

Feb. 10, 1959

W. J. CALDWELL
METHOD AND APPARATUS FOR PRESSURIZED SUPPLY
AND HIGH VELOCITY AIR CONTROL

2,872,858

Filed Sept. 28, 1950

7 Sheets-Sheet 1

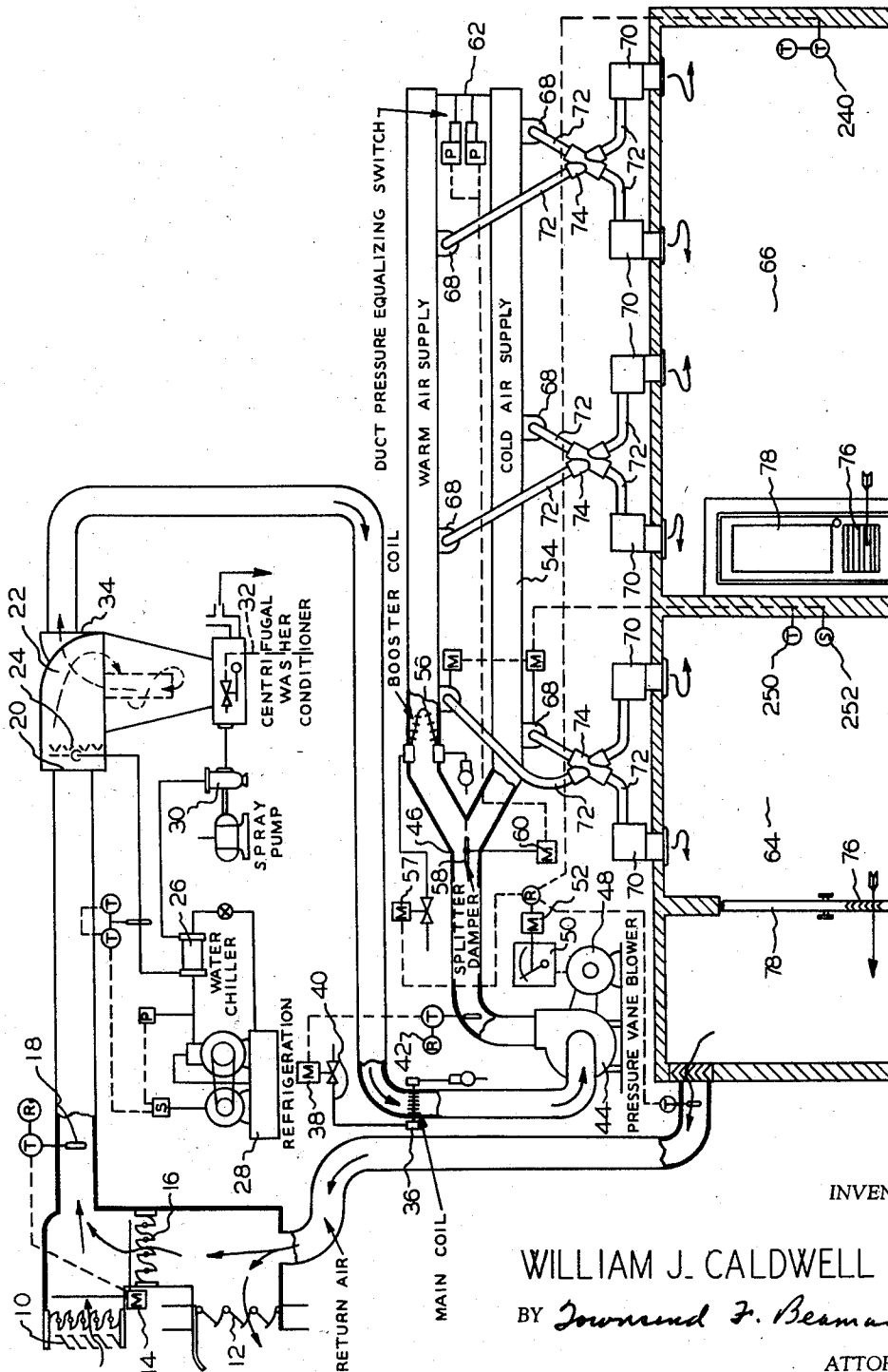


FIG. 1.

