AN ABSTRACT OF THE THESIS OF

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Title: AN ANALYSIS OF THE CONTROL AND ECONOMICS OF A THREE-PIPE HEAT PUMP SYSTEM

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The use of a three-pipe distribution system is known to impair the operating costs of an air conditioning system. Additional costs are encountered because heating water and cooling water are mixed in a common return pipe after use by the various air conditioning units. This investigation covers the design and installation of an automatic temperature control system to operate a heat pump air conditioning system utilizing a three-pipe distribution system. The investigation also was directed toward the study of the economics of the three-pipe system while in use providing heating and cooling to an industrial plant.

The experiment was performed with a centrifugal refrigeration compressor heat pump of 497 tons of refrigeration capacity. Tests were conducted at outside air temperatures of 42, 51.5, and 53° F. During each test, heating water flow, cooling water flow, heating supply water temperature, cooling supply water temperature, common return water temperature, and cooling return water temperature

were measured. Sufficient data were recorded, during each test, to determine the heating load, cooling load, and operating penalty due to the mixed flow condition in the common return. These results were correlated with outside temperature occurrence data to determine the additional annual operating cost encountered because of the use of a three-pipe system.

The additional annual owning cost of a conventional four-pipe system above that of the three-pipe system was determined. This cost included items for depreciation, interest, and taxes. A comparison between the annual owning and operating costs of the three-pipe system and the four-pipe system was made. It was determined that the three-pipe system installed in the particular industrial plant investigated exhibits a cost approximately one-third of that of the conventional system used for comparison.

It was determined that the three-pipe system would become relatively less costly to own and operate if applied to a system with similar loads in all areas, if used with a heat pump system, and if used with a system employing final heat transfer elements selected for as high a water temperature drop or rise as possible. The three-pipe system would become relatively more costly to own and operate if the system piping runs were short, if the individual air conditioning units were larger, and if a system employing separate heating and cooling sources was used.

An Analysis of the Control and Economics of a Three-pipe Heat Pump System

by

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I. INTRODUCTION

Purpose of the Thesis

The purpose of this thesis is to present the design, installation, and evaluation of an automatic temperature control system for a heat pump air conditioning system serving an industrial plant. The design portion will present the operation desired and the selection of equipment to afford this operation. The installation portion will present the main methods utilized in the field assembly of the selected control components. The evaluation phase will present an economic evaluation of certain phases of the air conditioning system as installed and controlled by the automatic temperature control system. The evaluation will compare the operating and owning costs of the distribution system as installed with those of a more conventional system.

The air conditioning system is an especially interesting installation to examine. It is unique in that the heat source for the heat pump is energy wasted in an industrial process. The system also uses the relatively uncommon three-pipe distribution system for heated and chilled water. This distribution system has been both praised and criticized for its initial and operational cost considerations.

Concept of the Air Conditioning System

The arrangement of the system is shown in Figure 1. The basic component of the air conditioning system is the heat pump. This consists of a centrifugal refrigeration compressor using refrigerant 113, an evaporator heat exchanger, and a condenser heat exchanger. The three portions of the heat pump are the product of a single manufacturer and form an integrated unit. The centrifugal compressor is a two stage direct drive machine of the hermetic type. The evaporator is a shell and tube water chiller with refrigerant in the shell and system chilled water in the tubes. The condenser is a dual circuit shell and tube heat exchanger with refrigerant in the shell and water in the two tube circuits. One condenser circuit contains system heating water and the other circuit contains well water. The three portions were assembled, tested, and dismantled for shipping at the manufacturer's factory. Upon arrival, they were reassembled, charged with refrigerant, and tested. A single heat pump with a capacity of 479 tons of refrigeration was installed with provision for a second unit to be added when future buildings are constructed and the system extended.

The water circulated in the evaporator serves as the heat source for the heat pump. Chilled water leaves the evaporator at a design temperature of 42 degrees F and is pumped to the cooling

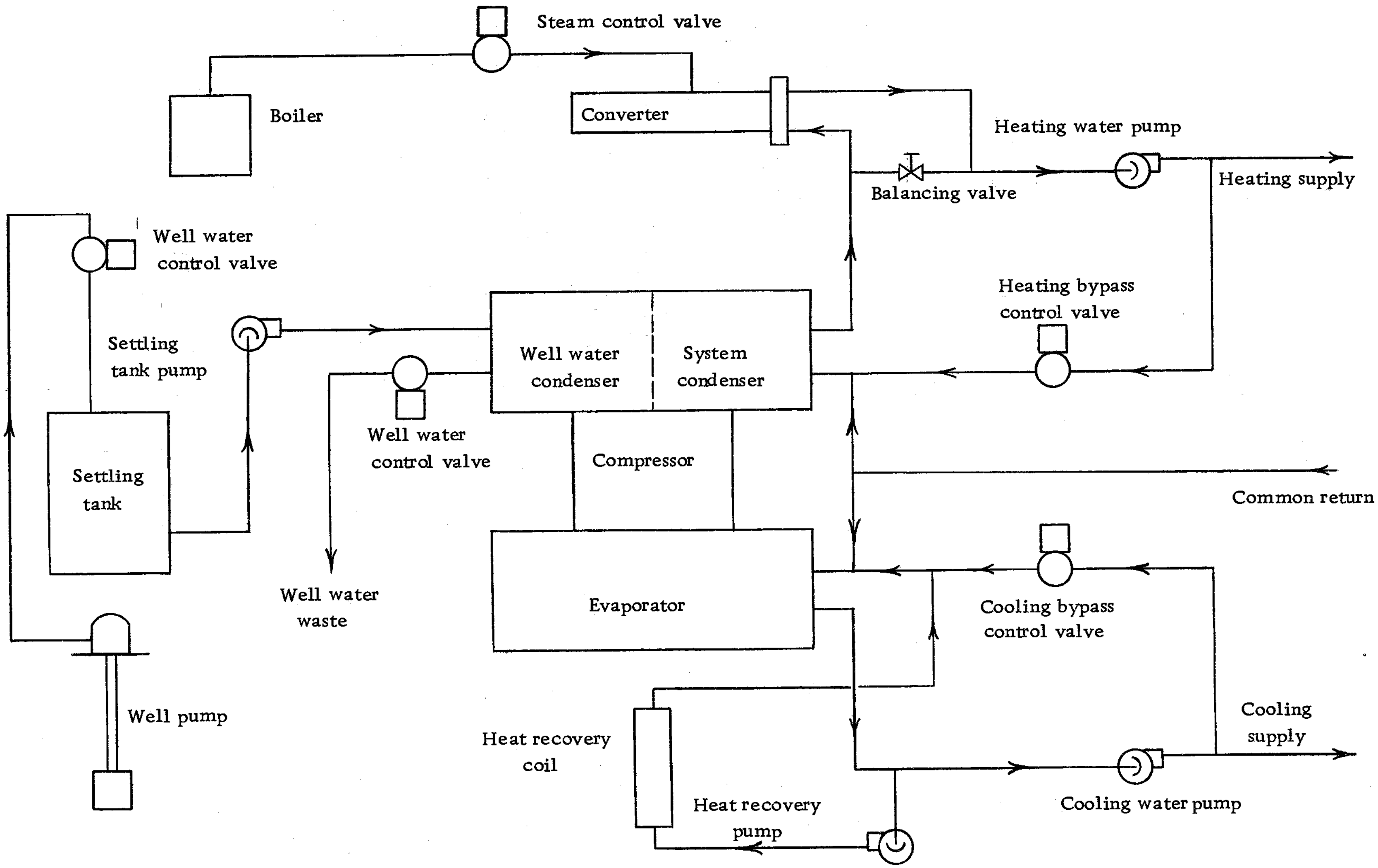

Figure 1. Basic arrangement of the heat pump system components.

coils of the air conditioning units. It is also pumped to the heat recovery coils to scavange waste industrial process energy if needed to meet the total system heating requirement. The condenser has two water circuits. One provides heating water for the air conditioning units. The other uses well water as a heat sink during the cooling cycle. Additional heat may be added to the water during the heating cycle by the steam converter if the heating load requires additional capacity above that furnished by the heat pump.

If the heating and cooling loads imposed on the heat pump by the air conditioning units are equal, use of the heat recovery coils or well water condenser circuit is not required. Heat is removed from the system chilled water and added to the system heating water. This is true regardless of the magnitude of the heating and cooling loads, so long as they are equal. Operation of the heat recovery coils or the well water waste is only required when one system load exceeds the other.

If the air conditioning system heating load exceeds the cooling load, sufficient heat is no longer available from the water being chilled for use by the space supply units. The additional heating requirement must be made up from a heat source outside of the air conditioning system. This source is the heat recovery coils. Chilled water is circulated through the coils, where it is warmed and returned to the evaporator inlet to mix with water returning from the space system. This increases the heat available to the heat pump for transfer to the heating water flowing through the condenser. Well water flow through the condenser does not occur under this condition since the net requirement is for heating.

If the air conditioning system cooling load exceeds the heating load, heat removed from the chilled water and transferred to the condenser can no longer be removed by the system heating water. The additional heat sink requirement must be made up by using well water. As previously stated, the well water is warmed in the condenser and wasted. Chilled water flow to the heat recovery coils does not occur under this condition since the net requirement is for cooling.

It is essential that the presence of a simultaneous heating and cooling load within the air conditioning system be realized. Some manufacturing area might require cooling because of high internal loads such as lighting or process machinery heat liberation. At the same time, an office area might require heating due to heat losses through walls and windows exceeding the internal gains. With this realization, it can be seen that the heat pump does not actually change from a heating load to a cooling load. It actually operates to afford simultaneous heating and cooling at almost all times. The change is rather one of degree in which the heat pump operates in three zones: where the heating load exceeds the cooling load, where the heating load equals the cooling load, and where the heating load is less than the cooling load. If heating is the predominate load, the heat recovery coil heat source is used. If the heating and cooling loads are equal, neither heat source or heat sink is utilized. If cooling is the predominate load, the well water heat sink is used.

