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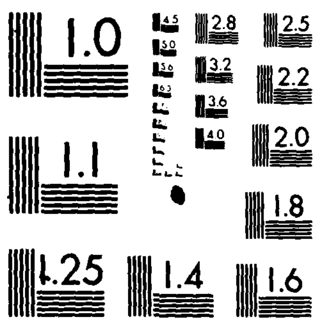

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TECHNICAL REPORT E-177
February 1982

VALIDATION DATA FOR MECHANICAL SYSTEM ALGORITHMS USED IN BUILDING ENERGY ANALYSIS PROGRAMS

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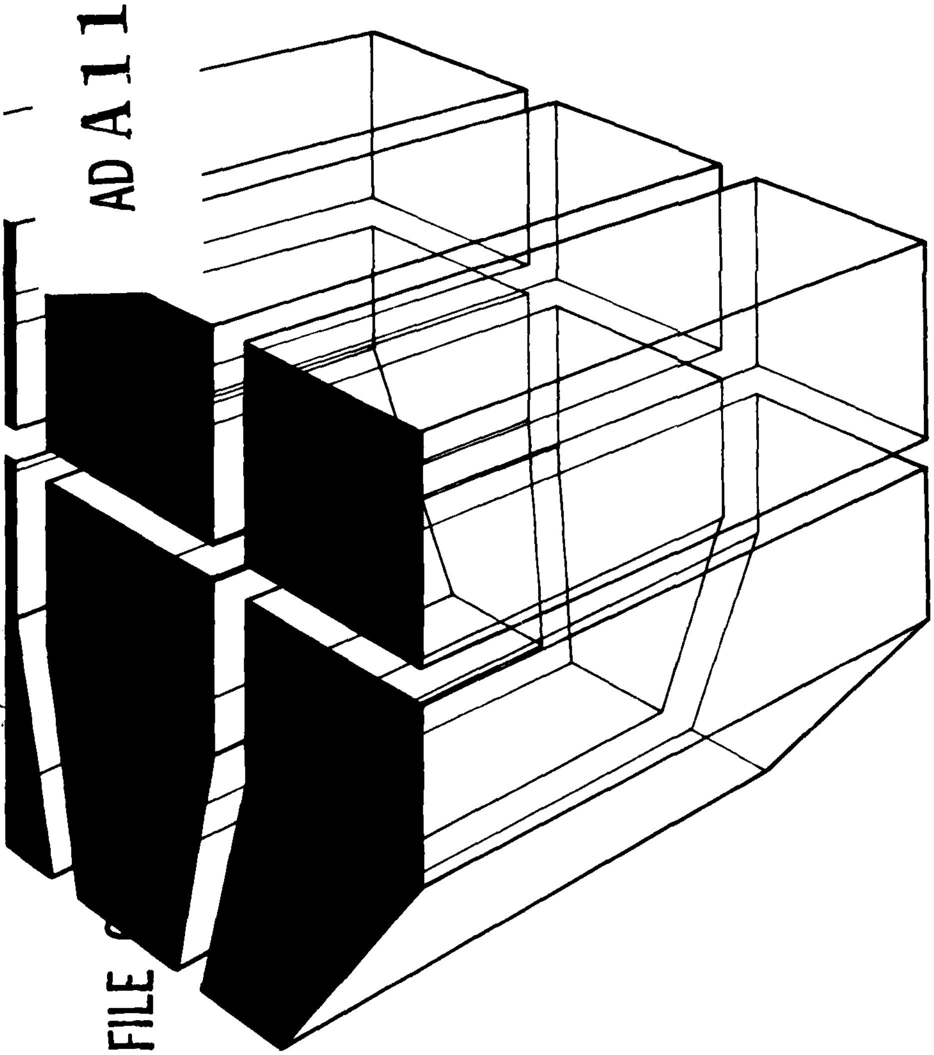

by
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This report describes a study which collected data developers can use to validate the algorithms used by building energy analysis computer programs to model system and central plant performance of typical heating, cooling, and ventilating (HVAC) systems. Building energy analysis programs compute (1) loads imposed on building environmental control systems for each zone, (2) loads imposed on central equipment (using the zone loads and system performance algorithms), and (3) the primary energy demanded by the central- and zone-specific components.

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When gathering data to validate the algorithms which model systems and plants, the precise loads imposed on the environmental control systems had to be determined. This was considered difficult to do if the systems being studied were operating in the field. For this reason, an experimental HVAC system was constructed at the U.S. Army Construction Engineering Research Laboratory (CERL). This experimental system housed four zones; loads within each zone were monitored and controlled. A number of HVAC systems were implemented to condition the experimental structure. Testing involved collecting all operating data for each system for a variety of zone loads. The output generated by system and plant algorithms can be used by developers to directly compare actual system performance (i.e., systems responding to known loads).

The information collected during this study made it possible to compare actual system performances with computer-generated results of system performances. In addition, this study identified some problems common to HVAC systems, including (1) often unreliable controls performance such as drifting away from calibration and poor repeatability, and (2) as-delivered components which do not operate at manufacturers' specifications.

FOREWORD

This investigation was performed for the Department of Energy (DOE), Division of Conservation, Solar Energy and Architectural and Engineering Branch, under contract IAA EM-78-1-01-4207. The DOE Technical Monitor was Mr. Howard Ross.

The work was performed by the Energy Systems (ES) Division of the U.S. Army Construction Engineering Research Laboratory (CERL). The following persons contributed substantially to this work: Larry Brand, Jim Weiner, Bruce Drolin, Chang Shon, Don Leverenz, and Nina Bergan. Dr. W. F. Stoecker of the University of Illinois, Department of Mechanical and Industrial Engineering, assisted in the planning and design of the experimental facilities. Mr. Richard Donaghy is Chief of CERL-ES.

COL Louis J. Circeo is Commander and Director of CERL and Dr. L. R. Shaffer is Technical Director.

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1 INTRODUCTION

Background

Escalating fuel costs, the uncertain availability of some energy sources, and the political overtones of the world energy market have created a need for building owners and operators, as well as Federal and State governments, to have accurate tools for predicting the energy performance of buildings.

Several large computer programs model the heat transfer characteristics of buildings and the equipment and components of heating, ventilating, and air conditioning (HVAC) systems. The Building Loads Analysis and System Thermodynamics (BLAST) program developed by the U.S. Army Construction Engineering Research Laboratory (CERL) and DOE-2, developed for the Department of the Energy[1] are such programs and are in the public domain. However, these programs have never been fully validated.

It now appears that Federal and State governments will take an active role in using building energy analysis programs to develop codes and standards to enforce energy conservative designs and to mandate extensive analyses of the options open to HVAC designers. Thus, a CERL-designed study, funded by the Department of Energy, was initiated to collect validation data for building energy performance (BEP) programs.

The many calculations of BEP programs are segmented according to particular modeling efforts. For example:

1. A loads section, which deals with building envelope heat transfer, infiltration and ventilation, and all other gains or losses contributing to the resultant heating or cooling loads.

2. A systems section, which integrates the building loads and the characteristics of the HVAC system and generates the demands imposed on its central plant components.

3. A plants section, which reduces the loads on individual components to actual energy usage by each piece of equipment.

Validation of the loads section has been reasonably successful due to a variety of experiments which compare actual conditions to the computer predictions. Performance data of individual components of the system are not always

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[1] D. C. Hittle, The Building Loads Analysis and System Thermodynamics (BLAST) Program Users Manual, Volumes I and II, Technical Report (TR) E-153/ADA072272 and ADA072273 (U.S. Army Construction Engineering Research Laboratory [CERL], June 1979); and DOE-2 Users Manual, Version 1.2 (Department of Energy, May 1980).

made available by the manufacturer, especially under part-load operation. System behavior is the area most open to scrutiny. The systems section of BEP programs can substantially affect the program results and, thus far, studies of systems effects have not been as extensive as other segments of BEP programs. Using data developed from the operation of an actual system along with the values of the loads the system is operating against will enable direct comparisons of the output of systems and plant algorithms to an actual systems performance.

Computer Modeling

BEP programs must be practical if they are to help reduce energy use. If, for example, a commercial office building heated and cooled with a large central system were to be modeled, the mathematics involved in system simulation would be complex. The solution would require many iterative processes; the results would be exact. The accuracy of the answer would be bounded by the accuracy of the models.

However, some clever shortcuts have been introduced into subroutines which model systems. These simplified algorithms substantially reduce the computational time and cost of systems modeling and are believed not to appreciably affect modeling accuracy. These abbreviated models allow timely and inexpensive computer simulations. Hence, designers of large structures can afford to base their final selections on BEP simulations of various design options.

Systems

One way to validate systems models is to quantify the difference between results from rigorous, extensive system modeling and the fast, efficient models incorporated in most BEP programs. Another way is to quantify the difference between systems models presently in use and the performance of actual systems. This latter approach was used by CERL in its BEP program validation study. This study was designed so the data of actual system performance would show the effects of (1) component part-load performance; (2) the nonlinearity, hysteresis, and poor repeatability of various controllers; (3) the cascading effects of a single component operating poorly; and (4) all the dynamic effects inherent to a multicomponent system.

Objective

The objective of this study was to collect performance data on a variety of HVAC systems and components to permit developers to evaluate the accuracy and correctness of algorithms used in building energy analysis programs.

Approach

This study had the following steps:

1. A full-scale experimental HVAC system was designed and constructed. This system could be reconfigured into a wide variety of system types and was fully instrumented. Heating and cooling loads on the system could be completely controlled.

2. For each of a variety of systems, steady-state experiments were performed to measure the performance of each system component and of the overall system responding to a variety of heating and cooling loads.

3. Results of the HVAC system tests were tabulated in sufficient detail to allow the authors of building energy analysis programs to compare measured system performance with the results of their simulation procedures.

4. Sample comparisons between measured performance data and simulation results were made to establish an appropriate procedure for using the data produced in Step 3.

Outline of Report

Chapter 1 summarizes the problems associated with validating building energy analysis programs and presents the strategy of the validation effort described in this report.

Chapter 2 describes the design and engineering of the experimental facility.

Chapter 3 gives the results of testing individual components within the system.

Chapter 4 gives the results of the data collection to be used for validation work.

Chapter 5 highlights the unexpected difficulties encountered during the set-up and operation of the test HVAC system.

Appendix A contains the manufacturers' specifications for the various components used in the HVAC system. This information can be used to build input files for building energy analysis programs.

Appendix B is a collection of various algorithms. These algorithms were gathered early during the experimental design to ensure that enough validation data would be collected by the test facility.

2 EXPERIMENTAL DESIGN

Overview

The general design criteria for the HVAC system tested during this study were to reflect as closely as possible a typical system in the field operating under reasonable loads and experiencing a variety of outdoor conditions.

The decisions made in the design and engineering of the HVAC test facility were based on the following considerations:

1. The HVAC system of the test facility should be capable of representing a typical commercial-scale system.

2. The selection procedures and engineering guidelines used to design the HVAC system should be based on popular engineering practice.

3. The assembly and operation of the HVAC system should be typical in every respect.

However, some concessions, not believed to affect the representativeness of the system, were necessary because of practical construction limitations. These will be discussed in Chapter 5.

HVAC System Configuration

Central air-handling systems are used to condition multiple areas -- each area is subject to a unique load profile. The number of zones per air handler is constrained only by practical limits. A four-zone system was selected for the experimental apparatus as representative of a typical commercial central air-handler system.

It was recognized that the performance of central station air handlers could be evaluated without coupling the air handler to a multiple zone assembly. Rather, a simplified approach could be taken, where the air handler would be subjected to various inlet conditions, and the output would be monitored and reported. However, it was believed something could be learned from an experimental facility which actually housed a multiple zone structure with conventional thermostats which reponded to zone conditions.

The zones were loaded by supplying heating or cooling energy directly to the air within the zone. To do this, each zone housed a heating and cooling fan/coil unit and was coupled to a central air-distribution system. The zones were loaded by the fan/coil units which were controlled by the experimenter; the air-distribution systems responded to this load under zone thermostat control. For fan/coil system experiments, this process was reversed. That is, the load was supplied by the air-distribution system (as controlled by the experimenter), and the fan/coil units responded to this load under room thermostat control.

Size Requirement

Two factors influenced the sizing of the experiment and its various components:

1. The air system and energy delivery rates were sized so as to be representative of systems found in practice.

2. The system was small enough so the total four-zone experiment, with all its mechanical equipment, fit into CERL's high-bay area. The following set-up was chosen: a four-zone heating and cooling air system, with each zone having a design maximum heating load and cooling load of 36,000 Btu/hr (10.5 kW). The mock four-zone building conditioned by the HVAC system was sized at 150 sq ft (14 m^2) per zone with a ceiling height of 10 ft (3 m). This work structure was small compared to areas having similar cooling loads, but necessary if the experiment was to fit within the allocated space.

Zone Design

The four zones were arranged in an L-shape (Figure 1) and were identical in design. The zones were designed to allow enough space for the air streams from the fan/coil units and supply air to mix well. They were constructed to minimize (1) heat transfer to other spaces, and (2) infiltration. The zones were 15 X 10 X 10 ft high (4.6 X 3 X 3 m). Heating and cooling fan/coil units were placed in the center of each zone. The air inlet into each zone was on the ceiling and the return air outlet was on the wall 12 in. (0.3 m) above the floor. The air supply and return positions were located to insure that the fan/coil conditioned air and room supply air mixed well. The thermostats (which sensed zone air temperature and controlled the HVAC system) were located just above the return air outlet. Each zone has a 4- X 4-ft (1.2- X 1.2-m) airtight access port.

All walls were 2- x 6-in. (0.05- X 0.15-m) stud construction, 16 in. (0.4 m) on center, and had 4 in. (0.10 m) of fiberglass insulation between studs. The floor and ceiling were of similar construction, but had vermiculite poured between joints. A vapor barrier was attached on the outer surface of the four zones, just under the surface paneling. The exterior of each zone was covered with 5/8-in. (0.016-m) plywood. The interior walls and ceiling were finished with 5/8-in. (0.016-m) wall board. The zone floor and roof were 5/8-in. (0.016-m) plywood.

Built-Up Air Handler

The built-up air handler provided conditioned air at regulated temperatures to meet the zone's heating or cooling loads.

Internal gains within each zone were generated by two fan/coil units which supplied a maximum of 36,000 Btu/hr (10.5 kW) of heating or cooling; the cooling and heating capacity of the air handler had to be sufficient to meet these internal gains plus the load due to the introduction of outdoor air. The cooling-coil design load assumed an outdoor air ventilation rate of 400 cfm (0.19 m^3/s) for each zone; that is, a maximum total outdoor air-flow rate of 1600 cfm (0.75 m^3/s). The design condition for outdoor air was 94°F (34.5°C) dry bulb and 78°F (25.6°C) wet bulb. The design condition for the return air was 75°F (23.9°C) dry bulb and 68°F (20.0°C) wet bulb. The water temperature rise through the cooling coil in the air handler was designed at 10°F (5.6°C) when the inlet water temperature was 45°F (7.2°C).

The heating coil was designed to raise the temperature of 6400 cfm (3.0 m^3/s) of air -- a blend of 4800 cfm (2.25 m^3/s) of air at 68°F (20.0°C) and 1600 cfm (0.75 m^3/s) of air at 0°F (-18°C) -- from 51 to 92°F (10.6 to 33.3°C); the coil was supplied with 180°F (82.2°C) water. The maximum outdoor air rates were 400 cfm (0.19 m^3/s) per zone or 1600 cfm (0.75 m^3/s) totally.

The fan in the air handler was fitted with variable inlet vanes to control the capacity.

The air handler mixing box was set so at maximum flow the outdoor air fraction was 0.25. At reduced flow rates this outdoor air fraction remained relatively constant.

The air handler, which was purchased commercially, was a blow-through type. It had a six-row, 16 fin per inch cooling coil; a two-row 16 fin per inch heating coil; forward-curved fan blades; inlet vanes; and a 10-hp (7.4-kW) motor. The mixing chamber was a combination mixing chamber/filter bank (Figure 2).

Ventilation Requirements

The central station air handler drew air through a mixing box used to regulate the proportions of outdoor and return air. It was of interest to control the conditions of the outdoor air, as seen by the experimental air handler, over a wide variation, irrespective of the actual outside weather conditions. To do this, the outdoor air was passed through an Outdoor Environment Simulator (OES), which permitted heating, cooling, and some humidity control of actual outdoor air. The OES also allowed the outdoor weather conditions, as seen by the air handler, to remain constant despite changes in the outdoor weather conditions.

The OES

The input to the air handler was either return air from the zones, outdoor air, or a mixture of these air sources. So outdoor air conditions could be controlled, the experiment included an OES between the actual outdoor air intake and the inlet to the air handler. The OES consisted of a cooling coil, a heating coil, and a humidifier; its purpose was to simulate outdoor conditions other than actual conditions.

The OES had four separate functions, but operated at any or all of them simultaneously (though this was not usually desirable). It was a humidifier, dehumidifier, heat source, and heat sink. In the summer, the OES simulated summer, spring, and fall conditions between 50 and 100°F (10 and 38°C) over a wide range of relative humidity. In winter, it simulated winter, spring, and summer conditions between the given outdoor air temperature and 100°F (38°C), in addition to elevating the relative humidity. Simulating winter air during the summer was considered too costly and was thus overlooked. With this limitation in mind, the OES components were chosen as follows:

1. The cooling coil was a six-row coil with a face dimension of 33 X 55 in. (0.84 x 1.4 m). It was designed to accept 6400 cfm (3.0 m^3/s), 67°F (19.5°C) wet-bulb air and extract energy at the rate of 940,000 Btu/hr (27.7 kW), with a sensible to total heat ratio as low as 0.6. The chilled water supply temperature was 45°F (7.2°C); this water temperature rose 10°F (5.6°C) through the coil.

2. The heating coil was a two-row coil with a face dimension of 18 x 57 in. (0.46 x 1.4 m). It was designed to accept 6400 cfm (3.0 m^3/s) of air at 51°F (10.6°C), and heat to 90°F (32°C) when supplied with 180°F (82°C) hot water.

3. The humidifer was a single tube "Dri-Steam" type. It could use 15 psi (103.4 kPa) steam to humidify air from 90°F (32°C), 15 percent relative humidity (RH) to 90°F (32°C), 80 percent RH using about 130 lb (59 kg) of steam per hour.

Duct Design

A duct system was designed to deliver air from the OES to the air handler, from the air handler to the four zones, and return the air from the zones to the air-handler mixing box. The ducts were sized using typical guidelines of recommended velocities not exceeding 2000 fpm (10 m/s) in the mains and 1200 fpm (6 m/s) the branches.

Figure 3 shows the final design. Note that (1) the OES could condition either outside or lab air, or it could be bypassed altogether; (2) each zone and the hot deck had a venturi in the duct to determine the flow rate; and (3) the ducting to each zone was geometrically similar to other zones, eliminating the need for balancing dampers (as was verified by the zone air flow venturis). The ducts were constructed of galvanized sheet metal; and the venturis were made of welded carbon steel.

Air-Delivery System

Each zone was supplied with conditioned air ducted from the air handler to meet the heating and cooling loads supplied by the fan/coil units. The design of the distribution system was such that it could operate as a constant volume or a variable air volume (VAV) system in either a dual duct or single duct mode, with an option for terminal reheat.

VAV Operation

The system used two identical VAV boxes for each zone, one on the cold air supply, and one on the hot air supply. The VAV boxes were located just before the reheat coil. The boxes were designed to deliver a maximum 1600 cfm (0.75 m^3/s). A pneumatic volume regulator (PVR) was used to control the VAV boxes so that a specific air-delivery rate, corresponding to a pneumatic signal from the thermostat, was maintained irrespective of fluctuating air velocity and upstream pressure.

The dual duct VAV system was designed to provide heating and cooling capabilities to the same zone simultaneously. Each box was designed to allow 1600 cfm (0.75 m^3/s) of conditioned air at maximum load. The control strategy inverted and shifted the signal from the thermostat to the VAV box on the hot deck, producing the proper control of two air boxes under signal thermostat control.

Constant Volume System

Two constant volume systems were tested: (1) a single duct system with reheat, and (2) a dual duct system which blended warm and chilled air.

The reheat system operation required closing the air boxes on the hot deck and blocking the hot deck entirely at the air handler. Zone thermostats controlled the three-way mixing valves which supplied hot water to the reheat coils.

The dual duct system tested in this study did not employ constant volume mixing boxes, but rather a simpler mixing box made by locking two dampers out of phase. The zone thermostat dictated whether the supply air came (1) totally from one deck, (2) from various mixtures from both decks, or (3) totally from the other deck.

These simpler mixing boxes are typical of the hardware used until recently, when constant volume mixing boxes became available. The flow through the simple mixing box is subject to variation depending on the demands of other zone mixing boxes; however, zone temperature control remains satisfactory.

Boiler Hot Water Plant

The hot water supply plant provided heat for one or more of the following purposes: (1) a cooling load for the zones, (2) the heating coil in the central air handler, (3) the heating coil in the OES, and (4) the reheat coils. The design included a total of 144,000 Btu/hr (42.2 kW) for the zone loads, heating coils, or reheat coils. The remaining heating capacity was used by the OES.

The boiler/hot water plant had one boiler with one steam to water converter situated above it. Heating energy from the steam was transferred by the converter to circulating hot water. The heat exchanger was designed for a water temperature rise of 40°F (22°C), a supply water temperature of 180°F (82°C), and a flow of 30 gpm (1.89×10^{-3} m^3/s). A heat exchanger rated at a 40 gpm (2.52×10^{-3} m^3/s) flow rate and 44°F (22.2°C) temperature rise or a 10 gpm (6.30×10^{-4} m^3/s) flow rate and 150°F (83°C) temperature rise was selected.

A package steam boiler provided steam to the heat exchanger. The rated output of the unit was 360,000 Btu/hr (102.5 kW). The rated input was 450,000 Btu/hr (132 kW).

The boiler control had a pressure stat which sensed the steam pressure and controlled the gas valve. These controls were entirely electric. The steam/hot water converter received a modulated flow of hot water to accept heat from the steam. Modulation, under automatic control, attempts to maintain the temperature of the circulating hot water (Figure 4).

Chiller -- Reciprocating Compressor

A nominal 20 ton (70 kW) capacity chiller was selected to accomplish system cooling. This unit used R-22 as a refrigerant. The motor was a hermetic three-phase 480-V inductive type. The four-cylinder reciprocating compressor was fitted with three-stage cylinder unloaders.

Cylinder unloading was controlled by a three-stage aquastat located on the chilled water inlet. Four cylinders operated when inlet water was at the aquastat setting or above. When inlet water was about 2°F (1°C) below the setting the aquastat deenergized two of the four cylinders. When inlet water was about 4°F (2°C) below the setting the aquastat deenergized three of the four cylinders. The machine cycled off when inlet water fell to about 6°F (3°C) below the setting.

Cooling Tower

A cooling tower was selected based on its ability to reject the heat exchanged in the condenser of the chiller. As part-load operation was usual, and as weather conditions varied, a tower with capacity control was desired to produce a more constant return water temperature.

A nominal 360,000 Btu/hr (105.5 kW) cooling tower was selected for this study. The unit controlled capacity by means of scroll dampers under aquastat control (Figure 5). The unit delivered full capacity at the rated conditions of 78°F (25.6°C) wet-bulb temperature, 95°F (35°C) entering water temperature, and 85°F (29.5°C) leaving water temperature.

Water entered the tower at the top, fell through the spray nozzles, and collected in a trough at the bottom. Air was forced upward against the falling water to promote evaporation. The tower was complete with an automatic make-up valve and a variable bleed rate valve. Air forced through the tower was regulated so as not to cool the water below the nominal setting on the water temperature controller.

Fan/Coil Design

Each zone had one heating and one cooling fan/coil unit to provide a heating and cooling load during one phase of the experiment, and to control room temperature during another phase.

The design criteria for the heating fan/coil units called for a unit that would provide a 3-ton (10.5-kW) zone cooling load while providing enough air flow to ensure good mixing. The supply water temperature was 180°F (82°C) and the zone air temperature was 70°F (21°C). The water flow rate necessary to transfer energy at a rate of 36,000 Btu/hr (10.5° kW) with a water temperature drop of 40°F (21.1°C) was 1.8 gpm (1.1×10^{-4} m^3/s). Fan/coil units selected to meet these criteria circulated 150 cfm (0.07 m^3/s) of conditioned air.

The design criteria for the cooling fan/coil units called for a unit which would provide a 3-ton (10.5-kW) zone heating load while providing a high-volume flow to mix with the low-volume flow of hot air supplied by the air handler or terminal reheat system.

Unit ventilators were selected to meet these criteria, accepting a flow of less than 10 gpm (6.3×10^{-4} m^3/s) water at 45°F (7.2°C) and heating it to 55°F (12.8°C) while cooling the air. The circulation rate was 600 cfm (0.28 m^3/s).

Piping Layout

The piping system was designed to provide heated and cooled water to the fan/coil units, the air handler, and the OES coils. Design considerations included space limitations, accessibility of sensors and valves, maximum fluid velocities, and minimum pressure drops. The final design used a piping bus consisting of parallel supply and return lines for both hot and chilled water. The bus was positioned about 12 ft (4 m) above the floor. At each piece of equipment, the piping dropped to the appropriate level and ran horizontally for a sufficient length to satisfy the requirements of the venturis, thermometer wells, and mixing valves. Venturis were placed in the supply line to each apparatus which transferred heat to or from water, and resistance temperature measuring devices were placed at each water coil inlet and exit (Figure 6).

Copper piping was selected for all plumbing because it was cost effective. Closed-cell foam pipe insulation was fitted over all piping.

Control Station

The control station had two major functions: (1) it allowed the viewer to get a general picture of what was occurring within the system at any particular point and moment; and (2) it was the central data collection location. The control station included seven sections (Figure 7). Each was a 6-ft (1.8-m) high by 3-ft (0.9-m) plexiglass panel supported on an aluminum frame. A low work area built on the front side of each panel extends to the rear and became a shelf for transducers and instruments.

Four identical panels, one for each zone, were dedicated to monitoring flow sensing devices and control valve pressures. These devices included: one manometer for the air venturi, three manometers for water venturis, one gage for thermostat pressure, and two gages for mixing valve controller pressure signals to zone fan/coils.

A fifth panel held two manometers, one connected to the hot deck venturi and one measuring static pressure increase across the fan.

A sixth panel was a manometer bank. It displayed the differential pressures relating to the water flow through all the venturis not directly connected with zone coils. These included flow through the chilled water and condenser water bundles within the chiller, the OES hot and cold coils, the air-handler hot and cold coils, the converter, and all pumps.

The seventh panel contained the pneumatic controllers associated with this experiment, such as the zone thermostat resets. To accommodate the multipurpose nature of this project, pneumatic three-way switches were mounted at this panel. These switches changed the controlling strategies of the HVAC system. Air temperatures controlled by water coils were adjustable from this location.

Measurement Techniques

This experiment required seven types of measurement: water temperature, air temperature, water flow, air flow, electric power, differential pressure, and relative humidity. The technique for measuring these quantities is described below.

Temperature Measurements

Both water and air temperature measurements were made using four-wire platinum probes of the resistance temperature device (RTD) type. The probes were inserted into the center of the flow stream. The water probes were encased in stainless steel to protect them from water damage.

The temperature was measured by measuring the resistance of the platinum wire probe (which varied in a prescribed manner as a function of the probe temperature). The actual resistance of a platinum probe at any one temperature is a property of the physical dimensions of the platinum wire probe. The change in resistance of the probe with temperature, however, is only a function of temperature. This temperature dependence, which can be expressed as a quadratic equation, is known to five significant figures; thus, highly accurate temperature measurements can be made using platinum probes. Table 1 shows the variation in resistance of a platinum wire probe which has a resistance of 100 ohms at 32°F (0°C) over the temperature ranges from 32 to 212°F (0°C to 100°C), which was the range of interest for this experiment. Table 1 is expressed as a quadratic function relating temperature to resistance:

$$R = 100\ (1 + 3.909 \times 10^{-3}\ (T) - 5.87 \times 10^{-7}\ (T)^2) \qquad \text{[Eq 1]}$$

where T is the temperature of the probe (°C) and R is the resistance of the probe. Since the coefficient of T and T^2 in Eq 1 are functions of platinum and not the size of the probe, Eq 1 can be written in the following form for probes with other than 100-ohm resistances at 32°F (0°C):

$$R = R_0(1 + 0.003909\,(T) - 5.87 \times 10^{-7}\,(T)^2) \qquad \text{[Eq 2]}$$

where R_0 is the actual resistance of the platinum probe at 32°F (0°C). For each probe used in this experiment, the value of the resistance at 32°F (0°C) was measured to within 0.01 ohms. Using Eq 2 and the values of R_0 allows for the calculation of the fluid temperature from the measured probe resistance to $\pm$ 0.1°F (0.05°C), provided the probe resistance can be measured to within 0.1 ohm. Rewriting Eq 1 for T as a function of R, Eq 2 can be written as:

$$T = (-0.003909 + ((-0.003909)^2 - 4(5.87 \times 10^{-7})(1 - \frac{R}{R0}))^{1/2})$$

$$(2(5.87 \times 10^{-7}))^{-1} \qquad \text{[Eq 3]}$$

Resistance Measurements

Because of the high degree of accuracy desired for this experiment, the platinum probes were used in a four-wire resistance measurement configuration. In this measurement, a known current is passed through the platinum probe through one set of wires. A second set of wires is used to read the voltage drop across the platinum wire probe induced by the known current. The voltage is read by a high impedance meter; thus no appreciable current passes through the leads used for the voltage measurement, and the problems associated with lead-wire resistance are eliminated. The digital voltmeter within the data logging system was specifically set up to make these four-wire resistance measurements. The current source and the voltage measurements were all made within the voltmeter and converted directly to resistance. The accuracy of this measurement was 0.01 percent of full scale.

Flow Measurements

Both air and water flow measurements were made by incorporting venturis in the flow stream at the appropriate points (Figure 8). The flow of water or air through the venturi caused a pressure drop between the inlet and throat of the venturi proportional to the square of the rate of flow. This pressure differential was input by pneumatic lines to a differential pressure transducer which converted the pressure differentials to a direct current (DC) voltage. The DC voltage was then read by the digital voltmeter. Tees were inserted in the pneumatic lines before the pressure transducers, allowing the manometers at the control panel to monitor the venturi pressures. The flow rate through the venturi as a function of the pressure drop between the inlet and the throat of the venturi is given in Eq 4. Each venturi was bench-tested four times. The test results were regressed on to determine the constant C. A micrometer was used to measure the two characteristic diameters of the venturis. The value of C was computed analytically and compared to the experimentally determined value. Close agreement was observed.

$$\text{Flow} = C\sqrt{\text{diff.pressure}} \qquad \text{[Eq 4]}$$

Electric Power Measurements

Electric power was measured using a watt transducer and appropriately placed potential and current transformers (PT and CT, respectively). For single-phase power measurements, the CT was placed on the power line supplying the instrument being measured. It provided a current proportional to the current passing through it. The current from the CT and the voltage from the PT were then input to a watt transducer which produced an output current proportional to the power consumed by the monitored device. This current passed through a known resistance; the voltage drop was measured by the digital voltmeter. For three-phase power measurements, a three-phase power transducer was used in a similar arrangement as single-phase measurements. Again, the output from the transducer was a current proportional to the three-phase power being consumed by the device to which the PTs and CTs were connected. This current passed through a known resistance and the voltage drop was measured by the digital voltmeter. The power consumed by the device being measured (p) is given by the expression:

$$p = k \times v \qquad \text{[Eq 5]}$$

where v is the voltage drop across a known shunt resister read on the digital voltmeter and k is the scaling factor relating the voltage drop across the known register to watts.

Differential Pressure Measurements

Differential pressure was measured using a pressure transducer in the same manner as the differential pressure was measured across the venturis in the air and water flow tests. The two pneumatic lines from the two pressures for which the differential was required were run to a differential pressure transducer. The transducer then produced an output voltage proportional to the difference in the two pressures on its input. Therefore, the pressure differences can be expressed as

$$p = k \times v \qquad \text{[Eq 6]}$$

where v is the voltage reading on the digital voltmeter and k is the scaling factor from the pressure transducer.

Humidity Measurements

Relative humidity was measured using a thin film capacitance humidity probe which produced an output voltage from 0 to 100 millivolts (mV) proportional to the relative humidity of the air in the vicinity of the probe. The voltage output was read by the digital voltmeter. The percent RH is found from the equation

$$rh = 1000 \times v \qquad \text{[Eq 7]}$$

where v is the voltage reading on the voltmeter.

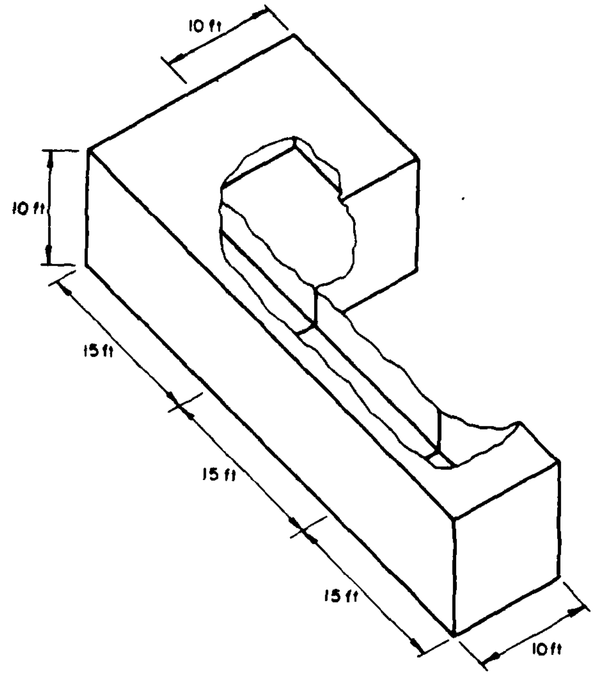


Figure 1. L-shape arrangement for the four-zone experiment. (Metric conversion: 1 ft = 0.3 m.)

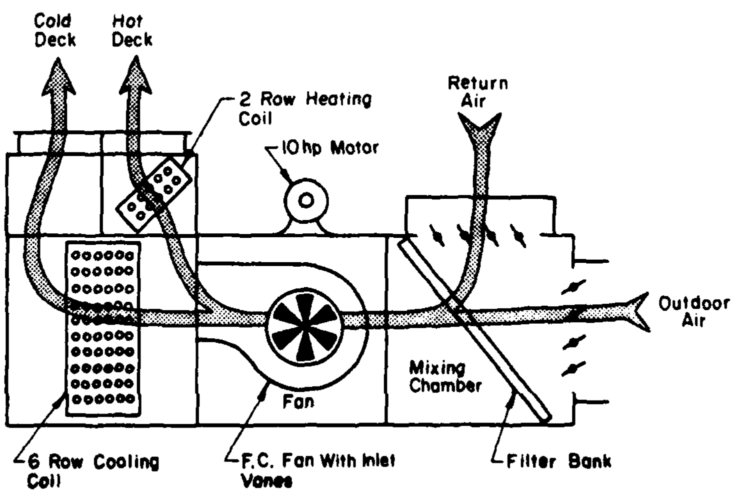


Figure 2. Central station air handler.

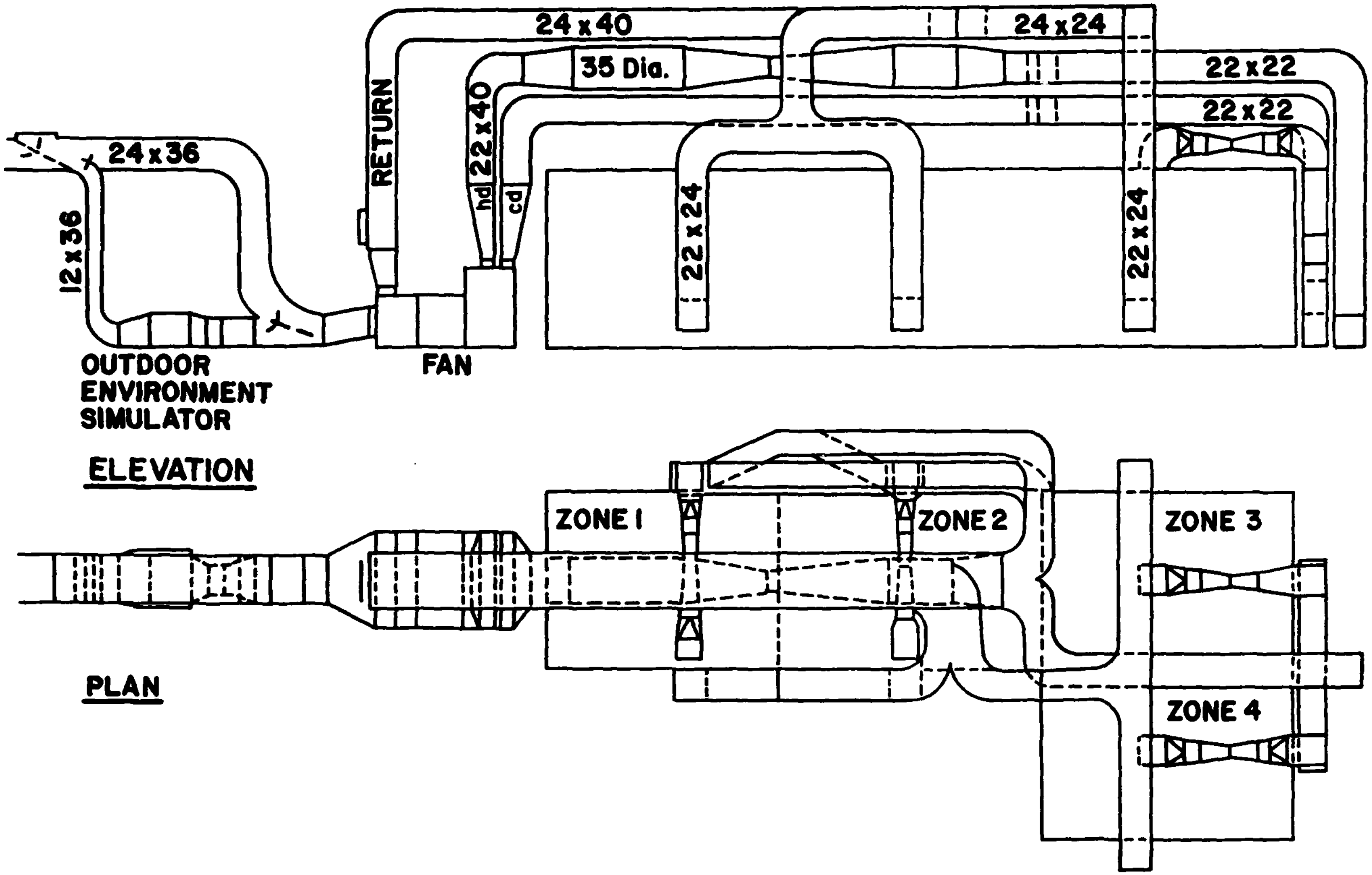


Figure 3. Ductwork.

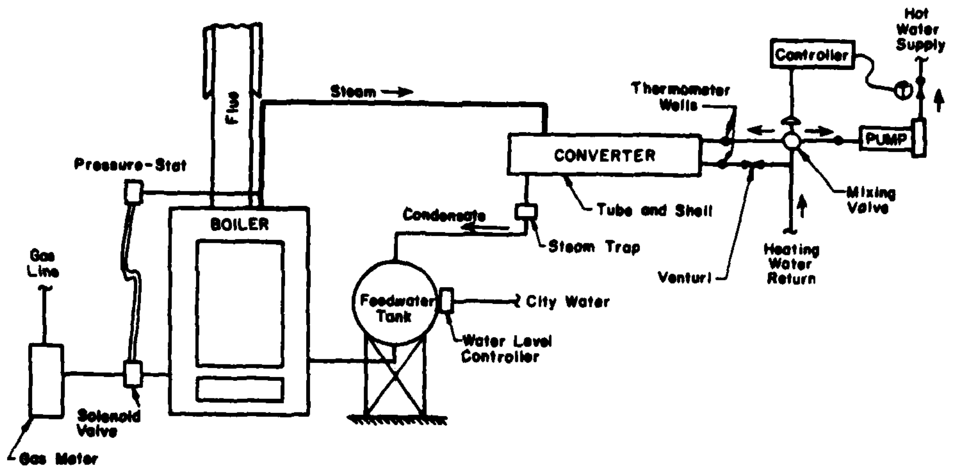


Figure 4. Heating water system.