INVENTOR

WILLIAM J. CALDWELL

BY *Howard F. Beaman*

ATTORNEY

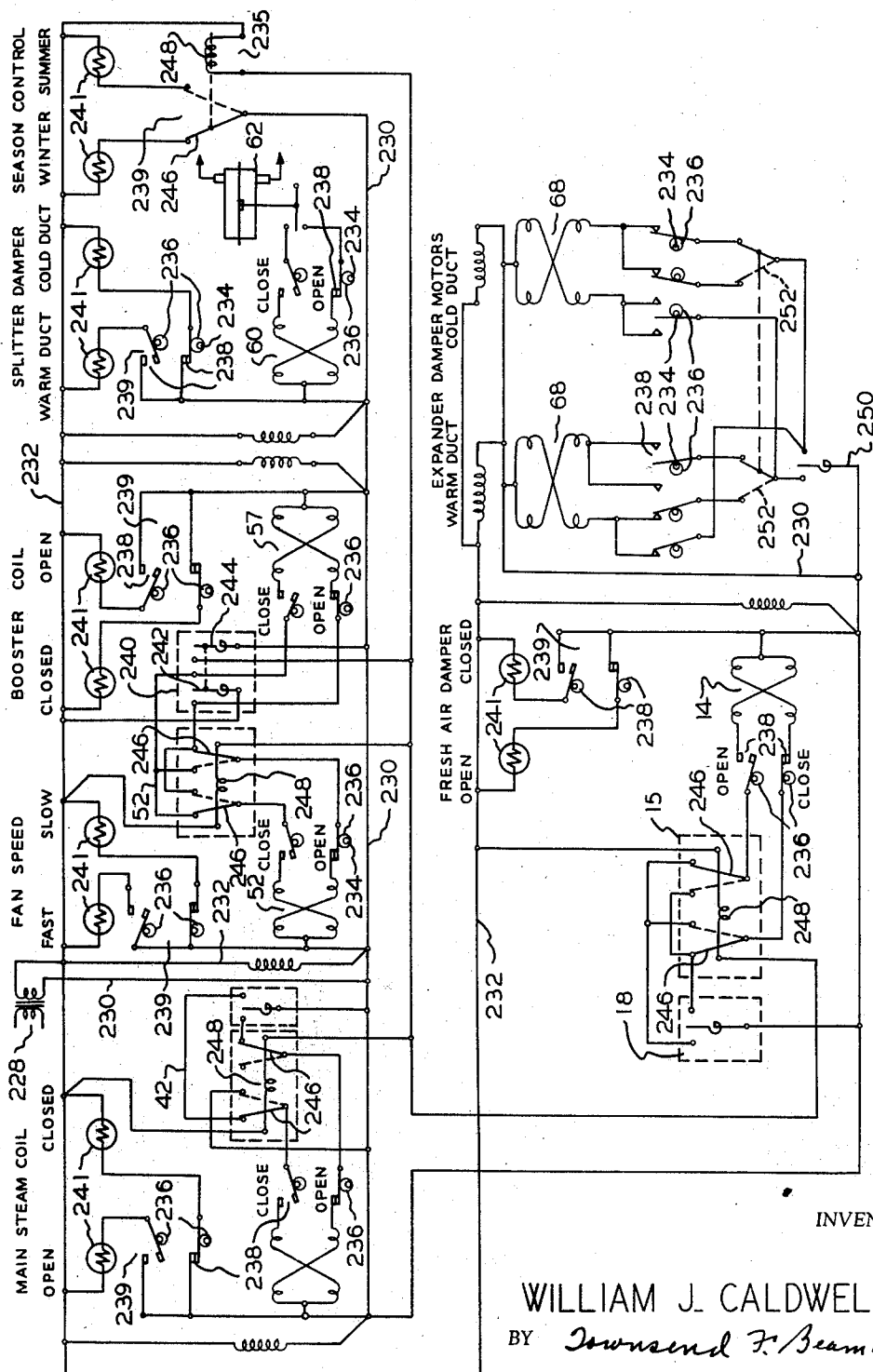
Feb. 10, 1959

W. J. CALDWELL
METHOD AND APPARATUS FOR PRESSURIZED SUPPLY
AND HIGH VELOCITY AIR CONTROL

2,872,858

Filed Sept. 28, 1950

7 Sheets-Sheet 2



FIFTH

INVENTOR:

WILLIAM J. CALDWELL
BY *Townsend F. Beaman*
ATTORNEY

Feb. 10, 1959

W. J. CALDWELL
METHOD AND APPARATUS FOR PRESSURIZED SUPPLY
AND HIGH VELOCITY AIR CONTROL

2,872,858

Filed Sept. 28, 1950

7 Sheets-Sheet 3

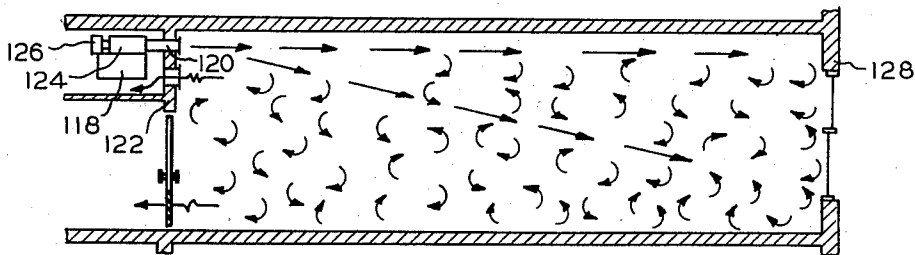


FIG. III.

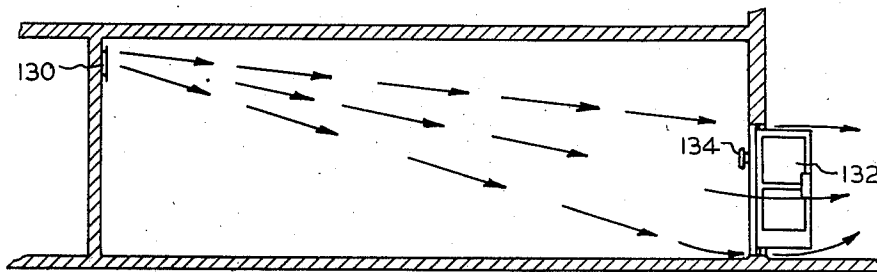


FIG. IV.

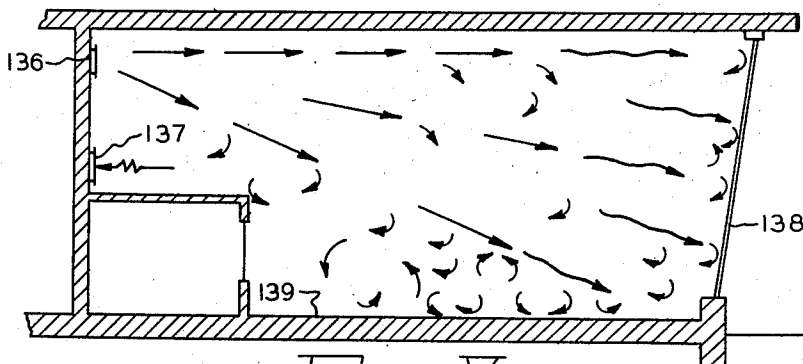


FIG. V.

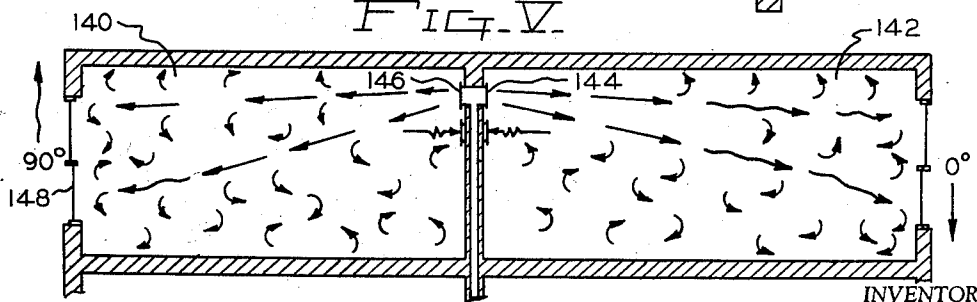


FIG. VI.