The heat recovery coil heat source is a unique feature of the system. The manufacturing process performed in the plant requires substantial amounts of heat treating. The heat treating furnaces are gas-fired and vented through the roof in a normal fashion. All furnaces are grouped in a separate portion of the building which is isolated from the general manufacturing area. The only connection is by two corridors provided for the movement of personnel and product. Excessive amounts of heat are liberated by the furnaces to the surrounding area. This can cause an extremely high space temperature in the upper portion of the area. Temperatures as high as 120 degrees F are experienced. It is this heat liberated to the space which is utilized to warm the heat recovery coils. Heat is in no way extracted from the flue gasses. The warm air near the ceiling is moved by two heat recovery fans across a heat recovery chilled water coil in the discharge of each fan. The coils, as previously stated, serve as a heat source for the heat pump. The air is then either discharged outside or recirculated to the space. The amount of heat available from this source is sufficient to provide the entire heating requirement of the plant at all conditions above approximately

35 degrees F outside. A boiler is provided to meet requirements in excess of the heat pump capacity.

The heated and chilled water from the heat pump is circulated throughout the plant for use by the space air conditioning units. The distribution system is of the three-pipe type. A heating water supply pipe carries water to the heating control valves and a chilled water supply pipe carries water to the cooling water control valves of the air conditioning units. Heated or chilled water is used as required by each individual unit to meet the needs of the area it serves. Water from the valves passes through the tubes of a finned heat exchange coil in the individual unit. Air passed across the outside of the coil is either warmed or cooled as required. When the water leaves the coil it enters a common return pipe for return to the heat pump. The three-pipe system name is derived from the use of a heating supply pipe, a cooling supply pipe, and a common return pipe.

The three-pipe system is less expensive to initially install as it does not require separate heating and cooling return pipes and requires fewer control valves. It does, however, introduce certain operating inefficiencies. When the entire air conditioning system is utilizing heating in each unit, there is no flow in the cooling pipe. The heating pipe carries the entire system flow which is utilized by the units as needed. The heating water leaves the units and enters the common return which acts as a heating return. In a similar manner, when the entire air conditioning system is utilizing cooling exclusively in each unit, the common return acts as a cooling return. If the system requires some heating and some cooling, the common return receives both heating and cooling water. Those units requiring heating discharge their heating water to the return. Those which require cooling likewise discharge their cooling water to the same common return. The return pipe therefore carries a mixture of heating and cooling water back to the heat pump.

When the return reaches the heat pump, an amount equal to the heating supply flow enters the condenser. An amount equal to the cooling supply flow enters the evaporator. The temperature of the common return under this mixed flow condition is between the temperature of the heating water leaving those coils on heating and the cooling water leaving those coils on cooling. This requires the condenser to add more heat to the portion of the water flowing through it than if the return was at heating coil discharge temperature. In a like manner, the evaporator must remove more heat than if the water entering it was at cooling coil discharge temperature. The mixed temperature of the common return will vary between cooling and heating coil discharge temperatures in proportion to the amount of each entering the pipe. If the flow is predominately cooling return, then the mixed temperature will approach this temperature. If the flow is primarily from the discharge of coils on heating, then the common return will be near the heating return temperature.

This varying of the mixed temperature with varying load removes the operating cost penalty at times when the system is on full heating or on full cooling. At these times, the return is at the temperature it would be if it were used exclusively for the predominate load. This operational characteristic lets the capacity of the heat pump be selected without a penalty due to the three-pipe system.

At other times, when the system is at a condition with simultaneous heating and cooling flow, the operational cost penalty is present. In assessing the penalty, it should be remembered that the system is a heat pump. If cooling is the predominate load, the evaporator is penalized by the higher return temperature due to heating water being introduced into the return. After the heat pump has removed the heat from the evaporator, it must be liberated in the condenser. If the heat to be liberated in the condenser exceeds the heating requirement of the system water circuit, the heat must be wasted. The heating load existing in the condenser therefore does not penalize the system so long as it is smaller than the cooling load in the evaporator. Similar conditions exist when the load is predominately heating. The penalty is then due exclusively to the return entering the condenser being at a lower temperature than the heating coil discharge temperature. A maximum operational cost penalty would exist when heating and cooling loads were equal.

Concept of the Temperature Control System

The automatic temperature control system utilized for controlling the air conditioning system is of the pneumatic type. The control air supply is taken from a large air compressor supplying process air to the manufacturing processes. The control consumption is an extremely minor amount of the compressor capacity. There are times when the process compressor will be inoperative and a standby compressor was supplied. The standby compressor is sized to supply the control system exclusively. Means were provided to accomplish automatic changeover to the standby unit and reversal to the plant compressor depending on operation of the plant unit. Control air is reduced in pressure and filtered before entering the control system piping. The control air supply is provided at 20 psig. The nominal operating range of most of the control equipment is 3 psig to 13 psig.

The control equipment which operates the air conditioning components was selected generally to fail to a certain position. All valves and other components associated with supplying heat were selected as normally open devices to provide heat on air failure. Those valves passing cooling water were selected as normally closed to stop the flow of cooling on air failure. Outside air dampers were likewise arranged for normally closed operation. This concept was generally followed throughout the control system so that in the event of an air failure, heat could be provided by manually cycling fans and pumps. This would allow continued operation of the system on a manual basis and would prevent possible freezing temperatures occurring in the event of an unexpected air failure.

The pneumatic system utilized a modulating form of operation. In the case of a room thermostat, typical operation would be a 3 psig output at 72 degrees F room temperature and a 13 psig output at 75 degrees F with uniform graduation between. If the thermostat were connected to a heating valve, the valve would be open at 3 psig, closed at 13 psig, and proportionally open at pressures between.

Many of the control functions were performed utilizing sensors and controllers. The sensor is a nonadjustable transmitter only and varies its output from 3 psig to 15 psig as the measured temperature shifts over a span of 200 degrees F. The controller is connected to the sensor output and adjusted to control to a particular value of pressure. The effect is that the controller actually controls to the value of the temperature existing at the sensor. The setting on the controller is expressed in degrees F and the intermediate transmission pressure is neglected. This offers the advantage of having the adjustment at the controller located remotely from the location at which the temperature is measured. The system as installed utilizes distances as great as 500 feet between sensor and controller locations.

The system also allows temperatures at the sensor location to be read at any desired location by connecting a pressure gage calibrated in degrees F to the sensor output. This technique is used extensively in gathering and displaying system temperatures at various control operational panels.

In the design portion of the thesis it should be noted that the system design is primarily conceptual. This is true of automatic temperature control systems as they are principally qualitative in nature. The only quantitative considerations are those of determining valve sizes. Selecting valve sizes is an extremely simple procedure through the use of monographs and tabular information. The qualitative considerations are the difficult portion of a control system installation as little formal material is available covering the subject. The field is also a dynamic one in that equipment available is constantly changing. This causes the design engineer to rely primarily on experience and judgment. The design section of this thesis is notably lacking in quantitative information as little was required.

The general objectives of the control system design were to afford proper and economical operation of the air-conditioning system, to maintain manufacturing area space temperatures at 75 degrees F ± 3 degrees F, to maintain office and cafeteria space temperatures at 75 degrees F ± 1.5 degrees F, to enable the operator to easily operate the system, to afford minimum maintenance costs for the control system, and to install the system at the least possible cost consistent with the other objectives. The control system is extensive in scope and physical arrangement. Sufficient centralization of operational controls and information was provided to enable the operator to analyze and adjust the system with a minimum of effort.

II. DESIGN

General Factory Area Supply Systems

The general factory area is air conditioned by 15 air supply units. These units are called upon to perform three functions. The first function is to provide heated or cooled air to the space for the maintenance of proper environmental conditions. The second function is to maintain a minimum building temperature during periods when the space is partially or completely unused. The third function is to provide a proper quantity of outside air to match the amount utilized in the exhaust of certain manufacturing equipment.

The general factory area supply systems are located immediately below the roof of the manufacturing area and are composed of three components. The first component is a pair of outside air intake and recirculated air dampers. The second component is a single finned heating and cooling coil. This coil is supplied with heated or chilled water within the tubes depending upon the requirements of the space. The air from the dampers is drawn over the outer surfaces of the coil to be heated or cooled. The third component is the centrifugal fan which moves the air through the system and discharges it into the general factory area.

The dampers are linked together in such a manner that as the outside air damper moves from the closed position to the open position, the recirculated air damper moves from the open position to the partially open position. This partial movement of the recirculated air is necessitated by the system requirement that the unit not handle full outside air across the heating and cooling coil with the associated possibility of freezing and damage to the coil. The outside air damper is closed when the supply system is off. When the supply system is operating, the outside damper moves to a minimum ventilation position, provided that the exhaust requirement of the manufacturing equipment is not present. If this requirement is present, the dampers move to a maximum ventilation position.

The dampers are operated by pneumatic damper actuators. The selection of the actuators is based upon damper size with the actuator used having an area rating of approximately twice the actual damper area. This underrating of the actuator was done to assure smooth positioning of the dampers even under severe operating conditions which might result in increased friction. This method of underrating of the actuator was used in lieu of a positive positioning relay as it was less expensive. The control air to the supply system controls is provided by an electrically operated pneumatic relay wired to the supply fan motor terminals. If the supply fan is off, the relay is deenergized, the control air supply is stopped, and the supply system controls are vented to atmosphere. This results in outside damper closure. If the supply fan is operating, the relay is energized and control air is made available to a second relay wired to the motor terminals of the exhaust fan and to a manually adjustable positioning switch. The output of the positioning switch is connected to the normally open port of the exhaust fan relay. The output of the supply fan relay is connected to the normally closed port of the exhaust fan relay. If the exhaust fan is off, the output of the positioning switch passes through the exhaust fan relay to the damper actuator and positions the outside damper to the minimum ventilation position. If the exhaust fan is operating, the output of the supply fan relay passes through the exhaust fan relay and positions the outside damper to the maximum ventilation position. The outside and return air dampers are of the parallel blade multi-louvre type. They are installed at right angles to each other and the blades arranged so that the airstreams are directed at each other. This was done to cause impingement of the airstreams and assist in mixing to provide a uniform temperature entering the system heating and cooling coil.

