to facilitate changing speeds in small increments through the total range. In conjunction with inlet vanes, this permits precise adjustment of air flow rate, a valuable asset in coil research. The fan is of the pull-through rather than blow-through type, a requirement dictated by the washer and air flow measuring chamber; also it is the arrangement most prevalent in well-designed systems.

<u>Air Purification</u>. The recommendation of complete air purification equipment in addition to the washer was based on the significance of air recovery in present and future industrial research. Included will be a disposable type viscous filter, permanent dry filter, electronic filter, activated

carbon filter, and air washer and scrubber.

Air Flow and Air Temperature Measurement. The apparatus for measuring air temperature and quantity through the unit is designed on principles prescribed by industry Standards. The accuracy of all data taken from the built-up unit will be dependent on the accuracy of this measuring chamber.

Cooling Coils. The chilled water and direct expansion cooling coils were designed for a nominal capacity of five tons, or 60,000 Btu/hr when the entering air is at 75° dry-bulb and 50 per cent relative humidity. This was found to require more rows of coil surface than required when the air entering the coil is a mixture of outside as well as return air, which is the present practice. However, one primary purpose of the laboratory is to study air recovery methods, or means of purifying return air from conditioned spaces for re-use. In this case the air entering the coil will be at substantially the same conditions as those maintained in the conditioned spaces.

Heating Coils. Three coils are required in the system: tempering coil, preheater coil, and reheater coil. The tempering coil will be exposed to freezing temperatures and should be of the non-freeze type. All coils were sized for the extreme conditions of their respective operating cycles to give greater versatility.

Auxiliary Equipment. A five-ton capacity hermetically sealed motor-compressor unit, using alternating current was selected. This was judged to be of the utmost reliability, a primary concern with experimental equipment. The water-cooled condensing unit will supply refrigerant both to the direct expansion chiller and to the direct expansion cooling coil. The refrigerant condenser will obtain circulating water from an existing pump circuit adjacent to the installed unit. A circulating water pump was selected for the washer which in other cycles will also serve the hot and chilled water coil. A steam heater was placed in the water circuit in parallel with the chiller.

Dampers. Hand-adjustable opposed-blade dampers will be placed at the junction of the outside and return air ducts, and opposed-blade type dampers were recommended for face and bypass control at the direct expansion cooling coil and hot and chilled water coil.

B. AIR PURIFICATION EQUIPMENT

Based on the research reported in a previous section, a modern central fan type air conditioning unit should include a disposable type filter, an electronic filter, and an activated carbon filter. All of the other schools contacted had disposable type filters in their laboratory air conditioning units, and North Carolina State College reported an electronic filter in addition. The application of electronic and activated carbon filters is relatively new in the air conditioning field. In the past, the scientific aspects of atmospheric impurities were not well understood, nor was the demand for a solution to the problem of atmospheric pollution imperative. Recently air purification problems have come into prominence. There are clear prognostications that research and development of instruments and techniques, as well as extension of basic knowledge, would be highly rewarding. Installing the proposed facilities would place the Virginia Polytechnic Institute in an advantageous position. Research would be economical, yet of great value.

Arrangement. The disposable type filter should be placed upstream of the conditioning coils at the head of the apparatus. Placed upstream of the electronic filter, the disposable type filter will allow longer periods of operation between cleaning cycles. A permanent type, mechanical filter is incorporated on the outlet side of electronic filters to catch loose particles flaking off the collector plates in instances of excessive build-up between cleaning cycles. The air leaving the electronic unit is of maximum purity, assuring maximum life for the activated carbon filter downstream. Should the activated carbon filter have the task of removing particulate matter from the air stream, its capacity for adsorbing gases and vapors would be diminished proportionally. Also, it is comparatively expensive to renew or reactivate the carbon.

The following sections describe the specifications which should be met by filters installed in the proposed V.P.I. unit.

DISPOSABLE TYPE FILTER

The selection of a disposable type filter should provide not only an air cleaner serving as a standard operating component of the built-up air conditioning unit, but in addition the frame holding the filter should be of a standard size so that many different brands and media may be installed and tested.

Selection. The filter recommended is an American Air Company "Amer-glas" replaceable air filter.(10) The set of filters are mounted in permanent frames. Each frame is 25 in. high x 20 in. wide x 2-13/16 in. thick. This is a standard size used throughout the industry for replaceable type air filters. Since each filter mounted in the 25 x 20 frame has a rated capacity of 1000 cfm, two in parallel will be required to clean the nominal 2000 cfm of the air conditioning system. Each frame will hold two 1 in. cells, one 2 in. cell, or single cells of 1 in. thickness. Each time the filter cell collects its dust load, it is discarded and replaced with a new one. When cells are used in multiple in series, only the front cell is discarded; the rear cell is moved forward and the clean replacement filter put in the rear. Retaining latches for securing the filter are furnished with each frame.

The initial resistance of a 2 in. cell is 0.10 in. water gage. The unit frames can be riveted to the supports with the number 7 tinner's rivets furnished. The inside perimeter of the frames is covered with a felt seal. The media of the filter is placed in a fiberboard casing between perforated metal grilles. The filtering media is formed of continuous, slightly curled, interlaced glass filaments held in place with a thermoplastic bond. The average diameter of the filaments is from 25 to 30 microns. A 2 in. thick pad contains approximately 37,000 linear feet of fiber per square foot. Each fiber is coated with a proprietary water-soluble viscous oil called "Viscosine." The oil forms an adhesive film to hold collected dust, and obtains its efficiency by the impingement principle. The oil is non-flammable and remains a fluid-jell which will not drip at temperatures below 150°. It is non-volatile, odorless and, since it will not support bacteria, sanitary. The density of the media is graduated from front to back to increase the effective life, and should be installed with the course side facing upstream.

In order to fit in a fairly narrow duct, the filter frames are mounted on clip-type supports furnished by the manufacturer at an angle of 67-1/2 degrees slanting down the duct from the position at which they would span the cross-section. A sufficiently large access door must be provided so that the filters may be serviced from the front. Also, an inclined tube manomenter should be installed either directly at the duct, or remotely at the central panel to indicate static pressure drop as the air stream flows across the filter bank. Research will be necessary to set up a direct correlation between increased loading of the filter cells and the optimum time to change. In the past, it was recommended that filters be changed when the air flow rate was lowered enough to cause noticeable loss of capacity of the conditioning coil.

ELECTRONIC FILTER

Selection. The recommended electronic filter is the Minneapolis-Honeywell F34-D-102 Standard Package Electronic Air Cleaner, which at 2000 cfm has an efficiency of 93 per cent as determined by the National Bureau of Standards blackness test, using atmospheric dust. The manufacturer should provide a factory trained representative to check out the installation and provide initial start-up service. The unit consists of the following:[62]

(1) Power pack to convert 110-130 volt, 60 cycle power supply to 6,500 volts direct current for collecting plates and 13,000 volts direct current for the ionizing section, housed in a metal case with cover secured by time delay safety switch, magnetic circuit breaker, milliammeter, indicating light, and current-adjustment control.

(2) Galvanized steel cabinet with built-in pan and access panel.

(3) Air cleaner cell complete with aluminum-mesh dry-type after filter.

(4) Hand-cranked traveling washer.

(5) Support cradle for suspension on framework.

(6) Duct-access-door safety switch.

(7) Hand-operated sprayer for applying dirt emulsifier.

(8) One gallon of dirt-emulsifier.

The overall height of the unit, having the suspension cradle rather than legs, is 29-3/8". Overall width with power pack is 35-15/16". Depth is 30-1/16". Duct connection sizes entering and leaving the unit are: width, 21-11/16"; height, 20".

A dry-type mechanical filter about 1/2" thick is included in the downstream section of the electronic filter casing to trap any particles leaving the collector plates in case of excessive build-up. A separate mechanical filter should be installed upstream of the precipitator to remove the bulk of impurities and prolong the service life of the precipitator between cleanings.

Construction Details.[53] Included with each electronic filter are a through-duct connector and water hose with fittings to supply water to the traveling washer, which is a water spray manifold which travels across the face of the cleaner on rails, driven by a hand-cranked chain. The unit is housed in a galvanized steel cabinet with built-in drain pan. Removing the power pack reveals an access panel. A duct access door, not furnished with the precipitator, with an opening not smaller than 18 in. x 18 in., should be installed on the upstream side of the unit, as close to the precipitator as possible. The access door provides a means for manually spraying the dirt-emulsifier solution on the collector plates. Furnished with the unit is a safety switch to be attached at the access door, which ensures that all power is cut off if the door is inadvertently opened before taking the unit out of service.

Immediately downstream of the washer assembly is the air cleaning cell. The ionizer section consists of vertical ionizing wires 0.012 in. in diameter alternating with capsule-type ground electrodes having a thin cross-section to eliminate turbulence in the flow of air through the cell. Immediately downstream from the ionizing section are the collector-plates spaced 5/12 in. apart. Alternate plates are charged with positive potential, and every other plate is grounded. Since practically all particulate matter leaves the ionizing section with a positive charge, the grounded negative plates will collect most of the impurities.

The duct on each side of the precipitator should slope down toward the drain pan, with a minimum slope of 1/2 in. in 2 feet of slope. Minimum distance from the precipitator to the heating coil upstream should be 2 ft. This slope returns any water that may get into the duct during washing. At some point upstream of the precipitator a screen with small mesh or a mechanical filter should be installed to prevent arcing due to insects and large particles getting into the air cleaner. There should be a perforated baffle plate ahead of the washer for two reasons: First, to insure even distribution of air flow across the face of the precipitator; and second, to positively insure that no one may come in contact with the ionizers by crawling along the duct. There should be no way of getting to the air cleaner without opening the electrically interlocked door. When suspension-mounted, the unit is supported from the ceiling by four 1/2 in. steel rods, one in each corner of the cradle. The cradle should be checked for true level so that the precipitator will drain properly during washing. The duct should be attached inside the collars on the cabinet and the joints calked to prevent water leakage during washing. Since the precipitator is on the suction side of the system fan, any seams between the precipitator and the fan must be made air-tight to prevent dirty air being drawn into the ventilating system.

The wash water connection is 7/8 in. O.D. copper tubing. The supply pipe should be large enough to carry 6-1/4 gallons per minute. The recommended temperature is 140°. Full city water pressure of 40 to 80 psig should be available for satisfactory operation of the washer. It is recommended that a strainer with a 324-hole-per-sq. in. screen be provided in the water supply pipe. The wash cycle time is 1/2 min. per ft. of air cleaner width. The built-in drain pan has a 2 in. pipe nipple connection. The drain system must be capable of handling 6-1/4 gallons per minute. Drain seals should conform to the best plumbing codes to prevent dirt, water, or air from entering the air-handling system. By means of hose clamps provided with the units, the hose is connected inside the duct to the traveling washer at one end and to the through-duct connector at the other end.

ACTIVATED CARBON FILTER

Selection. The recommended activated carbon filter is the Minneapolis-Honeywell CF1A1 Renew-Air Charcoal Filter, complete with frame.(62) Each filter shall have an efficiency of 95-97-1/2 per cent at 1,000 cfm with a 0.2 in. water gage pressure drop, and an activated coconut shell carbon content of 45 pounds. Two cells will be required since the nominal capacity of the built-up air handling system is 2,000 cfm. Each filter consists of a frame, adsorber cell, and locking clamps. For this original installation, each cell would be shipped with a 14 gage steel frame which is 24 in. high x 24 in. wide x 8-3/4 in. deep. The multiple frames are riveted together, with rivets supplied, inside the ductwork to form a filter bank. A felt strip is cemented to the inside of the turned flange to insure a tight seal against air leakage.

The adsorber cell is formed of 24 gage perforated steel with continuous vertical folds, welded together with spacer bars and heavy reinforcing angles to insure a uniform depth of the activated charcoal throughout the filter. The charcoal is retained in the cell by caps, top and bottom, sealed with sponge rubber sheets and fastened to the cell with sheet metal screws. The cell works equally well with air flow in either direction. An access panel at least 26 in. square must be provided in the ductwork on the opposite side of the frame from the flange, for installation and removal of the adsorber cell. That is, if the access panel is to be upstream from the cell, then the part of the frame which has the corner clamps should be forward of the part of the frame having the retaining flange.

To install, the upright cell should slide into the frame with the label facing out so that the felt on the cell seals against the flange on the frame. Four clamps, one for each corner, are provided with the unit to lock the cell in its frame. A suitable dust filter should be installed upstream of the filter to prevent plugging of the activated carbon pores. Care also should be taken that no oil vapors are emitted by filters upstream from the adsorber. A test panel is provided which, after a specific period of operation, is returned to the manufacturer, who will determine the residual adsorption capacity and estimate the expected service life to be obtained from the filter. Cells are reactivated by the manufacturer for a nominal charge.

C. STEAM HEATING COILS

General. Selection of the tempering, preheater, and reheater coils was based on the principles considered most sound as outlined in the Review of Literature. The recommended coils are of the extended surface type; the 5/8 in. O.D. seamless copper tubes have helically wound smooth aluminum fins mechanically bonded to the tubes. The optimum combination of heat transfer rate and air friction indicates a fin spacing of approximately fourteen per inch for this application. The recommended coils have uniform appearance and size, and in addition are similar to the hot and chilled water coil and the direct expansion coil, all being made by the same manufacturer. Coil casings in all cases are flanged on both faces and template punched with 9/31 in. diameter holes spaced on 3 in. centers 3/4 in. from the edge of the casing flange all around, for attachment of duct or flexible connections.

It is standard practice in the air conditioning industry to speak of velocity of air across a coil as the total cfm divided by the face area of the coil. This in no instance is directly indicative of the actual velocity of the air through the free area. It will be noticed that even if coils have approximately the same face area, the fin spacing, tube spacing, and number of rows may be quite different. With the exception of the tempering coil, the coils were selected so that the air velocity would not exceed 500 fpm at the coil face area when the unit is handling 2000 cfm.

The accompanying schematic diagram illustrates good practice in piping connections for the steam heating coils in the proposed laboratory. Slightly superheated steam at approximately 5 psig pressure can be supplied to the apparatus by throttling with a pressure reducing valve the 75 psig steam now available. The float and thermostatic traps should be capable of discharging 2-1/2 times the rated condensate flow. Though designed primarily for low pressure service, the coils can operate at steam pressures from 1 to 200 psig and temperatures up to 388°. In manufacture each coil should be hydrostatically tested at 1000 psig.

Three heating coils are required for the proposed built-up unit: tempering coil, preheater coil, and reheater coil. The tempering coil is used to raise the temperature of the entering air above the freezing point. The preheater coil is used to raise the temperature of the air from that leaving the tempering coil to such a temperature that in passing through the water sprays of the washer the air will approach adiabatic saturation, obtaining a moisture content corresponding to the required

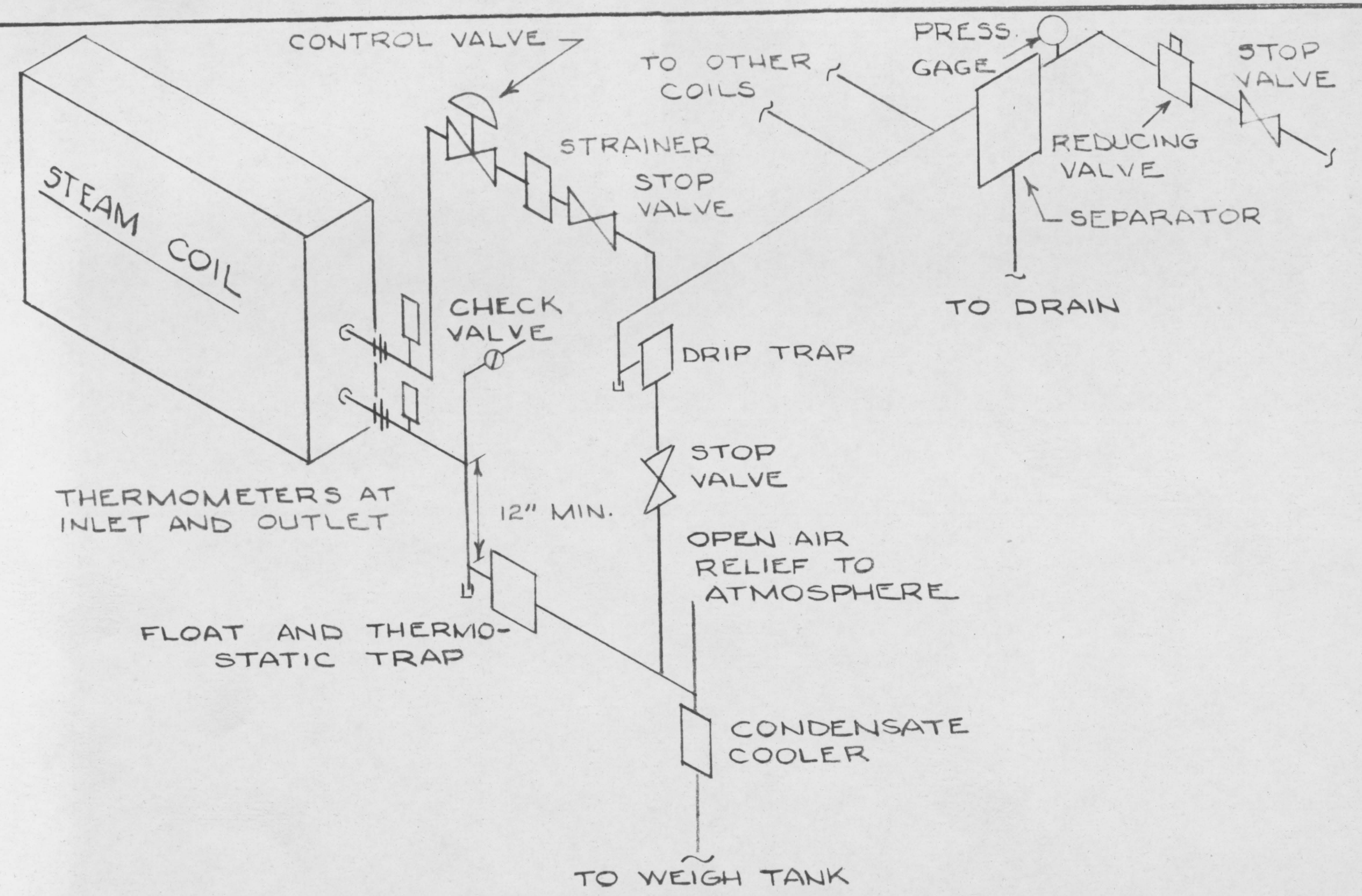


FIGURE 2. SCHEMATIC DIAGRAM OF TYPICAL STEAM COIL PIPING CONNECTIONS

dewpoint temperature. The preheater coil, therefore, supplies heat as necessary to control the dewpoint temperature. The reheater coil is used to raise the temperature of the air leaving the washer to that necessary to maintain the desired temperature in the conditioned space.[29]

It should be pointed out that the functions of the tempering coil and preheater coil could not be performed by a single coil. When handling freezing air, the steam coil should be the non-freeze, tube-within-a-tube type. The steam valve to the tempering coil should be wide open whenever the air entering the coil is below 35°. The full open valve is necessary to insure that the entire surface is heated with no cold spots; that no air may pass through the coil without being heated; and that the condensate will not freeze before leaving the coil tubes and entering the header which is outside the duct space. The tempering coil should have only enough capacity to heat air which enters at 0° to about 35°. Were it designed to have more capacity, the system would be out of control when the entering air rises to about 35° with the steam valve still wide open. As the temperature rises above 35° automatic control valves are designed to modulate rapidly the steam valve to the closed position. In addition to the tempering coil there must be a heating coil which can be regulated throughout the entire range of its capacity, and should have sufficient capacity to heat the air leaving the tempering coil to any temperature dictated by the heating cycles of the apparatus.

Graphical solutions using the Carrier normal and high temperature psychrometric charts facilitated the determination of capacity requirements for the preheater and reheater coils. The sensible heat ratio in typical conditioned spaces on the winter cycle was found always to exceed the value of 0.50,[36] which was therefore selected as a conservative value applicable to the proposed experimental work. Since the heat supplied to the apparatus must be removed at the test room by refrigeration equipment, the capacity of the heating and humidifying equipment was kept at a reasonable figure. It was arbitrarily decided that the air leaving the reheater should be held to about 100^{o}. Constructing the room load line from the datum of 75^{o} and 50 per cent relative humidity with a sensible heat ratio of 0.5, the required dewpoint temperature of the air supplied to the conditioned space was found to be 68.5^{o}. For a washer efficiency of 100 per cent, this would require that the air enter the washer at 134^{o}. Since the washer will be fabricated from design data developed in this thesis, the operating efficiency is indeterminate. However, commercial washers having two banks of nozzles with opposed spray characteristically have efficiencies ranging from 90 to 95 per cent.[29] Consequently, a design figure of 90 per cent will be used.

When the air outside is 0°, the air leaves the tempering coil and enters the preheater coil at 36°. The air would be heated to 134° before entering the washer. At 90 per cent efficiency, the air would be saturated adiabatically along the wet-bulb temperature line and leave the washer at a temperature of 75° (dry-bulb temperature).

$$\text{Washer Efficiency} = \frac{\text{(DBT entering washer) - (DBT leaving washer)}}{\text{(DBT entering washer) - (Temp. at saturation)}}$$

$$0.9 = \frac{134 - \text{DBT}}{134 - 68.5}$$

$$\text{DBT} = 75^\circ \text{ leaving washer.}$$

The processes referred to are illustrated on the attached sketch. In order to maintain the conditioned space at the specified conditions, and realizing that the reheater coil supplies sensible heat only, the conditioned air leaving the washer must be on process line CD.

The solution was obtained graphically. The adiabatic saturation process in the washer, at 90 per cent efficiency, is shown on the chart as B'C', which is 90 per cent of the total process length B'M. The air must leave the preheater coil, then, at 144° and will leave the washer at 78° dry-bulb temperature.

It is believed that selection of the preheater coil based on the above processes will provide a coil of sufficient capacity for all design conditions of operation.

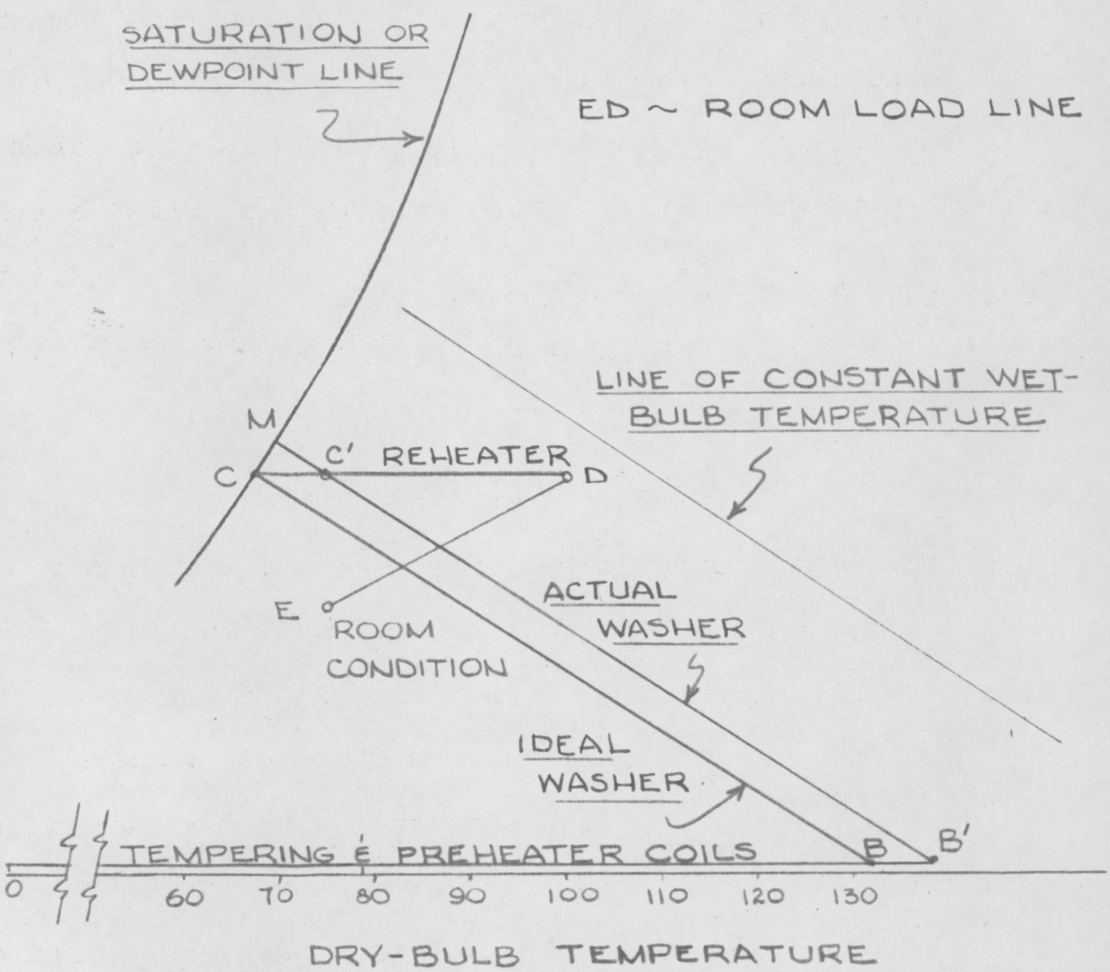


FIGURE 3. ILLUSTRATION OF GRAPHICAL SOLUTION OF HEATING COIL CAPACITY BY USING THE PSYCHROMETRIC CHART

Only if the washer efficiency were less than 90 per cent would the heating capacity be insufficient. There is little chance that the outside temperature ever would be as low as 0° while the equipment is being operated, giving a margin of capacity because of the correspondingly higher temperature of air entering the preheater. Should the sensible heat ratio increase, then the required temperature leaving the preheater would decrease.

The reheater coil will be sized to cover the maximum requirements of the processes, which would be for a 100 per cent washer efficiency. In that case the air would leave the washer and enter the reheater coil at 68.5°, and would leave the reheater at 100°. For washer efficiencies less than 100 per cent and for room sensible heat ratios greater than 0.5, the heating requirement of the reheater would be less than at the above conditions.

The required capacity of the two coils is based on handling the equivalent of 2000 cfm of air at standard conditions, or 9000 pounds per hour:

Preheater Coil Capacity, Btu/hr:

$$Q = W \times C_p \times \Delta t$$

Where: W = weight of air through coil, lb/hr
C_p = specific heat of air, Btu/(lb) ($^{\circ}$F)
Δt = temp. rise across coil, $^{\circ}$F

$$Q = 9000 \times 0.24 \times (144-36)$$

$$Q = 233{,}000 \text{ Btu/hr}$$

Reheater Coil Capacity, Btu/hr:

$$Q = W \times C_p \times \Delta t$$

$$Q = 9000 \times 0.24 \times (100 - 68.5)$$

$$Q = 68{,}000 \text{ Btu/hr}$$

Tempering Coil Capacity, Btu/hr:

$$Q = W \times C_p \times \Delta t$$

$$Q = 9000 \times 0.24 \times (36 - 0)$$

$$Q = 77{,}600 \text{ Btu/hr}$$

The heater for the washer water, to be used in lieu of the preheater coil for demonstration purposes, should have a capacity at least as great as that of the preheater coil in order to perform equivalent functions.

The capacity of the reheater coil must be checked for the summer cycle, for which its capacity is predicated on the:

(1) Lowest sensible heat ratio anticipated

(2) Lowest apparatus dewpoint temperature available

(3) Air quantity handled

The lowest typical sensible heat ratio encountered in commercial practice is 0.65.(29) By constructing the 0.65 load line slope on the psychrometric chart with the datum room conditions at 75° dry-bulb temperature and 50 per cent relative humidity, it can readily be seen that the temperature rise required for summer cycle reheat would always be far less than the winter design differential of 31.5°. In addition, the heating capacity of the coil characteristically increases as the entering air temperature drops, and the air across the coil in summer cycles is colder than the air across the coil in winter cycles.

The recommended method of installation of each piece of equipment will be according to the best current practice. This has educational value for the students as well as practical value in operation. The condensate will be cooled and collected for weighing at the outlet of each coil, before being drained to waste. When a test is not in progress on the coils, or a heat balance of the system is not required for the test in progress, the condensate will simply be cooled and drained directly to waste.

TEMPERING COIL

Coil Selection. The tempering coil selected should conform to the specifications of performance and capacity developed in the preceding section. It will be placed in the outside air duct of the proposed unit for the purpose of heating air which is below the freezing point to a temperature of not less than 36°, to insure that air entering the other parts of the built-up unit will always be above freezing. It will be capable of handling 2000 cfm of air. It should conform as closely as possible to the dimensions of the outside air duct to avoid extreme duct transitions. Since there will never be moisture on the fins and since the tubes and fins are so well spaced that air friction drop will be small, the allowable face velocity is not a critical factor.

The recommended tempering coil for the proposed unit is the Aerofin Type B Non-Freeze Coil. It is the tube-within-a-tube, non-stratifying type, with the tubes pitched within the casing to give more effective drainage and less possibility of water hammer. The steam supply and condensate return are at the same end of the coil, with the tubes slanting upward from the header; this counterflow arrangement for flow of steam and condensate permits throttling the steam, theoretically without danger of freezing the condensate even though the air entering the coil is below freezing. Nevertheless, experienced control engineers[63] say that it is possible to freeze the coil when the steam flow is throttled; therefore, the proposed unit will be so designed that whenever the entering air is 35^{o} or less the steam valve will be full open.

The coil was selected from information given in Aerofin Bulletin B-54.[7] The coil most suitable for the requirements of fitting the space available would have a nominal tube length of 24 in. with an overall casing width or height of 20-9/16 in. and a face area of 2.8 sq. ft. The table on page 6 of the bulletin gives the final temperature of air leaving the coil using saturated steam at 5 psig when the entering air temperature varies from -30° to 70° and the face velocity varies from 300 fpm to 800 fpm. By interpolation it was found that air entering the coil at 0° with a face velocity of 714 fpm would leave the coil at 35.8°. The coil is described as "NI," which denotes a single row of tubes with minimum tubes in the face. The condensate discharged would be 80.4 lb./hr. The approximate required capacity of the coil will be 77,6000 Btu/hr.

Interpolation from the tables on page 5 shows an air friction drop of 0.0586 inches of water gage for the air at an average temperature through the coil of 18°.

A summary of coil information follows:

Coil requirements: to heat 9000 lb/hr of air from 0° to 36°

Overall casing depth for one row: 5 in.
Overall casing width: 20-9/16 in.
Nominal tube length: 24 in.; overall casing length, 32-1/2 in.
Face area: 2.8 sq. ft.
Face velocity: 714 fpm
Tube face: 7
Supply connection: 1-1/2 in.
Return connection: 1 in.
Air friction drop: 0.0586 in. water gage

The tube face has seven 5/8 in. O.D. copper tubes with helically wound smooth aluminum fins mechanically bonded to the tubes. The casing of the tempering coil should be mounted squarely in the duct, because the tubes have a built-in pitch for proper drainage. A reducing valve and separator with drain should be located upstream from the tempering coil. A pressure gage and thermometer should be located directly downstream from the steam separator. A thermometer should be placed in the condensate line leaving the coil.[20] Then should come a float and thermostatic trap capable of handling two and one-half times as much condensate as the coil discharges,[7] or a total of 201 lb/hr.