WILLIAM J. CALDWELL

BY

Lowmnd F. Beamau
ATTORNEY

Feb. 10, 1959

W. J. CALDWELL
METHOD AND APPARATUS FOR PRESSURIZED SUPPLY
AND HIGH VELOCITY AIR CONTROL

2,872,858

Filed Sept. 28, 1950

7 Sheets-Sheet 4

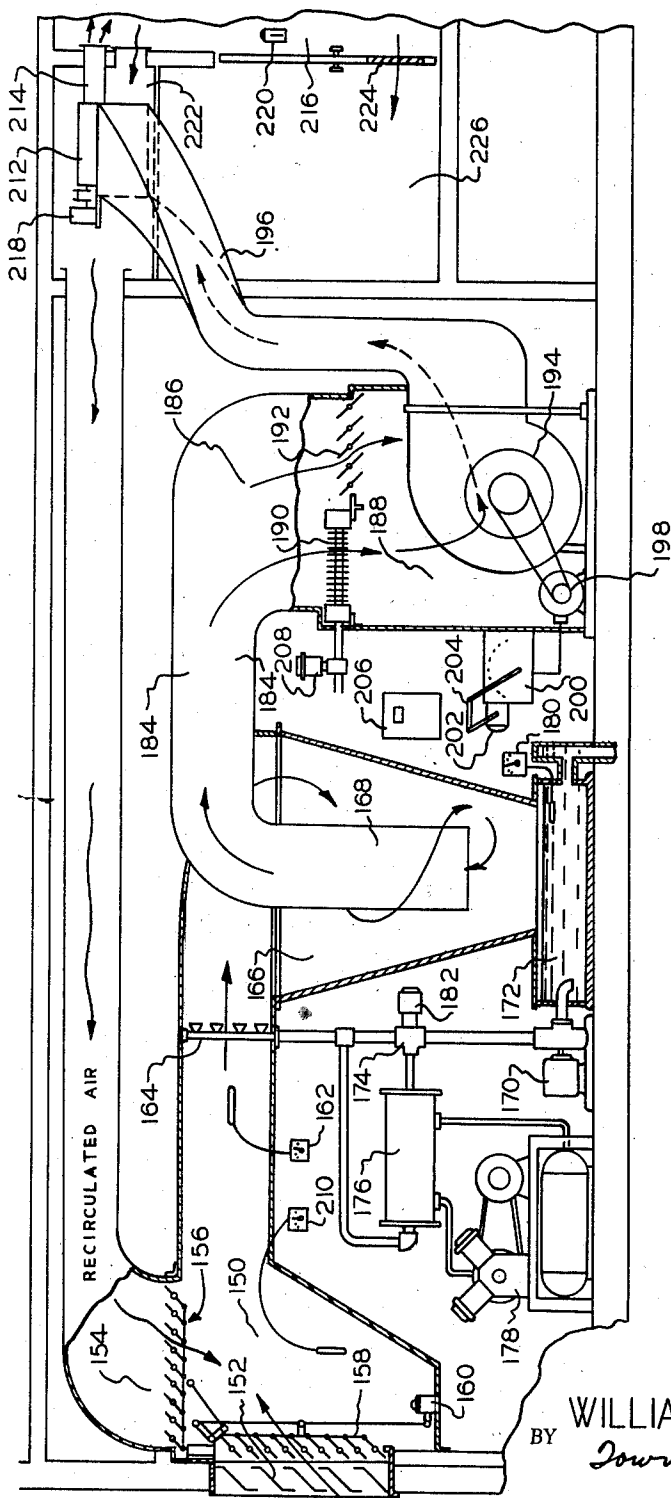


FIG. VII-

INVENTOR

BY WILLIAM J. CALDWELL
Downs and Beam
ATTORNEY

W. J. CALDWELL
METHOD AND APPARATUS FOR PRESSURIZED SUPPLY
AND HIGH VELOCITY AIR CONTROL

7 Sheets-Sheet 5

Filed Sept. 28, 1950



WILLIAM J. CALDWELL
BY *Townsend F. Beaman*

ATTORNEY

Feb. 10, 1959

W. J. CALDWELL
METHOD AND APPARATUS FOR PRESSURIZED SUPPLY
AND HIGH VELOCITY AIR CONTROL

2,872,858

Filed Sept. 28, 1950

7 Sheets-Sheet 6

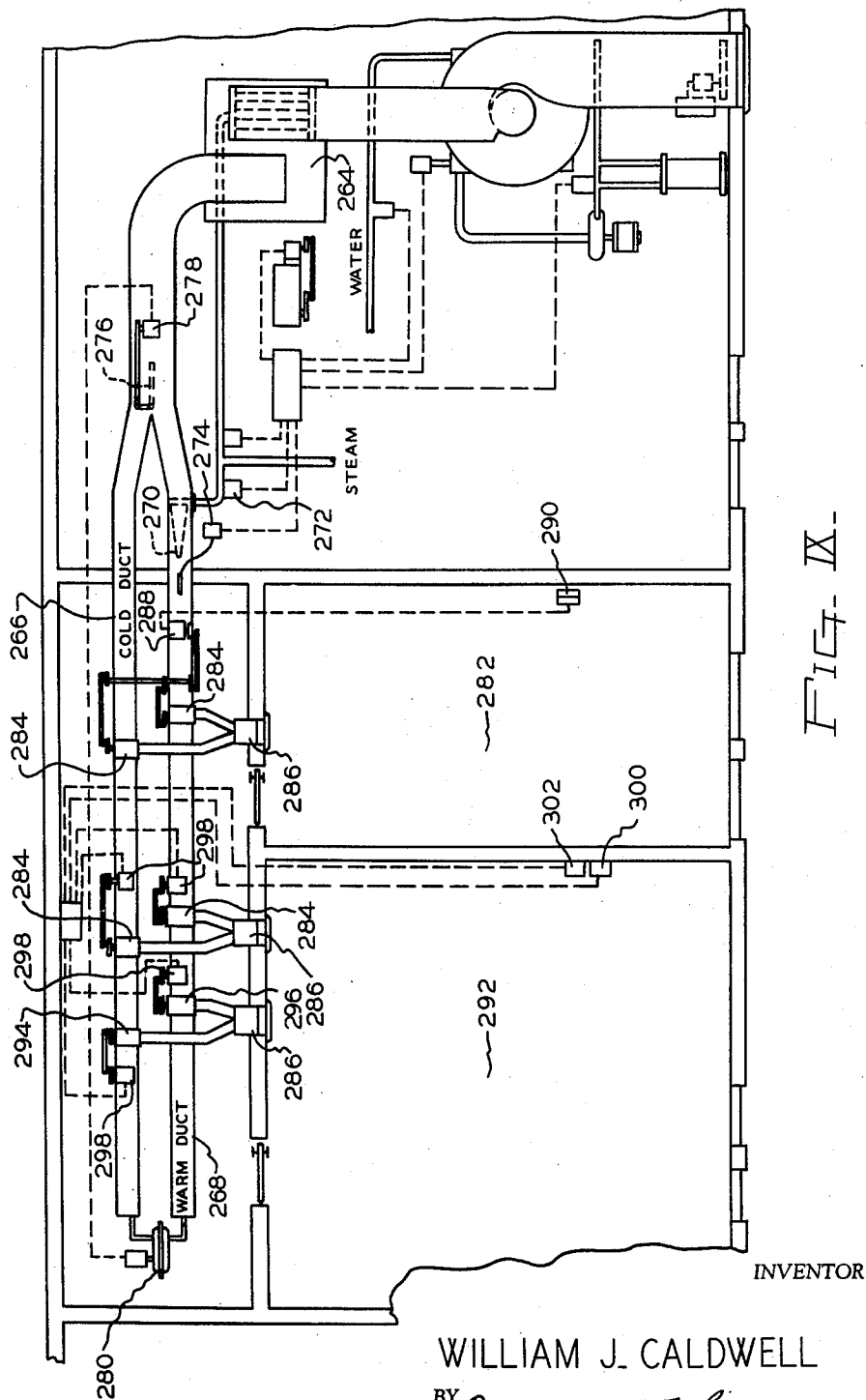


FIG. IX.