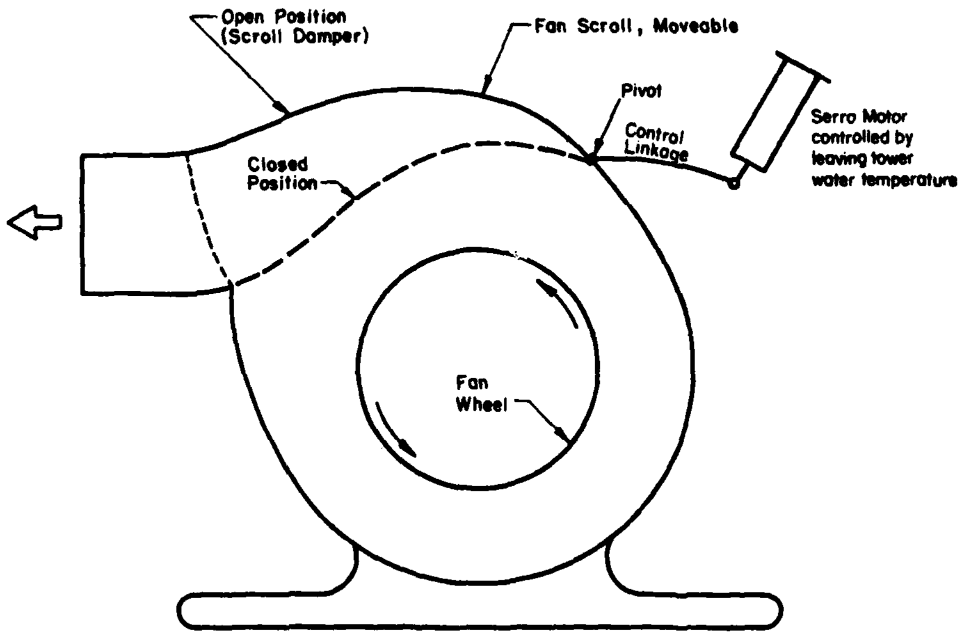
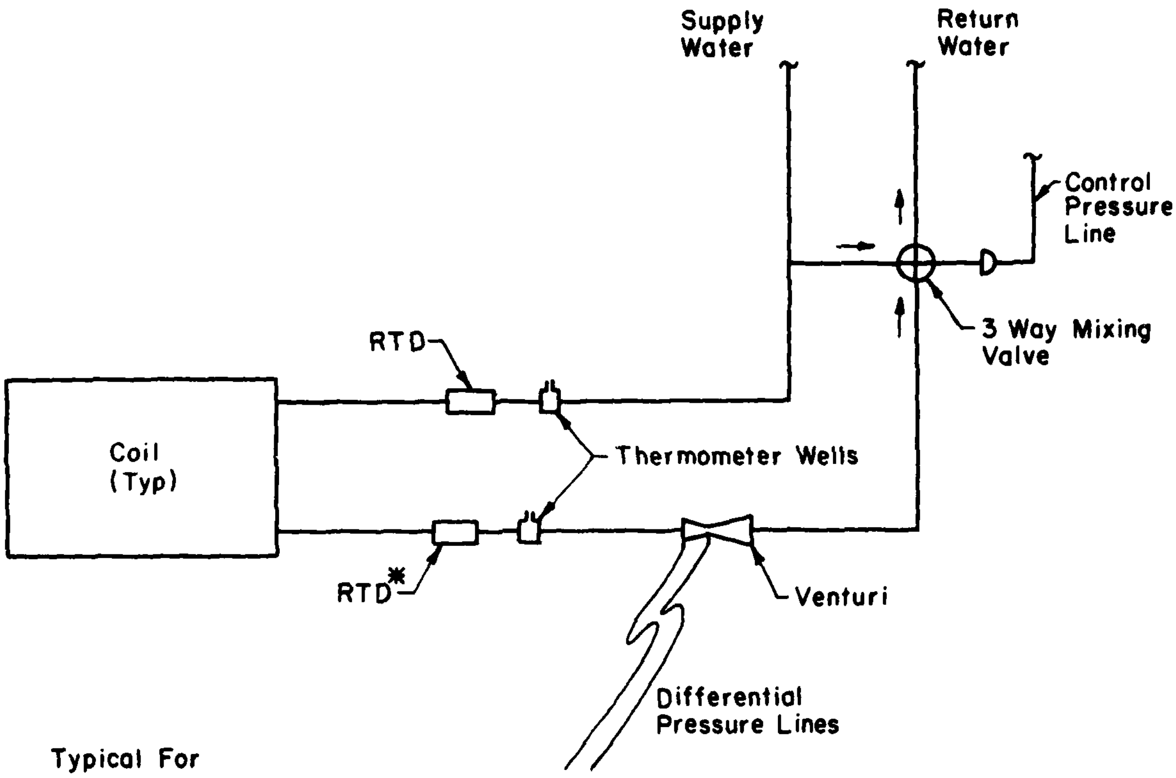


Figure 5. Fan with scroll dampers.



Typical For

- Air Handler Coils
- OES Coils
- Fan Coils
- Reheat Coils
- Converter Tube Bundle
- Chiller Condenser and Cold Water Tube Bundles

* RTD - Resistance Temperature - Measuring Device

Figure 6. Piping for heat exchanging devices.

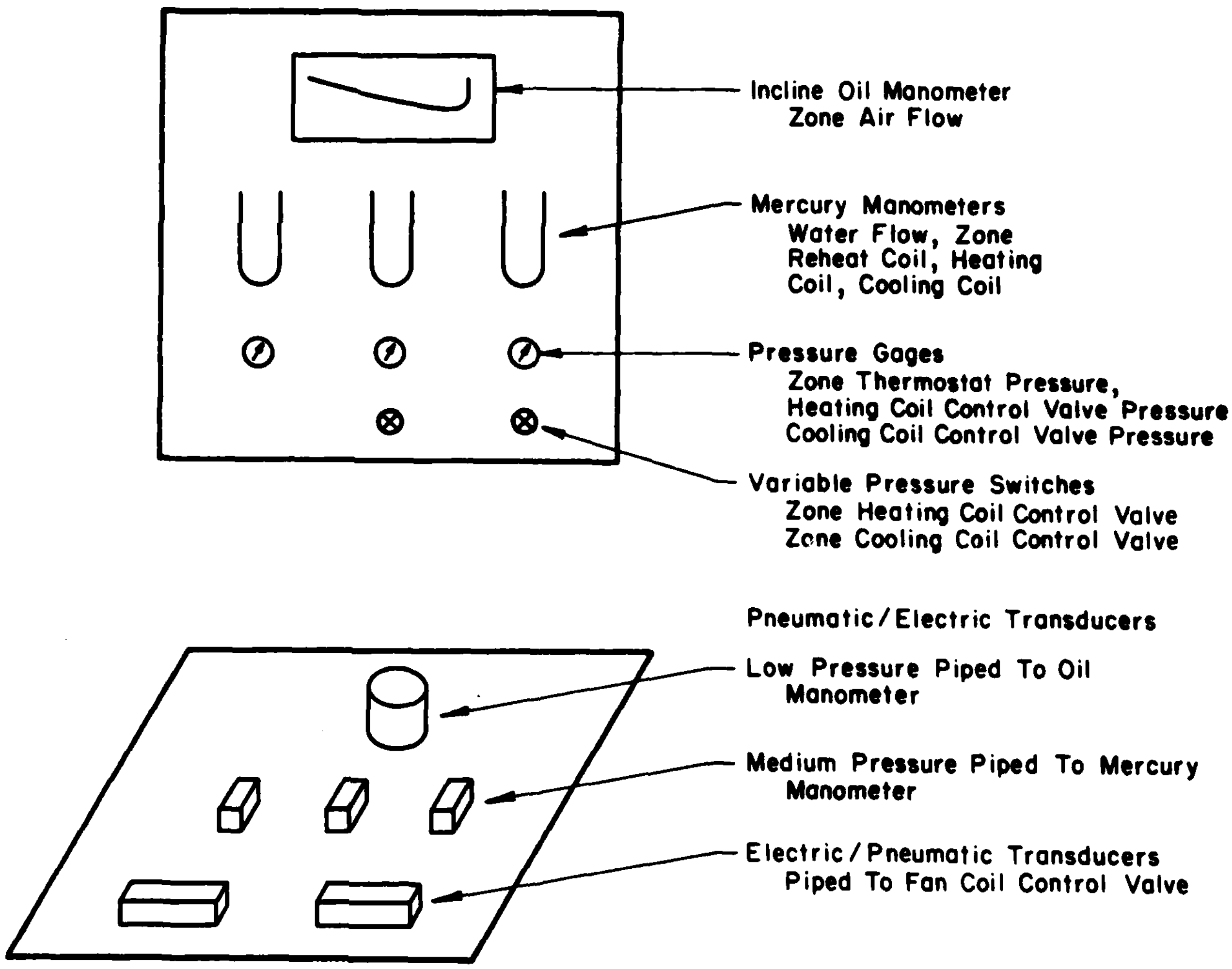


Figure 7. Typical one-zone control station.

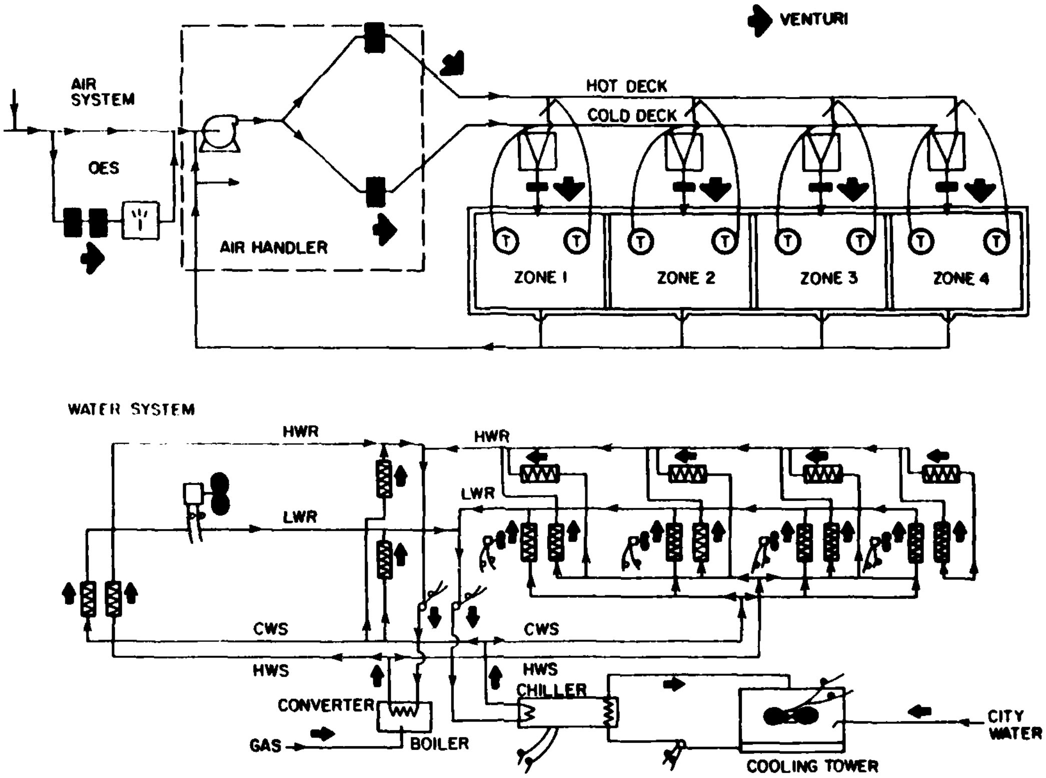


Figure 8. Placement of venturis.

Table 1

Platinum Wire Probe Resistance Variation

Temperature (°C)	Resistance (ohms)
0	100
10	103.90
20	107.79
30	111.67
40	115.54
50	119.40
60	123.24
70	127.08
80	130.90
90	134.71
100	138.50

3 COMPONENT TESTING

This chapter contains various component tests completed at the test facility. Generally, the objective of this experiment was to vary the input(s) and report the output. There were components such as the chiller where precise control of all inputs was not possible. In cases such as this, the component test reported whatever input(s) occurred and the corresponding output.

Test 1 -- Fan Performance of a Fan With Inlet Guide Vanes Operating Within a Built-Up Air Handler

Objective

1. To test the fan sufficiently to generate a family of fan curves corresponding to different positions of the inlet guide vanes.

2. To observe the inlet guide vanes operating under automatic control.

Description

The fan used in this test was a forward-curved blade, 16-in. (0.4 m) wheel centrifugal fan outfitted with inlet guide vanes. (Inlet guide vanes rotate the air entering the fan in the direction of fan rotation, causing less resistance to the fan's movement and reducing the power necessary to keep the fan rotating.)

Initially, the rotative speed of the fan was set by means of an adjustable pulley to make sure the fan capacity could deliver the maximum design flow, which, for this system, was 6400 cfm (3.0 m^3/s). The static pressure measured across the fan at design flow was 2.2 in. (590 kPa) water column (wc). To generate fan curves, the dampers within the ductwork system were gradually closed and the flow rate, static pressure, and electrical power were recorded over the entire range of the damper positions (wide open to fully closed). Five fan curves were generated, one having the inlet vanes wide open, one having the inlet vanes fully closed, and the remaining three having the inlet vanes between the extreme positions.

Two automatic inlet vane controllers were tested: pneumatic proportional control and electronic reset control.

Pneumatic proportional control involved three pieces of hardware: (1) a differential pressure transmitter which output 3 to 15 psig (20 to 103 kPa) linearly proportional to 0 to 5 in. (0 to 1250 Pa) wc differential pressure input, (2) a pneumatic motor fixed to the inlet vane control shaft, and (3) a receiver/controller with an adjustable gain and offset which received inputs from the differential pressure transmitter and a set-point pressure source, and output a signal to the pneumatic motor.

Electronic reset control involved four pieces of hardware: (1) a pneumatic to electric pressure transducer which output 0 to 5 V linearly proportional to 0 to 10 in. (0 to 2500 Pa) wc input, (2) an electric to pneumatic transducer which output 0 to 15 psi (0 to 103 kPa) corresponding to 0 to 10 V input, (3) a pneumatic motor fixed to the inlet vane control shaft, and (4) an electronic analog circuit (Figure 9) which increased or decreased the electrical output to the electric to pneumatic transducer controlling the inlet guide vanes until the electrical input corresponding to the differential static pressure was at the desired value.

Results

The fan performance curves are in Figure 10. Figure 11 shows fan efficiency over the five fan curves in Figure 10. Figure 12 shows the fan performance under automatic inlet guide vane control for both pneumatic proportional control and electronic reset control.

Test 2 -- Boiler Operation, Capacity, Efficiency, and Stand-By Losses

Objective

To measure the steady-state output of the boiler, the boiler efficiency, and the magnitude of the stand-by losses.

Description

The test procedure was as follows:

1. Using the test facility, a load was imposed on the boiler that caused boiler pressure to remain constant just below the pressure at which the controller closed the gas valve. The steady-state output at the converter (q_{conv}) was recorded along with the rate of gas consumption (q_{gas}).

2. With the flow of water through the converter completely stopped and the boiler energized, the gas valve was monitored and the lengths of time the valve remained opened and closed were recorded. (This condition is called idling or operating at no load.) After many cycles, the data were used to calculate the fraction of the total cycle time (x) the gas valve remained open to meet the stand-by load. The remaining fraction of time could then be equated to the output measured at the converter in Step 1.

3. Heat recovered from combustion (q_{out}) (stand-by loss [q_{loss}] plus converter output) ($q_{out} = q_{conv} + q_{loss}$) was calculated by dividing the output at the converter by the fraction of time the gas valve was closed during Step 2 ($q_{out} = q_{conv}/x$).

4. Boiler combustion efficiency was calculated by dividing the heat recovered from combustion by the recorded rate of gas consumption from Step 1 (q_{out}/q_{gas}).

5. Stand-by losses or the idling load was calculated by multiplying heat recovered from combustion by the fraction of time the gas valve remained open from Step 2 ($q_{loss} = q_{out}$ (1-x)).

Results

Table 2 shows the results of the boiler testing.

Discussion

The unit nameplate ratings were: 450,000 Btu/hr (132 kW) input, 360,000 Btu/hr (106 kW) output. These ratings implicitly claim 80 percent conversion efficiency (conversion efficiency refers to energy from gas to steam). CERL testing measured an output of 250,000 Btu/hr (73 kW) at an efficiency of 78 percent. Testing occurred with a boiler pressure of 5 psig (34.4 kPa). The nameplate rating probably applies to a boiler pressure of 0 psig (0 kPa), which would account for the small deficiency in the unit efficiency. However, boiler output measured only 70 percent of the nameplate value. The hot water system consisting of the boiler and a steam to water converter had a standby load of 24,000 Btu/hr (7000 W).

It is doubtful a boiler such as this operating in the field would ever be monitored. Rather, it would operate efficiently over its useful life without alerting anyone to the fact it operates at about 70 percent of nameplate capacity. It is equipment such as this boiler which has caused many engineers to use large safety factors when sizing HVAC equipment.

Test 3 -- Chiller/Cold Water Generator

Objective

To observe the performance of the chiller operating at maximum-rated load and through the full range of part-load operation.

Description

The chiller test involved seven measurements:

1. Flow rate of chilled water through the chiller.

2. Inlet temperature of the chilled water.

3. Outlet temperature of the chilled water.

4. Flow rate of condenser cooling water through the chiller.

5. Inlet temperature of the condenser water.

6. Outlet temperature of the condenser water.

7. Electrical power demanded by the chiller.

The flow rates of condenser water and chilled water through the chiller were precisely adjusted to duplicate catalog rating conditions. The load on the chiller was reflected in the temperature of the returning chilled water. The load was controlled using the capabilities of the HVAC system.

Testing involved 20 trials, each of which subjected the chiller to a different load. The range of loads tested was between a no-load state and the maximum obtainable output.

The chiller unloads accordingly: (1) its four compressor cylinders are active when the return water temperature is at or above the nominal chilled water set point, (2) two cylinders are deactivated when return water temperature falls below 2°F (1.1°C) of the nominal setting, (3) three cylinders are deactivated when return water temperature falls 4°F (2.2°C) below the nominal setting, and (4) the machine cycles off when return water temperature falls 6°F (3.3°C) below the nominal setting. In summary, the chiller operates nominally at 100, 50, or 25 percent of the maximum capacity.

Unless the load on the chiller is exactly the capacity of one-, two-, or four-cylinder operation, the machine automatically cycles between two of the four possible operating modes.

Results

During part-load testing, the seven test measurements were sampled every half minute as the chiller cycled between operating modes. The duration of the test was selected after observing the results of the 30-s data samples; that is, a period having many cycles and ending so no fractions of a cycle remained. Averages of the seven measurements were taken over the test duration. The cooling load and power demand were normalized to the full load and full-load power demand. Test results are given in Figure 13 and in Table 3.

Discussion

The cold water generator, as installed, required a substantial amount of work before the unit operated near the catalog capacity or efficiency. The unit was advertised as bench tested before leaving the factory; however, according to the manufacturer, this particular chiller was assembled before the manufacturer's bench-testing policy. The superheat of the refrigerant was measured at 25°F (14°C). The measurement was taken with thermocouples mounted to the compressor inlet and just downstream of the expansion valve. The temperature difference measured at these two points should have been 5°F (2.8°C), as recommended by the manufacturer's personnel. After the expansion valve was adjusted (resulting in the recommended 5°F [2.8°C] temperature difference), the unit capacity and efficiency dramatically increased to a value just under catalog values, but satisfactory considering tolerances allowed in the catalog and in the resolution of the instrumentation.

One of two solenoids located within the compressor head which disable the inlet valves (thus accomplishing compressor unloading) never worked correctly. The instrumentation of the HVAC test facility brought this problem to the surface and the manufacturer replaced the faulty part under warranty.

When the machine was initially energized -- after an extended period of rest which allowed both the chilled water and the condenser water loops to come to thermal equilibrium with the surroundings -- the compressor accomplished more refrigeration than the motor was designed for. The machine's capacity was 20 tons (70 kW), nominally, when the exiting chilled water was 44°F (6.7°C) and the entering condenser water was 85°F (29°C). When the machine ran with 70°F (21°C) inlet chilled water and 70°F (21°C) inlet condenser water, the unit capacity was 24 tons (84 kW), nominally, which, after several minutes of operation, caused the motor thermal protection controller to deenergize the motor. This problem was corrected by installing a controller which would sense inlet chilled water temperature and prevent the chiller from operating at full capacity (energizing one of the two unloading solenoids) until the inlet chilled water temperature dropped below 60°F (20°C). This was not a factory-installed controller.

Test 4 -- Hot Water Piping Losses

Objective

To quantify the rate energy is transferred from the hot water piping network to the surroundings.

Description

It was of interest for this test to ensure that the various heat exchangers tied into the hot water system did not contribute to the test measurement; that is, of interest was the steady-state rate of heat transfer from the piping. To do this, the three-way mixing valves which controlled the flow of hot water to each heating coil in the system were set to totally divert hot water from each heating coil and recycle water back to the hot water generator. In addition, all fans which forced air through the heating coils in the system were deenergized.

During this test, the three-way diverting valve at the steam to hot water converter was fixed at various positions. The hot water pump was energized and data were collected after a steady-state condition was observed. Measurements of flow through the converter as well as the temperature difference of the water entering and leaving the converter were taken to determine the steady-state rate of heat input.

Results

Test results are given in Table 4. A value of 0.088 MBtu/hr°F (0.046 kW/°C) was used for a mean UA* product for the piping network.

Discussion

One difficulty occurred during the measurement of hot water through the converter -- the flow rate was very low. The accuracy of flow measurement with venturis declines at very low rates. There is a discrepancy between the results obtained on 28 January 1980 and those obtained on 11 February 1980. This was probably caused by a small air bubble entrapped in the lines connecting the taps on the venturi and the manometer/transducer taps. The slight air bubble would cause a small pressure difference across the manometer, thus, affecting the flow calculation. At typical flow rates during testing, this entrapped air bubble would have had an insignificant effect on the accuracy of the calculation of flow; however, at very low flow rates, appreciable error

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* For a definition of UA, see Table 4, p 54.

would result. For this reason, the results of testing on 28 January were not considered. The value of 0.088 MBtu/hr^{o}F (0.046 kW/^{o}C) was derived from the test results on 11 February 1980.

Test 5 -- Enthalpy Logic Device

Objective

To observe the performance of the enthalpy logic device.

Description

The enthalpy logic device had four pneumatic inputs: outdoor air temperature and relative humidity, and return-air temperature and relative humidity. The output was either "low" (0 psi) for 100 percent outdoor air, or "high" (15 psi [103 kPa]) for minimum outdoor air.

The device was intended to allow 100 percent outdoor air to be introduced at the mixing box if the outdoor air was cooler than the return air and if the outdoor air enthalpy was less than return air enthalpy.

Four cases were tested covering all possible combinations of return air and outdoor air temperature and enthalpy. Testing was accomplished by controlling the pneumatic inputs to the device and observing the output.

Results

Test results are given in Table 5.

Discussion

The enthalpy logic device did not operate properly. Had it worked correctly, another serious shortcoming would have probably caused less than satisfactory performance; that is, the accuracy of the four transmitters coupled to the device. Component testing of the temperature transmitters revealed that, as installed, these transmitters did not follow the precise temperature to pressure relationship advertised by the manufacturer. The relative humidity transmitters were observed to output grossly different values when tested in a common environment. (For example, in an environment measured to be 62 percent RH, the output of the three pneumatic humidity transmitters was as follows: 63, 88, and 96 percent RH.) The discrepancies found in the pneumatic temperature and humidity transmitters made it unrealistic to expect the enthalpy logic device to discern precise states of enthalpy.

Test 6 -- Zone Thermostats

Objective

To observe the performance of the four zone thermostats as installed.

Description

1. All four thermostats were set at 70°F (20.9°C) and the pneumatic output recorded.

2. A set screw within the thermostat was adjusted so all four thermostats output 10 psi (68.9 kPa) in a 70°F (20.9°C) environment.

3. The zone temperatures were changed slowly, and the thermostat outputs recorded to observe the thermostat gains. (Since the gain and the set point are not physically related in the device, the gains observed are "from the factory" values.)

Results

Test results are given in Figure 14 and Table 6.

Discussion

The gain settings, as received from the factory, were relatively close. The gain adjustments are accomplished by sliding a pivot within the thermostat; however, to set a precise gain is very difficult. The zone thermostats were precisely adjusted using the instrumentation and flexibility of the test facility. It was of interest to have various throttling ranges for testing purposes. But, based on experience gained from this adjustment work, it would not be reasonable to task service personnel to make similar adjusts in the field because of the sensitivity of the adjustment and the lack of precise control over the surrounding environment.

Figure 14 shows the behavior of the four zone thermostats, two of which are linear and two slightly parabolic. These four graphs were generated using the data in Table 6 by a best fit, least squares approximation to a second-order polynominal. It should not be considered significant that two of the four thermostats are not represented by a near linear relation. All four thermostats are satisfactory for HVAC control applications.

Test 7 -- Pneumatic Control Drift

Objective

To document the problems of drifting experienced with the fluidic receiver/controller.

Description

During this test, six receiver/controllers were calibrated and the various gains and set points were documented. One week after the initial calibration, the state of the control devices was investigated. Testing was accomplished by uncoupling the receiver/controllers from the input signal (temperature transmitters, humidity transmitters, etc.) and recoupling a variable pressure source to the input (Figure 15). Mercury manometers were tied to both the input and output ports of the receiver/controllers and the output pressure was recorded for various input pressures.

Results

Test results are given in Table 7.

Discussion

Unfortunately, the problems (i.e., drifting off calibration) experienced with all six receiver/controllers were never corrected. Personnel from the control manufacturer inspected the facility and found no abnormalities in the equipment, set-up, or operation. (The receiver/controllers were calibrated on a daily basis when the test facility was operating, which generally resulted in their satisfactory performance.)

Test 8 -- Pneumatic Temperature Transmitters

Objective

To observe the performance of various pneumatic temperature transmitters over the applicable range of temperatures.

Description

This test monitored: (1) the output pressure of the transmitters with pressure/electric transducers, and (2) the surrounding temperature at the transmitter. The local environment was gradually heated or cooled and the profiles recorded.

Results

Test results are given in Figure 16.

Discussion

The particular temperature transmitters used in this test were the "one-pipe bleeding" variety. Their pressure/temperature sensitivity can be fine tuned using a sliding pivot. The absolute range of operation is fine tuned by turning a set screw. These transmitters are advertised as factory calibrated. The position of both adjustments set at the factory is identified by a dab of paint. As shown in Figure 16, the transmitters were not delivered precisely calibrated. The graphs of the transmitter performance make it apparent that transmitters operating in the field are not precisely calibrated. However, since receiver/controllers are adjustable and capable of whatever input-output linear relation is necessary for control action, no problems resulted from the lack of precision of the temperature transmitters.

Test 9 -- The Effects of the Total Cooling Coil Load on a Single-Zone Temperature Controller

Objective

To observe if the relationship between zone temperature and cooling effect changes when the total load on the cooling coil changes significantly.

Description

The test was identical to the VAV zone temperature controller test (Test 11), except two trials were recorded, one where the total load on the cooling coil was low, and another where the total cooling coil load was high.

Results

Test results are given in Figure 17.

Discussion

The test results are better explained by noting the cold deck temperature controller test (Test 10). An increased load on the cooling coil, first reflected by an increase in the cold deck temperatures, caused the controller to divert more chilled water to the coil. Often the new cold deck temperature was cooler than the original cold deck temperature, but the controller hysteresis prevented another correction of the chilled water flow rate. This is primarily responsible for the trend observed in this test.

The anticipated result of this test was that an increased total cooling load would elevate the cold deck temperature and reduce the cooling effect of the zone temperature controller. The opposite was observed. Another factor contributing to the increased cooling effect as the total cooling load increased was the usual variation of room temperature controller performance from one trial to the next.

Test 10 -- Cold Deck Temperature Controller

Objective

To observe the performance of the cold deck temperature controller over a wide range of loads imposed on the cooling coil.

Description

The cooling coil load was controlled by controlling the air temperature entering the coil. The air flow rate through the coil remained near constant. The following data were recorded: air temperature entering and leaving the coil and air flow rate through the coil. Testing involved 13 trials which ranged from light loads to full load (defined as the load which caused the cold deck controller to supply the cooling coil with maximum chilled water).

Results

Test results are given in Figure 18.

Discussion

This controller behaved in a nonpredictable manner while giving reasonable control over the cold deck temperature (within 2°F [1.10°C]). The controller gain was set as high as possible while still operating in a stable manner. Hysteresis in the chilled water mixing valve and the poor repeatability of the receiver/controller both contributed to the poor correlation between the cold deck temperature and the cooling coil load. Chilled water varied during the test and further contributed to the noncorrelatable relation observed. The chiller operated under a three-stage automatic control. When the load on the chiller was between the capacities of two sequenced stages (for example, the chiller operated at 100, 50, or 25 percent nominal operation, and a load of 75 percent), the chiller would cycle between the two bordering stages. This, in turn, caused the chilled water temperature to vary several degrees during steady-periodic operation, further affecting the cold deck temperature. The cycling period was on the order of 5 min.

Test 11 -- VAV Zone Temperature Controller

Objective

To determine the behavior of a zone temperature controller as the zone load traveled from its minimum to maximum extremes.

Description

The zone temperature controller (Figure 19) encompassed:

1. A pneumatic thermostat (acting as a temperature transmitter).

2. A pneumatic volume regulator.

3. An air box or valve.

Each room thermostat output a pneumatic signal directly proportional to the zone temperature according to the following expression:

$$P_{out} = (T_{zone} - T_{sp})\,(g) + c \qquad \text{[Eq 8]}$$

where:

P_{out} is the output pneumatic signal
T_{zone} is the zone temperature
T_{sp} is the set-point temperature
g is the thermostat gain
c is the thermostat output when zone temperature is at the setting.

The volume regulator was a reset control device. Air flow through the VAV box was monitored by the volume regulator by means of a pitot probe at the air box outlet. The volume regulator modulated the air box to satisfy the following relationship between the air flow through the box and the thermostat signal:

1. If $P_{stat} < 8$ psi, flow = minimum setting

2. If $P_{stat} > 13$ psi, flow = maximum setting

3. If 8 psi $< P_{stat} <$ 13 psi, then

$$\text{flow} = \text{max flow}\ ((1-(\text{min fraction})^2(P_{stat} - 8))/5 + (\text{m hr fraction})^2)^{1/2} \qquad \text{[Eq 9]}$$

where P_{stat} is thermostat pressure (psi).*

The temperature of the supply air was 55°F (12.6°C), nominally, during this test. The cold deck temperature should be influenced by a changing zone load since the change in the zone load affects the return air temperature; however, this particular cold deck controller did not respond to the changing zone load.

Results

Test results are given in Figure 20.

Discussion

The linear relation between cooling effect and zone temperature is the result of two effects. First, as zone temperature increases, the rate of cold air increases proportional to the square root of increasing temperature. This is caused by the performance of the PVR (Figure 21), which positions the cooling air damper to match the thermostat pressure to the pitot tube pressure differential. Second, as the zone temperature increases, the temperature difference between the supply air and the zone air temperature increases, thus increasing the cooling effect. For this test, supply air remained near constant at 55°F (12.6°C). Zone temperature varied from 75 to 80°F (23.6 to 26.4°C), corresponding to a change in thermostat pressure from 8 to 13 psi (55 to 89 kPa). Figure 18 shows the combined effects of the PVR, proportioning flow to the square root of increasing temperature, and the increasing temperature differential between supply air and the conditioned space.

In general, the pneumatic controls on the test facility did not display good repeatability and were observed to drift out of calibration over a period of several days. Of eight pneumatic volume regulators installed, two were inoperative and all eight were installed incorrectly, causing improper control. When the facility controllers were replaced and the installation problems corrected, these pneumatic volume regulators were the most repeatable pneumatic controls on the test facility.

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* 1 psi = 6.894 kPa.

The room thermostats performed satisfactorily over months of testing. The thermostat outputs corresponding to zone temperature were somewhat stocastic. The general pressure temperature relationship of the thermostat remained stable, but some random drift was always present, which caused problems in reporting thermostat settings. These thermostats had very sensitive gain adjustments, which made the task of setting specific throttling ranges difficult.

Note the near linear relationship of the zone temperature and cooling effect in Figure 20. Programs such as BLAST and DOE-2 assume a linear relationship for this algorithm.[2]

Test 12 -- Control Valves and Control Air Boxes

Objective

To record the flow rate through (1) a VAV air box and (2) a three-way mixing valve regulating conditioned water to a coil.

Description

The test procedure for the VAV air box was as follows: A variable pressure source was monitored by a data acquisition system and connected to the pneumatic motor on the air box. The air flow through the air box was also monitored. The variable pressure source was increased slowly and incrementally from 0 psi (0 kPa), while air flow and control pressure were recorded. When the control pressure well exceeded the range of the pneumatic motor, the process was repeated while decreasing the control pressure.

The test procedure for the three-way mixing control valve was identical to the air box test, except the variable control pressure was connected to the valve accuator, and the water flow directed through the coil was monitored.

Result

The relationship of the control pressure (signal) and the flow rates (controlled response) are presented graphically in Figures 22 and 23.

Discussion

As expected, nonlinearity, and hysteresis are present in the relationship of control pressure and the controlled flow rate. Most programs which model HVAC control systems assume linear, hysteresis-free performance of control devices responding to pneumatic signals.

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[2] D. C. Hittle, The Building Loads Analysis and System Thermodynamics (BLAST) Users Manual Volumes I and II, TR E-153/ADA072272 and ADA072273 (CERL, June 1979), and DOE-2 Users Guide Version 2.1 (Department of Energy, May 1980).

Test 13 -- Variable Speed Fan Drive

Objective

To record the performance of a retrofitted variable speed drive powering an air handler fan.

Description

The variable speed drive assembly (a variable width pulley-belt device) was designed to be mounted over the existing fan and motor shafts. The width of the pulley on the motor shaft was controlled by a servo motor, the width of the pulley on the fan shaft changed with changing belt tension.

Testing involved recording standard fan curves (static pressure vs flow) and curves of electrical power vs flow for various rotative speeds of the fan.

Results

The performance of the variable speed drive operating at four distinct rotative speeds is presented in Figures 24 and 25.

Discussion

A problem developed in the operation of the drive system. The manufacturer emphasized good pulley alignment was necessary for extended belt life and vibration-free service. This drive system was accurately aligned at the initial setup with proper tension on the belt where the motor drive is at the smallest size and the spring-loaded, driven pulley was at its maximum position. However, as the device increased the fan speed, the belt tension changed. This caused the entire motor-adjustable pulley assembly to flex (the motor base is mounted to the fan cabinet, which displaces elastically when stress imposed by the motor mounts increases); this upset the alignment. At the elevated fan speed, the drive system continued to perform. However, excess belt wear was observed, the driver pulley center (allowed to float axially on the fan shaft) became too hot to touch, and motor power substantially succeeded the power requirements of an equivalent fixed speed drive. It was not possible to stiffen the motor support, so it was not conclusive whether the poor performance was due to this alignment problem. At low fan speed (900 rpm), the 3600-cfm (1.7-m^3/s), 1.2-in. (300-Pa) wc, 3.5-kW operating point has an overall efficiency of 14 percent. At 1300 rpm, the 7200-cfm (3.4-m^3/s), 2.4-in. (600-Pa) wc, 9.6-kW operating point had an overall efficiency of 21 percent. Overall efficiency includes motor losses. This same fan motor enjoyed overall efficiency greater than 40 percent with a fixed pulley drive.

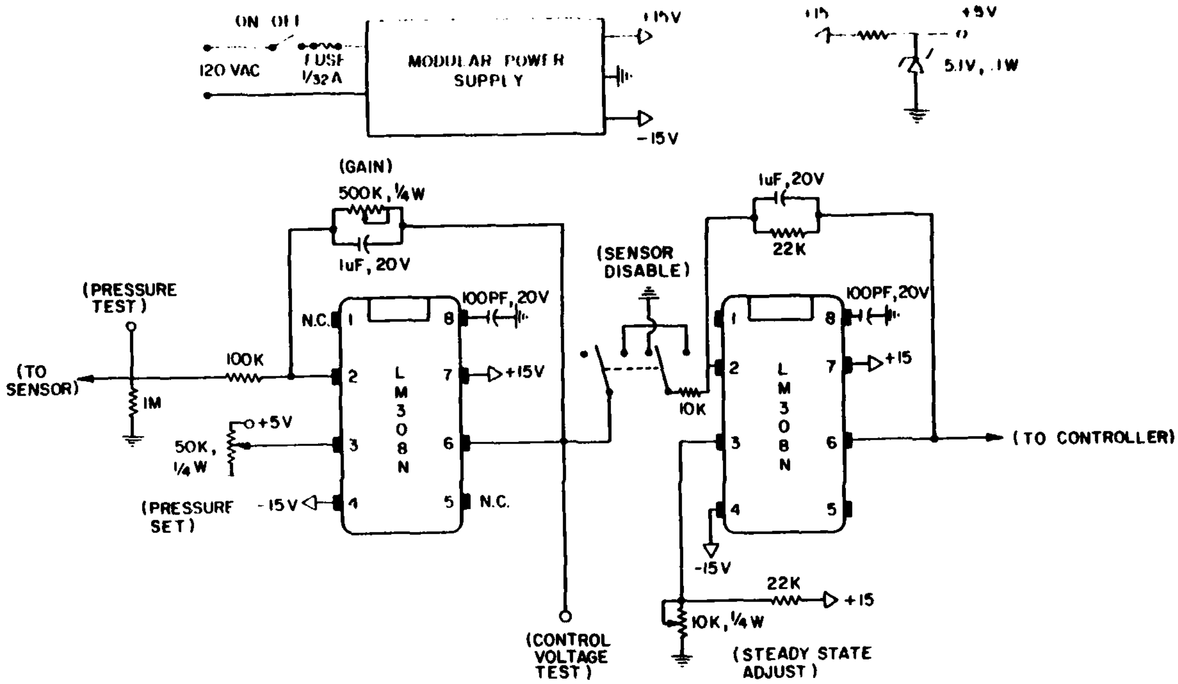


Figure 9. Control schematic -- inlet vane reset controller.

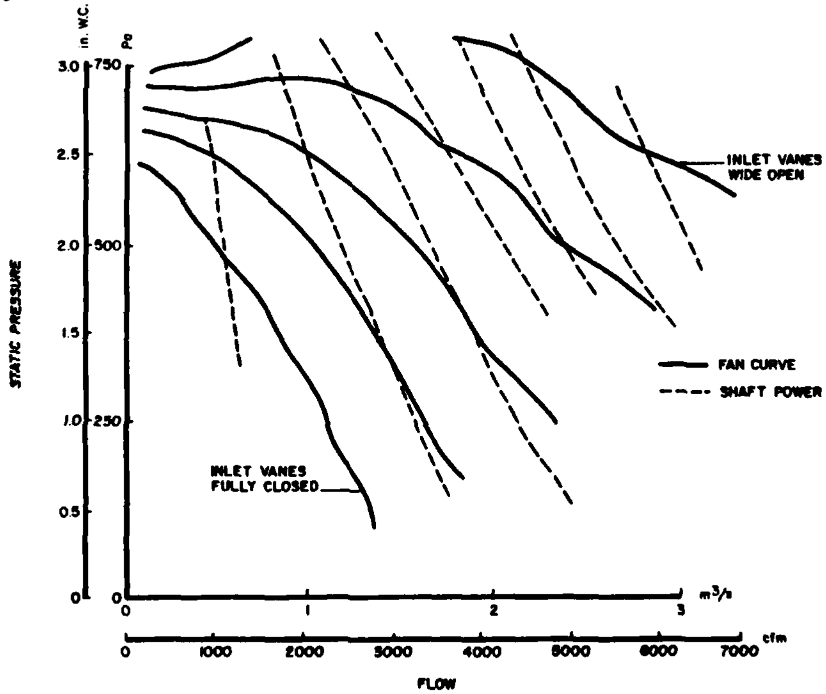


Figure 10. Fan curves for inlet vane equipped fan -- inlet vanes at various positions.

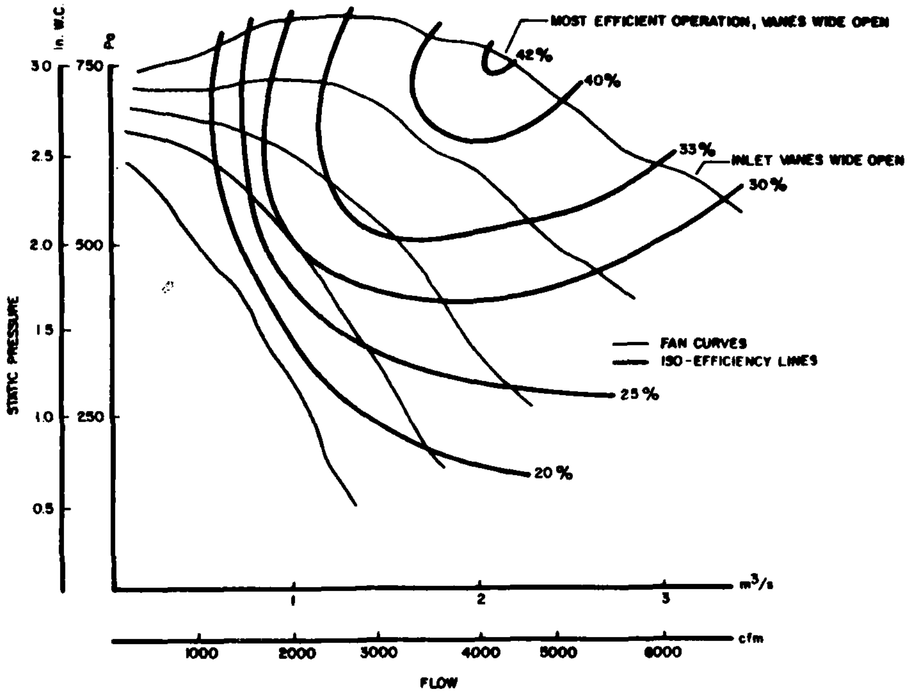


Figure 11. Fan efficiency.

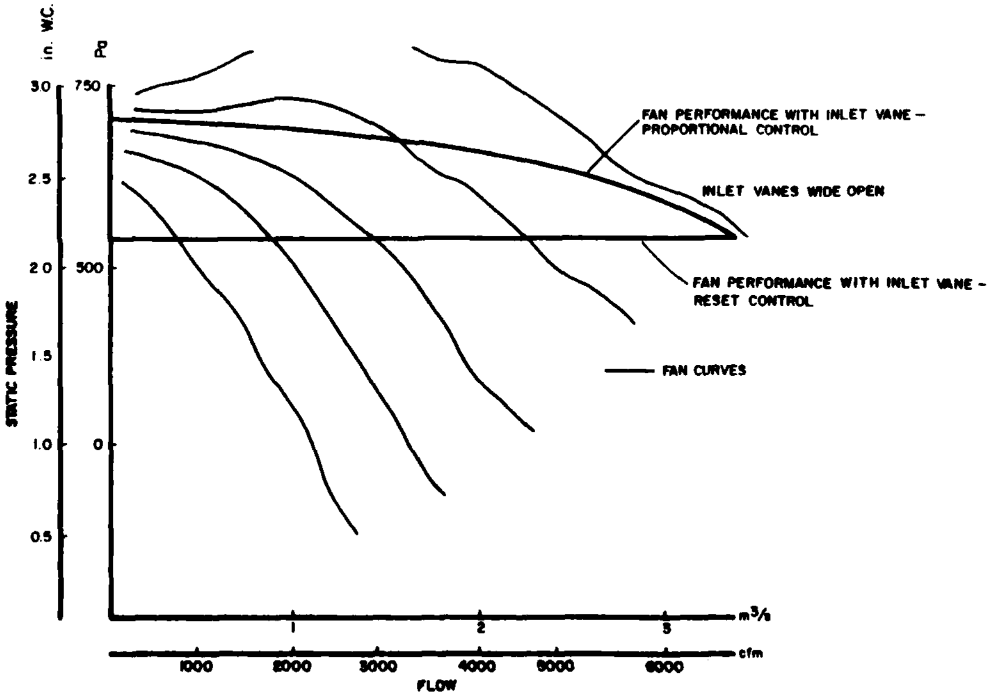
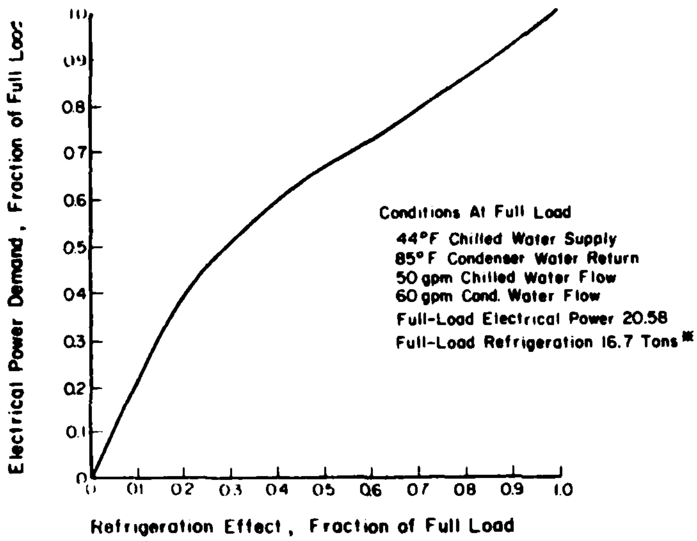


Figure 12. Fan curves and fan operation under automatic control.



