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VAV SYSTEM WITH INVERTER-DRIVEN AHU FOR HIGH-RISE OFFICE BUILDING IN TROPICAL CLIMATES—A CASE STUDY

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ABSTRACT

In this case study, the application of adjustable-frequency inverters to drive the various fan and pump motors for the variable-air-volume (VAV) system in a high-rise office building in hot, humid climates is described. Methods for outside air admission and building pressure control are demonstrated. Energy conservation aspects are discussed.

INTRODUCTION

The fans of VAV systems are traditionally driven by "squirrel cage" motors that run at constant speed and the air flow is controlled by the inlet guide vanes of the fans. An alternative for airflow control is to use variable-width fans.

With the introduction of adjustable-frequency inverters, ordinary fans for VAV systems can still be driven by "squirrel cage" motors, but their speed can be adjusted for airflow control in relation to the condition of outside air, building pressure, and temperature.

For demonstration, the VAV system being installed in a high-rise bank building in the central commercial area of Hong Kong, where the climate is hot and humid, is described herewith.

BACKGROUND DATA

The building is a 40-story high-rise presently under construction and having a total gross floor area of approximately 387,500 ft² (36,000 m²). A typical office floor has an area of approximately 7500 ft² (697 m²), as shown in Figure 1. The design data for the HVAC system on a typical office floor are given in Table 1.

Chilled water is supplied to each floor by three 60-hp (44.8 kW) secondary chilled water pumps, and heating water is supplied by two 30-hp (22.4 kW) secondary heating water pumps. Primary air is supplied to each floor by two 25-hp (18.7 kW) primary air-handling units (PAU) and exhaust air is removed from each floor by two 20-hp (14.9 kW) exhaust air fans. Supply air to a typical office floor is supplied by one 15-hp (11.2 kW) air-handling unit (AHU). The above pumps and fans, totaling 41 units, are driven by "squirrel cage" motors equipped with adjustable-frequency inverters (AFI). The schematic of the entire system is given in Figure 2 and the data of the above fans are in Table 2.

HVAC SYSTEM FOR TYPICAL OFFICE FLOOR

Chilled Water and Heating Water Systems

Chilled water and heating water are supplied separately through risers to each floor by secondary chilled water pumps and secondary heating water pumps, respectively. These pumps are controlled by adjustable-frequency inverters in conjunction with direct digital controllers (DDC), which are subsystems of the facility's central supervisory (FCS) system, to maintain a predetermined minimum differential pressure at the coils that have the lowest differential pressure. The chilled water and heating water systems are given in Figures 3 and 4, respectively.

Primary Preconditioned Outside Air System

Primary preconditioned outside air is supplied by primary air-handling units (PAU) through primary air shafts to each floor. Each of these PAU is started when any of the AHU on the floors served by the same PAU is started. With reference to Figure 5, primary preconditioned outside air is controlled in the following manner:

Summer/winter mode is selected by software in the DDC by detecting the outside air temperature using sensor TT-2. In summer, when the fan is started, the primary air temperature sensor (TT-1), via the control algorithm in the DDC, modulates the chilled water control valve (V-1) to maintain the primary air temperature at the predetermined value. When the outside air humidity, sensed by humidity sensor HT-1, is above a preset value, the control algorithm in the DDC resets the predetermined value of the primary air temperature downward. In winter, when the fan is started, the primary air temperature, which is reset by the outside air temperature (sensed by temperature sensor TT-2), modulates the heating water control valve (V-2) to maintain the primary air temperature at the predetermined value.