Feb. 10, 1959

W. J. CALDWELL
METHOD AND APPARATUS FOR PRESSURIZED SUPPLY
AND HIGH VELOCITY AIR CONTROL

2,872,858

Filed Sept. 28, 1950

7 Sheets-Sheet 7

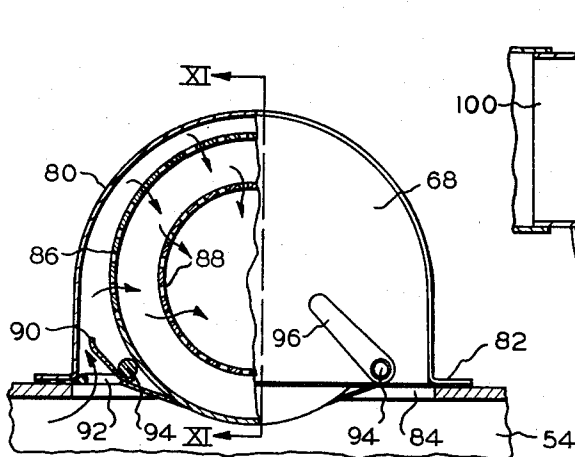


FIG. X.

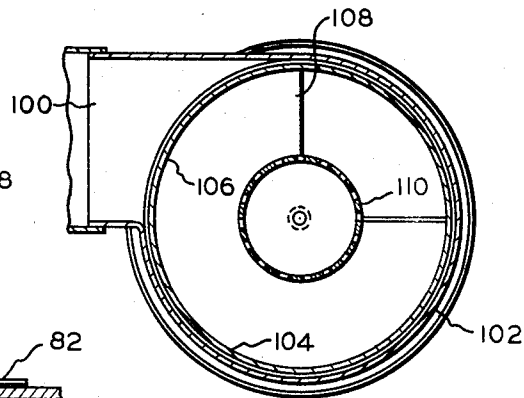


FIG. XIII.

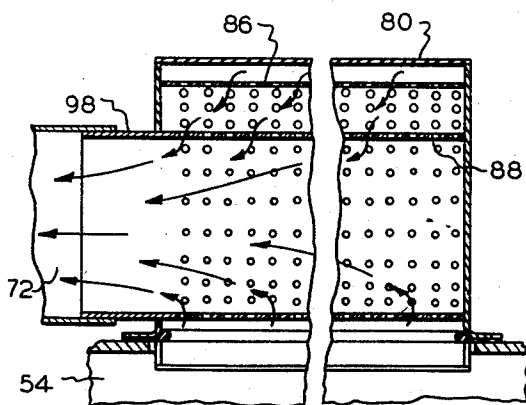


FIG. XI.

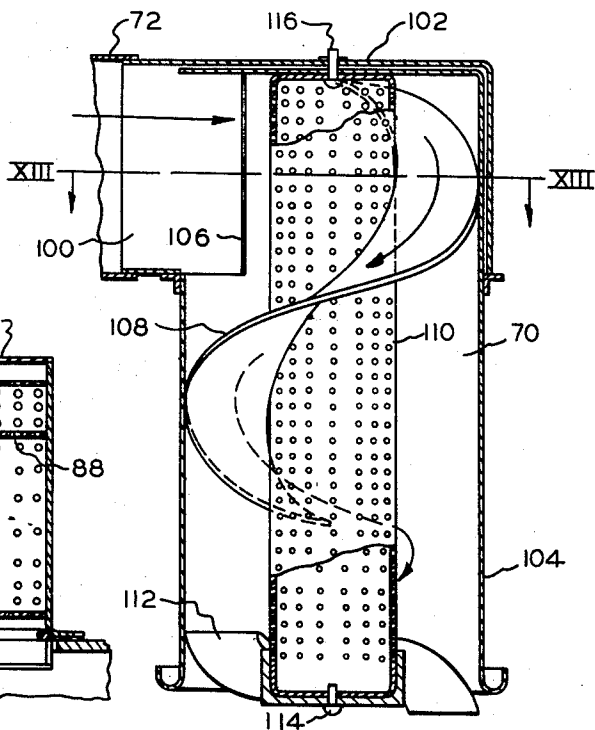


FIG. XII.

INVENTOR

WILLIAM J. CALDWELL

BY *James F. Beaman*

ATTORNEY

1

2,872,858

METHOD AND APPARATUS FOR PRESSURIZED SUPPLY AND HIGH VELOCITY AIR CONTROL

William J. Caldwell, Independence, Mo., assignor to
Townsend F. Beaman, Jackson, Mich.

Application September 28, 1950, Serial No. 187,258

11 Claims. (Cl. 98—33)

The present invention relates to improvements in air control systems for air conditioning, screening and blanketing and related purposes. In general, the invention is concerned with an improved pressurized supply, high velocity, air control system and improvements in apparatus used in the system.

In air conditioning of rooms and buildings it is the usual practice to introduce large masses of slow moving air through numerous outlets. With this method the thermal updrafts and downdrafts from temperature differentials may be the dominant forces and cause temperature stratification. Also, all doors and windows and other openings become, under this method, points of air infiltration which necessitate the employment of vestibules, revolving doors, uneconomical location of inlets and other remedial structures and equipment to combat the undesirable effects of such infiltration.

The system, according to the present invention, employs high velocity air streams to impart to the room or building air mass a substantial amount of kinetic energy. This causes the mass to be in a state of turbulence, in which all the air is in commotion, but without any pronounced directional effect. Also, the energy input of the high velocity air streams is substantially greater than any resulting thermals, thus avoiding appreciable stratification even in rooms of multistory heights.

With our improved system, as the temperature at all levels and areas is substantially uniform, there is no need for carefully locating return air openings near windows and doors in order to employ the return air motion as a means of withdrawing cold air from the occupied spaces near these normal sources of cold air accumulation. In contrast, it only becomes necessary to return a volume of air to the conditioning apparatus for reconditioning which is roughly equivalent to the amount supplied to the rooms. This permits wide selection in locating the return air outlets and makes it possible to take advantage of the economies of being able to place the outlets near the ceiling and close to the main return air path.

As a component in our improved system we employ an efficient air pressure producing device which makes it possible to subject the building to sufficient pressure to offset normal infiltration and substitute an exhaustion of air from the interior of the building. This eliminates cold drafts near windows and doors originating from the usual infiltration of outside air and simultaneously keeps out air borne dirt. An amplification of this principle may be effectively applied to building entrances, either to permit elimination of vestibules or offset drafts, where heavy traffic keeps doors open more or less continuously. In entrances where the normal wind pressure would overcome the normal pressure within the building, our system, in one of its forms, contemplates an automatic regulation of the velocity, or pressure, or both, of the air being directed toward the entrance to overcome the exterior condition striking the entrance.

One of the outstanding advantages of our system of

2

air control resides in the improvements in condensation control. In air conditioning systems of conventional design, the walls and window surfaces become areas of least air movement. This may result in objectionable condensation on windows where satisfactory comfort humidities are attempted. With our system, under high velocity application, the walls and window areas become the points of maximum air movement.