* Testing Prior To Lowering Refrigerant Superheat From 25°F To 10°F. Chiller COP Measured 2.74 Prior To Adjustment of Expansion Valve, and 3.17 After the Adjustment. Part-Load Operation Did Not Change Trends.

Figure 13. Chiller part-load operation. (Metric conversion: [°F-32]/1.8 = °C.)

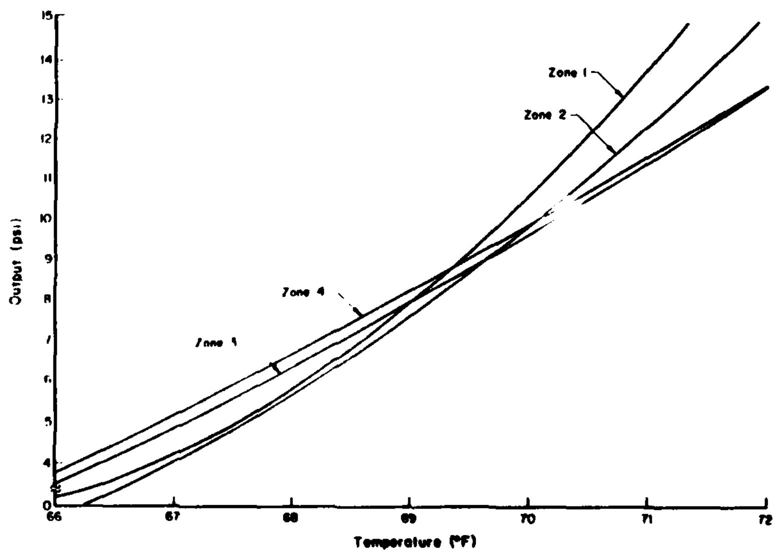


Figure 14. Zone thermostat.

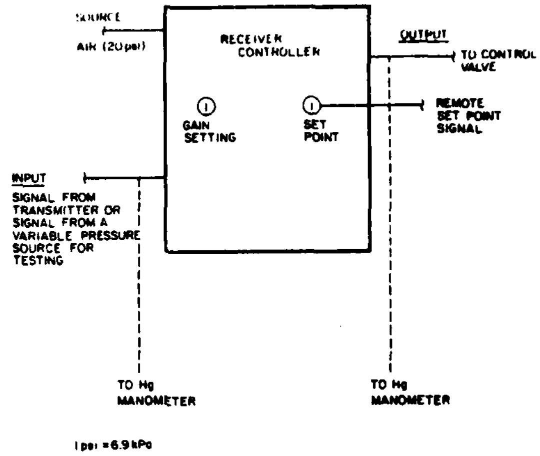


Figure 15. Receiver controller testing.

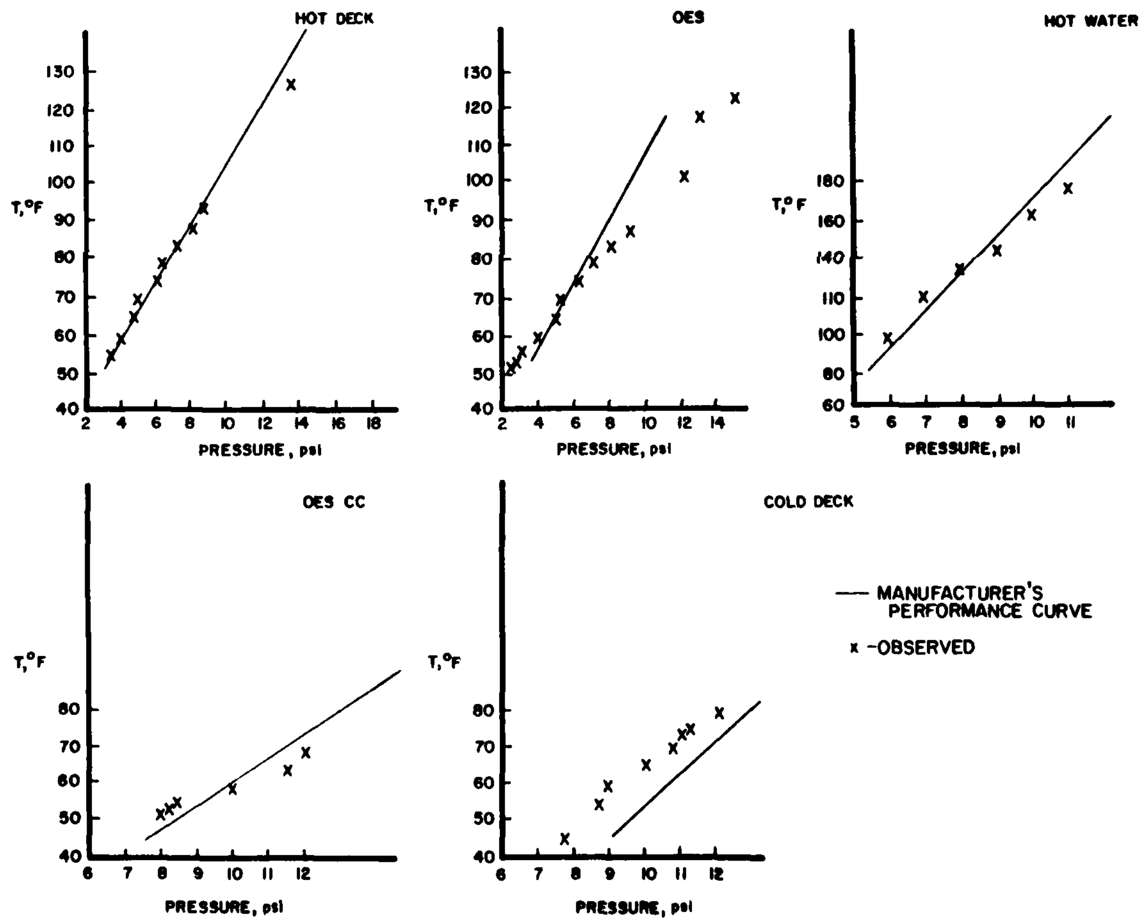


Figure 16. Pneumatic temperature transmitter. (Metric conversions: 1 psi = 6.9 kPa; [°F-32]/1.8 = °C.)

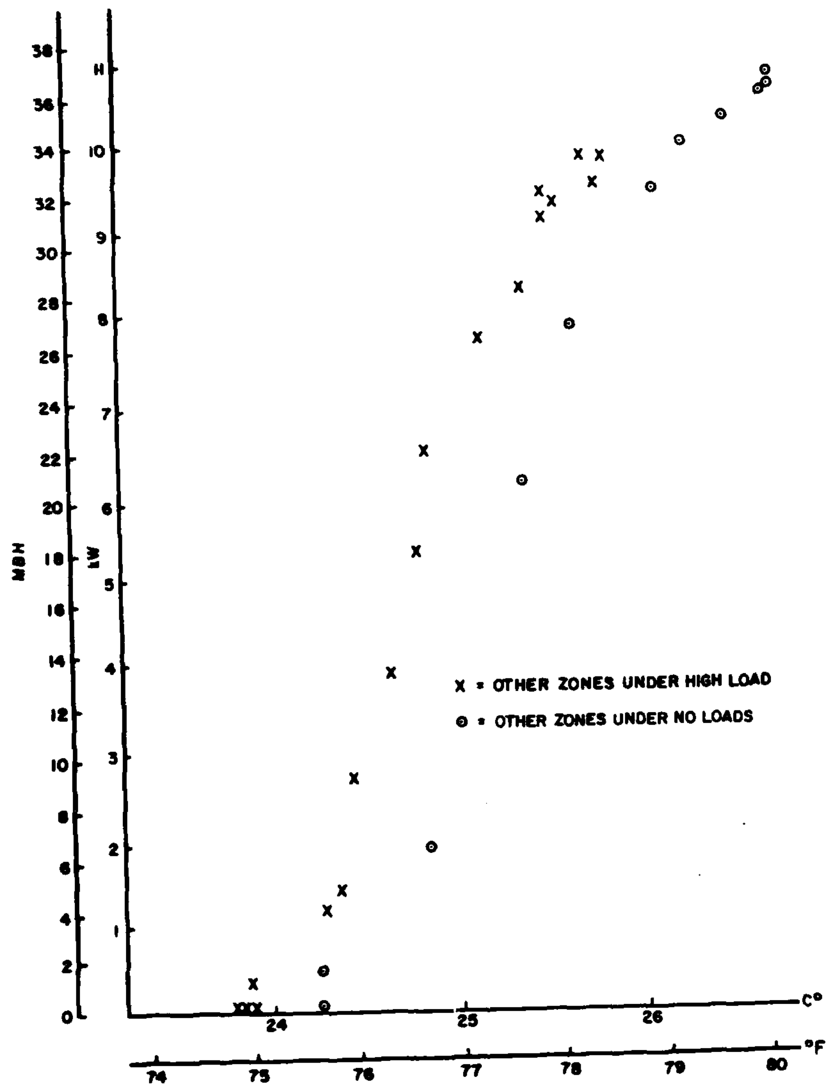


Figure 17. Single-zone temperature controller results.

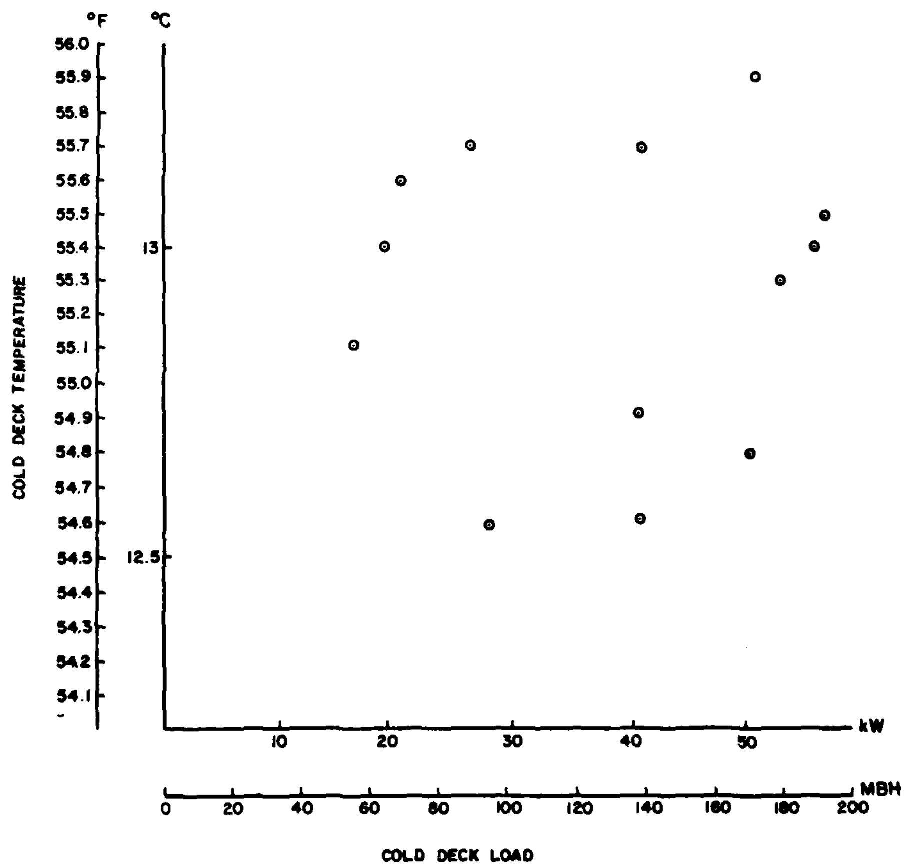


Figure 18. Cold deck temperature controller results.

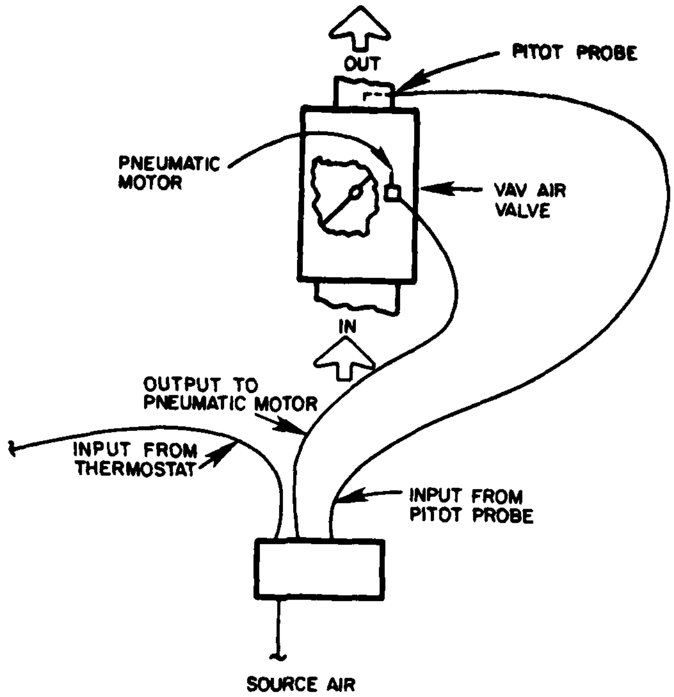


Figure 19. Pneumatic volume regulator (PRV).

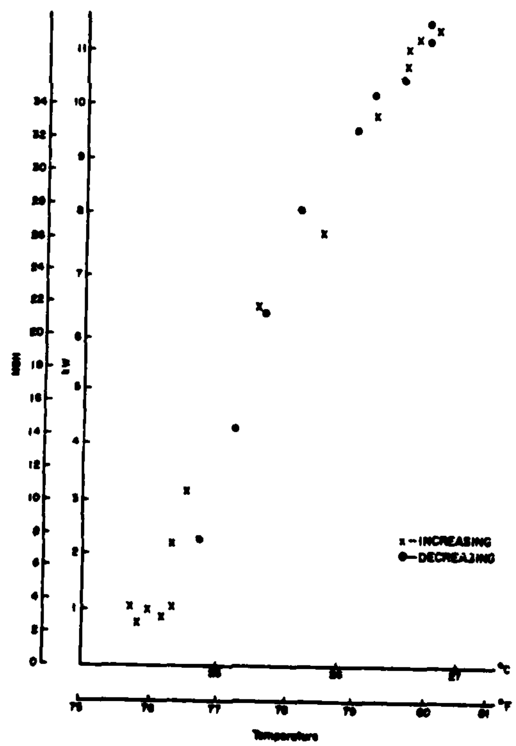
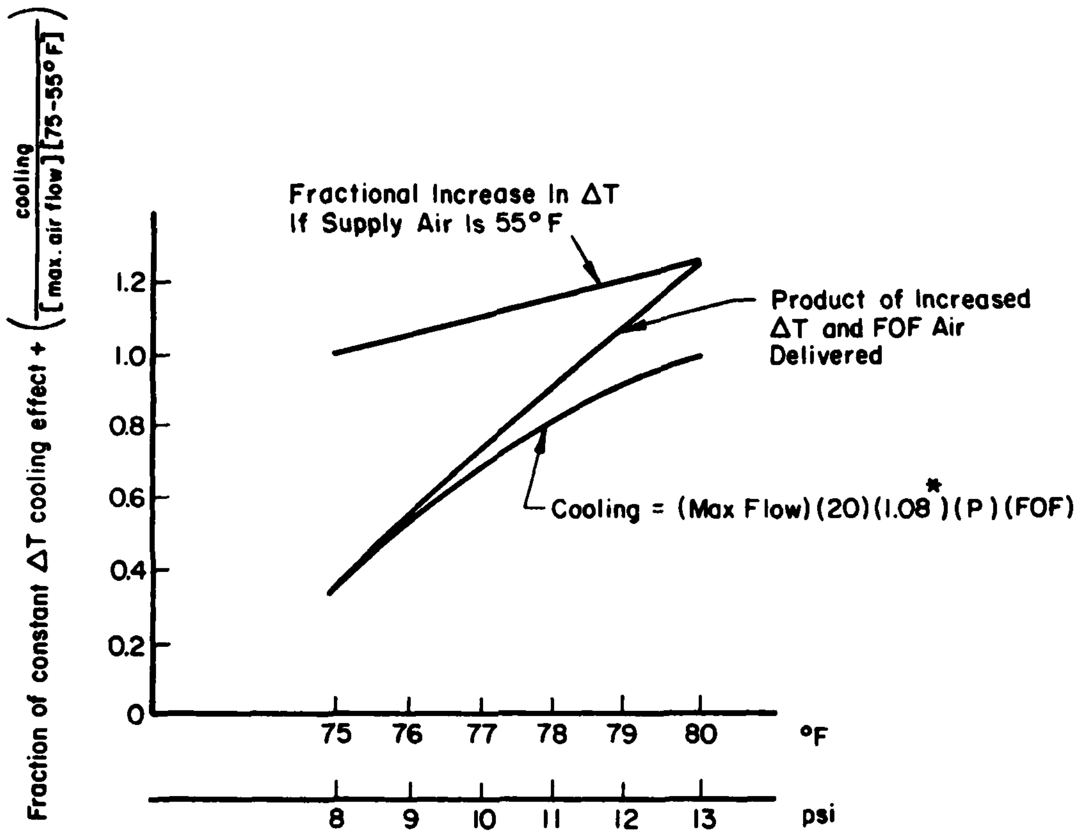


Figure 20. VAV zone temperature controller results.



Fraction of Maximum Air Flow (FOF)
When :

1) Minimum Flow Set At 1/3 Maximum
2) Thermostat Gain Is 1 psi/°F, and
3) Thermostat Set At 8 psi - 75°F

$$\text{(FOF) Fraction of Flow} = \sqrt{\frac{1-(\text{min frac.})^2}{5}\ (\text{psi}-8) + (\text{min fvac})^2}$$

*Sea Level

Figure 21. Combined effects of PVR. (Metric conversions: 1 psi = 6.9 kPa; [°F-32]/1.8 = °C.)

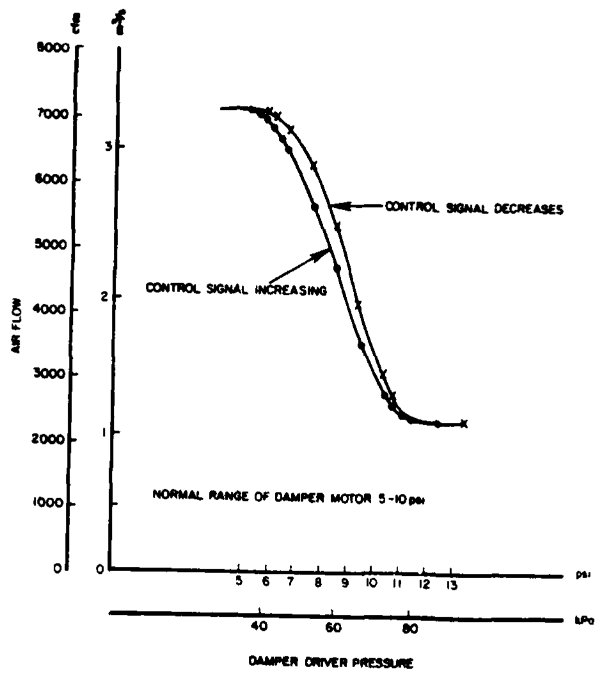


Figure 22. Control pressure -- flow rate relation.

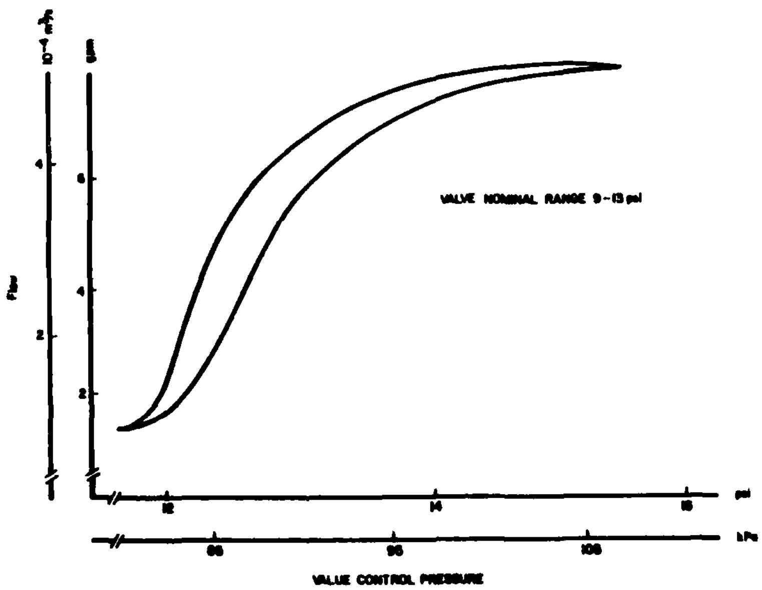


Figure 23. Water flow vs control pressure.

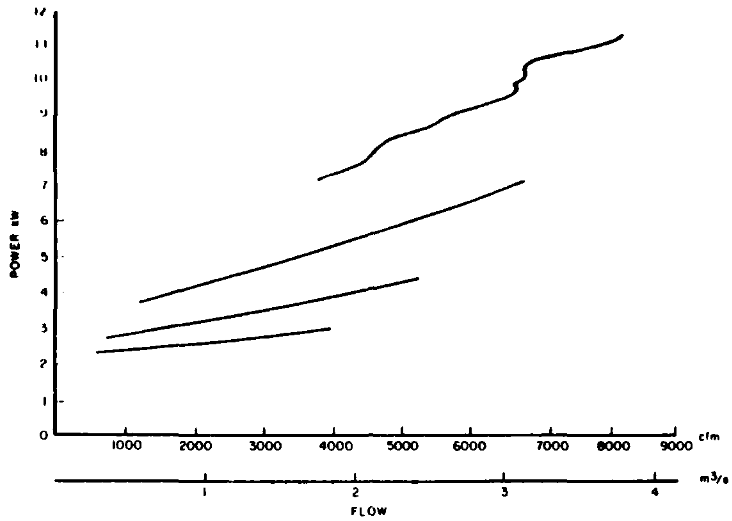


Figure 24. Electrical Power -- air flow relation.

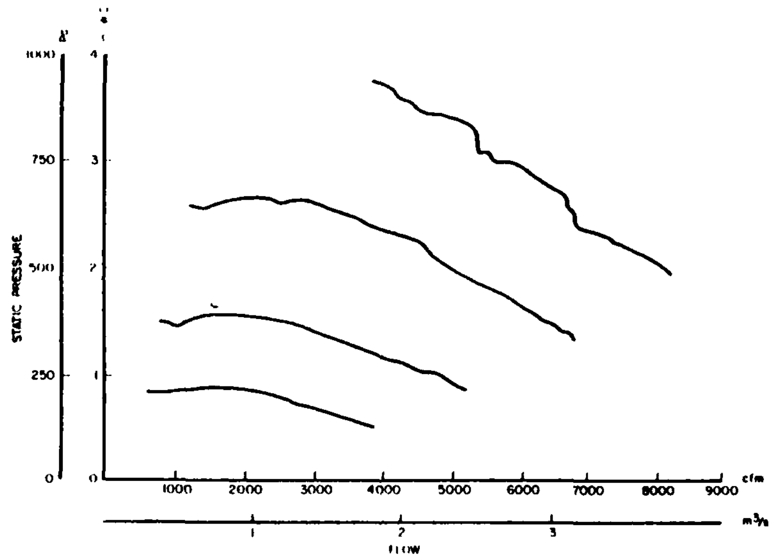


Figure 25. Fan curves -- variable speed drive.

Table 2

Boiler Stand-by Losses

Off Time	On Time
7 min 37s	51s
7 min 35s	48s
7 min 31s	48s
7 min 32s	47s

Average off time: 7 min 34s   Average on time: 48s

Table 3

Examples of Chiller Testing*

T--COLD--IN (Test Averages) ^{o}F	T--COLD--OUT (^{o}F)	T--COND--IN (^{o}F)	T--COND--OUT (^{o}F)	FLOW--COLD gpm	FLOW--COND (gpm)	POWER (W)	QREF (Btu/hr)	QCOND (Btu/hr)	COP
55.19	45.04	84.53	94.38	48.34	62.01	22690.75	245426.35	309015.00	3.17
55.65	45.23	84.43	94.40	48.34	62.15	22736.06	244803.71	310083.02	3.16
55.66	45.52	84.43	94.39	48.34	62.11	22768.04	245496.75	309601.06	3.16
55.66	45.53	84.44	94.41	48.34	62.04	22743.99	244928.31	309681.11	3.16
55.67	45.54	84.43	94.41	48.32	62.10	22735.01	244933.94	309950.54	3.16
55.66	45.55	84.43	94.41	48.35	62.10	22729.19	244525.84	310119.33	3.15
55.65	45.56	84.42	94.41	48.34	62.09	22757.73	244278.79	310363.51	3.15
55.67	45.55	84.42	94.41	48.34	62.14	22735.27	244700.65	310576.70	3.15
55.66	45.55	84.42	94.40	48.38	62.15	22734.74	244825.02	310492.38	3.16
55.66	45.56	84.41	94.41	48.34	62.05	22721.53	244388.45	310562.00	3.15
55.67	45.56	84.42	94.40	48.36	62.04	22743.99	244691.00	309679.98	3.15
55.67	45.57	84.40	94.39	48.35	62.02	22751.39	244452.21	309990.91	3.15
55.69	45.56	84.42	94.39	48.36	62.00	22730.25	245156.54	309437.78	3.16
55.69	45.56	84.41	94.38	48.36	62.00	22740.56	245036.97	309441.06	3.16
55.69	45.55	84.40	94.38	48.35	61.99	22738.97	245194.04	309392.30	3.16
55.70	45.56	84.39	94.37	48.35	61.92	22732.36	245240.48	309080.90	3.16
55.70	45.57	84.40	94.38	48.36	61.96	22731.84	245245.87	309255.17	3.16
55.70	45.58	84.40	94.37	48.37	61.97	22718.89	244997.12	309319.33	3.16
55.70	45.57	84.39	94.37	48.35	61.97	22736.06	245080.78	309431.00	3.16
55.69	45.58	84.40	94.37	48.36	62.01	22749.54	244715.06	309529.00	3.15
55.70	45.58	84.40	94.37	48.36	62.03	22708.32	244949.12	309465.13	3.16
55.70	45.59	84.39	94.36	48.35	62.04	22709.64	244540.23	309492.95	3.16
55.71	45.58	84.40	94.37	48.35	62.08	22738.44	245005.39	309730.85	3.16
55.70	45.59	84.39	94.36	48.36	62.00	22724.70	244692.80	309463.75	3.16
55.72	45.58	84.39	94.36	48.34	61.99	22708.85	245048.58	309246.81	3.16
55.72	45.58	84.40	94.36	48.33	61.98	22742.93	244992.56	308938.70	3.16
55.72	45.58	84.40	94.36	48.34	61.98	22766.19	245185.94	309074.99	3.16
55.73	45.59	84.41	94.36	48.34	62.00	22745.58	245178.90	308863.51	3.16
55.74	45.60	84.40	94.37	48.35	62.01	22756.94	245344.61	309516.90	3.16
55.73	45.60	84.40	94.36	48.34	61.92	22763.55	245047.23	308636.89	3.16
55.75	45.62	84.40	94.38	48.33	61.97	22783.37	245015.32	309291.47	3.15
55.78	45.62	84.40	94.36	48.33	61.99	22748.75	245668.40	309099.17	3.17

*Metric conversions: (^{o}F-32)/1.8 = ^{o}C; 1000 Btu/hr/3.412 = kW.

Table 4

Hot Water Piping Losses

Circulating Hot Water Temperature °F(°C)	Laboratory Air Temperature °F(°C)	1000 Btu/hr (kW)	UA* 1000 Btu/hr°F (kW/°C)
186 (84)	64 (17)	7.94 (2.3)	0.068 (0.036)
181 (81)	68 (19)	9.56 (2.8)	0.085 (0.045)
155 (67)	68 (19)	8.13 (2.4)	0.093 (0.049)
133 (55)	68 (19)	5.72 (1.7)	0.088 (0.046)

*Computed as follows: UA = Q/ T
Q(1000 Btu/hr)
F(°F)

Table 5

Enthalpy Logic Device

Case	T_{ODA} %psi	H_{ODA} %psi	E_{ODA} Btu/lbm	T_{RA} °F/psi	H_{RA} %/psi	E_{RA} Btu/lbm	Output pressure/action	Comments
TRA > TODA ERA < EODA	85° 13.2	40% 7.8	32	90°F 13.8	50% 9	38.6	0 psi 100% ODA	Correct Operation
TRA > TODA ERA < EODA	85° 13.2	70% 11.4	40.8	90°F 13.8	50% 9	38.6	0 psi 100% ODA	Incorrect Operation
TRA < TODA ERA > EODA	85° 13.2	40% 7.8	32	80°F 12.6	80% 12.6	38.6	12 psi Minimum Outdoor Air	Correct Operation
TRA < TODA ERA < EODA	80° 12.6	60% 10.2	34	75°F 12	50% 9	28.2	0 psi 100% ODA	Incorrect Operation

*T = temperature; H = relative humidity; E = enthalpy; RA = return air; ODA = outdoor air.
Pneumatic input from temperature transmitters, psi = T 0.12 + 3 (1 psi = 6.9 kPa).
Pneumatic input from humidity transmitters, psi = %v.h 0.12 + 3 (1 psi = 6.9 kPa).

Table 6

Zone Thermostats*

ZONE 1		ZONE 2		ZONE 3		ZONE 4	
TEMPERATURE	PRESSURE	TEMPERATURE	PRESSURE	TEMPERATURE	PRESSURE	TEMPERATURE	PRESSURE
67.7	5.00	66.6	3.75	67.1	5.00	66.9	5.00
67.8	5.00	66.8	3.75	67.4	5.75	67.1	5.25
68.0	6.00	66.9	3.75	67.6	6.00	67.1	5.25
68.0	6.00	66.9	3.75	67.8	6.50	67.1	5.25
68.3	6.50	67.1	4.00	68.1	6.75	67.3	5.75
68.4	6.75	67.3	4.50	68.3	7.00	67.6	6.00
68.6	7.00	67.5	5.00	68.4	7.00	67.8	6.25
68.7	7.25	67.8	5.00	68.6	7.25	68.0	6.75
68.8	7.50	68.0	5.50	68.8	8.00	68.3	7.00
68.9	7.75	68.1	5.75	69.0	8.00	68.5	7.25
69.2	8.25	68.3	6.25	69.1	8.25	68.6	7.75
69.3	8.50	68.5	6.75	69.3	8.50	68.8	8.00
69.4	8.75	68.7	7.00	69.5	9.00	69.0	8.00
69.4	9.00	68.9	7.50	69.6	9.25	69.2	8.50
69.5	9.25	69.1	7.50	69.8	9.75	69.4	8.75
69.7	9.75	69.2	8.00	70.0	10.00	69.6	9.00
69.8	10.00	69.4	8.25	70.2	10.50	69.7	9.25
69.9	10.25	69.6	9.00	70.4	10.75	69.9	9.75
70.1	10.75	69.7	9.00	70.5	11.00	70.1	10.00
70.1	11.00	69.8	9.75	70.6	11.00	70.3	10.75
70.1	11.00	70.1	10.00	70.7	11.50	70.4	10.50
70.4	11.25	70.2	10.00	70.9	11.50	70.6	10.75
70.4	12.00	70.3	10.50	71.0	11.75	70.8	11.25
70.5	12.00	70.5	11.00	71.1	12.00	71.1	11.75
70.7	12.50	70.6	11.50	71.3	12.25	71.1	12.00
70.8	13.00	70.8	11.75	71.3	12.25	71.2	12.00
70.9	13.25	71.0	12.00	71.5	12.75	71.5	12.00
71.0	13.50	71.1	12.75	71.6	13.00	71.6	12.50
71.1	14.00	71.3	13.00	71.7	13.00	71.6	12.75
71.1	14.50	71.3	13.00	0.0	0.00	71.8	13.00

$\frac{70.45 - 68.0}{12.0 - 6} = 0.41^{o}F/psi$ $\frac{71.0 - 68.3}{12 - 6} = 0.45^{o}F/psi$ $\frac{71.1 - 68.35}{12 - 7} = 0.55^{o}F/psi$ $\frac{71.15 - 67.6}{12 - 6}$ $0.59^{o}F/psi$

throttling range ^{o}F (gain[^{o}F/psi] x 5 psi)

2.0°F	2.3°F	2.8°F	3°F

* Metric conversions: 1 psi = 6.9 kPa; (^{o}F-32)/1.8 = ^{o}C.

Table 7

Record of Receiver/Controller Drift Over 72 Hours

	Input (psi)*	Output as Calibrated (psi)+	Output Observed After 72 Hours+
Circulating hot water	7	10	19
	9	9	19
	11	8	14
Hot deck	4	12	19
	10	8	18
	11	7	14
Cold deck	7	7	14
	10	10	14
	12	12	17
OES heating coil	5	14	19
	9	11	14
	11	7	14
OES cooling coil	8	8	10
	10	11	12
	13	13	15

*Input pressure (signal from temperature transmitter) was controlled from a variable pressure source and measured by a mercury manometer.
+Output pressure (control pressure for mixing valves) was observed on pressure gages internal to the receiver/controller.

Metric conversion: 1 psi = 6.9 kPa.

4 TEST RESULTS

Central Air Systems

Tables 8 through 26 list the performance data for the various systems tested, and Figures 26 through 29 are line schematics. These data are given in tabular form to simplify comparisons with computer-generated performance data. Each text series description includes:

1. A description of the system being tested, including the capacities of various components and electric motor ratings (where applicable).

2. A description of the control schedule and reset schedules (when used).

3. A table containing control set points, throttling ranges, fraction of outdoor air, and all other parameters necessary to assemble a complete input file for building energy analysis programs.

Modular Systems

Roof-Mounted Packaged Cooling and Ventilation Unit

The packaged unit acquired for testing was sized to condition the four-zone test facility exactly as the built-up system. Each zone of the four-zone VAV system is supplied a maximum 1600 cfm (0.75 m^3/s) of chilled air to offset the 36,000 Btu/hr (10.5 kW) internal gain. In addition, the packaged unit chills 1600 cfm (0.75 m^3/s) -- one-fourth of the total flow -- of warm, humid, outdoor air during design conditions. The packaged unit was to have the following capabilities according to the specifications of the purchase order:

1. A forward-curved blade fan or air foil blade fan belt driven by a 7-1/2 hp motor having the capability of moving 6400 cfm (3.0 m^3/s) at 2.3 in. (575 Pa) wc external to the unit.

2. Discharge dampers automatically controlled to maintain a preset static pressure external to the unit (in response to VAV air boxes throttling down air flow rates).

3. An economizer package capable of blending cool outdoor air with return air to maintain discharge air temperature.

4. Sufficient cooling capacity to chill 6400 cfm (3.0 m^3/s) of air from 80°F (27°C) dry bulb/67°F (19°C) wet bulb to 55°F (13°C) (approximately 20 tons [70 kW]).

5. An adjustable outdoor air/return air mixing box.

6. A discharge air thermostat control and unloading capability.

Table 27 contains unit's manufacturer's performance data and specifications.

The test set-up involved modifying the existing ductwork connecting the return air ducting and supply air ducting to the packaged unit. As loads changed in the zones, the VAV boxes would increase or throttle down the rate of conditioned air to the zone, thus changing the rate of flow through the unit.

In summary, the packaged unit had to deliver chilled air (nominally 55°F [13°C]) at rates varying from 1600 to 6400 cfm (0.75 to 3.0 m^3/s) using refrigeration or outdoor air free-cooling, when possible. Other capabilities included the control of static pressure external to the unit, blending outdoor air with return air at a ratio of 25/75 respectively, and the enthalpy control logic (Figure 30).

The purchase order specifications for the fan within the packaged unit called for the fan to be set up to move 6400 cfm (3.0 m^3/s) at 2.3 in. (575 Pa) wc external to the unit. As-delivered, the fan was adjusted precisely to accomplish this flow/static pressure operating point.

Fan testing involved monitoring flow, static pressure, and electrical power while the air flow rate was varied from its maximum (6400 cfm at 2.3 in. wc [3.0 m^3/s at 575 Pa] to a minimum of about 1000 cfm (0.5 m^3/s).

Test results are presented in two graphs, total flow vs static pressure (Figure 31) and total flow vs electric power (Figure 32).

Figure 31 shows a flow-static pressure relation of particular significance. The arrangement of the two foward-curved blade fans result in a very flat region of the flow-static pressure fan curve. Thus, as flow is reduced in a VAV mode, no appreciable rise in static pressure results. Furthermore, there is no need for static pressure control with discharge dampers and no danger of overpressurization of the ducting.

As air flow is throttled from 6400 to about 1000 cfm (3.0 to 0.5 m^3/s), power is reduced by about 50 percent of full-load power. This is a substantial reduction, but not as effective as the literature generally suggests.

The packaged unit was considered to be operating at full load when all cylinders of the compressor were energized, both condenser fans were operating, and no hot-gas bypass was present. Tests of full-load operation were recorded for both wet and dry coils and for air flow rates less than design. Air flow rate, refrigeration effect, dry and wet bulb temperatures of air entering and exiting the DX coil, and electrical power demanded by the compressor and both condenser fans were recorded and are presented in Table 28 for wet coil operation, Table 29 for dry coil operation.

The packaged unit was considered operating at part load when two of the four cylinders of the compressor were deenergized, leaving two cylinders active. Air flow rate, refrigeration effect, air dry and wet bulb temperatures entering and exiting the DX coil, and electrical power were recorded and are presented in Table 30.

Controls

The principal function of the packaged unit is to supply a constant temperature chilled air for VAV service. Control is possible by operating the chiller at four-cylinder (full) capacity, two-cylinder (50 percent) capacity, or two-cylinder operation with hot-gas bypass. Control is accomplished as follows:

1. A proportional temperature controller activates in sequence (a) the ODA dampers modulated from the minimum position to fully open, (b) the chiller at 50 percent capacity, and (c) the chiller at 100 percent capacity until the deck temperature is satisfied. The ODA dampers open only if the enthalpy logic device senses ODA enthalpy below the controller setting.

2. A hot-gas bypass system artificially loads the evaporator with hot refrigerant gas. The flow of hot gas is controlled by a constant pressure expansion valve which passes hot refrigerant gas when evaporator pressure falls below the valve setting.

3. A summer/winter change over thermostat which disables the compressor and condenser fans when ODA falls below the controller setting.

4. A thermostat which disables one of the two condenser fans when ODA falls below about 65°F (18°C).

Several switches were necessary for the fan and compressor to energize. After studying the wiring schematics, the placement of these switches was noted.

As-delivered, the fan rotative speed was precisely correct to deliver design flow at the design static pressure. Belt tension was noted as proper.

Expansion valve superheat was measured by measuring the suction pressure at the compressor and by measuring the temperature of the refrigerant vapor leaving the evaporator. The superheat was precisely the value recommended by the manufacturer.

The two forward-curved fans supplied in the packaged unit operated along a very flat region of the fan curve (flow vs static pressure) as the VAV system reduced the flow of air. Design flow (6400 cfm (3.0 m^3/s) resulted in about 2.4 in. (600 Pa) wc external to the unit. After throttling the flow down to 1000 cfm (0.5 m^3/s), the static pressure increased to about 2.6 in. (650 Pa) wc, a very small change. Thus, there was no need to control static pressure with discharge dampers.

The unit was equipped with a floating-type static pressure controller which, if engaged, would never sense a significant change in static pressure and thus would never change the position of the discharge dampers.

The packaged unit was equipped with a hot-gas bypass to provide capacity control at light loads. This system directs some of the hot high-pressure refrigerant gas past the condenser heat exchanger to a constant pressure expansion valve and into the evaporator coil.

The unit was tested with and without the hot gas bypass in operation (Table 29). When the unit was tested with the hot-gas bypass operational, the full-load capacity was reduced about 30 percent and at no load the unit would not cycle off, but rather continually operated at two-cylinder (50 percent) capacity.

The hot-gas bypass option was deenergized for the unit testing. The reduction in full-load capacity and efficiency, and low-load operating efficiency, make this an unattractive option.

Heating Fan/Coil Test

This test involved a heating fan/coil unit under control of a two-position thermostat and an electric zone valve. Increasing zone temperature caused the zone thermostat to open, thus closing an electric water valve, stopping the flow of hot water to the fan/coil unit. The heating capacity of the unit was 36,000 Btu/hr (10.5 kW) when supplied 2 gpm (0.00012 m^3/s) of 155°F (68°C) heating water and operating in a 68°F (20°C) environment. The unit was tested at several part-load conditions as well as the full-load condition. (The part-load condition results when the system cycles. In this example, the two-position controller cycles, causing the heating water supplied to the fan coil unit to cycle.)

The particular zone valve used for this test used a heat motor to open the valve. This device operated in a different manner than other types of automatic valves, such as solenoid valves. When the two-position thermostat called for heating, current would pass through the zone thermostat to a winding internal to the valve producing heat. This winding surrounding a heat-sensitive piston-cylinder which expanded when hot, causing the valve to open. When this winding was sufficiently hot, an additional temperature controller internal to the valve opened the circuit and flow of current through the heating element stopped, presumably not affecting the valve position. However, this controller did display some abnormalities due to the internal thermal breaker.