The chilled water control valve (V-1) is a two-way, normally open type and the heating water control valve (V-2) is a two-way, normally closed type; both of them are interlocked in the closed position when the fan is stopped. They are actuated by compressed air sup-

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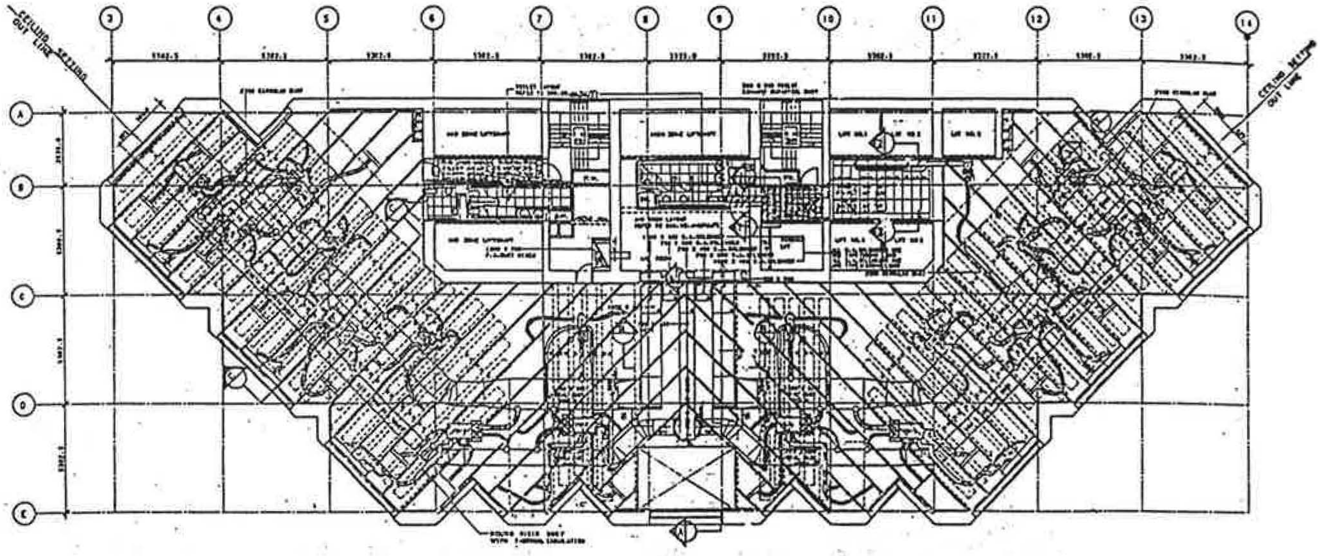


Figure 1 Typical office floor

TABLE 1
Design Data For HVAC Systems on Typical Office Floor

(a) Outside Conditions:		Summer 90.9°F (32.7°C) db, 82.0°F (27.8°C) wb Winter 45.0°F (7.2°C) db
(b) Internal Conditions:		Summer 76.1°F (24.5°C) db, 55% to 65% RH Winter 70.0°F (21.1°C) db, RH not specified
(c) Occupancy Density:		75.3 ft² (7.0 m²) per person
(d) Ventilation Rate:		10 cfm (4.72 L/s) per person
(e) Chilled Water Temperatures:		Supply 45.0°F (7.2°C) Return 54.5°F (12.5°C)
(f) Heating Water Temperatures:		Supply 103.1°F (32.5°C) Return 90.5°F (32.5°C)
(g) Primary Air:		1000 cfm (472.2 L/s)
(h) Exhaust Air:	Maximum Flow Rates	667 cfm (314.8 L/s) (total of 2 systems)
(i) Supply Air:		Cold 11,000 cfm (5194 L/s) Hot 4400 cfm (2078 L/s)

plied from the air main (M) and controlled by the DDC through the analog electronic/pneumatic transducers E/P-1 and E/P-2, respectively.

Two pressure sensors, PT-1 and PT-2, sense the static pressure two-thirds of the way downstream in the primary air shaft and transmit signals to the DDC. Via the DDC control algorithm, the lower signal controls the speed of the fan motor through the adjustable-frequency inverter to maintain the primary preconditioned outside air system pressure at the predetermined value.

Primary preconditioned outside air is admitted to an office floor in the AHU room by a constant-air-volume (CAV) terminal unit, which, while normally closed, is open when the supply power to the AHU for that floor is on, as shown in Figure 7.