Another advantage of our improved system, due to the high air mass activity, is the more effective conduction of air temperature to encountered surfaces. Thus, cold floors may be overcome without resorting to panel heating which is an effective solution to the problem but does not lend itself to cooling in the summer.

A further advantage of high velocity system, as herein claimed, over standard low velocity systems, is concerned with the ability to perform, with equal efficacy, heating in the winter and cooling in the summer with the same system and apparatus. In the low velocity system, for heating, in order to prevent cold spots at perimeter areas from natural thermals and stratification, the hot air should be introduced at baseboard level. This is not the proper place to introduce cool air for cooling at low velocity. With our system the kinetic energy of the entering air produces such rapid and thorough diffusion that the same system is equally effective for both heating and cooling.

The apparatus required for our improved system of air conditioning includes the use of both standard and special design. Entering air is first passed through a centrifugal air washer-conditioner and thence to a pressure vane blower. From the blower, the air is preferably carried in specially designed pressure duct work to expander silencer units from which it is discharged through the air control nozzles into occupied areas being conditioned. Standard heating, refrigerating and control devices complete the system.

It is to be understood that the present invention is considered an improvement over the system and apparatus disclosed in the Caldwell Patent No. 2,053,391, granted September 8, 1936.

Thus the object of the present invention is to provide an improved system of high velocity air control and improved apparatus for carrying out the system; the combination, construction and arrangement of parts embodying specific objects and advantages which will more fully appear from the following specification and the appended claims.

In the drawings:

Fig. I is a schematic layout of our improved system,
Fig. II is a view similar to Fig. I showing the control phase of the improved system,

Fig. III is a diagrammatic showing of the air diffusion in a room embodied within our improved system,

Fig. IV is a view similar to Fig. III illustrating the application of the system to the control of entrances,

Fig. V is a diagrammatic elevation of a room to illustrate the manner in which the diffusion of the high velocity air controls condensation and cold surfaces,

Fig. VI is a diagrammatic treatment of air movement in similar rooms under different thermal conditions due to exterior temperature,

Fig. VII is a diagrammatic view of a simplified form of our system as compared to Fig. I,

Fig. VIII is a schematic layout of the control phase of the system of Fig. VII,

Fig. IX is a schematic diagram of a two-duct distributing system embodying the principles of the present invention.

Fig. X is an end view partially shown in section of the expander-silencer,

Fig. XI is a view taken on line XI—XI of Fig. X,

Fig. XII is a vertical section of the centrifugal air control nozzle, and

Fig. XIII is a sectional view taken on line XIII—XIII of Fig. XII.

Referring to Fig. I of the drawings, sufficient fresh air from the outside air is drawn in through storm louvers 10 to pressurize the area being air conditioned and prevent infiltration of outside air. To prevent excessive pressure within the area, an adjustable internal pressure relief damper 12 is provided to the outside air which is adjusted to vent to outside any internal pressure in excess of the specific amount required to effect the desired infiltration control. A suitable damper motor 14 controls the louvers 10 as well as the louver 16 in the return air duct. Control of the motor 14 is under the regulation of the thermostat and direction reversing relay unit 18 located in the mixed fresh and return air being conducted to the inlet end 20 of the centrifugal washer and conditioner 22.

The temperature of the water spray at 24 is controlled by the water chiller 26 in heat exchange relationship with the refrigeration unit 28. A pump 30 circulates the spray water collected in the sump of the washer 22 together with the make up water admitted at 32 through the chiller 26 and thence to the spray head 24. Conditioned air leaving the outlet 34 is drawn through the heating coil 36 controlled by the motor actuated valve 38 in the steam line 40 and regulated by the thermostat and direction reversing relay unit 42.

The pressure vane blower 44 is designed to develop air velocities in the duct work 46 in the order of 5000'/min. with velocities up to 6000'/min. entirely practical with velocities as high as 12,000'/min. being contemplated and capable of being produced and adapted to our system. A variable speed motor 48 drives the blower 44 at the desired speed through variable speed controls 50 actuated by the damper motor and direction reversing relay unit 52.

It will be understood that the cross-sectional area of the duct work 46 is exceptionally small as compared with a conventional low velocity system. In the two-duct arrangement shown, the paralleling sections 54 and 56 conduct air at different temperatures. This is accomplished through the use of a booster heating coil 55 in the section 56 downstream from the splitter damper 58 actuated by the motor 60 under the control of the pressure sensitive switch unit 62 to equalize the pressure in the duct sections. A motor valve 57 controls the flow of heat to the coil 55.

Adjacent to the areas at which the conditioned air is to be consumed, as for example the rooms 64 and 66, are located volumetric expander and silencer units 68 connected with the duct sections 54 and 56. The units 68 function to reduce the air velocities to the order of 1700 to 3000'/min. between the duct work 46 and the discharge nozzles 70. The secondary ducts 72, leading to the nozzles 70, are small in section and may conveniently be of flexible construction to facilitate installation between the joists. Mixing junctions 74 may be used to mix the air being received from the sections 54 and 56 prior to discharge into the rooms 64 and 66.

As shown, grills 76 are provided in the doors 78 to return the air to the intake side of the return air duct controlled by the louvers 16.

Referring to Figs. X and XI, one form of the expander-silencer unit 68 is shown in detail. The outer casing 80 is flanged at 82 to be supported in an opening 84 in the duct 54, for example. Concentrically supported within the casing 80 are cylinders 86 and 88 of which the cylinder 88 may be perforated throughout its entire surface while the cylinder 86 is only partially perforated. Adjustable dampers 90 extending along the entire length of the casing 80 control the flow of air from the duct 54 through the elongated slots 92. The damper

shafts 94 have actuating levers 96 for connection to the damper motor in any suitable manner. An imperforated extension of the cylinder 88 provides an outlet connection 98 to which the secondary duct 72 is attached.

By regulating the damper 90 the desired amount of air will flow under pressure through the slots 92 to be expanded and silenced by its passage through the perforations of the cylinders 86 and 88.

One form of air control nozzle 70 is shown in Figs. XII and XIII in which the air reduced velocity passes through the duct 72 to the inlet 100 of the fixed volute head 102. Supported for rotation in the head 102 is a barrel 104 having an inlet port 106 adapted to register with the inlet 100 to provide a volume action on the air supplied to the nozzle. A spiral guide vane 108 may be located between the inner wall of the barrel 104 and the acoustic core 110 which is in the form of a perforated cylinder preferably filled with a suitable sound absorbent material. An outlet vane assembly 112 is held in position by a screw 114 at the lower end of the core 110. A screw 116 holds the core 110 and barrel 104 assembly to the fixed volute head 102.

In Fig. III is diagrammatically shown the manner in which a jet of high velocity air is projected from an interior wall toward exterior windows. A small supply duct 118, carrying air up to velocities of 6,000'/min. or more, is connected to a fin type air control nozzle 120 in the wall 122 through the expander-silencer unit 124. A damper motor 126 operates the damper in the unit 124 in the manner described with reference to Figs. X and XI. The friction created with the room air mass dissipates the kinetic energy of the air from the jet 120 and puts all the air of the room into motion. The velocity of the jet and the diffusion angle of the air stream as defined and controlled by the nozzle will permit control of the impact against the far wall 128 so that any tendency to "bounce" off the wall 128 is controlled. As indicated by the arrows, the room air will be in constant motion without directional effects, with the diffusion of conditioned air into the room air mass coming from the kinetic energy imparted into the room air mass.