A near constant temperature air supply was delivered to the test zone to produce a heating load. The magnitude of the heating load was changed by regulating the rate of the cold air supply. Trials of various rates of hot water supplied to the fan/coil unit indicated that about 2 gpm (0.00013 m^3/s) of 150°F (65°C) water was sufficient to meet the design load of 36,000 Btu/hr (10.5 kW) of heating in a 68°F (20°C) environment. During testing, the hot water supply varied between 150 and 160°F (65 and 71°C). All conditions were sampled at a 30-s interval and several cycles were observed before steady periodic results were formulated.

The zone thermostat was set to call for heating (switch close) at 68°F (20°C). The thermostat would open (disabling heating) at 71.8°F (22.1°C). No drift from this set point or control range was observed over weeks of testing. However, the zone temperature swing was substantially larger than the controller swing (about 8°F [4.4°C]), due to the slow response of the zone valve.

Figure 33 shows the flow of hot water to the fan/coil vs time. Almost 1 min elapsed before the valve initially responds to the thermostat signal. Hot water flow rate reached design after about 2-1/2 min; however, a peculiar characteristic of the valve and internal thermal breaker caused the valve to close briefly for about a half minute before resuming the full open position. About 3-1/2 min elapsed until the valve was fixed in the open position. This caused the zone temperature to dip below the temperature which triggered the zone thermostat.

The same lagging response of the zone valve caused the zone temperature to exceed the temperature which caused the thermostat switch to open (disabling heating). As previously mentioned, the zone thermostat had about a 4°F (2°C) throttling range; however, the thermostat control valve system resulted in a zone temperature swing of about 8°F (4°C). Figure 34 shows the temperature time history of one cycle of the fan/coil control system. Note the small dip in zone temperature as heating begins. This is due to the valve characteristic pictured in Figure 31.

The fan/coil was exposed to four loads where the hot water supply temperature remained near constant. The fan/coil was tested once where the supply water temperature was elevated. The results are presented graphically in Figure 35. The elevated supply water temperature had little effect on the heating system performance. For modeling purposes, it appears the average for zone temperature varies from 67 to 77°F (19 to 25°C), corresponding to the design load of 36,000 Btu/hr (10.5 kW) and a no-load condition, respectively. This would be considered a 10°F (5.5°C) throttling range of the heating system. However, zone temperature was very transient during testing as indicated in Figure 35 for all loads.

In summary, the heating system experienced about 8°F (4°C) swing in steady periodic operation. This was due mainly to the slow response of the electric hot water valve and in part to the throttling range of the thermostat. The heating system experience about a 10°F (5.5°C) swing in average temperature, which corresponded to a no-load to full-load operation.

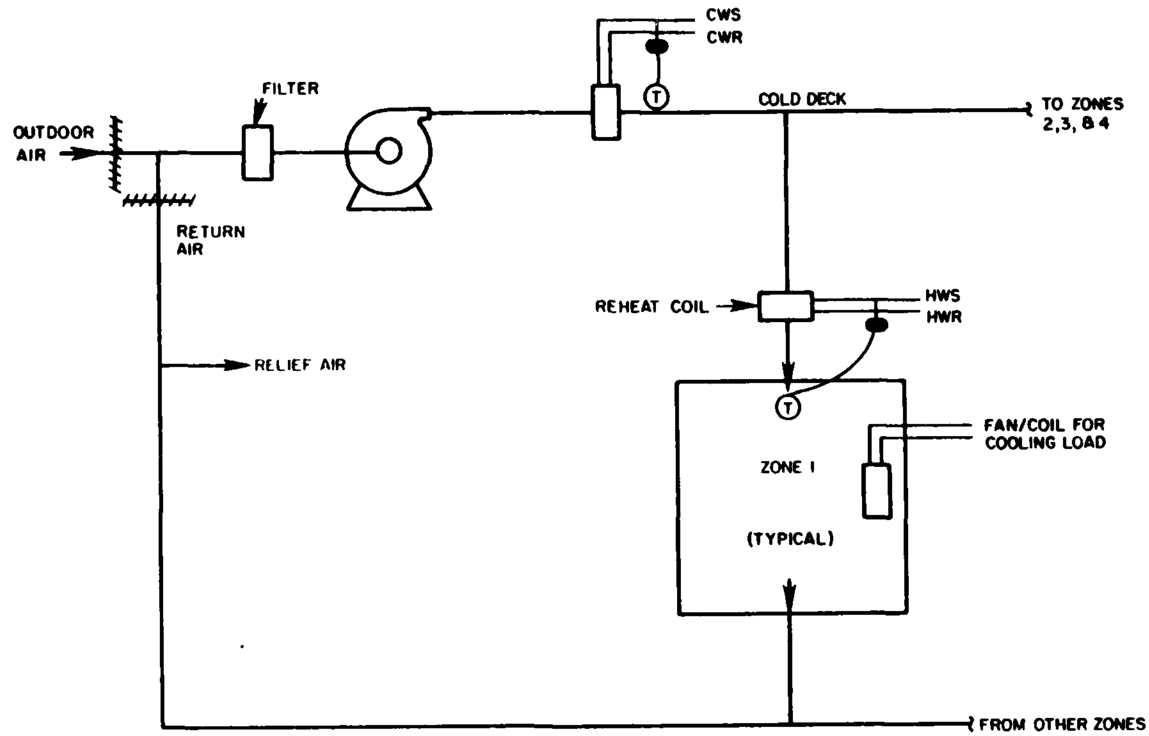


Figure 26. Schematic -- terminal reheat.

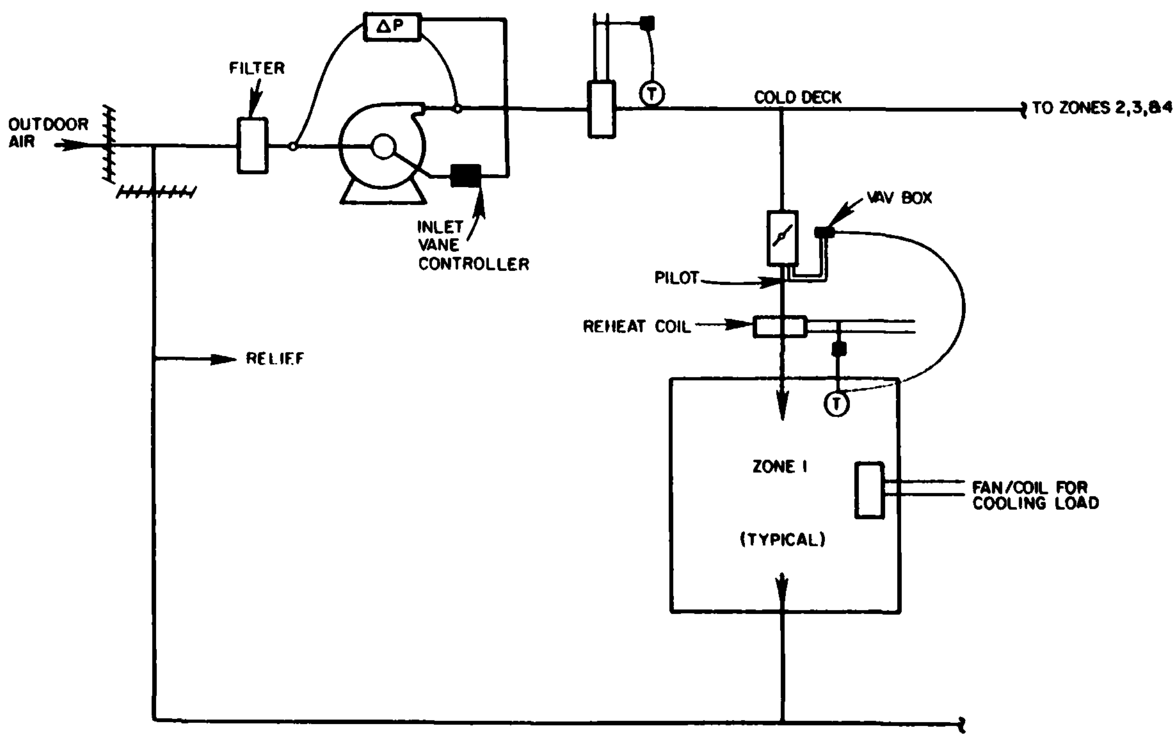


Figure 27. Schematic -- VAV with reheat.

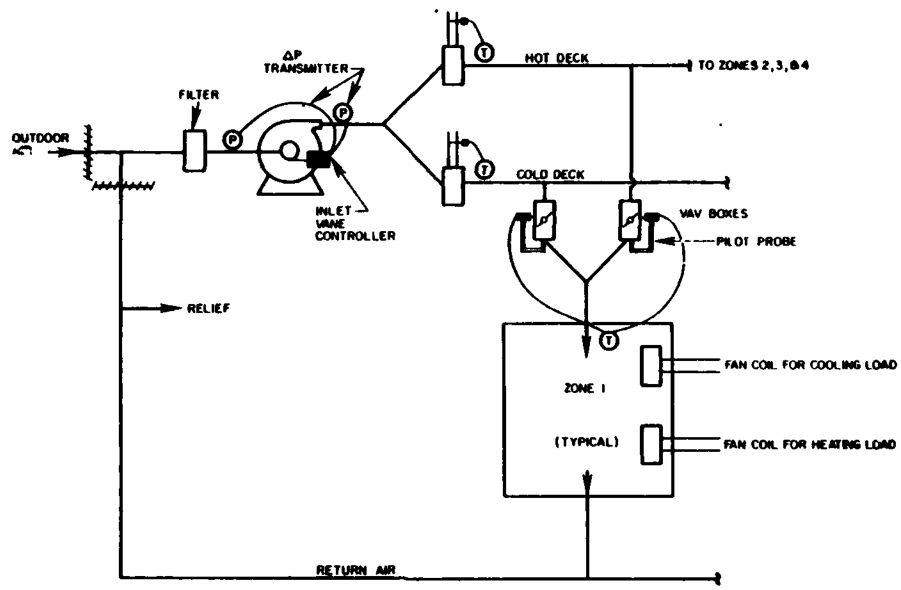


Figure 28. Schematic -- dual duct VAV.

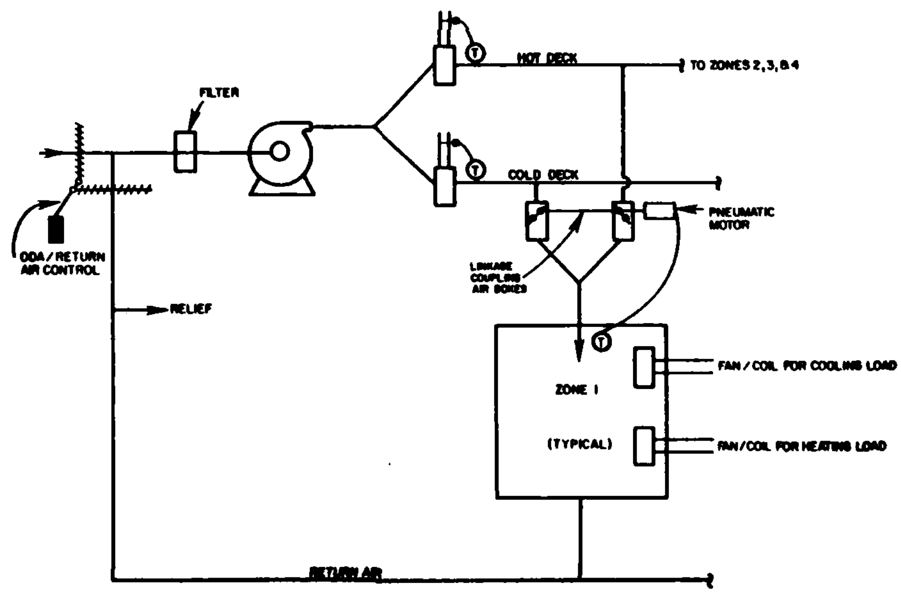


Figure 29. Schematic -- dual duct/multizone.

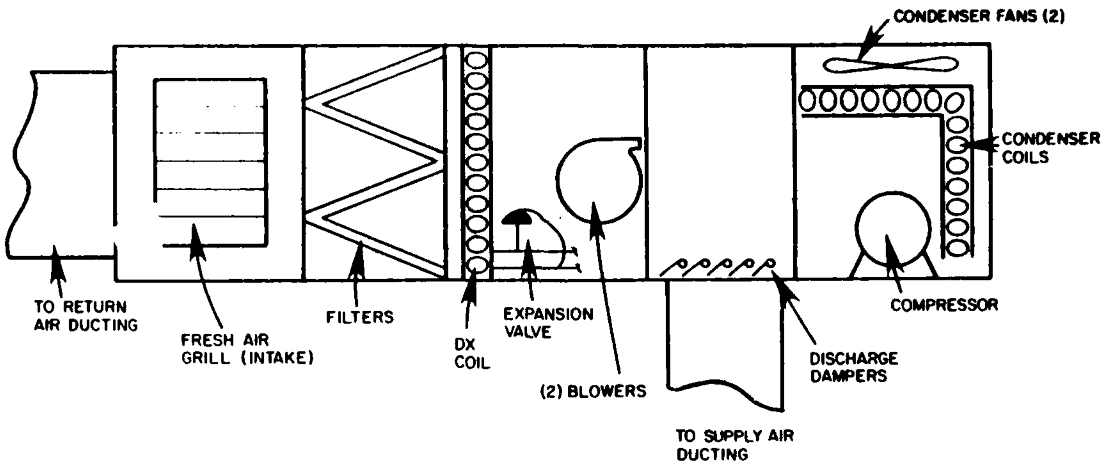


Figure 30. Roof-mounted packaged cooling and ventilation unit.

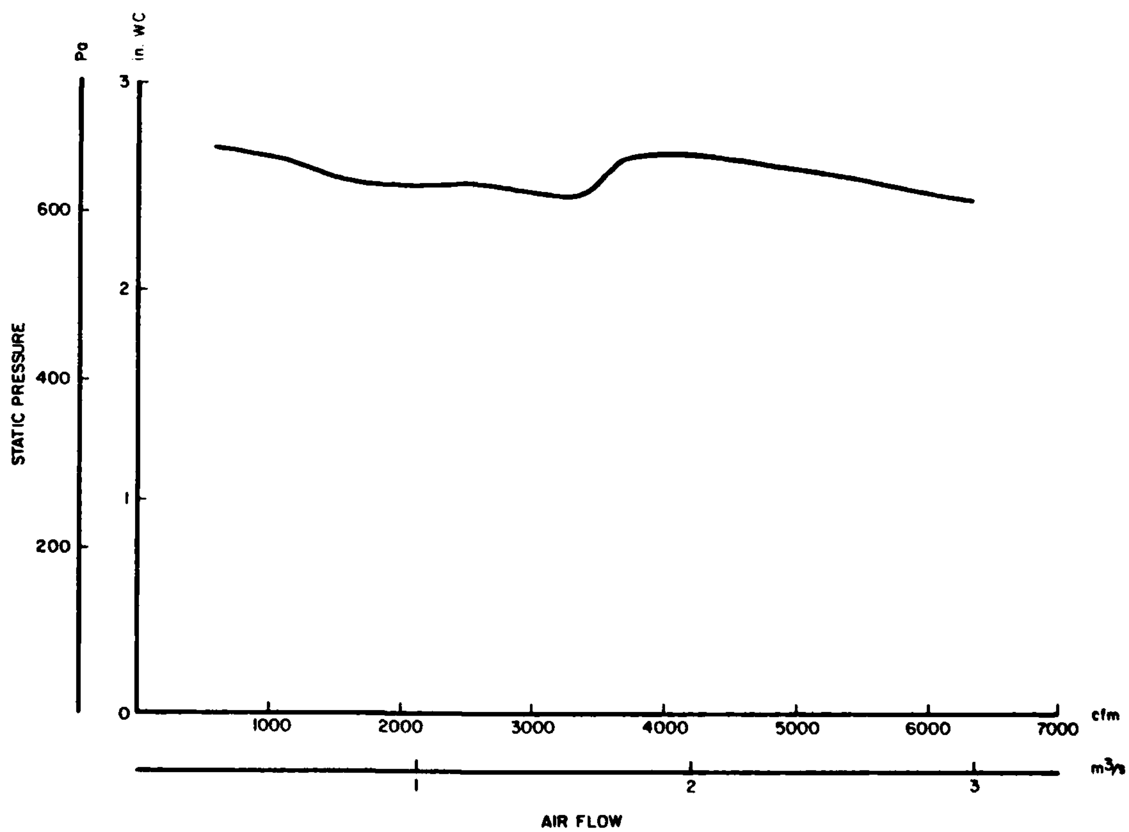


Figure 31. Total flow vs static pressure.

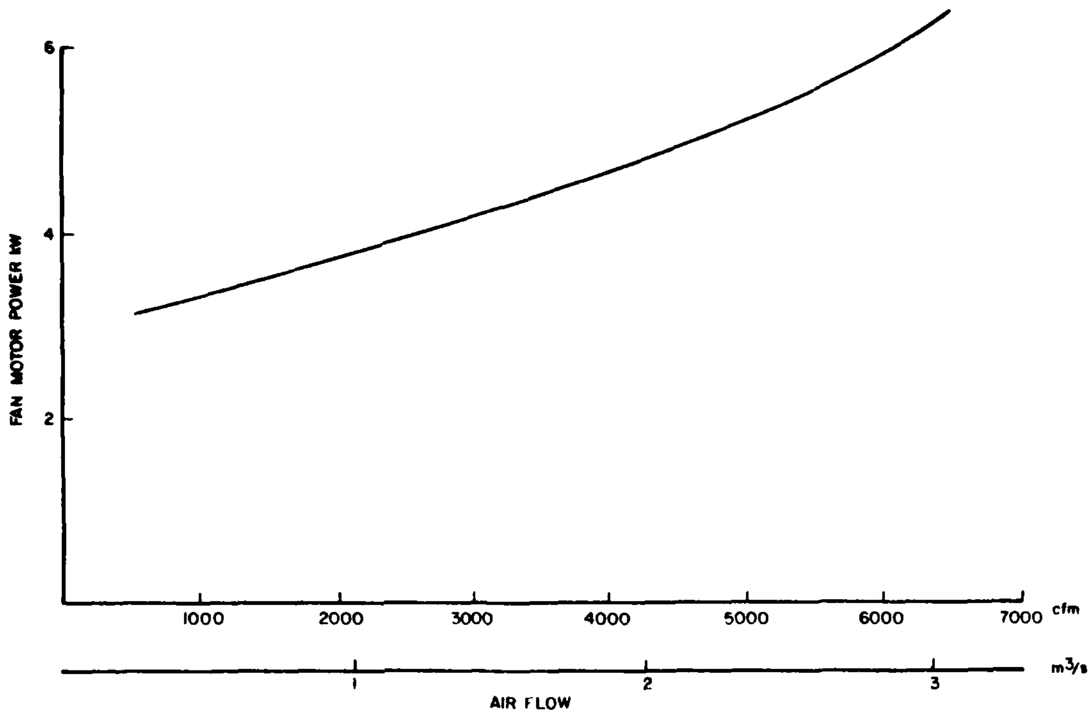


Figure 32. Total flow vs electric power.

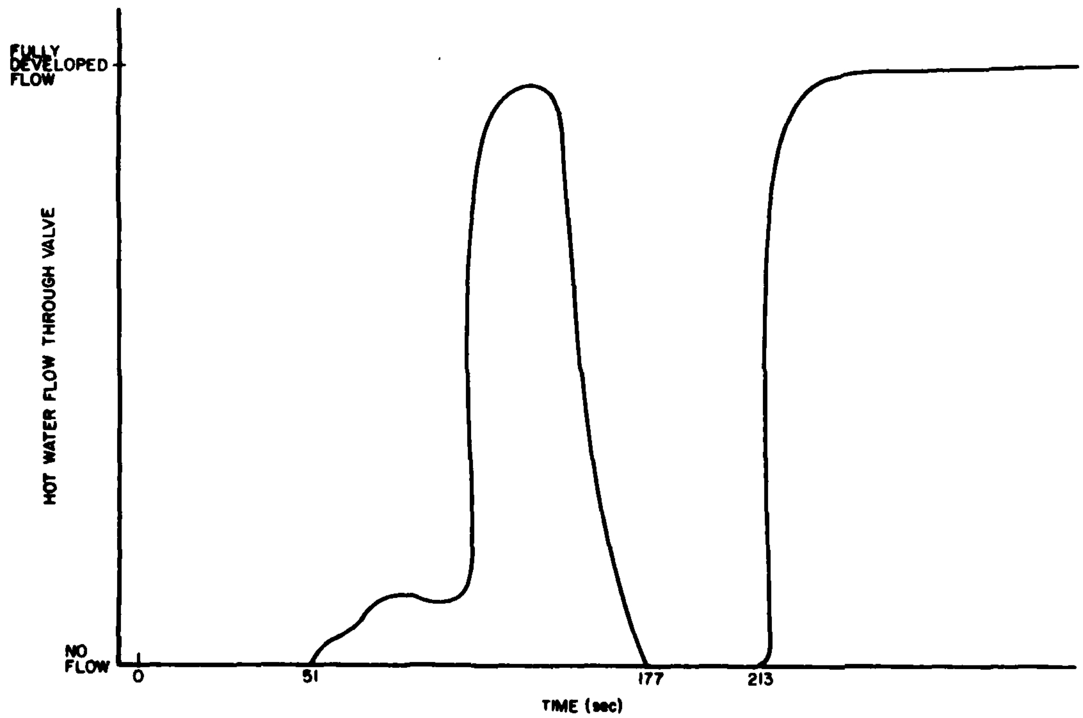


Figure 33. Flow of hot water to the fan/coil vs time.

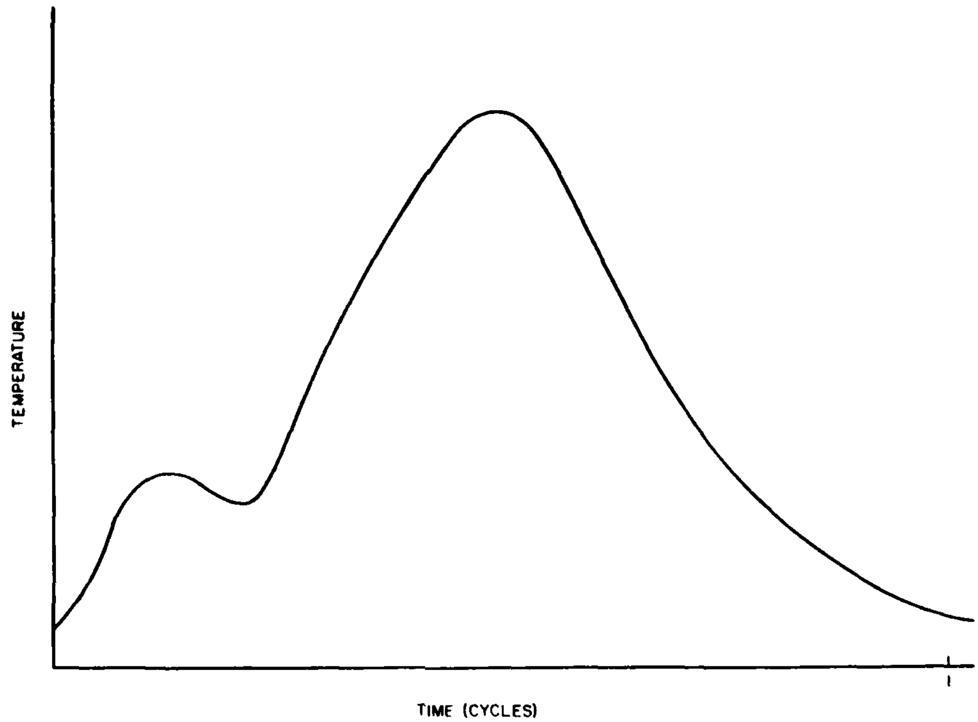


Figure 34. Temperature time history of one cycle of the fan/coil control system.

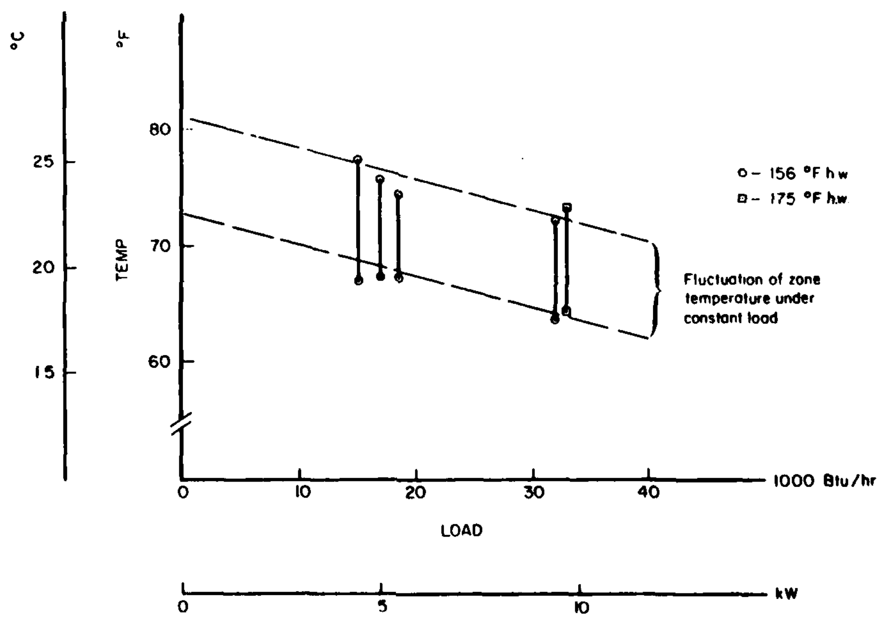


Figure 35. Fan/coil test results.

Table 8

VAV With Reheat Test Series -- Index

Test No.	System Description
1	Basic system, all low loads
2	Basic system, 2 low, 2 medium loads
3	Basic system, 2 high, 2 low loads
4	Basic system, all high loads
5	Basic system, all medium loads
6	High zone set points, high loads
7	High zone set points, high loads
8	High zone set points, low loads
9	25% ODA, medium loads
10	Free cooling, high loads
11	Cold cold deck, high loads
12	Cold cold deck, low loads
13	Cold cold deck, low loads
14	25% ODA, low loads
15	High zone set points, low loads
16	High zone set points, high loads
17	Low throttling ranges, high loads
18	Low throttling ranges, low loads
19	Low throttling ranges, fan capacity control, high loads
20	Low throttling ranges, fan capacity control, low loads

Table 9

VAV With Reheat Test Series -- Test System Description

COOLING COIL: Six-row counter flow, 33 x 57 in. (0.76 x 1.45 m).

CHILLER: 20-ton (70 kW) cold water generator (manufacturer's specification at full-load COP = 3.4).

TOWER: 30-ton (10 x 5 kW) with capacity control by means of scroll dampers, equipped with a 5-hp (3.7 kW) motor.

FAN: One 15 in. fan-coil fan in blow-through arrangement 6400 cfm (3 m^3/s), 2.75 in. (0.7 KPa), 1320 rpm (manufacturer's specification: E = power to air/shaft power 34%), fan is 40% efficient, motor is 85% efficient.

PUMPS: conditioned water, 1-1/2 hp (1.1 kW); chilled water, 1-1/2 hp (1.1 kW); and hot water, 1 hp (0.745 kW).

Basic System -- Control Settings

ROOM TEMPERATURE CONTROLLERS:

VAV air box under control of a volume regulator -- maximum flow 1600 cfm (0.75 m^3/s); minimum flow fraction 40%; reheat value sequenced with VAV box controller.

Set point (median of control range) 76°F (24°C), nominal; exact set point reported.

Throttling range: as small as possible for stable operation, exact value reported.

Chilled water temperature: 44°F (7°C), nominal.

Cold deck temperature: 55°F (13°C) nominal; exact value reported, throttling range as small as possible reported.

Fan: no capacity control.

Tower: return water temperature 80°F (27°C) nominal.

Hot water temperature: 150°F (64°C) nominal reported; controlled by a mixing value diverting water through a converter, as necessary.

Boiler pressure controller: 5 psi (35 kPa) nominal.

Air-handler mixing box: 0% outdoor air.

Table 10

VAV With Reheat Test Series Data -- Tests 1 Through 8*

VAV -REHEAT SYSTEM TOTALS	1	2	3	4	5	6	7	8
Loads (Cooling)	4.9	37	66	131	71	134	133	51
Electrical	51	49	64	90	62	87	87	56
Gas	103	67	68	39	34	39	39	46
ELECTRICAL								
Chiller	28	26	35	57	36	57	55	30
Tower	5.2	5.3	5.8	6.0	5.2	5.9	5.9	5.1
Fan	11	12	14	18	13	15	19	11
Pump-Cold Water	3.6	3.6	3.5	3.5	3.5	3.5	3.4	3.5
Pump-Hot Water	1.6	1.7	1.5	1.4	1.5	1.4	1.4	1.8
Pump-Cond Water	3.9	3.9	3.9	3.9	3.9	3.9	3.9	3.9
BOILER								
Stand-by Loss	23	23	23	23	23	23	23	23
Distribution Loss	6.8	5.7	7.7	7.5	3.8	7.7	7.5	3.8
Load	51	24	22	0	0	0	0	9
Fuel-Gas	103	67	68	39	34	39	39	46
CHILLER								
Load	74	71	105	150	95	156	148	95
Flow-Cond.	63.7	65.6	65.9	65.9	66.0	64.3	62.6	62.4
Flow-Cold	51.8	48.5	49.2	49.7	48.3	52.2	49.2	51.4
Temp-Cold In	49.0	49.0	49.8	50.5	49.5	50.8	50.5	48.9
Temp-Cold Out	46.3	46.3	45.8	44.5	45.9	44.8	44.5	45.5
Temp-Cond In	75.6	75.7	83.0	84.7	82.1	84.3	84.6	79.6
Temp-Cond Out	78.4	78.6	87.0	90.8	85.8	90.4	90.3	83.2

* Units: ~~loads --~~ thermal (MBH), loads -- electrical (MBH), temperature (°F), flows -- water (gpm). Metric conversions: 1000 Btu/hr/3.412 = kW; (°F-32)/1.8 = °C.

Table 10 (Cont'd)

ZONES	1	2	3	4	5	6	7	8
1 - Temp Supply Air	72.9	71.5	69.9	55.6	55.3	52.0	52.4	63.4
1 - Flow-Air	731	663	715	1530	1001	1141	1356	691
1 - Temp Zone	73.9	72.6	71.8	76.1	74.1	80.2	80.3	78.6
1 - Load - Cooling	0.8	0.8	1.5	34	20	35	34	11.5
1 - Load - Reheat	13.6	11.5	12.2	0	0	0	0	5.0
2 - Temp Supply Air	69.5	70.0	67.6	55.8	56.0	52.5	57.6	58.2
2 - Flow - Air	652	708	697	1356	792	1027	1306	719
2 - Temp Zone	72.2	73.3	70.7	77.2	75.5	80.3	80.8	76.3
2 - Load - Cooling	1.9	2.5	2.3	32	17	31	33	14
2 - Load - Reheat	10	12	10	0	0	0	0	0
3 - Temp Supply Air	74.1	56.8	55.9	56.3	56.9	52.9	58.0	59.4
3 - Flow-Air	614	756	1279	1376	759	1220	1393	604
3 - Temp Zone	74.0	77.6	78.8	79.4	77.5	81.6	82.4	79.7
3 - Load - Cooling	1.3	17	32	34	17	38	37	13
3 - Load - Reheat	12	0	0	0	0	0	0	0
4 - Temp Supply Air	74.6	56.6	55.7	56.0	56.4	52.8	57.9	61.2
4 - Flow - Air	687	802	1295	1336	838	1015	1193	672
4 - Temp Zone	75.9	75.8	77.1	77.7	75.4	80.3	80.8	77.4
4 - Load - Cooling	1.0	17	30	31	17	30	30	12
4 - Load - Reheat	15	0	0	0	0	0	0	3.8

* Units: loads -- thermal (MBH), loads -- electrical (MBH), temperature (°F), flows -- water (gpm). Metric conversions: 1000 Btu/hr/3.412 = kW; (°F-32)/1.8 = °C.

Table 10 (Cont'd)

AIRHANDLER	1	2	3	4	5	6	7	8
Temp - Return Air	73.0	73.7	74.7	76.1	74.4	78.1	78.7	76.6
Temp - Mixed Air	73.1	73.1	73.9	76.0	74.1	78.0	78.7	76.4
Temp - O.D.A.-dry bulb	66.8	64.2	66.4	66.7	66.4	68.1	69.1	70.8
Temp - O.D.A. - Wet bulb								
% ODA	2	6	9	1	4	2	0	5
Δ T Across Fan	1.9	1.9	2.2	2.0	2.0	1.6	1.7	1.8
Δ P Across Fan		3.4	3.2	2.7	3.3	3.1	2.8	3.4
% Hum. Before Cold Coil				20	22	18	16	17
% Hum. After Cold Coil				47	48	51	38	39
Flow - Air Over Coil	2684	2929	3986	5598	3389	4402	5249	2686
Flow - Chilled Water	15.0	14.9	20.5	25.3	18.9	50.8	19.8	12.5
Temp - Chilled Water - In	46.7	46.6	46.2	44.9	46.3	45.4	44.8	46.3
Temp - Chilled Water - Out	55.6	56.3	56.4	57.0	56.1	50.8	59.4	58.2
Temp - Cold Deck	53.3	54.0	53.9	54.6	53.6	49.9	55.0	54.2
CONTROLS								
Zone 1 - Set Point	74	74	74	74	74	80	80	80
Zone 1 - Thr. Range	7	7	7	7	7	7	7	7
Zone 2 - Set Point	77	77	77	77	77	82	82	82
Zone 2 - Thr. Range	10	10	10	10	10	10	10	10
Zone 3 - Set Point	76	76	76	76	76	80	80	80
Zone 3 - Thr. Range	12	12	12	12	12	12	12	12
Zone 4 - Set Point	74	74	74	74	74	79	79	79
Zone 4 - Thr. Range	10	10	10	10	10	10	10	10
Cold Deck - Set Point	58	58	58	58	58	58	58	58
Cold Deck - Thr. Range	5	5	5	5	5	5.5	5	5
Hot Water - Set Point								
Hot Water - Thr. Range								
Cooling Tower - Set Point, Nom.								
Free Cooling								
Enthalpy Logic								
Deck Reset								
Fan Capacity Control								

* Units: loads -- thermal (MBH), loads -- electrical (MBH), temperature (°F), flows -- water (gpm). Metric conversions: 1000 Btu/hr/3.412 = kW; (°F-32)/1.8 = °C.

Table 11

VAV With Reheat Test Series Data -- Tests 9 Through 16*

VAV-REHEAT SYSTEM TOTALS	9	10	11	12	13	14	15	16
Loads (Cooling)	99	118	111	136	31	37	32	122
Electrical	68	48	81	88	55	59	53	76
Gas	37	38	37	39	73	62	68	38
ELECTRICAL								
Chiller	37	16	52	57	30	34	27	46
Tower	5.4	4.8	5.5	5.9	5.3	5.3	5.2	5.5
Fan	16	18	15	17	11	11	11	16
Pump-Cold Water	3.5	3.7	3.6	3.6	3.5	3.5	3.6	3.5
Pump-Hot Water	1.4	1.4	1.4	1.4	1.7	1.7	1.8	1.4
Pump-Cond Water	3.9	3.9	3.9	3.9	3.9	3.9	4.0	3.9
BOILER								
Stand-by Loss	23	23	23	23	23	23	23	23
Distribution Loss	5.5	6.9	6.1	7.4	2.2	2.3	2.8	7.0
Load	0	0	0	0	32	23	28	0
Fuel-Gas	37	38	37	39	73	62	68	38
CHILLER								
Load	60	28	113	153	84	60	81	140
Flow-Cond.	66.3	66.5	62.8	63.0	64.5	65.3	62.4	62.4
Flow-Cold	51.3	54.0	52.0	52.1	51.3	51.7	51.8	49.5
Temp-Cold In	49.2	48.4	50.6	50.8	49.8	48.8	48.8	50.6
Temp-Cold Out	46.9	47.7	45.6	45.0	46.6	46.5	45.8	45.0
Temp-Cond In	67.5	75.6	83.5	84.1	75.4	72.7	73.4	83.1
Temp-Cond Out	70.1	76.5	88.8	90.1	78.8	75.3	76.3	88.5

* Units: loads -- thermal (MBH), loads -- electrical (MBH), temperature (^{o}F), flows -- water (gpm). Metric conversions: 1000 Btu/hr/3.412 = kW; (^{o}F-32)/1.8 = ^{o}C.

Table 11 (Cont'd)

ZONES	9	10	11	12	13	14	15	16
1 - Temp Supply Air	56.7	56.0	52.0	52.0	63.2	58.2	67.5	57.7
1 - Flow-Air	1205	1567	1167	1364	655	626	668	1232
1 - Temp Zone	75.8	75.9	75.7	75.9	74.5	73.3	79.5	81.5
1 - Load - Cooling	25	34	30	35	8	10	9	32
1 - Load - Reheat	0	0	0	0	8.5	5.0	7.6	0
2 - Temp Supply Air	57.0	56.3	52.7	52.4	62.5	57.4	66.0	58.2
2 - Flow - Air	1140	1228	887	1165	706	670	681	1080
2 - Temp Zone	76.4	74.4	78.6	77.8	72.5	71.0	77.2	83.4
2 - Load - Cooling	24	24	25	32	8	10	8	29
2 - Load - Reheat	0	0	0	0	8.8	6.0	7.4	0
3 - Temp Supply Air	57.4	56.7	52.9	52.8	63.7	60.7	66.6	58.4
3 - Flow-Air	1204	1292	1177	1363	594	546	611	1290
3 - Temp Zone	78.1	78.7	77.7	78.5	75.1	74.7	78.4	82.4
3 - Load - Cooling	27	31	32	40	7	8	8	34
3 - Load - Reheat	0	0	0	0	7.0	5.2	5.6	0
4 - Temp Supply Air	57.2	56.4	52.9	52.7	61.8	60.2	66.5	58.3
4 - Flow - Air	1102	1275	984	1173	683	654	662	1101
4 - Temp Zone	76.0	77.1	76.1	76.9	72.3	72.2	77.0	80.7
4 - Load - Cooling	22	29	25	31	8	8	8	27
4 - Load - Reheat	0	0	0	0	7.5	6.7	6.9	0

* Units: loads -- thermal (MBH), loads -- electrical (MBH), temperature (^{o}F), flows -- water (gpm). Metric conversions: 1000 Btu/hr/3.412 = kW; (^{o}F-32)/1.8 = ^{o}C.

Table 11 (Cont'd)

AIRHANDLER	9	10	11	12	13	14	15	16
Temp - Return Air	74.5	74.8	74.6	74.9	72.1	71.5	76.2	79.2
Temp - Mixed Air	66.1	54.3	74.3	74.8	71.7	66.1	75.9	79.1
Temp - O.D.A.-dry bulb	41.8	54.4	61.9	62.9	63.6	50.2	64.3	64.3
Temp - O.D.A. - Wet bulb								
% ODA	26	99	1	1	5	26	3	1
Δ T Across Fan	0.3†	4.1†	1.7	1.7	1.8	0.8†	1.4	1.5
Δ P Across Fan	3.1	2.9	3.2	2.9	3.4	3.4	3.2	2.9
% Hum. Before Cold Coil	24	44	17	17	20	24	19	17
% Hum. After Cold Coil	36	50	43	44	47	48	39	40
Flow - Air Over Coil	4651	5362	4215	5065	2638	2495	2621	4703
Flow - Chilled Water	12.6	5.3	48.9	49.1	44.6	44.2	11.0	18.4
Temp - Chilled Water - In	47.3	48.0	46.1	45.3	47.2	47.0	46.4	45.7
Temp - Chilled Water - Out	56.0	55.6	50.9	51.1	50.3	49.2	59.0	59.6
Temp - Cold Deck	56.4	55.4	49.9	50.3	49.2	48.7	55.8	57.0
CONTROLS								
Zone 1 - Set Point	74	74	74	74	74	74	80	80
Zone 1 - Thr. Range	7	7	7	7	7	7	7	7
Zone 2 - Set Point	77	77	77	77	77	77	80	80
Zone 2 - Thr. Range	10	10	10	10	10	10	10	10
Zone 3 - Set Point	76	76	76	76	76	76	80	80
Zone 3 - Thr. Range	12	12	12	12	12	12	12	12
Zone 4 - Set Point	74	74	74	74	74	74	79	79
Zone 4 - Thr. Range	10	10	10	10	10	10	10	10
Cold Deck - Set Point	55	55	49	49	49	49	58	58
Cold Deck - Thr. Range	5	5	5	5	5	5	5	5
Hot Water - Set Point			126	149	74	77	75	140
Hot Water - Thr. Range	38	38	38	38	38	38	38	38
Cooling Tower - Set Point, Nom.								
Free Cooling	25%	✓				25%		
Enthalpy Logic								
Deck Reset								
Fan Capacity Control								

* Units: loads -- thermal (MBH), loads -- electrical (MBH), temperature (°F), flows -- water (gpm). Metric conversions: 1000 Btu/hr/3.412 = kW; (°F-32)/1.8 = °C.

†probably not correct

Table 12

VAV With Reheat Test Series Data -- Tests 17 Through 21*

VAV -REHEAT SYSTEM TOTALS	17	18	19	20	21			
Loads (Cooling)	126	3.0	148	2.3	46			
Electrical	94	57	95	50	26			
Gas	37	71	39	97	52			
ELECTRICAL								
Chiller	60	32	62	27	0.4			
Tower	6.5	5.2	6.3	4.8	4.8			
Fan	18	11	18	8	11			
Pump-Cold Water	3.5	3.5	3.4	3.6	3.8			
Pump-Hot Water	1.4	1.6	1.4	1.8	1.7			
Pump-Cond Water	4.0	3.9	3.9	4.0	3.9			
BOILER								
Stand-by Loss	23	23	23	23	23			
Distribution Loss	6.2	1.8	7.2	2.0	3.5			
Load	0	46	0	51	14			
Fuel-Gas	37	71	39	97	52			
CHILLER								
Load	163	86	178	74	2.9			
Flow-Cond.	63.4	63.5	66.7	64.5	66.7			
Flow-Cold	45.4	51.0	49.5	52.6	58.0			
Temp-Cold In	51.4	49.1	51.4	48.8	46.6			
Temp-Cold Out	44.3	46.1	44.2	46.0	46.7			
Temp-Cond In	85.4	82.5	84.8	79.4	57.5			
Temp-Cond Out	92.3	85.6	91.6	82.0	57.4			

* Units: loads -- thermal (MBH), loads -- electrical (MBH), temperature (°F), flows -- water (gpm). Metric conversions: 1000 Btu/hr/3.412 = kW; (°F-32)/1.8 = °C.