Exhaust Air System

Exhaust air is removed by exhaust air fans through washroom exhaust air shafts from each floor.

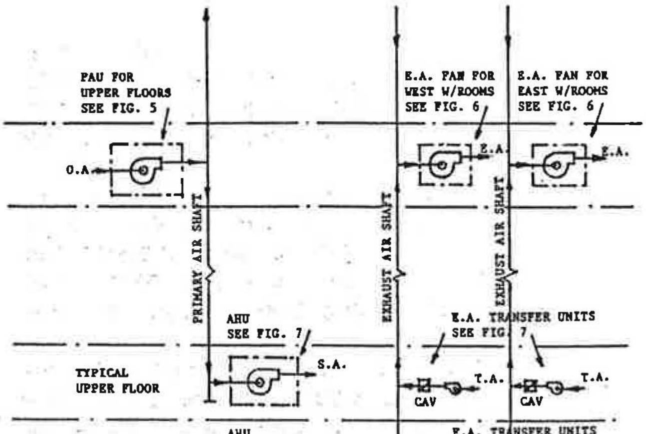


Figure 2 Schematic of the entire VAV system

Each of these exhaust air fans is tied to the PAU serving the same floor, so that they start and stop simultaneously. With reference to Figure 6, exhaust air is controlled in the following manner:

A static pressure sensor, PT, installed in the main exhaust air shaft two-thirds of the way from the exhaust air fan, detects the static pressure therein and transmits a signal to the DDC, which, via the control al-

TABLE 2
Fan Data

Primary Air Fan (Double Inlet, Double Width, Backward Curved)	
Designed Airflow Rate	: 13,000 cfm (6139 L/s)
Static Pressure	: 6 in. w.g. (1.5 kPa)
System Effect	: 0
Speed	: 1700 rpm
Output Power	: 18.1 hp (13.5 kW)
Total Efficiency	: 0.71
Exhaust Air Fan (Single Inlet, Single Width, Backward Curved)	
Designed Airflow Rate	: 11,480 cfm (5422 L/s)
Static Pressure	: 5.5 in. w.g. (1.375 kPa)
System Effect	: 0
Speed	: 1250 rpm
Output Power	: 13.3 hp (9.9 kW)
Total Efficiency	: 0.85
Supply Air Fan (Double Inlet, Double Width, Backward Curved)	
Designed Airflow Rate	: 11,000 cfm (5194 L/s)
Static Pressure	: 4.5 in. w.g. (1,125 kPa)
System Effect	: 0.23 in. w.g. (0.057 kPa)
Speed	: 1675 rpm
Output Power, Total	: 11.65 hp (8.69 kW)
Output Power, Per Unit of Floor Area	: 1.55 hp/1000 ft² (12.47 W/m²)
Total Efficiency	: 0.72

gorithm, sends an analog signal to the adjustable-frequency inverter to control the speed of the exhaust air fan motor to maintain static pressure at the predetermined value.

Exhaust air is transferred from the two washrooms on an office floor to the main exhaust air shafts by two sets of exhaust air transfer fans and constant-air-volume (CAV) terminal units, one set for each washroom, which are tied to the AHU for that floor, so that they start and stop simultaneously, as shown in Figure 7.

Supply and Return Air System

On a typical office floor, supply air is supplied by an air-handling unit and distributed to each zone through cold and hot supply air ducts by 12 dual-duct VAV terminal units and 7 single-duct VAV terminal units, as shown in Figure 1. Each of the VAV terminal units is pneumatically controlled by a zone thermostat. The VAV terminal units are pressure independent at all airflow rates between maximum and minimum settings regardless of inlet static pressure and supply air temperature and humidity changes, and their characteristics are shown in Table 3.