The supply system coil is a combination heating and cooling coil. The coil is provided with heated or chilled water depending upon the requirements of the space. If the supply system is inoperative, the flow to the coil is locked out. This is to assure that the common heating and cooling return does not receive any water that has not been through a supply system coil load.

The space thermostat is a pneumatic thermostat of the modulating type. Its output is connected to the pilot port of the positive positioning relay of the chilled water valve actuator, and through a pneumatic diverting relay to the positive positioning relay of the heating water valve actuator. The air supply to the chilled water valve is obtained from the relay actuated by the supply fan. This relay also provides a signal to operate the diverting relay. When the supply fan is off, relay removes air from the cooling valve actuator causing it to close, and actuates the diverting relay so that it provides full supply air pressure to the heating valve actuator causing it to close. When the supply fan is operating, the heating and cooling valve actuators operate in sequence to meet the space requirements. The actuators are selected so that in the event of control air failure, the heating valve will fail open and the cooling valve will fail closed. Each actuator is equipped with a positive positioning relay to assure that the valves operate in sequence regardless of varying system water pressures. The heating and cooling valves have single seated globe pattern bodies, are equipped with stainless steel seats, and have stainless steel stem guided plugs with an equal percentage flow characteristic. The valves are sized to pass the desired flows at the desired pressure drops. The pressure drop varies with the unit location in relation to the pumps, and this is accounted for in each unit's valve selection.

The unit fan is operated from an operational system described under another section.

Loading Dock Area Supply Systems

The two loading docks are heated by a single unit heater in each area. These units are called upon to perform the function of providing spot heating to these high heat loss areas.

The unit heaters are located immediately below the roof adjacent to the loading dock doors. They are composed of two components. The first component is a finned heating coil. The coil is supplied with heated water within the tubes. Recirculated air from the space is drawn over the outer surfaces of the coil to be heated. The second component is the centrifugal fan which moves the air through the system and discharges it adjacent to the loading dock door.

The space thermostat is a pneumatic thermostat of the modulating type. Its output is connected through an electrically operated pneumatic relay to the valve actuator of the heating valve. The relay is interlocked to the supply fan and closes the valve if the fan is inoperative, or allows the thermostat to modulate the valve if the fan is operating. The valve is of the same construction and is selected in the same manner as those provided for the factory supply units.

The fan is operated by the room thermostat so that it functions whenever the heating valve just begins to open. The fan is further interlocked by a pneumatic diverting relay to the supply fan of the closest general factory area supply unit. If the factory area unit is inoperative, the loading dock unit heater is inoperative, and if the factory area unit is operating, the loading dock unit heater may operate from its respective room thermostat.

Office Entrance Area Supply System

The entrance to the office area is electrically heated by a single system in that area. Electric resistance heat was used as the system is remotely located from any heating water piping and the system would have to operate at periods when the central system was inoperative.

The supply system is located above the ceiling in the entrance area and is composed of two components. The first component is the electric resistance heating coil. Recirculated air from the space is drawn over the coil to be heated. The second component is the centrifugal fan which moves the air through the system and discharges it into the entrance area.

The space thermostat is composed of a sensor located in the recirculated air and a pneumatic controller. The output from the controller operates three pressure switches in sequence which in turn actuate contactors supplying power to the electric resistance heating coil. A high limit thermostat in the heating coil interrupts the contactor control circuit if temperatures within the coil are excessive for safe operation. The contactor control circuit is also interruptive by the unit operational switch.

The fan is operated by an "on-off" switch located in the area served. In the "on" position, the fan operates and the coil operates subject to its control system. In the "off" position, the coil contactors are inoperative, and the fan is off unless operated by the purge thermostat. The purge thermostat is installed adjacent to the downstream face of the heating coil and will operate the fan regardless of switch position if excessive temperatures develop. Purge thermostat actuation occurs most frequently on system shutdown, when residual heat in the coil warms the air above the purge thermostat setting. The fan is restarted and operates long enough to purge the unit of the residual coil heat.

Cafeteria Area Supply System

The cafeteria area is air conditioned by a single multizone air supply system. The unit is called upon to perform two functions. The first function is to supply heated or cooled air to maintain proper environmental temperatures. The second function is to provide the proper amount of air for ventilation of the space.

The cafeteria supply system is located on a balcony in the factory area immediately adjacent to the cafeteria and is composed of five components. The first component is a pair of outside and recirculated air dampers. The second component is a centrifugal fan which moves air through the system. The third component is a heating coil which heats air passing through it to the hot plenum. The fourth component is a cooling coil which heats air passing through it to the cold plenum. The fifth component consists of three sets of hot plenum and cold plenum mixing dampers. These dampers mix heated and cooled air in varying proportions for delivery to the space as required by room conditions. A separate set of hot and cold plenum mixing dampers is provided for each of three zones within the space; the south perimeter, the west perimeter, and the north perimeter. Each zone is controlled by a separate room thermostat. This was done as the space is quite large and each perimeter has varying heat losses and heat gains due to orientation and unequal amounts of windows.

The outside air damper is under control of a ventilation time switch in the central panel and is allowed to open to a maximum position during programmed periods. The amount of opening, when programmed, is selected by a gradual acting pneumatic switch at the unit panel. When the ventilation time switch is in the unprogrammed position, positioning of the outside damper reverts to another gradual acting switch mounted in the central control panel. This is set for an amount of outside air equal to the minimum ventilation requirement. If the fan is off, an electrically operated pneumatic relay removes the air supply from the positive positioner of the damper operator and closes the outside damper.

Fan operation is proven by a differential pressure control measuring the pressure rise across the fan. When the fan is operating, a pilot at the central panel is energized.

The heating valve is of the normally open modulating type constructed and sized as previously described. The valve is modulated by a controller to provide a hot plenum temperature which varies from 120 degrees F at 10 degrees F outside to 80 degrees F at 70 degrees F outside on a straight line schedule. When the fan is stopped, a pneumatic relay closes the valve.

The cooling valve is of the normally closed modulating type constructed and sized as previously described. The valve is modulated by a controller to provide a cold plenum temperature which varies from 50 degrees F at 90 degrees F outside to 70 degrees F at 35 degrees F outside. An override of the plenum schedule is provided from the south perimeter room thermostat such that when the cold plenum damper is full open and the zone temperature continues to rise, the cooling coil valve is modulated open above the position necessary to satisfy the plenum schedule. When the fan stops, a pneumatic relay closes the cooling valve.

Sensors in the outside air, hot plenum, cold plenum, and mixed air transmit pneumatic signals to the plenum controllers, temperature indication gages in the unit mounting panel, and to the temperature indication gages in the central panel. Modulating room thermostats in the south and north zones modulate damper motors which operate their respective hot and cold plenum zone dampers in a cooperating manner. When the room is cold, the hot plenum damper is open and the cold plenum damper is closed. When the room is warm, the hot plenum damper is closed and the cold plenum damper is open. The west zone operates in the same manner except that a room temperature sensor and a controller are used in lieu of a room thermostat. This was done because no wall was available for mounting of a room thermostat. The sensor was mounted 18 inches directly below an air supply ceiling diffuser where room air being induced by the high velocity discharge would pass over it.

Industrial Relations Area Supply System

The industrial relations area is conditioned by a single multizone air supply system. The unit is called upon to maintain proper environmental temperatures in several zones and to provide outside air for ventilation.

The industrial relations area supply system is located on the factory area balcony adjacent to the cafeteria area supply system. The arrangement of system components is the same as the cafeteria system.

The outside air damper is under control of a separate ventilation time switch in the central panel and is allowed to open during programmed periods. If the supply fan is off, the damper closes. The hot and cold plenums are controlled to provide the same schedules and by the same type of control system utilized for the cafeteria system. Fan operation is proven and reported to the central panel as described for the cafeteria system. Temperature sensors in the outside air, hot plenum, cold plenum, and mixed air transmit pneumatic signals to their associated controllers and the temperature indication gages in the unit panel and central panel.

Pneumatic modulating room thermostats in each of four zones modulate their respective zone damper motors. One zone serving the secretaries' area is provided with an electric resistance booster coil. This coil is controlled by an electric room thermostat which operates a contactor on the power supply to the coil. The contactor is interlocked by a pressure switch to fan operation so that the coil is inoperative if the fan is off. An integral coil high limit is also arranged to prevent coil operation on excessive coil temperature.

Heat Treating Area Supply and Exhaust System

The heat treating area is conditioned by a system of eight supply fans introducing outside air for cooling, six exhaust fans removing warm air from the space, and two heat recovery fans which remove warm air from the space and either exhaust it or return it to mix with the outside air being introduced. The area is used for heat treating of manufactured parts and contains several large furnaces used in the process. Heat liberation from the equipment is great and the space requires cooling regardless of the outside temperature. The system was therefore designed for heat removal from the space at all times and no provision was made to supply heating. A further design requirement was that a slight negative pressure be maintained so that no air would move from the heat treating area into the general manufacturing area carrying odors and fumes. This area serves as the heat source for the central heat pump. Heat recovery coils are installed in the air leaving the heat recovery fans and controlled to afford the heat required by the system.

The system components are contained in a monitor running the length of the heat treating area roof. Supply distribution duct work was provided along each side of the area with grilles arranged to distribute air toward the center of the area.

Control of space temperature is accomplished by a room temperature sensor and a controller which operate a step control. The step control starts the supply fans in sequence on rising room temperature, operating as many as required to maintain the desired temperature. The exhaust fans and heat recovery fans are also operated in sequence so that supply and exhaust fans are operated in pairs. This action assures that the proper balance is maintained between air supply and air exhaust.

The heat recovery fans are the last fans to be actuated from the step control for the maintenance of space temperature. The heat recovery fans, however, are also operated whenever heating is required by the heat pump system. Since this operation could result in an unbalance of air supplied and exhausted, the heat recovery fans are provided with exhaust and recirculation dampers. If the fan operates above 60 degrees F outside temperature, the dampers are positioned to exhaust, and if the fan operation occurs below that point, the dampers position to recirculate the air to the space. This feature is provided as the supply fans do not operate in conjunction with the heat recovery fans when the heat recovery fans are operated from the heat pump system.