Ordering specifications: Aerofin Type B Non-Freeze Coil, 24 in. Nominal Tube Length, 20-9/16 in. Overall Casing Width, Series Number N1, Aluminum Fins.

PREHEATER COIL

Coil Selection. The preheater coil selected should conform to the specifications of performance and capacity developed in the section entitled "Steam Heating Coils." The primary purpose of the preheater coil is to supply heat as necessary to control the dewpoint temperature of air leaving the washer. The preheater will not be exposed to freezing air, and need not be of the non-freeze type. It will be capable of handling the equivalent of 2000 cfm.

It should conform as closely as possible to the dimensions of the pieces of equipment adjacent to it.

The recommended preheater coil for the proposed unit is the Aerofin Type B Flexitube Coil. Though the casing is increased in depth as a result, this type of heater has unique offset bent tube design which permits individual tube expansion and contraction, and absorbs the strains within the heater itself.

The minimum temperature of air entering the coil will be 36°. The preheater must be able to heat this air to 144°.

The coil was selected from information given in Aerofin Bulletin B-54.(7) The coil most suitable for the requirements of the space would have a nominal tube length of 36 in. with an overall casing width of 20-9/16 in. and a face area of 4.2 sq. ft. Interpolating from the table on page 6, it was found that air entering the coil at 36° with a face velocity of 476 fpm would leave the coil at 141.4°. The coil is rated at conditions of 5 psig saturated steam. When using steam with 5° of superheat, the coil would have sufficient capacity for the requirements. The coil is described as "W2", which denotes two staggered rows with moderate spacing of the rows in the tube face. The condensate discharged would be approximately 242 lb./hr.

The approximate required capacity of the coil will be 233,000 Btu/hr.

Interpolation from the tables on page 5 shows an air friction drop of 0.113 inches of water gage for the air at an average temperature through the coil of 90°.

A summary of coil information follows:

Coil requirements: to heat 9000 lb/hr of air from 36° to 144°
Overall casing depth for two rows: 10 in.
Overall casing width: 20-9/16 in.
Nominal tube length: 36 in.; overall casing length, 44-1/2 in.
Face area: 4.2 sq. ft.
Face velocity: 476 fpm
Tube face: 9 in first row; 8 in second row
Supply connection: 2 in.
Return connection: 1-1/2 in.
Air friction drop: 0.113 in. water gage

The tubes are 5/8 in. O.D. copper with helically wound smooth aluminum fins mechanically bonded to the tubes. For proper drainage, the casing should be pitched 1/2 in. toward the return connection. The supply piping should be 2 in. nominal steel, and the drain piping should be 1-1/2 in. Fittings and specialties should be mounted in the same order as that specified for the tempering coil. The float and thermostatic trap should be capable of discharging two and one-half times full capacity coil condensate, or about 605 lb/hr.

Ordering specifications: Aerofin Type B Flexitube Coil, 36 in. Nominal Tube Length, 20-9/16 in. Overall Casing Width, Series Number W2, Aluminum Fins.

REHEATER COIL

Coil Selection. The reheater coil selected should conform to the specifications of performance and capacity developed in the section entitled "Steam Heating Coils." The purpose of the reheater is to raise the temperature of the air leaving the washer on winter cycles to that necessary to maintain the desired temperature in the conditioned space. On the summer cycle it is used to maintain the temperature of the conditioned space when dehumidification requires excessive chilling of the air. Since the winter processes demand the greatest capacity, the coil will be sized for the winter cycle. The reheater will be capable of handling the equivalent of 2000 cfm of air. It should conform as closely as possible to the dimensions of the washer adjacent to it.

The design, or maximum, requirement of the reheater coil is that it heat air which enters 68.5° to a temperature of 100°. The recommended coil for the proposed unit is the Aerofin Type B Flexitube Coil, which is similar in construction to the preheater coil recommended. The only limitation on air velocity over the face area of the coil is that it not be so high that the air friction loss becomes excessive.

The coil was selected from information given in Aerofin Bulletin B-54.(7) The coil most suitable for the requirements of the space would have a nominal tube length of 30 in. with an overall casing width of 20-9/16 in. and a face area of 3.5 sq. ft. Interpolating from the table on page 6, it was found that air entering the coil at 68.5° with a face velocity of 570 fpm would leave the coil at 108°. The coil is rated at conditions of 5 psig saturated steam. The coil is class "W1," which means one row with tubes moderately spaced. The condensate discharged when operating at the required capacity of 68,000 Btu/hr. would be about 71 lb/hr.

Interpolation from the tables on page 5 shows an air friction drop of 0.0719 inches of water gage for the air at an average temperature through the coil of 84°.

A summary of coil information follows:

Coil requirements to heat 9000 lb/hr of air from 68.5 to 100°
Overall casing depth: 10 in.
Overall casing width: 20-9/16 in.
Nominal tube length: 30 in.; overall casing length, 38-1/2 in.
Face area: 3.5 sq. ft.
Face velocity: 570 fpm
Tube face: 9
Supply connection: 1-1/2 in.
Return connection: 1 in.
Air friction drop: 0.0719 in. water gage

The coil consists of a single row of nine 5/8 in. O.D. copper tubes with helically wound smooth aluminum fins mechanically bonded to the tubes. For proper drainage, the casing should be pitched in the duct 1/2 in. toward the return connection, which is on the opposite side from the supply. Fittings and specialties should be mounted in the same order as that specified for the tempering and preheater coils. The float and thermostatic trap should be capable of discharging two and one-half times full capacity coil condensate, or approximately 180 lb/hr.

Ordering Specifications: Aerofin Type B Flexitube Coil, 30 in. Nominal Tube Length, 20-9/16 in. Overall Casing Width, Series Number W1, Aluminum Fins.

D. HOT AND CHILLED WATER COIL

Philosophy of Coil Selection. The usual method for selecting a chilled water coil is to select a coil that will cool the required amount of air from one wet-bulb temperature to a lower wet-bulb temperature when having a definite amount of refrigerant available at a given temperature. Then the heating capacity for the winter cycle of the coil when using hot water at the maximum temperature available is checked against the capacity specified. In most cases, the air conditioning requirements are such that a coil large enough for the cooling load has a surplus heating capacity. The same pump flow rate is used on both cycles.

The method used in this thesis was to select a chilled water coil that would cool the nominal cfm of air between those entering and leaving wet- and dry-bulb conditions which simulate typical commercial practice, when supplied with the available capacity of refrigerant. Though primarily a trial-and-error procedure, in each revision of the previous trial the factors that changed with, say, a change in number of coil rows, were quickly analyzed for their effect.

Design Criteria. The selected coil should be preferably of the same width and length as the direct expansion coil mounted immediately upstream. In the selection of the chilled water coil, all valves used are conservative to allow freedom of variation of the values both above and below the mean values. Aerofin Type "R" coil was recommended over the Type "C" coil. Rather than having return bends connecting all tubes, the Type "R" coil is cleanable, with removable steel header cover plates in order that the inside of the tubes may easily be inspected and studied for fouling. In addition, the effect of coil bypass factor may be demonstrated and studied by installing a hand shutoff valve at each pass. With an eight row coil the flow can be regulated to pass through from one to eight of the rows in singular sequence. The washer circulating pump will also be available for this coil, insuring abundant circulating capacity.

Selection. The recommended heating and cooling water coil for the proposed laboratory is the Aerofin Type "R" Removable Header Water Coil.(6) This coil should never be exposed to freezing air. Information about the coil follows:

Overall casing depth for 8 rows: 15 in.
Overall casing width: 20-9/16 in.
Nominal tube length: 36 in.; overall casing length, 40-1/2 in.
Face area: 4.03 sq. ft.
Face velocity: 496 ft./min.
Tube face: 12
Pressure drop through water circuit of coil: 1.25 ft. water gage
Inlet and outlet connections: 2 in. male pipe tap

Nominal cooling capacity of coil: 60,000 Btu/hr.
Air friction drop through dehumidifying coil: 0.765 in. water gage
Number fins per inch of tube length: 8
Number passes of refrigerant: 8

At a load of 60,000 Btu/hr for the cooling cycle, a minimum of 23 gpm must be circulated through the water chiller and coil. The number of passes will be the same as the number of rows for maximum water velocity. With a 12 tube face coil, the manufacturer's tables indicate a velocity of 2.4 ft/sec. An average of commercial applications shows a sensible heat ratio of 0.80.(29) Using this value and the water velocity just determined, Aerofin Bulletin C-58, page 11 shows a heat transfer coefficient "k" of 197 Btu/(hr) (sq. ft. face area) (^{o}F) (row).

The condition of the air leaving the cooling coil will be a function of the condition of air entering the coil, the coil sensible heat ratio, and the total amount of heat removed. The typical sensible heat ratio already selected is 0.80. Design air conditions entering the coil will be taken as 75^{o} dry-bulb and 50 per cent relative humidity. This decision was made to enhance the research possibilities of the coil, bearing in mind that more efficient air purification apparatus will make such conditions of coil entering air the rule rather than the exception in the near future. It is significant to note that more coil cooling surface will be required when the entering air is all return air than when outside air is being mixed with the return air, or all outside air is entering the coil. The leaving condition of enthalpy was found by use of the following formula:

(1) $Q_T = 4.5 \times \text{cfm} \times (h1 - h2)$

Where: Q_T = total heat removed by coil, Btu/hr
4.5 = constant by which ft^3/min are converted to lb/hr
h_1 = enthalpy of entering air, Btu/lb
$h2$ = enthalpy of leaving air, Btu/lb

(2) $h_1 - h_2 = \dfrac{Q_T}{4.5 \times \text{cfm}}$

(3) $h_2 = h_1 - \dfrac{Q_T}{4.5 \times \text{cfm}}$.

(4) $h_2 = 28.20 - \dfrac{60{,}000}{4.5 \times 2000}$.

(5) $h_2 = 28.20 - 6.67$

(6) $h_2 = 21.53$ Btu/lb.

The wet-bulb temperature corresponding to 21.53 Btu/lb is found from the Carrier psychrometric chart to be 52.2°F, and the intersection of the coil load line shows the leaving dry-bulb temperature to be 53.5°F. Water enters the coil at 45° and leaves at 55°. For counterflow conditions in the coil, the logarithmic mean temperature difference can be calculated or found on the manufacturer's chart to be 13.5. The number of rows of coil depth required are found from the Aerofin formula in Bulletin C-58 on page 8:

$$(1)\ \text{Rows depth} = \frac{Q_T}{K \times M.E.D. \times F.A.}$$

Where: Q_T = total heat load, Btu/hr.
K = coil heat transfer coefficient, Btu/(hr) (sq. ft. face area) (°F) (row)
M.E.D. = logarithmic mean effective temperature difference of water and air, °F
F.A. = face area, sq. ft.

$$(2)\ \text{Rows depth} = \frac{60{,}000}{197 \times 13.5 \times 4.03}$$

$$(3)\ \text{Rows depth} = 5.6$$

To condition air on the cooling cycle a 6-row coil will be required.

The coil will also be used for heating, preferably having the same capacity as the steam preheater coil so that it can be used as an alternate for the preheater. The preheater coil has a capacity of 233,000 Btu per hr. The Aerofin removable header water coil is not recommended for temperatures above 160°. It would be desirable to have a coil which can handle 180° water if possible; however, the feature of having removable headers is of more importance than the higher temperature, if both cannot be obtained. Water will enter the coil at 160° and leave at 140°. Air will enter the coil at 36° and leave at 144°. There will be counterflow between the entering air and water. The logarithmic mean temperature difference between the air and water will be 47°. The sensible heat ratio for the heating coil is 1.0. From page 11 of Aerofin Bulletin C-58 the "k" factor was found to be 160 Btu/(hr) (sq. ft. face area) (^{o}F) (row). The number of rows required can be calculated as follows:

$$\text{Rows depth} = \frac{Q_T}{K \times M.E.D. \times F.A.}$$

$$\text{Rows depth} = \frac{233,000}{160 \times 47 \times 4.03} = 7.67$$

Therefore, an 8-row coil should be used.

On the heating cycle the quantity of water flow required is as follows:

(1) $Q_T = \text{gpm} \times 488.4 \times \Delta t.$

Where: Q_T = total heat transferred, Btu/hr
gpm = flow through coil, gal/min
488.4 = conversion constant for changing gal/min to lb/hr, derived as follows:
(a) density water at 170^{o} = 60.8 lb/ft^3
(b) $\frac{60.8 \text{ lb/ft}^3}{7.48 \text{ gal/ft}^3} = 8.14$ lb/gal
(c) gpm x 8.14 lb/gal x 60 min/hr = gpm x 488.4
Δt = temperature rise of water through coil, ^{o}F

(2) $\text{gpm} = \frac{233,000}{483.4 \times 20} = 23.9$ gpm.

The air friction drop will be greatest when the coil is dehumidifying. It was found from the Aerofin chart in Bulletin C-58 on page 12 to be 0.765 in. water gage.

The head loss in the water circuit of the coil, through both headers and eight passes, was found on page six of Bulletin R-50 to be 1.25 feet water gage.

The removable header water coil recommended for the proposed laboratory is easily inspected and cleaned by simply removing a single cover plate at each end of the coil, exposing every tube and making it possible to clean each tube from either end of the coil. The coil is made of round seamless phosphorized copper tubing having an outside diameter of 5/8 in. O.D. Aluminum helical fins are 13/32 in. high and are wound under high pressure to insure a mechanically tight bond. The finned tubes are staggered in the direction of air flow. Tubes are not pitched in the casing. It is advisable, therefore, to pitch the casing on installation to insure perfect drainage.

The removable cover plate and baffles are of formed and welded tank steel. The flanges of the cover plate are bolted to the tube sheets by means of heat treated bolts with gaskets between tube sheet and header cover. The tube sheets are hot rolled carbon steel. Tubes are made tight in each tube sheet by belling and flaring. This method eliminates any ferrules or other projections and presents a streamline design for water flow. Type "R" coils are designed for 100/sq.in. working pressure and 160° maximum temperature on the water side. "Right" or "left" hand of the coil need not be specified because the water inlet and outlet connections can be used interchangeably to provide counterflow between air and water. It is recommended in the proposed unit that each header be drilled and tapped between each division plate, so that a gate valve and piping may be connected to the common discharge line. Pipe unions should be installed to facilitate clearing the pipe and valves when removing the coil headers. Each valve should be near the bottom of the header. Air vent valves may be inserted in place of the plugs at the top of each pass on both headers. Adding the outlet valves at each pass will facilitate demonstration of bypass factor of a coil by varying the number of rows in service from one to eight. The faces of the coil flanges are template punched for attaching ductwork.

Ordering Specifications: Aerofin Type "R" Removable Header Water Coil. 36 in. Nominal Tube Length, 20-9/16 in. Casing Width, Series No. 88, 8 pass, Aluminum Fins.

E. DIRECT EXPANSION COOLING COIL

Selection. The recommended direct expansion cooling coil for the proposed unit is the Aerofin Type "D". Information about the coil and selection procedure follows:(5)

Overall casing depth for 6 rows: 10 in.
Overall casing width: 20-9/16 in.
Nominal tube length: 36 in.; overall casing length, 40-1/2 in.
Face area: 4.06 sq. ft.
Face velocity: 493 ft./min.
Tube face: 12
Pressure drop through refrigerant circuit of coil: 12 psi
Pressure drop through refrigerant distributor: 1 psi
Liquid, or inlet, connection: 1-1/2 in. I.D. to fit 1-1/8 in. O.D. copper tubing
Vapor, or suction, connection: 2-1/8 in. O.D.
Nominal cooling capacity of coil: 60,000 Btu/hr.
Type of refrigerant distributor: centrifugal header
Air friction drop through dehumidifying coil: 0.578 in. water gage
Number fins per inch of tube length: 8
Number passes of refrigerant: 6
Normal charge of Refrigerant 12: 8.2 lb.

The same design conditions for air entering the coil and coil sensible heat ratio were used for the direct expansion cooling coil as for the chilled water coil. In selecting a direct expansion coil, there are a number of variables which act as criteria of design. For example, the wet- and dry-bulb conditions entering the coil, the air volume, and the cooling load may be specified. From this, the number of rows, size and number of units, refrigeration suction temperature, dry-bulb temperature leaving the coil, and the air friction may be determined.

The Aerofin Corporation publishes empirical data with which four types of solutions may be applied, according to the design information. The solution for the coil recommended for the proposed laboratory is based on specifications which call for a definite ratio of sensible heat to total heat.

As was the case with the chilled water coil, the room and coil sensible heat ratio are 0.80. By drawing a straight line on the psychrometric chart, as in the selection of the chilled water coil, the apparatus dewpoint temperature was found to be approximately 50.7°, the leaving wet-bulb temperature 52.2°, and the leaving dry-bulb temperature 53.5°. From the Aerofin table of total heat values, approximately 6.46 Btu must be removed from each pound of dry air conditioned. The number of rows required is found by utilizing the following formula:

$$(1)\ L = \frac{Q_T}{cfm \times (\ TH)}$$

Where: L = manufacturer's notation, no units
Q_T = total heat removed from air, Btu/hr
cfm = total volume of air through coil, ft^3/min
TH = Total Heat removed from air, Btu/lb

$$(2)\ L = \frac{60{,}000}{2000 \times 6.46} = 4.65$$

From the manufacturer's tables, 4.65 is slightly greater than the value requiring a 6-row coil, the maximum depth manufactured. For the 6-row coil, a symbol "M" shows that the value 2400 should be substituted in the following formula to find the required refrigerant suction temperature:

$$(1)\ t_r = t_c - \left(\frac{Q_T}{M \times F.A.}\right)$$

Where: t_r = refrigerant suction temp., °F
t_c = coil apparatus dewpoint temp., °F
Q_T = total heat removed from air, Btu/hr
M = manufacturer's notation, no units
$F.A.$ = face area of coil, ft^2

$$(2)\ t_r = 50.7 - \frac{60{,}000}{2400 \times 4.06}$$

$$(3)\ t_r = 50.7 - 6.2$$

$$(4)\ t_r = 44.5°F.$$

Since the value used in determining the number of rows was greater than the range given, performance tables for Aerofin coils having a face velocity of 500 fpm and different saturated suction temperatures at the coil outlet were consulted. The lowest entering wet-bulb proposed by the tables is 65°. By extrapolation, however, it is seen that the above refrigerant temperature of 44.5° is about right. It is noteworthy that the proposed coil must operate at lower sensible heat ratios than at present are common in comfort air conditioning. In practical applications this presents the choice of changing to a lower dry-bulb temperature and higher relative humidity, thus keeping the same effective temperature, or else using reheat.

The reason for the discrepancy in the rating tables is that it is always customary at the present time to use a certain amount of outside make-up air, if only to provide pressurization of the conditioned spaces. It is expected that in the future such a thing will be necessary only in limited areas of a commercial building, and no air will be required for ventilation because of highly efficient recovery apparatus.

The recommended coil will be furnished with an Aerofin patented centrifugal header type distributor. The standardized use of eight fins per inch gives efficient operating results under varying operating conditions.[5] To avoid the use of long single passes which would result in high pressure drops, each of the 12 tubes in the face of the coil represents the start of an individual refrigerant circuit supplied from the centrifugal distributor. In the recommended coil the refrigerant will make six passes before entering the common suction header, located on the same side of the coil as the distributor. The suction piping should be connected to the outlet located at the bottom of the coil as it is mounted in the duct, and the alternate connection should be plugged. Upstream from the centrifugal distributor should be placed, in order, the thermostatic expansion valve with external equalizer, solenoid valve, 80-mesh screen strainer, and

Sporlan "See-All." The external equalizer extends from a point below the diaphragm of the expansion valve to the coil suction line at a point adjacent to the gas bulb regulating superheat. The "See-All" not only provides a sight glass but also indicates the presence of moisture in the system. Leading consulting engineers commonly require it in all refrigerant circuits.

Fabrication of tubes and fins is similar to the other coils already described. After manufacture, the coil is tested to an air pressure of 250 psig. Then the coil is heated to a temperature of 180° and the pressure inside the coil reduced to 0.11 inches of mercury absolute. The unit is valved tight under this vacuum for a sufficient time to insure that no leaks are present, purged with inert gas, then sealed and crated.

Ordering Specifications: Aerofin Type D Direct Expansion Coil, 36 in. Nominal Tube Length, 20-9/16 in. Casing Width, Series No. 86, Aluminum Fins.

F. AIR WASHER-DEHUMIDIFIER

General. It was found that none of the commercially available washers matched the required capacity of the proposed unit. Also, it was felt that fabrication by local craftsmen would be less expensive than purchasing a commercial unit. Details of design of the washer-dehumidifier for the proposed laboratory will be described in this section. The basic principles incorporated into the design have been developed through many years of washer utilization by the industry, and are practiced by one or more of such leading manufacturers as Buffalo Forge,[23] Carrier,[28] Trane,[67] and Westinghouse.[49]

Criteria of Design. The unit must perform each of the following actions either continuously in operation or else during a specific cycle:

(1) Scrub impurities from the air.

(2) Dissolve entrained vapors and gases and hold in solution until the water is changed.

(3) Prevent particles of moisture from leaving the apparatus with the conditioned air.

(4) Saturate the air adiabatically.

(5) Heat and humidify the air.

(6) Cool and dehumidify the air.

In order to accomplish these things, the design was based on the following criteria:

(1) The casing must be sturdy in construction and must resist disintegration by corrosion and rust on the interior surfaces.

(2) The casing must be properly protected with insulation and coated with a vapor barrier to prevent sweating.

(3) Air entering the washer must be distributed uniformly over the entire cross-sectional area.

(4) Spray must be contained at both inlet and outlet by proper eliminators.

(5) The operating velocity must be high enough to throw the wetted particulate matter against the eliminator surfaces, yet not so high that moisture is blown off the lips back into the air stream.

(6) To insure efficient cleaning of the air, the scrubber plates should present a large impingement surface, which means narrow slots and multiple bends.

(7) The circulating pump must provide sufficient head pressure for good atomization at the spray nozzles.

(8) The length of spray chamber must be long enough to perform its most critical function, dehumidification, with a high efficiency.

(9) The washer tank must be deep enough to allow settling of impurities before recirculation of the water.

Description of Operation. Air will be drawn into the washer through closely spaced inlet baffles which cause enough pressure drop to insure distribution of air across the entire face area. In addition the baffles prevent backlash of spray water from the atomizing section. The air enters the spray chamber where it comes into intimate contact with atomized water from the evenly spaced spray nozzles. Soluble vapors are absorbed by the water and particulate matter is thoroughly wetted. Heat transfer between the water and air takes place according to the relative temperature of each. If adiabatic cooling or humidification and heating of the air takes place, then some moisture will be evaporated and compensated by automatic feed at the float control. If dehumidification and cooling takes place, then some moisture will be condensed from the air and will leave at the tank overflow connection. Leaving the spray chamber, the air impinges against metal plates which deflect the air stream and cause the heavy, wetted particulate matter to be deposited. Several changes of direction insure complete deposition of particulate matter. The scrubber plates are continuously flushed from above by flooding nozzles. Following the scrubber plates are eliminator plates which should not be wet by the flooding nozzles. The scrubber and eliminator plates in applications requiring moderate air cleaning are made up of approximately six 60-degree bends altogether. The eliminator plates have projecting lips to entrap entrained moisture which is thrown against the plates as the air stream changes direction.

Data. Pertinent data are as follows:

Nominal capacity of washer: 2000 cfm
Inside face area of washer: 4.0 sq. ft.
Inside width: 33 in.; inside height: 17-1/2 in.
Depth of tank: 10 in.; depth of water in tank: 6 in.
Length of unit: 95 in., inside of framing.
Number of spray nozzles: 10
Capacity of spray nozzles: 20 gpm
Number of flooding nozzles: 10
Capacity of flooding nozzles: 10 gpm
Circulating pump: 30 gpm at 70 ft. total head
Estimated air friction drop: 0.34 in. water gage

Casing. The top and side panels should be of 18 gage galvanized steel, reinforced with galvanized steel angles. Panel sheets should be joined by inside grooved seams. A marine light with vapor-proof globe and protective guard should be installed on the interior overhead. A thorough coating of chlorinated rubber-bitumen type paint should be applied to all inside and outside surfaces of the unit except the baffles, scrubber-eliminator plates, and piping. The outside of the casing should be insulated with 2 in. rigid insulation with vapor barrier. A coating of sealing compound should be applied next to the casing to fill all joints and prevent air leakage to the casing.

Tank. The tank should be of either 3/16 in. black steel plate or 16 gage galvanized steel. If of black steel, all seams should be heavy, continuous welds. If of galvanized sheet, the joints should be lapped, riveted, and soldered. It is imperative that all joints, rivets, and seams of the casing and tank be absolutely and permanently water-tight by being welded, or soldered over. Because of the great length of the proposed washer for its small capacity, a tank 10 in. deep will be sufficient. The design water level should be not less than 6 in. deep.

Weir. Located near the front of the tank, the weir consists of a heavy sheet steel bulkhead made rigid with angles at its top and made water-tight by riveting and soldering, or by welding, depending on whether it is of galvanized or black steel construction.

Inlet Baffles. The inlet baffles should be made of 24 gage galvanized steel, the vertical plates mounted on 3 in. centers and rigidly supported. The baffles should be removable without disturbing other parts of the unit. The baffles consist of three deflections.

Spray Nozzles. Spray nozzles are brass with a patented centrifugal whirling chamber. The recommended size of 3/8 in. x 3/16 in. (shank x orifice) may be purchased from the Buffalo Forge Company or Carrier Corporation. Carrier performance tables[28] show that with 25 psig pressure each nozzle can cover an area of approximately 115 sq. in. This gives a spray pattern of about 12 in. diameter. The spray piping should be of standard galvanized steel, sized large enough for easy attachment of the spray nozzles, and capped on the ends. There should be a pressure gage on the inlet header outside the washer to indicate water pressure. Should any nozzle become clogged, as indicated by an interrupted atomizing pattern, the head may be easily removed for cleaning.

Flooding Nozzles. The flooding nozzles located above the scrubber plates should be 1/4 in. x 3/16 in., obtainable from the Buffalo Forge Company. There should be ten nozzles, each spaced on 3 in. centers. The supply pressure at the nozzles should be reduced by a throttling globe valve on the inlet header so that each nozzle will discharge about 1 gpm. This will require only about 5 psig pressure at the nozzle header. A pressure gage should be installed on the low side of the throttling valve to indicate this pressure. The flooding nozzle area must be sealed from the atomizing chamber of the washer to prevent bypass of air flow. An access door with plastic inspection window should be placed on the washer above the flooding nozzle section.

Eliminators. The scrubber and eliminator plates are of integral construction. The six-bend sheets should be of one-piece, 24 gage galvanized steel, with 60-degree bends. This means that the air will be deflected alternately to right and left at angles of 30-degrees from straight ahead. Like the inlet baffles, the blades should be rigidly supported, yet capable of removal without disturbing the rest of the washer. The lips, made by hooking the trailing edge of each bend, should be small at the first bend, increasing in distance of protrusion towards the last bend. This will prevent flooding water from the nozzles above from being carried by the air stream to other bends. The flooding water should by no means wet the last two plates, which serve strictly for elimination of entrained moisture. The first four plates serve principally for scrubbing the wetted particulate matter from the air.

Inspection Door. An inspection door with plastic window should be fitted on a side panel. The panel must be reinforced to support the door. Standard construction for heavy-duty, water-tight doors should be followed. To prevent condensation and reduce heat transfer, the door should be insulated with 2 in. of cork. A foam rubber gasket should be affixed to the inside perimeter of the door, and tightness insured by four lever-operated cam locks.

Suction Screens. The removable suction screen should be made of bronze wire of smaller mesh than the 3/16 in. nozzle orifices. The screen should be installed immediately upstream of the weir to strain all water leaving the washer.

Service Connections. The unit should have a 3/4 in. brass float valve and a 3 in. overflow connection with P-trap. The overflow must be shielded against continuously draining spray from the atomizing chamber. A 3 in. drain should be located at the bottom of the tank. A check valve should be installed in the float circuit to prevent feed-back. Flanges should support all piping entering the washer casing.