Return air is returned to the AHU through the space above the false ceiling. Supply air to the washrooms is the return air from the office space through transfer ducts. With reference to Figure 7, supply air on a typical office floor is controlled in the following manner:

The supply air fan is started and stopped by the optimum start/stop program of the FCS system using

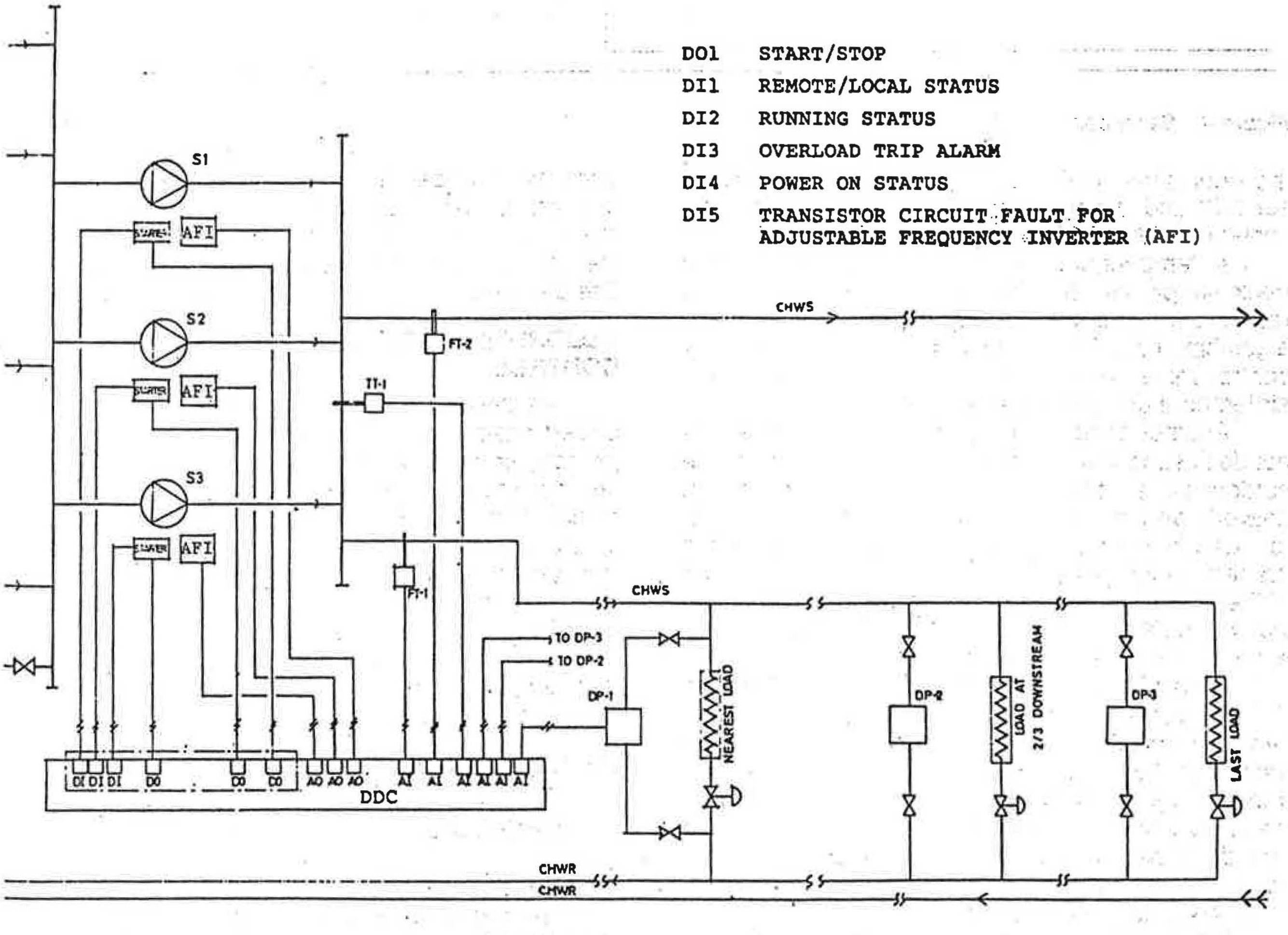


Figure 3 Secondary chilled water system

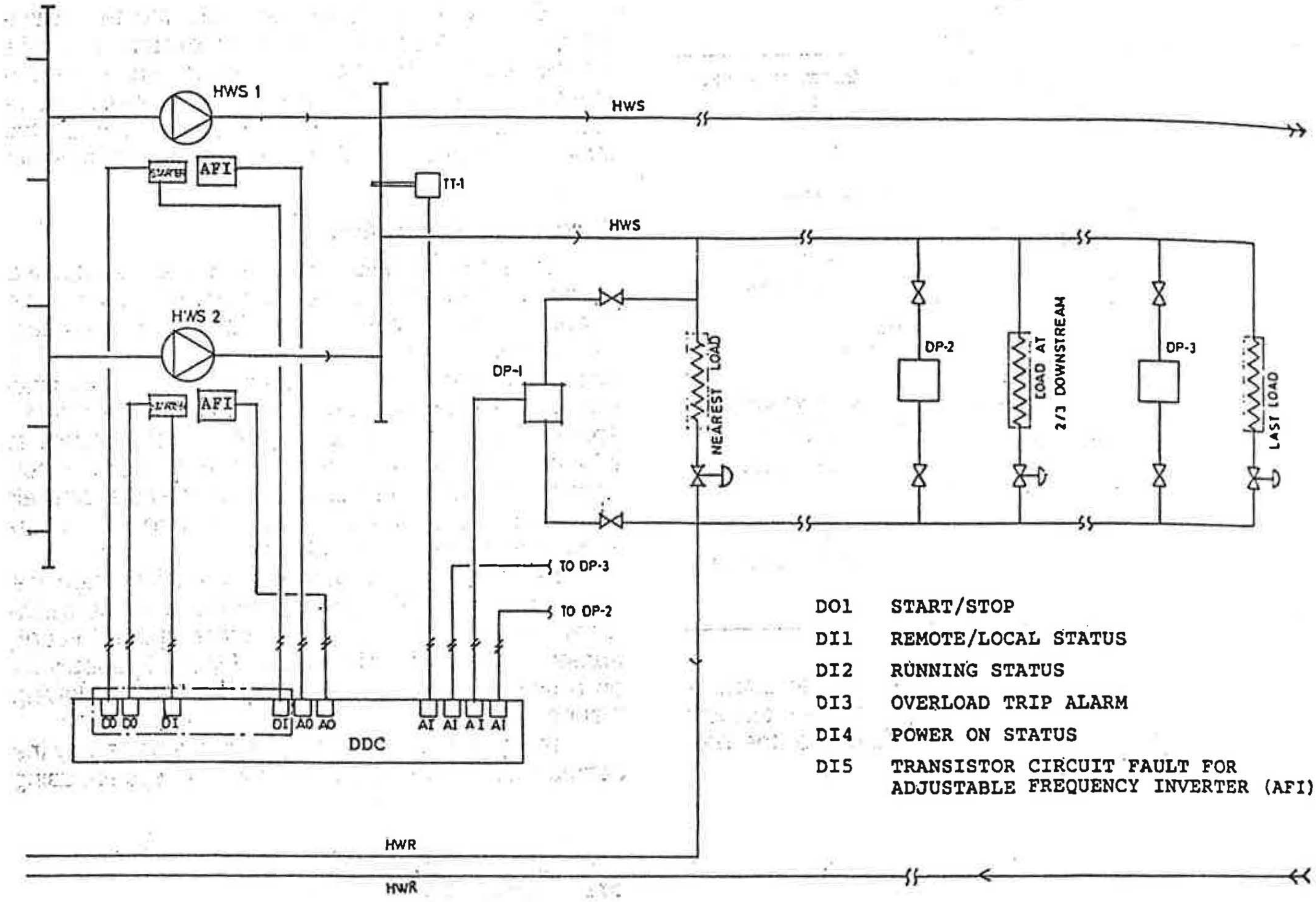


Figure 4 *Secondary heating water system*

the signals transmitted by the space temperature sensor (TS) and the outdoor air temperature sensor (not shown) via the peer communication bus.