Excessive temperatures might be encountered in the return air duct due to furnace operation. A temperature sensor and a controller are arranged to override the space temperature control and supply an additional quantity of outside air to prevent the return air exceeding 120 degrees F. Each supply, exhaust, and heat recovery fan is equipped with a motorized damper and pneumatic actuator in the fan discharge to close when the fan is inoperative. This damper prevents air bypassing through the unit when it is off.

Manual "on-off-automatic" switches are provided for each fan to allow manual override of the automatic system. The switches are mounted on a control panel located on the wall of the heat treating area. The panel is provided with a graphic floor plan of the area and the switches are mounted on the plan in locations corresponding to their actual location. A pilot is located adjacent to each switch to enable the operators to easily determine which fans are operating. The pilots are of the low voltage transformer type selected to provide maximum lamp life. The panel is also equipped with temperature indicators reading room temperature, return air temperature, and two air supply temperatures at different locations in the supply duct.

Heat Source

As mentioned before, the heat source for the central heat pump system is the heat liberated by the furnaces in the heat treating area. When heating is required by the heat pump, the two heat recovery fans are started. Each fan is equipped with a chilled water coil in its discharge. The coil has a face damper controlling the amount of warm return air passing through it. A bypass damper is also provided which cooperates with the face damper so that constant air flow through the fan is maintained as the air flow through the chilled water coil is varied. The amount of air passed through the coil is controlled in a modulating manner by the heat requirement of the central heat pump system. The dampers are provided with a pneumatic damper motor which is equipped with a positive positioning relay to assure positive movement regardless of varying damper loads.

Heat Sink System

The central heat pump system heat sink is provided through the use of well water. When the cooling requirements of the heat pump system exceed the heating requirements, heat must be disposed of. This is accomplished by heating well water and wasting it. The well water is pumped by the well pump into the settling tank which is vented to atmosphere. The settling tank level is controlled by a modulating pressure control. The pressure control is arranged to sense level by connecting it to a bubble pipe into which compressed air is fed through a restriction. The lower end of the bubble pipe is located near the bottom of the tank so that the pressure in the bubble pipe is equal to the head of water in the tank. The output of the level control starts the well pump on low level and stops it on high level through a pressure switch.

The level control also modulates a six-inch control valve on the well pump discharge. This valve is controlled in such a manner that when the well pump starts, the valve is ten percent open and remains so until the tank rises to the level which stops the well pump. If the level drops below the level at which the well pump starts, the valve modulates toward the open position and reaches a full open position when the settling tank level is at the bottom of its operating span. This action is accomplished through the use of a pneumatic ratio relay and positioning switch. The well pump discharge valve is an industrial type double seated control valve with top and bottom guided plug and stainless steel trim. This material affords maximum operating life under the sandy well water conditions. The well pump is started with the discharge valve only ten percent open to prevent large flows on startup which would carryover excessive quantities of sand into the settling tank.

The settling tank is equipped with a high level and a low level float switch. These are set above and below the normal operating range of the tank level control system. The high level float switch stops the well pump regardless of other controls and lights an alarm pilot at the heat pump control panel. The low level float switch stops all pumps drawing water from the settling tank and lights a low level alarm pilot at the heat pump control panel.

Three pumps draw water from the settling tank. Two provide water to the manufacturing process. One supplies water to the central heat pump system for use as a heat sink. The pump is operated whenever a heat sink is required and the pump capacity is controlled by a modulating control valve on the well water waste.

Central Heat Pump System

The central heat pump is a centrifugal refrigeration compressor with a single water circuit chiller as an evaporator and a dual water circuit heat exchanger as a condenser. The evaporator chills the system chilled water for use in the air conditioning units or to be passed through the heat recovery coils as a source of heat for the system. The condenser heats system water in one circuit to provide heating water for use in the air conditioning units and well water in the other circuit to act as a heat sink for the system.

The main control concept is to regulate heat pump compressor capacity from the chilled water temperature leaving the evaporator and being supplied to the system. A thermostat in the heating water supply to the air conditioning units operates the heat source and heat sink controls. If the heating water falls in temperature, the compressor is not pumping enough heat and chilled water is circulated through the heat recovery coils to be warmed. This warmed water is then introduced into the evaporator raising the chilled water return temperature which causes the compressor to pump more heat to the condenser and raise the heating water temperature. If the heating water rises in temperature, the compressor is pumping more heat than can be utilized by the air conditioning system. Well water is then introduced into the second circuit of the condenser to remove the excess heat. The action of the heat source and heat sink systems is arranged so that they occur in sequence and not simultaneously.

The chilled water supply thermostat is a recording controlling instrument with a 12 inch chart. The control point of the instrument is reset by outside air temperature so that it is 42 degrees F at 90 degrees F outside and 50 degrees F at 35 degrees F outside. This reset is provided to reduce the thermal head against which the heat pump must operate during the period when heating is the predominate load. The outside signal is provided by a sensor and a controller. The chilled water instrument is provided with automatic reset to remove the droop of the control point from the set point due to varying load. The automatic reset feature allows the use of a wide proportional band for operational stability without the attendant droop. The automatic reset feature does cause a problem during system shutdown when the chilled water temperature rises well above the set point. This rise causes the automatic reset to force the set point downward to compensate for the high chilled water temperature. When operation is resumed, the heat pump operates attempting to reach the low set point. This results in excessively low chilled water temperature with safety shutdown of the unit from the freeze-up thermostat. To avoid this problem, a relay was provided which eliminates the automatic reset system when the heat pump is inoperative.

On heat pump start-up, the capacity is limited by a manual positioning switch to allow the heat pump time to chill the water in the piping system. A time delay relay is provided to switch the manual positioning switch out at the end of ten minutes and return control to the chilled water supply instrument. A time switch was also provided to limit the heat pump capacity during periods at the beginning and end of each shift when transient loads might be present. This action is accomplished by switching the manual positioning switch into the pneumatic control circuit as on system startup.

The capacity control signal to the heat pump was arranged to load the machine in a gradual slow manner and to unload the machine at as rapid a rate as the chilled water controlling instrument indicated. This was accomplished through the use of a capacity tank in the pneumatic control line immediately before its connection to the heat pump control console. A restriction was placed ahead of the tank so that as pressure was increased, a delay occurred from the time the controlling instrument increased its output until the capacity tank filled and the control signal equalled the instrument output. The fast unloading was provided by installing a check valve in parallel with the restriction to allow air from the tank to the controller to bypass the restriction.

The heat pump control console contains the safety controls furnished as standard equipment by the manufacturer which consist of an oil failure control, an evaporator low refrigerant pressure control, a condenser high refrigerant pressure control, a chilled water flow control, a chilled water low temperature control, and a current limiting control. All of these controls stop the heat pump when an unsafe condition is reached with the exception of the current limiting control. This device is connected to the motor power conductors through a current tap transformer and acts as a high limit for the amount of power consumed by the heat pump. The device limits the pneumatic control signal supplied to the capacity control mechanism operator when its setting is exceeded. The setting is variable with the adjustment located on the control console and expressed as a percentage of full load.

The actual capacity control mechanism is a set of vane dampers in the inlet to the centrifugal compressor. The vane dampers are operated by a pneumatic operator arranged to open the vanes to increase capacity or to close the vanes in a modulating manner and throttle the flow of refrigerant to decrease capacity. An elapsed time indicator is provided to record the number of hours of compressor motor operation for maintenance purposes.

The hot water supply temperature leaving the heat pump is controlled by a controller with a sensor in the heating line. The control point is reset from the outside air temperature to provide 110 degrees F heating supply at 50 degrees F outside air temperature with uniform graduation to 95 degrees F heating supply at 65 degrees F outside air temperature. The heating supply temperature is prevented from controlling at temperatures above 110 degrees F. The output of the heating supply controller next passes through a relay which is energized if the hot water pump is operating and functions to close the heat recovery coil if deenergized. From this relay the signal is transmitted to operate the heat recovery coil dampers if heating is required or to operate the well water valve to the condenser if heating is not required.

The well water valve signal from the temperature controller may be overriden by a signal from the condenser refrigerant pressure such that the signal requiring the greatest valve opening will be passed by a pneumatic selector relay. The selector relay output passes through a relay arranged to close the well water valve if the heat pump is inoperative. After leaving this relay the signal passes to a pneumatic ratio relay which selects the higher portion of the signal and expands it for use by the well water valve. A pressure switch connected to the well water valve signal starts the settling tank pump whenever the valve begins to open and stops the pump when well water condensing is no longer required.

The condensing system is augmented by two refrigerant condenser pressure controls. The first is set at 95 degrees F and operates when the hot water supply pump is inoperative. The second is set at 120 degrees F and operates when the hot water supply pump is operative. The higher condensing temperature is utilized to afford maximum heating water temperature during the winter. The lower temperature is used during the summer season so that the thermal head against which the heat pump must operate is reduced. The relay operated from the heating pump selects which signal is utilized to operate the well water valve.

A steam converter heat exchanger is provided in the heating supply after it leaves the heat pump. This converter provides the additional heat required for the system above the capacity of the heat pump. The converter discharge temperature is reset from the outside temperature by a controller, an outside sensor, and a sensor in the final heating water discharge to the system. An electrically operated pneumatic relay is provided to close the steam valve if the heating pump is off.

The various pumps associated with the heat pump system are interlocked to function only when the system is operative. The hot water pump is off above 70 degrees F outside temperature. The chilled water pump is off below 35 degrees F outside temperature. The heat recovery coil pump is off above 65 degrees F outside temperature. The outside signals are provided by pressure switches which are operated by a controller.

Two bypass valves are provided to assure minimum flow through the evaporator and condenser heating circuit. The air conditioning units use a variable amount of heating or cooling water depending on loads existing. If the load swings predominately one way, flow will be reduced to a small amount in the other circuit. This would prevent the sensing elements in the low flow circuit from functioning properly. A differential pressure controller is installed measuring the differential pressure across the evaporator. This control operates the chilled water bypass valve to maintain a minimum chilled water flow through the evaporator. A similar system is installed to operate the heating water bypass valve from differential pressure across the condenser heating circuit.