Air Friction. Based on commercial experience, the air friction drop through the washer will be approximately 0.34 in. water gage.[23]

Circulating Pump. The pump should be equal to the Buffalo Forge Company "1½ CCL" pump described in another section. It has a single suction and is bronze-fitted. At 30 gpm it will develop the required 70 feet of head. Since page 10 of Bulletin 975-F shows that 1.33 horsepower would be required, the manufacturer recommends a 1-1/2 horsepower, 3-phase, 60-cycle, 208-220 volt, 1750 rpm open motor.[24]

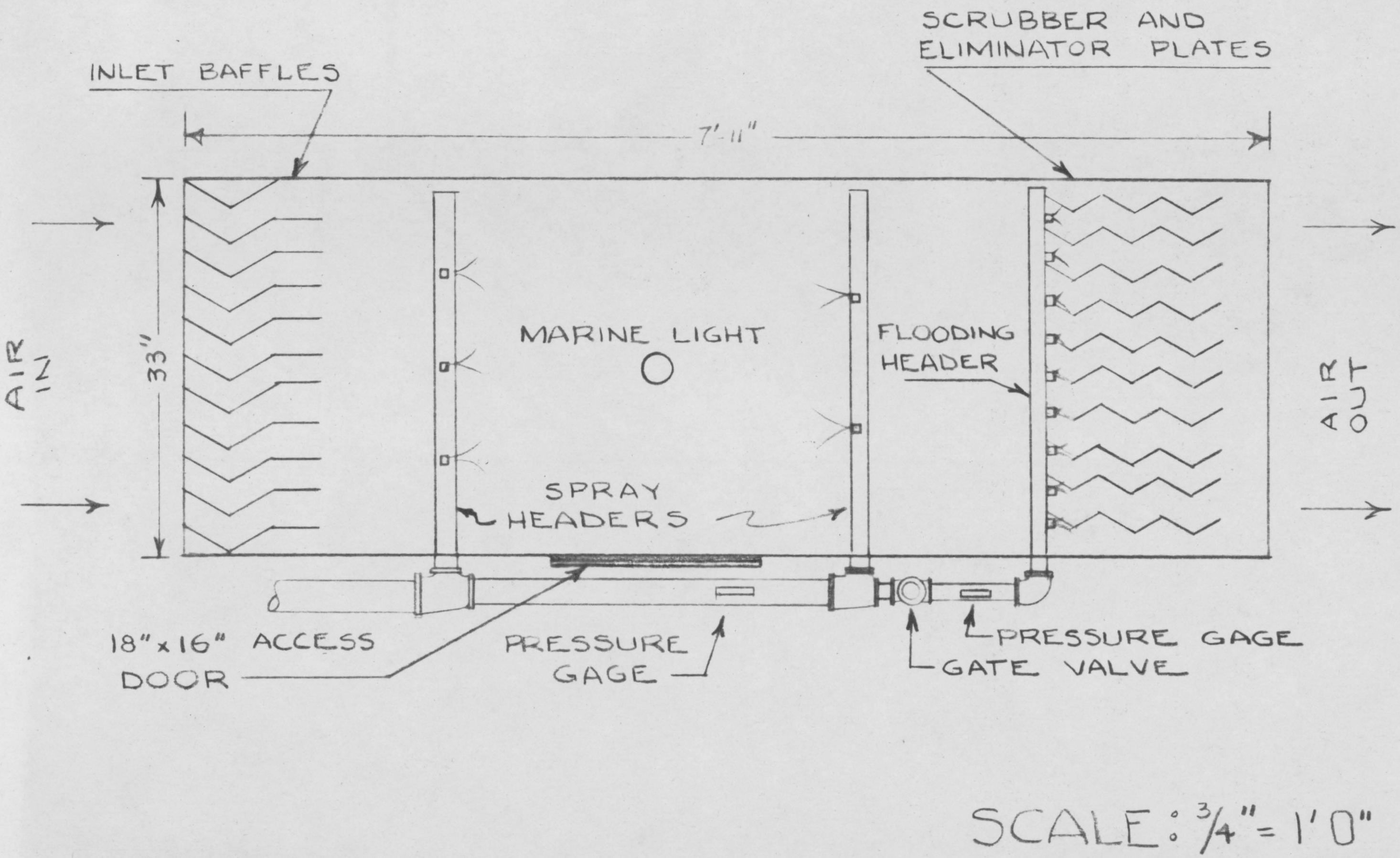


FIGURE 4. PLAN OF PROPOSED WASHER

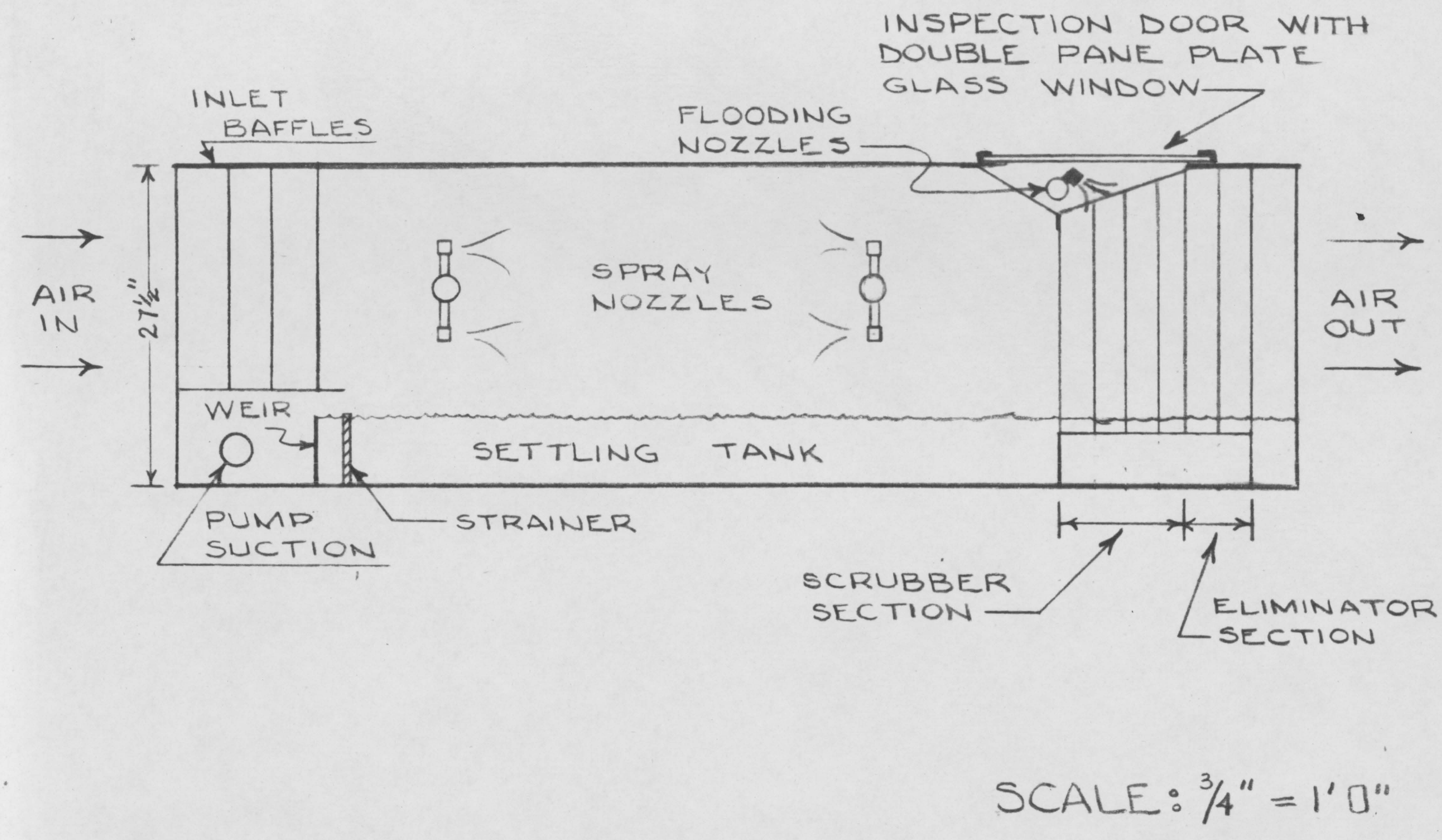


FIGURE 5. ELEVATION OF PROPOSED WASHER

G. AIR FLOW AND AIR TEMPERATURE MEASURING EQUIPMENT

The air flow measuring apparatus will consist of a nozzle chamber designed according to the standards of the air conditioning industry. By making use of the specifications given in A.S.R.E. Standard 16-56, together with reports on the extensive research which led to the adoption of that standard, it has been possible to design an apparatus for the proposed unit which will insure the utmost accuracy possible in any experimental work of this nature. In conjunction with the air flow measuring apparatus, a temperature measuring apparatus was designed, based on the same authoritative sources. Further, in order that dry-bulb and wet-bulb temperatures might be determined on each side of each componet of the built-up system, grids of copper-constantan thermocouples will be mounted.

In measuring the wet- and dry-bulb temperatures at such vital points as the A.S.R.E. chamber, mercury-in-glass thermometers calibrated by the Bureau of Standards and graduated in increments of 0.01 degree Fahrenheit should be used. This degree of accuracy is warranted because of the special sampling tube supplying air to the thermometers. In other instances, thermocouples should be used to measure the temperature.

A.S.R.E. CHAMBER

The air flow and air temperature measuring apparatus was designed in strict compliance with the specifications set forth in A.S.R.E. Standard 16-56.[35] Because of the brevity of specification details in the Standard, it was necessary to consult the original sourcework of D. D. Wile for specific details of design, or to perform original design work.

Important specifications governing the design of the apparatus for the proposed laboratory are as follows: [69] [20]

Center to center of nozzles	3 x average nozzle dia., min.
Duct wall to center of nozzle	1.5 x nozzle dia., min.
Upstream diffuser to nozzle inlet	1.5 x nozzle dia., min.
Nozzle outlet to downstream diffuser	2.5 x nozzle dia., min.
Approximate free areas of diffusers	40 per cent of face area
Area of venturi throat	1.5 x total nozzle area
Inside dia. of sampling tube	3 inches
Air velocity through sampling tube at thermometers	1000 fpm
Maximum nozzle discharge velocity	7000 fpm at throat
Minimum nozzle discharge velocity	3000 fpm at throat

Temperature Measuring Section. The apparatus consists of two separate parts: the temperature measuring section and the air flow measuring section. Although the temperature measuring section is not always necessary, because of the many thermocouple grids employed in the built-up unit, it has been found that this section is more reliable than such grids.[69]

For accuracy in temperature measurement of air in a duct which has different velocities as well as different temperatures, both horizontal and vertical mixing vanes are employed. These vanes provide mainly for making the velocity uniform so that the average of the temperatures in the venturi throat will indicate the true condition of the air. Regardless of how accurately the temperatures across the duct cross-section are averaged, differences in velocities would make the reading inaccurate. A scaled sketch of the proposed design of the air mixers is shown. The vanes should be approximately 4.25 in. wide x 1.5 in. deep, and set to deflect the air at 45 degrees. The vanes are arranged to divide the air flow into a number of small streams and then divert these streams across each other. To a certain extent, the greater the number of vanes, the better the mixing. It is important that the vanes extend only about halfway across the duct with the remaining area blocked off to allow space into which the deflected air streams can flow. The air velocity through the mixer face area should be about 1000 fpm. The resistance will be about 0.5 in. water gage for each mixer.[69]

The venturi arrangement serves to minimize non-uniformity of the air stream velocity and also reduces the required size of sampling tube. It is desirable to make the venturi throat as small as possible without developing excessive head loss.

A satisfactory size is about 1.5 times the total nozzle area <u>in excess of the area occupied by the sampling tubes</u>. The velocity head at the venturi throat is partially recovered depending on the angle of the discharge flare. Typically, the head loss would be approximately 1/4 of the nozzle head when all nozzles are in use. The section should be externally well insulated, including the sampling tube and fan. There is no official data available on the optimum design of the sampling tube placed in the venturi throat. The accompanying sketch shows a scaled plan from which the sampling tube for the proposed laboratory may be constructed. The basic premise on which it was designed was the fact that the tube leading the air to the sampling fan was to have a minimum inside diameter of three inches. The purpose of the fan is to withdraw a representative sample of air from each small cross-sectional area of the venturi through which the air flows so that a single dry-bulb and a single wet-bulb thermometer placed at the inlet of the fan will give a true indication of average air conditions. The fan discharges its air into the flow measuring chamber.

Since the area of the three-inch tube is 7.06 sq. in. and the specified velocity of air through the tube is 1000 fpm, the required capacity of the fan is 49 cfm. Since the fan discharges the air into the flow measuring chamber well upstream of the nozzles, the amount of air diverted to the fan is not important, within the limits prescribed. It was arbitrarily decided that the combined area of the sampling holes should equal or slightly exceed the area of the three-inch tube. Following a reasonable proportion of parts, the three-inch tube should be placed horizontally in the venturi throat, with six extensions above and six extensions below. The entire area of the throat could be sampled in this way by 30 intake holes, 6 in the three-inch tube and 2 in the extension tubes. Precisely, the holes should be 35/64 in. diameter, but it would serve as well to make them 9/16 in., the next standard size larger. The extension tubes should be of thin wall construction at 3/4 in. O.D. The combined area of the sampling apparatus would be approximately 99 sq. in. To this should be added 145.5 sq. in., which is one and one-half times the total nozzle area of the flow measuring chamber. This gives 244.5 sq. in., which is the required total area of the venturi throat. Since the inside dimensions of the duct are 17 in. x 17 in., the venturi throat should be 17 in. wide x 14.5 in. high. Though the construction just outlined would conform to the present standards, the writer of this thesis is not convinced that careful observation in the proposed laboratory would not permit a better design to be worked out, which could be submitted for official adoption by the industry.

Air Flow Measuring Section. The principal dimensions of the air flow measuring section are based on the size of each nozzle, if more than one is used, and the total number of nozzles. As was seen previously, the free area of the venturi throat was based on the total nozzle area employed.

The two criteria of design which were followed were the specifications that the nozzle discharge velocity should be between the limits of 3000 and 7000 fpm at the throat. To find the total nozzle throat area for the least possible pressure drop for the above conditions, a nominal air flow rate of 2000 cfm at standard conditions and a throat velocity of 3000 fpm were used as follows:

(1) $V = 4005\sqrt{h}$.

Where: V= velocity, ft/min

4005= constant

h= velocity head, in. water gage

(2) $4005\sqrt{h} = 3000$

(3) $h = \left(\frac{3000}{4005}\right)^2 = 0.564$ in. water gage.

The required nozzle area for this velocity head, or pressure drop across the nozzles, may be found as follows:

(1) $Q = 1096 \times C \times A \times \sqrt{h \times v_h}$

Where: Q= air flow, ft^3/min

1096= constant

C= coefficient of discharge

A= area of nozzle, ft^2

h = static press. difference across nozzle, in. water gage

$\sqrt{n}$ = specific volume of air at standard conditions ft^3/lb.

(2) $2000 = 1096 \times 0.99 \times A \times \sqrt{0.564 \times 13.34}$

(3) $A = \dfrac{2000}{1096 \times 0.99 \times 2.74}$

(4) $A = 0.674$ sq. ft. $= 97.0$ sq. in.

The coefficient of discharge, C, varies slightly with diameter of nozzle, velocity, temperature, and pressure of the air, but for the values used in this design, the coefficient is constant at 0.99.(69) Using the same equation as that above, or $V = 4005\sqrt{h}$, it was found that the pressure drop across the nozzles would be 3.07 in. water gage at a throat velocity of 7000 fpm. It is known that the system resistance may be assumed to vary as the square of the volume rate of flow through the airway components.(29) To avoid exceeding the design capacity of the fan, this maximum velocity and maximum pressure drop would be employed only at reduced air flow rates. To give versatility to the unit, a nozzle should be installed so that the unit may be tested at this maximum pressure drop, and hence greatest degree of experimental accuracy,(69) at a minimum of 500 cfm of air measured at standard conditions. The area of the required nozzle is calculated as follows:

(1) $Q = 1096 \times C \times A \times \sqrt{h \times \sqrt{n}}$.

(2) $500 = 1096 \times 0.99 \times A \times \sqrt{3.07 \times 13.34}$.

$$(3)\ A = \frac{500}{1096 \times 0.99 \times 6.39}$$

(4) A = 0.0721 sq. ft. = 10.38 sq. in.

(5) Diameter = 3.63 in.

This nozzle has the area required to give greatest accuracy when measuring the smallest increment of flow for which the proposed built-up unit is designed. It was decided to let the remainder of the nozzle area, or 97.00 less 10.38, be divided in four parts, to give acceptable increments to the rating tests. Dividing 86.62 sq. in. among four nozzles would give each a diameter of 5.25 in. Incidentally, it can be shown that the area required to measure the nominal flow of 2000 cfm at standard conditions with maximum throat velocity and maximum pressure drop would require only two of the 5.25 in. nozzles. By specification,[20] the upstream diffuser will be a minimum of 8 in. from the nozzle inlet, and the downstream diffuser will be a minimum of 14 in. from the nozzle outlet. The inlet of the chamber to the upstream diffuser should be a reasonable distance, say 9 in. The duct leaving the downstream diffuser should make a gradual transition of not more than a 30-degree angle to the fan inlet.[38] An inclined tube manometer, graduated to read in increments of 0.01 in. water, should be installed on the chamber as shown in the sketch. If desired, a second one could be placed in parallel. A scaled sketch is also included which shows a cross-section view of the proposed chamber at the throat of the nozzle assembly. All dimensions are in compliance with the Standard.

<u>Diffusers</u>. Using perforated plates as diffusing baffles, both upstream and downstream from the nozzle, makes it possible to confine the nozzles in a comparatively short duct length.[69] Based on the specifications, perforated plates having 1/4 in. holes staggered on 3/8 in. centers giving a free area of 40 per cent of the face area should be used. The approach velocity heads lost was given as 8.4.[69] Since the air flow chamber will be 38 in. square, the area will be 1444 sq. in. The friction drop through each diffuser may be calculated as follows:

(1) $V = 4005\sqrt{h}$.

Where: V = velocity of air, ft/min

$$V = \frac{2000 \text{ ft}^3/\text{min} \times 144 \text{ in}^2/\text{ft}^2}{1444 \text{ in}^2} = 200$$

4005 = constant

h = press. drop, in. water gage.

(2) $h = \left(\frac{200}{4005}\right)^2 = 0.0025$ in. water gage.

Drop across perforated baffle, with 40 per cent free area:

8.4 x 0.0025 = 0.021 in. water gage.

<u>Air Friction Drop</u>. The air friction drop through the proposed chamber may be tabulated as follows. The values are based on similar, previous designs.[69]

Horizontal mixers..................................	0.5
Vertical mixers....................................	0.5
Venturi (1/4 of nozzle head when all nozzles are in use).................................	0.141

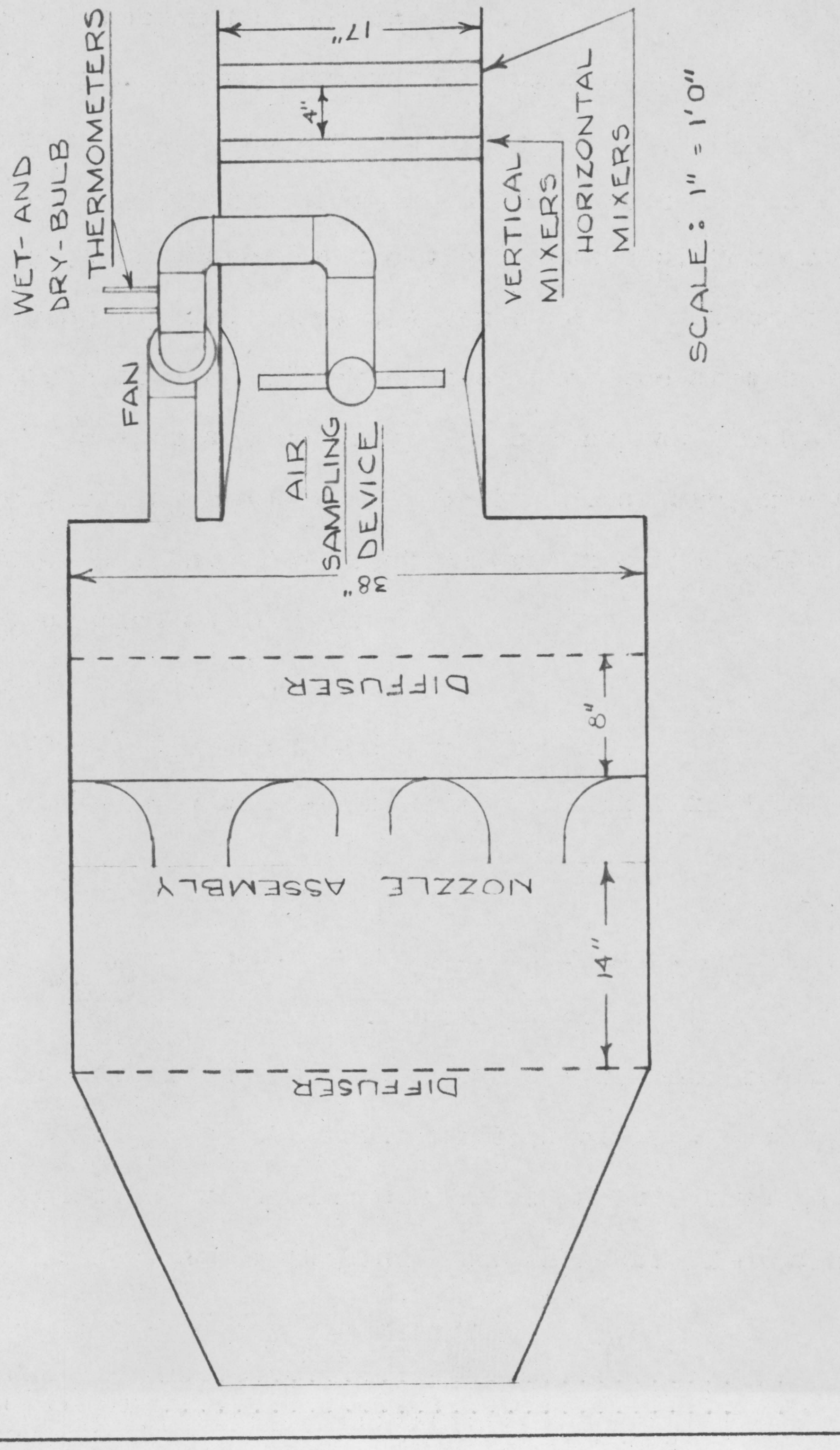
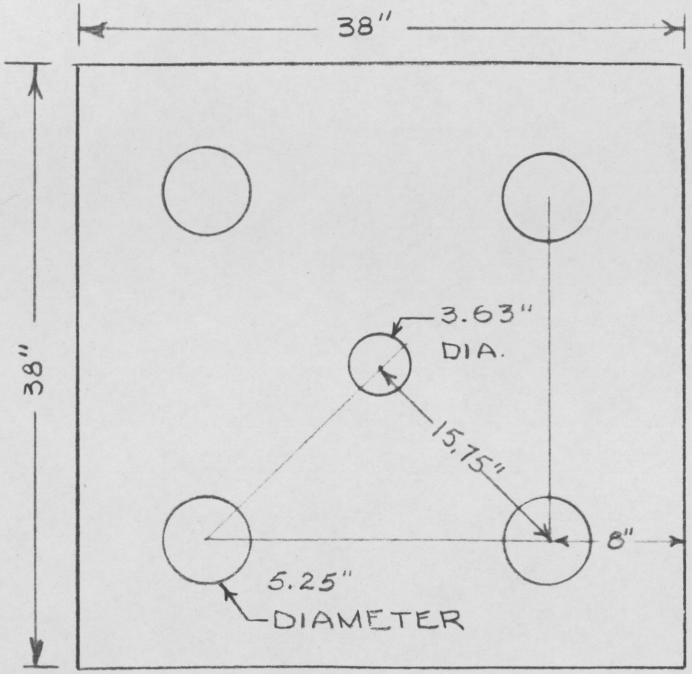


FIGURE 6. PLAN OF PROPOSED AIR TEMPERATURE AND AIR FLOW MEASURING APPARATUS



SCALE: 1" = 1'0"

FIGURE 7. CROSS-SECTION VIEW AT THROAT OF NOZZLE ASSEMBLY OF AIR FLOW MEASURING APPARATUS

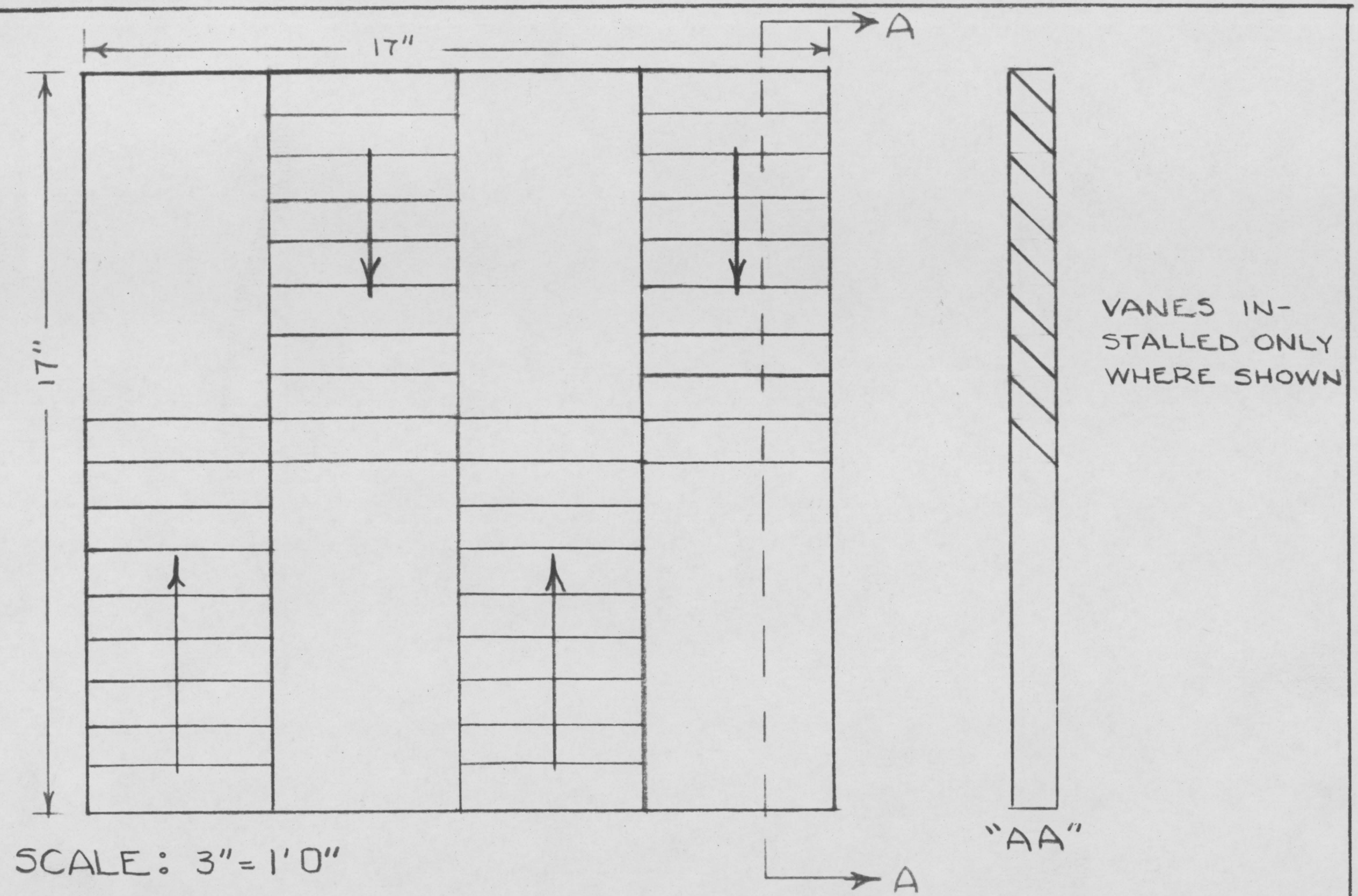


FIGURE 8. AIR MIXER FOR AIR TEMPERATURE MEASURING APPARATUS

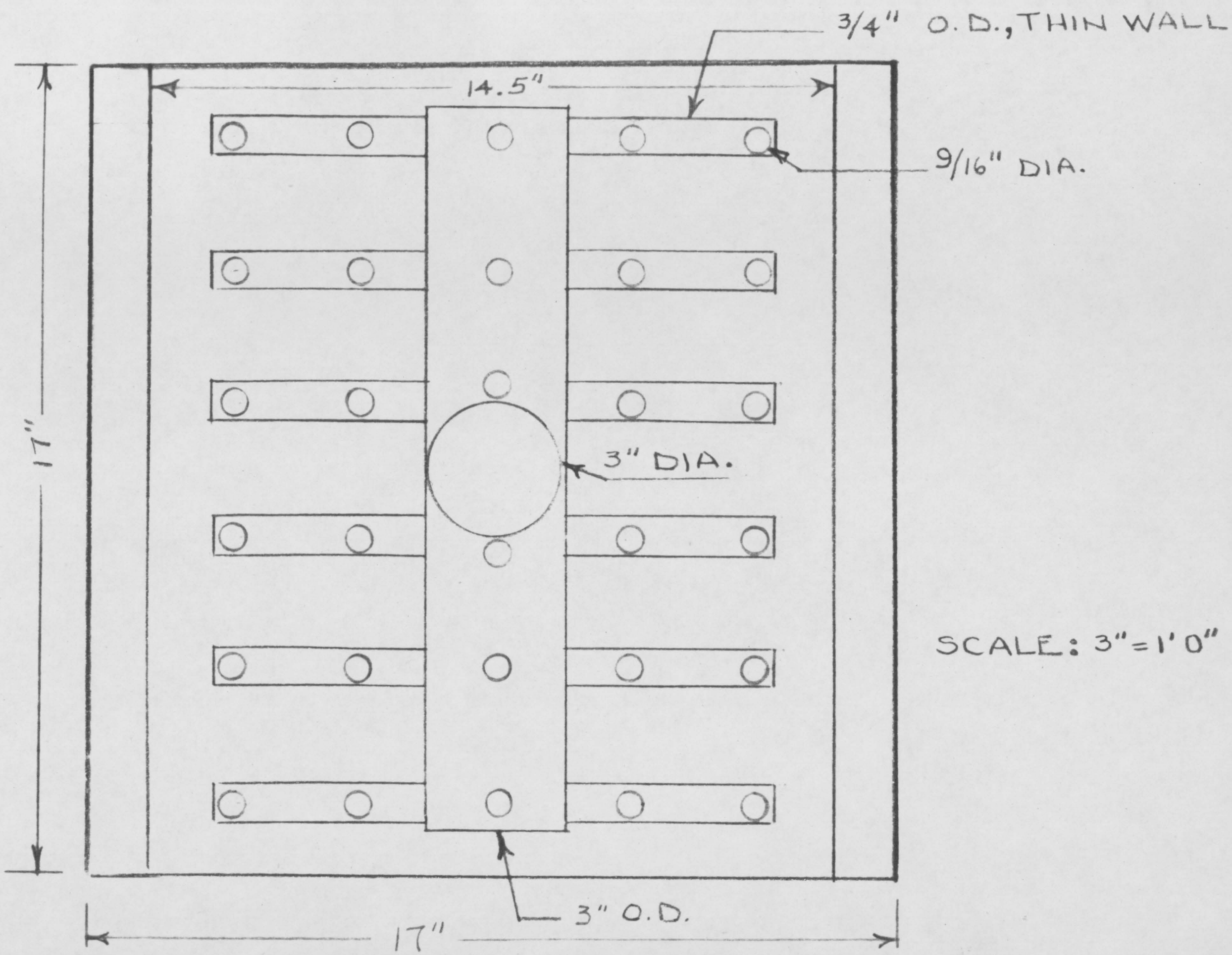


FIGURE 9. SAMPLING TUBE FOR AIR TEMPERATURE MEASURING APPARATUS

Upstream diffuser	0.021
Nozzles (3000 fpm at nozzle throat)	0.564
Downstream diffuser	0.021
Total drop, in. water gage	1.747

THERMOCOUPLES

In this section the measurement of air temperature by means of thermocouples will be described. Then a brief section will describe the application of thermocouples to other temperature measurements on the built-up unit.

A simple wire latticework should be constructed in the duct to support the thermocouple junctions. The couples should be connected in parallel to give an average of all the temperatures sensed. An effort should be made in designing the duct to have the air velocity at the section being measured as uniform as possible. The accuracy in using the average of each thermocouple indication is dependent on equal velocity of air past each couple. In setting up the unit a Pitot-static traverse may be used to check the velocity across the duct section. However, finding variations in velocity shows only that accuracy in measuring temperature will not be good unless the duct can be re-designed to make the velocity uniform at all points.

Past experience of the Mechanical Engineering Staff should be utilized in constructing the grids. There is no standard recommendation. There is no upper limit as to how many couples should be placed in a given cross-section, and installing more couples does not require more points in the rotary selector switch when the couples are wired in parallel.

Good practice should follow closely the information summarized by D. D. Wile in the article, "Measurement of Temperature in the Laboratory by Means of Thermocouples."[70]

The temperature within an insulated pipe can be ascertained by measuring the temperature of the pipe surface. Soft solder applied with a soldering iron is the most satisfactory method of attaching thermocouples to refrigerant lines and other metallic surfaces. The leads should enter the solder separately so that the junction will be at the metal surface and not outside of it.