Fig. IV illustrates the manner in which our system may avoid the use of vestibules and revolving doors. By controlling and directing the air stream entering through the jet 130, it becomes possible to block the entrance 132 with a cone of conditioned air preventing the inward movement of the outside air. In cases where normal pressurization is insufficient to offset abnormal wind impact, a fan speed thermostat 134 located at the entrance 132 may be employed to adjust the air stream velocity to an amount necessary to be equal to and in opposite direction to the impinging wind.

Referring to Fig. V the diffusion of air in a room of multi-story height is illustrated. A thorough diffusion of conditioned air is created by the kinetic energy imparted thereto by jet 136. Considerable air movement is generated against the large glass area 138, thus minimizing fogging and condensation on its inside surface in cold weather. Likewise, air motion along the floor surface 139 conducts heat thereto, thus keeping floors laid directly on grade comfortably warmed. Since a uniform temperature level is achieved, the return air opening 137 can be located wherever convenience or structural economies dictate, since the only function is to return an air mass for reconditioning roughly equivalent to the mass being supplied to the room.

In Fig. VI is diagrammatically shown two similar rooms 140 and 142 having similar input of high velocity air through jet nozzles 144 and 146. With temperatures of 90° F. outside the window 148 of room 140 and 0° F. outside the room 142, the thermals outside the rooms 140 and 142 would be in the direction indicated by arrows. Within the rooms 140 and 142 the diffusion of air will be substantially the same in each room and sub-

stantially independent of the oppositely acting outside thermals. For this reason the air action indicated by arrows in the rooms 140 and 142 represents similar conditions. This situation remains the same whether warmed or refrigerated air is supplied to the room. Accordingly, substantially equal results are obtained in winter and summer with the same apparatus and system of operation.

A simplified arrangement of our system is shown in Fig. VII wherein the admission of mixed air to the plenum 150 is through the intake storm louver 152 and the recirculation duct 154. Dampers 156 and 158 under the operation of the damper motor 160, regulate the amount of outside and recirculated air in accordance with the demands of the duct thermostat 162. Water is sprayed at 164 into the mixed air passing to the centrifugal washer-conditioner 166 and out the axial discharge tube 168. A pump 170 draws water from the sump 172 and discharges it through a three-way valve 174 directly to the spray head 164 or indirectly through the spray water chiller 176 in heat exchanger relationship with the refrigerating apparatus 178. The immersion thermostat 180 controls the resulting relative humidity in the conditioned space by virtue of controlling the wet bulb temperature through an electrically controlled valve 182.

The washed air is carried in the duct 184 to the inlet 186 of the fan cabinet 188 where it is passed through the finned tube heating coil 190. In the summer, to lower the resistance through the unused heating coil 190, manual adjusted by-pass louvers 192 are provided.

Pressure vane blower 194 has its intake connected to the interior of the cabinet 188 and its exhaust into the pressurized supply duct 196. The motor 198 is preferably a wound rotor, slip ring variable speed type with its secondary electrical input, and hence rotative speed, controlled by well known manual face plate speed controller 200. Automatic operation of the controller is provided by a damper motor 202 connected thereto by the link 204. A central control panel 206 contains low voltage power transformers and reversing relays to change control circuits for seasonal changes. The admission of steam or hot water to the coil 190 is under the regulation of a motor operated valve 208. To operate the reversing relays to change seasonal control, the duct thermostat 210 is located in the outside air stream.

Connected into the duct 196 is shown an expander-silencer unit 212 to which the nozzle 214 is closely coupled for discharging the high velocity air into the room 216. The damper motor 218 controls the air flow to the unit 212 under the regulation of the room thermostat 220. Return air inlet 222 connects with the duct 154. Alternate return paths are indicated at 224 and 226.

In the operation of the system of Fig. VII, when the controls are set for winter operation, duct thermostat 162 gradually closes the outside damper 158 as the outside temperature falls. However, the damper 158 is never fully closed, as enough outside air is always introduced into the system to maintain a slight interior pressure within the building at all times. When the controls are set for summer operation, the outside air damper 158 reverses in operation, being almost closed at maximum outside temperature and gradually opening as the outside temperature reaches the desired inside temperature. This reversal of operation is effected by reversing relays (Fig. VIII) which are actuated by the season thermostat 210, whose remote bulb is in the outside air stream entering the plenum chamber 150.

As in the case of the system of Fig. I, the fan 194 pressurizes the low pressure air entering the cabinet 226 and discharges it at high velocity, in the order of 6,000'/min., or higher. The expander-silencer unit 212 reduces its velocity to that required for the treatment of the room 216, in the order of 1700 to 3000'/min. The unit also functions as a volume control device by linking it to the damper motor under the control of the room thermostat. In this manner individual room control is achieved.

Since it is not necessary to carefully balance the pressure differentials between supply and return with our high velocity system, the pressure created by the supply system causes the room air to seek its own path back to the conditioning apparatus. Thus, a path can be established through corridors, or the space above a furred ceiling on the corridor carrying the supply duct can be used for this purpose. For example, the corridor 226 may be used and each door provided with a louver 224 with the return duct 154 opening into the corridor 226.

While we propose to employ standard temperature control devices in our system, we employ such devices in an improved manner and under a principle of control which has not heretofore been employed to our knowledge. This principle involves the use of "floating" type thermostatic controls and the simultaneous variations of both air temperature and volume.

The floating type thermostat is recognized as potentially the most sensitive of the mechanical type of elements. In such instruments the contact on the bimetallic element swing between two fixed contacts. In basic principle it is presumed that contacts on the "hot side" will increase the heat supply and contact on the "cold side" will cause it to decrease and this swing back and forth will continue in diminishing cycles until just the proper supply of conditioned air is achieved for the specific requirements then existing and the bimetal element is no longer disturbed thermally and remains neutral between the two fixed contacts.

This all presupposes that a system is used that will permit a prompt variation in heat supply, plus a fast response in the thermostat. Actually, very few systems provide the means to fully use the floating principle. In conventional air conditioning systems the fans are generally so taxed as to effect distribution and pressure differentials to effect it are too small to allow a change in air volume at the source. Temperature control is therefore generally limited to changing the air temperature only. The problem is further complicated by the fact that maximum air delivery is required in summer and the distributing system must be designed for this condition, making it difficult to establish the necessary pressure differentials with a lower delivery in winter.

It is in this field that our principle permits full play of the floating principle in that both the air volume and air temperature are varied simultaneously. Further, whereas with conventional systems the room thermostat operates in quiet air, or the "skin effect" next to the walls, this area is the region of maximum air movement in our system with the result that a fast response is obtained.