A control panel is provided in the immediate area of the heat pump. All controls not requiring mounting on equipment are mounted within the panel. A multi-color graphic flow diagram of the heat pump system is provided on the surface. Pilot lights are provided to show operation of the hot water supply pump, chilled water supply pump, heat recovery pump, well pump, settling tank pump, two process well water pumps, and heat pump. Temperature indication gages are provided to show temperatures of the well water supply, well water waste, system return, chilled water entering evaporator, chilled water supply, heating water entering condenser, heating water supply leaving condenser, heating water supply leaving converter, chilled water leaving heat recovery coil, and outside.

Operational Control Center

The operational control center for the system is a control panel located in the building engineer's office. The panel is two feet deep, seven feet high, and six feet wide. Switches, pilot lights, temperature indicators, and adjustments necessary for system analysis and operation are mounted on the panel.

The first function of the operational system is the starting and stopping of the various air conditioning systems. Six seven-day calendar dial time switches are connected to the six channels of a signal distribution panel with a circuit for each of the air conditioning units except for the industrial relations unit. The operator may set each of the time switches for a different time schedule. Any unit or number of units may then be programmed for any one of the programs by inserting a pin into the proper location in the signal distribution panel. Each air conditioning unit is also provided with a three-position "on-off-automatic" switch to allow the operator to manually override the time control system. A pilot is provided for each air conditioning unit to indicate unit operation. A second pilot is provided for each unit to indicate when the air filter media is exhausted. These switches and pilots are arranged on a graphic floor plan of the entire plant in a position corresponding to their actual location.

A similar system is provided for the industrial relations unit except that it is not connected to the signal distribution panel. A separate time switch is provided for operation of this unit. A second separate time switch is provided to control periods of maximum ventilation of the industrial relations unit.

The second function is the maintenance of minimum temperatures within the plant when operation is not programmed. This is accomplished by six room temperature sensors which operate in conjunction with six controllers. The controllers are set at 50 degrees F. One sensor is located in the industrial relations area, one in the lunch room, and four in the factory area. They restart units in the area immediately adjacent to their locations. Temperature indication of space temperatures at each sensor location is provided at the central panel. These are located on the plant diagram.

The third function of the operational system is to operate air conditioning units when process exhaust units are operated. A pilot is provided for each of the process exhaust units. These pilots are located on the plant diagram. Relays are provided to start the associated supply units when any process exhaust operates.

A separate time switch is provided to control periods of operation of the steam boiler. A pressure gage is also provided on the panel indicating boiler steam pressure.

Temperature indication is provided of the mixed air temperature, hot plenum temperature, and cold plenum temperature of the industrial relations and cafeteria units. The 12 temperatures indicated at the heat pump area control panel are repeated at the central panel. Heat treating area room and ceiling temperatures are also indicated at the central panel. The outside air temperature is indicated. Graphic flow diagrams are provided showing the heat pump flow, and the industrial relations and cafeteria units. The temperature indicators are located on the flow diagrams corresponding to their actual location. Pilots showing pump and heat pump compressor operation are also located on the flow diagram.

A three-pen temperature recorder is located on the central panel. It is arranged so that any temperatures indicated at the central panel may be recorded for analysis or trouble shooting. Up to three temperatures may be recorded simultaneously. The central panel is also provided with a clock to indicate time of day.

III. INSTALLATION

Control Mounting

Controls that must be mounted on the equipment served, such as damper motors and automatic valves, were installed thereon. Other control items such as relays, controllers, switches, pilots, and temperature indicators were mounted in panels. The panels are of two types, cabinets and cubicles.

The cabinets are 24 inches wide, 32 inches high, and 8 inches deep. The cabinets are constructed of steel with hinged doors. Locks were provided and keyed alike. A subpanel slightly smaller than the outside dimensions is bolted to the back of the panel. Equipment was mounted, wired, and piped with the subpanel removed. The subpanel was inserted into the panel for final external connection. The sides, top, and bottom of the cabinet are provided with knockouts to allow entrance of external connections at any point.

The cubicles are 24 inches wide, 24 inches deep, and 84 inches high. They are free standing rather than wall mounted. The cubicles are mounted on a four inch steel base. The base acts as a raceway between sections. The central operational control panel is made up of three cubicles. The heat pump panel is a single cubicle. The front of each cubicle is hinged and the rear is removable. Control equipment was mounted on the sides of the cubicles.

Items that do not require adjustment or access during normal operation were mounted inside the panels. Items requiring accessibility such as switches, pilots, and temperature indicators, were mounted in the door of the panels. Wiring and piping from these items were arranged in a manner to allow the door to be opened for service. Wiring was terminated at numbered terminal strips. Control piping inside the panels was done with color coded plastic tubing.

Control Piping

Control piping was done with two materials, copper tubing and plastic tubing. Copper tubing was used where lines were exposed and was of the hard drawn type. This exposed installation occurred in the immediate area of equipment where lines ran singly to different devices, such as damper motors and valves. Fittings used were primarily of the sweat type assembled with soft solder. Compression fittings were used at joints subject to disassembly such as at instrument connections.

Plastic tubing used in the installation was made of polyethylene. Different colors were used in a random manner to aid line identification. A metallic raceway with removable cover was installed along the length of the plant. One end terminated at the central operational panel and the other at the heat pump area panel. All lines possible were installed within the raceway. Fittings employed with the plastic tubing were of two types. The first was the compression type as utilized for the copper tubing. The second type of fitting used was of the barbed insert type. These fittings are especially manufactured for use with polyethylene tubing. They have an outside diameter equal to the inside diameter of the tubing. A sharp edged concentric ring protrudes from the surface. When the fitting is inserted into the tubing, the ring grips the tubing tightly.

Control Wiring

Wiring required for the control system utilized standard methods and materials to meet building codes. Every effort to reduce wiring requirements at the project was made. Panels were completely wired prior to delivery to the project. System design attempted to reduce field wiring to a minimum. This was done to render as economical an installation as possible.

IV. EVALUATION

Testing Procedure

The diagrammatic arrangement of the system and location of the measurement points is shown in Figure 2. Tests were performed at different times with the air conditioning system in actual operation serving the needs of the various areas. The test procedure started with recording the outside air, heating water supply, cooling water supply, and common return temperatures. The heating water flow was measured as was the cooling water flow of the four stations. The heating bypass was closed during the heating flow measurement.

The heating water supply was next stopped by closing a manual valve and stopping the heating supply pump. This prevented any heating water from being supplied to the air conditioning units and any heating water return from the units entering the common return. The common return became the cooling water return under this condition. The temperature of the cooling return was recorded. This procedure gives a valid value of cooling return temperature as it was performed rapidly and the space had enough thermal inertia to prevent changes in cooling water flow during the time used.

Flows were measured with a Rinco Engineering Company M50-50 flow meter and venturi stations installed in the piping. Temperatures were measured with Honeywell Incorporated LP914A

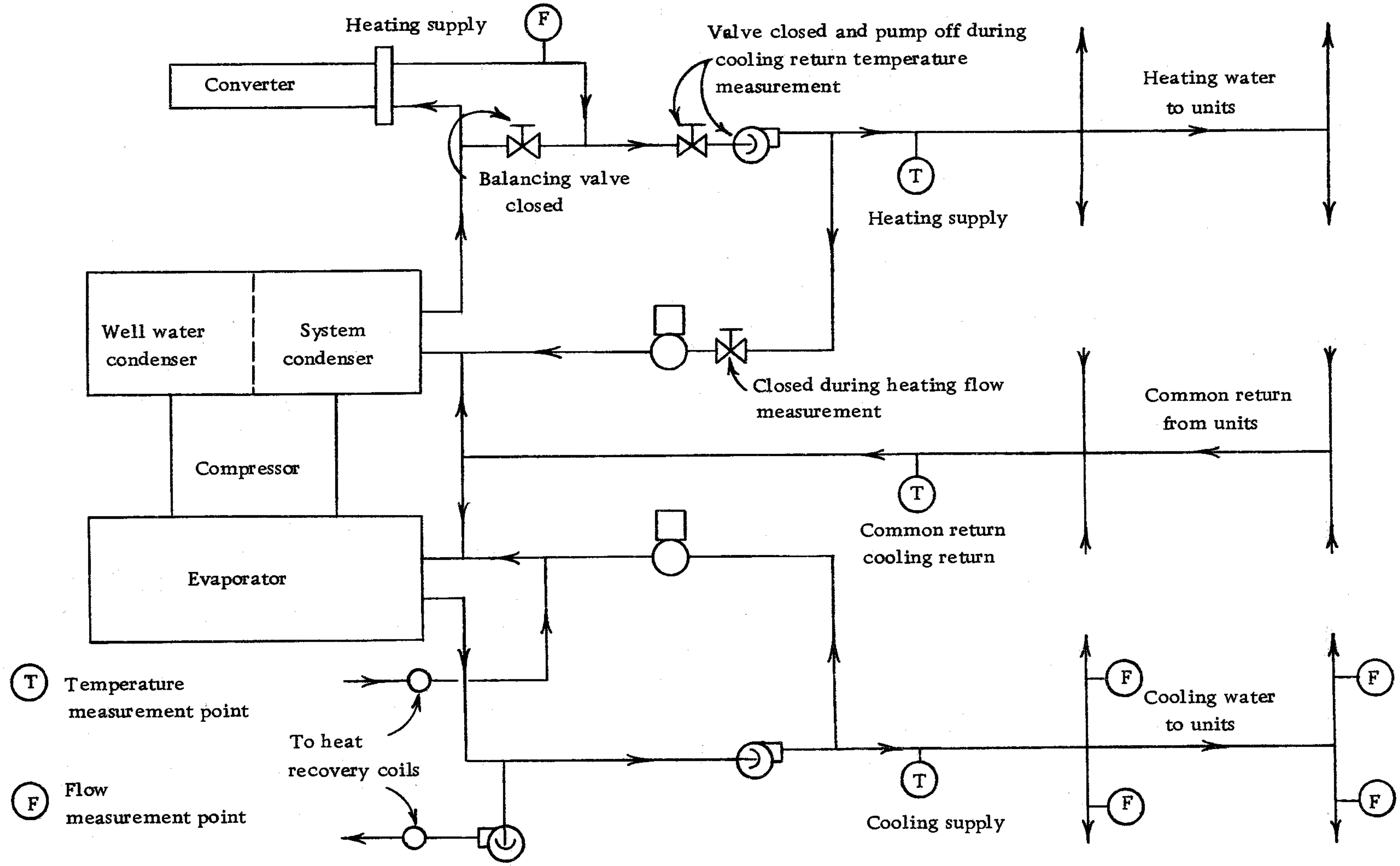

Figure 2. System arrangement and measurement point location.

temperature sensors.