Rotary Switch. On the central control panel there should be a rotary switch to which all of the thermocouple leads are brought. All of the thermocouples in a single grid should be wired in parallel so that only one switch connection is taken. The switch should be equal to the Brown Series 911A1B Switch Assembly manufactured by the Minneapolis-Honeywell Regulator Company.[54] This switch is moisture-proof with laminated brushes fastened to a rotating switch arm; the brushes move across the contact segments to make positive low resistance contacts. The assembly should be flush-mounted on the panel. The air conditioning built-up unit should be completed before purchase of the switch since the exact number of points required is indeterminate, pending inevitable modifications.

Potentiometer. It is recommended that the existing recording strip chart potentiometer previously used for heat pump research be used in the proposed unit. It has a variable span from 0-1 and 0-51 millivolts. The recorded emf can be converted to temperature by using standard emf-temperature tables. The instrument was manufactured by the Brown Instrument Company, Division of Minneapolis-Honeywell Regulator Company, Philadelphia, Pennsylvania. This instrument will record the temperature of any thermocouple circuit connected to it by the selector, or rotary, switch.

Thermocouple Locations. In order that the dry-bulb and wet-bulb temperatures might be determined at the inlet of each component of the built-up system, a grid of copper-constantan thermocouples should be mounted at each of the following locations:

(1) In the return air duct

(2) Tempering coil inlet

(3) Tempering coil outlet

(4) Preheater coil inlet

(5) Direct expansion coil inlet

(6) Washer inlet

(7) Reheater coil inlet

Dry-bulb temperatures should be taken by soldering and taping copper-constantan thermocouples at the following locations:

(1) Water entering condenser

(2) Water leaving condenser

(3) Suction vapor entering refrigerant heat exchanger

(4) Suction vapor entering compressor

(5) Compressor discharge

(6) Liquid entering refrigerant heat exchanger

(7) Liquid leaving refrigerant heat exchanger

(8) Outlet of chiller

(9) Outlet header of direct expansion coil

(10) Water entering and leaving condensate coolers on tempering coil, preheater coil, and reheater coil

(11) Water entering circulating pump

(12) Water leaving circulating pump

(13) Water leaving chiller

(14) Water leaving water heater

(15) Water leaving mixing valve

(16) Water leaving chilled and hot water coil

(17) Water leaving washer

(18) Steam supply to the steam heating coils

(19) Condensate leaving tempering coil

(20) Condensate leaving preheater coil

(21) Condensate leaving reheater coil

H. FAN SYSTEM AIRWAY COMPONENTS

The duct system for the proposed unit must be capable of conveying 2000 cfm of return air, 2000 cfm of outside air, or 2000 cfm of a mixture of return and outside air to the built-up conditioning unit; and of conveying 2000 cfm of conditioned air from the outlet of the unit to the insulated test room. Supply and return duct connections to the test room will be made at the location of an inspection door high on the north wall of the room. The ductwork should be designed to present a neat appearance; have minimum airway resistance consistent with space allotments; be of sturdy, airtight construction; and be well insulated to prevent exchange of heat between the conditioned air and the outside.

Based on the research reported in the Review of Literature, this section will cover the design of the ductwork for the proposed system, and the details of selection, application, and design necessary to incorporate the fan and dampers into the proposed built-up unit.

DUCTWORK

The design of the ductwork was based on the best standard contemporary practice. The sizes of ducts are set by the maximum air velocities which can be used without causing undue noise or without causing excessive friction loss. Larger ducts would reduce frictional losses, but the space and investment requirements offset the power saving at the fan. For the proposed unit, the ducts were sized using the equal friction method. The ductwork on the suction side of the fan was sized at 0.06 in. water gage per 100 ft. of length, and the ductwork on the discharge side of the fan was sized at 0.08. This is standard practice in the industry, based on practical economics. The maximum resistance for the unit will occur when handling all outside air. The two branches of duct at the suction side were not balanced in friction loss by sizing since they can be balanced by the dampers installed.

In general, the duct layout was made as direct as possible, sharp bends were avoided, and the duct cross-section, if rectangular, had a low aspect ratio. Contractions in the proposed duct were designed not to exceed an included angle of 30 degrees, and expansion-type transitions were designed not to exceed an included angle of 20 degrees. Standard recommendations designate that each side of an expanding duct have a slope not exceeding one inch in seven, which is just over eight degrees, or an included angle of about 16 degrees;[29] another recommendation designates that each side should not diverge more than 10 degrees, or an included angle of 20 degrees.[67]

The pressure loss due to gradual contractions is completely negligible, and is very small with gradual expansions. The difficulty was not so much to avoid excessive pressure loss, but to convey the air properly into the equipment. Duct design is still a specialized field without specialists; standard procedure follows a few basic approximations and simple principles, some of questionable accuracy.

The length of straight duct was taken from center-to-center measurements of the scaled drawing on Plate 1. The new system of calculating equivalent length of elbows, as shown in the 1956 Heating, Ventilating, Air Conditioning Guide, was utilized. The aspect ratio of an elbow, unlike a duct run, is always equal to the depth divided by the width, where the width of the elbow is always the dimension which lies in the same plane as the radius of the elbow. The optimum value of aspect ratio is 1.0.[67] When an elbow is followed by a duct whose length is at least four times the equivalent length of the elbow, no splitter vanes are required if the centerline radius ratio is 1.5.[67]

The air circuit was taken from the screen at the gooseneck of the outside air intake, through the tempering coil into the unit proper, through all the components of the duct system to the fan, and from the fan discharge to the duct discharge at the test room. A tabulation of the total static pressure losses of the proposed duct airway circuit follows:

	In. Water Gage
Ductwork	0.315
Tempering coil	0.059
Disposable filter, clean	0.100
Electronic filter with dry after-filter	0.290
Activated carbon filter	0.200
Preheater coil	0.113
Direct expansion coil, dehumidifying	0.578
Chilled and hot water coil, dehumidifying	0.765
Washer	0.340
Reheater coil	0.072
Horizontal mixers in temperature measuring chamber	0.500
Vertical mixers in temperature measuring chamber	0.500
Upstream diffuser in flow measuring chamber	0.021
Downstream diffuser in flow measuring chamber	0.021
Subtotal	3.874
Static Pressure Loss at Minimum Design Velocity of Air Through Measuring Chamber:	
Venturi at 1/4 of nozzle pressure loss	0.141
Measuring nozzles	0.564
Total pressure loss for 3000 fpm velocity	4.579
Static Pressure Loss at Maximum Design Velocity of Air Through Measuring Chamber:	
Venturi at 1/4 of nozzle pressure loss	0.765
Measuring nozzles	3.070
Total pressure loss for 7000 fpm velocity	7.709

Since the velocities at the fan outlet and the duct discharge into the test room are approximately equal, there was no static pressure regain to account for. The resistances shown in the table are typical for the specific components which were recommended in their respective design sections of this thesis. By the very nature of the selection, however, they are typical of the equipment represented. In most cases the equipment of any leading manufacturer, if substituted for that which was recommended for the proposed unit, would have almost identical characteristics. The resistances shown are for the nominal design air flow of 2000 cfm. At lower quantities the resistance would decrease greatly. In addition, the static pressure drop across the nozzles of the air flow measuring chamber will increase the static pressure loss from 0.564 to 3.07 in. water gage, depending on whether the nozzle throat velocity is the minimum or the maximum value. Any velocity in the designed range of 3000 fpm to 7000 fpm is acceptable; greater accuracy is ascribed to results using the greater pressure drop.(69) Another item that would make the total loss less than indicated in the tabulation is that the resistance of each dehumidifying coil was taken when wet. In practice, one will always be dry.

Since the above measurements would be conducted in a laboratory at 2100 feet altitude, the maximum static pressure required for standard conditions of discharge would be 7.709 divided by 0.92,(28) or 8.38 in. water gage. The minimum static pressure required for 2000 cfm air flow corrected to standard atmospheric pressure would be 4.579 divided by 0.92, or 4.98 in. water gage. Though it would be desirable to have a fan that develops 8.38 in. static pressure, one is not commercially available. The selected fan is rated at 6.0 in. static pressure. Even at the full 2000 cfm flow the measurements will be in the range of highly reliable accuracy, and at reduced rates of flow, some of the nozzles can be blanked off with tape to secure higher pressure drops.

Duct Insulation. All flat surfaces of the proposed unit exposed to transfer of heat should be insulated with two inches of glass fiber equal to Owens-Corning Fiberglas Vapor-Seal Duct Insulation,[41] which is designed especially for use on the outside of air conditioning ducts. It is a board-like material of fibrous glass, with a factory-applied vapor barrier of asphalt and kraft paper. Standard panels are 24 in. x 48 in. It can be used where the temperature does not exceed 150°. A flexible glass fiber with vapor barrier should be applied to the curved surfaces. In all cases the vapor barrier should be on the outside of the insulation.

Ductwork Thickness. The sheet metal for the duct system should be galvanized steel or aluminum. In either case the thickness should be as specified for the greatest duct dimension in the latest A.S.H.R.A.E. Guide governing duct installation practice. Plate 1 shows correct gage sizes.

Workmanship. A special effort should be made, because of the exceptionally high pressures and the experimental nature of the work, to make the joints absolutely airtight, preferably by soldering. All joints should be pressure-tested before the insulation is applied. All joints in the insulation should be plugged to prevent air from reaching the duct surfaces. The duct insulation should be so applied that any component of the built-up unit may be removed without disturbing the adjacent insulation.

Special Appliances. All access panels should contain a window of double plate glass with a one-inch air space, hermetically sealed. Wherever deemed appropriate for viewing the internal equipment, a similar window should be installed. There should be a small marine-type light inside the duct, mounted at the top, to illuminate the area viewed through the glass. One side of all duct elbows should be a removable plastic panel so that the air flow pattern can be visually studied when a eucalyptus smoke pot is inserted in the upstream duct. This feature must be consistent with the important specification that the ducts be completely airtight, regardless of any other considerations.

A corrosion resistant drain pan of copper should be fitted beneath the direct expansion coil and the chilled water coil to catch condensed moisture. The pan should slope and drain into a graduated cylinder, which can be drained to waste after the condensate volume and temperature are determined. All drain connections should be 2 in. nominal size. The pan and graduated cylinder should be insulated with 2 in. of glass fiber with vapor barrier.

The metalwork supports for all components of the proposed system should be capable of being leveled. Each component must be tested for level before operation can commence.

This is important because most of the equipment must drain properly, either internally or externally, for proper operation.

The flanged joints of each component where possible should be connected to the ductwork with removable nuts and bolts, with a neoprene rubber gasket making the required airtight joint.

DAMPERS

The face damper for the dehumidifying coils is of opposed-blade type construction. The dehumidifier bypass damper should, if possible, balance the resistance through the parallel circuit, which consists of the direct expansion and chilled water coils. However, if the resistance through the bypass alone were to equal the coil resistance, the bypass duct would be less than one inch in height. Therefore, a perforated baffle should be installed in the bypass duct, which should have a height of six inches and a width equal to the coils bypassed. The six-inch height was selected to give substantial size to the bypass dampers. In order that the bypass air volume match the air flow characteristics of the face dampers, the bypass damper should have a minimum of two opposed blades. After installation of the coils, a perforated plate can be drilled to match the resistance of the coil airway circuit. The resistance of the perforated baffle is based on per cent of free area. The required resistance through the bypass may be converted to "approach velocity heads lost" by dividing the required resistance in inches of water by the velocity head. Then the following table may prove helpful in determining the per cent of free area of the holes in the perforated baffle:(69)

Per Cent of Free Area	20	25	30	35	40
Approach Velocity Heads Lost	51	30	19	12	8.4

The dampers in the outside air and return air ducts should be the opposed-blade, manually operated type. With the aid of the air flow measuring apparatus, the dampers, which should be mechanically interconnected, may be calibrated.

There should be a set of inlet vanes, purchased with the fan, at the fan inlet. Inlet vanes are manually operated to adjust fan volume without great loss in fan efficiency.

FAN

Determination of optimum fan size usually involves balancing the minimum size against the outlet velocity, sound, efficiency, and class of construction. There are two types of construction offered:

(1) Class I fans are designed for operation against a maximum of 3-3/4 in. water gage total pressure when handling air at 70° at sea level and at a minimum tip speed of 10,000 fpm. Anti-friction bearings are standard, with sleeve bearings available.

(2) Class II fans are designed for operation against a maximum of 6-3/4 in. water gage total pressure when handling air at 70° at sea level and at a maximum tip speed of 13,000 fpm. They are equipped with anti-friction bearings only.

Fans operating at the higher static pressures generally have greater allowable outlet velocities. The higher the static pressure the greater the speed and air quantity at which the maximum efficiency occurs. Since the best sound characteristics are obtained at the point of maximum efficiency, it is evident that outlet velocity cannot be used as a criterion for selection from the standpoint of sound. Rather, for minimum resultant noise levels, the fan should be selected at the point of maximum efficiency and a smooth, easy conversion should be made from the fan outlet to the desired duct size.

The fan recommended for installation in the proposed built-up unit is a Carrier Model 27 CN. Its design and operating characteristics satisfy the requirements found desirable in the Review of Literature. There were a number of factors to be considered in determining the optimum fan size. First, the requirements of the system were determined: air quantity, static pressure, air density, and prevailing sound level. The selection then involved balancing the minimum size against the outlet velocity, sound level, efficiency, and class of construction. For a fan running at constant speed, the maximum noise occurs at free delivery conditions, with decreasing noise level at less volume and increased pressure. Minimum noise level occurs in the region of maximum efficiency. When a fan must deliver a given cfm of air at a higher static pressure, its speed must be increased, and the static pressure varies as the square of the speed.(67) The noise level also will increase. A lower noise level can be obtained by selecting a larger fan which, to deliver the same amount of air at the same pressure, can run at a lower speed and yet operate at the point of maximum efficiency. Two geometrically similar fans, such as the products of the same manufacturer, are alike in all proportions and in all details regardless of their size, and have nearly similar performance characteristics and efficiency curves.

The Carrier selection tables give data on fan performance in that part of the range of maximum efficiency, which for a backward curved blade fan is quite wide. The final choice was made from three suitable fans. Each would render satisfactory service; the problem was to decide which would best serve the purposes of the proposed unit. The data for each is tabulated below. No interpolations were made.

Size	Wheel Diameter	Cfm	Outlet Velocity	RPM	BHP	Maximum BHP	Corrected Maximum BHP
245	12-1/4	2064	2400	3464	2.81	3.17	2.97
270	13-1/2	2080	2000	3018	2.88	3.4	3.13
300	15	2048	1600	2629	2.97	3.82	3.51

All of the above data is at the highest rating of 6 in. static pressure. Model 245 could use a three horsepower motor, whereas the other models would require the next larger, or five horsepower, size. The lowest velocity pressure is developed at the outlet of the largest fan. Since the limit for Class II fans is 6-3/4 in. total pressure, of the three fans described Model 300 would have the greatest static pressure available, though the difference is negligible and is not significant enough to publish. Having the same efficiency as the other fans, Model 300 would have the lowest noise level because of the lowest outlet velocity and lowest speed. For these reasons, Model 300 is recommended for the proposed unit. With its five horsepower motor, it would be possible to experiment with higher air flow rates and static pressures than would be possible with the other machines.

In determining the direction of rotation, the fan is viewed from the side opposite the inlet. "Arrangement Number 2, Single Width, Single Inlet" is recommended. This means that the wheel is overhung, with the shaft bearings supported in an external bracket mounted on the fan housing. The fan will be belt driven. Information about the fan follows:[(28)]

Model number: Carrier Model 27 CN, Size 300 with inlet vanes
Capacity: 2000 cfm nominal
Motor drive: 5 HP direct current shunt-wound
Fan inlet: 19 in. diameter
Fan outlet: 11-3/4 in. wide x 16-3/8 in. deep
Base to centerline of fan inlet: 19-1/2 in.
Overall width to attached fan pulley: 28-3/4 in.

Overall height of fan: 35-7/8 in.
Position for delivery: counter-clock-wise top horizontal
Construction: arrangement number 2, single width, single inlet
Connections to duct: at site, 4 in. airtight asbestos at inlet and outlet.

Fan Details. Steel bracing is included where required to provide stiffness to the housing and a rigid support for the bearings. The fan has a slip joint inlet and a flanged outlet. The blades are die-formed from alloy steel and are welded to the shroud. The wheels are statically and dynamically balanced. A solid backplate forms a rigid support for the curved wheel blades of the single inlet rotor. Standard practice is to rivet the blades to the backplate, though welding may be specified. On the inlet side of the wheel a large, smoothly curved shroud strengthens the wheel and guides the flow of air within the wheel. An inlet bell of the same curvature as the wheel shroud guides the air into the inlet. Inlet vanes are to be included as an accessory. They impart a spin to the incoming air so that it enters the wheel at the angle of greatest efficiency. Precise volume control is possible without sacrificing fan performance. The pitch adjusting mechanism of the vane assembly is on the inside of the inlet bell out of the air stream. The cantilevered vane shafts are connected to a common control ring to which is attached a single manually controlled operating lever.

Though complete volume control can be had by moving the position of the variable inlet vanes, a direct-motor drive is recommended to allow testing of the fan itself. The motor should be a five horsepower, shunt-wound type with a rheostat-controlled external resistance in series with the shunt field. [29]

I. SECONDARY CIRCUIT COMPONENTS

Considering the well-designed, factory assembled packaged chilling units available, it may seem strange that one was not recommended for the proposed built-up unit. The reason is that none was appropriate since the proposed unit has special requirements. The washer-dehumidifier requires a large flow rate and high head for its tonnage rating, compared with the usual applications for which packaged chillers are designed.

COMPRESSOR

The compressor for the proposed unit must meet several basic requirements. It must be of sturdy construction and completely dependable. The capacity should match the nominal requirements of the conditioning coils and dehumidifying washer. To eliminate leaks, belts, and maintenance the compressor and driving motor should be a hermetically sealed unit. The recommended refrigerant is R-12.

The compressor recommended for the proposed built-up unit is the Carrier Model 6D47. Ratings are based on condenser water and suction gas temperatures in accordance with A.S.R.E. Standard 14-41, which specifies the use of a liquid-suction heat exchanger when using Refrigerant 12 at suction temperatures below 30°. The information which follows is based on a saturated suction temperature of 40°:

Type: semi-hermetic, field serviceable
Evaporator temperature range: high (for air conditioning service)
Refrigerant: R-12
Capacity: 63,600 Btu/hr.
Power input: 4.1 kilowatts
Number of cylinders: 4
Speed: 1750 rpm
Bore and stroke: 2 in. x 1-7/16 in.
Displacement at 1750 rpm: 18.3 cu. ft./min.
Condenser: to be selected separately
Oil charge: 7 pints
Oil gage: bulls-eye type sight glass
Suction gas connection: 1-1/8 in. O.D. female
Discharge gas connection: not given
Capacity control: none
Controls furnished:
(1) Magnetic across-the-line starter with overload protection
(2) Dual pressurestat
(3) Compressor thermal protector
Wiring: 3/60/208-220
Control wiring: 220 volts

The compressor has automotive type pistons, with two compression rings and one oil ring. Positive lubrication of connecting rods, main bearings, and cylinder walls is achieved by a direct drive oil pump which is automatically reversible to provide oil pressure regardless of the direction of the rotation. The suction manifold contains a fine mesh strainer, as does also the lubrication circuit.

The hermetically sealed compressor motor is cooled by the suction vapor and is protected against unusual and prolonged high temperature conditions by a thermal protector on the motor housing. The starting equipment, dual pressurestat, and compressor thermal protector are installed at the factory. A terminal board is provided for field connection of thermostat and the solenoid valve for the refrigerant liquid line.

Ordering specifications: Carrier Model 6D47 compressor for 3 phase, 60 cycle, 208-220 volt service. Using Refrigerant 12 with a suitable condenser having water entering at 75° and leaving at 95°, the compressor must have a capacity not less than 60,000 Btu per hr. at a suction temperature of 40°.

CONDENSER

The compressor selected is normally sold with a shell-and-coil type condenser as a pre-assembled condensing unit. The shell-and-coil type is sealed shut; it cannot be inspected, and can be cleaned only by circulating a chemical solution through the tubes. Because of the importance of having a condenser in which the effect of fouling by water deposits can be measured and studied, a cleanable type shell-and-tube condenser is recommended.

Manufacturers' catalogs rate condensers according to the refrigeration load of the evaporator, with correction factors for various compressor suction and condensing temperatures. The actual heat to be removed by the condenser can be found from page 422 of the 1949 A.S.R.E. Data Book.[57] With a condensing temperature of 102° and a suction temperature of 40° , 228 Btu per min. per ton of refrigerating effect occuring in the evaporator, or a total of 68,400 Btu per hr., must be removed by the condenser.

Selection was made from Acme Catalog 235.[2] It is planned that the existing laboratory circulating pump with constant level tank and cooling tower will be used in operating the proposed unit. The pump has a capacity of approximately 10.3 gpm. To keep within this limitation, a 20° rise through the proposed condenser was used as one of the design factors. Also, the compressor already selected was rated on the basis of water entering the condenser at 75° and leaving at 95°, with a suction temperature of 40° . The condensing temperature will be 102°. On page 2 of Acme Catalog 235, Model J-500 was selected. Information about the condenser follows:

Condenser model number: J-500
Basic rating: 5.5 tons
Number tubes: 24
Number passes: 8
Water pressure drop: 2.6 psi
Total effective surface: 40.7 sq. ft.
Pump down capacity: 65.91 lb. R-12
Outside diameter of shell: 8-1/4 in.
Effective tube length: 41 in.
Overall length: 45-1/2 in.
Water connections: 1 in. F.P.T.
Gas inlet connection: 1-1/8 in. O.D.S.
Liquid outlet connection: 1/2 in. F.P.T.

In nesting the tubes, space is left at the bottom of the condenser for storage space so that a separate receiver will not be required. The pump down capacity is based on 80 per cent of the net contents when the Refrigerant 12 is at 102° liquid temperature.

Straight-through tube construction permits simple mechanical cleaning upon removal of the cast iron water heads. Integral fins on the copper tubes have an outside surface 300 per cent as great as bare tubes, which increases the heat transfer value in nearly direct proportion. The tubes are silver soldered permanently into chemically treated steel tube sheets which are welded to the condenser shell. Access space should be left at one end of the condenser for replacing tubes when necessary.

Ordering specifications: The condenser should be equal to Acme Model J-500. It should have the capacity to remove not less than 68,400 Btu per hr., or a basic rating of 5 tons under the following conditions: water entering at 75° and leaving at 95°, 102° condensing temperature, and 40° compressor suction temperature.

WATER CHILLER

The basic requirement for the chiller selected was that it serve the high rate of flow needed for the washer circuit without excessive friction drop in the water circuit, and yet have ample refrigerating capacity for the low flow when serving the chilled water coil. In addition it should be of high quality and of accessible construction to facilitate experimental research.

The chiller was selected from Acme Catalog No. 600-A.[1] The chiller is of the direct expansion, "dry," type rather than the flooded type. The tubes do not have the star-shaped aluminum insert. The insert gives more heat transfer per unit length of tube than the plain tube, but the resulting shorter chiller length would cause an excessive friction drop in the water circuit.

The Acme chiller has straight-through tubes with no bends or joints. The refrigerant space is in the tubes and heads; the water space is in the shell around the tubes. With multi-pass heads at each end of the tubes, pressure drop is minimized. The velocity of the water circulating within the shell and over the outside of the tubes is controlled by one of several standard baffle spacings to produce the most efficient heat transfer. Baffles cut off short and widely spaced give low pressure drop for systems having high flow rates; fuller baffles, closely spaced, increase fluid velocity for maximum heat transfer in systems having low flow rates. The designation used to indicate the order of increasing baffle spacing is K, L, and M. Chillers with K spacing have the closest baffle spacing and the greatest pressure drop for a given rate of flow.

Positive oil return is assured because there are no bends or joints in the tubes to obstruct refrigerant flow. There are no soldered joints, minimizing the danger of refrigerant leakage. The straight-through, full-size tubes can be mechanically cleaned by removing the heads. If ruptured by accidental freeze-up, the tubes are replaceable. Sufficient areaway must be open at either end of the chiller to facilitate this removal.

Selection Procedure. To select the proper chiller the procedure outlined in the Acme catalog was followed precisely. To do this the logarithmic mean temperature difference between the water and the refrigerant was determined. The temperature of the refrigerant will be constant; allowing for the usual 2° rise in the piping between the evaporator, or chiller, and the compressor suction due to pressure drop, the evaporating temperature at the chiller was taken as 42°. To supply 60,000 Btu per hr. to the washer with 30 gpm will require a temperature drop of 4° for the water entering the chiller. Chilled water will leave the chiller at 48°.

$$\text{L.M.T.D.} = \frac{\text{G.T.D.} - \text{L.T.D.}}{\log_e \frac{\text{G.T.D.}}{\text{L.T.D.}}}$$

Where:

L.M.T.D. = logarithmic mean temp. difference, °F

G.T.D. = greatest temp. difference; or inlet water temp. minus evaporating temp., °F

L.T.D. = least temperature difference, or outlet water temp. minus evaporating temp., ° F

$$\text{L.M.T.D.} = \frac{(52 - 42) - (48 - 42)}{\log_e \frac{10}{6}}$$

$$\text{L.M.T.D.} = \frac{4}{\log_e 1.667} = 7.85$$

Refrigerant 12 will be used. On page 13 of Acme Catalog 600-A the 8L type chiller was selected; it has an 8 in. diameter shell and moderate baffle spacing. From the rating chart, at 30 gpm and a mean effective temperature difference of 7.85, the chiller shows a loading of approximately 800 Btu/(hr.)(sq. ft.). The surface required in sq. ft. is calculated as follows:

$$\text{Surface required, sq. ft.} = \frac{\text{Capacity, Btu/hr.}}{\text{Loading, Btu/(hr.)(sq.ft.)}}$$

$$\text{Surface required} = \frac{60,000}{800} = 75.0 \text{ sq. ft.}$$

Page 12 shows that Acme Model No. DXH-809 will handle the requirements, having 76 sq. ft. of effective tube area. From the rating chart, the water pressure drop obtained at 30 gpm flow is 0.64 ft. head per ft. of shell length, or a total of 5.76 ft.

In checking chiller performance when using the water for the chilled water coil, the chiller will deliver the five tons capacity so long as the flow rate does not fall below 23 gpm. This will require a temperature drop of 5°. Any chiller installed in the proposed unit should have the ability to meet specifications for both washer and chilled water coil.

Other information about the chiller follows:

Minimum rated flow capacity: approximately 17 gpm
Maximum rated flow capacity: approximately 56 gpm

Number of refrigerant circuits: 1
Number of refrigerant tubes: 44
Minimum Refrigerant 12 charge: 27 lb.
Volume of Shell space: 15 gallons
Total operating weight with water: 710 lb.
Length from tube sheet to tube sheet: 8 ft. 11-3/4 in.
Overall length: 9 ft. 10-3/4 in.
Outside diameter of shell: 8-5/8 in.
Inside diameter of shell: 8 in.
Number passes of refrigerant: 6
Suction connection: 2 in. I.P.S.
Liquid connection: 3/4 in. I.P.S.
Water connections: 2 in. F.P.T.

It should be noted that several limitations governed the above selection. The thermostatic expansion valve must be set for 8° superheat. The entering temperature difference must be 8° or higher, and the leaving temperature difference must be 5° or higher (in the above selection they were 10° and 6° respectively); the mean effective temperature difference must be at least 7°; and the refrigerant temperature must never be allowed to drop below 30°.

Construction Details. The tubes are seamless copper 3/4 in. O.D. and the shell is seamless steel. The heads are cast steel and the tube sheets are flange quality carbon steel. The water baffles are hot rolled steel, terneplate, which indicates an alloy coating of approximately 4 parts lead to 1 part tin. The design working pressures on both tube and shell side are 150 psig, with test pressures twice as great. Though Acme is using 3/4 in. flexible foamed plastic of the closed cell type for new packaged chillers, for the purposes of the proposed unit a more durable covering should be used, which Acme furnishes. Saddle brackets are furnished with the chiller to fit the insulated covering, when the insulation is ordered initally. The chiller head has a connection for the external equalizer line of the thermostatic expansion valve.

Accessories. The chiller requires one thermostatic expansion valve, with external equalizer connected on the chiller head where provided. Upstream from the expansion valve in series should be placed a Sporlan "See-All" flow and moisture indicator, solenoid valve, filter-dryer, and refrigerant shut-off valve.

The refrigerant solenoid valve should be controlled by the temperature of the water leaving the chiller, to maintain it always at a constant setting. Depending on the amount of water being circulated, the return water temperature to the chiller will change. To insure positive protection against freeze-up, an immersion thermostat should be placed in the chiller near the point at which the water leaves. The thermostat customarily shuts down the compressor when the water temperature falls to 36°.[67] Another device which helps prevent freezing of the chiller is the normal cycling of the low-pressure cutout on the compressor.

Though it is best when feasible to provide full flow of water through the chiller at all times to give further assurance against freezing, it is not possible in the proposed unit. Full pressure at the washer is necessary for proper atomization at the spray nozzles, yet the water temperature must be modulated; so a mixing valve is used which bypasses part of the water through the heater, which is off-cycle. It is necessary to watch the operation of the chiller controls closely during periods of low dehumidifying load at the washer since the flow through the chiller will be reduced as the three-way mixing valve bypasses more water around the chiller.

Ordering specifications: Acme Chiller Model No. DXH-809. The chiller shell shall be complete with not less than 3 in. of rock cork laid in hot asphalt and covered with a 20 gage steel jacket; insulation head covers shall be made of 20 gage steel filled with commercial rock wool. With a refrigerant temperature of 42^{o} the chiller shall be able to cool 30 gpm of water 4^{o} when entering at 52^{o}, and to cool 23 gpm of water 5^{o} when entering at 53^{o}.

WATER HEATER

The water heater was selected from the Taco catalog, "Commercial and Industrial Heat Exchangers."[66] The proposed heater, designed to operate on slightly superheated 5 psig steam, must have sufficient capacity to transfer not less than 233,000 Btu per hr. to the air heating coil and to the washer.

The design of the built-up unit hot water coil was based on water entering at 180° and leaving at 160°. On page 6 of the Taco catalog are capacity ratings for the "RS" series of steam-to-water heaters which operate on 5 psig steam pressure with a 20° rise on the water side. The capacity rating of a given converter decreases as the supply water temperatures increases since the steam-water temperature differential becomes less. The best selection is the RS-632, at 340,000 Btu per hr. Other information about the converter follows:

Model number: RS-632
Steam supply pressure: 5 psig
Tubes: 3/4 in. copper of U-bend construction
Water connections: 2 in. F.P.T.
Steam inlet connection: 4 in. flanged
Condenste connection: 1-1/2 in F.P.T.
Outside diameter of shell: 6-5/8 in.
Overall length of converter: 41-7/8 in.