A temperature sensor (TT-1) senses the cold deck supply air temperature, which is reset by the return air temperature sensor (TT-3) via the control algorithm of the DDC, and modulates the chilled water control valve (V-1) to maintain the cold deck supply air temperature at the predetermined value.

Another temperature sensor (TT-2) senses the hot deck supply air temperature, which is reset by the outdoor air temperature via the peer communication network, and modulates the heating water valve (V-2) to maintain the hot deck supply air temperature at the predetermined value. The construction and operation of the chilled water control valve (V-1) and the heating water control valve (V-2) are similar to that for the primary air system described above.

Several differential pressure sensors, (PT-1) and (PT-2), installed in the cold and hot ducts at representative spots, detect the static pressures with reference to the building pressure on that floor and transmit signals to the DDC. Via a control algorithm, the DDC selects the lowest signal and controls the speed of the fan motor through the adjustable-frequency inverter to maintain the supply air system pressure at a predetermined value.

The primary air and exhaust air CAV terminal units, as described above, are open- and closed type and are actuated by compressed air supplied from the air main (M) and are controlled by the DDC through the digital electronic/pneumatic transducer (EP-1). The CAV terminal units are also pressure independent.

ALGORITHM FOR FAN AND PUMP SPEED CONTROL

As described above, the speeds of secondary chilled water pumps, secondary heating water pumps, primary air fans, exhaust air fans, and supply air fans are controlled by adjustable-frequency inverters in conjunction with DDC to maintain predetermined differential pressures or static pressures in their respective systems. With reference to Figure 8, block diagram (a) shows the control algorithm for secondary chilled water and secondary heating water pumps, block diagram (b) shows the algorithm for the exhaust air fans, and block diagram (c) shows the algorithm for primary air and supply air fans.

SPACE ENVIRONMENT CONTROLS

Some of the space environment controls of the above HVAC system are as follows:

Space Humidity Control

As mentioned above, the predetermined value of the primary preconditioned outside air system tem-