Load and Penalty Calculations

With the observed information, the heating load, cooling load, heating return temperature, penalty to the cooling system, and penalty to the heating system may be calculated.

The heating water flow rate, FR_{hws}, was calculated as

$$FR_{hws} = gpm_{hws}\left(\frac{ft^3}{7.48\ gal}\right)\left(\frac{1}{v_{f-hws}}\right)\left(\frac{60\ min}{hour}\right) \qquad (1)$$

where

gpm_{hws} = observed heating water supply flow, gal/min

v_{f-hws} = specific volume of the heating supply, lb_m/ft^3.

FR_{hws} has the units of lb_m/hour.

The cooling water flow rate, FR_{cws}, was calculated as

$$FR_{cws} = gpm_{cws}\left(\frac{ft^3}{7.48\ gal}\right)\left(\frac{1}{v_{f-cws}}\right)\left(\frac{60\ min}{hour}\right) \qquad (2)$$

where

gpm_{cws} = observed cooling water supply flow, gal/min.

v_{f-cws} = specific volume of the cooling supply, lb_m/ft^3.

FR_{cws} has the units of lb_m/hour.

The common return flow rate, FR_{cr}, was obtained by adding the results of equation (1) and equation (2).

$$FR_{cr} = FR_{hws} + FR_{cws} \tag{3}$$

FR_{cr} has the units of lb_m/hour.

The total heat contained in the common return, Q_{cr}, was calculated as

$$Q_{cr} = FR_{cr}(h_{f - cr}) \tag{4}$$

where

$h_{f - cr}$ = enthalpy of the common return, Btu/lb_m

Q_{cr} has the units of Btu/hour.

The total heat contained in the cooling return, Q_{cwr}, was calculated as

$$Q_{cwr} = FR_{cws}(h_{f-cwr}) \tag{5}$$

where

$h_{f - cwr}$ = enthalpy of the cooling return, Btu/lb_m

Q_{cwr} has the units of Btu/hour.

The total heat contained in the heating return, Q_{hwr}, was obtained by subtracting the results of equation (5) from the results of equation (4).

$$Q_{hwr} = Q_{cr} - Q_{cwr} \tag{6}$$

Q_{hwr} has the units of Btu/hour.

The enthalpy of the heating water return, h_{f-hwr}, was calculated as

$$h_f = Q_{hwr}\left(\frac{1}{FR_{hws}}\right) \tag{7}$$

h_{f-hwr} has the units of Btu/lb_m.

The temperature of the heating water return, T_{hwr}, was obtained by finding the temperature which yields the calculated value of h_{f-hwr}.

The cooling load, L_c, represents the cooling utilized in the space and was calculated as

$$L_c = FR_{cws}(h_{f-cwr} - h_{f-cws}) \tag{8}$$

where

h_{f-cws} = enthalpy of the cooling supply, Btu/lb_m.

L_c has the units of Btu/hour.

The heating load, L_h, represents the heating utilized in the space and was calculated as

$$L_h = FR_{cws}(h_{f-hws} - h_{f-hwr}) \tag{9}$$

where

$$h_{f\text{ - }hws} = \text{enthalpy of the heating supply, Btu/lb}_m.$$

L_h has the units of Btu/hour.

The penalty to the cooling load, P_c, represents the additional cooling that the heat pump must perform because the common return is warmer than it would be if it were a cooling return only. P_c is calculated as

$$P_c = FR_{cws}(h_{f\text{ - }cr} - h_{f\text{ - }cwr}) \tag{10}$$

P_c has the units of Btu/hour.

The penalty to the heating load, P_h, represents the additional heating that the heat pump must perform because the common return is cooler than it would be if it were a heating return only. P_h is calculated as

$$P_h = FR_{hws}(h_{f\text{ - }hwr} - h_{f\text{ - }cr}) \tag{11}$$

P_h has the units of Btu/hour.

All values of v_f and h_f were obtained from Keenan and Keyes (4). The observed and calculated values of all data are shown in Table 1.

The values for the cooling load, L_c, and heating load, L_h, are plotted as a function of outside temperature in Figure 3. This shows that the heating and cooling loads are equal at 48 degrees F outside

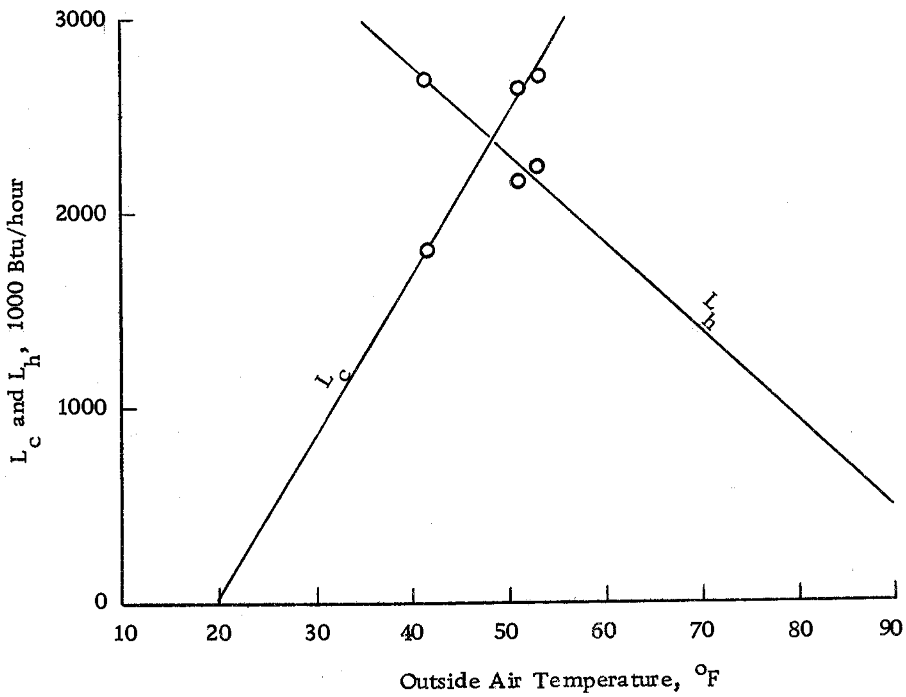

Figure 3. Heating and cooling loads.

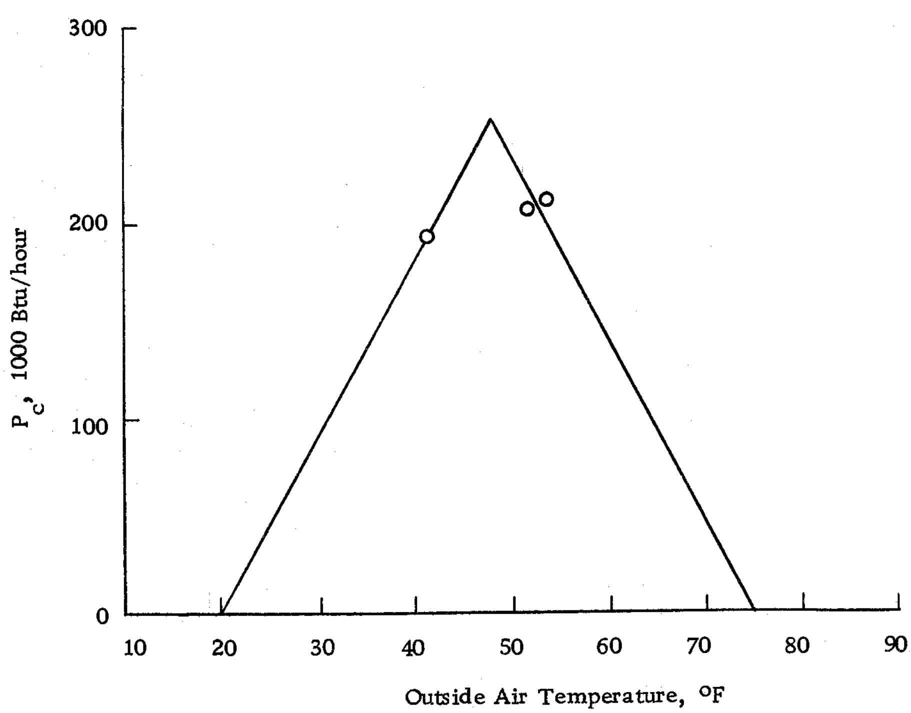

Figure 4. Penalty to the cooling system.

Table 1. Observed data and calculated results.

Observed Data				
Temperatures				
Outside air	°F	53.0	51.5	42.0
Heating supply	°F	102.0	102.0	110.0
Cooling supply	°F	42.0	42.0	41.0
Common return	°F	56.0	56.0	56.5
Cooling return	°F	55.0	55.0	55.0
Flows				
Heating supply	gal/min	108.0	104.0	109.0
Cooling supply	gal/min	419.0	417.0	261.0
Calculated Results				
FR_{hws}	1000 lb_m/hr	53.7	51.7	54.2
FR_{cws}	1000 lb_m/hr	210.0	208.5	131.0
FR_{cr}	1000 lb_m/hr	263.7	260.2	185.2
Q_{cr}	1000 Btu/hr	6,340.0	6,260.0	4,540.0
Q_{cwr}	1000 Btu/hr	4,850.0	4,810.0	3,020.0
Q_{hwr}	1000 Btu/hr	1,490.0	1,450.0	1,520.0
h_{f-hwr}	Btu/lb_m	27.8	28.1	28.1
T_{hwr}	°F	59.7	60.1	60.1
L_c	1000 Btu/hr	2,730.0	2,710.0	1,840.0
L_h	1000 Btu/hr	2,260.0	2,160.0	2,700.0
P_c	1000 Btu/hr	211.0	207.0	195.0
P_h	1000 Btu/hr	202.0	208.0	191.5

temperature and that the outside temperature must drop to 20 degrees before the need for cooling is not present. The curve of the heating load, L_h, appears to be excessively flat in that it indicates the need for heating above 75 degrees F outside air temperature which is the temperature maintained within the plant.