The pressure drop for the rated capacity is given as 1.0 psi. But only 24 gpm need to be circulated to deliver the required load, so the above pressure drop is high. The actual pressure drop, however, is unimportant so long as it is less than that through the chiller in the parallel water circuit. To maintain accurate control of flow by means of the mixing valve on the summer cycle, a balancing cock should be placed at the heater inlet and adjusted to balance the pressure drop through the chiller circuit. On the winter cycle it would not be good practice to bypass hot water through the chiller. Therefore, a bypass line with stop valve and with balancing cock should be installed in parallel with the chiller.

The converter should have a float and thermostatic trap in the condensate line. In the water circuit beyond the mixing valve should be a compression tank with a Bell & Gosset "airtrol" fitting. Since the compression tank is the "point of no pressure change" in the system, the tank should be located above the hot water coil with connecting piping pitched at least 1/8 in. per ft. This will put it on the suction side of the pump, and insure that a positive pressure exists at all times on the components of the water circuit. The tank will be isolated from the washer circuit when the hot water supply and return valves are closed. The compression tank should have a sight glass to indicate water level. Make-up water may be taken from the washer tank when needed. The pressure relief valve should be located at the converter discharge, with no valves between the relief valve and the tank itself.

Ordering specifications: Taco RS-632 Converter. The steam-to-water heat exchanger should have sufficient capacity to heat 24 gpm of water from 160°to 180°, or a minimum of 233,000 Btu per hr.

CIRCULATING PUMP

Pumps are selected on the basis of the quantity of water to be handled and the total head required. The standard procedure[29] for determining the total head, including friction losses and vertical lift, is as follows:

(1) Decide on the pipe circuit water velocity, not exceeding 10 fps, from tables which give the unit pressure loss for any flow rate and pipe size selected.

(2) From a scale drawing, make a summation of all friction losses in pipes, fittings, and valves.

(3) Determine the head loss of sprays and loss through other equipment; if more than one circuit is being used, find the loss through the circuit with maximum head loss.

(4) Determine the net vertical distance the water must be lifted, whether on pump discharge or suction.

(5) Summarize all values as total feet of water head required of the pump for a specific flow rate.

For the proposed built-up unit, the pump will supply water to the washer circuit which contains the chiller and mixing valve; and also the hot and chilled water coil, which contains the chiller and mixing valve. The chiller and heater are in parallel, but the chiller has by far the greatest friction drop. On cooling cycles, the circuit through the water heater will act as bypass for the chiller; andon heating cycles, a separate line will act as bypass for the heater. The mixing valve is at the outlet of the heat exchangers and acts to control the amount of flow through either circuit. A balancing cock will be placed at the inlet to the water heater so that the pressure drop will be the same across the heater and chiller circuits. The pressure drop across the mixing valve at any position will be equal to the pressure drop across the chiller, and the heater. This is a basic principle of control, as given in the Engineering Manual of Automatic Control, page 6-20.(33) If the valve drop is much smaller than through the chiller, effective control will be obtained only when the disc is near one or the other of the two seats. The mid-portion of the valve lift will be relatively ineffective.

Because of the atomizing spray nozzles, the washer circuit will have a greater friction drop than the water coil. The flow rate through the washer circuit was found to be 30 gpm. The net vertical distance through which the water must be lifted is only about 0.75 ft. From the friction chart on page 229 of Modern Air Conditioning, Heating, and Ventilating,(29) using a 2 in. nominal size steel pipe will give a velocity just under 3 ft. per sec., which is satisfactory. The friction loss is 3.4 ft. of water per 100 ft. of length. The chart is based on seamless steel pipe in open water circuits. Page 232 gives the friction loss in pipe fittings in terms of equivalent feet of straight pipe. Tabulation of the estimated total required head is as follows:

Equivalent straight length of fittings:	
2" elbows, 5.2 x 9	46.8
2" gate valves, full open, 1.1 x 4	4.4
Estimated length of 2" pipe	32.0
Estimated total equivalent length, feet	83.2

(3/4 ft. loss/100 ft.) x 83.2 ft. = 2.8 ft.

Friction head drop through piping and fittings	2.8
Friction head drop through chiller	5.8
Friction head drop through mixing valve	5.8
Friction head drop through spray nozzles	57.8
Net static head drop	0.8
Total required head in ft. of water gage	73.0

When the pump is serving the hot and chilled water coil instead of the washer, the flow can be throttled down to the required amount.

The pump selected was the Buffalo Forge "1½ CCL." The code of the series designates the pump to be close-coupled with a large discharge head. It has a single suction and is bronze-fitted. At 30 gpm it is rated at 70 ft. of head. Since page 10 of Bulletin 975-F shows that 1.33 horsepower will be required, the manufacturer recommends a 1-1/2 horsepower, 3-phase, 60-cycle, 208-220/440 volt, 1750 rpm open type motor. It should be remembered that the estimated head required for the pump should be checked when the final layout is made to be sure that the head rating of the pump is not exceeded. Though gate valves are generally installed on the suction and discharge, it is not absolutely necessary in the proposed system because of the small size of the circuit and the positive head on the pump suction.

Ordering specifications: Buffalo Forge Pump 1½ CCL with 1-1/2 hp, 3/60/208-220, 1750 rpm open type motor. The pump shall be able to deliver 30 gpm at a total head of 70 feet.

ACCESSORIES

Thermostatic Expansion Valve. A Sporlan thermostatic expansion valve was selected from catalog 56-C.[64] To insure obtaining full capacity from the valve, a liquid line heat exchanger will be installed. The other components of the refrigerant circuit were checked to ascertain whether or not the net pressure drop across the valve port exceeded 60 psi. Since it did not, there will be no reduction from the capacity listed in the selection chart. The valve is fitted with an external equalizer which should be mounted close by the remote bulb of the valve. Theoretically, only one valve is required to serve both the chiller and the direct expansion coil. However, it is recommended that a valve be installed in each place to simulate the usual type of installation encountered.

Ordering specifications: Sporlan Thermostatic Expansion Valve, Type PFE-6-G with 1/4 in. port size, 5/8 in.

O.D.F. inlet connection, and 7/8 in. O.D.F. outlet connection. The length of capillary tubing normally furnished is 5 ft., but may be specified longer.

Heat Exchanger. A heat exchanger was selected from page 5 of Acme Catalog No. 300-A.[4] It will be used to increase system efficiency by subcooling the liquid refrigerant from the system condenser. Besides preventing flash gas at the thermostatic expansion valve, subcooling increases the cycle efficiency. Superheating suction vapor prevents liquid carry over to the compressor. Liquid flows through the tubes, which have an extended bar-type fin surface, and vapor flows through the shell. The external diameter is 4-1/2 in. and the overall length 31-1/2 in.

In normal operation there will be approximately 60° difference between the entering liquid and the entering gas. This will add 17° superheat to the gas and subcool the liquid 10°. The suction side pressure drop will be only 0.095 psi.

Ordering specifications: Acme Model HX5 Suction Line Heat Exchanger. The liquid connections are 5/8 in. O.D.S. and the suction gas connections are 1-3/8 in. O.D.S.

Refrigerant Filter-Drier. The recommended filter-drier for the proposed unit is the Sporlan "Catch-All."[64] It should be placed in the liquid line upstream from the solenoid valve. The main purpose is to trap foreign particles in the system that might obstruct the refrigerant control valves and damage the compressor. The secondary purpose is to absorb any moisture circulating with the refrigerant. Moisture will freeze at the thermostatic expansion valve. Moisture also reacts chemically with the lubricating oil and refrigerant to form acids; once started, the action is progressive, and causes corrosion to internal surfaces. The desiccant type granules prevent the passage of particles nine microns or over in size.

Ordering specification: Sporlan Catch-All Filter-Drier, Type C-305-S with 5/8 in. O.D.F. solder connections. The unit is 3 in. in diameter; 9-1/4 in. in length; the effective volume of desiccant is 30 cu. in.; and the surface filtering area is 53 sq. in.

Strainer. As discussed in the Review of Literature, regardless of any other strainers in the circuit, immediately preceding the solenoid valve should be an 80-mesh strainer. A Sporlan strainer was selected.[64] The screen is seam welded and the ends reinforced or capped with a metal stamping which prevents splitting. The removable monel plain woven wire cloth is contained in a brass shell with steel cap and flange. External parts are cadmium plated. Overall length is 5-5/8 in. There is negligible pressure drop.

Ordering specifications: Sporlan Straight Line Strainer, Type No. 2005, with 5/8 in. O.D.F. connections. 80-mesh with 12 sq. in. screen area.

Solenoid Valve. A Sporlan solenoid valve[64] was selected of the pilot operated, normally-closed type, in which the valve remains in the closed position when the coil is de-energized. The valve must be installed upright in a horizontal line since the weight of the plunger assists in closing the valve. Any voltage may be specified for the coil.

Ordering specifications: Sporlan Solenoid Valve Type 73S having a 3/8 in. orifice and 5/8 in. O.D.F. — 7/8 in. O.D.M. connections. At 100 per cent rated voltage the maximum operating pressure differential is 300 psi.

Flow and Moisture Indicator. A Sporlan "See-All" flow and moisture indicator with 5/8 in. O.D.F. connections should be installed in the liquid line. The moisture-sensitive pad behind the glass changes color when touched by traces of moisture; this device is a patented feature not yet available from any other manufacturer.

Condensate Coolers. The required three condensate coolers were selected from page 32 of the Taco catalog.[66] Since the tempering coil, preheater coil, and reheater coil all may be in service at the same time, each will need a condensate cooler. The Taco cooler has a steel cylindrical shell with 3/4 in. copper U-bend tube bundles. The single head is cast iron. Condensate is cooled from 200° to 100°

using two pounds of cooling water for each pound of condensate. Selection of the coolers was based on a cooling water temperature of 70°. The tempering and reheater coils will each use the smallest size, which has a shell 4-1/2 in. in diameter and an overall length of 29-1/2 in. The preheater coil will use the next to smallest size, which has the same diameter with an overall length of 41-1/2 in.

The condensate enters the shell on top. The cooling water exit piping should loop up in order to keep the tubes always filled with water. The condensate drain line should make an inverted U-turn before draining into the respective weigh tank, to keep the shell sealed with water. The inverted U should not extend higher than the cooler, however, since an air vent is fitted on top. Water from the existing laboratory circulating pump and constant-level tank should be used for the coolers. It will have sufficient capacity to serve the condenser as well as the coolers.

Ordering specifications: Two Taco Condensate Coolers, No. CC422 and one Taco No. CC432. At the rate of two pounds of 70° cooling water for each pound of condensate the coolers shall have the capacity to cool 81 lb/hr, 102 lb/hr, and 199 lb/hr respectively.

V. DESIGN OF THE AUTOMATIC CONTROL SYSTEM

The proposed air conditioning unit could operate in all the cycles outlined without automatic control. But it is evident that automatic controls have become an indispensable and vital adjunct of air conditioning units. For this reason, they certainly should not be omitted from the proposed unit, especially since educational considerations are paramount. Not only can the students observe the elements of a control system in actual operation in a multitude of winter and summer cycles, but they can actually touch controllers and actuators and watch the controlling pressures change, as well as study the technique of a job well laid out. First impressions are lasting impressions.

When doing research in the controlled climate test chamber or when performing tests on individual conditioning components, the automatic controls will provide precise accuracy even when unattended and insure reproducible experimental conditions.

Standard safety devices which would have been included in any case, such as the compressor automatic high pressure cut-out switch, will be described in a later section.

Immediately preceding the description and illustration of each main cycle to be demonstrated by the unit is a compilation of basic definitions in pneumatic control terminology.

A. PNEUMATIC CONTROL TERMINOLOGY

Common terms applicable to pneumatic control systems will be defined.[37]

Mains: air lines carrying air at a constant supply pressure, usually 15 to 20 psig, to furnish the power for actuating valves, motors, and dampers.

Branch lines: air lines in which a varying air pressure is maintained by the action of the controllers in order to position valves, dampers, and similar devices.

Control point: that degree of temperature, relative humidity, or pressure which it is desired to maintain, and at which the controller is set. Also called "set point."

Proportioning control, or graduate action: a mode of control in which the valve or damper may assume any position, from fully closed to fully open, which corresponds to the temperature, relative humidity, or pressure measured by the controller.

Compensated control: a method of control in which the condition maintained at one point automatically changes as conditions at another point change.

Lag: the period of delay occurring between one action and that which follows in the chain of events comprising the sequence of operation of an automatic-control system. "Over-all lag" is the time required for a change in conditions at one point to become apparent or cause an effect at another point.

<u>Droop or drift</u>: a persistent deviation of the controlled condition from the set point, resulting from a change in load conditions.

<u>Pull-through</u>: a system arranged so that the air passes through the conditioning apparatus before entering the inlet of the fan.

<u>Reset controls</u>: devices for maintaining a pressure or temperature in the heating or cooling medium which is automatically changed as some other condition changes; that is, the controller is automatically reset to a different control point.

<u>Control hunt</u>: a condition of extreme instability, in which the controlled condition changes in a rhythmic manner from a valve much larger to one much smaller than that desired, as the controller overcorrects in each direction.

<u>Two-position control</u>: a mode of control in which the heating or cooling equipment is either on or off, with no graduate action.

CONTROLLERS

<u>Controllers</u>: control devices which measure changes in temperature, relative humidity, or pressure and initiate action by other units, or actuators, in the system to counteract these changes. The controller is supplied with air pressure from a main, and delivers a pressure to the

branch according to the action needed to control the measured condition.

Direct-acting controller: a device which increases air pressure on an increase in the controlled variable.

Humidity controller: a controller which measures and controls the relative humidity of air in a space or duct. "Humidostat" and "Humidistat" are patented names.

Insertion thermostat, or insertion humidostat: controller with extended element which can be inserted into a duct or other enclosure within which the temperature or relative humidity is to be controlled.

Limit control: controllers employed in the control system so as to keep the temperature, pressure, or relative humidity in the duct within some limit determined by considerations of safety, comfort, or economy. A high-limit control prevents operation of equipment when it would cause dangerous or undesirably high temperature, pressure, or relative humidity. A low-limit control supplements the action of other controllers to keep the temperature, pressure, or relative humidity in the duct above a minimum value; or prevents circulation of air in the system until its condition is at a selected minimum level.

Master controller: a controller which measures conditions at one point and resets the control point of another, or submaster, controller.

<u>Remote-bulb thermostat</u>: a thermostat having an insertion element in the form of a liquid- or vapor-filled bulb connected by flexible capillary tubing to a bellows or diaphragm in the case.

<u>Reverse-acting controllers</u>: a controller which reduces air pressure on an increase in temperature, pressure, or relative humidity.

<u>Snap-acting controller</u>: device which opens and closes a small air valve with a rapid snap movement.

<u>Summer-winter thermostat</u>: two-temperature thermostat which may be set at the same or two different control points, for control of heating in one season and of cooling in the other, with manual or automatic change-over. It commonly has direct action for heating control and reverse action for cooling control.

<u>Thermostat</u>: a standard term for any temperature controller.

<u>Throttling range</u>: a term commonly used for the differential of a graduate-acting pneumatic controller; also, the change in the controlled condition necessary for the controller to operate the actuator through its full range.

<u>ACTUATORS</u>

<u>Actuator</u>: the final control element which starts or stops or varies the operation of conditioning equipment, in response to a controller; it may be a control valve, damper motor, or a relay.

Control motors: actuators which can move loads, such as dampers, from one definite position to another in response to the controller.

Control valves: power-operated valves; the power units may be control motors, solenoid coil and plunger assemblies, or the like.

Electric-pneumatic relay: a small, electrically operated diverting valve, specifically designed to divert air from one control line to another.

Graduate relay: pneumatic relay capable of maintaining in one air line, usually known as the branch line, an air pressure proportional to the pressure in another line, usually known as the pilot line. They may be direct-acting, increasing branch-line pressure on an increase in pilot-line pressure, or reverse-acting, reducing branch-line pressure on an increase in pilot-line pressure.

Mixing valve: three-way valve connected so that fluid from two inlets is mixed in various proportions before flowing out through the single outlet.

Motorized value: a control valve whose power unit is identical or closely similar to a standard control motor.

Normally closed valve: valve which goes to the closed position and is held there by spring action when pneumatic power is cut off.

Normally open valve: valve which goes to the open position and is held there by spring action when pneumatic power is cut off.

Pneumatic-electric relay: relay which opens and closes an electric circuit in response to a pneumatic pressure change.

Positive-positioning relay: relay to supply to pneumatic motors or control valves whatever air pressure is necessary to drive the actuator to the definite position called for by the controller.

Positive relay: pneumatic relay which translates gradual change in pilot pressure to positive-action, snap-acting change in branch pressure; it is essentially a small, pneumatic diverting valve with snap action.

Relay: electric- or pneumatic-control device for converting one kind or degree of control action to another; for instance, changing direct action to reverse action; switching two or more load circuits in response to a single controller; or coordinating the demands of two or more controllers, so as to produce a single, consistent control action or sequence of actions.

B. PROPOSED OPERATING CYCLES WITH ILLUSTRATIONS

WINTER CYCLES

Cycles of operation which are common in central fan system applications will be described and illustrated. The first one, the simplest and least expensive, is able to control only the dry-bulb temperature. Then a cycle will be described which can add moisture to the air, but cannot control the humidity within close limits. The more complex cycles will show how the temperature and humidity can be controlled within very close limits, even when the sensible heat ratio of the conditioned space changes. The only apparatus in a central fan system designed to operate in freezing air is the tempering coil. Therefore, the tempering coil will be shown in the outside air duct in all cases. To save expense, it is of course possible to forego the tempering coil if the outside air is closed off enough in cold weather to insure that the mixture of outside and return air entering the equipment will always be above the freezing point.

Control of Temperature with Hot Water Coil. The air is heated by the hot water coil. The circulating water is heated in the steam heat exchanger. The pump and fan operate at all times. Changing demands for heat in the conditioned space are met by throttling the steam supply to the heater to change the outlet water temperature.

Automatic Control: The direct-acting space thermostat modulates the supply of steam to the water heater according to space demand for heat. The direct-acting low-limit thermostat in the fan discharge maintains the air supplied to the conditioned space at a predetermined minimum level to avoid cold drafts when the space temperature is satisfied. Wide fluctuations in temperature of the conditioned air are avoided by the low-limit thermostat. No humidity control is provided, which in comfort applications normally would cause some distress. A variation would be to use the steam preheater coil instead of the water coil.

A direct-acting thermostat with remote bulb located in the outside air stream, shielded from radiant heat of the tempering coil, begins to close the steam supply to the tempering coil when the outside air temperature exceeds 35°. At outside air temperatures less than 35° the tempering coil valve is full open, and as a further precaution the coil is of the tube-within-a-tube, non-freeze type.

A modern cycle increasingly applied can be demonstrated as an alternative for the above, and has been included in the proposed unit, as shown on Plate 2. On the summer cycle the thermostat-controlled mixing valve bypasses water through the water heater. On the winter cycle the same automatic controls will modulate the water temperature at the hot water coil. To avoid excessive pressure at the refrigeration chiller, water diverted around the water heater will flow through a pipe bypass fitted with balancing cock.

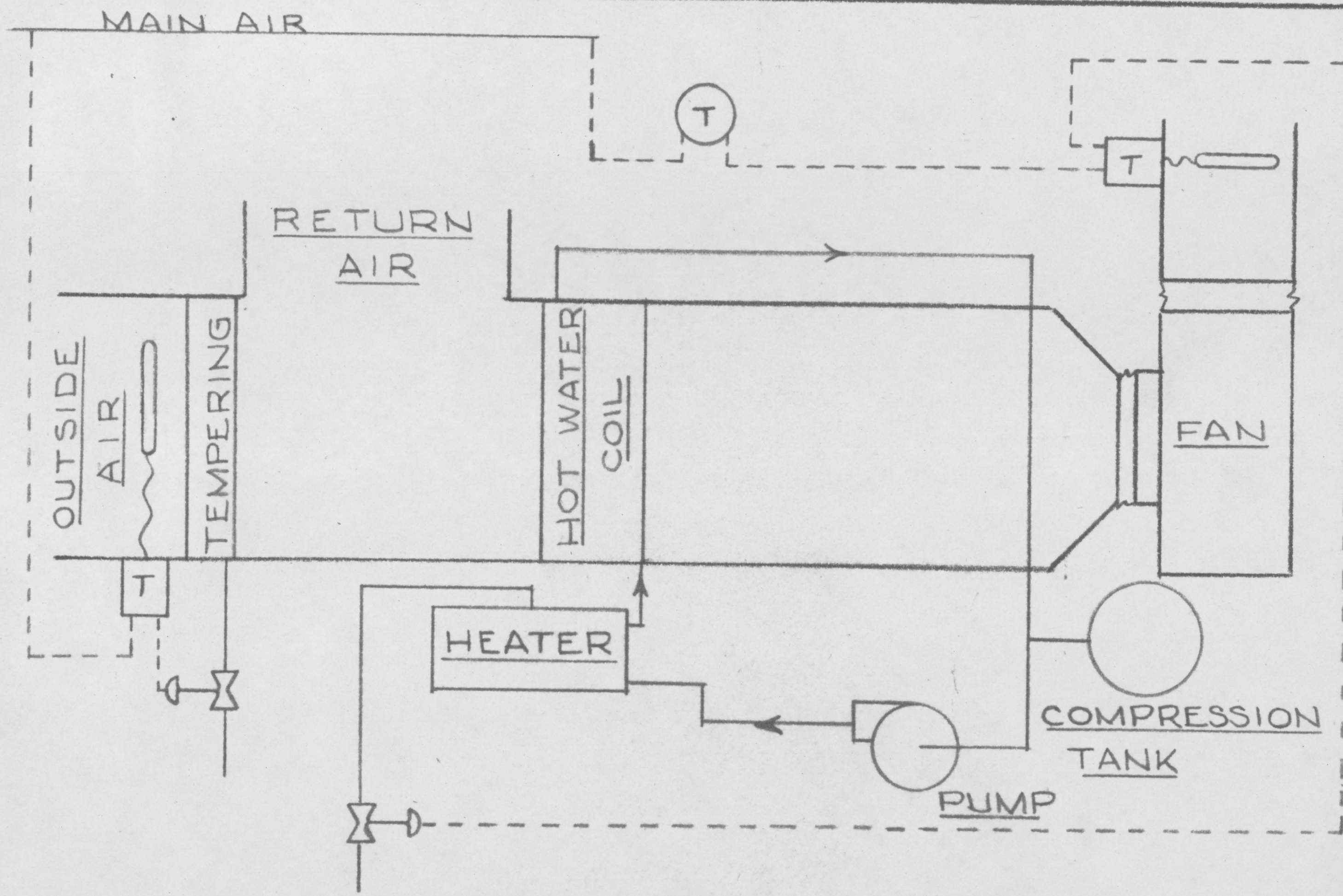


FIGURE 10. CONTROL OF TEMPERATURE WITH HOT WATER COIL

<u>Control of Temperature and Humidity with Heating Coil and Spray Humidifier</u>. The air is heated by the steam preheater coil. Changing demands for heat in the conditioned space are met by throttling the steam supply to the coil. Humidity is added to the air by operating the water pump. Water can be atomized in the spray nozzles, or humidity can be added by letting the flooding nozzles of the washer wet the scrubber plates. The humidifying efficiency should not be greater than about 50 per cent; if the complete washer were used the air would approach the saturation point, which would cause excessive humidity in the conditioned space.

Automatic Control: The direct-acting space thermostat modulates the supply of steam to the steam preheater coil. The circulating water pump, which is operated in an on-off manner through a pneumatic-electric relay by the reverse-acting space humidity controller, supplies the spray humidifier. This system provides good control of heating. The low-limit thermostat in the fan discharge air is not always necessary. For example, it could be omitted if the outside air intake were so small that it would not appreciably affect the mixture temperature. The control of humidity in this system is not precise.

A variation of this system would be to allow the humidity controller to operate a solenoid valve in the city water supply line to the spray humidifier, while the pump delivers water to the hot water coil via the steam water heater. The excess water would be drained to the sewer.

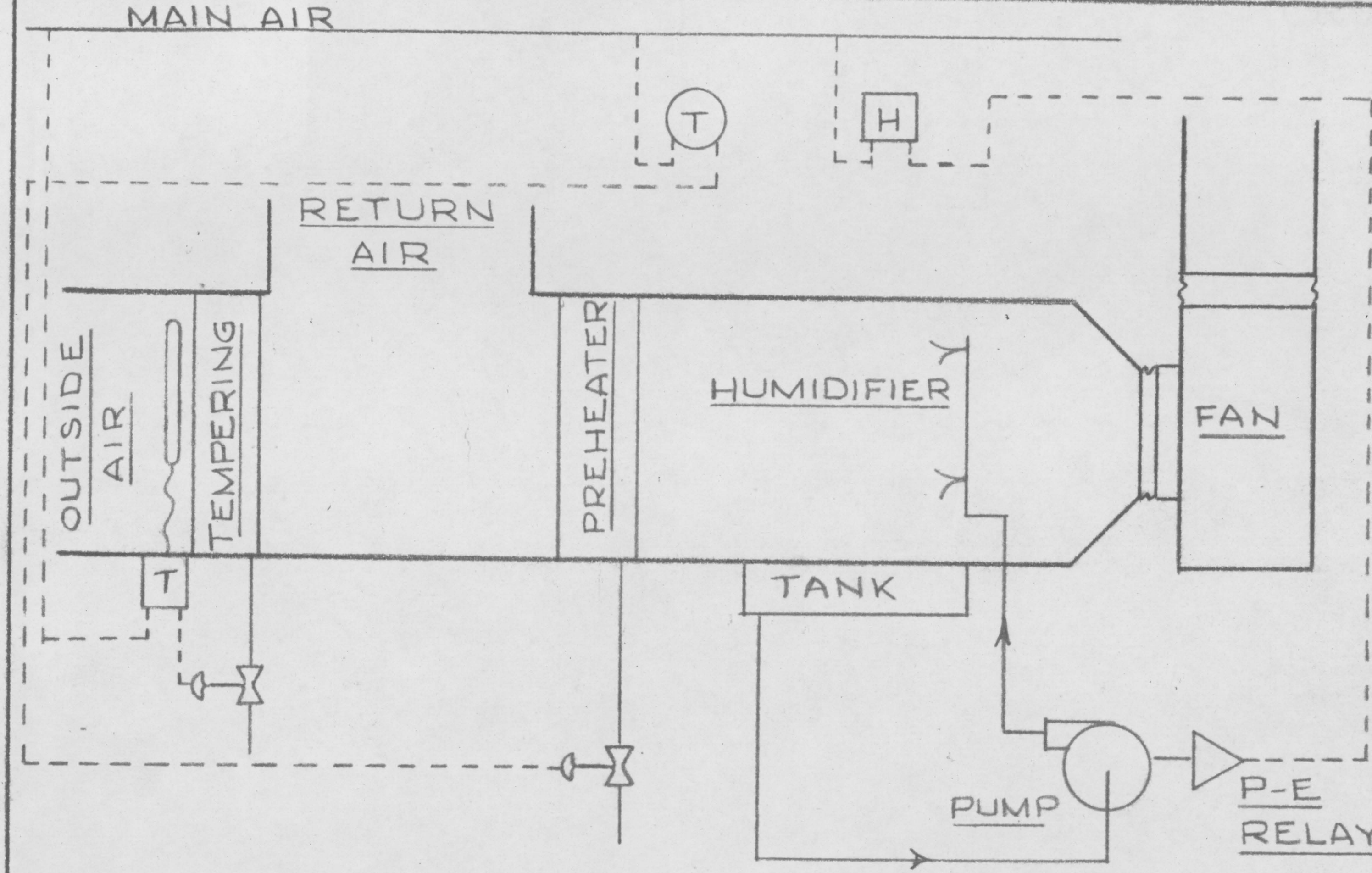


FIGURE 11. CONTROL OF TEMPERATURE AND HUMIDITY WITH HEATING COIL AND SPRAY HUMIDIFIER

Dewpoint Control with Heating Coil and Humidifying Washer. The air is heated enough by the steam preheater coil so that after adiabatic saturation in the washer, it will be at the desired condition. The purpose is to demonstrate "dewpoint control." This is an important cycle for demonstrating the control of a central system which supplies air to various zones, each having its own booster heating coil to furnish the required amount of sensible heat.

Automatic Control: The dewpoint temperature of the conditioned air leaving the washer is maintained at a predetermined setting by a direct-acting dewpoint thermostat, with remote bulb located in the discharge air from the washer, which modulates the steam supply to the heating coil. The washer pump and system fan operate at all times. This system provides good humidity control for commercial multi-zone installations in which each zone has a booster coil controlled by the zone thermostat. For a reasonable supply air differential for the zone having the lowest sensible heat ratio, the required dewpoint temperature would be determined and set. The air capacity for each zone would be designed for the prevailing condition of sensible heat ratio and dewpoint temperature supplied. Changes in sensible heat ratio in any zone would cause the relative humidity to change slightly from the design figure since the original dewpoint setting would not provide the correct amount of moisture. This system would give consistently good control of humidity in industrial applications where the sensible heat ratio as well as the total heat load are steady. In commercial applications, where the sensible heat ratio does change, good control of humidity is obtained by a reverse-acting humidity controller which compensates for change in sensible heat ratio in a representative conditioned space by resetting the control point of the submaster dewpoint thermostat.

With the master humidity controller, this system represents superior control design. It provides excellent control of both temperature and humidity for all conditions of air entering the apparatus normally encountered, and for all changes of sensible heat ratio in the conditioned space.