Table 12 (Cont'd)

ZONES	17	18	19	20	21			
1 - Temp Supply Air	56.1	74.1	55.4	74.6	55.8			
1 - Flow-Air	1485	642	1614	665	612			
1 - Temp Zone	77.2	74.7	78.4	75.8	72.8			
1 - Load - Cooling	34	0.7	40	1.0	11			
1 - Load - Reheat	0	12	0	13	3.5			
2 - Temp Supply Air	56.4	75.8	55.7	74.9	51.0			
2 - Flow - Air	1242	699	1325	713	653			
2 - Temp Zone	79.0	75.7	79.7	75.9	70.0			
2 - Load - Cooling	30	0.8	34	0.7	13.5			
2 - Load - Reheat	0	13	0	13	0.5			
3 - Temp Supply Air	57.1	76.2	56.1	75.4	57.9			
3 - Flow-Air	1378	552	1538	756	566			
3 - Temp Zone	78.2	75.8	79.1	75.3	74.9			
3 - Load - Cooling	32	0.1	38	0	10			
3 - Load - Reheat	0	11	0	14	5			
4 - Temp Supply Air	56.6	70.9	55.7	71.9	57.5			
4 - Flow - Air	1444	682	1461	706	651			
4 - Temp Zone	75.9	72.7	77.7	72.8	72.7			
4 - Load - Cooling	30	1.4	35	0.7	11			
4 - Load - Reheat	0	10	0	11	5			

* Units: loads -- thermal (MBH), flows -- air (cfm), temperature (°F). Metric conversions: 1000 Btu/hr/3.412 = kW; 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C.

Table 12 (Cont'd)

AIRHANDLER	17	18	19	20	21			
Temp - Return Air	77.0	75.2	77.4	74.8	70.6			
Temp - Mixed Air	77.0	75.5	77.5	75.1	42.6			
Temp - O.D.A.-dry bulb	79.5	78.5	68.9	66.0	42.6			
Temp - O.D.A. - Wet bulb	58	55	47	52				
% ODA	0	7†	0	0	100			
Δ T Across Fan	2.7	2.7	1.7	1.3	5.1			
Δ P Across Fan	2.6	3.4	1.9[1]	1.9[1]	3.5			
% Hum. Before Cold Coil	14	14	12	12	42			
% Hum. After Cold Coil	35	32	28	25	44			
Flow - Air Over Coil	5549	2575	5928	2840	2482			
Flow - Chilled Water	27.0	13.7	29.2	52.6	0.8			
Temp - Chilled Water - In	45.0	46.5	44.7	48.8	51.4			
Temp - Chilled Water - Out	56.9	57.3	56.3	46.0	55.1			
Temp - Cold Deck	52.9	53.6	52.5	55.2	48.0			
CONTROLS								
Zone 1 - Set Point	75	75	75	75	74			
Zone 1 - Thr. Range	5	5	5	5	6			
Zone 2 - Set Point	76	76	76	76	77			
Zone 2 - Thr. Range	8	8	8	8	10			
Zone 3 - Set Point	76	76	76	76	76			
Zone 3 - Thr. Range	4	4	4	4	12			
Zone 4 - Set Point	73	73	73	73	75			
Zone 4 - Thr. Range	6	6	6	6	10			
Cold Deck - Set Point	57	58	57	57				
Cold Deck - Thr. Range	5	6	6	6				
Hot Water - Set Point	160	110	181	70				
Hot Water - Thr. Range	38	38	38	38				
Cooling Tower - Set Point, Nom.	81	81	81	81	81			
Free Cooling					✓			
Enthalpy Logic								
Deck Reset								
Fan Capacity Control			✓	✓				

* Units: loads -- thermal (MBH), flows -- air (cfm), flows -- water (gpm), temperature (°F), set point -- median of control range. Metric conversions: 1000 Btu/hr/3.412 = kW; 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C.

[1]Cold Deck Static Pressure controlled at 1.9" H_2O

†Probably not this high

Table 13

Terminal Reheat Test Series -- Index

Date	Test No.	System Description
5-16-#1	1	Basic system: no loads
5-16-#2	2	Basic system: no loads
5-16-#3	3	Basic system: 2 high, 2 low loads
5-19-#1	4	Basic system: all high loads
5-20-#1	5	Basic system: all high loads
5-21-#1	6	Basic system: all medium loads
5-22-#1	7	Basic system: all high loads, no ODA
5-22-#2	8	Basic system: no loads, no ODA
5-22-#3	9	Basic system: no loads, 33% ODA, 60°F (15°C) cold deck
5-22-#4	10	Basic system: high loads, ODA, 60°F (15°C)
5-30-#1	11	Basic system: medium loads, enthalpy logic, raining outside
6-02-#1	12	Basic system: high loads, cold deck reset
6-02-#2	13	Basic system: 2 high loads, 2 no loads, cold deck reset
6-02-#3	14	Basic system: no loads, cold deck reset
6-03-#1	15	Basic system: low loads, 52°F (11°C) cold deck, no ODA
6-03-#2	16	Basic system: high loads, 52°F (11°C) cold deck, no ODA
	17	Basic system: all high loads, free cooling
	18	Basic system: no loads, free cooling
	19	Basic system: low set points, no loads
	20	Basic system: low set points, high loads

Table 13 (Cont'd)

Test No.	System Description
21	Basic system: different zone set points, high loads, no ODA
22	Basic system: different zone set points, no ODA, low loads
23	Basic system: zone throttling ranges wide, high loads
24	Basic system: zone throttling ranges wide, no loads, 50°F (10°C)
25	Basic system: zone throttling ranges wide, no loads, 50°F (10°C) cold deck
26	Basic system: zone throttling ranges wide, high loads, 50°F (10°C) cold deck
27	Basic system: zone throttling ranges wide, high loads, 160°F (10°C) hot water
28	Basic system: zone throttling ranges wide, no loads, 160°F (10°C) hot water

Table 14

Terminal Reheat Test Series -- Test System Description

COOLING COIL: Six-row counter flow 33 x 57 in. (0.76 x 1.45 m).

CHILLER: 20-ton (70-kW) cold water generator, manufacturer's specification at full load COP = 3.4.

TOWER: 30-ton (100-kW) with capacity control by means of scroll dampers; equipped with a 5 hp (3.7 kW) motor.

FAN: One 15-in. fan/coil fan in blow-through arrangement 6400 cfm (3 m^3/s): 2.75 in. (0.7 kPa); 1320 rpm manufacturer's specification E = power to air/shaft power 34%); fan is 40% off; motor is 86% off.

PUMPS: conditioned water, 1-1/2 hp (1.1 kW); chilled water, 1-1/2 hp (1.1 kW); and hot water, 1 hp (0.745 kW).

Terminal Reheat

BASIC SYSTEM CONTROL SETTINGS:

Air volume: 1600 cfm (0.75 m^3/s) per zone nominal.

Room temperature controller set point (median of control range): 78°F (26°C) nominal; throttling range as small as possible reported.

Deck temperature controller: 55°F (13°C) nominal; throttling range as small as possible.

Chilled water temperature: 44°F (7°C) nominal.

Hot water temperature: 180°F (82°C) nominal; throttling range as small as possible reported.

Tower return water temperature: 80°F (27°C) nominal.

Boiler pressure controller: 5 psi (35 kPa) nominal.

Outdoor air: 30% nominal.

Table 15

Terminal Reheat Test Series Data -- Tests 1 Through 8*

TERMINAL-REHEAT SYSTEM TOTALS	1	2	3	4	5	6	7	8
Loads (Cooling)	0	0	72	139	138	76	138	1
Electrical	80	81	83	96	92	106	106	105
Gas	229	228	143	75	81	152	72	228
ELECTRICAL								
Chiller	44	45	47	59	55	68	67	66
Tower	5.3	5.5	5.6	6.8	6.8	7.9	8.3	8.3
Fan	21.5	21.6	21.6	21.2	21.3	21.3	21.7	21.8
Pump-Cold Water	3.5	3.5	3.5	3.4	3.4	3.4	3.4	3.4
Pump-Hot Water	1.5	1.5	1.5	1.5	1.5	1.4	1.5	1.5
Pump-Cond Water	4.0	4.0	3.9	4.0	4.0	3.9	3.9	3.9
BOILER								
Stand-by Loss	23	23	23	23	23	23	23	23
Distribution Loss	9.3	8.9	9.5	9.3	9.4	8.3	8.9	8.6
Load	146	146	79	26	31	87	24	146
Fuel-Gas	229	228	143	75	81	152	72	228
CHILLER								
Load	102	102	107	154	138	203	202	188
Flow-Cond.	62.6	63.1	64.5	63.2	63.9	62.9	63.4	62.8
Flow-Cold	48.5	48.4	48.5	47.4	46.8	50.2	48.3	46.0
Temp-Cold In	49.7	50.0	50.1	51.4	51.1	53.2	53.6	51.4
Temp-Cold Out	45.5	45.8	45.6	45.0	45.2	45.2	45.2	43.2
Temp-Cond In	84.1	84.1	84.1	85.8	85.3	86.2	86.6	86.2
Temp-Cond Out	88.4	88.3	88.5	92.1	91.0	94.2	94.5	93.9

* Units: loads -- thermal (MBH), loads -- electrical (MBH), temperature (°F), flows -- water (gpm). Metric conversions: 1000 Btu/hr/3.412 = kW; (°F-32)/1.8 = °C.

Table 15 (Cont'd)

ZONES	1	2	3	4	5	6	7	8
1 - Temp Supply Air	78.1	78.2	57.9	59.6	61.0	69.0	62.2	78.5
1 - Flow-Air	1666	1672	1679	1642	1645	1645	1692	1698
1 - Temp Zone	77.5	77.6	79.3	79.1	79.8	79.1	80.5	77.9
1 - Load - Cooling	0	0	36	34	34	18	33	0
1 - Load - Reheat	37	37	1	5	7	23	9	38
2 - Temp Supply Air	76.4	76.5	59.5	60.0	60.6	65.3	58.1	75.8
2 - Flow - Air	1608	1614	1618	1581	1588	1588	1591	1602
2 - Temp Zone	78.1	78.2	81.2	80.7	80.9	78.5	80.2	77.4
2 - Load - Cooling	0	0	36	32	34	22	35	0
2 - Load - Reheat	35	35	4	5	7	17	1	34
3 - Temp Supply Air	76.2	76.2	76.0	60.2	59.7	67.2	60.4	75.9
3 - Flow-Air	1683	1692	1681	1657	1664	1669	1719	1729
3 - Temp Zone	78.0	78.1	77.9	80.2	79.5	78.9	80.5	77.7
3 - Load - Cooling	0	0	0	36	36	21	36	1
3 - Load - Reheat	37	34	34	6	5	21	6	37
4 - Temp Supply Air	77.6	77.5	77.5	62.7	63.2	70.8	62.0	77.0
4 - Flow - Air	1710	1715	1672	1583	1587	1638	1638	1729
4 - Temp Zone	77.8	77.9	77.9	81.3	81.0	78.9	80.[illegible]	77.5
4 - Load - Cooling	0	0	0	34	34	15	34	0
4 - Load - Reheat	37	37	37	10	12	26	8	37

* Units: loads -- thermal (MBH), flows -- air (cfm), temperature (^{o}F). Metric conversions: 1000 Btu/hr/3.412 = kW; $1 m^3/s$ = 2119 cfm; $(^{o}F-32)/1.8 = {}^{o}C$.

Table 15 (Cont'd)

AIRHANDLER	1	2	3	4	5	6	7	8
Temp - Return Air	77.5	77.5	78.3	79.1	79.0	78.9	79.8	77.7
Temp - Mixed Air	69.5	71.3	72.0	75.3	74.0	78.5	79.5	77.6
Temp - O.D.A.-dry bulb	59.3	58.2	57.3	66.9	64.3	78.1	76.4	80.6
Temp - O.D.A. - Wet bulb	54	56	56	63	62	62	65	66
% ODA	45	33	30	32	33	45	1	0
Δ T Across Fan	1.5	2.5	3.1	1.2		2.2	2.0	2.6
Δ P Across Fan	2.4	2.4	2.4	2.4	2.5	2.4	2.3	2.3
% Hum. Before Cold Coil	39	40	42	41	42	21	30	29
% Hum. After Cold Coil	63	68	71	80	81	28	79	76
Flow - Air Over Coil	6667	6692	6606	6463	6484	6540	6640	6758
Flow - Chilled Water	17.7	18.3	18.4	26.0	24.5	38.8	32.0	26.2
Temp - Chilled Water - In	46.0	46.1	46.1	45.4	45.6	45.7	45.7	43.7
Temp - Chilled Water - Out	56.9	57.2	57.3	57.1	57.0	55.5	57.8	58.0
Temp - Cold Deck	54.8	55.0	55.0	54.6	54.5	52.6	54.9	54.4
CONTROLS								
Zone 1 - Set Point	78	78	78	78	78	78	78	78
Zone 1 - Thr. Range	3	3	3	3	6	5	5	5
Zone 2 - Set Point	78	78	78	78	78	77	77	77
Zone 2 - Thr. Range	5	5	5	5	4	3	3	3
Zone 3 - Set Point	78	78	78	78	78	79	79	79
Zone 3 - Thr. Range	5	5	5	5	7	4	4	4
Zone 4 - Set Point	78	78	78	78	78	78	78	78
Zone 4 - Thr. Range	7	7	7	7	6	4	4	4
Cold Deck - Set Point	57	57	57	57	57	57	57	57
Cold Deck - Thr. Range	5	5	5	5	5	6	5	5
Hot Water - Set Point	181	181	181	181	181	173	173	173
Hot Water - Thr. Range	38	38	38	38	38	38	38	38
Cooling Tower - Set Point, Nom.	81	81	81	81	81	81	81	81
Free Cooling								
Enthalpy Logic								
Deck Reset								
Fan Capacity Control								

* Units: loads -- thermal (MBH), flows -- air (cfm), flows -- water (gpm), temperature (^{o}F), set point -- median of control range. Metric conversions: 1000 Btu/hr/3.412 = kW; 1 m^3/s = 2119 cfm; (^{o}F-32)/1.8 = ^{o}C.

Table 16

Terminal Reheat Test Series Data -- Tests 9 Through 16*

TERMINAL-REHEAT SYSTEM TOTALS	9	10	11	12	13	14	15	16
Loads (Cooling)	0	136	84	130	64	0	0	135
Electrical	96	100	120	93	84	65	106	109
Gas	178	43	102	57	123	157	244	95
ELECTRICAL								
Chiller	58	62	78	54	45	27	63	68
Tower	7.4	7.8	13	9.2	9.2	7.2	11.8	12.2
Fan	21.8	20.9	20.0	20.9	20.9	20.9	21.7	21.7
Pump-Cold Water	3.7	3.6	3.8	3.7	3.8	3.9	3.8	3.8
Pump-Hot Water	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
Pump-Cond Water	3.9	3.9	4.0	4.0	4.0	4.0	4.0	4.0
BOILER								
Stand-by Loss	23	23	23	23	23	23	23	23
Distribution Loss	8.6	8.5	8.7	8.6	8.6	9.2	8.4	8.8
Load	107	2	48	13	64	75	159	42
Fuel-Gas	178	43	102	57	123	137	244	95
CHILLER								
Load	150	164	267	154	152	90	218	239
Flow-Cond.	62.5	62.8	63.5	65.2	64.9	64.8	64.5	64.3
Flow-Cold	53.2	51.2	54.1	53.8	53.8	57.7	57.2	57.2
Temp-Cold In	50.9	51.1	56.8	51.2	51.3	49.4	53.3	52.8
Temp-Cold Out	45.3	44.7	47.0	45.9	46.0	46.8	45.7	44.4
Temp-Cond In	86.1	86.4	84.3	78.3	78.4	77.2	80.1	80.6
Temp-Cond Out	92.2	93.3	94.8	84.5	83.7	80.3	88.2	89.6

* Units: loads -- thermal (MBH), loads -- electrical (MBH), temperature (°F), flows -- water (gpm). Metric conversions: 1000 Btu/hr/3.412 = kW; (°F-32)/1.8 = °C.

Table 16 (Cont'd)

ZONES	9	10	11	12	13	14	15	16
1 - Temp Supply Air	79.6	62.3	68.1	62.4	62.5	80.4	78.5	62.4
1 - Flow-Air	1619	1621	1558	1625	1630	1641	1686	1683
1 - Temp Zone	78.9	82.1	80.8	81.0	81.1	80.0	77.7	80.3
1 - Load - Cooling	0	33	21	32	32	0	0	32
1 - Load - Reheat	29	0	13	4	3	21	41	13
2 - Temp Supply Air	76.3	61.9	64.4	60.3	60.4	77.1	76.5	58.7
2 - Flow - Air	1538	1539	1518	1578	1586	1595	1642	1645
2 - Temp Zone	78.0	83.2	79.1	80.6	80.6	78.8	77.8	79.5
2 - Load - Cooling	0	34	24	32	32	0	0	33
2 - Load - Reheat	24	0	7	0	0	16	39	7
3 - Temp Supply Air	76.7	63.1	68.3	62.1	77.6	78.7	76.4	60.0
3 - Flow-Air	1649	1648	1551	1617	1614	1634	1678	1677
3 - Temp Zone	78.5	82.3	80.4	81.5	79.6	80.5	77.9	80.2
3 - Load - Cooling	0	35	20	33	0	0	0	37
3 - Load - Reheat	27	2	14	4	32	19	40	9
4 - Temp Supply Air	77.8	63.0	68.6	62.6	76.7	78.3	76.4	62.4
4 - Flow - Air	1631	1569	1552	1581	1601	1666	1740	1637
4 - Temp Zone	78.1	81.9	79.5	81.2	78.1	78.9	76.8	80.1
4 - Load - Cooling	0	23	19	33	0	0	0	35
4 - Load - Reheat	27	0	14	5	29	19	39	13

* Units: loads -- thermal (MBH), flows -- air (cfm), temperature (°F).
Metric conversions: 1000 Btu/hr/3.412 = kW; 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C.

Table 16 (Cont'd)

AIRHANDLER	9	10	11	12	13	14	15	16
Temp - Return Air	78.5	81.9	79.9	80.4	79.2	79.0	77.3	79.8
Temp - Mixed Air	79.0	81.1	76.6	76.0	75.6	76.3	77.3	79.7
Temp - O.D.A.-dry bulb	80.1	79.8	76.7	64.8	66.6	69.5	76.1	75.9
Temp - O.D.A. - Wet bulb	66	66	71	64	64	66	67	66
% ODA	34	40	98	28	28	28	0	3
Δ T Across Fan	2.7	1.8	2.8				2.1	1.9
Δ P Across Fan	2.4	2.3	2.5	2.4	2.4	2.3	2.3	2.3
% Hum. Before Cold Coil	30	29	53	49	48	51	32	29
% Hum. After Cold Coil	67	68	87	87	87	77	84	83
Flow - Air Over Coil	6437	6376	6179	6400	6431	6535	6746	6642
Flow - Chilled Water	16.0	17.5	51.0	24.6	22.4	5.9	54.8	54.8
Temp - Chilled Water - In	45.7	45.2	47.5	46.4	46.4	46.4	46.1	44.8
Temp - Chilled Water - Out	64.1	64.0	57.6	58.9	58.9	69.9	53.4	52.8
Temp - Cold Deck	60.6	60.0	57.4	58.0	58.3	67.5	52.4	51.7
CONTROLS								
Zone 1 - Set Point	78	78	78	78	78	78	78	78
Zone 1 - Thr. Range	5	5	5	5	5	5	5	5
Zone 2 - Set Point	77	77	77	77	77	77	77	77
Zone 2 - Thr. Range	3	3	3	3	3	3	3	3
Zone 3 - Set Point	79	79	78	78	78	78	78	78
Zone 3 - Thr. Range	4	4	6	6	6	6	5	5
Zone 4 - Set Point	78	78	78	78	78	78	78	78
Zone 4 - Thr. Range	4	4	4	4	4	4	4	4
Cold Deck - Set Point	63	63	58	*	*	*	52	52
Cold Deck - Thr. Range	5	5	6				6	4
Hot Water - Set Point	171	173	181	159	159	159		172
Hot Water - Thr. Range	38	38	38	38	38	38	38	38
Cooling Tower - Set Point, Nom.	81	81	81	81	81	81	81	81
Free Cooling								
Enthalpy Logic			✓					
Deck Reset				✓	✓	✓		
Fan Capacity Control								

* Units: loads -- thermal (MBH), flows -- air (cfm), flows -- water (gpm), temperature (^{o}F), set point -- median of control range. Metric conversions: 1000 Btu/hr/3.412 = kW; 1 m^3/s = 2119 cfm; $(^{o}F-32)/1.8 = {}^{o}C$.

Table 17

Terminal Reheat Test Series Data -- Tests 17 Through 24*

SYSTEM TOTALS	TEST 17	TEST 18	TEST 19	TEST 20	TEST 21	TEST 22	TEST 23	TEST 24
Loads (Cooling)	151	9	8	153	142	8	135	8
Electrical	72	75	98	119	102	96	100	97
Gas	61	222	197	43	67	192	82	227
ELECTRICAL								
Chiller	35	38	58	76	61	55	60	57
Tower	7.1	7.1	10.5	12.9	10.2	9.5	9.4	8.9
Fan	21.1	20.7	21.0	20.7	22.1	22.0	21.6	21.5
Pump-Cold Water	3.7	3.7	3.6	3.7	3.6	3.6	3.6	3.6
Pump-Hot Water	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
Pump-Cond Water	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0
BOILER								
Stand-by Loss	23	23	23	23	23	23	23	23
Distribution Loss	9.4	9.2	9.3	9.3	8.9	9.0	8.6	9.2
Load	15	141	101	1	20	118	32	145
Fuel-Gas	61	222	197	43	67	192	82	227
CHILLER								
Load	98	105	161	241	173	152	170	157

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.

Table 17 (Cont'd)

CHILLER (Cont'd)	TEST 17	TEST 18	TEST 19	TEST 20	TEST 21	TEST 22	TEST 23	TEST 24
Flow-Cond.	61.7	61.3	62.4	61.5	62.3	62.1	63.1	63.2
Flow-Cold	48.3	47.9	46.9	48.5	45.3	45.5	46.3	46.3
Temp-Cold In	49.9	50.3	51.3	53.0	51.4	57.3	57.3	51.4
Temp-Cold Out	46.0	46.0	44.5	43.1	43.8	44.7	44.0	44.6
Temp-Cond In	77.1	77.0	79.6	83.7	79.1	78.7	78.8	78.2
Temp-Cond Out	81.1	81.4	86.4	93.7	86.7	85.4	86.1	84.9
ZONES								
1 - Temp Supply Air	61.8	79.7	73.2	58.2	57.7	72.9	63.5	79.9
1 - Flow Air	1640	1614	1618	1596	1699	1702	1647	1645
1 - Temp Zone	82.1	79.2	73.4	80.6	78.0	73.0	81.4	78.8
1 - Load - Cooling	35	0	0	36	35	0	32	0
1 - Load - Reheat	7	38	26	0	0	26	10	39
2 - Temp Supply Air	58.0	75.4	71.0	58.3	57.7	71.9	60.2	75.3
2 - Flow - Air	1594	1570	1590	1565	1673	1676	1604	1604
2 - Temp Zone	80.8	77.8	73.2	81.4	78.4	74.2	81.1	77.4
2 - Load - Cooling	35	4	0	36	35	4	33	0
2 - Load - Reheat	0	33	24	0	0	27	4	33
3 - Temp Supply Air	59.3	770	70.5	58.9	59.4	75.3	59.9	75.1

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.

Table 17 (Cont'd)

ZONES (Cont'd)	TEST 17	TEST 18	TEST 19	TEST 20	TEST 21	TEST 22	TEST 23	TEST 24
3 - Flow-Air	1634	1608	1654	1627	1735	1742	1677	1679
3 - Temp Zone	81.3	99.2	72.8	80.7	79.1	77.0	79.9	77.1
3 - Load - Cooling	37	0	0	38	36	0	35	0
3 - Load - Reheat	3	36	25	0	2	25	4	35
4 - Temp supply Air	60.0	76.2	72.2	58.6	68.1	79.6	65.7	78.2
4 - Temp Zone	81.1	77.6	93.0	80.4	84.9	80.5	84.1	79.4
4 - Load - Cooling	37	0	0	32	33	0	33	0
4 - Load - Reheat	5	34	26	1	18	40	14	38
AIR HANDLER								
Temp - Return Air	80.2	77.9	73.2	20.0	79.6	76.3	80.9	78.1
Temp - Mixed Air	64.2	64.8	72.8	80.2	79.4	76.0	79.4	76.9
Temp - ODA - Dry bulb	64.1	64.8	72.3	80.5	79.3	79.1	76.5	74.9
Temp - ODA - Wet bulb	59	60	68	73	62	61	57	57
% ODA	99	100	45	33	33	0	33	33
Δ T Across Fan	36	3.5	2.3	2.2	2.4	2.8	1.1	1.7
Δ P Across Fan	2.5	2.6	2.4	2.4	2.1	2.1	2.1	2.1
% Hum. Before Cold Coil	51	51	44	59	24	26	16	17
% Hum. After Cold Coil	86	86	86	86	62	61	41	40

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.

Table 17 (Cont'd)

AIR HANDLER (Cont'd)	TEST 17	TEST 18	TEST 19	TEST 20	TEST 21	TEST 22	TEST 23	TEST 24
Flow - Air Over Coil	6453	6403	6532	6333	6767	6230	6533	6580
Flow - Chilled Water	18.9	194	28.7	36.7	24.6	23.9	25.3	26.3
Temp - Chilled Water - In	46.4	46.7	44.7	43.5	44.3	45.0	44.4	45.0
Temp - Chilled Water - Out	56.7	56.9	56.9	57.1	59.7	59.2	58.8	58.1
Temp - Cold Deck	55.3	55.3	55.3	55.8	55.5	55.4	54.8	54.4
CONTROLS								
Zone 1 - Set Point	78	78	72	72	71	71	81	81
Zone 1 - Thr. Range	5	5	7	7	5	5	6	6
Zone 2 - Set Point	77	77	72	72	74	74	77	72
Zone 2 - Thr. Range	3	3	4	4	3	3	8	8
Zone 3 - Set Point	78	78	71	71	77	77	78	78
Zone 3 - Thr. Range	6	6	4	4	4	4	7	7
Zone 4 - Set Point	78	78	71	71	82	82	80	80
Zone 4 - Thr. Range	4	4	6	6	6	6	10	10
Cold Deck - Set Point		60	58	58	60	60	59	59
Cold Deck - Thr. Range		6	6	6	6	6	6	6
Hot Water - Set Point		172	174	174	172	172	171	171

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.

Table 17 (Cont'd)

CONTROLS (Cont'd)	TEST 17	TEST 18	TEST 19	TEST 20	TEST 21	TEST 22	TEST 23	TEST 24
Hot Water - Thr. Range	38	38	38	38	38	38	38	38
Cooling Tower - Set Point, Nom.	81	81	81	81	81	81	81	81
Free Cooling								
Enthalpy Logic								
Deck Reset								
Fan Capacity								

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.

Table 18

Terminal Reheat Series Data -- Tests 25 Through 28*

TERMINAL-REHEAT SYSTEM TOTALS	TEST 25	TEST 26	TEST 27	TEST 28
Loads (Cooling)	0	139	111	0
Electrical	104	102	99	99
Gas	254	116	102	208
ELECTRICAL				
Chiller	63	61	59	59
Tower	9.7	9.3	9.3	9.3
Fan	21.5	21.6	21.3	21.3
Pump-Cold Water	3.8	3.8	3.6	3.6
Pump-Hot Water	1.5	1.5	1.5	1.5
Pump-Cond Water	4.0	4.0	4.0	4.0
BOILER				
Stand-by Loss	23	23	23	23
Distribution Loss	9.1	9.2	6.9	7.0
Load	166	58	50	132
Fuel-Gas	254	116	102	208

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8°C; 1000 Btu/hr 3.412 = kW.

Table 18 (Cont'd)

	TEST	TEST	TEST	TEST
CHILLER				
Load	211	195	211	160
Flow-Cond.	63.3	62.4	61.8	61.7
Flow-Cold	57.8	52.9	44.6	44.4
Temp-Cold In	51.2	49.9	51.7	51.6
Temp-Cold Out	43.9	42.6	46.8	44.4
Temp-Cond In	78.7	78.8	78.9	78.9
Temp-Cond Out	86.6	86.5	86.2	86.0
ZONES				
1 - Temp Supply Air	78.9	61.5	65.5	77.2
1 - Flow-Air	1650	1657	1641	1641
1 - Temp Zone	78.1	80.5	80.4	77.3
1 - Load - Cooling	0	34	27	0
1 - Load - Reheat	44	16	14	35
2 - Temp Supply Air	75.1	58.3	62.4	74.6
2 - Flow - Air	1611	1614	1600	1600
2 - Temp Zone	77.0	80.5	80.1	76.8
2 - Load - Cooling	0	34	28	0
2 - Load - Reheat	40	11	9	32

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (^{o}F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (^{o}F-32)/1.8^{o}C; 1000 Btu/hr/3.412 = kW

Table 18 (Cont'd)

	TEST 25	TEST 26	TEST 27	TEST 28
3 - Temp Supply Air	74.2	67.5	61.9	73.7
3 - Flow-Air	1685	1685	1671	1671
3 - Temp Zone	76.3	79.0	79.1	76.0
3 - Load - Cooling	0	3.2	29	0
3 - Load - Reheat	40	10	9	32
4 - Temp Supply Air	76.7	64.2	67.4	75.4
4 - Flow - Air	1683	1610	1613	1657
4 - Temp Zone	77.7	81.7	82.6	77.3
4 - Load Cooling	0	34	27	0
4 - Load - Reheat	42	21	18	33
AIRHANDLER				
Temp - Return Air	77.2	79.4	79.7	76.6
Temp - Mixed Air	76.3	76.3	78.6	76.7
Temp - ODA Dry Bulb	74.8	70.9	76.7	77.1
Temp - ODA - Wet Bulb	58	58	59	60
% ODA	33	33	33	33
Δ T Across Fan	1.9	0.2**	1.4	2.4
Δ P Across Fan	2.1	2.1	2.1	2.1

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (^{o}F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (^{o}F-32)/1.8^{o}C; 1000 Btu//hr/3.412 = kW.

**Probably not correct.

Table 18 (Cont'd)

	TEST	TEST	TEST	TEST
% Hum. Before Cold Coil	18	24	20	20
% Hum. After Cold Coil	48	63	49	49
Flow - Air Over Coil	6628	6666	6576	6570
Flow - Chilled Water	52.8	45.7	26.6	26.6
Temp - Chilled Water - In	44.3	43.0	44.9	45.0
Temp - Chilled Water - Out	51.6	51.3	58.5	58.1
Temp - Cold Deck	49.8	49.1	54.6	54.4
CONTROLS				
Zone 1 - Set Point	81	81	81	81
Zone 1 - Thr. Range	6	6	6	6
Zone 2 - Set Point	77	77	77	77
Zone 2 - Thr. Range	8	8	8	8
Zone 3 - Set Point	78	78	78	78
Zone 3 - Thr. Range	7	7	7	7
Zone 4 - Set Point	80	80	80	80
Zone 4 - Thr. Range	10	10	10	10
Cold Deck - Set Point	50	50	57	57
Cold Deck - Thr. Range	6	6	6	6

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8°C; 1000 Btu/hr/3.412 = kW.

AD-A115 182 CONSTRUCTION ENGINEERING RESEARCH LAB (ARMY) CHAMPAIGN IL F/G 13/1
VALIDATION DATA FOR MECHANICAL SYSTEM ALGORITHMS USED IN BUILDI--ETC(U)
FEB 82 W DOLAN IAA-EM-78-1-01-4207
UNCLASSIFIED CERL-TR-E-177 NL

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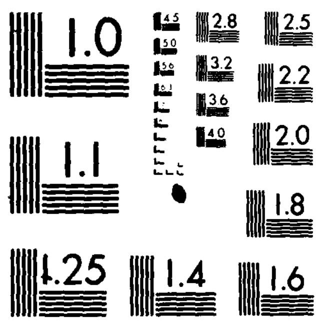

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS 1963-A

Table 18 (Cont'd)

	TEST	TEST	TEST	TEST
Hot Water - Set Point	180	180	140	140
Hot Water - Thr. Range	38	38	38	38
Cooling Tower - Set Point.	Nom.81	81	81	
Free Cooling				
Enthalpy Logic				
Deck Reset				
Fan Capacity Control				

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8°C; 1000 Btu/hr/3.412 = kW.

Table 19

Dual Duct VAV Test Series -- Index*

Test Number	Description
1	Basic system, heating loads all zones
2	Basic system, heating loads all zones, 33% outdoor air
3	Basic system, 2 cooling loads, 2 heating loads, 33% outdoor air
4	Basic system, no zone loads, 33% outdoor air
5	2 cooling loads, 2 heating loads, economizer, chiller off
6	2 cooling loads, 2 heating loads, economizer, chiller on
7	2 cooling loads, 2 heating loads, no outdoor air
8	Same as test No. 7, cold deck setting up 5°F, hot deck setting down 5°
9	Same as test No. 8
10	2 cooling loads, 2 heating loads, deck settings returned, 30% ODA
11	2 cooling loads, 2 heating loads, zone set points changed
12	Mild zone loads, no fan capacity control, no ODA
13	2 cooling loads, 2 heating loads, no fan capacity control
14	2 cooling loads, 2 heating loads, deck reset

*Metric conversion: $(^{o}F-32)/1.8 = {}^{o}C$.

Table 20

Dual Duct VAV -- Test System Description*

Cooling Coil: Six-row counter flow 33 X 57 in.

Heating Coil: Two-row 18 X 57 in.

Chiller: 20-ton cold water generator, water cooled. Manufacturer's full load COP -- 3.4

Tower: 30-ton, 5 hp tower with scroll dampers for capacity control.

Fan: One 15-in. forward-curved blade fan in a blow-through arrangement fitted with inlet guide vanes. The static pressure across the fan could be controlled by positioning the inlet guide vanes (effecting the fan capacity) using an electronic reset controller (proportional plus integral control). From manufacturer's specifications 6400 cfm at 2.75 in. wc, 1320 rpm, 40% efficiency. The 10-hp fan motor was about 86% efficient at this load.

Pumps: Chilled water pump 50 gpm nom, 1-1/2 hp; hot water pump 15 gpm nom, 1 hp; condenser water pump 75 gpm nom, 1-1/2 hp.

Zone Air Boxes: Two VAV boxed controlled by one thermostat and calibrated to deliver a minimum 400 cfm per zone (25% of design maximum). This calibration was observed to drift.

Basic System Control Settings

Air Volume: 1600 cfm per zone nominal, 400 cfm minimum

Zone Set Points, Deck Set Points, as noted.

Chilled Water: 44°F nominal

Tower Water: As noted.

Outdoor Air: As noted.

Boiler Pressure Controller: 5 psi nom.

* Metric conversions: 1 in. = 25.4 mm; 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1 psi = 6.9 kPa; 1 hp = 0.74 kW.

Table 21

Dual Duct VAV Test Series Data -- Tests 1 Through 6*

DUAL DUCT-VAV							
SYSTEM TOTALS	1	2	3	4**	5	6	7
Zone Loads (Cooling)	0	0	36.3	0	45.7	46.1	48.9
Zone Loads (Heating)	75.5	40.6	31.5	0.3	27	31.4	36.4
Electrical (kW)	5.91	4.7	4.5	2.8	4.1	9.2	18.3
Gas	145	174	151	118	160	163	108
ELECTRICAL							
Chiller (kW)	0.11	0.11	0.11	0.11	1.55	10.46	10.95
Tower (kW)						1.45	1.45
Fan (kW)	3.26	4.04	3.89	2.19	3.53	3.60	3.68
Pump-Cold Water (kW)						0.98	0.92
Pump-Hot Water (kW)	0.53	0.53	0.52	0.53	0.50	0.51	0.51
Pump-Cond Water (kW)						1.13	1.12
BOILER							
Stand-by Loss	24	24	24	24	24	24	24
Distribution Loss	13	10	10	5	12	12	11
Load	90	113	95	69	102	104	61
Fuel-Gas	145	174	151	118	160	163	108
CHILLER	off	off	off	off	off		
Load						10.2	58.4
Flow-Cond						60.9	60.0
Flow-Cold						47.5	44
Temp-Cold In						47.6	50.1
Temp-Cold Out						47.3	47.4
Temp-Cond In						63.9	79.6
Temp-Cond Out						69.3	82.3

* Units: Loads -- thermal (MBH), Loads -- electrical (kW), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.

** Zone load data and component electrical data are accurate. Some data on air handler performance are not consistent.

Table 21 (Cont'd)

DUAL DUCT-VAV

SYSTEM TOTALS	1	2	3	4	5	6	7
AIRHANDLER							
Temp - Return Air	68.7	64.6	67.0	67.8	72.3	71.8	73.5
Temp - Mixed Air	68.3	49	49.2	46.8	48.0	48.9	71.5
Temp - ODA-dry bulb	68.2	38.3	38.4	39.3	47.2	46.9	47.7
r.h. - ODA rh (%)	17	40	42	44	53.6	57	54
% ODA	6	59	62	74	93	92	8
T Across Fan	2.2	2.0	2.3	3.3	2.1	2.1	2.1
P Across Fan	2.0	1.9	1.9	2.2	2.1	2.0	2.0
% Hum. Before Cold Coil	13	22	21	28	41	45	17
% Hum. After Cold Coil	off	--	--	--	off	57	37
Flow - Air Over Cooling	741		2201	592	2096	2072	2351
Flow - Air Over Heating	3808	5064	3154	1062	2296	2435	2206
Flow - Chilled Water						3.3	14.5
Flow - Heated Water	6.7	7.2	7.1	5.2	6.3	6.5	2.8
Temp - Chilled Water - In						47.6	47.9
Temp - Chilled Water - Out						51.1	55.3
Temp - Heating Water - In	128	122	120	132	135	135	138
Temp - Heating Water - Out	105	93.2	96.4	108	107	107	103
Temp - Cold Deck	70.7	--	51.2	49.0	51.4	50.9	54.2
Temp - Hot Deck	88	69.6	75.8	91.0	45.5	84.8	90.2
Load (Water) - Cold Deck						6	53
Load (Water) - Hot Deck	76	103	86	64	89	92	50
CONTROLS							
Zone 1 Set Point	76	76	76	76	73	73	73
Zone 1 - Thr. Range	4.5	4.5	4.5	4.5	4.5	4.5	4.5
Zone 2 - Set Point	70	70	70	70	72	72	72
Zone 2 - Thr. Range	4.7	4.7	4.7	4.7	5.6	5.6	5.6

* Units: Loads -- thermal (MBH), Loads -- electrical (kW), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.

** Zone load data and component electrical data are accurate. Some data on air handler performance are not consistent.

Table 21 (Cont'd)

DUAL DUCT-VAV

SYSTEM TOTALS	1	2	3	4	5	6	7
Zone 3 - Set Point	70	70	70	70	72	72	72
Zone 3 - Thr. Range	6.3	6.3	6.3	6.3	6.5	6.5	6.5
Zone 4 - Set Point	69	69	69	69	72	72	72
Zone 4 - Thr. Range	10.6	10.6	10.6	10.6	7.3	7.3	7.3
Cold Deck - Set Point					55	55	55
Cold Deck - Thr. Range					8	8	8
Hot Deck - Set Point	90	90	90	90	90	90	90
Hot Deck - Thr. Range	13	13	13	13	11	11	11
Hot Water - Set Point	125	125	125	135	135	135	135
Hot Water - Thr. Range	10	10	10	10	10	10	10
Cooling Tower - Set Point, Nom.	off	off	off	off	off	70	80
Free Cooling					X	X	
Enthalpy Logic							
Deck Reset							
Fan Capacity Control (2-in. wc)	X	X	X	X	X	X	X

ZONES							
1 - Temp Supply Air	85.2	69.6	74.3	73.2	82.0	82.6	86.4
1 - Flow-Air	1241	1345	1586	223	1118	1186	1058
1 - Temp Zone	70.4	64.2	65.5	70.0	70.8	70.7	71.0
1 - Load - Cooling	--	--	--	--	--	--	--
1 - Load - Heating	19.8	7.8	15.1	.3	13.5	15.3	17.7
2 - Temp Supply Air	85.2	69.7	74.5	lost	82.0	83.0	87.1
2 - Flow-Air	1057	1263	1469	lost	1086	1155	1059
2 - Temp Zone	69.5	63.2	64.2	70.6	70.5	70.1	70.8
2 - Load - Cooling	--	--	--	--	--	--	--
2 - Load - Heating	18	8.9	16.4	--	13.5	16.1	18.7
3 - Temp Supply Air	83.1	69.4	55.8	69.2	55.5	55.1	57.6
3 - Flow-Air	1341	1823	1375	114	1209	1185	1343
3 - Temp Zone	68.3	62.9	69.9	69.2	73.5	73.3	74.3
3 - Load - Cooling	--	--	20.9	--	23.7	23.5	24.3
3 - Load - Heating	21.6	12.7	--	--	--	--	--
4 - Temp Supply Air	83.5	69.6	55.3	69.8	54.8	54.4	57.1
4 - Flow-Air	890	1333	917	196	978	981	1096
4 - Temp Zone	66.7	61.8	70.8	68.4	75.6	75.6	76.1
4 - Load - Cooling	--	--	15.4	--	22.0	22.6	22.6
4 - Load - Heating	16.1	11.2	--	--	--	--	--

* Units: Loads -- thermal (MBH), Loads -- electrical (kW), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.