In summer operation the main thermostat varies both the speed of the pressure vane blower 194 and the temperature of the water leaving the spray heads 164. This is done through the damper motor 202 on the fan speed controller 200 and the three way valve 182. As the outside temperature increases, the fan speed also increases and the spray water temperature decreases. Thus, the total volume of conditioning medium supplied to the building is metered in accordance with weather conditions. With the floating type temperature control so applied, it is not necessary to employ an outside thermostatic element to "anticipate" requirements. Rather, inside results dictate the supply.

In winter operation the fan speed control is opposite to summer control; increasing as the outside temperature decreases. At the same time the main thermostat also increases the heat supplied by heating coil 190 in the fan cabinet as the outside temperature falls. Further, the 3-way valve 182 controlling the spray water temperature is disconnected from the main thermostat in winter and placed under the control of the immersion thermostat 180 whose element is placed in the water sump 172. This effects humidity control for the building by controlling the wet bulb temperature.

This seasonal control circuit change is readily effected by reversing relays, shown on the diagram Fig. VIII. Further, this seasonal switchover can be made automatically by operating the reversing relays with a duct thermostat 210, whose element is placed in the outside air stream entering the intake plenum 150. This is very advantageous in spring and fall seasons, when heating may be required during part of the day but cooling required during the balance of the day. During spring and fall, therefore, the by-pass dampers 192 in the fan cabinet 188 are closed and steam (or hot water) is supplied to control valve 208 on the main heating coil 190. Thereafter, the system will swing from winter to summer control, as required, in accordance with the setting of the seasonal thermostat 210.

In Fig. II the electrical wiring diagram of the system of Fig. I is fully disclosed except for such ancillary circuits as embodied in the test panel and program circuits.

Transformer 228 reduces the voltage to that required for the operation of the standard damper motor, relays and thermostats we employ in our system with the low voltage conductor being indicated at 230 and 232.

The damper motor units employed to operate various dampers, valves and motor regulator are of well known construction and constitute no part of the present invention. Preferably, the units are substantially geared down to enable regulation between their extreme positions when used in combination with our use of floating type thermostat and the high rate of response to control which characterizes our system. Throughout Fig. II, the slowly rotated shaft of each unit is indicated at 234 and carries cams 236 which actuate the electrical switches 238.

The master thermostat 240 located in the room 66 preferably comprises a temperature unit 242 and a season unit 244, whose bimetallic elements are linked together so as to maintain a definite operating range relationship with each other regardless of manual resetting of the thermostat for the purpose of changing the maintained room temperature. During operation of the system in the winter part of the cycle the contact arms of the switches 246 are in the full line position and remain in that position during the normal range of operation of the thermostatic temperature unit 242. However, in the event the temperature continues to rise beyond a predetermined level, the thermostat season unit 244 closes the circuit in which it is located to energize the coils 248 to move the switches 246 into the dotted line position shown for summer operation of the system.

During the normal operation of air conditioning the room 64, the thermostat 250 will control the proportion of air drawn from the ducts 54 and 56 with the switches 252 in the dotted line position shown in Fig. II. However, should a conference be held in room 64 attended by a large number of people requiring a more frequent change of air, the switches 252 may be manually thrown into the full line position shown which will result in the damper of the unit 68 being opened wider to increase the amount of air being admitted to the room 64 through the nozzles 70.

Associated with each main control damper motor is an auxiliary switch arrangement 239 actuated by the motor drive shaft, whose switches are set to operate when the damper motors reach either extreme of their arc of rotation. Utilizing switches 238 of limit switch arrangements 239, circuits are established through indicator lamps 241. These lamps are then assembled into an indicator panel for the purpose of allowing visual observation of the functioning of all of the main control devices from one position, thereby detecting improper functioning or failures.

Relay 235, also actuated by the season thermostat 244 provides means for indicating through associated lamps 241 which season cycle is in operation.

Mixed air thermostat 18 controls damper motor 14,

with the ancillary circuits passing through relay 15 for the purpose of reversing the circuit relationship between summer and winter.

In Fig. VIII is shown the electrical diagram of the simplified system disclosed in Fig. VII. The reference characters applied to the damper motor units of Fig. VII have been applied to Fig. VIII to designate the electrical circuit of each such unit. As heretofore described with reference to Fig. II, each damper motor unit has a shaft 254 upon which cams 256 are carried for operating the switch members 258. The master thermostat 260 controls the operation of the motor valve 208 which admits the heat to the coil 190. The reversing relays 262 are operated by the coils 264 to move the switches 266 from the full line winter position shown to the dotted line summer position under the control of the season control setting thermostat 210 in the manner with reference to Fig. II.

Referring to Fig. IX, the discharge duct from the pressure vane blower cabinet 264 splits into two smaller ducts 266 and 268, one being referred to as the "Cool Duct," and the other, the "Warm Duct," respectively.

Positioned across the warm duct 268 is a special finned tube, hairpin shaped booster coil 270. This booster coil serves to effect the difference in temperature required for this supply principle. In summer operation, air emerges from the fan cabinet 264 sufficiently chilled to properly cool the building, the degree of chilling being controlled by a master thermostat. The booster coil 270 in the warm duct is then supplied with sufficient steam (or hot water) to raise the air temperature in the warm duct to approximately the temperature generally being maintained, or slightly higher. This control is effected by controlling the motorized valve 272 serving the booster coil 270 by a duct thermostat 274.

In winter operation the air leaving the fan cabinet 264 is maintained only slightly above the general building temperature. The booster coil 270 then maintains the air temperature in the warm duct 266 from 40° to 60° above this temperature. Under winter control, the booster coil control valve 272 is controlled by the master or zone thermostat.

At the point where the duct from the fan cabinet splits into two ducts, a swinging splitter damper 276 is positioned. The position of this splitter damper controls the relative amounts of air supplied to the two ducts. The splitter damper is linked to a damper motor 278 to make this function automatic. At the extreme end of the twin duct run a duct volume proportioning control 280 is located. This consists of a diaphragm type floating switch with the two sides of the diaphragm subjected to the pressures within the two ducts by means of connecting tubing.

In operation, if that portion of the building served by the two ducts 268 and 270 in question is making principal demand from one duct only, the resulting drop in pressure in the overloaded duct is transmitted to the proportioning control 280, which thereupon re-positions the splitter damper 276 to divert a major portion of the air supply into the starved duct until its normal pressurization is restored.

Each room of the building is supplied from both supply ducts. As shown, the small room 282 has a silencer-expander 284 connected to each duct 266 and 268 and the outlets are brought together and thereupon connected to one air control nozzle 286. These expander units are linked to a damper motor 288 in such a manner as to operate in opposition; one closing as the other opens. The room thermostat 290 controls this damper motor 288 so as to introduce either warm or cool air, or any mixture thereof as required to satisfy the thermostat. Each room, therefore, becomes a completely independent zone.