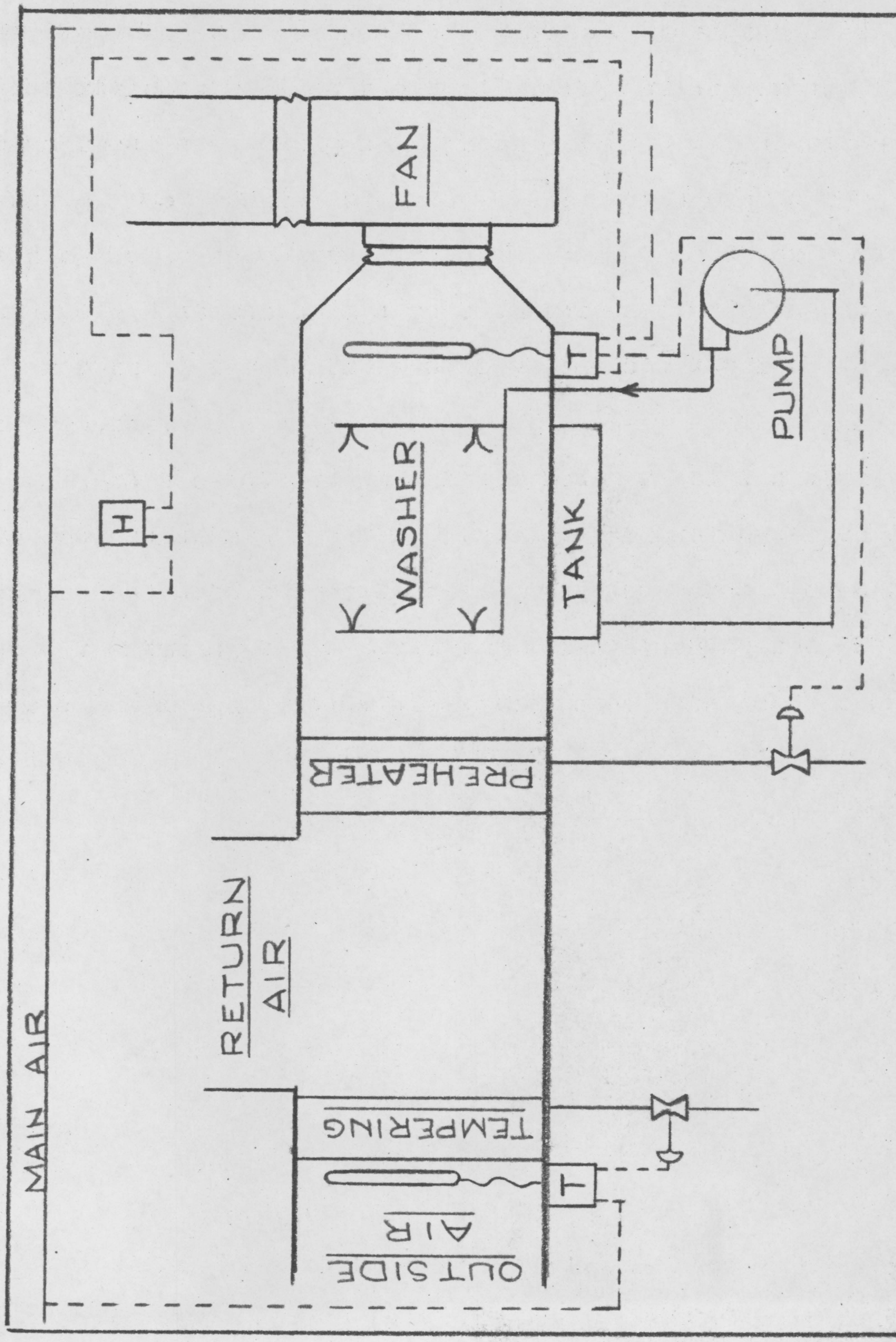


FIGURE 12. DEWPOINT CONTROL WITH HEATING COIL AND HUMIDIFYING WASHER

<u>Control of Temperature and Humidity with Steam Preheater Coil, Humidifying Washer, and Reheater Coil</u>. The air is heated enough by the steam preheater coil so that after adiabatic saturation in the washer, it will be at the desired dewpoint temperature. The dewpoint temperature is roughly taken as the temperature of the air leaving the washer. When enough heat is added to the air leaving the washer to satisfy the conditioned space, the relative humidity of the conditioned space will be correct if the correct dewpoint was chosen. The dewpoint temperature can be raised or lowered by controlling the flow of steam to the preheater coil, and the sensible temperature of the conditioned space can be controlled by varying the flow of steam to the reheater coil. The pump and fan operate at all times.

Automatic Control: The direct-acting space thermostat modulates the steam supply to the reheater coil to maintain space temperature. The dewpoint temperature of the air leaving the washer is maintained at a predetermined setting by a direct-acting dewpoint thermostat, with remote bulb located in the discharge air from the washer, which modulates the steam supply to the preheater coil. This system provides excellent control of temperature at all times. Designed to condition a single zone, the unit provides excellent control of air conditions at the fan discharge. Zone humidity would be under control as long as the sensible heat ratio does not vary, as in industrial applications. In commercial applications, where the sensible heat ratio does change, good control of humidity is obtained by a reverse-acting humidity controller which compensates for change in sensible heat ratio in the conditioned space by resetting the control point of the submaster dewpoint thermostat. A limitation of this cycle is that the desired leaving air dewpoint can be maintained only so long as the entering wet-bulb temperature at the apparatus is lower than the wet-bulb temperature at the desired dewpoint. The exception would occur when the entering air at the apparatus is only moderately chilly but is extremely moist -- a rare condition.

A common variation of this cycle would be to substitute a steam water heater in the washer circuit in place of the preheater coil.

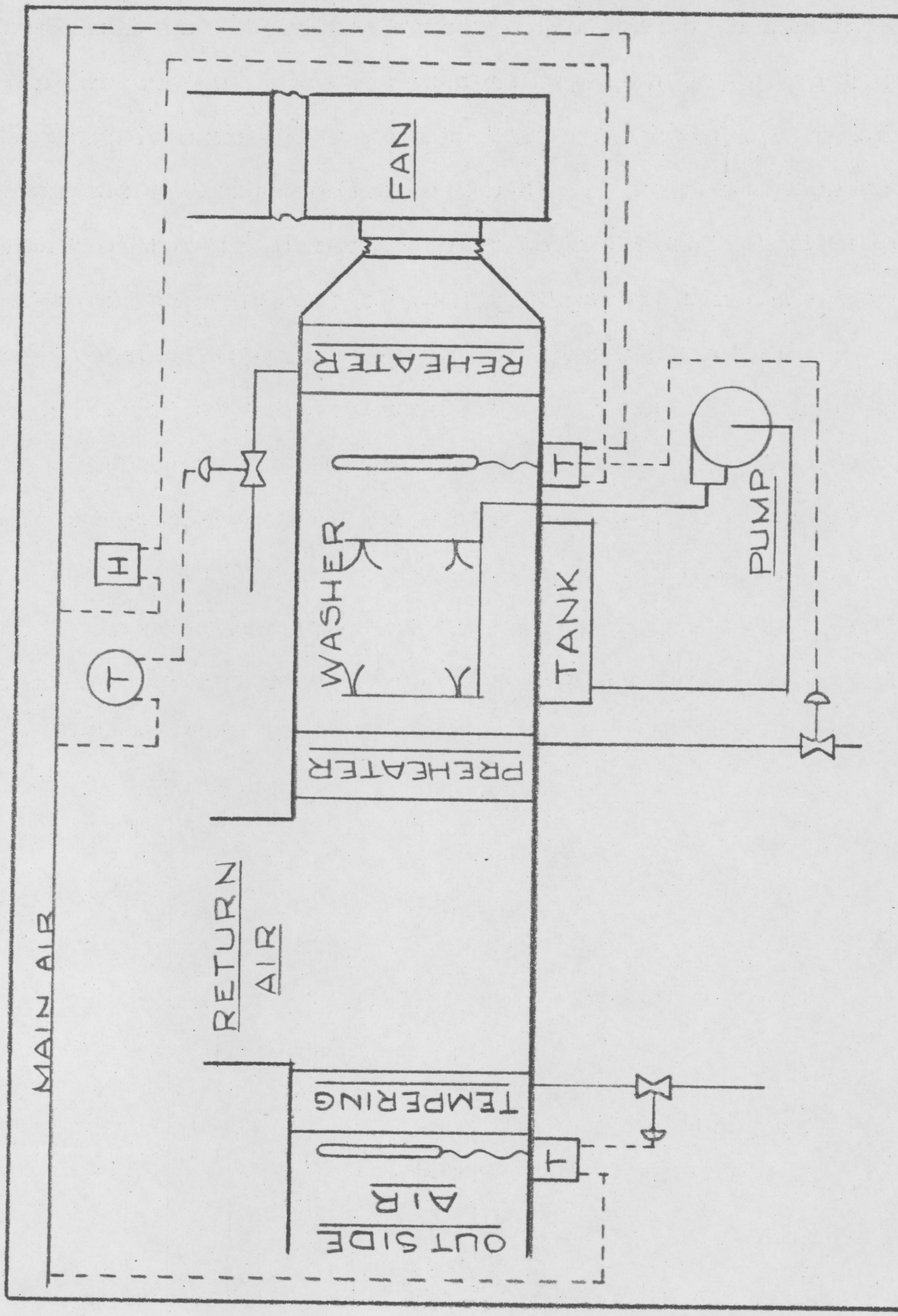


FIGURE 13. CONTROL OF TEMPERATURE AND HUMIDITY WITH STEAM PREHEATER COIL, WASHER, AND REHEATER

Notes on Design. It will have been noted that all of the winter systems use a constant volume of circulated air. This illustrates a good principle of design. Many systems have been installed which maintain a constant temperature of air leaving the conditioning apparatus, and have a modulating volume damper on each zone duct runout. Two serious disadvantages are inherent in a system of this kind:

(1) Since the discharge air temperature is fixed, it is not possible to provide cooling by using outside ventilation.

(2) As the heating load decreases in mild weather, the closing of the dampers at each conditioned space impairs ventilation of those spaces, since the volume of circulating air is progressively reduced.

The central fan tends to build up a higher static pressure as the zone dampers are closing, which raises the air velocity and noise level prohibitively. However, the latter faults are generally attenuated by installing a static pressure regulating damper at the fan inlet.

SUMMER CYCLES

Cyles of operation which are common in central fan system cooling applications will be described and illustrated. The first cycles described are simple and inexpensive means of obtaining control of temperature, but have no positive control over humidity. Some of the cycles which can closely control the temperature can effect dehumidification when desired, or humidification in the case of the evaporative type cooler; but the equipment has no positive control over humidity. Most of the cycles described admit outside ventilating air and mix it with return air before it enters the conditioning equipment. However, no outside air should be admitted on those cycles having face and bypass dampers; otherwise, outside air would be able to enter the conditioned space directly without passing through the dehumidifying coil.

The more complex cycles will show how the temperature and humidity can be controlled within very close limits, even when the sensible heat ratio of the conditioned space changes. The cycles described are basic. Some of the many variations which are in common use and which can be demonstrated with the same equipment will also be mentioned.

Control of Temperature with Evaporative Cooler. Outside air only enters the spray chamber of the washer and is cooled by adiabatic saturation. The purpose of this cycle is to obtain intimate contact between the warm, dry entering air and the recirculating water. The cycle is controlled by operating the circulating pump. Since the cooling process takes place at a constant wet-bulb temperature, the amount of cooling that can be obtained depends on the relative humidity and temperature of the outside entering air, and on the saturating efficiency of the apparatus. Sensible heat of entering air is converted to latent heat as spray water evaporates. All outside air -- no return air -- is used, to preclude the build-up of humidity to an uncomfortable level in the conditioned space. An alternative to the washer for small, compact systems is to combine the spray with a mat-type filter. This system is simple and inexpensive, but for comfort conditioning is applicable only to arid regions with a consistently low wet-bulb temperature. In commercial practice this system is always practical and highly effective in such applications as green houses. Formerly it was widely used in textile mills, where a constant high humidity was the prime consideration. In textile mills the internal latent and sensible loads are constant; therefore, the supply air would maintain the relative humidity of the mill at a constant level while the sensible temperature would fluctuate up and down with the outside wet-bulb temperature. Of course, with a high outside wet-bulb temperature, the combination of heat and humidity in the mill would create an unbearable effective temperature for the personnel.

For experimentation with the equipment, the efficiency of the washer may be measured by using first the flooding nozzles on the scrubber plates only, then the first bank spraying with the direction of air movement, then the second bank spraying against the direction of air movement, and finally the entire assembly in action, which would give maximum efficiency.

Automatic Control: The direct-acting space thermostat controls the washer water pump in an on-off manner by means of a pneumatic-electric relay. The fan operates continuously.

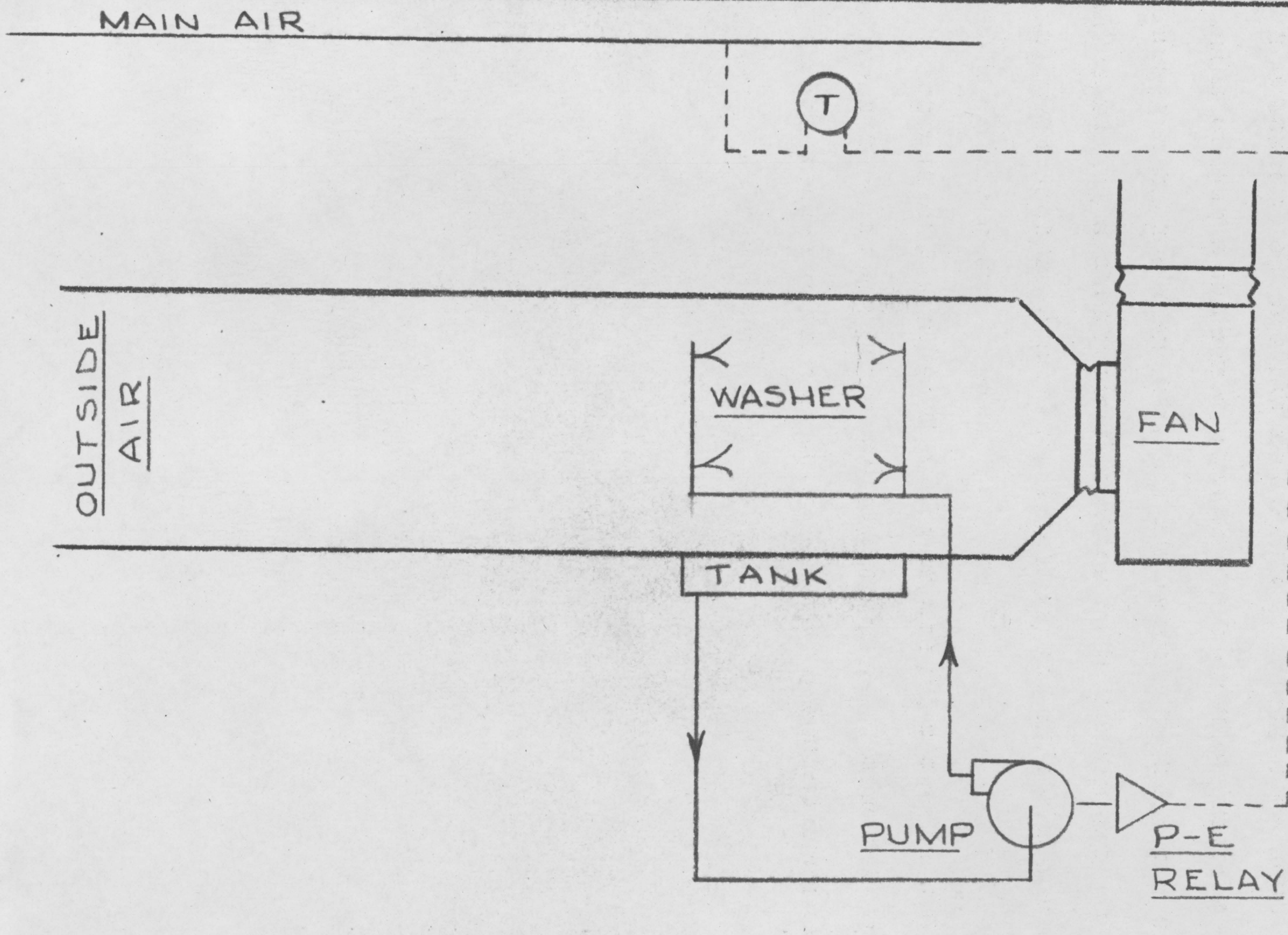


FIGURE 14. CONTROL OF TEMPERATURE WITH EVAPORATIVE COOLER

<u>Control of Temperature with Direct Expansion Coil</u>. The mixture of outside air and return air is cooled and dehumidified by a direct expansion coil. The compressor of the condensing unit is on pump-down control. This system provides no control of humidity. Flow can be throttled with the refrigerant manual shut-off valve.

Automatic Control: The direct-acting space thermostat operates the solenoid valve in the refrigerant line in a two-position manner by means of a pneumatic-electric relay. The compressor of the cycle operates on pump down control; that is, the compressor cycles on and off automatically in response to a pressure-sensitive switch in the suction line to maintain a predetermined range of evaporating pressure corresponding to the desired evaporating temperature. If the system fan is operated continuously, the high moisture content of the ventilating air drawn in and the moisture re-evaporated from the off-cycle coil will cause an excessive build-up of humidity in the conditioned space. For this reason, manufacturers recommend that the fan cycle on and off with the solenoid valve, though the result is a "dead" feeling in the space with consequent stratification of the air during the off cycle.

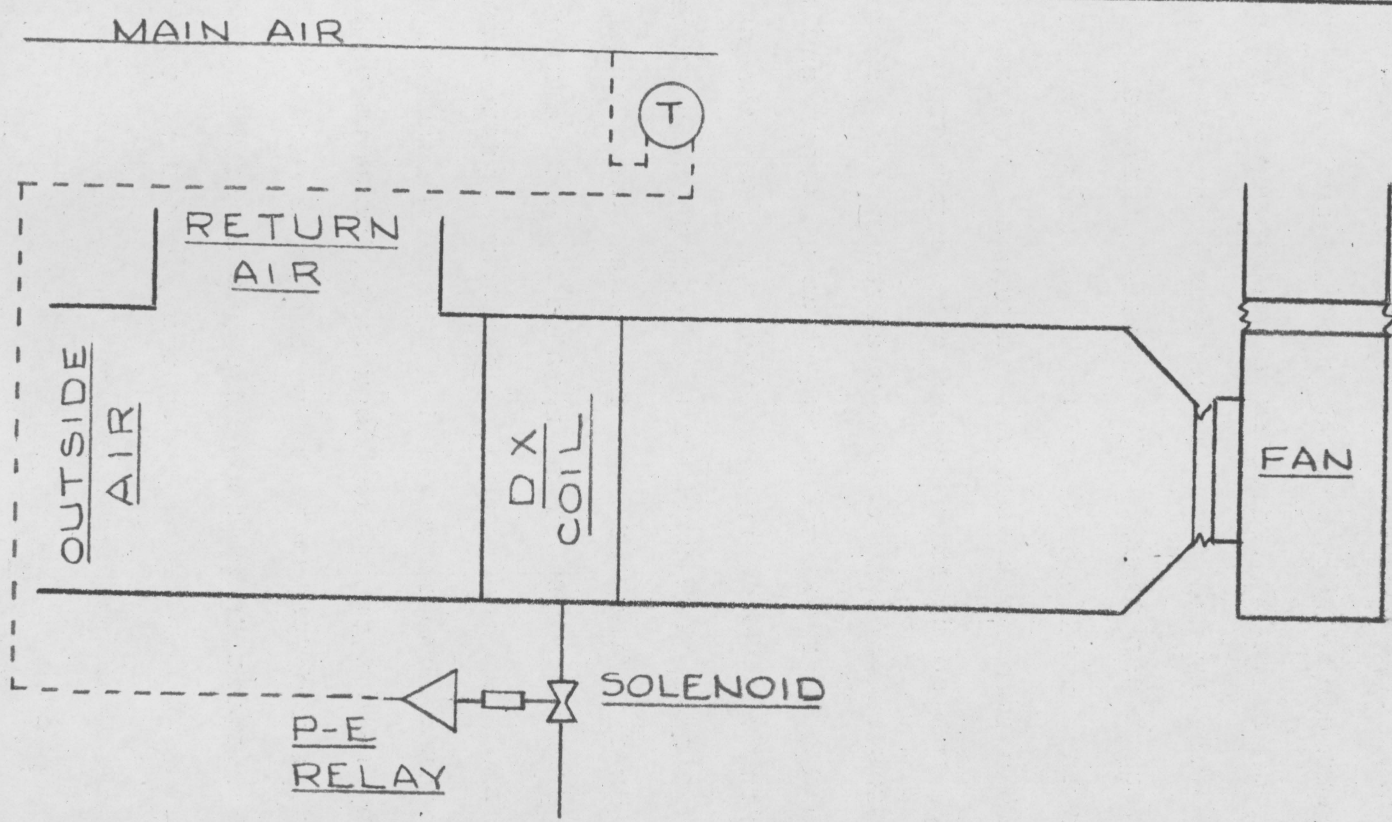


FIGURE 15. CONTROL OF TEMPERATURE WITH DX COIL

Control of Temperature with Chilled Water Coil. The mixture of outside air and return air is cooled and dehumidified by the chilled water coil. The circulating water pump and the system fan operate continuously. Water leaving the chiller is maintained at constant temperature by automatic control: a thermostat with remote bulb in the chiller outlet water opens and closes the solenoid valve admitting refrigerant to the chiller. The compressor operates on pump down control. Part-load coil capacity is obtained by throttling the water to the coil or by mixing the chiller water with bypass water to reduce the temperature of the water entering the coil. This system provides control of temperature only; control of humidity is poor. At part-load capacity, the average water temperature in the coil rises, which gives a higher apparatus dewpoint temperature and less dehumidification capacity. Significantly, comfort applications characteristically show an increase in the proportion of latent heat at part-load conditions. An alternate cycle would be to cycle the pump on and off with the space thermostat, and hold the chilled water temperature constant. For dehumidification the chilled water coil is less efficient than the direct expansion coil because the refrigerant is at a higher temperature, and characteristically shows a temperature rise through the coil. If the system fan is operated continuously, and yet the pump cycles on and off, then the high moisture content of the ventilating outside air drawn in will cause an excessive build-up of humidity in the conditioned space.

Automatic Control: The direct-acting space thermostat modulates the position of a three-way mixing valve at the outlet of the chiller so that the temperature of water supplied to the coil will vary as the space load. To avoid the possibility of freeze-up when most of the water is bypassing the chiller and very little flowing through, the mixing valve in commercial practice would be relocated so that it bypasses the flow of water around the cooling coil and allows full flow at all times through the chiller. There is no difference in performance of the coil, however. The mixing valve is located in the position shown to facilitate demonstration of an important cycle in which modulation of washer water temperature is desired. Though this system provides good control of temperature, control of humidity is poor. At less than full capacity cooling, the supply water temperature is modulated upward, which gives a higher apparatus dewpoint temperature and less dehumidification capacity. In laboratory work, the effect of bypass factor can be demonstrated by varying the number of active rows.

In an alternate system not employing the mixing valve, the direct-acting space thermostat could operate the circulating water pump in an on-off manner by means of a pneumatic-electric relay. The temperature of the chilled water entering the coil will be constant at the temperature preset on the chiller immersion type direct-acting thermostat.

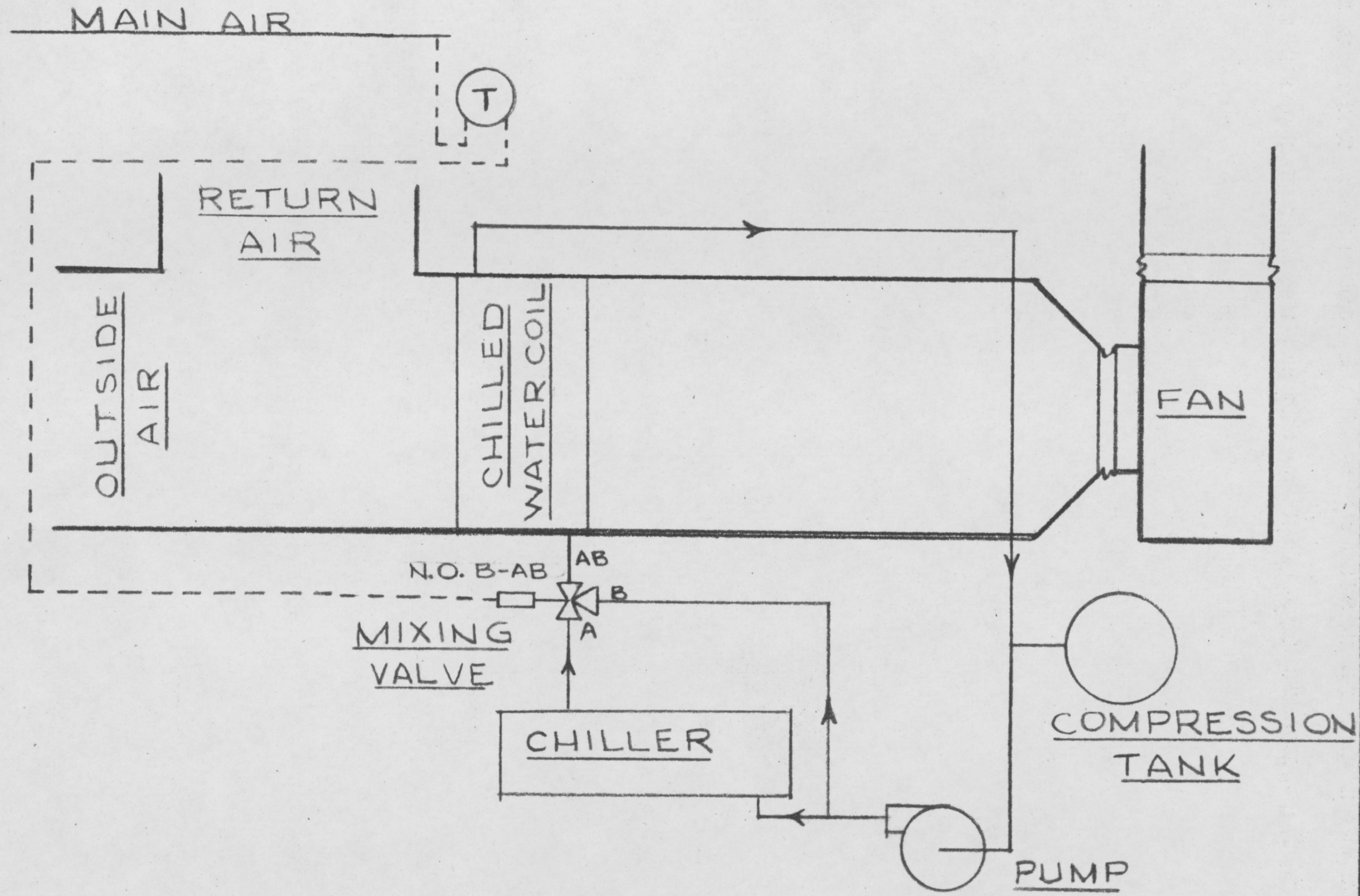


FIGURE 16. CONTROL OF TEMPERATURE WITH CHILLED WATER COIL

<u>Control of Temperature and Approximate Control of Humidity by Direct Expansion Coil and Face and Bypass Dampers</u>. Return air from the conditioned space is cooled and dehumidified by the direct expansion coil. The compressor is on pump down control. Capacity of the coil can be varied by means of the refrigerant shut-off valve, or by operating the face and bypass dampers. Closing in on the coil face dampers opens the bypass dampers a corresponding amount so that a constant quantity of air will be circulated by the fan, which operates continuously. This system gives good control of temperature and fair control of humidity. A characteristic feature of comfort installations is the decrease of sensible heat ratio with decrease in total cooling load; that is, the human load begins to predominate after the sun load goes down. This system provides greater dehumidification of the air at low load conditions than at full capacity. At full load, dehumidification is sacrificed for sensible cooling. The system fan may be operated continuously to prevent stratification of air in the conditioned space, and to give to it a "live" feeling. No outside air should under any circumstances be admitted to the equipment since it would be possible for the high moisture content of the outside air to bypass the dehumidifying coil and enter the conditioned space directly.

An alternate system would be to let the cooling be done by a chilled water coil supplied by a circulating pump which operates at all times.

Automatic Control: In the normal position, the bypass damper is open and the face dampers are closed. Using sequence control, the direct-acting space thermostat operates the solenoid valve in the refrigerant line in a two-position manner by means of a pneumatic-electric relay, and modulates the damper motor. On a demand for cooling at the space thermostat, the solenoid valve is energized and opens, and the damper motor modulates the face dampers toward the wide open position as the bypass damper is closed. When satisfied, the thermostat modulates the damper motor to close the face dampers and open the bypass damper. Sequence control of the solenoid valve insures termination of refrigeration flow to the coil when the face dampers reach the position at which further restriction of the air flow would introduce danger of frost forming on the coil. A positive positioning relay at the damper motor insures that the damper blades assume and hold a predetermined position for each increment of branch line pressure regardless of changes in air pressure on the dampers and mechanical idiosyncrasies of the linkage and bearings.

In an alternate system using a chilled water coil and continuously operating water pump, the damper action is controlled by the direct-acting space thermostat just as with the direct expansion coil. However, danger of frosting of the coil is not present because the refrigerant temperature is always above freezing. Whereas the refrigerant temperature in a direct expansion coil is substantially constant during transfer of heat, the refrigerant temperature in a chilled water coil rises during transfer of heat. Further, the chilled water is usually supplied at a higher temperature. Therefore, the chilled water type of coil shows a lower dehumidifying efficiency than the direct expansion coil.