** Zone load data and component electrical data are accurate. Some data on air handler performance are not consistent.

Table 22

Dual Duct VAV Test Series Data -- Tests 7 through 11*

DUAL DUCT-VAV

SYSTEM TOTALS	8**	9	10	11**	12**	13**	14**
Zone Loads (Cooling)	46.4	41.3	39.7	39.4	39.5	38.7	37.8
Zone Loads (Heating)	34.3	34	32.5	12.3	16.5	16.1	16.7
Electrical (kW)	19.1	15.4	9.4	16.8	15.3	18.7	16.1
Gas	96	83	136	72	88	82	72
ELECTRICAL							
Chiller (kW)	10.9	7.15	1.44	9.42	7.93	11.32	9.04
Tower (kW)	1.45	1.45	1.49	1.43	1.43	1.42	1.47
Fan (kW)	4.09	4.17	3.80	3.30	3.39	3.38	3.46
Pump-Cold Water (kW)	.95	.96	.97	.94	.94	.95	.96
Pump-Hot Water (kW)	.51	.53	.54	.54	.52	.53	.52
Pump-Cond Water (kW)	1.11	1.12	1.13	1.12	1.12	1.13	1.11
BOILER							
Stand-by Loss	24	24	24	24	24	24	24
Distribution Loss	8	5	4	5	5	6	5
Load	52	42	83	33	46	41	33
Fuel-Gas	96	83	136	72	88	82	72
CHILLER							
Load	53.5	57.1	26	65.7	41.5	51	48
Flow-Cond	59.7	60.2	60.7	64.8	65.1	64.8	65.1
Flow-Cold	45.7	45.2	46.4	43.7	43.7	44.1	46
Temp-Cold In	49.7	49.3	47.3	49.7	48.7	48.2	49.4
Temp-Cold Out	47.4	47.0	46.6	46.7	47.1	45.9	47.3
Temp-Cold In	88.4	68.3	53.9	78.5	74.7	78.6	76.1
Temp-Cond Out	91.1	71.0	54.7	81.5	76.6	81.3	81.8
Temp - Return Air	73.9	72.6	71.1	75.6	75.1	76.2	75.7
Temp - Mixed Air	71.1	69.9	54.3	71.3	61.3	67.1	71.1
Temp - ODA-dry bulb	47.8	45.2	41.6	59	60.9	62.8	59.2
RH - ODA-rh	53	46	70	50	46	48	49
% ODA	11	10	57	26	97	68	27
Δ T Across Fan	2.0	2.1	2.1	3.2	3.2	3.2	3.1
Δ P Across Fan	1.97	1.95	2.00	3.29	3.35	3.33	3.26

* Units: loads -- thermal (MBH), Loads -- electrical (MBH), temperatures (°F), flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; 1000 Btu/hr/3.412 = kW; (°F-32)/1.8 = °C.

** Zone load data and component electrical data are accurate. Some data on air handler performance are not consistent.

Table 22 (Cont'd)

DUAL DUCT-VAV

SYSTEM TOTALS	8**	9	10	11**	12**	13**	14**
% Hum. Before Cold Coil	17	14	30	33	41	36	22
% Hum. After Cold Coil	32	28	37	70	62	71	63
Flow-Air Over Cooling	2640	2370	1870	1730	1570	1550	1670
Flow-Air Over Heating	2430	2840	2940	950	1160	1180	1260
Flow-Chilled Water	9.3	9.4	5.8	19.2	16.8	13.9	20
Flow-Heated Water	2.0	1.9	6.5	2.0	3.4	2.9	2.3
Temp-Chilled Water-In	47.9	47.6	46.8	47.4	47.4	46.5	47.4
Temp-Chilled Water-Out	59.2	57.8	51.8	53.6	51.3	52.7	53.6
Temp-Heating Water-In	137.5	128.6	124.2	128.4	127.8	126.9	125
Temp-Heating Water-Out	95.9	89.3	99.9	100.9	104	103	97.3
Temp-Cold Deck	58.4	57.2	51.5	51.2	50.2	51	51.6
Temp-Hot Deck	85.9	82.4	80.4	92	90.6	90.9	87.2
Load (Water)-Cold Deck	53	47	14	54	31	43	60
Load (Water)-Hot Deck	44	37	79	28	41	35	28

CONTROLS							
Zone 1 Set Point	73	73	73	68	68	68	68
Zone 1-Thr. Range	4.5	4.5	4.5	6.5	6.5	6.5	6.5
Zone 2-Set Point	72	72	72	64	64	64	64
Zone 2-Thr. Range	5.6	5.6	5.6	6.7	6.7	6.7	6.7
Zone 3-Set Point	72	72	72	76	76	76	76
Zone 3-Thr. Range	6.5	6.5	6.5	6.5	6.5	6.5	6.5
Zone 4-Set Point	72	72	72	78	78	78	78
Zone 4-Thr. Range	7.3	7.3	7.3	10.6	10.6	10.6	10.6
Cold Deck-Set Point	60	60	60	54.5	54.5	54.5	54.5
Cold Deck-Thr. Range	8	8	8	4.5	4.5	4.5	4.5
Hot Deck-Set Point	85	85	85	92	92	92	92
Hot Deck-Thr. Range	11	11	11	4	4	4	4
Hot Water-Set Point	140	130	130	130	130	130	130
Hot Water-Thr. Range	10	10	10	10	10	10	10
Cooling Tower-Set Point, Nom.	85	No Cntl	No Cntl	80	80	80	

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), temperatures (°F), flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; 1000 Btu/hr/3.412 = kW; (°F-32)/1.8 = °C.

** Zone load data and component electrical data are accurate. Some data on air handler performance are not consistent.

Table 22 (Cont'd)

DUAL DUCT-VAV

SYSTEM TOTALS	8**	9	10	11**	12**	13**	14**
Free Cooling				X			
Enthalpy Logic							
Deck Reset						X	
Fan Capacity Control	X	X					
ZONES							
1 - Temp Supply Air	83.7	80.2	78.9	78.4	82.6	83	78.7
1 - Flow-Air	1169	1298	1387	368	538	516	497
1 - Temp Zone	70.8	69.6	69.4	65.9	68	68.2	65.8
1 - Load - Cooling	--	--	--	--	--	--	--
1 - Load - Heating	16.5	15	14.3	5.4	8.8	8.5	7.3
2 - Temp Supply Air	84.3	80.8	79.3	77.2	80.5	80.4	76.7
2 - Flow-Air	1161	1430	1454	520	482	491	649
2 - Temp Zone	70.1	68.5	67.8	65.4	66.4	66.5	64.6
2 - Load - Cooling	--	--	--	--	--	--	--
2 - Load - Heating	17.8	19	18.2	6.9	7.6	7.6	9.4
3 - Temp Supply Air	60.4	60.0	55.9	58.4	57.4	58.1	58.6
3 - Flow-Air	1494	1324	1037	947	886	897	930
3 - Temp Zone	75.2	74.4	74.3	77.5	77.5	77.6	77.6
3 - Load - Cooling	23.4	20.7	19.7	20.2	19.8	19.5	19.1
3 - Load - Heating							
4 - Temp Supply Air	60.0	59.6	55.3	58.3	57.0	57.8	58.4
4 - Flow-Air	1245	1155	936	846	826	827	853
4 - Temp Zone	76.7	76.0	75.0	78.6	78.5	78.7	78.6
4 - Load - Cooling	22.5	20.6	20	19.2	19.7	19.2	18.7
4 - Load - Heating							

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), temperatures (°F), flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; 1000 Btu/hr/3.412 = kW; (°F-32)/1.8 = °C.

** Zone load data and component electrical data are accurate. Some data on air handler performance are not consistent.

Table 23

Dual Duct/Multizone -- Index

Test Number	Description
1	Basic system, high cooling loads all zones, 10% ODA.
2	Basic system, moderate cooling loads all zones, 40% ODA.
3	2 zones under no load, 3 zones with moderate cooling load.
4	2 zones under no load, 2 zones with moderate cooling loads, warm ODA.
5	Cooling loads all zones, hot deck off.
6	Cooling loads all zones, new zone set points.
7	Heating loads 2 zones, cooling loads 2 zones, no ODA.
8	Cooling loads, 25% ODA, moderate temperature.
9	No zone loads, 25% ODA, moderate temperature.
10	Heating loads 2 zones, cooling loads 2 zones, 25% ODA.
11	Cooling loads 2 zones, no load 2 zones.
12	High cooling loads, all zones.
13	High cooling loads, all zones, 40% ODA.
14	High cooling loads, all zones, 5% ODA.

Table 24

Dual Duct/Multizone -- Test System Description

A heating and cooling all-air system where dampers control whether zone supply air passes through the air handler cooling coil, the air handler heating coil, or a combination of both.

Cooling Coil: Six-row counter flow 33 x 57 in.*

Heading Coil: Two-row 18 x 57 in.

Chiller: 20-ton cold water generator, water cooled.
Manufacturer's reported full load COP = 3.4.

Tower: 30-ton, 5 hp tower with scroll dampers for capacity control.

Fan: One 15-in. forward-curved blade fan in a blow-through, built-up air handler. Fan speed was set at 1320 rpm. (The fan-motor speed ratio is adjustable in steps by means of the variable diameter pulley. The fan rotative speed was set so as the fan capacity was 6400 cfm at the highest resistance condition.)

Zone Air Supply: Hot and cold supply air dampers were controlled by one pneumatic motor energized by the zone thermostat (standard multizone control and standard dual duct control except for the dual duct mixing boxes which also control the mixed air delivery rate).

Heating Plant: A natural gas boiler (name plate ratings of 450 000 Btu/hr input, 360 000 Btu/hr) was controlled by a pressurestat operating between 4 and 6 psig. A steam-to-water converter supplied the heating for the heating water supply. A three-way mixing valve and a pneumatic receiver/controller controlled the portion of the circulating hot water which passed through the converter, thus accomplishing temperature control of the circulating hot water.

Basic Control Settings:

Fan capacity -- 1400 cfm (nominal)

Chilled water -- 44°F, 8°F, control range

All others as noted.

*Metric conversions: 1 in. = 25.4 mm; 1 hp = 0.14 kW; 1 m^3/s = 2119 cfm; 1000 Btu/hr = 3.412 kW; 1 psi = 6.9 kPa; °F = (°C x 1.8) +32.

Table 25

Dual Duct/Multizone Test Series Data -- Tests 1 Through 7*

	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6	Test 7
SYSTEM TOTALS							
Zone loads cooling	133	80	45	64	104	113	58
Zone loads heating							30
Total electrical	31.2	26.2	25.9	35.2	28.2	30.8	26.5
Gas	44	81	100	0	0	0	122
ELECTRICAL							
Chiller	18.2	14.4	14.0	22.4	15.8	16.1	14.0
Tower	3.39	2.5	2.6	3.66	3.1	3.5	2.94
Fan	6.94	6.61	6.57	6.43	6.59	6.51	6.91
Pump - cold	1.08	1.07	1.08	1.10	1.05	1.04	1.04
Pump - cond.	1.14	1.15	1.15	1.14	1.14	1.14	1.14
Pump - hot	.50	.51	.52	.52	.49	.48	.49
BOILER							
Stand-by	23	23	23	Off	Off	Off	23
Load	.11	40	55		5	12	72
Fuel-gas	43.6	80.8	100	0	0	0	121.8
CHILLER							
Load	193	133	128.9	246	163	197	129
Flow - cold	56	54	55	57	54	53	54
Flow - cond	60	60	60	62	62	62	62
Temp - cold in	50.3	50.5	50.6	54.1	51.6	50.7	50.5
Temp - cold out	43.4	45.6	45.9	45.4	45.6	43.3	45.7
Temp cond - in	78.6	77.3	76.7	85.2	77.3	78.9	77.0
Temp cond - out	86.5	82.9	82.1	95.2	83.8	86.9	82.4
CONTROL							
Zones							
1 set point/thr. range	73.2/4.1	73.5/4.1	72.5/4.1	73.9/3.5	75.2/3.5	77.1/3.5	74.1/3.5
2 set ppoint/thr. range	70.6/4.3	70.0/4.3	70.9/4.3	70.7/3.7	70.7/5.7	71.4/3.7	70.8/3.7
3 set point.thr. range	72.6/3.5	71.9/3.5	72.0/3.5	74.0/3.7	73.6/3.7	78.1/3.7	79.2/3.7
4 set point/thr. range	75.2/6	70.6/6	70.6/6	72.3/3	72.1/3	73.7/3	73.7/3
Hot deck set pt/thr.	78/16	87/6	81/6	Off	Off	Off	96/8
Cold deck set pt/thr.	51/4	54/7	54/7	56/4	54/4	54/4	55/5
Hot water set pt/thr.	150	125	125	Off	Off	Off	160
Tower set pt (thr = 10°F)	80	80	80	80	80	80	80
Fuel cooling							
Deck reset							

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.

Table 25 (Cont'd)

	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6	Test 7
ZONES							
1 zone temp	75.1	73.6	72.0	73.6	75.6	77.8	73.3
1 supply temp	58.2	64.1	71.4	72.4	62.4	62.5	82.1
1 Air flow	1883	1957	1837	1773	1899	1902	1892
1 load cooling	34.7	20.2	1.3	2.4	27.4	31.8	18.1
1 load heating							
2 zone temp	75.3	73.5	70.4	71.6	72.2	73.1	71.1
2 supply temp	55.6	61.1	68.2	68.9	57.0	57.0	77.5
2 air flow	1697	1693	1703	1616	1572	1605	1730
2 load cooling	36.2	22.9	4.2	4.7	26	28.1	
2 load heating							
3 zone temp	73.9	73.3	73.4	75	74.9	80.2	80.8
3 supply temp	58.3	63.2	63.1	59.5	61.4	65.3	66.0
3 air flow	1791	1651	1665	1696	1731	1607	1695
3 load cooling	30.4	18.2	18.7	28.8	25.4	26.2	27.2
3 load heating							
4 zone temp	76.2	73.1	72.7	74.1	73.2	75	74.8
4 supply temp	59.4	61.6	60.9	57.6	57.9	58.2	59.4
4 air flow	1730	1526	1586	1552	1530	1520	1768
4 load cooling	31.6	19.1	20.4	28.0	25.6	27.9	29.7
% ODA	10	40	47	62	41	40	4
ΔT across fan	2.5	2.4	2.4	2.4	2.5	2.5	2.5
ΔP across fan	2.1	2.0	2.0	2.0	2.1	2.0	2.1
% r.h. before c.c.	16	25	33	48	43	41	41
% r.h. after c.c.	72	76	79	100	100	100	99
Flow cold deck	5680	4812	3987	4520	5358	5143	3979
Temp cold deck	50.7	51.9	51.9	54.3	54.1	53.9	54.0
Flow hot deck	1422	2016	2804	2118	1374	1490	3105
Temp hot deck	82.4	87.5	86.7	84.3	77.8	80.8	94.0
Temp hot water in	151.7	122.1	122.7	93.6	76.2	76.5	154.8
Temp hot water out	89.8	100.3	101.8	83.5	76.4	77.0	107.8
Temp cold water in	43.9	45.9	46.3	45.9	45.9	43.9	46.1
Temp cold water out	51.2	52.5	52.1	54.1	56.2	56.2	56.2
Load (water) heat coil	7	38	53	--	--	--	65
Load (water) cool coil	174	122	113	224	150	175	115
Return air temp	74.8	72.7	72.3	74.9	74.3	76.6	75.9
Mixed air temp	75.3	70.6	68.8	82.5	75.5	78.8	76.2
Temp ODA d.b.	79.8	67.4	64.9	87.2	77.2	82.1	83.1
r.h. ODA	35	54	65	87	77	37	34

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.

Table 26

Dual Duct/Multizone Test Series Data -- Tests 8 Through 14*

	Test 8	Test 9	Test 10	Test 11	Test 12	Test 13	Test 14
SYSTEM TOTALS							
Zone loads cooling	98.9	6.2	41.7	46.7	129.5	96.2	98.0
Zone loads heating		1.4	16.6				
Total electrical	27.5	21.5	48.7	76.8	32.7	35.6	32.3
Gas							
ELECTRICAL							
Chiller	14.67	9.62	14.9	15.79	19.6	23.12	18.84
Tower	3.19	2.34	2.65	2.71	3.81	3.76	3.76
Fan	7.00	6.80	6.93	6.91	6.64	6.45	6.95
Pump - cold	1.04	1.06	1.04	1.04	1.04	1.09	1.05
Pump - cond.	1.13	1.17	1.15	1.14	1.14	1.15	1.15
Pump - hot	Off	.52	.52	.52	Off	Off	Off
BOILER							
Stand-by		23	23	23			
Load		70	60	48			
Fuel-gas	Off	119	106	91	Off	Off	Off
CHILLER							
Load	142	105	111	131	265	251	187
Flow - cold	54	54	54	54	53	56	53
Flow - cond	62	62	62	62	62	62	62
Temp - cold in	50.6	50.0	50.5	50.8	52.9	56.7	51.6
Temp - cold out	45.3	46.4	46.4	45.9	45.2	47.8	44.6
Temp cond - in	77.6	75.4	76.1	76.4	84.7	87.4	85.2
Temp cond - out	83.5	79.4	80.9	82.1	93.0	98.1	92.9
CONTROLS							
Zones							
1 set point/thr. range	74.1/3.5	74.4/2.8	74.4/2.8	74.7/2.8	76.4/2.8	76.4/2.8	76.4/2.8
2 set point/thr. range	70.1/3.7	70.9/3.4	70.9/3.4	70.9/3.4	71.3/3.4	71.3/3.4	71.3/3.4
3 set point/thr. range	79.2/3.7	74.8/3.9	74.8/3.9	74.0/3.9	74.3/3.9	74.3/3.9	74.3/3.9
4 set point/thr. range	73.7/3	72.3/3.2	72.3/3.2	72.6/3.2	72.4/3.2	72.4/3.2	72.4/3.2
Hot deck set pt/thr.	Off	87/3	90/3	--	--	--	--
Cold deck set pt/thr.	55/6	55/6	55/6	54/6	54/6	54/6	54/6
Hot water set pt/thr.	Off	140/10	140/10	140/10	Off	Off	Off
Tower set pt (thr = 10°F [-12°C])	80	80	80	80	80	80	80
Fuel cooling							
Deck reset							

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.

Table 26 (Cont'd)

	Test 8	Test 9	Test 10	Test 11	Test 12	Test 13	Test 14
ZONES							
1 zone temp	76.1	75.0	73.4	75.2	70.8	77.3	76.4
1 supply temp	65.6	75.7	79.5	75.3	60.7	64.3	65.5
1 air flow	2085	1870	1909	1907	1797	1785	2013
1 load cooling	23.8				33.9	25.4	23.9
1 load heating		1.4	12.5	0			
2 zone temp	73.0	71.5	70.9	72.2	78.7*	76.3*	75.2*
2 supply temp	58.1	70.9	72.9	70.4	59.0	61.6	59.6
2 air flow	1701	1746	1810	1745	1648	1593	1678
2 load cooling	27.7	1.3		3.4	35.5	25.7	28.4
2 load heating			4.1				
3 zone temp	79.9	74.2	75.7	75.7	76.6	75.5	75.4
3 supply temp	67.2	72.9	64.5	64.4	60.1	63.2	64.2
3 air flow	1635	1706	1721	1755	1719	1686	1773
3 load cooling	22.7	2.4	20.9	21.6	31.1	22.7	21.7
3 load heating							
4 zone temp	73.4	72.6	74.4	74.6	76.3*	74.7	74.0
4 supply temp	10.4	71.2	63.3	63.0	59.0	61.1	60.5
4 air flow	1739	1703	1710	1711	1540	1499	1626
4 load cooling	24.7	2.5	20.8	21.7	29.2	22.4	24
4 load heating							
% ODA	0	25%	25%	20%	4%	40%	5%
Δ T across fan	2.5	2.4	2.4	2.5	2.5	2.4	2.5
Δ P across fan	2.0	2.0	2.1	2.1	2.1	2.2	2.0
% r.h. before c.c.	42	46	45	43	45	53	48
% r.h. after c.c.	99	100	100	100	100	100	100
Flow cold deck	5180	2916	3639	4061	6242	5603	5464
Temp cold deck	55.8	53.0	53.3	53.3	56.1	57.8	55.9
Flow hot deck	1980	4106	3511	3058	461	960	1625
Temp hot deck	78.1	88.1	88.2	88.4	79.6	82.3	78.7
Temp hot water in	Off	135.6	133.8	134	Off	Off	Off
Temp hot water out	Off	104.9	101.9	98.1	Off	Off	Off
Temp cold water in	45.5	46.8	46.6	46.3	45.7	48.2	45.1
Temp cold water out	58.4	53.6	54.6	54.9	58.3	58.1	57.4
Load (water) heat. coil	0	60	50	42	Off	Off	Off
Load (water) cool. coil	135	85	101	114	186	232	163
Return air temp	75.6	73.4	74.0	74.7	77.0	76.0	75.4
Mixed air temp	75.6	72.8	73.6	74.0	77.3	80.4	76.0
Temp ODA d.b.	80.4	71.7	71.4	70.9	84.2	87.3	86.4
r.h. ODA	36	47	48	47	84	87	86

* Units: Loads -- thermal (MBH), Loads -- electrical (MBH), Temperatures (°F), Flows -- water (gpm). Metric conversions: 1 m^3/s = 2119 cfm; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.
**Out of zone controller range.

Table 27

Manufacturer's Data on Packaged Roof-Mounted Unit

Power demand	22.6 kW
Air flow	6400 cfm (3 m^3/s)
External static pressure	2.3 in. wc (1573 Pa)
Fan rpm	1160
Motor hp	7 1/2 (5.6 kW)
Fan shaft power	4.6 hp (3.4 kW)
Total cooling	237,000 Btu/hr (69.5 kW)
Sensible cooling	164,600 Btu/hr (48.2 kW)
Condenser fans	two 1-1/2 hp units (2[1.1 kW])
Capacity control	100%/50%/0
Compressor	4 cylinder
Compressor motor	25 hp (19 kW)
Hot-gas bypass option	Yes
Entering air dry bulb temperature	80°F (26.7°F)
Entering air wet bulb temperature	67°F (19.4°C)
Leaving air dry bulb temperature	56.5°F (13.6°C)
Leaving air wet bulb temperature	55.3^{3}°F (12.9°C)
Outdoor air dry bulb temperature	95°F (35°C)

Table 28

Full-Load Performance (Wet Coil)

Air flow (cfm)	6310	5390	4250
Refrigeration effect (1000 Btu/hr)	249	244	230
dry bulb temperature air before coil (°F)*	86.9	86.4	84.9
wet bulb temperature air before coil (°F)	71.7	71.7	71.7
dry bulb temperature air after coil (°F)	61.3	59.2	56.5
wet bulb temperature air after coil (°F)	60.5	58.5	55.8
Power-compressor and cond. fans (kW)	28.6	29.0	30.0

* Same as outdoor air temperature. Metric conversions: 2119 cfm = 1 m^3/s; (°F-32)/1.8 = °C; 1000 Btu/hr/3.412 = kW.

Table 29

Effect of Hot-Gas Bypass on Full-Load Performance (Dry Coil)

	With Hot-Gas ByPass	Without Hot-Gas Bypass
Air flow (cfm)	6380	6320
Cooling (1000 Btu/hr)*	139	181
Temperature -- air entering coil (^{o}F)	80.8	83.6
Temperature -- air heating unit (^{o}F)	60.5	57.5
Electrical power (kW)**	34.5	34.5
Fan static (in. wc)	2.5	2.5

* Cooling equals refrigeration effect less fan motor power
** Includes compressor, condenser fans, and main fan. Metric conversions: 2119 cfm = 1 m^3/s; 1000 Btu/hr/3.412 = kW; (^{o}F-32)/1.8 = ^{o}C; 249 in. wc = Pa.

Table 30

Part-Load Performance* (Dry Coil)

Air flow	6320	5850	4590	3575	2110
Cooling (1000 Btu/hr)**	181	163	108	83	46
Temperature--mixed air (^{o}F)	83.6	80.9	78.8	75.6	71.0
Temperature--supply air (^{o}F)	57.5	55.3	57.2	54.2	50.4
Temperature--outdoor air (^{o}F)	62.9	63.6	62.2	64.4	64.2
Fan static (in. wc)	2.5	2.5	2.6	2.6	2.6
Power (kW)$^{+}$	34.5	34.4	14.7	14.0	14.8

* The unit was tested as follows: four zones at high loads, three zones at high loads, one zone at high load, and all zones loads light. Metric conversions: 2119 cfm = 1 m^3/s; 1000 Btu/hr/3.412 = kW; (^{o}F-32)/1.8 = ^{o}C; 249 (in. wc) = Pa.
** Cooling equals refrigeration effect less fan motor power.
$^{+}$ Power includes fan power, condensor fans, and compressor.

5 SET-UP AND OPERATING EXPERIENCES

During the several years the HVAC test facility was being constructed and operated, many unanticipated problems surfaced. Most of these involved difficulties setting up conventional equipment to operate properly. Engineers who compile and use large BEP programs should be interested in these problems because, typically, computer models of HVAC systems assume that equipment and controls operate to closer tolerances than may be found in actual practice.

The test facility was an assembly of thoroughly instrumented conventional equipment. The instrumentation enabled the actual system performance to be studied, which is how the many small problems were detected. Without instrumentation, the system would most likely have accomplished some heating and cooling; however, in most configurations, it would most certainly have been an energy glutton and would have been unable to deliver the maximum capacities the designers specified.

Pneumatic Controls

Pneumatic controls displayed the poorest performance of all the components in the test HVAC system. The following problems or mistakes occurred during CERL's testing:

1. One mistake on the master plan was noted by CERL engineers. It was corrected before the work got underway.

2. One substantial piping error was uncovered and corrected by CERL personnel.

3. One pneumatic motor controlling the air-handler mixing box was rigged backwards. After the control contractor was informed, the problem was corrected.

4. The four thermostats did not meet the contract specification which stated the controllers should have adjustable gain and remote set-point capability. The remote set-point feature was relaxed in favor of a conventional zone thermostat with adjustable gain as a suitable replacement.

5. All six receiver/controllers drifted excessively; that is, daily calibration was necessary for satisfactory control. The drifting was substantiated and recorded repeatedly. A representative of the control manufacturer inspected the job site and, in particular, the receiver/controller hardware, and found no evidence of faulty installation or operation (such as oil in the air supply). The supply pressure was monitored and found to be well within its tolerable limits. The drifting problem was never resolved. One of the receiver/controllers did not function after it was installed and was replaced by the control contractor.

6. The temperature transmitters which output a pressure corresponding to a temperature were not calibrated accurately. This, however, would only

result in a problem if the HVAC system were set up with precise throttling ranges and set points (as many designs call for and as many BEP program users assume, but which is rarely the installed practice).

7. Three low-pressure transmitters were bench tested. The first two were inoperative and were replaced. The third did not perform up to catalog claims; however, it was functional and was used in a control loop.

8. Reverse-acting accumulators were used to control multiple VAV boxes from a signal thermostat (dual duct VAV). The thermostat signal was to be transformed as shown in Figure 36. These devices drifted, which seriously undermined the zone control strategy by causing an excessive blend of warm and chilled air. (This drifting was small compared to the receiver/controller drift.)

Fan

The original fan equipped with inlet guide vanes tested less than 20 percent efficient with the guide vanes wide open. This fan was a backward-incline type centrifugal fan, the type least suited for VAV service because of the poor power savings when the air-delivery rate is dampered. The principal reasons for this poor efficiency were:

1. The design of the inlet guide vanes resulted in an appreciable reduction in the fan inlet area when they were at the wide-open position.

2. The manner in which one of the two fan bearings was positioned and supported at the fan inlet again appreciably reduced the fan inlet area. Since the original fan did not meet the 50 percent efficiency stated in the contract, CERL requested a replacement. The manufacturer initially proposed substituting the motor with a larger one to boost the fan capacity, but, after some persuasion, the manufacturer agreed to replace the fan. The substitute fan was a forward-curved blade fan with a significantly improved mechanism for inlet guide vane control (a mechanism which positioned the vanes away from the fan inlet at the wide-open position). Under testing conditions similar to those under which the original fan was evaluated, the new fan measured 44 percent efficient at its best operating point (1300 rpm). The only problem experienced with this fan was that half the inlet guide vane linkage became uncoupled from the control shaft, causing the inlet vanes to remain closed on one side of the fan. As expected, the fan capacity dramatically decreased. This problem could occur in the field and continue for the life of the air handler without being detected. The test facility's instrumentation, however, readily diagnosed the problem.

Automatic Control of Inlet Guide Vanes

For inlet guide vanes to reduce fan power, the vanes must be controlled according to a system variable, such as (1) static pressure across the fan, (2) static pressure at some point in the duct work system, or (3) the highest or lowest zone thermostat signal corresponding to at least one zone demanding maximum air flow.

Generally, static pressure is the control variable governing inlet vane modulation, and optimal placement of the static pressure sensor is downstream of the fan at the point where, if a minimum pressure is maintained, no zone will be starved for air.

Pneumatic proportional control was the first type of control tested. This control was set up to output a pressure midway between the control pressure range of the inlet vane motor (2 to 7 psi [13 to 89 kPa]) when the input to the controller corresponds to the set-point static pressure. As the static pressure deviated from the set-point value, the controller output a higher or lower pressure to the inlet vane motor, whichever was appropriate. The static pressure control range (the static pressure range which corresponded to the vanes wide open/vanes fully closed) was made as small as possible, limited only by the stability of the control system.

The pneumatic proportional control system had:

1. A differential pressure transmitter where the low side probe was positioned at the ducting immediately behind the fan and a high side probe was placed at the furthest point downstream of the fan that was still common to all four zones.

2. A receiver/controller with adjustable gain and offset,

3. A pneumatic motor coupled to the inlet vane control shaft.

The control differential pressure at the point downstream of the fan and the fan inlet was 2 in. (500 Pa) wc. The receiver/controller gain was set as high as possible without the system becoming unstable.

The test results were less than satisfactory. The receiver/controller gain was set as high as possible; however, the static pressure throttling range still remained excessively wide, which undermined the power savings expected from automatic control. It was hoped the static pressure would be controlled ±0.2 in. (50 Pa) wc around the set-point of 2 in. (500 Pa) wc. However, testing showed ±1 in. (250 Pa) wc was as low as the control system could do. Figure 12 shows the resultant performance, which is only slightly better from a power reduction standpoint than operation with no control. Figure 37 shows expected and actual power savings, which should be noted by computer modelers and designers. (The expected performance is noted as the performance of the electronic reset controller.)

Electronic reset control was the second type of control system tested. This control system modulated the inlet guide vanes which attempted to hold in on the static pressure set point. It was a dynamic control system in which the controller had an adjustable gain, set point, and damping ratio.

One manufacturer markets such a controller, but it is only available in an entire packaged HVAC system. CERL designed and assembled an electronic reset controller and interfaced a pneumatic to electric transducer for differential static pressure and an electronic to pneumatic transducer for control of the pneumatic motor coupled to the inlet control shaft for about $500. (See Figure 9 for the control schematic, Figure 38 for the pneumatic to electric transducer performance, and Figure 39 for the electric to pneumatic transducer performance.) An option not used was to employ an electric motor in place of the electric to pneumatic transducer and pneumatic motor for control of the inlet vane control shaft. For this control system, the differential pressure was sampled across the fan and the set point was noted as 2.2 in. (0.055 m) wc.

This control system performed successfully when tested. One adjustment was necessary after 1 month of operation because of an instability in the system, but the adjustment was simple and good control resumed. Previous experience with the differential pressure to electric transducers would indicate these are the weak link in the control system. The electric to pneumatic transducer was not an accurate, repeatable piece of hardware. However, a reset control system is a searching type controller, which is not bothered by the inexactness of the electric to pneumatic transducer.

Chiller

The cold water generator was ready for operation as delivered. It required one electrical line and four plumbing connections. However, the unit, as installed, required a substantial amount of work before its performance matched catalog capacity and efficiency. The manufacturer advertised the unit as bench tested before leaving the factory, but it was later revealed that this unit was manufactured just before the manufacturer's bench testing policy went into effect. As installed, the superheat of the refrigerant was 25°F (13.8°C). As recommended by the manufacturer, the superheat temperature difference should be 5°F (2.7°C), measured in the manner described in the section above on component testing. The superheat is controlled by an adjustable expansion valve. As installed, the unit operated with an efficiency 80 percent of the catalog value. After the expansion valve was adjusted so the superheat was 5°F (2.7°C), the unit efficiency was within the tolerances of the catalog.

One of two solenoids in the compressor head never worked properly. (This solenoid should have disabled one cylinder of the four-cylinder compressor by restraining the movement of the inlet valve.) The HVAC instrumentation identified this problem, and the manufacturer replaced the faulty part under warranty. It is doubtful this problem would have been recognized at a typical field installation.

Problems occurred when the machine was energized and both cold and condenser water loops were at about 70°F (20°C). When the return chilled water temperature was 70°F (20°C) and the inlet condenser water temperature was 70°F (20°C), the machine's capacity was about 25 tons (88 kW) of refrigeration (20 tons nominal at typical conditions of 54°F [12.2°C] return chilled water temperature and 80°F [27°C] condenser water temperature). However, the 20 hp (15 kW) motor could not sustain the load. As a consequence of startup with a 70°F (21.1°C) condenser and chilled water loop temperature, the motor thermal protection controller would deenergize the motor. This problem was corrected by installing a controller which did not allow the chiller to operate at full capacity until the inlet chilled water temperature fell below 60°F (15°C). This controller is not standard equipment or factory installed, but may be needed in many field applications.

Boiler

The boiler was tested at 78 percent efficient for steam generation at 5 psi (34 kPa), or about 225°F (106°C). As typical of most gas-heat diversion devices, the nameplate claimed 80 percent efficiency. It should be noted that the operating condition for nameplate data is steam generation at 212°F (99°C). Thus, the boiler, as installed, operated as efficiently as the nameplate claimed. (Efficiency was measured by summing heat output and standby losses and dividing by the heat content of the raw fuel demanded.) The boiler output capacity was measured at 250,000 Btu/hr (73 kW). The boiler nameplate claimed 360,000 Btu/hr (105 kW). There appeared to be no overlooked adjustment or improvement, as evidenced by the boiler's operating efficiency. Thus, it is quite probable a boiler like this could be installed and operate for its useful life without the fact that its capacity was 70 percent of the nameplate claim ever surfacing.

VAV Boxes/PVR

The VAV boxes were sold complete with a PVR* for air flow control. This device modulated the air box to match a rate of supply air to a particular thermostat signal, assuming the air system had sufficient capacity. The PVR eliminated fluctuating air flows. These occur through dampers in a fixed position and are caused by changes of other dampers on the same fan system (that is, varying flow rates related to changes in the static pressure at the air box).

The following problems occurred:

1. As installed, two of eight PVRs were totally inoperative and were replaced without charge.

2. All eight PVRs were installed on a vertical surface. This device is position sensitive and must be installed horizontally. When the PVR is mounted vertically, it will not restrict the air-delivery rate below 30 percent of the maximum rate (which is only a problem if the device is programmed to throttle air-delivery rates lower than 30 percent). Correct operation resumed when the device was horizontally mounted. However, without the test facility, the problem may have gone undetected for its service life.

3. The input range to the PVR, as stated in the manufactor's literature, from the zone thermostat corresponding to minimum to maximum air-delivery rates was 8 to 13 psi (55 to 89 kPa). However, this pressure range was not exact. Several PVRs did not respond until inputs exceeded 9 psi (62 kPa), while several responded to inputs as low as 7 psi (48 kPa). This is only a problem in an arrangement such as dual duct VAV systems, where one zone thermostat controls two PVRs sequentially, or in a VAV system with reheat, where one zone thermostat controls both a VAV box and a reheat valve. The thermostat signal for the dual duct VAVs was piped directly to one PVR, and inverted and offset (if necessary), and connected to another PVR. If one PVR was sensitive from 7 to 12 psi (48 to 82 kPa) and the other sensitive from 9 to 14 psi (62 to 96 kPa), the chilled air would blend with the heated air, operation

* Pneumatic volume regulator.

not usually intended to occur with dual duct VAV. The test facility at CERL easily determined the precise range and sensitivity of the PVRs. Thus, setting up a dual duct VAV or a VAV with reheat was not a problem (except the pneumatic control equipment drifted). In the field, however, it may be difficult to be sure if sequential controls are operating properly.

Pneumatic to Electric Transducers

Pneumatic to electric (P/E) transducers are essential elements in computer controlled systems and computer-aided data acquisition systems. There are several types of P/E transducers, all of which correlate a force (pressure over an area) to a displacement. The measurement of the displacement varies, the most common measuring technique uses strain gages. The transducers usually are complete with the electronic hardware which convert the strain gage reading to a voltage or current proportional to the input pressure.

The P/E transducers used on the CERL test facility output a voltage directly proportional to the gage or differential pressure input (depending on the type of transducer). Several categories of P/E transducers were used.

1. Very low differential pressure transducers which output 0 to 5 V corresponding to an input differential pressure of 0 to 2 in. (0 to 0.5 kPa) wc (low range) and 0 to 10 in. (0 to 2.5 kPa) wc (high range).

2. Low differential pressure transducers which output 0 to 5 V corresponding to an input differential pressure of 0 to 3 psi (0 to 20 kPa).

3. Medium gage pressure transducers which output 0 to 5 V corresponding to 0 to 30 psi (0 to 206 kPa) input.

4. Medium differential pressure transducers which output 0 to 5 V corresponding to 0 to 50 psi (0 to 344 kPa) input.

Substantial problems and failures occurred in all the above categories. During 1 year, more than half of the P/E transducers had to be repaired or replaced. All transducers required regular calibration (the transducers have span and 0 adjustments). The test facility has manometers piped parallel to each transducer, which made calibration possible. This probably would not be the case in a typical field HVAC system. In conclusion, substantial improvements in reliability and repeatability are necessary before computer control interfaced with moderately priced P/E transducers will be suited for HVAC service.

Electric to Pneumatic Transducers

The electric to pneumatic (E/P) transducers were plagued with problems similar to those of the P/E transducers. Two of eight had to be replaced. These transducers were not precision devices and did not require calibration. Their output was not linearly proportional to the input. However, E/P transducers are well suited to certain control strategies such as reset control.

On the CERL test facility, the reset controller for the inlet guide vanes used an E/P transducer and did not have any problems.

Figure 36 shows the output pressure corresponding to the input voltage. The transducer gain between 7 and 10 V input was much greater than the remaining operating range. Reset control uses a searching scheme which is not bothered by the wide variation in gain. Other control strategies may be troubled by the gain variation.

Flow Measurement

Venturis were used to monitor air and water flow. Differential pressure which develops across the venturi inlet and throat was sensed by both P/E transducers, where (1) the electric signal supplied a data acquisition system, and (2) the manometers allowed visual inspection of the differential pressure.

The following difficulties were encountered:

1. The P/E transducers required frequent calibration and replacement.

2. The original venturis were grossly machined and unacceptable. Some of the shortcomings were: (a) low-pressure taps mispositioned upstream of the venturi throats, (b) high-pressure taps mispositioned on the threaded section of the venturis, (c) throat areas on identical venturis differing enough as to be obvious by visual inspection, and (d) threaded sections milled away on some venturis).

3. The circulating water systems of the test facility were equipped with many automatic air vents and air separators; however, periodically, air found its way into the venturi lines. This caused a differential pressure in the absence of flow. This entrapped air was expelled by bleeding the venturi lines. The problem was that it was not possible to rely on the previous flow measurement if after a flow measurement was taken and the pumps shut down, the manometer recorded a differential pressure. After considerable work bleeding lines wherever possible, the problem was generally resolved.

4. Venturis used on the experiment were measured with a micrometer and calibrated with a bucket and scale test facility. The venturi ends were female pipe threads; thus, it was questionable whether the characteristic inlet diameter was the pipe inlet diameter or the diameter of the female pipe threads (which was the pipe outer diameter). However, this was not a problem because the data from the bench test were used to derive the characteristic multiplier for each venturi. This method accounted for inconsistencies or abnormalities in the venturi and determined the venturi constant due to friction.

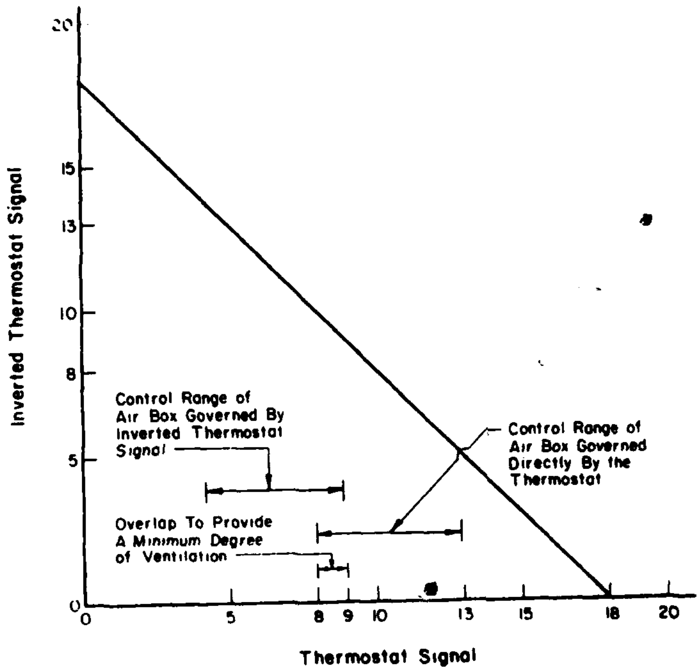

Figure 36. Dual duct control having one thermostat with a direct and inverted control.

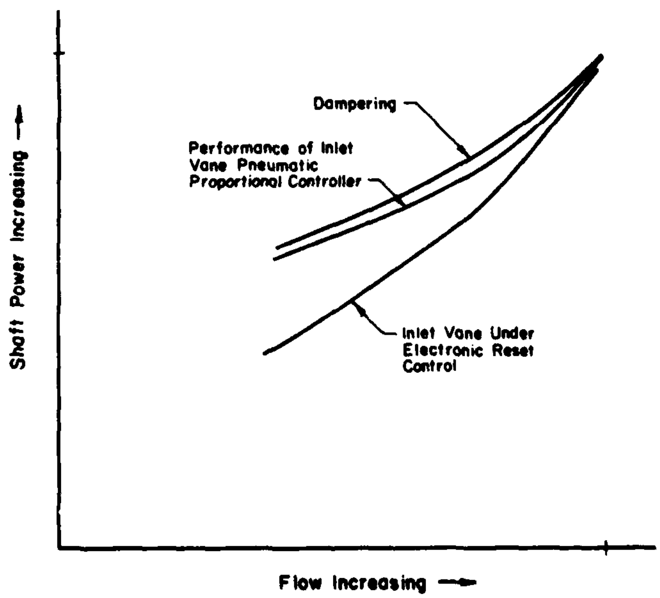

Figure 37. Power requirements at reduced flow rates in a VAV system.