Figure 4 shows the curve of the penalty to the cooling system, P_c, for various outside air temperatures. This curve passes through the calculated values and crosses the absissa at 75 degrees F and 20 degrees F. The value of 75 degrees F was selected because this is the outside temperature at which the heating load would cease to exist. If there is no heating load present, there can be no heating water flow present. Without heating water flow, the common return becomes the cooling return and the penalty ceases to exist. The value of 20 degrees F was selected since the cooling load does not exist below this temperature.

Figure 5 shows the penalty to the heating system, P_h, for various outside air temperatures. This curve passes through the calculated values and crosses the absissa at 20 degrees F and 75 degrees F. The value of 20 degrees F was selected because this is the outside air temperature at which the cooling load, L_c, becomes zero. The value of 75 degrees F was selected since the heating load does not exist above this temperature.

Figure 6 is a composite of P_c above 48 degrees F outside

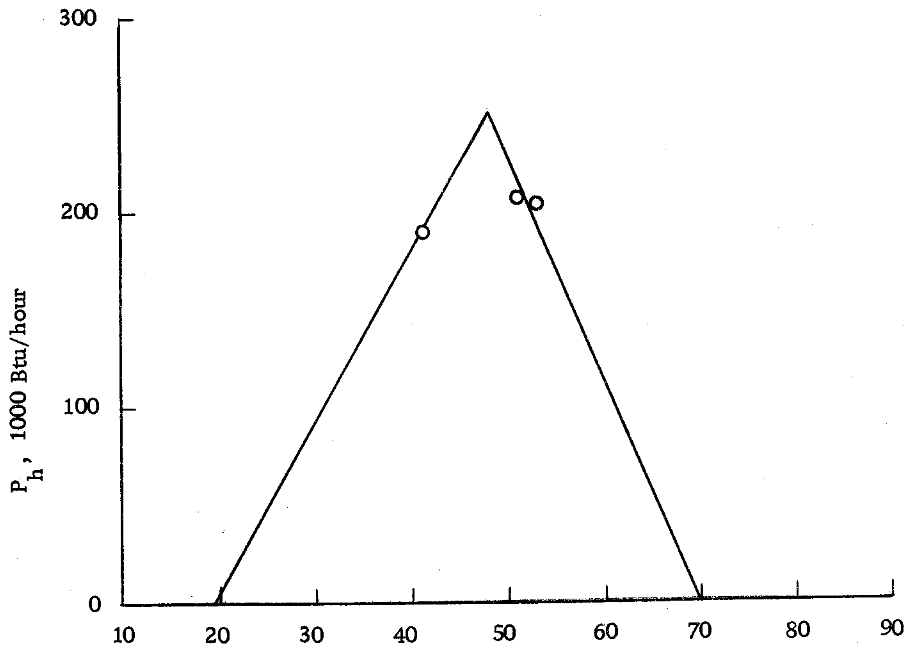

Figure 5. Penalty to the heating system.

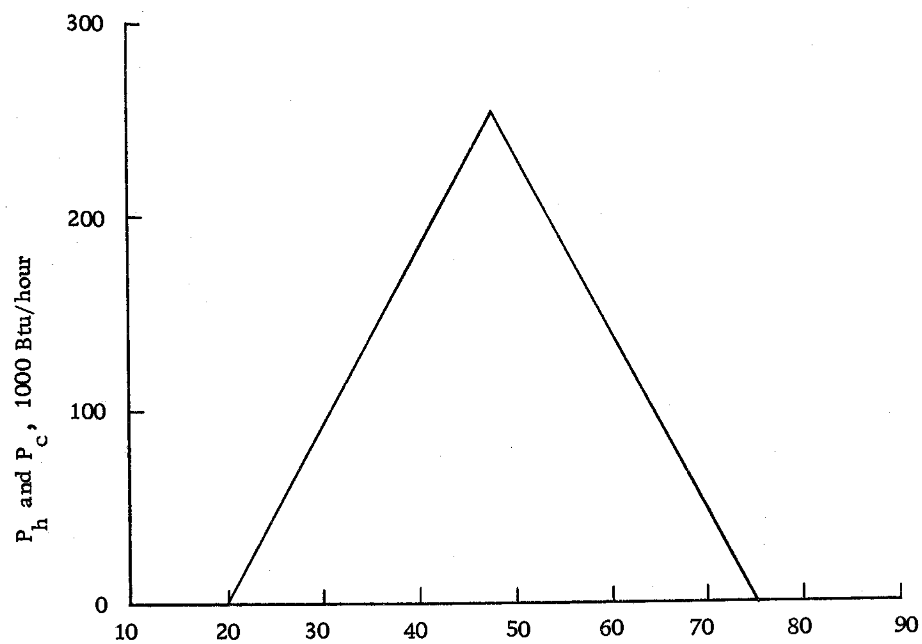

Figure 6. Penalty to the air conditioning system.

temperature from Figure 4 and P_h below 48 degrees F from Figure 5. The penalty is zero at 20 degrees F outside temperature, rises to a maximum value at 48 degrees F when the heating and cooling loads are equal and falls to zero at 75 degrees F. With this information it is possible to determine the annual penalty to the system when the frequency of occurrence of each outside air temperature is known.

Three-pipe System Owning Costs

The three-pipe distribution system is utilized by the air conditioning system. This system has a lower initial cost than the four-pipe system used for comparison. For the three-pipe system, only one return pipe is required and fewer control components are used. The system is shown in Figure 7. Since the four-pipe distribution system has a greater cost, the initial cost of the three-pipe system will be neglected and only the additional cost of the four-pipe system will be evaluated.

Four-pipe System Owning Costs

The four-pipe distribution system is used for comparison with the three-pipe system. It differs in that separate returns are provided for cooling and heating flow. The addition of a second return does result in extra initial cost which will be considered in evaluating owning costs. The four-pipe system is shown in Figure 8. The

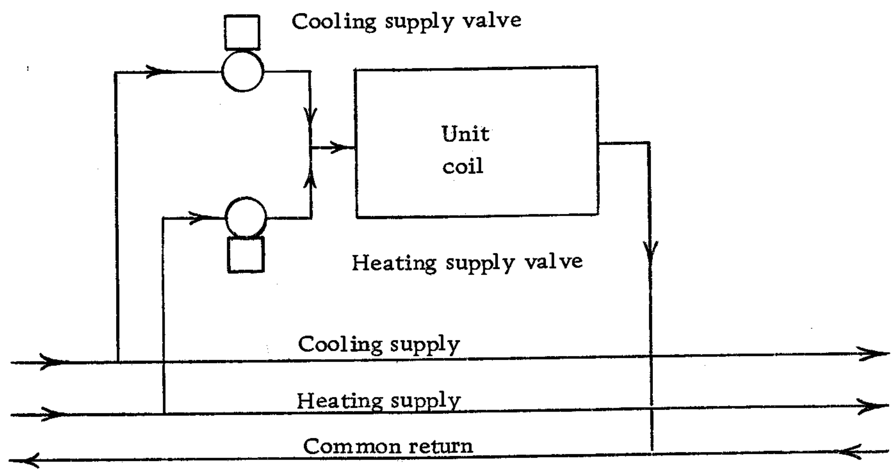

Figure 7. General arrangement of a three-pipe unit.

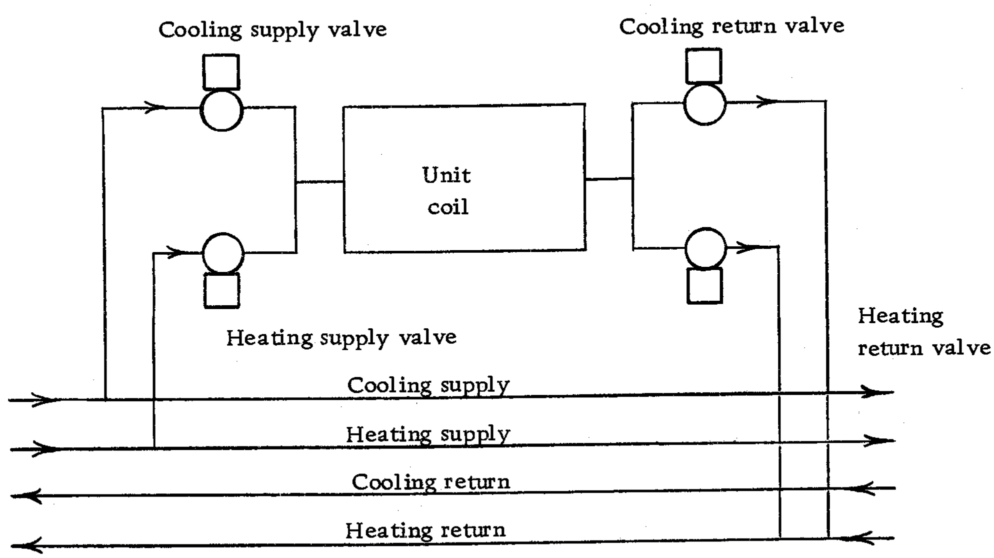

Figure 8. General arrangement of a four-pipe unit.

system also requires the addition of automatic control valves to isolate the return lines. The valves are arranged so that they operate in the dead band when neither supply valve is open. The valves cooperate such that when one is open the other is closed. The return valves are positive acting and are operated by a positive acting pneumatic relay. The relay is actuated by the signal operating the supply valves.