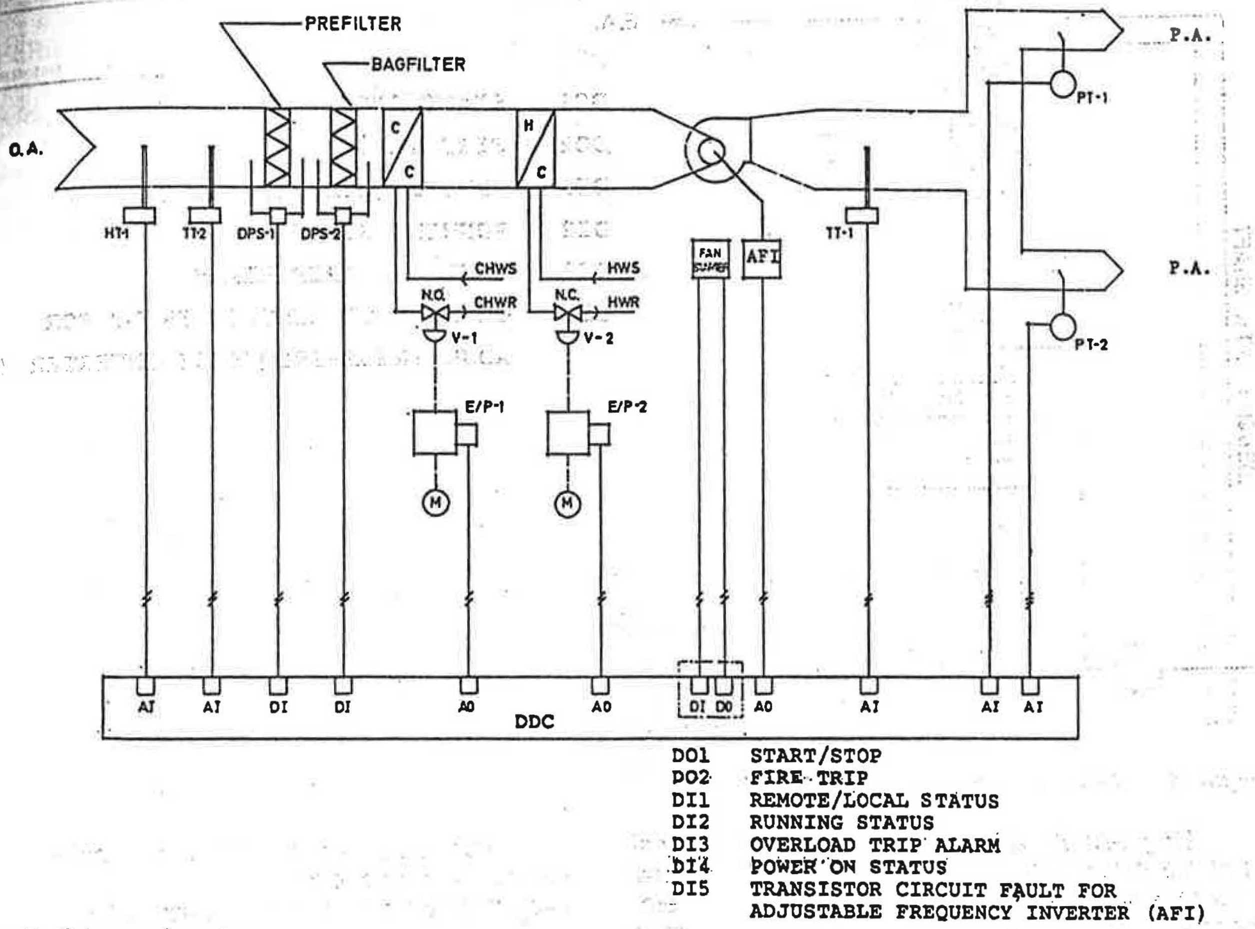


Figure 5 Primary air system

perature is reset downward when the outdoor air humidity is above a preset value. Therefore, after this colder primary air is mixed with return air and reheated in the AHU, the resulting supply air maintains its humidity within a predetermined range.

Building Pressure Control

With reference to Figure 7, primary preconditioned outside air is admitted through a CAV terminal unit to a typical office floor and exhaust air is removed from that floor by two sets of exhaust air transfer fans and CAV terminal units. These terminal units are manually adjusted to preset the ratio of exhaust air flow to primary preconditioned outside air flow such that the building static pressure is maintained at the predetermined value, which is just enough to prevent air infiltration through doors, staircases, etc.

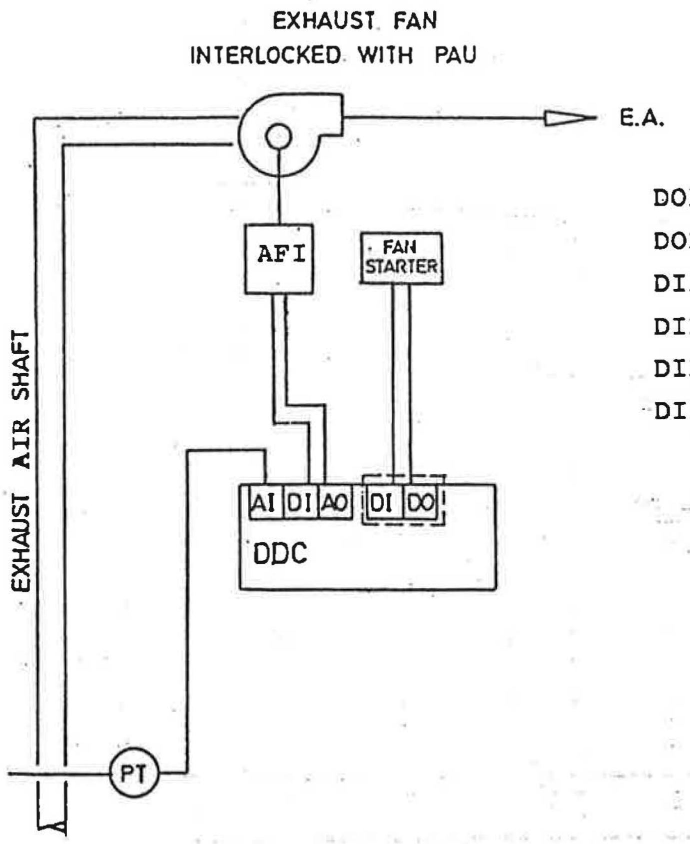
Making Up for Air Filter Pressure Losses