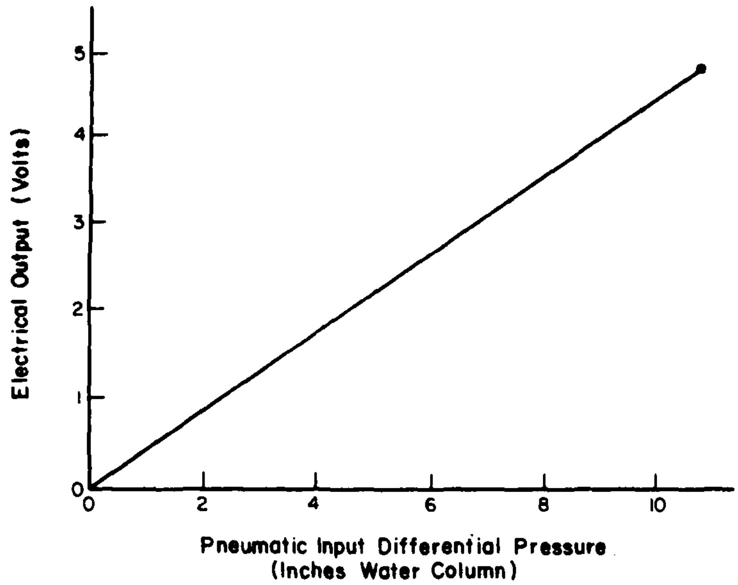

Figure 38. P/E transducer input for fan capacity controller (inlet vanes). (Metric conversion: 1 in. wc = 269 Pa.)

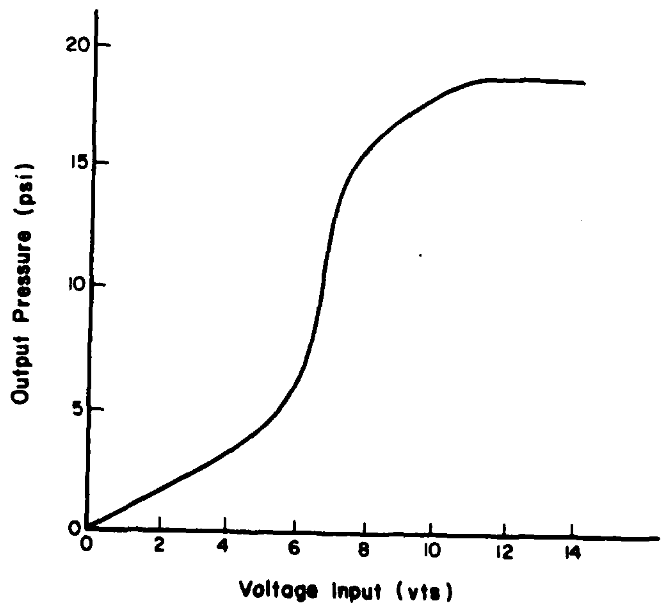

Figure 39. E/P transducer output of fan capacity controller (inlet vane).

6 CONCLUSIONS AND RECOMMENDATIONS

Conclusions

1. This study developed data not previously available on the performance of different HVAC systems operating under a wide range of controlled loads and conditions. This information makes it possible to compare measured system performance with the results of computer simulations used for predicting system performance.

2. In addition to developing validation data, this study identified some problems common to HVAC systems. Two of the more important of these problems are:

a. Unreliable controls performance; e.g., drifting away from calibration and poor repeatability.

2. As-delivered components which do not operate at manufacturers' specifications.

3. Complex control systems of the type tested proved difficult to maintain. While these systems are theoretically capable of improving system efficiency, simple control systems can often produce nearly equal energy efficiency.

4. Careful attention should be given to part-load performance of fans when designing VAV systems. Multiple forward-curved fans were observed to operate well at reduced air flows, even at low flow in the so-called surge region.

Recommendations

1. The HVAC system and component performance data developed in this study should be used with confidence by developers and users of energy analysis methods and programs.

2. Complex control systems should not be used unless regular, skilled maintenance is available. Function indicators should be part of central system control panels.

3. Use of permanently installed indicating air flow meters should be considered in field HVAC applications to ensure that desired mixtures of outdoor and return air are achieved. Alternatively, accurate temperature indicators in the return air, outdoor air, and mixed air streams could be used. These instruments will allow rapid and frequent readjustments which are likely to be required to avoid higher than intended outdoor air flow rates.

4. Because "as-received" HVAC components (chillers, boilers, fan controls) may not perform as specified without field adjustment, performance indicating meters (flow meters, thermometers, power meters) should be used to test and adjust equipment in the field. Provision for temporary or permanent

installation of these meters is required and field performance tests are essential if poor system efficiencies are to be avoided.

5. More reseach and development is needed to improve the reliability and maintainability of HVAC control components and systems.

APPENDIX A:

COMPONENT SPECIFICATIONS*

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The PVR is used on HTB or HCB units to precisely control airflow regardless of changing system static pressure.

An isolated non-bleed diaphragm accepts a separate pneumatic signal from the thermostat. This will reset the velocity (flow) controller section in proportion to the thermostat throttling range between 8 and 13 psi.

The thermostat signal will proportion the output signal between adjustable maximum and minimum setpoints. If a minimum flow is required for reheat or ventilation, the minimum adjustment will control airflow at the preset minimum flow setting. A portion of the thermostat signal throttling range is cut off at the lower end when the minimum flow setting is utilized. The remaining throttling range above minimum flow will continue to reset the controller to maintain airflow as required by the thermostat.

With main air pressure applied to the "M" port, the actuator will position the damper to the cfm required to satisfy the signal applied to the port "T". If the thermostat signal remains constant, indicating a stable load condition, the actuator will maintain the necessary damper position. If system static pressure changes, the velocity sensing section will immediately reposition the damper actuator to maintain a constant flow. If the thermostat signal changes, indicating a change in space loading, the PVR repositions the actuator.

A normally open flow damper will open when main control air to the "M" port is cut. This is a "fail-safe" method to maintain airflow through the terminal when control air is lost and also accomplishes morning warm up simply.

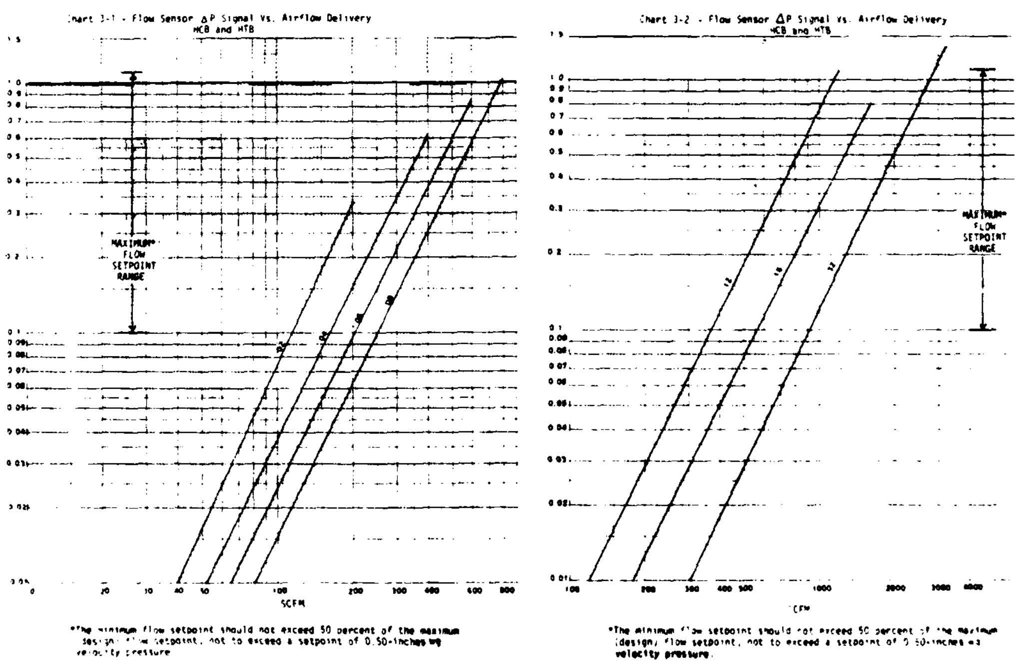
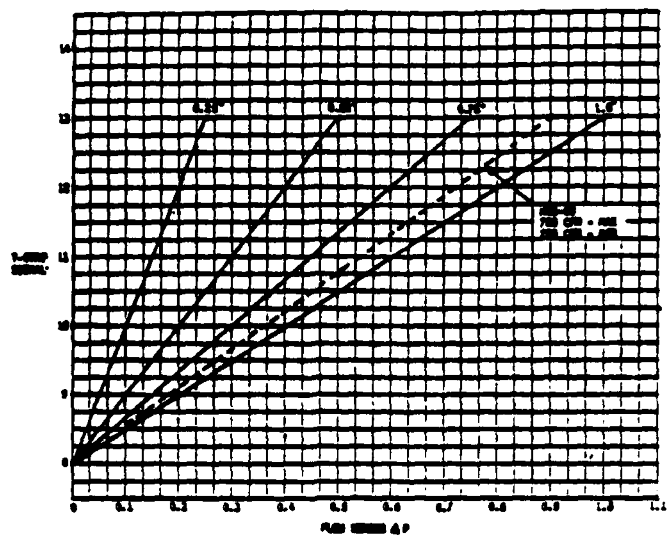


Figure A1. PVR performance (reset velocity controller) used to control the VAV air boxes.

PVR Operation Curves



Flow Sensor ΔP Signal Versus Airflow Delivery (Chart 3-1 and Chart 3-2)

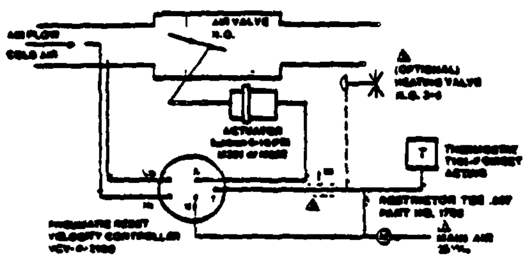
Chart 5-1 shows the relationship of room thermostat branch pressure to flow sensor ΔP signal. This ΔP signal from a flow sensor is shown by unit size in Charts 3-1 and 3-2. The ΔP signal from the flow sensor varies as a squared relationship with airflow delivery through the unit. For example, an HCB-08 unit will have a ΔP signal from the flow sensor of 0.9 inches wg at maximum 750 cfm setting and 0.1 inches wg at minimum 250 cfm setting according to Chart 3-1.

On Chart 5-1 the dashed line is the PVR operation as it modulates over the thermostat's throttling range of 8 to 13 psi. The maximum setting of 0.9 inches wg (750 cfm) is supplied at thermostat signal of approximately 13 psi. As the load in the space decreases, the thermostat signal also decreases causing the PVR to cut air delivery to the space on the dashed line. Eventually, the load decreases to a point where the P signal reaches the minimum flow setting. The dashed line approximates the thermostat signal of approximately 8.6 psi results in a minimum flow requirement of 250 cfm. If the thermostat signal continued to decrease, the PVR would maintain the minimum flow setting.

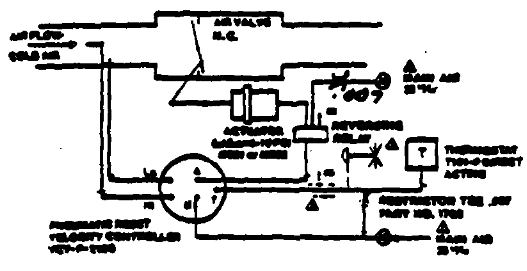
Be careful. The PRV has close to linear operation throughout the 8 to 13 psi throttling range. But the thermostat signal versus ΔP signal is approximate because of manufacturing tolerances. This is important in dual minimum stop applications where a given pneumatic signal is sent to the PVR to provide a minimum cfm air delivery.

Figure A1. (Cont'd).

CONNECTION SCHEMATICS



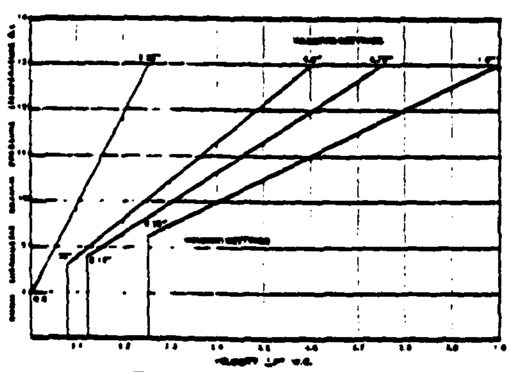
NORMALLY OPEN AIR VALVE



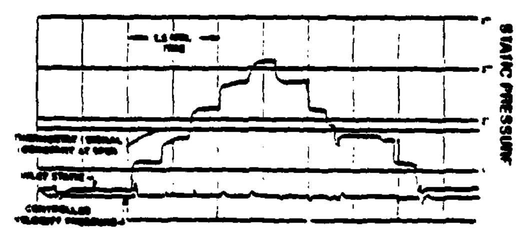
NORMALLY CLOSED AIR VALVE

1. Cutting main air here will open air valve for warm-up.
2. A 3-way pneumatic switch (or solenoid) to block the thermostat signal and exhaust the "T" port will position the air valve to the minimum flow setpoint to maintain ventilation. Connecting the thermostat in a "two pipe" configuration (separate main air supply to the thermostat) would permit cutting this main from a central location accomplishing the same result.
3. If perimeter heat or reheat is required, a valve or pressure electric switch should be located here. The air valve will remain at the minimum position until thermostat branch pressure exceeds 8 psi. At this pressure heating will be off.
4. Cutting main air here will close air valve for economy in unoccupied space.

OPERATION CURVES



TYPICAL OPERATION CURVES



CONSTANT VOLUME CONTROL CHART

Figure A1. (Cont'd).

WEIGHT (LBS.)				FAN MOTOR HORSEPOWER						
APPROX. SHPG.	APPROX. OPER.	HEAVIEST SECTION (PAN)	CFM ESP	0" ESP	1/8" ESP	1/4" ESP	3/8" ESP	1/2" ESP	WATER OUTLET CONNS.	MAKE-UP CONN SIZE
1090	1420	810	9600	3	5	5	5	5	3"	1"

WET BULB	68		70		72		75		78		80	
WATER	90/80	95/85	90/80	95/85	90/80	95/85	95/85	103/85	95/85	103/85	95/85	103/85
	117	167	99	152	80	140	117	72	90	56	61	40

Figure A2. Variable capacity cooling tower (tower capacity controlled via scroll dampers and an aquastat).

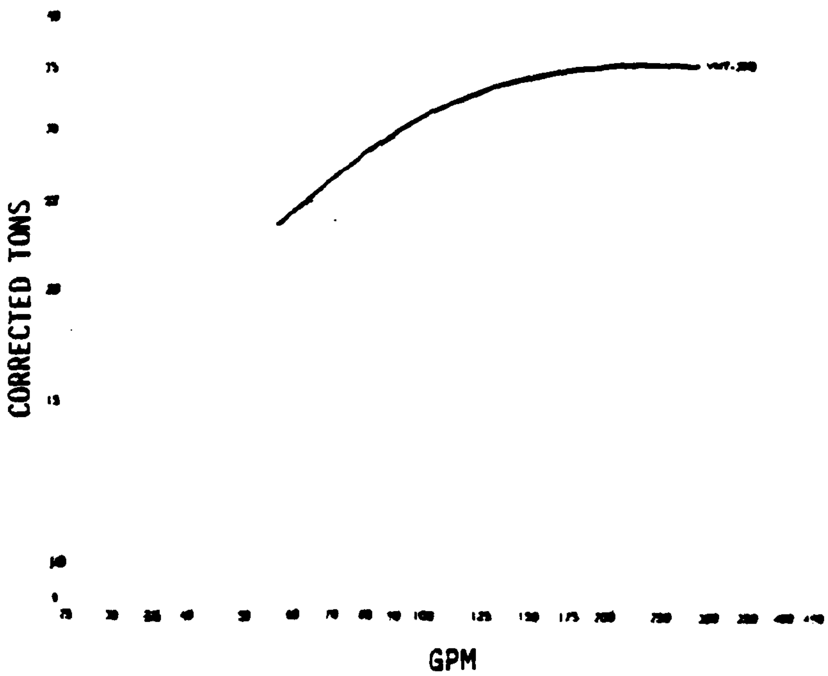


Figure A3. Tower capacity at water flow rates greater than rated conditions.

EDB	EWB	EWT	GALLONS PER MINUTE														
58	65	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	70	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	75	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	80	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	85	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
61	65	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	70	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	75	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	80	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	85	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	90	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
64	70	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	75	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	80	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	85	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	90	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	95	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
67	75	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	80	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	85	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	90	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	95	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
70	80	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	85	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	90	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	95	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
73	80	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	85	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	90	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	95	40	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		45	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

Figure A4. Performance of cooling fan/coils within each zone.

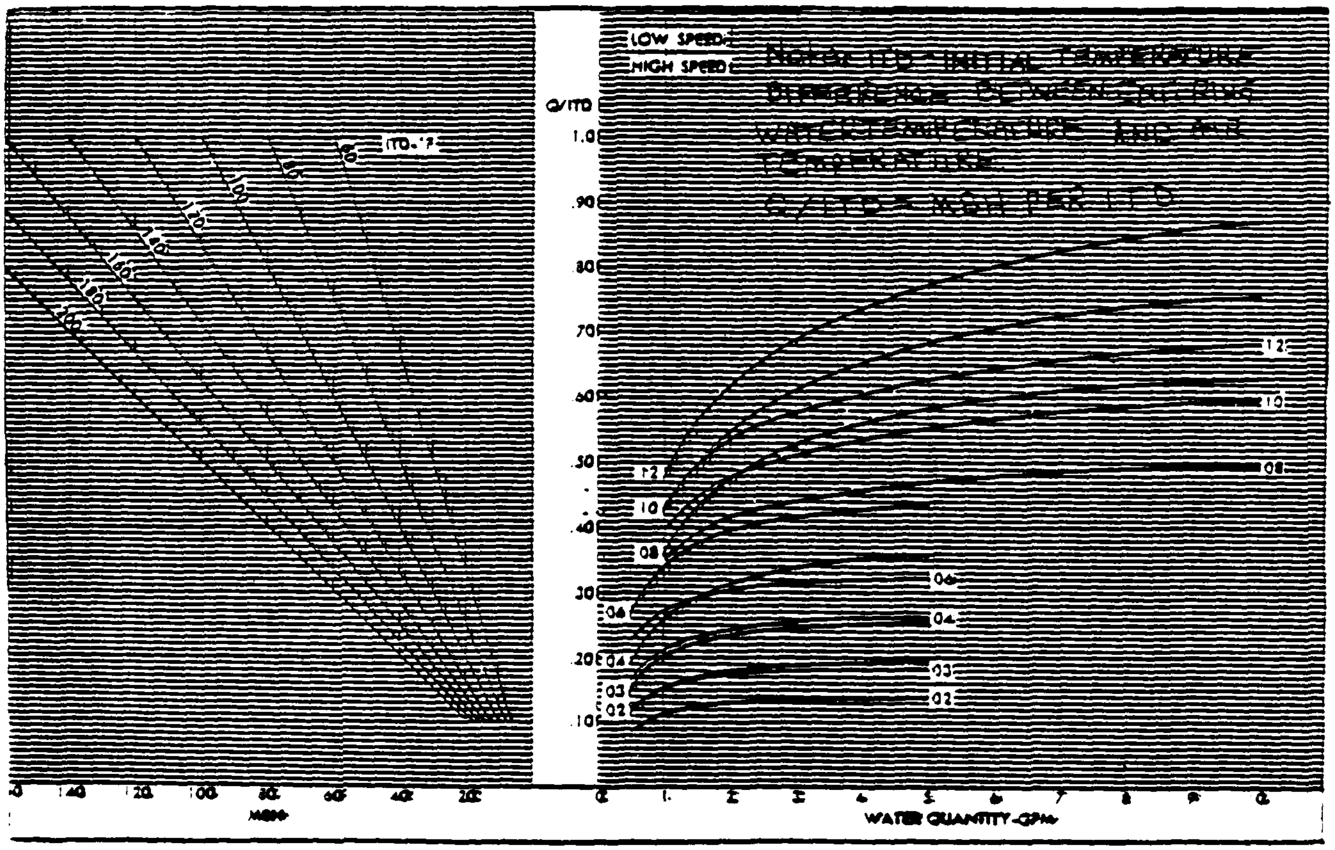


Figure A5. Performance of heating fan/coils within each zone.

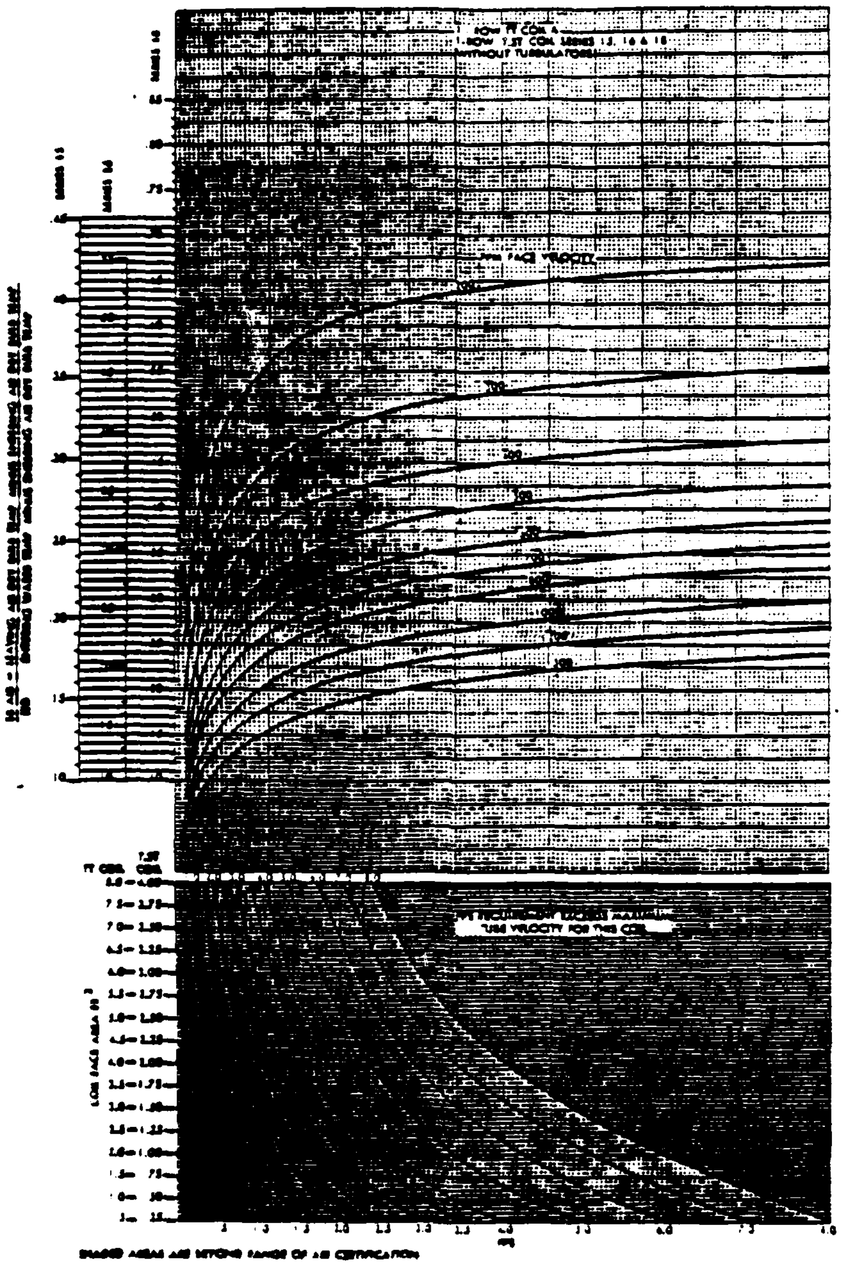


Figure A6. Performance of reheat coils serving each zone.

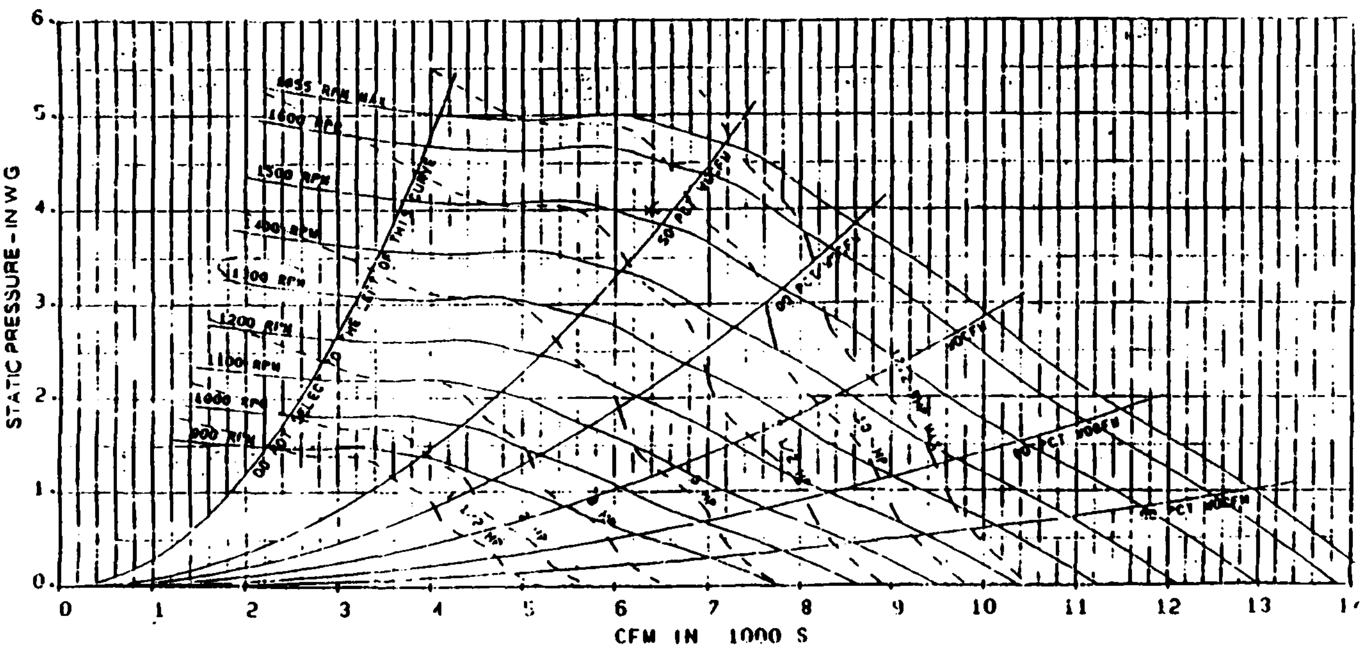


Figure A7. Fan curves for built-up air handler.

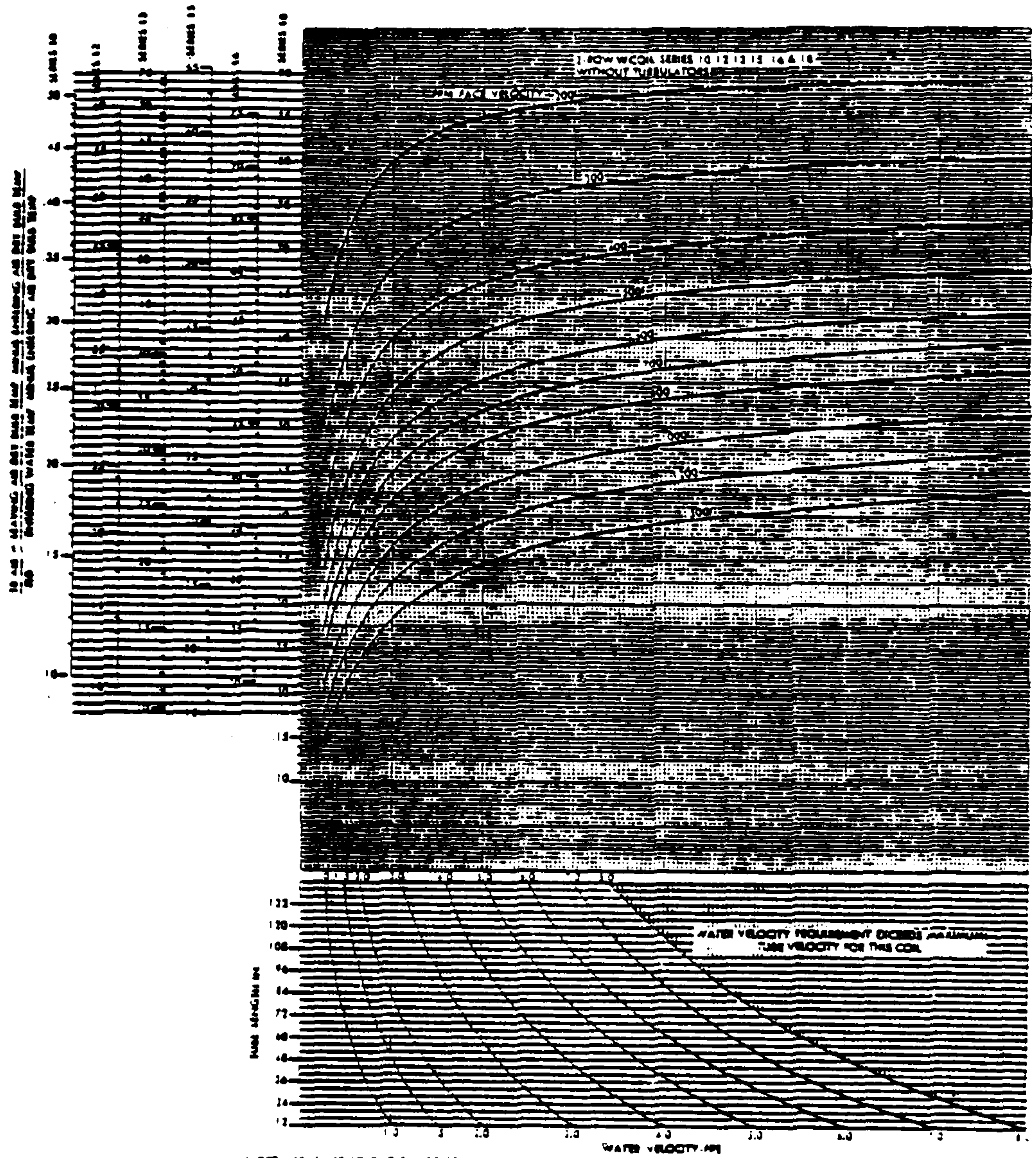


Figure A8. Heating coil, built-up air handler.

CHILLED WATER COOLING COIL CAPACITIES

75
85
EDB
62
EWB

75/62 EDB/EWB — 85/62 EDB/EWB

FPS	ROWS	WTR	EWT
10	1	2, 4, 6, 8, 10	[illegible]
10	2	2, 4, 6, 8, 10	[illegible]
10	4	2, 4, 6, 8, 10	[illegible]
10	8	2, 4, 6, 8, 10	[illegible]
12	1	2, 4, 6, 8, 10	[illegible]
12	2	2, 4, 6, 8, 10	[illegible]
12	4	2, 4, 6, 8, 10	[illegible]
12	8	2, 4, 6, 8, 10	[illegible]

NOTE: (1) MSH = MBH/FT² Coil Face Area. WTR = Water Temperature Rise, degrees F. FPS = Water Velocity, Ft. Second. LDB = Leaving Dry Bulb degrees F. LWB = Leaving Wet Bulb degrees F. EWT = Entering Water Temperature, degrees F.
(2) When using turbulators make selection based on double the actual water velocity.

Figure A9. Cooling coil, built-up air handler (1).

CHILLED WATER COOLING COIL CAPACITIES

Face velocity	Rows	FPS	75/62 EDB/EWB	85/62 EDB/EWB
16	1	2	[illegible]	[illegible]
		4	[illegible]	[illegible]
		6	[illegible]	[illegible]
		8	[illegible]	[illegible]
		10	[illegible]	[illegible]
	2	2	[illegible]	[illegible]
		4	[illegible]	[illegible]
		6	[illegible]	[illegible]
		8	[illegible]	[illegible]
		10	[illegible]	[illegible]
	4	2	[illegible]	[illegible]
		4	[illegible]	[illegible]
		6	[illegible]	[illegible]
		8	[illegible]	[illegible]
		10	[illegible]	[illegible]
	8	2	[illegible]	[illegible]
		4	[illegible]	[illegible]
		6	[illegible]	[illegible]
		8	[illegible]	[illegible]
		10	[illegible]	[illegible]
20	1	2	[illegible]	[illegible]
		4	[illegible]	[illegible]
		6	[illegible]	[illegible]
		8	[illegible]	[illegible]
		10	[illegible]	[illegible]
	2	2	[illegible]	[illegible]
		4	[illegible]	[illegible]
		6	[illegible]	[illegible]
		8	[illegible]	[illegible]
		10	[illegible]	[illegible]
	4	2	[illegible]	[illegible]
		4	[illegible]	[illegible]
		6	[illegible]	[illegible]
		8	[illegible]	[illegible]
		10	[illegible]	[illegible]
	8	2	[illegible]	[illegible]
		4	[illegible]	[illegible]
		6	[illegible]	[illegible]
		8	[illegible]	[illegible]
		10	[illegible]	[illegible]

75 / 85 EDB — 62 EWB

NOTE: MBH = MBH/FT² Coil Face Area. WTR = Water Temperature Rise, degrees F. FPS = Water Velocity Ft. Second. LDB = Leaving Dry Bulb degrees F. LWB = Leaving Wet Bulb degrees F. EWT = Entering Water Temperature degrees F.
When using turbulators, make selection based on double the actual water velocity.

Figure A10. Cooling coil, built-up air handler (2).

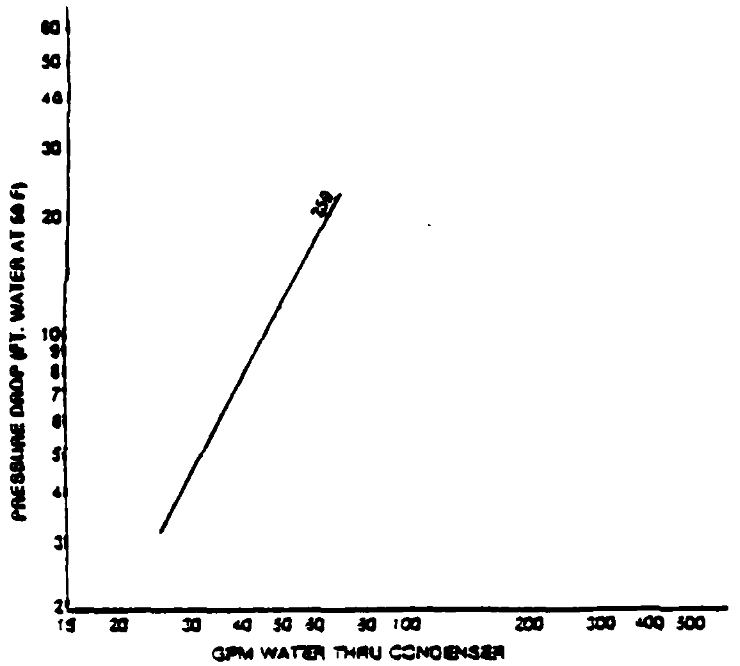
	[illegible]
COMPRESSOR, NUMBER/ MODEL NO.	1/CRHM-200
EVAP MODEL	EVP 283
STORAGE CAPACITY (GAL)	6.5
CONDENSER MODEL	CDS 250
STORAGE CAPACITY (GAL)	2.5
STD. REFRIGERANT CHARGE (LB) (COMPRESSOR & SYSTEM)	32
STD. UNIT OIL CHARGE (PTS) (COMPRESSOR & SYSTEM)	11.5
OPERATING WEIGHT (LBS) (REFRIGERANT & WATER)	1208
SHIPPING WEIGHT (LBS) (SKID & REFRIGERANT)	1325
UNIT CAPACITY STEPS	
2 STEP	100-50
3 STEP	100-50 25
4 STEP	N/A
REFRIGERANT	22

	[illegible]
REFRIGERANT	22
STD. REFRIGERANT CHARGE (LB)	15
OPERATING WT (LBS) (REF & WATER)	[illegible]
SHIPPING WT (LBS) (SKID & HOLDING CHARGE)	1110

Figure A11. Chiller specifications.

MODEL NO.	ENTERING CONDENSER WATER TEMPERATURE (°F)																			
[illegible] R-22	75				80				85				90				95			
	TONS	EVAP GPM	KW	COND GPM	TONS	EVAP GPM	KW	COND GPM	TONS	EVAP GPM	KW	COND GPM	TONS	EVAP GPM	KW	COND GPM	TONS	EVAP GPM	KW	COND GPM
40	20.0	48.0	19.4	60.6	19.4	46.5	20.0	59.5	18.8	45.[illegible]	20.7	58.5	18.2	43.6	21.4	57.6	17.6	42.2	22.2	56.[illegible]
42	20.[illegible]	49.[illegible]	19.[illegible]	62.2	20.1	48.3	20.3	61.3	19.6	47.0	21.0	60.4	18.9	45.4	21.7	59.4	18.3	43.9	22.5	56.5
44	21.5	51.6	19.9	64.1	20.9	50.1	20.5	63.1	20.3	48.[illegible]	21.1	62.2	19.7	47.2	21.9	61.3	19.1	45.8	22.8	60.[illegible]
45	21.8	52.3	20.0	64.9	21.2	50.8	20.6	63.8	20.6	49.4	21.3	62.7	20.0	48.0	22.1	62.0	19.4	46.6	23.0	6[illegible].4
46	22.1	53.0	20.1	65.7	21.5	51.7	20.7	64.7	21.0	50.4	21.4	63.7	20.4	48.9	22.3	63.0	19.8	47.5	23.2	62.4
48	22.6	54.[illegible]	20.3	66.8	22.1	53.0	21.0	66.2	21.6	51.6	21.[illegible]	65.6	21.0	50.5	22.6	64.9	20.5	49.2	23.5	64.[illegible]
50	23.2	55.[illegible]	20.6	68.5	22.6	54.4	21.2	68.1	22.2	53.2	21.9	6[illegible]	21.[illegible]	52.0	22.8	66.9	21.2	50.9	23.8	66.[illegible]

Figure A12. Chiller performance (1).



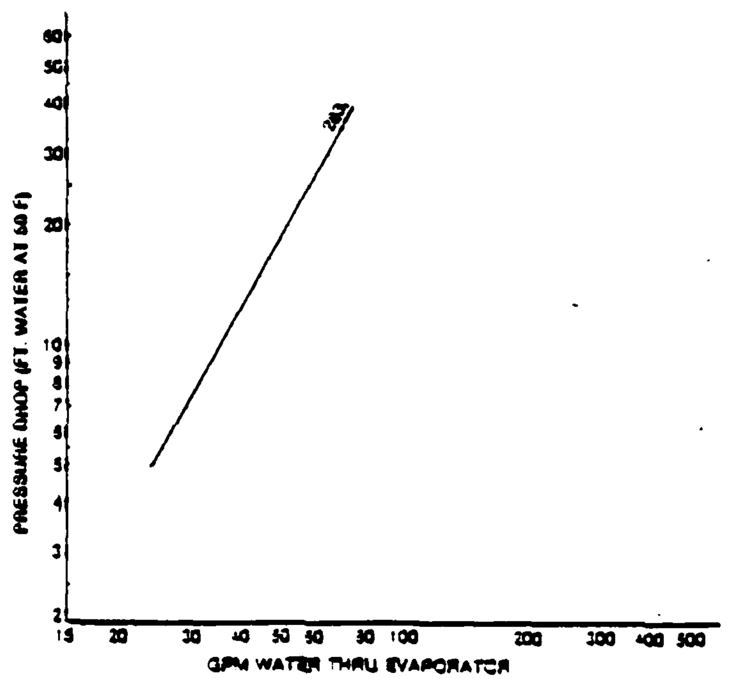


Figure A13. Chiller performance (2).

APPENDIX B:

ALGORITHMS

Overview

This appendix contains many common algorithms found in building energy analysis programs. These algorithms were collected and examined before this experiment was designed to ensure the data collected by the experiment were sufficient for validation. This collection of algorithms is not a complete listing.

Source and Reference materials in this appendix are identified as follows:

BLAST 1.2 -- D. C. Hittle, The Building Loads Analysis and System Thermodynamics (BLAST) Program, Users Manual and Reference Manual, TR E-119/ADA048734 and ADA048982 (CERL, December 1977).

BLAST 2.0 -- D. C. Hittle, The Building Loads Analysis and System Thermodynamics (BLAST) Program, Version 2.0 Users Manual, Volumes I and II, TR E-153/ADA072272 and ADA072273 (CERL, June 1979); and E. Sowell, The Building Loads Analysis and System Thermodynamics (BLAST) Input Booklet, E-154/ADA072435 (CERL, June 1979).

DOE-2 -- DOE-2 Users Guide, Version 2.1 (Department of Energy, May 1980).

ARI 410-72 -- Standard for Forced-Circulation Air-Cooling and Air-Heating Coils, ARI Standard 410-72 (Air Conditioning and Refrigeration Institute, 1978).

Fan Engineering -- Fan Engineering (Buffalo Forge Co., 1970).

ASHRAE -- "Procedures for Simulating the Performance of Components and Systems for Energy Calculations," ASHRAE Handbook of Fundamentals (American Society of Heating, Refrigeration, and Air-Conditioning Engineers [ASHRAE], 1977).

Consultants Computation Bureau -- Consultants Computation Bureau Total Energy Plant Simulation Program (Consultants Computation Bureau, San Francisco, CA).

Contents

COMPONENT: Mixing Box

DESCRIPTION: A mixing air plenum is a device which performs adiabatic mixing of two streams of moist air. There are five different methods of controlling the rate of outdoor air introduced into the air system through this box:

1. Fixed percent
2. Fixed amount
3. Temperature economic cycle
4. Return air economy cycle
5. Enthalpy economy cycle.

INPUT: Hod = outdoor air enthalphy, kJ/kg

Hra = Return air enthalpy, kJ/kg

SA = Supply air volume flow rate, m^3/s

Tdm = Desired mixed air temperature, ^oC

Tod = Outdoor air dry-bulb temperature, ^oC

Tra = return air dry-bulb temperature, ^oC

Wod = Outdoor air humidity ratio, kg/kg

Wsa = Supply air humidity ratio, kg/kg.

OUTPUT: Tmix = Mixed air temperature, ^oC.

Wmix = Mixed air humidity ratio, kg/kg.

Min = Minimum fraction of outdoor air introduced.

CALCULATION METHOD: Tmix = (Tod)(Fod) + (Tra)(1-Fod)

Wmix = (Wod)(Fod) + (Wsa)(1-Fod)

Where: Fod = fraction of outdoor air introduced according to the control strategy, kg/kg.