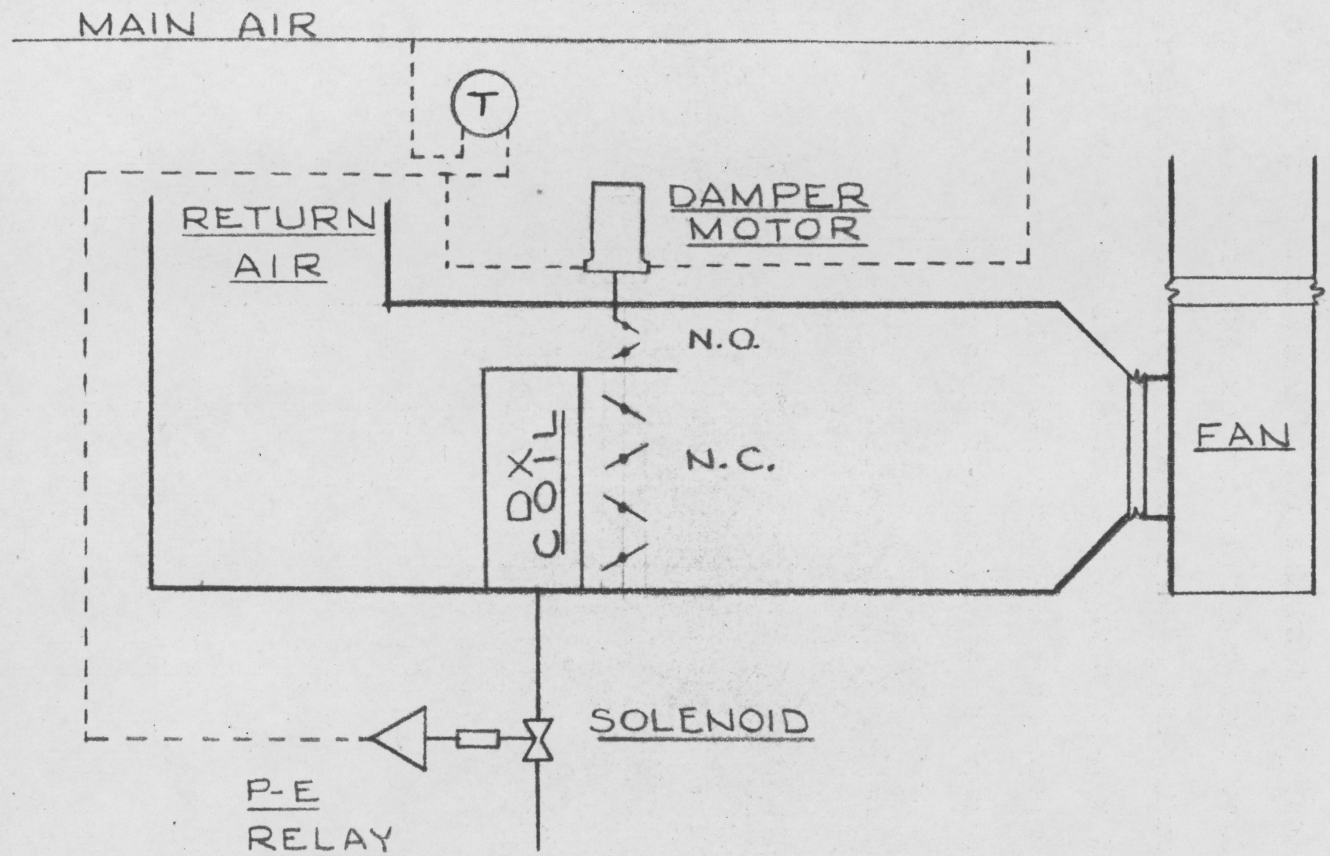


FIGURE 17. CONTROL OF TEMPERATURE AND APPROXIMATE CONTROL OF HUMIDITY BY DX COIL AND DAMPERS

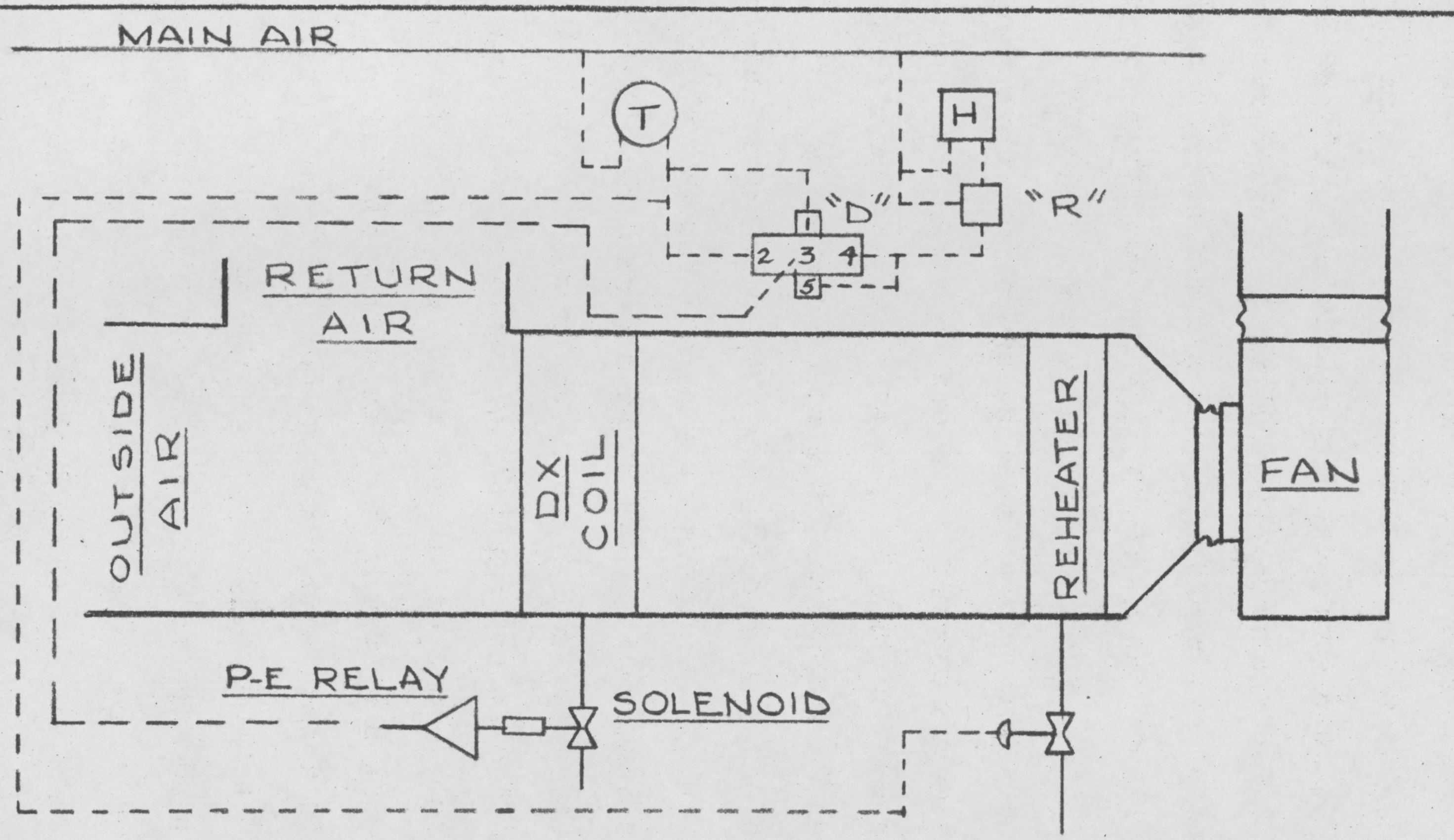
Control of Temperature and Humidity by Direct Expansion Coil and Reheater. The mixture of outside air and return air is cooled and dehumidified by the direct expansion coil. The compressor of the condensing unit is on pump down control. The system fan operates continuously. Since the dehumidication process occasionally removes too much sensible heat, the air must be heated by the steam reheater coil. The capacity of both coils can be manually controlled by shut-off type valves. This system provides good control of temperature and humidity under ordinary comfort conditions. A variation of this system which would operate at less cost would be to provide reheat by means of return air bypassing the cooling coil. Another variation would employ hot discharge vapor from the compressor instead of steam in the reheater coil. Both variations would have a limited capacity, but in commercial practice are worth investigating.

Alternate systems which accomplish the same objective as the direct expansion coil and which can be demonstrated by the proposed built-up unit are the use of a chilled water coil with pump turned on and off, and the use of a chilled water coil with mixing valve and continuously operating pump. Also, the direct expansion coil can be used with the hot water coil acting as the reheater.

Automatic Control: The system using the direct expansion coil will be described first. A diverting relay allows selective control from two master controllers over the refrigerant line solenoid valve. A reversing relay is necessitated in the line between the diverting relay and the reverse-acting humidity controller since all impulses to the relay from the master controllers must be direct-acting. A pneumatic-electric relay at the solenoid valve converts the modulating impulse from the diverting relay to two-position control. On a rise in relative humidity in the conditioned space, the humidity controller energizes the solenoid valve, and provides full capacity dehumidification until the space humidity is satisfied. Temperature control is provided by a direct-acting modulating space thermostat, which modulates the steam supply to the reheater coil when heating is required to balance the excessive cooling necessary to effect dehumidification, or operates the solenoid valve to the direct expansion coil in an on-off manner when the space demands more cooling than is required to effect proper dehumidification.

When using a chilled water coil pump, impulses from the master controllers control cooling and dehumidifying through a diverting relay. A pneumatic-electric relay at the chilled water pump provides two-position control. On a rise in relative humidity in the conditioned space, the humidity controller energizes the chilled water pump and provides full capacity dehumidification until the space humidity is satisfied. Temperature control is provided by a direct-acting modulating space thermostat; it modulates the steam supply valve to the reheater coil when heating is required to balance the excessive cooling necessary to effect dehumidification, or operates the chiller water pump in a two-position manner when the space demands more cooling than is required to effect proper dehumidification.

When a mixing valve is used in the chilled water circuit, the water pump operates continuously and the diverting relay allows selective control from two master controllers over the three-way mixing valve. The normal position of the mixing valve is to bypass all water around the chiller. On a rise in relative humidity in the conditioned space, the humidity controller modulates the mixing valve to permit more flow of water through the chiller, and to permit less to bypass. Advantages of the mixing valve are constant operation of the chilled water circulating pump and modulation of cooling according to space load. Effective dehumidification generally requires modulation to the position of full flow of circulating water through the chiller, to give the minimum coil apparatus dewpoint. The control of humidity is precise because modulation of the mixing valve provides control of dewpoint temperature of air leaving the dehumidifying coil.



"D" ~ DIVERTING RELAY
"R" ~ REVERSING RELAY

FIGURE 18. CONTROL OF TEMPERATURE AND HUMIDITY BY DX COIL AND REHEATER

Control of Dewpoint Temperature with Dehumidifying Washer. The mixture of outside air and return air is chilled and dehumidified by intimate contact with the chilled water spray of the washer. The purpose is to demonstrate dewpoint control. This is an important cycle for demonstrating the control of conditions at a central system which supplies air to various zones, each having its own booster heating coil to furnish the required amount of sensible reheat. Dewpoint temperature leaving the apparatus is controlled by mixing the water leaving the chiller with water bypassed around the chiller to obtain the proper washer temperature. Full flow through the pump is required at all times to get proper atomization of water in the washer spray nozzles. The system fan operates continuously.

Automatic Control: A direct-acting dewpoint thermostat with remote bulb located in the outlet of the dehumidifying washer varies the temperature of supply water to the washer by modulating a three-way mixing valve at the chiller outlet. A reverse-acting humidity master controller modulates the setting of the dewpoint thermostat which becomes the sub-master controller. This system provides good humidity control for multi-zone commercial installations in which each zone has a reheat coil controlled by the respective zone thermostat. The air flow quantity for each zone would correspond to the prevailing condition of sensible heat ratio, and the humidity controller would be placed in the zone requiring the lowest entering dewpoint temperature; that is, that zone having the lowest sensible heat ratio. With changing conditions of sensible heat ratio in the various zones, the relative humidity would change slightly from the design figure in all zones but the one containing the humidity controller. The value of the humidity controller is that in a given installation most zones will vary in a similar degree throughout the day.

In industrial applications, generally the internal sensible heat ratio and total load are constant. A typical example is the textile industry.(51) When such is the case, the humidity reset master controller could be omitted. For a textile mill, this modern system would provide supply air at constant dewpoint temperature at all times, regardless of air conditions entering the washer, to provide constant relative humidity and dry-bulb temperature inside the mill. Economies in the required size of the central unit and in the size of ductwork are obtained by evaporative cooling with compressed air-water atomizers scattered throughout the mill.

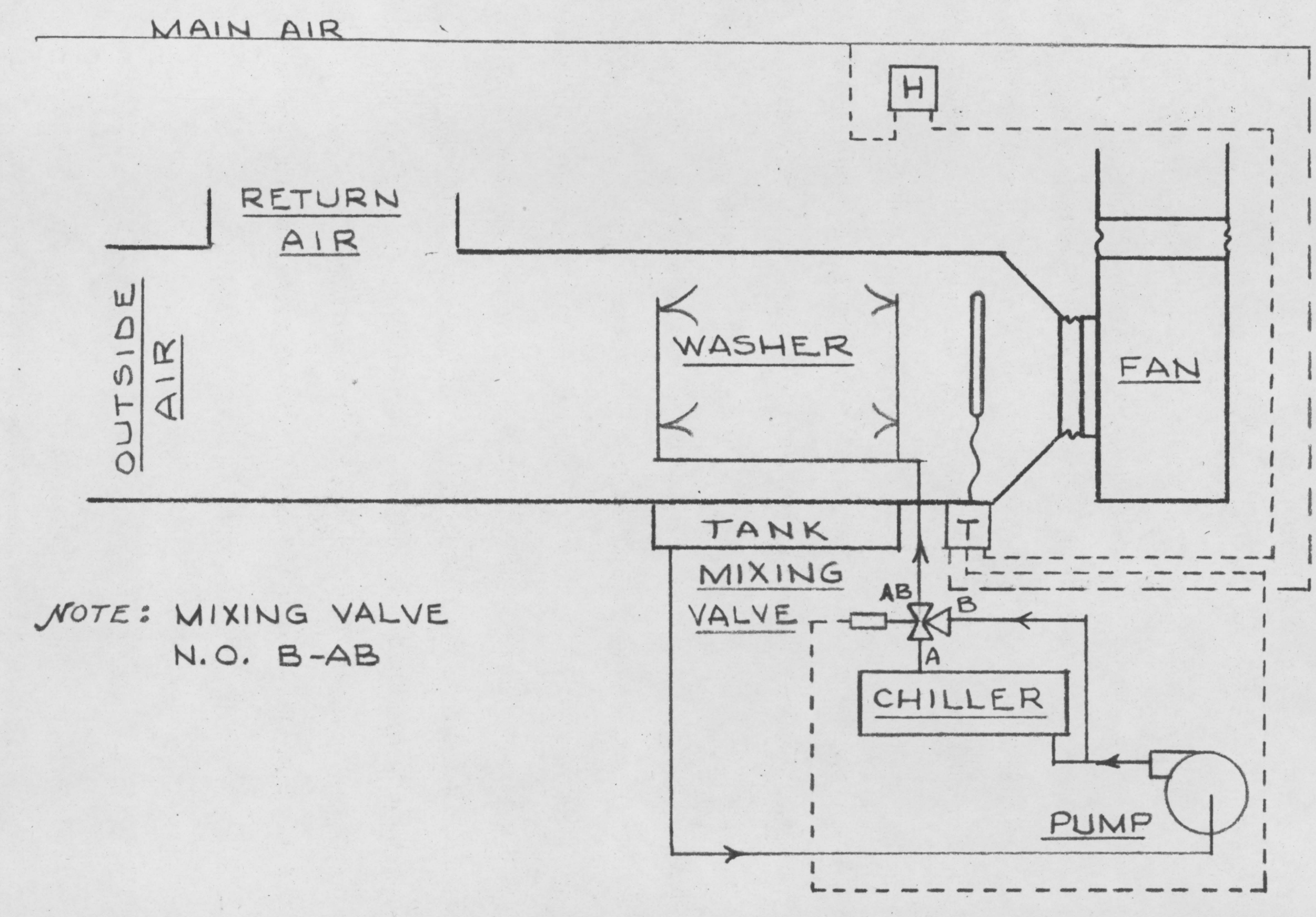


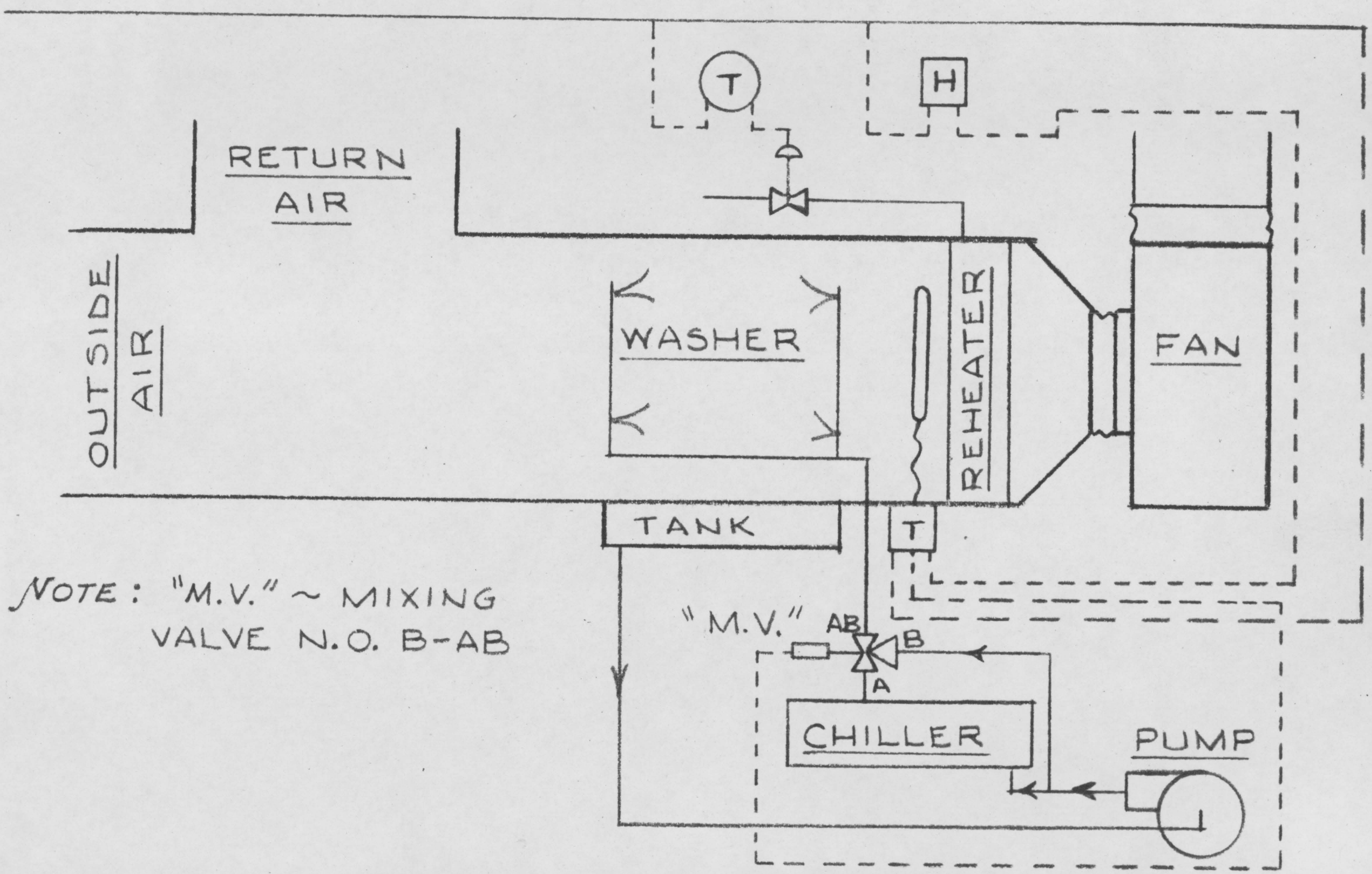
FIGURE 19. CONTROL OF DEWPOINT TEMPERATURE (WASHER)

Control of Temperature and Humidity with Dehumidifying Washer and Reheater. The mixture of return air and outside air is chilled and dehumidified by intimate contact with the chilled water spray of the washer. The temperature of water entering the washer can be controlled by mixing the water leaving the chiller with water bypassed around the chiller. Full flow through the pump is necessary at all times to get proper atomization in the spray nozzles of the washer. Water leaving the chiller is automatically maintained at a predetermined setting by a thermostat with remote bulb in the chiller outlet water. The compressor is on pump down control. The steam reheat coil is controlled by throttling its steam supply when too much sensible heat is removed by the dehumidifying process. The system fan operates continuously.

Automatic Control: A direct-acting dewpoint thermostat with remote bulb located in the outlet of the dehumidifying washer varies the temperature of supply water to the washer by modulating a three-way mixing valve at the chiller outlet. A reverse-acting space humidity controller modulates the setting of the dewpoint thermostat, the submaster controller, to regulate space relative humidity. A direct-acting space thermostat modulates steam supply to the reheater coil to control space temperature. This system is designed to condition a single zone, and provides precise, excellent control of temperature and humidity for all conditions of air entering the apparatus.

Because of the dewpoint control furnished by the washer, even with very dry outside air and no latent heat gain in the conditioned space, it would never be necessary to control cooling capacity with the space thermostat. This is in contrast with a system using a direct expansion coil and reheater coil, which must have control over sensible cooling when entering air at the apparatus is exceptionally dry. This system provides accurate control of temperature and humidity regardless of the condition of entering air to the conditioning apparatus, or change of sensible heat ratio in the conditioned space. In conclusion, this system represents superior control design.

In those industrial applications where the sensible heat ratio of the conditioned space does not change, the humidity reset master controller is not necessary.



NOTE: "M.V." ~ MIXING
VALVE N.O. B-AB

FIGURE 20. CONTROL OF TEMPERATURE AND HUMIDITY WITH WASHER AND REHEATER

C. THE INTEGRATED CONTROL SYSTEM

CONSOLIDATION OF CONTROLLED CYCLES

Integrating all the basic cycles just described and illustrated into a composite layout was worked out in a form by which the details of installation of all control components, and the construction of a central graphic panel, might be accomplished. This drawing is shown on Plate 2 inside the back cover. On the drawing the components of the built-up unit are schematically presented for simplicity and clarity rather than drawn to actual scale.

It is planned that the master temperature and humidity controllers will be mounted in the test room. An alternate location would be the return air duct from the test room. Between the master controllers and the actuators on the components of the built-up unit, all control lines should be routed through the central graphic panel. On the panel will be placed the stop and bleed cocks so that all of the summer and winter cycles may be set up and observed in operation. Those gages and controllers which function as part of the operating cycles, such as the compressor suction and discharge gages and safety switches, should also be mounted on the graphic panel.

THE PNEUMATIC CONTROL SYSTEM

This section will describe the equipment, material, and services necessary for the proper installation of the pneumatic system of automatic temperature and humidity regulation. It is recommended that the entire control system as described in this section be installed by mechanics regularly employed by the control manufacturer. The description is so written that it may serve as a guide in writing specifications to secure bids. In addition, the proper model numbers for the pieces of equipment, as listed by the Minneapolis-Honeywell Regulator Company, were obtained to establish the exact criteria of selection. Valve bodies and face and bypass dampers, though purchased from the control manufacturer, would be installed by employees of the Mechanical Engineering Department prior to installation of the control system. The control manufacturer normally selects the proper size valve for which he is supplying the control, based on the specified conditions of pressure and flow rate.

Air Piping. The compressed air piping throughout shall be hard drawn copper tubing and shall be run exposed and properly secured to the building structure. The entire air piping system shall be tested under 30 psig pressure and should not leak more than 10 per cent overnight.

Reducing Station. The present 80 psig air pressure in the laboratory should be reduced to the desired control system operating pressure and filtered by a Honeywell Q197A reducing valve and filter station.

Valves. Steam control valves for the tempering, preheater, and reheater coils and the water heater shall be of the modulating type equipped with spring-loaded self-adjusting Teflon packing. The 150 psi cast bronze valve bodies shall have replaceable seats, removable composition discs and contoured plugs to provide equal percentage flow characteristics. Valves to be Honeywell type K0514C or equal, and sized by the control manufacturer.

The three-way hot and chilled water mixing valve shall be of the modulating type equipped with spring loaded Teflon packing. Valve body shall have a one-piece contoured plug to give linear flow characteristics and constant total flow throughout the full plug travel. It should be equal to the Honeywell type K0516C.

Thermostats. The space thermostat, equal to the Honeywell type T0910A, shall be equipped with adjustable throttling range, bi-metal sensing element, bimetallic thermometer, locking type metal cover, concealed lock type adjustments for limiting or locking the temperature setting range, and plug-in air gage connection for calibration purposes or visual indication of branch line pressure. The remote bulb thermostats for outside air and fan discharge air shall be equal to Honeywell type LO900A. The submaster dewpoint controller to be equal to Honeywell type LO900B. Thermostats should be equipped with adjustable throttling ranges, key type adjustments, locking type covers, and at least five feet of capillary tubing.

Humidity Controller. The space humidity controller, equal to the Honeywell type HO900B, shall be of the reverse-acting modulating type equipped with adjustable throttling range, hair element, visible scale, and locking type metal cover.

Damper and Damper Motor. Dampers for face and bypass operation, equal to the Honeywell type D42C, shall be constructed of channel iron frames, 16 gage galvanized opposed type blades, and brass sleeve type bearings. The face and bypass damper operator, equal to the Honeywell type MO900B, shall be equipped with a Grad-U-Trol relay to adjust the air pressure operating range and to provide positive positioning of the dampers regardless of external load conditions.

Relays and Accessory Equipment. An electric-pneumatic relay, equal to the Honeywell type RO400A, shall de-energize the entire control system when the system fan is de-energized. A diverting relay, equal to the Honeywell RO48B, shall permit the higher of two pressures to control the appropriate actuators. A reversing relay between the humidity controller and the diverting relay shall reverse the controller air pressure (since impulses at the diverting relay must be direct acting). It shall be equal to the Honeywell type R095B. Pneumatic-electric relays equal to the Honeywell L404H shall be furnished for control of the city water solenoid, direct expansion coil solenoid, and the system water pump.

Graphic Control Panel. Based on Plate 1, inside the back cover, a factory-built "colorgraphic" panel shall be furnished and installed which provides a color-coded schematic process diagram of the entire air conditioning built-up unit. It shall be constructed of 3/16 in. gage steel with turnbacks on the top and sides and shall be painted front and back with a primer coat. The front shall have two additional coats of enamel baked on. The size and color shall be decided later.

A complete piping and duct layout of the entire unit shall be reverse engraved on 3/16 in. Lucite with the engraved portions, consisting of symbols, ductwork, piping, lettering, etc., colored according to the function it symbolizes. For example, the sealed-in color for heating shall be red; for cooling, blue; for ductwork, black; for control piping, yellow; for background, light tan, gray, or green.

The layout shall consist of ductwork, tempering coil, fresh and return air dampers, disposable filter, electronic filter, activated carbon filter, preheater coil, direct expansion coil, hot and chilled water coil, face and bypass dampers, washer, reheater coil, air and temperature measuring chambers, centrifugal fan, pump, water chiller, water heater, steam and water piping, hand valves, controls and control valves, and pneumatic interconnecting tubing. Stopcocks and stop-and-bleed cocks, adjustable from the face of the panel and located in the pneumatic tubing circuits, shall enable the operator to set up various heating and cooling cycles under automatic control. Flush mounted air gages located in the pneumatic tubing circuits indicating high pressure air, low pressure air, system air, thermostat branch line pressure, humidity controller branch line pressure, relay branch line pressures, and controlled device pressures shall enable the operator to observe the complete operation of all control components.

Relays and accessory equipment shall be factory-mounted on the back of the panel and pre-wired to a numbered terminal strip.

The control equipment, as all other equipment, shall be guaranteed against original defects in workmanship and material for at least one year. The hermetic compressor shall have the standard five-year warranty.

VI. CONCLUSION

The work of this thesis began with the problem of determining the type of system and equipment that should be acquired for the Mechanical Engineering Laboratory at the Virginia Polytechnic Institute to facilitate demonstration and research in air conditioning. The thesis concludes by proposing the construction of a built-up unit. The Review of Literature summarized the research which was necessary to plan the unit so that it would be modern and useful for years to come. The Design Section included calculations, sketches, selection of major components and accessory equipment, and design of the functioning unit.

Because of the conservative nature of the air conditioning industry and because of the versatility and flexibility incorporated into the unit as a result of the search for future trends, the unit will be modern for a long time. It will be simple to change the components of the unit without altering the basic design. For example, in following the evolution of extended surface coil design, new coils can be substituted for old ones.

After this proposal is approved, preparation of shop drawings can be made immediately from the sketches and scaled drawings of the thesis, and the materials can be immediately procured for erection. The writer believes that the proposed unit would provide this school, at moderate cost, with facilities for demonstration and research which would be unsurpassed by any educational institution in the country.

A. LIMITATIONS

The basic limitation was the probable amount of money available for the proposed unit. This eliminated a chemical dehumidifier, which would allow return air to the unit to simulate hot, arid conditions, and allow demonstration of dehumidification of air by a method other than cooling it below the dewpoint.

Because of its small size, the compressor does not have variable capacity control. Make-shift methods of capacity control were considered but rejected; nothing should be considered which would eliminate the complete reliability of a hermetically sealed compressor and motor drive. If in operating the proposed unit it is found desirable to operate the compressor continuously and yet vary the load on the evaporator, then one of the methods of compressor loading outlined in the Review of Literature may easily be incorporated. The writer recommends the "hot-gas bypass with liquid quench." The bypass valve should be a manual throttling type, and the liquid quench should be regulated by a thermostatic expansion valve with remote bulb on the compressor suction pipe.

The fact that the unit is not able to demonstrate dual-duct systems is not important.

The design of the unit was adjusted to the space available, as shown on Plate 1. The required services are available for the unit: alternating and direct current of proper phase and voltage, recirculated and fresh water, compressed air, and steam. Locations of outside air intake and drains to sewer are adequate.

The design of the proposed unit was based on instructions that a separate bypass duct for the return air around the conditioning coils should not be installed because of the difficulty in making the dampers leak-proof. Such a duct is in use on the University of Illinois unit. However, notwithstanding the inconvenience of blanking off the bypass duct when air flow measurement through the main duct must be exact, such a bypass would allow many more cycles to be demonstrated. In the proposed unit it was necessary to avoid drawing outside air into the system on summer cycles when a face and bypass damper is used at the conditioning coil. But this would not be a problem if the bypass duct, fitted with the same perforated plate and bypass dampers, were run from the return air duct to a point downstream from the conditioning apparatus. This bypassed return air is good for reheat. As before, the face and bypass dampers would be mechanically connected and the vane positions calibrated after installation.

In demonstrating automatic control systems, automatic damper control of outside air intake was not included. It would have little research value, and the fact that demonstrations will be held according to an academic schedule rather than according to weather conditions limits the value to undergraduates.

B. RECOMMENDATIONS

In connection with the air temperature measuring apparatus, it is advisable to experimentally determine the best design of air sampling device. This knowledge would also be a valuable contribution to general instrumentation techniques. The design developed in this thesis was based on the published information of industrial laboratories. Changing the spacing of the holes and tubes and substitution of tapered tubes should be tried. The sampling device is a critical part of the entire circuit.

C. DISCUSSION OF RESULTS

As seen on Plate 1 the proposed unit includes the following: outside air intake at existing roof opening; tempering coil in outside air duct; manual dampers to control return air and outside air to conditioning unit; system of ductwork connecting test chamber to the built-up conditioning unit; duct windows to permit observation of all processes; impingement type disposable filter; activated carbon filter; preheater coil; direct expansion coil; hot and chilled water coil; face and bypass dampers; washer and scrubber; reheater coil; air temperature measuring chamber; air flow measuring chamber; and centrifugal fan with inlet vanes.

Many important winter and summer cycles found in industrial, commercial, and residential applications can be demonstrated with the unit. In addition, all of the cycles can be automatically controlled with the proposed pneumatic system.

Special emphasis was placed on the methods for obtaining efficient purification of the air. Washers are still applicable to large installations, provided that means can be found for removing particulate matter and vapors insoluble in water. The major limitation of washers--not being able to remove the worst offender in comfort work, tobacco smoke--

has led to the use of an electronic filter in combination with the activated carbon filter. Emphasis on air purification makes the proposed unit distinctive from existing units. The future trend in air conditioning will be to cut down on the required cooling and heating capacity of air conditioning units by purifying rather than exhausting contaminated air from the conditioned spaces.

All components of the proposed unit will meet the high standards of quality and efficiency found in industry. For the student, this fact has a wholesome significance in running performance tests.

A primary fault of most experimental work is that the observer cannot observe what is taking place. The proposed unit was planned to include observation windows of double pane plate glass, hermetically sealed, with marine-type lights to illuminate the interior processes.