The additional return line required for the four pipe system was selected on the same sizing basis as used for the heating supply line. The heating supply is smaller than the cooling supply as installed. The existing return line is sized to handle the larger cooling flow. If an additional return line was added, it would only have to be sized to handle the smaller heating flow. The piping lengths were selected to match those existing in the distribution system. The piping material was standard schedule 40 black iron pipe with three-fourths inch fiberglass insulation installed with a canvas jacket. Unit costs for piping represent current prices existing in the Portland, Oregon area and were obtained from the engineering staff of the architects for the project. The additional return piping cost was calculated as

Piping Size Inches	Length Feet	Unit Cost Dollars/Foot	Cost Dollars
3/4	248	$3.85	$ 970.
1	304	4.25	1290.
1-1/4	264	4.40	1160.
1-1/2	140	4.70	660.
2	288	4.90	1410.
3	32	5.70	182.
4	280	6.70	1880.
		Additional Piping Cost	$7552.

The additional control cost was estimated to be $2905. The control cost was obtained from the manufacturer and an allowance of ten percent was added to cover the mechanical contractor's overhead and profit.

Normal trade channels are for the mechanical contractor to act as a subcontractor to the prime contractor. Typical bidding practice would result in approximately a ten percent addition by the prime contractor to cover his overhead and profit. The total initial cost for the four-pipe system above the three-pipe system basis was calculated as follows

Additional Piping Cost	\$7552.
Additional Control Cost	2905.
Prime Contractor's Cost	\$10457.
Prime Contractor's Overhead and Profit	1046.
Initial Cost to Owner	\$11503.

Having obtained the initial cost of the additional installation, yearly owning costs are computed as suggested by the American Society of Heating, Refrigerating and Air Conditioning Engineers (1). The annual fixed costs cover charges for depreciation, interest, and taxes. The estimated life of the equipment is 20 years. If a straight line method of depreciation is used, the annual depreciation charge may be calculated as follows

$$\text{Annual Depreciation Cost} = \frac{\$11,503}{20 \text{ years}}$$

$$= \$575$$

Interest charges cover the cost to the owner of capital invested in the additional components. An interest rate of six percent was assumed. The annual interest charge was calculated as follows

$$\text{Annual Interest Cost} = \frac{Y+1}{2Y} \text{(interest rate) (initial cost)}$$

$$= \$362.$$

where

Y = depreciation period.

Tax information was obtained from the Clackamas County tax assessor. Taxes are assessed at the rate of 91. 9 mills per dollar of assessed valuation in the area of the installation. Assessed valuation is equal to 25 percent of the true market value. True market value of a new installation is equal to the initial cost. Tax cost is calculated as follows

$$\text{Annual tax cost} = \$11,503.\ (0.25)(0.0919)$$

$$= \$\quad 264.$$

The total annual owning cost of the four-pipe system above that of the three-pipe system is the sum of previously calculated costs.

Annual Depreciation Cost	$575.
Annual Interest Cost	362.
Annual Tax Cost	264.
Annual Owning Cost	$1201.

Four-pipe System Operating Cost

In the four-pipe system, complete isolation is provided between the heating and cooling flows. If a supply unit is using heating, the cooling supply and return valves are closed. If a supply unit is using cooling, the heating supply and return values are closed. There is no mixing of heating and cooling with attendant operating cost penalty. The four-pipe system will therefore cost less to operate and its operating cost will be neglected. Only the additional operating cost of the three-pipe system will be evaluated.

Three-pipe System Operating Cost

When the air conditioning system is utilizing both heating and cooling, a penalty exists due to the mixture of heating water return and cooling water return. This penalty has been previously evaluated as a function of outside air temperature. Ommission of a factor for solar gains differs from the analysis made by Blossom (2) of apartment buildings. Since the manufacturing area is without windows and their attendant high solar gains, the neglect of solar conditions is justified. The American Society of Heating, Refrigerating, and Air Conditioning Engineers (1) neglects solar radiation in their estimations of operating cost.

The frequency of occurrence of outside air temperature was obtained from the United States Air Force (5). The data is based upon an observation period of 18 years and consists of average temperatures existing in each three-hour period of the day on a monthy basis. Using the outside temperature for each period, the value of the system penalty was obtained for each period. The outside temperatures and penalties are shown in Table 2. The average penalty was calculated as the arithmetic average of the penalties for each period. This average was used as the plant operates on a 24-hour

Table 2. Outside air temperature and penalty occurrence.

		Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.
12:00 AM to 3:00 AM	Average Temperature, ^{o}F	36.6	39.8	41.6	45.9	50.6	56.0	59.5	59.9	56.0	49.7	43.0	39.6
	Penalty, 1000 Btu/hour	146	174	190	228	226	173	141	138	173	231	202	172
8:00 AM to 6:00 AM	Average Temperature, ^{o}F	36.0	39.0	40.2	44.1	48.7	54.1	57.1	57.6	53.8	48.5	42.3	39.1
	Penalty, 1000 Btu/hour	140	168	177	212	240	181	163	159	193	241	196	168
6:00 AM to 9:00 AM	Average Temperature, ^{o}F	35.7	38.7	40.4	45.7	51.5	56.8	59.7	59.3	55.0	48.9	42.2	38.9
	Penalty, 1000 Btu/hour	138	164	179	226	214	168	140	143	182	238	195	166
9:00 AM to 12:00 PM	Average Temperature, ^{o}F	38.0	42.2	45.2	51.8	57.4	62.5	66.6	66.0	62.8	54.4	45.9	41.0
	Penalty, 1000 Btu/hour	158	195	222	212	161	114	77	82	111	188	228	185
12:00 PM to 3:00 PM	Average Temperature, ^{o}F	41.0	46.2	49.7	56.8	62.6	67.8	73.8	73.1	70.2	60.1	49.6	43.9
	Penalty, 1000 Btu/hour	185	231	231	167	113	66	11	17	44	136	232	210
3:00 PM to 6:00 PM	Average Temperature, ^{o}F	40.6	43.1	50.4	57.7	63.8	69.3	76.4	75.5	71.5	60.2	48.8	43.1
	Penalty, 1000 Btu/hour	181	203	224	158	102	52	0	0	32	135	239	203
6:00 PM to 9:00 PM	Average Temperature, ^{o}F	38.3	41.2	46.5	53.1	59.4	65.2	71.4	70.0	64.9	54.9	45.6	41.1
	Penalty, 1000 Btu/hour	161	186	233	200	142	89	33	46	92	183	225	185
9:00 PM to 12:00 AM	Average Temperature, ^{o}F	37.3	42.1	43.7	48.7	53.9	59.3	63.9	63.5	59.3	51.8	44.1	40.2
	Penalty, 1000 Btu/hour	152	194	208	240	192	143	101	105	144	212	212	178
Average Penalty, P_{ave}		162,000 Btu/hour											

basis. The total operating penalty, TOP, was calculated as

$$TOP = P_{ave}\left(\frac{52\ \text{weeks}}{\text{year}}\right)\left(\frac{5\ \text{days}}{\text{week}}\right)\left(\frac{24\ \text{hours}}{\text{day}}\right)$$

$$= 1,010,000,000\ \text{Btu/year}$$

where

P_{ave} = the average operating penalty, Btu/hour

The energy required to overcome the penalty was obtained by reference to Carrier Corporation (3). The power requirement to produce 1000 Btu/hour of heat pump output is 0.0765 kw. The energy charge at the plant was obtained from the serving utility. The charge was \$0.00487/kwhr and was obtained by dividing the total charge by the total power consumption. The charge therefore contains demand costs as well as consumption costs. The annual cost to overcome the three-pipe system operating penalty was calculated as

$$\text{Annual operating cost} = \frac{1,010,000,000\ \text{Btu}}{\text{year}}\left(\frac{0.0765\ \text{kw}}{1000\ \text{Btu/hr}}\right)\left(\frac{\$0.00487}{\text{kw hr}}\right)$$

$$= \$376.$$

Comparison and Conclusions

Comparison of the three-pipe system installed and the four-pipe system considered as an alterante was made by evaluating the owning and operating costs of each system. The costs calculated do not represent the entire costs of the systems. They represent the increased cost of the cited system above the other system. The comparison was made as follows

	Three-pipe system	Four-pipe system
Annual owning cost	$ 0.	$1201.
Annual operating cost	396.	0.
Annual owning and operating cost	$396.	$1201.

The choice of a three-pipe system for use in the project examined provided the least total owning and operating cost to the owner. The selection of this system represented sound engineering cost judgment by the designing engineers.

The three-pipe system is penalized by increased operating cost due to mixing of the heating return and cooling return in the common return pipe. This type of system will be less costly to operate if the penalty due to mixing is reduced. If the air conditioned space served by the system has the same load requirement in all areas, the penalty will approach a minimum value. Under this loading, heating would decrease to almost zero before cooling would begin to be required. The three-pipe system would therefore exhibit the least operating cost penalty when all areas served by it were of the same load characteristic.

The operating cost penalty is minimized in a heat pump system as the penalty to the lesser load need not be considered. If the system utilized a separate source for heating and a separate source for cooling, the operating cost penalties would be additive at all loads. Energy would not be available to the lesser load unless it were produced at an additional operating cost in a separate boiler or chiller.

The operating cost penalty would be minimized by selecting final heat transfer elements of the system with as high a cooling water temperature rise and heating water temperature drop as possible. This will allow the system to supply the maximum heating to the space with the minimum heating water flow rate, FR_{hws}. Likewise, maximum cooling may be provided with the minimum cooling water flow rate, FR_{cws}. A higher water temperature drop would also cause the heating return water to enter the common return at a lower temperature and a lower enthalpy, h_{f-hwr}. On a similar basis, the enthalpy of the cooling return, h_{f-cwr} would be at a higher value. This would cause the penalty to the cooling load, P_c, and the penalty to the heating load, P_h, to approach a minimum value according to equations (10) and (11).

Any factors that would reduce the four-pipe system's initial cost would reduce the economic advantage of the three-pipe system. Shorter piping lengths would accomplish this.

A reduction in the control cost would also lessen the economical advantage of the three-pipe system. Control cost is primarily governed by the number of instruments and valves provided. Fewer and larger units would reduce the three-pipe system advantage.

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