1. Fixed percent: Fod as output

2. Fixed amount: Fod = (MIN)(SA)

3. Temperature economizer cycle

If a. Tdm <Tod; Fod = MIN
b. Tdm >Tod; Fod = (Tdm-Tra)/(Tod-Tra)

4. Return air economy cycle

If a. Tra>Tod>Tdm; Fod = 1
b. Tdm>Tra>Tod; Fod = MIN
c. Tra>Tdm>Tod; Fod = (Tdm-Tra)/(Tod-Tra)
d. Tra = Tod; Fod = 1
e. Tod>Tra>Tdm; Fod = MIN
f. Tdm>Tod>Tra; Fod = 1
g. Tod>Tdm>Tra; Fod = (Tdm-Tra)/(Tod-Tra)

If a. Hod<Hra; use temperature economy cycle
b. Hod>Hra; Fod = MIN.

SOURCE: BLAST 1.2, ASHRAE

REFERENCE: ASHRAE

COMPONENT: Mixing Box Temperature Controller (Temperature Economizer Cycle)

DESCRIPTION: The air-handler mixing box can be scheduled to (1) introduce a fixed amount of outdoor air into a constant or varying air stream or (2) a fixed percent of outdoor air into a constant or varying air stream. The mixing box can also blend cool outdoor air with return air to maintain a constant temperature.

INPUT: If (1) temperature economizer cycle is specified and (2) outdoor air is cooler than the cold deck set point.

OUTPUT: Cold deck temperature is fixed at the cold deck set point, and load on the cooling coil is set to 0.

SOURCE: None

REFERENCE: None

COMPONENT: Boiler (1)

DESCRIPTION: This algorithm calculates the amount of fuel energy required by a boiler as a function of load, heating value of the fuel, and combustion parameters. The fuel consumption is determined by a second degree curve fit of manufacturers' data relating the fraction of full load on the boiler to the theoretical vs actual fuel consumption ratio.

INPUT: Ta = ambient air temperature, ^{o}C

HR = humidity ratio, kg/kg

Q = load on boiler, fraction of capacity, kW/kW

N = net energy output (load), kW.

OUTPUT: AFC = Actual fuel consumption, kW.

MODEL PARAMETERS (default values in parenthesis);

AF = air to fuel ratio, kg/kg (17.0)

HV = heating values of fuel, kJ/kg (46 520)

TS = stack temperature, ^{o}C (287.8)

Cpex = specific heat of exhaust gas, $\frac{kJ}{kgk}$ (1.005)

Cpw = specific heat of water, $\frac{kJ}{kgK}$ (4.19)

HVW = heat of vaporization of water at boiler conditions, $\frac{kJ}{kg}$ (2326.5)

A1-A3 = coefficients of a second degree curve fit determined from manufacturers' data relating the fraction of full load to the theoretical to actual fuel consumption ratio (0.6, 0, 0).

CALCULATION METHOD:

$$EFF = 0.87 - 1.25 \left(\frac{AF}{HV} [Ts - Ta] ((Cpex + Cpw \cdot HR) + HVW \cdot HR)\right)$$

$$\frac{N}{TFC} = EFF$$

$$\frac{TFC}{AFC} = A1 + A2 \cdot Q + A3 \cdot Q^2$$

SOURCE: BLAST 1.2

REFERENCE: Consultants Computation Bureau Total Energy Plant Simulation Program

COMPONENT: Boiler (2)

DESCRIPTION: This boiler algorithm calculates the fuel energy required by a boiler as a function of the load on the boiler. There are three basic types of boilers: (1) electric heaters, (2) modulating combustion air-supply rate, and (3) natural convection driving force for combustion air. This algorithm calculates the efficiency with which the boiler creates useful energy to satisfy the load from potential energy in the form of fuel. This conversion efficiency is a function of the load.

INPUT: L = percent of full load, dimensionless.

OUTPUT: N = conversion efficiency, percent.

MODEL PARAMETERS (default values in parentheses):

A0-A2 = Coefficients determined by a curve fit of supplied data comparing the percent full load and conversion efficiency (Type 1: 100, 0, 0; type 3: 60.46, - 0.0773, 0.00273; no default values for Type 2).

CALCULATION METHOD: $N = A0 + A1 \cdot L + A2 \cdot L^2$

SOURCE: ASHRAE

REFERENCE: None

COMPONENT: Cold Deck Control

DESCRIPTION: The temperature of the air stream in the cold air duct may be controlled through a specified control strategy. They are:

1. Fixed temperature
2. Inverse function of outdoor air temperature
3. Based on the zone requiring the coldest supply air temperature
4. Set to the zone 1 supply air temperature.

INPUT (by section):

1. Tf = fixed desired air temperature, ^{o}C
2. Toa = outdoor air temperature, ^{o}C
3. T_{1z} = lowest required zone supply air temperature, ^{o}C
4. Tz1 = zone 1 supply air temperature, ^{o}C
5. Q = cooling load, W, last hours.

OUTPUT: Tc = cold deck temperature, ^{o}C.

MODEL PARAMETERS (default values in parentheses):

Hc = capacity of cooling coil, W

Rt = controller throttling range, ^{o}C (4.0)

TXMIN, TYMIN, TXMAX, TYMAX = Two coordinates which define a ramp used in the direct or inverse reset control strategy, ^{o}C ([21.1, 18.3], [32.2, 12.8]).

CALCULATION METHOD:

1. Tc = Tf - Rt + (Rt/Hc)Q
2. Determine TY using Toa, a ramp function, and the inverse acting temperature controller routine, TX = Toa

   Tc = TY - Rt + (Rt/Hc)Q
3. Determine TY using Tlz, a ramp function, and the direct acting temperature controller routine, TX = Tlz

   Tc = TY - Rt + (Rt/Hc)Q
4. Tc = Tllz + Tz1.

SOURCE: BLAST 2.0

REFERENCE: None

COMPONENT: Cooling Tower (one unit)

DESCRIPTION: This cooling tower algorithm calculates the leaving water temperature from a cooling tower based on a second degree three-variable curve fit of leaving water temperature, entering water temperature, and the ambient air wet-bulb temperature. This model assumes one tower with a constant air and water flow.

INPUT: Tin = entering water temperature, ^oC

Twb = ambient air wet-bulb temperature, ^oC.

OUTPUT: Tout = leaving water temperature, ^oC.

MODEL PARAMETERS (default values in parentheses):

C1 - C9 = coefficients of a curve fit of physical data relating leaving water temperature, entering water temperature, and the ambient air wet-bulb temperature (86.2969, -5.8468, 0.1210, -2.1410, 0.1975, -0.003802, 0.009037, -0.000930, 0.000019).

CALCULATION METHOD: $Tout = C1 + C2 \cdot Twb + C3 \cdot Tw^2B + C4 \cdot Tin +$

$C5 \cdot Tin \cdot Twb + C6 \cdot Tw^2b \cdot Tin +$

$C7 \cdot Tin^2 + C8 \cdot Twb \cdot Tin + C9 \cdot Tw^2b$

$\cdot Tin^2$

SOURCE: ASHRAE

REFERENCE: None

COMPONENT: Hot Deck Controller

DESCRIPTION: This algorithm calculates the temperature of the air stream in the hot air supply duct based on the heating load and one of four control strategies:

1. Fixed temperature

2. Inverse function of outdoor air temperature

3. Based on the zone requiring the warmest supply air temperature

4. Set to the zone 1 desired supply air temperature.

INPUT: Q = heating load from the last hour, kW.

Ty = temperature determined using the inverse acting controller algorithm by resetting the deck temperature according to the outdoor air dry-bulb temperature, ^{o}C

Tyy = temperature determined using the direct acting controller algorithm by resetting the deck temperature according to the zone requiring the warmest supply air temperature, ^{o}C

Tz1 = desired suppply air temperature for zone 1, ^{o}C.

OUTPUT: Te = hot deck temperature, ^{o}C.

MODEL PARAMETERS (default values in parentheses):

Tf = specified hot deck temperatures, ^{o}C (40)
TH = throttling range of controller, ^{o}C (4)
HC = capacity of heating coil, kW (1000).

CALCULATION METHOD:

1. Fixed temperature

$$Te = Tf + TH - TH/HC \cdot Q$$

2. Inverse function of outdoor air temperature

$$Te = Ty + TH - TH/HC \cdot Q$$

3. Based on the zone requiring the warmest supply air temperature

$$Te = Tyy + TH - TH/HC \cdot C$$

4. Set to the zone 1 desired supply air temperature

$$Te = Tz1.$$

SOURCE: BLAST 2.0

REFERENCE: None

COMPONENT: Reciprocating Compression Chiller (1)

DESCRIPTION: This algorithm calculates the capacity of and the energy required by a reciprocating compressor for a given evaporating and condensing temperature through a second degree curve fit of empirical data using a least squares regression method.

INPUT: Tc = condensing temperature, ^{o}C

Te = evaporating temperature, ^{o}C.

OUTPUT: EELEC = electrical energy required to meet cooling load, kW

CAP = refrigerating capacity, kW.

MODEL PARAMETERS (default values in parentheses): CP1-CP9 = coefficients relating the electrical energy required by the compressor and the evaporating and condensing temperatures determined by a least squares fit of empirical data (11.0914, -0.2078, -0.006379, 0.4384, 0.006283, -0.00129, -0.001011, 0.000149, 0.000001)

CQ1-CQ9 = coefficients relating the refrigerating capacity of the compressor to the evaporating and condensing temperatures determined by a least squares fit of empirical data (191.6106, 7.4325, -0.002281, -1.6316, 0.07956, 0.002843, -0.001995, 0.000263, -0.000018).

CALCULATION METHOD:

$$P = CP1 + (CP2)\,Te + (CP3)\,Te^2 + (CP4)\,Tc + (CP5)\,TeTc + (CP6)\,Te^2Tc + (CP7)\,Tc^2 + (CP8)\,TeTc^2 + (CP9)\,Te^2Tc^2$$

$$Q = CQ1 + (CQ2)\,Te + (CQ3)\,Te^2 + (CQ4)\,Tc + (CQ5)\,TeTc + (CQ6)\,Te^2Tc + (CQ7)\,Tc^2 + (CQ8)\,TeTc^2 + (CQ9)\,Te^2Tc^2$$

SOURCE: ASHRAE

REFERENCE: None

COMPONENT: Reciprocating Compression Chiller (2)

DESCRIPTION: This chiller algorithm calculates the electrical energy required to meet a cooling load as a function of the leaving chilled water and condenser water temperatures. The algorithm uses curve fits to three sets of manufacturers data: (1) graphs relating available to nominal capacity ratio and the equivalent temperature difference, (2) graphs relating the full-load power consumption per unit output to the available to nominal capacity ratio, and (3) graphs relating the fraction to full-load power to the part-load ratio. The equivalent temperature difference is found from the actual chilled water temperature and condenser water temperature and manufacturers' data to determine the temperature difference deviation from the value at the rated capacity.

INPUT:

Q = load on chiller, kW

Tchw = temperature of cold water leaving the chiller, ^{o}C

Tcw = temperature of the water leaving the condenser, ^{o}C.

OUTPUT:

EELEC = electrical energy required to meet the cooling load, kW

ETOWER = load to the cooling tower, kW.

MODEL PARAMETERS (default values in parentheses):

Tcwr = temperature of the water leaving the condenser at the rated capacity as determined from manufacturers' data, ^{o}C (35)

TRATIO = ratio of the change in leaving condenser water temperature and the change in leaving chilled water temperature required to maintain a constant rated capacity, dimensionless (2.58)

Tchwr = chilled water supply temperature at the rated capacity as determined from manufacturers' data, ^{o}C (6.66)

B1-B3 = coefficients of a second degree curve fit relating the available to nominal capacity ratio and the equivalent temperature difference determined through a regression analysis of manufacturers' data (1.0, -0.0594, -0.0184)

NCAP = nominal cooling capacity of chiller, kW

C1-C3 = coefficients for a second degree curve fit relating the available to nominal capacity ratio to the actual full-load power consumption per unit of actual capacity; the values are determined through a regression analysis of manufacturers' data (1.61, -0.61, 0)

D1-D3 = coefficients of a second degree curve fit relating the fraction of full-load power to the part-load ratio; the values are determined by a regression analysis of manufacturers' data (0.1494, 0.9568, -0.11184)

NFLPR = nominal full-load power ratios, ratio of electrical energy required at nominal capacity, to nominal capacity, kW/kW (0.2275)

RMAX = maximum part-load ratio, fraction (1.05)

RMIN = minimum part-load ratio, fraction (0.1).

CALCULATION METHOD:

1. Calculate equivalent temperature difference

$$TEQ = ((Tcw - Tcwr)/TRATIO) - (Tchw - Tchwr)$$

2. Calculate ratio of available capacity to nominal capacity

$$ANCR = B1 + B2\ (TEQ) + B3\ (TEQ)^2$$

3. Calculate available capacity and part-load ratio

$$ACAP = ANCR\ (NCAP)$$
$$X = Q/ACAP$$

$$\text{if } RMIN < RMAX,\ PLR = X$$

$$\text{if } X < RMIN,\ PLR = RMIN$$
$$\text{if } X > RMAX,\ PLR = RMAX$$

4. Calculate full-load power consumption

$$FLPR = [C1 + C2\ (ANCR) + C3(ANCR)^2](NFLPR)$$

$$FLP = FLPR\ (ACAP)$$

5. Calculate actual power consumption

FFL = D1 + D2 (PLR) + D3 (PLR)

ELEC = FFL (FLP).

SOURCE: BLAST 2.0

REFERENCE: Consultants Computation Bureau Total Energy Plant Simulation

COMPONENT: Reciprocating Compression Chiller (3)

DESCRIPTION: This algorithm calculates the capacity of and the energy required by a reciprocating compressor for a given evaporating and condensing temperature through a second degree curve fit of empirical data using a least squares regression method.

INPUT: Tc = condensing temperature, ^{o}C

Te = evaporating temperature, ^{o}C.

OUTPUT: P = electrical energy required, kW

Q = refrigerating capacity, kW.

MODEL PARAMETERS (default values in parentheses): CP1-CP9 = coefficients relating the electrical energy required by the compressor and the evaporating and condensing temperatures determined by a least squares fit of empirical data (11.0914, -0.2078, -0.006379, 0.4384, 0.006283, -0.00129, -0.001011, 0.000149, 0.000001)

CQ1-CQ9 = coefficients relating the refrigerating capacity of the compressor and the evaporating and condensing temperatures determined by a least squares fit of empirical data (191.6106, 7.4325, -0.002281, -1.6316, -0.07956, 0.002843, -0.001995, 0.000263, -0.000018).

CALCULATION METHOD:

$$P = CP1 + CP2 . Te + CP3 . Te^2 + CP4 . Tc + CP5 . TeTc + CP6 . Te^2Tc + CP7 . Tc^2 + CP8 . TeTc^2 + CP9 . Te^2Tc^2$$

$$Q = CQ1 + CQ2 . Te + CQ3 . Te^2 + CQ4 . Tc + CQ5 . TeTc + CQ6 . Te^2Tc + CQ7 . Tc^2 + CQ8 . TeTc^2 + CQ9 . Te^2Tc^2$$

SOURCE: ASHRAE

REFERENCE: None

COMPONENT: Reciprocating Compression Chiller (4)

DESCRIPTION: This algorithm calculates the electrical energy required by a chiller using a third degree polynomial curve fit of empirical data which relates the fraction of full-load electrical energy consumed to the fraction of the full chiller load. The fraction of electrical load is multiplied by the full-load electrical energy to get the electrical energy required at the specific part-load condition. The cooling tower load is defined as the sum of the electrical energy required by the chiller and the chiller cooling load.

INPUT: Q = load on chiller, kW.

OUTPUT: EELEC = electrical energy required by the chiller, kW

ETOWER = cooling tower load, kW.

MODEL PARAMETERS (default values in parentheses):

A1-A3 = regression fit of manufacturers' data comparing fraction of full chiller load to fraction of full-load electrical energy required (0.1494, 0.9568, -0.11184)

CAP = capacity of chiller, kW

F = full-load electrical energy/CAP, dimensionless (0.2275).

CALCULATION METHOD: $FELEC = A1 + A2(Q/CAP) + A3(Q/CAP)^2$

$EELEC = FELEC\ (F)(CAP)$

$ETOWER = Q + EELEC$

SOURCE: BLAST 1.2, DOE 1.3

REFERENCE: ASHRAE

COMPONENT: Temperature Controller, Submaster, Hot or Cold Deck Direct-Acting Method

DESCRIPTION: This algorithm determines an output temperature as a function of an input schedule. The direct-acting method of controlling temperatures is used when the hot or cold deck temperature is reset according to the zone requiring the most heating or cooling, respectively. The ramp is specified in such a manner that it is not possible to reset the temperature below the minimum or maximum specified.

INPUT: TX = value of the independent variable by which the output variable is determined according to the specified ramp function, ^oC.

OUTPUT: TY = value of the dependent variable; the reset temperature, ^oC.

MODEL PARAMETERS: TXMAX, TXMIN = the maximum and minimum values of the independent variables which describe the control ramp, ^oC

TYMAX, TYMIN = Maximum and minimum values of the dependent variables which describe the control ramp, ^oC.

CALCULATION METHOD: if TX < TXMIN; TX = TXMIN

if TX > TXMAX; TX = TXMAX

TY = RYMAX - [(TYMAX - TYMIN)/(TXMAX - TXMIN)]/(TXMAX - TX)

SOURCE: BLAST 2.0

REFERENCE: None

COMPONENT: Two-Pipe Fan/Coil Systems

DESCRIPTION: This algorithm calculates the heat exchanged by a two-pipe heating or cooling fan/coil. In a heating fan/coil, the heat rejected by the unit is a function of the design supply air temperature, the zone air temperature, and the air flow rate.

The fan/coil cooling model includes both sensible and latent heat transfer. Within the algorithm, there is a curve fit relating the total cooling load to the sensible cooling load for two-pipe fan/coil units. This curve fit gives the ratio of total to sensible cooling as a function of the entering air dewpoint temperature, the entering air dry-bulb temperature, the entering water temperature, the air volume flow rate, and the water volume flow rate based on manufacturers' catalog data. The algorithm uses a single design point for the particular cooling coil. From that design point, the model decides which curve the coil falls closest to and uses that curve in the hourly simulation to determine the latent cooling load on the coil. The design point data, obtainable from the manufacturers' catalog, consists of the entering water temperature, entering air dry-bulb temperature, entering air wet-bulb temperature, leaving water temperature, leaving dry-bulb temperature, air volume flow rate, water volume flow rate, and the barometric pressure.

INPUT: Ma = mass flow rate of air, kg/s

Tdbe = entering air dry-bulb temperature, ^{o}C

Ts = supply air temperature, ^{o}C

Twb = entering air wet-bulb temperature, ^{o}C.

OUTPUT: Qs = sensible heat transferred, W

Qt = total heat transferred, W.

MODEL PARAMETERS (default values in parentheses):

General

B[1,1,1] - B[6,4,2] = coil curve-fit coefficients

K = 1

J

I	1	2	3	4
1	$-.6464 \times 10^{-1}$	0.5228×10^{-3}	-0.1067×10^{-2}	0.9632
2	0.3946	-0.6534×10^{-1}	0.7162×10^{-1}	0.4731×10^{-3}
3	-0.1143×10^{-2}	0.9864	0.4978	-0.9443×10^{-2}
4	0.7103×10^{-2}	0.3771×10^{-3}	-0.1091×10^{-2}	0.9928
5	0.5305	-0.3009×10^{-1}	-0.7337×10^{-1}	0.3149×10^{-3}

6	-0.1092×10^{-2}	1.0319	5.5645	0.1582

K = 2
J

I	1	2	3	4
1	0.6772×10^{-1}	0.5970×10^{-3}	-0.1138×10^{-2}	0.8875
2	0.2320	-0.9837×10^{-2}	0.7540×10^{-1}	0.5413×10^{-3}
3	-0.1210×10^{-2}	0.9654	0.3202	0.1504×10^{-4}
4	0.7043×10^{-1}	0.5911×10^{-3}	-0.1162×10^{-2}	0.9555
5	0.3221	0.5765×10^{-2}	0.7365×10^{-1}	0.4045×10^{-3}
6	-0.1124×10^{-2}	0.9566	0.5154	0.1807×10^{-3}

Cpa = specific heat of air, J/kgK (1.0)

CF1 - CF4 = volumetric flow rate correction factors (210, 330, 430, 600)

NR = number of rows (1 or 2)

RCpa = air density x specific heat, J/m^3 K (1.2×10^3)

TCpw = water density x specific heat, J/m^3 K (4.19×10^6).

Design point for cooling

Pb = barometric pressure, Pa (101000)

Tdbe = entering air dry-bulb temperature, °C (26.67)

Tdbl = leaving air dry-bulb temperature, °C (15.78)

Twbe = entering air wet-bulb temperature, °C (19.44)

Twe = entering water temperature, °C (7.22)

Twl = leaving water temperature, °C (12.56)

Va = volumetric flow rate of air, m^3/s (0.28)

Vw = volumetric flow rate of water, m^3s (1.87×10^{-3}).

CALCULATION METHOD: There are two distinct calculations involved in this algorithm: the amount of heat transferred into a space by a heating fan/coil, and the amount of heat removed by a cooling fan/coil. For heating:

$$Qt = (Ma)(Cpa)(Ts - Tdbe)$$

For cooling, the sensible heat removed is:

$$Qs = (Ma)(Cpa)(Tdbe - Ts)$$

The total heat removed (sensible and latent) is determined by:

$$\frac{Qt}{Qs} = \frac{(Rcp)(Vw)(Twl - Twe)}{(Va)(RCp)(Tdbe - Tdbl)}$$

This ratio is the actual ratio at the design conditions (using the design point values). The remainder of the algorithm compares this ratio to two others which describe two-pipe fan/coil models. It determines which model the design point most closely matches and uses that as the performance equation for this fan/coil. The total heat transferred at any condition may then be found.

$$D1 = (Vae - CF2)(Vae - CF3)(Vae - CF4)$$

$$D2 = (Vae - CF1)(Vae - CF3)(Vae - CF4)$$

$$D3 = (Vae - CF1)(Vae - CF2)(Vae - CF4)$$

$$D4 = (Vae - CF1)(Vae - CF2)(Vae - CF3)$$

where: $Vae = (Va)(2118.88)$

$$A1 = (D1/C1 \times B[1,1,NR]) + (D2/C2\ B[1,2,NR]) + (D3/C3 \times B[1,3,NR]) + (D4/C4\ B[1,4,NR])$$

$$A2 = (D1/C1\ B[2,1,NR]) + (D2/C2\ B[2,2,NR]) + (D3/C3\ B[2,3,NR]) + (D4/C4\ B[2,4,NR])$$

$$A3 = (D1/C1\ B[3,1,NR]) + (D2/C2\ B[3,2,NR]) + (D3/C3\ B[3,3,NR]) + (D4/C4\ B[3,4,NR])$$

$$A4 = (D1/C1\ B[4,1,NR]) + (D2/C2\ B[4,2,NR]) + (D3/C3\ B[4,3,NR]) + (D4/C4\ B[4,4,NR])$$

$$A5 = (D1/C1\ B[5,1,NR]) + (D2/C2\ B[5,2,NR]) + (D3/C3\ B[5,5,NR]) + (D4/C4\ B[5,4,NR])$$

$$A6 = (D1/C1\ B[6,1,NR]) + (D2/C2\ B[6,2,NR]) + (D3/C3\ B[6,3,NR]) + (D4/C4\ B[6,4,NR])$$

$$QRAT = \frac{1 + T1\ (A1 + [A2][T1] + [A3][T2])}{A4 + (A5/Vwe) + (A6/VWE^2)}$$

where: $T1 = Tdp - Twe$

where: Tdp = dewpoint temperature determined from a psychrometric chart as a function of Twbe, Tdbe, and Pb, ^{o}C

$T2 = Tdbe - Twe$

$Vwe = (Vw)(15850.3)$

Method: for NR = 1, one value of QRAT will be found. For NR = 2, another value of QRAT will be found. Compare Qt/Qs with QRAT to determine the value which is closest to the actual value and then use the corresponding value of NR as the one which best describes the system. Then,

$$Qt = (QRAT)(Qs)$$

SOURCE: BLAST

REFERENCE: ASHRAE

COMPONENT: Zone Temperature Controller

DESCRIPTION: The action of the heating or cooling mechanism for each zone is controlled by the zone temperature. The capacity of the heating/cooling system modulates linearly between prescribed zone temperatures.

INPUT: The schedule of zone temperatures and heating or cooling capacities are entered as control profiles.

Profile = (Q_1 at T_1, Q_2 at T_2,...Q_{10} at T_{10})

where: Q_1, Q_2...Q_{10} are various heating or cooling capacities (entered as negative values if cooling) corresponding to the zone temperatures T_1, T_2,...T_{10}.

CALCULATION METHOD: An energy balance method calculates the equilibrium point where the heating/cooling energy matches the loads.

OUTPUT: The results of the energy balance yield Q-heating/cooling energy input to the zone, and T zone.

NOTE: When deck reset is specified, some error results.

SOURCE: BLAST 2.0

REFERENCE: None

COMPONENT: Air-Handler Fan, Constant Volume System

DESCRIPTION: This algorithm calculates the electrical energy required by a fan as a function of the total pressure, the mass flow rate of air, and the fan efficiency. The electrical energy required is calculated one time and used for the period of the simulation.

INPUT: No hour by hour input.

OUTPUT: P = Power required by the fan, W.

MODEL PARAMETERS (default values in parentheses):

Fair = volumetric flow rate of air determined by adding the volumetric flow rates required by each zone, m^3/s

Nf = fan efficiency = air power/electrical power (0.7)

T_p = total pressure produced by the fan (static pressure and velocity pressure), Pa (625).

CALCULATION METHOD: $P = (T_p)(Fair)(1.033)/N_f$

where: 1.033 is a conversion factor.

SOURCE: BLAST 2.0

REFERENCE: Fan Engineering

COMPONENT: Air-Handler Mixing Box

DESCRIPTION: The mixing box is a device which performs adiabatic mixing of two streams of moist air: outdoor air and return air. The algorithm calculates the temperature and humidity ratios of the air leaving the mixing box based on the temperature and humidity ratio of the two input air streams and the ratio in which they are mixed.

INPUT: Fod = fraction of outdoor air determined by the control strategy, dimensionless

MIN = minimum Fod, dimensionless

Tod = outdoor air dry-bulb temperature, ^{o}C

Tra = return air dry-bulb temperature, ^{o}C

Wod = outdoor air humidity ratio, kg/kg

Wsa = supply air humidity ratio, kg/kg.

OUTPUT: Tmix = mixed air dry-bulb temperature, ^{o}C

Wmix = mixed air humidity ratio, kg/kg.

MODEL PARAMETERS: None.

CALCULATION METHOD: if Fod <MIN, let Fod = MIN

$$Tmix = Tod \cdot Fod + Tra\,(1 - Fod)$$
$$Wmix = Wod \cdot Fod + Wsa\,(1 - Fod)$$

SOURCE: BLAST 2.0

REFERENCE: ASHRAE

COMPONENT: Cooling Coil

DESCRIPTION: This algorithm calculates the total heat transferred from the air to the water through a chilled water cooling coil in two steps. The cooling coil model uses the design conditions (catalog data) at one operating point to determine the equivalent heat transfer area of the coil in the first step. The fluid flow rates and velocities, the air wet- and dry-bulb temperatures, the cooling load, and the barometric pressure are sufficient data to determine this area. The equivalent heat transfer area is then used in the second step with the current entering air and water temperatures and the flow rate of air to solve for the actual total heat transfer rate for the current condition. This calculation is based on the premise that the air is leaving the coil at the wet-bulb temperature.

INPUT: He = entering air enthalpy, J/kg

Mass = mass flow rate of air, kg/s

Tdbe = entering air dry-bulb temperature, ^{o}C

Twbe = entering air wet-bulb temperature, ^{o}C

Twe = entering water temperature, ^{o}C.

MODEL PARAMETERS (default values in parentheses):

Cp, air = specific heat of air, J/kgK (1000)

Hdpe = enthalpy at Tdpe determined from a psychrometric chart, J/kg

H = enthalpy at T determined from a psychrometric chart, J/kg

Ma = mass flow rate of air through the coil, kg/s

Pb = atmospheric pressure, Pa (101000)

Qtd = heat transferred by dry coil, W

Tdbe = entering air dry-bulb temperature, ^{o}C (29.4)

Tdpe = entering air dew-point temperature, ^{o}C (17.8)

Te = entering air dry-bulb temperature, ^{o}C (29.4)

T_L = leaving air dry-bulb temperature, ^{o}C (12.8)

Ts1-Ts2 = coil surface temperatures at the entrance and exit of the coil, respectively; determined from the ARI Standard 410-72 using CH = (Rmw + Rw)/(Cp,air,Raw)

Twe = entering water temperature, ^oC (7.2)

Va = face velocity of the air stream, m/s (2.5)

VOLa = volumetric flow rate of air, m^3/s

VOLw = volumetric flow rate of water, m^3/s (.00088 . VOLa)

Vw = velocity of the water in the coil, m/s (1.4)

CALCULATION METHOD: There are two major steps or phases in the development of this algorithm and a large number of smaller intermediate steps and calculations. The values listed under MODEL PARAMETERS are used in Phase 1, and the values under INPUT are used in Phase 2. Once the coil has been defined, Phase 1 may be omitted from the calculation. A psychrometric chart and ARI standard 410-72 are required.

Phase 1

Five values for the heat transfer resistance are defined:

Rad = air side resistance, dry coil, $^oC/W$

Raw = air side resistance, wet coil, $^oC/W$

Rmd = metal resistance, dry coil, $^oC/W$

Rw = refrigerant side film resistance $^oC/W$

$$Rad = \frac{0.0263}{Va^{0.504}}$$

$$Raw = \frac{0.0256}{Va^{0.542}}$$

$$Rmd = 2.24521 \times 10^{-3} + 0.0079\ Rad$$

$$Rmw = 2.24521 \times 10^{-3} + 0.00709\ Raw/P$$

where: P = ratio of total heat of air water vapor mixture to sensible heat of air water vapor mixture

$$P = (4.0138 - 2.413 \times 10^{-3} Pb) + (0.1065 + 8.7354 \times 10^{-7} Pb)(32.0 + 1.8\ TdPe) + (.00217 - 1.4303 \times 10^{-8} Pb)(Tdpe \cdot 1.8 + 32.0)^2$$

$$Rw = \frac{0.00592}{Vw^{.8}\ (1.352 + 1.98 \times 10^{-2} Twm)}$$

where: $Twm = (Tw_L - Twe)/2$

Face area = Fa = VOLa

Water flow area = WFA = $\frac{VOLw}{Vw}$

Surface area: calculation of the coil surface area requires knowing whether the coil is dry, partially wet, or completely wet:

I A dry coil satisfies this relation

$$Hab < H$$

II A partially wet coil satisfies this relation

$$He > Hab > H$$

III A wet coil satisfies this relation

$$Hab > He$$

where: $Hab = \frac{Tdpe - Twe + Y \;\; He + CH \;\; Hdpe}{CH + Y}$

where: CH = (Rmw + Rw)/(Cp, air + Raw)

Y = (Tw - Twe)/(He - H).

Other important parameters to determine the coil surface area:

Tab = temperature of air at dry-wet boundary, ^{o}C

Twb = temperature of refrigerant at dry-wet boundary, ^{o}C

Tab = Tdbe - (H e - H)/Cp,air

Twb = Tw - Y Cp,air (Tdbe - Tab).

To calculate A surface, the following methods are used according to the three categories (I, II, III) given above.

I A surface = $\frac{QTD(Rad + Rmd + Rw)}{DTMI}$

where: Qtd = Qt for a dry coil

DTMI = log mean temperature difference for Case I

$$DTMI = \frac{(Tdbe - Tw\ ) - (Tdb\ - Twe)}{n \dfrac{(Tdbe - Tw)}{(Tdbe\ - Twe)}}$$

II A surface = A dry + A wet =

((Ma . Cp,air (Tdbe - Tab) (Rad + Rmd + Rw))/
(DTMII)) + (((Qt - Ma Cp,air
(Tdbe - Tab)) (Rmw + Rw))/(DTMSII)

where: DTMII and DTMSII are log mean temperature differences for Case II.

$$DTMII = \frac{(Tdbl - Tw\ ) - (Tab - Twb)}{n\ \dfrac{Tdbe - Tw}{(Tab - Twb)}}$$

$$DTMSII = \frac{(Tdpe - Twb) - (Ts2 - Twe)}{n\ \dfrac{(Tdpe - Twb)}{(Ts2 - Twe)}}$$

where: Ts2 = the coil surface temperature at the exit of the coil as determined from ARI Standard 410-72 using CH = (Tdpe - Twb)/(Hab - Hdpe) (see MODEL PARAMETERS)

A surface = Qt (Rmw + Rw)/DTMSIII

where: DTMSIII = log mean temperature difference for Case III

$$DTMSIII = \frac{(Ts1 - Tw\ ) - (Ts2 - Twe)}{n\ \dfrac{(Ts1 - Tw)}{(Ts2 - Twe)}}$$

Where: Ts1 and Ts2 are the coil surface temperatures at the entrance and exit of the coil, respectively, found from the ARI Standard 410-72 using CH = (Rmw + Tw)/(Cp,air. Raw). The necessary values from Phase 1 are A surface, Fa, and WFA.

$$Qt = M\ air\ (he - h\ )$$

where: $h\ = he + (hsb - he)\ \cdot\ (1-e^{-SC})$

where: SC = number of transfer units

$$= \frac{Asurface}{M\ air\ \ Cp,air\ \ Rad}$$

where: Rad = 0.0263/(main/1.2 FA)

hsb = saturation enthalpy at Tsb

where: $Tsb = \dfrac{Tdb\ \ - Tdbe\ \ \cdot\ \ e^{-SC}}{1 - e^{-SC}}$

SOURCE: BLAST 2.0

REFERENCE: ARI Standard 410-72

COMPONENT: Fan

DESCRIPTION: This algorithm calculates a fan's electrical energy requirements as a function of the total pressure, the mass flow rate of air, and the fan efficiency. The electrical energy required is calculated once and used for the period of the simulation. This algorithm also calculates the temperature rise across the fan for the above condition (i.e., heat input from fan into air stream).

INPUT (default values in parentheses):

Q = air-flow rate determined by adding the air-flow rates required by each zone, m^3/s

N_f = fan efficiency = air power/electrical power (0.7)

H = total pressure produced by the fan (static pressure and velocity pressure), Pa (625).

OUTPUT: P = electrical power required by the fan, W

DT = temperature rise across fan, ^{o}C.

MODEL PARAMETERS (default values in parentheses): None

CALCULATION METHOD: $P = (H)(Q)/N_f$

$DT = H$ (Density of air at STP) $= H\ (1.2 \times 10^{-3})$

SOURCE: Fan Engineering

REFERENCE: BLAST 2.0

COMPONENT: Fan, VAV

DESCRIPTION: This algorithm calculates the fraction of full-load (design) electrical power required by a fan equipped with a volume control device as a function of the fraction of full (design) air flow required. This algorithm models three kinds of volume control devices:

1. Inlet vanes
2. Discharge dampers
3. Variable speed motor.

Each of these volume control methods are modeled by a polynomial curve fit to data provided by the manufacturer. A second part of this algorithm calculates the temperature rise across the fan (i.e., heat input by fan into the air stream) for the actual flow conditions.

INPUT: Q = air-flow rate, m^3/s

OUTPUT: P = electrical fan power, W

DT = temperature rise across fan, ^oC.

MODEL PARAMETERS (default values in parentheses):

A1-A5 = coefficients of a fourth order equation with one independent variable which relates, for an inlet vane controlled fan, the fraction of full-load power to the fraction of full mass flow rate. These values are determined by a least squares curve fit of inlet vane performance data (0.3071223, 0.30850535, -0.54137364, 0.87198823, 0)

B1-B5 = coefficients of a fourth degree equation with one independent variable which relates, for a discharge damper controlled fan, the fraction of full-load power to the fraction of full mass flow rate. These values are determined by a least squares curve fit of discharge damper performance data (0.37073425, 0.97250253, -0.34240761, 0, 0)

C1-C5 = coefficients of a fourth degree equation with one independent variable which relates, for a variable speed motor controlled fan, the fraction of full-load power to the fraction of full mass flow rate. These values are determined by a least square curve fit of variable speed motor performance data (0.0015302446, 0.0052080574, 1.1086242, -0.11635563, 0)

PD = fan power at design flow rate, W.

DTD = Temperature rise across fan at design conditions, ^oC

NOTE = QD, PD, and DTD can be input or obtained from air-handler fan (constant volume) model.

CALCULATION MODEL: $F = Q/QD$

$PF = A1 + (A2)\ F + (A3)\ F^2 + (A4)\ F^3 + (A5)\ F^4$
(inlet vanes)

$PF = B1 + (B2)\ F + (B3)\ F^2 + (B4)\ F^3 + (B5)\ F^4$
(discharge dampers)

$PF = C1 + (C2)\ F + (C3)\ F^2 + (C4)\ F^3 + (C5)\ F^4$
(variable speed motor)

$P = (pF)(PD)$

$DT = (PF)(DTD)$

SOURCE: BLAST 1.2*, 2.0

REFERENCE: None

*In BLAST 1.2 A5 = B4 = B5 = C5 = 0 (i.e., cannot be user specified).

COMPONENT: Cooling Tower

DESCRIPTION: This cooling tower algorithm defines the electrical energy required by the cooling tower, the outlet water temperature, the amount of heat energy dissipated as a function of the inlet water temperature, the load on the tower, the nominal capacity of the tower, the ambient air wet-bulb temperature, and a series of second degree curve fits to data relating the actual capacity of the tower with air and water temperatures. This model may be used for one or more cooling towers working simultaneously and with a fixed or variable water flow rate.

INPUT: Tin = entering water temperature, ^{o}C

TL = tower load, kW

Twb = wet-bulb temperature of the air, ^{o}C.

OUTPUT: EE = electrical energy required by the cooling tower and circulating pump, kW

PL = cooling load satisfied by the cooling tower, kW

Tout = leaving water temperature, ^{o}C.

MODEL PARAMETERS (default values in parentheses):

BPLR = best part-load ratio when using more than one cooling tower (0.4365)

CAP = capacity of cooling tower, kW

Cpw = specific heat of water, kJ/kgK (4.19)

C1 - C18 = six sets of three coefficients per set which are determined from a second degree curve fit of manufacturers' data relating the ambient air wet-bulb temperature, Twb, and the capacity correction factor, CCF (default values are tabulated below)

	AP, ^{o}C	C	C	C
(C1-C3)	3.33	191.800	-1.2127	0.001924
(C4-C6)	4.44	230.540	-1.4930	0.002426
(C7-C9)	5.55	126.235	-0.8070	0.001296
(C10-C12)	6.66	131.587	-0.8563	0.001400
(C12-C15)	7.77	86.736	-0.5564	0.0008975
(C16-C18)	8.88	70.128	-0.4497	0.0007258

D1-D3 = coefficients of a second degree curve fit relating the range, R, to the rating factor to capacity correction factor ratio, RF/CCF, from manufacturers' data (0, 5.139, 0)

FPOW = electrical energy required by the fan/CAP, kW/kW. (0.012)

PE = pump electrical energy required to tower cooling load ratio, kW/kW (0.013)

TU = tower unit = area required to cool 1 gpm at the 90-80-point, kW (1.465)

90-80-70 point = Tin = 90°F (32.2°C), Tout = 80°F (26.7°C). TWB = 70°F (21.1°C).

CALCULATION METHOD: This algorithm uses an iterative method to determine the leaving water temperature, Tout, that will satisfy the load, TL. The following is a step-by-step examination of that iterative procedure.

Step 1. Variable water-flow rate:
$R = Tin - Tout$
$Mw = TL/R \cdot Cpw$
(Go to Step 4)

Step 2. Fixed water-flow rate:
$Mw = CAP \cdot RWCC$

Step 3. $R = TL/(Mw \cdot Cpw)$

Step 4. $AP = Tout - Twb$

Step 5. if $AP \leq 2.78°C$, $AP = 2.78°C$

Step 6. The rating factor, RF, is the product of the capacity correction factor, CCF, and a curve fit of the range R.

if $AP < 3.33°C$, $AP = 3.33°C$
if $AP \geq 8.88°C$, $AP = 8.88°C$

Step 7. Find the value of CCF for the two values of AP nearest it according to the following equations (Twb in °K):

AP	Equation
3.33	$CCF = C1 + C2\ Twb + C3\ Twb^2$
4.44	$CCF = C4 + C5\ Twb + C6\ Twb^2$
5.55	$CCF = C7 + C8\ Twb + C9\ Twb^2$
6.66	$CCF = C10 + C11\ Twb + C12\ Twb^2$
7.77	$CCF = C13 + C14\ Twb + C15\ Twb^2$
8.88	$CCF = C16 + C17\ Twb + C18\ Twb^2$

For example, AP = 4.0, CCF lower = CCF for AP = 3.33
CCF upper = CCF for AP = 4.44

Step 8. Use a weighing factor to determine the true value of CCF from CCF upper and CCF lower:

For example, W upper = 4.44 - 4.0
W lower = 4.0 - 3.33

CCF = CCF upper - W lower + CCF lower - W upper

Step 9. $RF = CCF \cdot (D1 + D2 \cdot R + D3 \cdot R^2)$

Step 10. The rating capacity in TU needed to satisfy the load is:
$RC = RF \cdot Mw$

Step 11. Nominal cooling tower capcity in TU:
CAPTU = CAP/1.465

Step 12. For more than one cooling tower, the best part-load operation for this tower is defined:
$CAPTUH = BPLR \cdot CAPTU$

Step 13. For more than one cooling tower; if CAPTUH < RC, skip this step.
EU = electrical energy used = $0.5 \cdot FPOW \cdot CAP \cdot RC/CAPTUH$
(Go to Step 16)

Step 14. If CAPTU < RC skip this step.
$EU = FPOW \cdot CAP \cdot RC/CAPTU$

(Go to Step 16)

Step 15. If the capacity is insufficient (i.e., CAPTU < RC), increase the leaving water temperature and reduce the capacity required:
Tout = Tout + 1.11
if Tout $\geq$ Tin go to next step.
(Go to Step 4)

Step 16. Load satisfied, PL, is the smallest of

a. Load on the tower kW.
$RC \cdot 1.465$

b. Capacity of the tower, kW.
CAP

Step 17. $EE = EU + PE \cdot CAP.$

SOURCE: BLAST 2.0

REFERENCE: Consultants Computation Bureau Total Energy Plant Simulation

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