It is possible, particularly with contemporary designs featuring large window areas, to have a situation during

spring and fall in which it may be necessary to furnish some heat to north elevations at the same time that south exposures require cooling due to solar heat. Such a situation can be met with the two duct system since low enough temperature air can be furnished in the cool duct to cool the south exposure while warm enough air in the warm duct is serving the north exposure.

The accompanying diagram indicates each room having two expander sections as its source of supply. It follows that in a larger space several air inlets may be required. In such cases several pairs of expanders can be connected to still operate from one room thermostat, making such a space an independent zone.

In the room 292 of Fig. IX is shown one possible booster ventilator arrangement with our two duct system. At one time the room 292 might be used for a small supervisory meeting in which the normal air supply is adequate for the few people involved. Again, the same room might be used for a group gathering in which smoking is allowed. It might then be desirable to treble the air changes just for smoke purging.

To effect such an arrangement with the two duct system a separate damper motor is linked to each expander, shown in Fig. IX as 298. The reason for this is that the additional or "booster" supply of air must come from the duct carrying air at room temperature and this is a different duct in summer than in winter. Further, when a booster effect is desired the damper motors must effect a wider opening of the volumetric expander 284 to obtain greater air delivery from the expander. The damper motors 298 are multi-position motors whose arcs of rotation can be preselected from the setting of manual wall switch 302. Further, means must be provided to proportion the amount of air supplied from each duct in accordance with the temperature requirements of room 292. Hence each damper motor 298 is actuated by room thermostat 300 whether under normal operation or booster operation; the booster switch 302 merely preselecting the rotative arc through which each damper motor (and through each volumetric expander damper) operates.

I claim:

1. A method of pressurized supply high velocity air control comprising the steps of pressurizing conditioned air, distributing the pressurized air at velocities in the order of 5,000 to 12,000'/min., reducing the velocity of the air by a primary expansion to the order of 2,500'/min. at a point adjacent a zone of consumption, and conducting the air from the point of primary expansion and reduced velocity to a point of secondary expansion, effecting a secondary expansion of the air and discharging the same into the zone of consumption to provide a high degree of diffusion principally by the kinetic energy of the discharged air being transferred into the ambient air of said area, the initial commingling of said discharged air with the interior air of said zone taking place within said zone.

2. A method as defined in claim 1 wherein the volume of air conducted from the point of primary expansion to the point of secondary expansion is variable.

3. A method as defined in claim 1 including dividing the high velocity air into parallel streams of conditioned air at different temperatures, and providing temperature responsive regulation at the point of primary expansion prior to the conducting of the air to the point of secondary expansion.

4. A method of pressurized supply high velocity air control comprising the steps of pressurizing conditioned air, distributing the pressurized air at velocities in the order of 5,000 to 12,000'/min., dividing the air into parallel streams, exchanging heat in one of said streams to obtain a temperature differential in said streams, reducing the velocity of the air in said streams by a primary expansion to the order of 2,500'/min. at points adjacent a common zone of consumption, and conducting and mixing the air from the points of primary expansion and

reduced velocity to said common zone for a secondary expansion and discharge into said zone to provide a high degree of diffusion principally by the kinetic energy of the discharged air, the initial commingling of said discharged air with the interior air of said zone taking place within said zone.

5. A method as defined in claim 4 which includes the step of temperature responsive regulation of the volume of air at the points of primary expansion to control the temperature of said common zone.

6. A method as defined in claim 4 which includes the step of mixing the air from said points of primary expansion prior to discharge into the common area of consumption.

7. A method of air conditioning a room having an entrance to adjacent area comprising pressurization of the air within the room to offset normal infiltration from said adjacent area through the discharge of conditioned air at one velocity, and automatically increasing the velocity of the entering conditioned air upon opening of said entrance to said open adjacent area independently of pressure conditions within said room.

8. A method as defined in claim 7 wherein the conditioned air of increased velocity is directed toward said entrance.

9. In a high pressurized velocity air conditioning distribution system, a pair of pressurized ducts for distributing conditioned air to points adjacent living spaces, mixing chambers located between said ducts and said living spaces, secondary ducts, certain of said secondary ducts directly connecting said first ducts to said mixing chambers and certain other of said secondary ducts directly connecting said mixing chambers with said living spaces, means in said living spaces for controlling the drawing of air from said first ducts in differential quantities to maintain predetermined temperatures in said living spaces, a main supply duct for said pair of ducts, and automatically operating means for diverting more or less of the air from said supply duct to one or the other of said pair of ducts as the demand thereon varies in said living spaces.

10. In a high pressurized velocity air conditioning distribution system, a pair of pressurized ducts for distributing conditioned air to points adjacent living spaces, mixing chambers located between said ducts and said living spaces, secondary ducts, certain of said secondary ducts directly connecting said first ducts to said mixing chambers and certain other of said secondary ducts directly connecting said mixing chambers with said living spaces, means in said living spaces for controlling the drawing of air from said first ducts in differential quantities to maintain predetermined temperatures in said living spaces, and a main supply duct for said pair of ducts.

11. In a high pressurized velocity air conditioning distribution system, a pair of pressurized ducts for distributing conditioned air to points adjacent living spaces, mixing chambers located between said ducts and said living spaces, secondary ducts, certain of said secondary ducts directly connecting said first ducts to said mixing chambers and certain other of said secondary ducts directly connecting said mixing chambers with said living spaces, means in said living spaces for controlling the drawing of air from said first ducts in differential quantities to maintain predetermined temperatures in said living spaces, a main supply duct for said pair of ducts, and a booster coil in one of said pair of ducts.

References Cited in the file of this patent

UNITED STATES PATENTS

Re. 19,759	Davis	Nov. 19, 1935
986,731	McGerry	Mar. 14, 1911
1,045,419	Matula	Nov. 26, 1912

(Other references on following page)

11

UNITED STATES PATENTS

1,303,210	Klein	May 6, 1919
1,670,766	Gould et al.	May 22, 1928
1,783,005	Stewart	Nov. 25, 1930
1,832,638	Kitchen	Nov. 17, 1931
1,916,908	Stacey et al.	July 4, 1933
1,949,735	Bulkeley	Mar. 6, 1934
2,109,512	Stacey	Mar. 1, 1938
2,123,440	Schlafman	July 12, 1938
2,150,252	Shure	Mar. 14, 1939
2,203,590	Bock	June 4, 1940
2,226,510	Grant	Dec. 24, 1940
2,271,010	Honerkamp	Jan. 27, 1942
2,282,210	Plum	May 5, 1942

2,296,314
2,314,569
2,422,560
2,620,983
2,644,389
2,652,763

5

10

12

Slade	Sept. 22, 1942
Baker	Mar. 23, 1943
Palmer	June 17, 1947
Lyman	Dec. 9, 1952
Dauphinee	July 7, 1953
Emery et al.	Sept. 22, 1953

FOREIGN PATENTS

Great Britain	July 24, 1930
Great Britain	Jan. 16, 1931
Great Britain	July 4, 1932
Italy	June 24, 1932
Germany	Apr. 18, 1933