Several factors make this unit inherently different from the usual commercial installation. The first is the fact that as many different methods of conditioning the air as possible have been incorporated so that some of the components will be doing the same job, though not at the same time. Second, the proposed unit presents an irregular, "in and out" shape, whereas commercial systems have smooth, box-like shapes. Third, the unit is conspicuously instrumented for testing and research. Because of so many components in series, the fan must develop an exceptionally high static pressure. The high static pressure is also necessary for a high degree of accuracy in the air flow measuring chamber.

D. APPLICATION OF THE UNIT

Undergraduate. The proposed year-around air conditioning unit will have a direct application to the instruction of undergraduate students. It will permit them to do the following:

(1) Practice performance and efficiency tests on an air conditioning refrigeration unit.

(2) Observe and operate a system of pneumatic control for heating, cooling, humidifying, and dehumidifying air.

(3) Practice overall performance tests in order to make heat balances for different winter and summer air conditioning cycles.

(4) Become familiar with construction and details of steam, water, refrigerant, and control piping layout; and the arrangement of equipment and ducts.

(5) Practice performance tests on non-freeze and standard type extended surface heating coils.

(6) Make verifications of actual capacity of direct expansion dehumidifying coil as predicted by analytical and graphical computation.

(7) Study the reheating of cooler and dehumidifier air by use of a steam reheating coil.

(8) Practice performance tests on an extended surface direct expansion type cooling unit, and also study the effect of air velocity on performance.

(9) Practice performance tests on an extended surface chilled and hot water coil.

(10) Study and determine the effect of coil depth on bypass factor, and also the effect of varied water flow rates on performance of a chilled and hot water coil.

(11) Compare the heat transfer rate in finned coils having different type refrigerants: constant phase and change in phase by boiling.

(12) Study the effect of face and bypass dampers on coil leaving air conditions.

(13) Study the performance of an air washer as a humidifier and as a dehumidifier.

(14) Observe operation and change in efficiency of an air washer with one or two rows of spray nozzles and with the use of a scrubber.

(15) Study the effect of varying water temperatures on the performance of an air washer.

(16) Study the effectiveness of different types of air cleaners other than a washer-scrubber: viscous impingement filter, dry filter, electronic filter, and activated carbon filter.

(17) Approximate friction loss in various types of ductwork fittings and connections by air flow measurement instrumentation.

(18) Learn methods of precision measurement of dry-bulb and wet-bulb temperatures, and of air flow rate.

(19) Measure air friction loss through each piece of equipment.

(20) Study the operating characteristics of a backward curved centrifugal fan.

Graduate. An almost unlimited number of interesting possibilities in graduate research will be possible with the proposed apparatus. A few are listed:

(1) Verify and extend economic studies already begun on the advantages of installing sufficient air purification apparatus to eliminate the necessity of outside ventilating air, thus saving on the initial and operating costs of an air conditioning system.

(2) Study patterns of air flow from various registers and diffusers by mounting duct in the test room and inserting a smoke pot in the supply duct, with the purpose of developing improved diffusion techniques.

(3) Study air flow patterns through such fittings as duct turns by means of transparent side panel and smoke pot.

(4) Develop better methods of joining the transverse and longitudinal seams of sheet metal ducts.

(5) Observe the effects of fouling on the water side of the refrigeration condenser and steam water heater; and develop methods for counteracting scale formation.

(6) Develop good, reliable, simple means of adding humidity to the air in small systems not large enough for a washer, and in large systems not having a washer.

(7) Relate performance data to recommendations for changes in design in evolving improvements on any of the components of the built-up unit.

(8) Experiment with new solutions added to washer water to make it capable of removing impurities insoluble in water. The solution must lend no odor to the air nor affect the ability of the scrubber and eliminator plates to remove entrained particulate matter and moisture from the air.

(9) Construct an experimental room in the test chamber for investigating heat transmission and vapor migration through the building materials selected. The proposed unit would hold the outside of the structure at any selected temperature and relative humidity; a heater and humidifier would be placed inside the structure and automatically controlled. This is an alternative to the idea already planned for conditioning a model building in the test chamber with air from the built-up unit, while the artificial climate is provided by chilled brine and steam coils in the test chamber.

(10) Develop a water-soluble, viscous liquid for use on impingement type filters and electronic filter collecting plates which will not evaporate. This vapor at present clogs the capillary channels of activated carbon filters.

(11) Study the expediency and value of placing ultraviolet lights in the duct to destroy bacterial and emit negative ions.

General. For general research work of the Department, the College, and the Experiment Stations, the unit will provide a constant temperature, constant humidity test room which can be automatically held within close limits at any reasonable climate condition.

The two page vita has been removed from the scanned document. Page 1 of 2

The two page vita has been removed from the scanned document. Page 2 of 2

VIII. BIBLIOGRAPHY

1. ACME DRY-EX LIQUID CHILLERS, CATALOG NO. 600-A, Acme Industries, Inc., Jackson, Michigan, 1955.

2. ACME REFRIGERATION AND AIR CONDITIONING EQUIPMENT, CATALOG NO. 23-D, Acme Industries, Inc., Jackson, Michigan, 1954.

3. ACME SMALL TONNAGE CONDENSERS, CATALOG SERIES 235, Acme Industries, Inc., Jackson, Michigan, 1951.

4. ACME SYSTEM-ENGINEERED COMPONENTS, CATALOG NO. 300-A, Acme Industries, Inc., Jackson, Michigan, 1955.

5. AEROFIN HEAT-EXCHANGE SURFACE, BULLETIN D-51, Aerofin Corp., 101 Greenway Ave., Syracuse 3, N. Y., 1951.

6. AEROFIN REMOVABLE HEADER WATER COILS—TYPE "R", BULLETIN NO. R-50, Aerofin Corp., Syracuse 3, N.Y., 1950.

7. AEROFIN TYPE B NON-FREEZE AND TYPE B FLEXITUBE STEAM HEATING COILS, BULLETIN B-54, Aerofin Corp., Syracuse 3, N.Y., 1954.

8. AEROFIN WATER COILS--TYPE "C", BULLETIN NO. C-58, Aerofin Corp., Syracuse 3, N.Y., 1958.

9. Allen, J. R., J. H. Walker, and J. W. James, HEATING AND AIR CONDITIONING, McGraw-Hill Book Co., N. Y., 1946. Sixth Edition.

10. AMERICAN AIR FILTER BULLETIN NO. 216A, REPLACEABLE MEDIA AIR FILTERS, American Air Filter Co., Inc., Louisville, Kentucky, 1956.

11. AMERICAN AIR FILTER BULLETIN NO. 250-H, ELECTRO-MATIC MODEL F AUTOMATIC ELECTRONIC PRECIPITATOR, American Air Filter Co., Inc., Louisville, Kentucky, 1957.

12. AMERICAN AIR FILTER BULLETIN NO. 249, ROLLOTRON AIR FILTRATION UNIT, American Air Filter Co., Inc., Louisville, Kentucky, 1957.

13. AMERICAN AIR FILTER BULLETIN NO. 252-D, ELECTRO-CELL MODEL A STATIONARY PLATE TYPE ELECTRONIC PRECIPITATOR, American Air Filter Co., Inc., Louisville, Kentucky, 1955.

14. A.R.I. STANDARD 411-56, METHODS OF TESTING AND RATING FORCED-CIRCULATION AIR-COOLING AND AIR-HEATING COILS, Air-Conditioning and Refrigeration Institute, 1346 Connecticut Ave., N. W., Washington 6, D. C., 1956.

15. A.S.H.V.E. TRANSACTIONS, "A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work," American Society of Heating and Ventilating Engineers, 62 Worth St., N. Y. 13, N. Y., vol. 39, p. 225, 1933.

16. A.S.H.V.E. TRANSACTIONS, "Code for Minimum Requirements for Comfort Air Conditioning," American Society of Heating and Ventilating Engineers, N. Y. 13, N. Y., vol. 44, p. 27, 1938.

17. A.S.M.E. POWER TEST CODES, 1942, INFORMATION ON INSTRUMENTS AND APPARATUS, Part 2: Pressure Measurement; Chapter 5: Liquid Column Gages, The American Society of Mechanical Engineers, 29 West 39th St., N. Y., 1942.

18. A.S.M.E. POWER TEST CODES, SUPPLEMENT ON INSTRUMENTS AND APPARATUS, Part 2: Pressure Measurement; Chapters 2 and 3: Static and Total Pressure, etc., The American Society of Mechanical Engineers, N. Y., 1949.

19. A.S.M.E. POWER TEST CODES, TEST CODE FOR FANS, The American Society of Mechanical Engineers, N. Y., 1946.

20. A.S.R.E. STANDARD 16-56, METHODS OF RATING AND TESTING AIR CONDITIONERS, The American Society of Refrigerating Engineers, 234 Fifth Ave., N. Y. 1, N. Y., 1955.

21. AUTOMATIC REFRIGERANT CONTROLS, A HANDBOOK OF INFORMATION AND DATA, Alco Valve Co., 865 Kingsland Ave., St. Louis 5, Missouri, 1955.

22. Beckett, John C., "Ions' Role in Air Conditioning Takes on New Importance," HEATING, PIPING AND AIR-CONDITIONING, Jan., 1958.

23. BUFFALO AIR WASHERS, BULLETIN NO. 3142-E, Buffalo Forge Co., Buffalo 5, N. Y., 1952.

24. BUFFALO PUMPS, BULLETIN NO. 975-F, Buffalo Pump Division, Buffalo 5, N. Y., 1958.

25. Campbell, Margaret, Registered Graduate Nurse, Personal Interviews in June, 1953, New York City.

26. Carpenter, J. H., "Air Conditioning and Heating Laboratory Supplements Classroom Instruction," HEATING, PIPING, AND AIR-CONDITIONING, Nov., 1943.

27. CARRIER BULLETIN: AIR CONDITIONING FOR THE TEXTILE INDUSTRY, A.I.A. No. 30-F-2, Carrier Corporation, Syracuse, N. Y., 1951.

28. CARRIER ENGINEERS MANUAL, APPLIED EQUIPMENT VOLUME, Carrier Corp., Syracuse, N. Y., 1959.

29. Carrier, W. H., R. E. Cherne, W. A. Grant, and W. H. Roberts, MODERN AIR CONDITIONING, HEATING, AND VENTILATING, Pitman Publishing Corp., N. Y., 1959. Third Edition.

30. CARRIER DESIGN DATA, PAMPHLET 3X-51PD1, Carrier Corp., Syracuse, N. Y., 1951.

31. CONTROL MANUAL FOR HEATING, VENTILATING AND AIR CONDITIONING, Minneapolis-Honeywell Regulator Co., Minneapolis, Minn., 1950.

32. Dill, Richard S., "A Test Method for Air Filters," A.S.H.V.E. TRANSACTIONS, vol. 44, p. 379, 1938.

33. ENGINEERING MANUAL OF AUTOMATIC CONTROL FOR COMMERCIAL AIR CONDITIONING, Minneapolis-Honeywell Regulator Co., Minneapolis 8, Minn., 1957.

34. Faires, V. M., APPLIED THERMODYNAMICS, The Macmillan Co., N. Y., 1947.

35. FAN ENGINEERING, AN ENGINEER'S HANDBOOK (R. D. Madison, Editor), Buffalo Forge Co., Buffalo, N. Y., 1949. Fifth Edition.

36. Goodman, William, AIR CONDITIONING ANALYSIS, The Macmillan Co., N. Y., 1943.

37. Haines, J. E., AUTOMATIC CONTROL OF HEATING AND AIR CONDITIONING, McGraw-Hill Book Co., Inc., 1953.

38. HEATING VENTILATING AIR CONDITIONING GUIDE 1950, American Society of Heating and Ventilating Engineers, N. Y., 1950 (vol. 28).

39. HEATING VENTILATING AIR CONDITIONING GUIDE 1956, American Society of Heating and Air-Conditioning Engineers, Inc., 62 Worth St., N. Y. 13, N. Y., 1956, (vol. 34).

40. INDUSTRIAL VENTILATION, A MANUAL OF RECOMMENDED PRACTICE, Committee on Industrial Ventilation, American Conference of Governmental Industrial Hygienists, P. O. Box 453, Lansing, Michigan, (no date).

41. INSULATION DESIGN FOR THE AIR-CONDITIONED HOME, FIBERGLAS DESIGN DAtaBL6.A3, Owens-Corning Fiberglas Corp., Toledo 1, Ohio, 1955.

42. Jennings, B. H. and S. R. Lewis, AIR CONDITIONING AND REFRIGERATION, International Textbook Co., Scranton, Pa., 1950.

43. Jordan, R. C. and G. B. Priester, REFRIGERATION AND AIR CONDITIONING, Prentice Hall Inc., 1956. Second Edition.

44. Keenan, J. H. and F. G. Keyes, THERMODYNAMIC PROPERTIES OF STEAM, John Wiley & Sons, Inc., N. Y., 1936.

45. Knight, Richard B., Professor-in-Charge of Heating and Air Conditioning Curriculum, North Carolina State College, Personal Communication, Aug. 6, 1956.

46. Kusuda, T., "Graphical Method Simplifies Determination of Air Coil Wet Heat Transfer Surface Temperature," REFRIGERATING ENGINEERING, May, 1957, pp. 41-45.

47. Kusuda, T., THEORETICAL AND EXPERIMENTAL ANALYSIS OF WATER VAPOR TRANSMISSION THROUGH SORPTIVE SOLIDS, (Ph. D. Thesis), University of Minnesota, 1955.

48. Landgraf, G. F., "The Electronic Air Cleaner," IRON AND STEEL ENGINEER, Oct., 1953, pp. 91-110.

49. Lee, A. L., Jr., THE EFFECTS OF AIR DIRECTIONAL GUIDE VANES AND AIR TURBULATORS ON THE AIR FLOW AND CAPACITY CHARACTERISTICS OF TUBE AND FIN HEAT EXCHANGERS, (M.S. Thesis), Virginia Polytechnic Institute, June, 1959.

50. Lee, A. L., Jr., Development and Design Engineer, Westinghouse Air Conditioning Division, Personal Interview, July 7, 1959, Staunton, Virginia.

51. Macintire, H. J. and F. W. Hutchinson, REFRIGERATION ENGINEERING, John Wiley & Sons, Inc., N. Y., 1950. Second Edition.

52. Marks, L. S. (editor), MECHANICAL ENGINEERS' HANDBOOK, McGraw-Hill Book Co., Inc., N. Y., 1951. Fifth Edition.

53. MINNEAPOLIS-HONEYWELL ELECTRONIC AIR CLEANERS, SELECTOR SERIES, Minneapolis-Honeywell Regulator Co., Minneapolis, Minn., 1959.

54. MINNEAPOLIS-HONEYWELL PYROMETRIC SWITCHES, CATALOG 5800, Minneapolis-Honeywell Regulator Co., Industrial Division, Phila., Pa., 1951.

55. Nagy, Rudolph A., "Report on Ionization of Air on Health," AIR-CONDITIONING, HEATING, AND VENTILATING, Jan., 1959, p. 62.

56. Reed, P. B., AIR CONDITIONING REFRIGERATING DATA BOOK, DESIGN VOLUME, The American Society of Refrigerating Engineers, 234 Fifth Ave., N. Y. 1, N. Y., 1955, chapter 7. Ninth Edition.

57. REFRIGERATING DATA BOOK, BASIC VOLUME, The American Society of Refrigerating Engineers, N. Y., 1949. Sixth Edition.

58. Severns, W. H. and M. K. Fahnestock, "University of Illinois Installs New Experimental Conditioning Equipment," HEATING, PIPING AND AIR CONDITIONING, April, 1937, pp. 227-230.

59. Sharpe, Norman, REFRIGERATING PRINCIPLES AND PRACTICES, McGraw-Hill Book Co., Inc., N. Y., 1949.

60. Sharpe, Norman, "The Economies of Air Purification," REFRIGERATING ENGINEERING, Sept., 1953, pp. 959-963.

61. Sleik, Henry and Amos Turk, AIR CONSERVATION ENGINEERING, Connor Engineering Corp., Danbury, Conn., 1953.

62. Smith, Donald C., District Manager, Minneapolis-Honeywell Regulator Co., Personal Communication, June 18, 1959.

63. Smith, Donald C., District Manager, Minneapolis-Honeywell Regulator Co., Personal Interviews in May, 1959, Blacksburg, Virginia.

64. SPORLAN CONDENSED CATALOG NUMBER 56-C, Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Missouri, 1955.

65. SPORLAN CATALOG NO. 55, BULLETIN 20-10, Sporlan Valve Co., St. Louis 17, Missouri, 1954.

66. TACO COMMERCIAL AND INDUSTRIAL HEAT EXCHANGERS, Taco Heaters, Inc., 1160 Cranston St., Cranston 9, R. I., (no date).

67. TRANE AIR CONDITIONING MANUAL, The Trane Co., La Crosse, Wisconsin, 1955.

68. TRANE REFRIGERATION MANUAL, A PRACTICAL VOLUME ON THE INSTALLATION, MAINTENANCE AND SERVICE OF REFRIGERATION EQUIPMENT USED IN CONJUNCTION WITH AIR CONDITIONING SYSTEMS, The Trane Co., La Crosse, Wisconsin, 1956.

69. Wile, D. D., "Air Flow Measurement in the Laboratory," REFRIGERATING ENGINEERING, June, 1947.

70. Wile, D. D., "Measurement of Temperature in the Laboratory by Means of Thermocouples," REFRIGERATING ENGINEERING, Jan. and Feb., 1945.

71. Yaglou, C. P., E. C. Riley, and D. I. Coggers, "Ventilation Requirements," TRANSACTIONS A.S.H.V.E., vol. 42, pp. 133-162, 1938.

IX. TERMINOLOGY AND ABBREVIATIONS

Terminology and abbreviations which appear in the thesis are defined as follows:

A.R.I.: Air-Conditioning and Refrigeration Institute.

A.S.H.A.E., or formerly A.S.H.R.A.E. and A.S.H.V.E.: American Society of Heating and Air-Conditioning Engineers, Inc.

Btu: British thermal unit

cfm: cubic feet of air per minute. In this thesis cfm refers to rate of air flow when measured at standard conditions of temperature and pressure. Cfm does not mean rate of flow actually indicated in laboratory apparatus. Standard air is substantially equivalent to dry air at 70°F and 29.92 in. mercury barometric pressure.

degrees temperature: always degrees Fahrenheit unless specifically stated otherwise.

fpm: feet per minute.

in.: inch(es).

psia: pounds per square inch absolute pressure.

psig: pounds per square inch gage pressure.

sq.ft.: square feet

ABSTRACT

The purposes of this thesis were to investigate the contemporary field of air conditioning in industry and education; to determine what equipment should be recommended for installation in the Mechanical Engineering Laboratory of the Virginia Polytechnic Institute; and to design the components and controls for a unit which will be modern, practical, and educational. The components of the unit were designed or selected on the basis of the research summarized in this thesis in Chapter II, "Review of Literature."

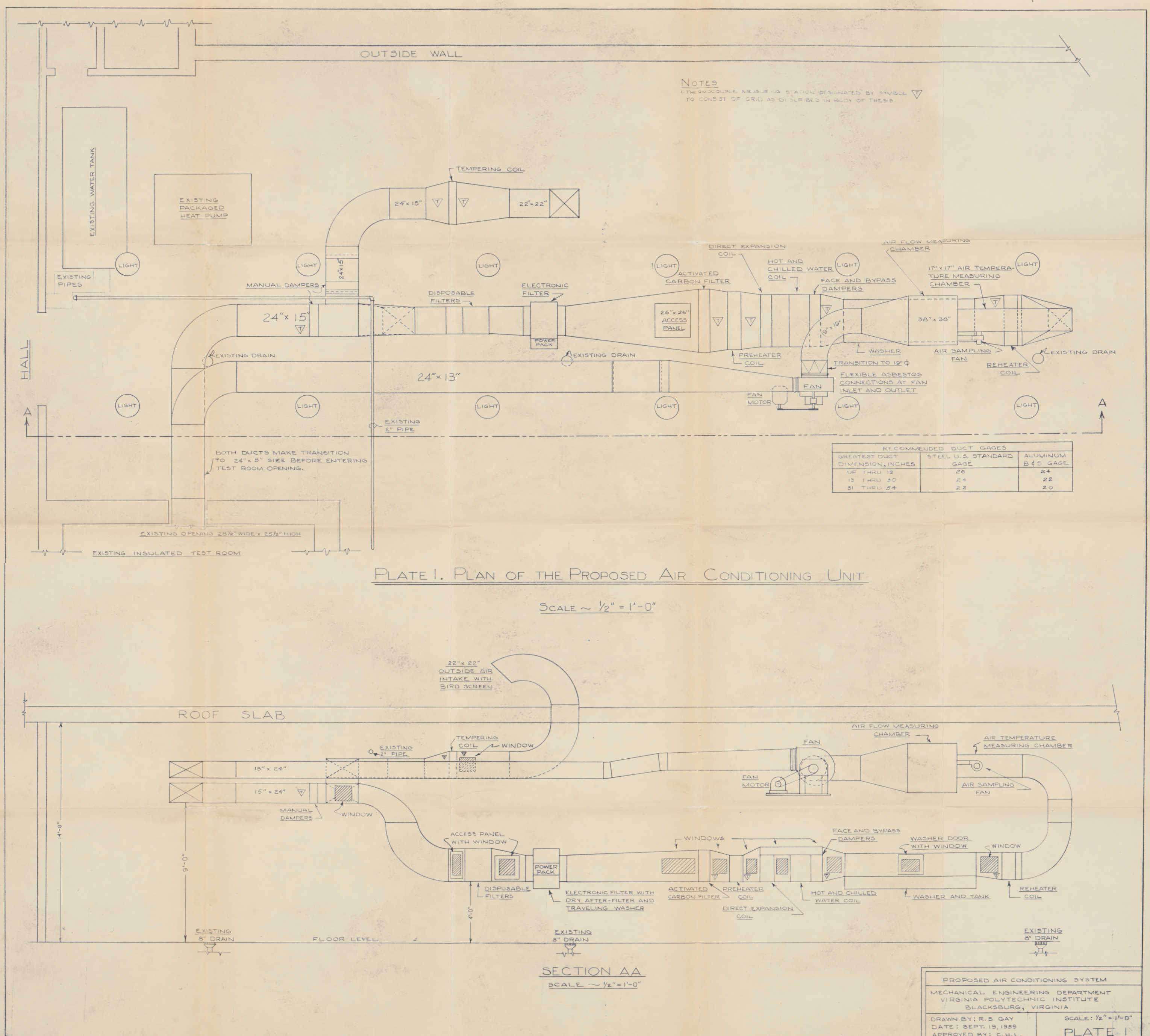
A central fan system was designed. Within the necessary design limitations, every effort was made to incorporate as many different methods as possible which demonstrate air conditioning processes, and every effort was made to enhance the effectiveness and attractiveness of the unit for its intended purposes of demonstration and research.

The main part of the conditioning apparatus will be supported in ductwork about four feet above the floor level so that weigh tanks, scales, condensate coolers, and drain lines may be placed underneath, thereby allowing the processes to be controlled and observed at normal eye level. In order to keep within the prescribed area, the air measurement chamber and fan will be mounted at a higher level. Ducts will carry air from the existing insulated test room to the unit and return conditioned air to the test room. A duct to

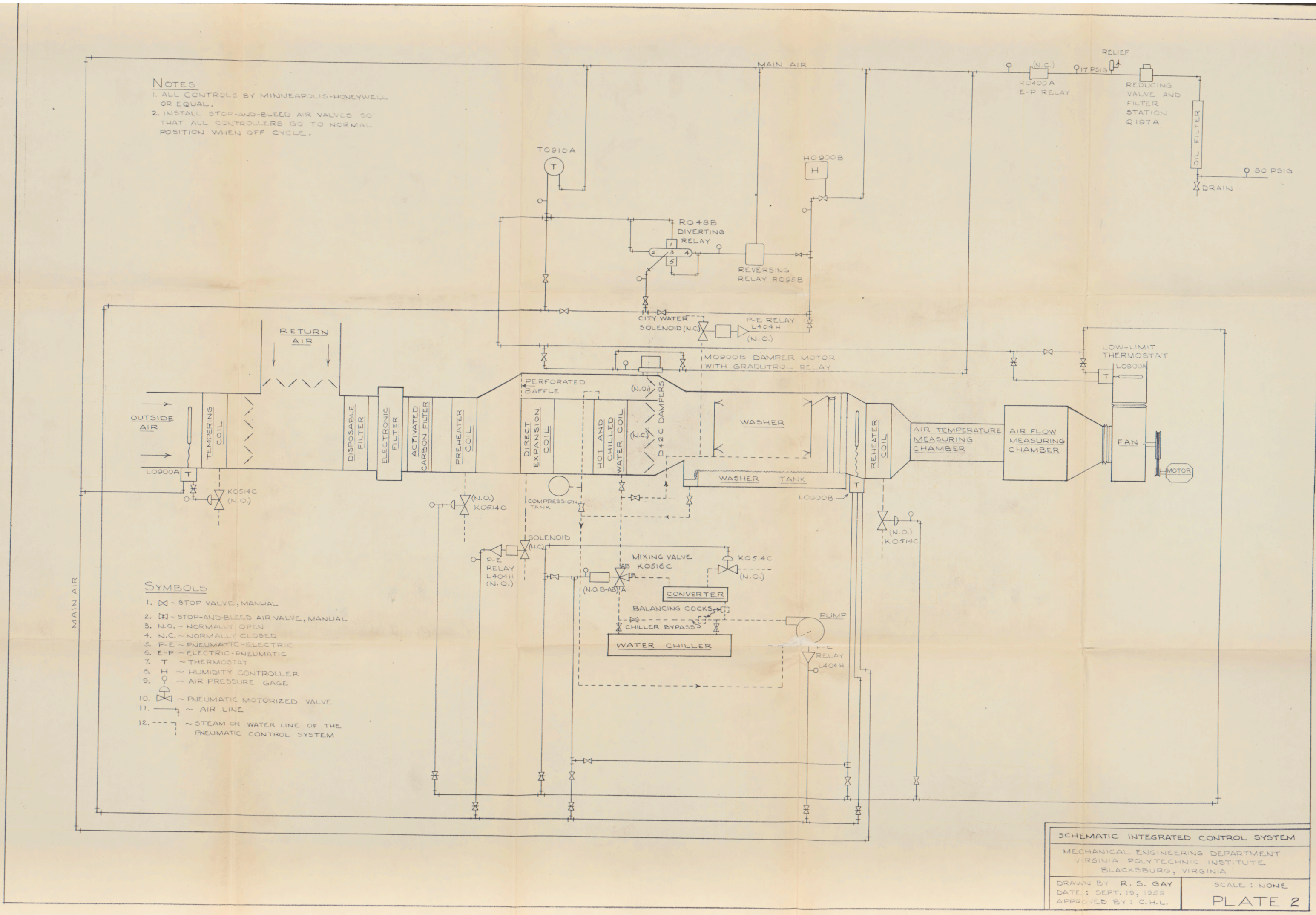
extend from an existing roof opening will be large enough to carry 2000 cfm and will contain a freeze-resistant tempering coil.

Manually adjusted opposed blade dampers control air quantities at the junction of the outside air duct and the return air from the test room. Air passing through the built-up unit will pass through the following components: disposable filter, electronic filter, activated carbon filter, steam preheater coil, direct expansion coil, hot or chilled water coil, opposed blade face and bypass dampers, washer, steam reheater coil, air temperature measuring chamber, air flow measuring chamber, and centrifugal fan with inlet vanes. Among the accessory components are a Refrigerant 12 refrigeration unit with water cooled condenser, water chiller, steam water heater, high-head circulating pump, and a wide assortment of valves and actuators for automatic control on any one of the many cycles being run.

Each piece of conditioning equipment operating on the summer cycle was designed for a nominal cooling capacity of five tons when the system is circulating the equivalent of 2000 standard cfm of air. The winter cycle equipment was designed to supply humidified air to the test chamber at not less than 100°F when 9000 pounds per hour of outside air at 0°F enters the equipment. The apparatus will be capable of maintaining within close limits any reasonable air temperature and humidity under all conditions of operation on both cooling and heating cycles.



RECOMMENDED DUCT GAGES		
GREATEST DUCT DIMENSION, INCHES	STEEL U.S. STANDARD GAGE	ALUMINUM B & S GAGE
UP THRU 12	26	24
13 THRU 30	24	22
31 THRU 54	22	20


NOTES
1. ALL CONTROLS BY MINNEAPOLIS-HONEYWELL OR EQUAL.
2. INSTALL STOP-AND-BLEED AIR VALVES SO THAT ALL CONTROLLERS GO TO NORMAL POSITION WHEN OFF CYCLE.
MAIN AIR
RELIEF
(N.C.)
17 PSIG
RO400A E-P RELAY
REDUCING VALVE AND FILTER STATION Q197A
OIL FILTER
30 PSIG
DRAIN
T0910A
H0900B
R0488 DIVERTING RELAY
REVERSING RELAY
CITY WATER SOLENOID (N.C.)
P-E RELAY L404H (N.O.)
M0900B DAMPER MOTOR WITH GRADUTROL RELAY
LOW-LIMIT THERMOSTAT L0900A
RETURN AIR
OUTSIDE AIR
TEMPERING COIL
L0900A
K0514C (N.O.)
DISPOSABLE FILTER
ELECTRONIC FILTER
ACTIVATED CARBON FILTER
PREHEATER COIL
PERFORATED BAFFLE
DIRECT EXPANSION COIL
HOT AND CHILLED WATER COIL
(N.C.)
(N.O.)
D42C DAMPERS
WASHER
WASHER TANK
REHEATER COIL
L0900B
AIR TEMPERATURE MEASURING CHAMBER
AIR FLOW MEASURING CHAMBER
FAN
MOTOR
COMPRESSION TANK
(N.O.) K0514C
SOLENOID (N.C.)
P-E RELAY L404H (N.O.)
MIXING VALVE K0516C
(N.O.B-AB)
K0514C (N.O.)
CONVERTER
BALANCING COCKS
CHILLER BYPASS
WATER CHILLER
PUMP
P-E RELAY L404H
MAIN AIR
SYMBOLS
1. ~ STOP VALVE, MANUAL
2. ~ STOP-AND-BLEED AIR VALVE, MANUAL
3. N.O. ~ NORMALLY OPEN
4. N.C. ~ NORMALLY CLOSED
5. P-E ~ PNEUMATIC-ELECTRIC
6. E-P ~ ELECTRIC-PNEUMATIC
7. T ~ THERMOSTAT
8. H ~ HUMIDITY CONTROLLER
9. ~ AIR PRESSURE GAGE
10. ~ PNEUMATIC MOTORIZED VALVE
11. ~ AIR LINE
12. ~ STEAM OR WATER LINE OF THE PNEUMATIC CONTROL SYSTEM
SCHEMATIC INTEGRATED CONTROL SYSTEM
MECHANICAL ENGINEERING DEPARTMENT
VIRGINIA POLYTECHNIC INSTITUTE
BLACKSBURG, VIRGINIA
DRAWN BY R. S. GAY
DATE: SEPT. 19, 1959
APPROVED BY: C.H.L.
SCALE: NONE
PLATE 2