As shown in Figures 5 and 7, respectively, prefilters and bag filters are installed on each PAU and AHU. Two differential pressure switches, DPS-1 and DPS-2, send alarm signals to the DDC when the prefilter and bag filter, respectively, are clogged. When the differential pressures across these filters increase, the inlet pressure to the fan and the air duct pressure decrease, and the pressure sensors transmit signals via the DDC control algorithm to increase the speed of the fan motor through the adjustable-frequency inverter to restore the system pressure in the air duct to the originally set value.

Response to Changing Loads

As shown in Figure 1, supply air is distributed through dual- and single-duct VAV terminal units on a typical floor.

Dual-Duct VAV Terminal Units These terminal units are installed in perimeter zones along outside walls through which heat is lost during winter. Also, the occupants in these zones are engaged mainly in desk work. When the AHU for that floor is started, compressed air is supplied to the control devices, while cold and hot supply air are supplied to the cold and hot air inlets, respectively. Each of them is controlled by a room-type zone thermostat to maintain the space condition by modulating the normally open damper in the cold air inlet and the normally closed damper in the hot air inlet through two pneumatic damper operators.



DO1 START/STOP
DO2 FIRE TRIP
DI1 REMOTE/LOCAL STATUS
DI2 RUNNING STATUS
DI3 OVERLOAD TRIP ALARM
DI5 TRANSISTOR CIRCUIT FAULT FOR ADJUSTABLE-FREQUENCY INVERTER (AFI)

Figure 6 *Exhaust air system*

Single-Duct VAV Terminal Units These terminal units are installed in interior zones not adjacent to any outside wall, such as elevator lobbies, corridors, etc. Heating is not required, even during the winter. When the AHU for that floor is started, compressed air is supplied to the control devices, while only cold supply air is supplied to the air inlets. Each is controlled by a room-type zone thermostat to maintain the space condition by modulating the normally open damper in the air inlet through a pneumatic damper operator.

When a zone thermostat setting is adjusted or the heating/cooling loads are changed, the VAV terminal unit in that zone is modulated, causing the static system pressure in the supply air ducts to change; the fan motor speed is changed by the adjustable-frequency inverter to restore the supply air system pressure to the predetermined value.

ENERGY CONSERVATION ASPECTS

Some of the energy conservation aspects of the above HVAC system are as follows.

Free Cooling for Night Purge

The DDC controls the VAV system on a typical office floor to introduce outdoor air at night any time when the following conditions are met:
—outdoor air temperature is above 50°F (10°C),
—space temperature is above 75°F (23.9°C),
—outdoor air temperature is below space temperature, and
—outdoor air dew point is below 60°F (15.6°C).

Night purge stops when one or more of the following conditions is met:
—outdoor air temperature is below 50°F (10°C), and/or
—space temperature is below 75°F (23.9°C), and/or
—outdoor air temperature is less than 5°F (2.8°C) cooler than the space temperature, and/or
—outdoor air dew point is above 60°F (15.6°C).

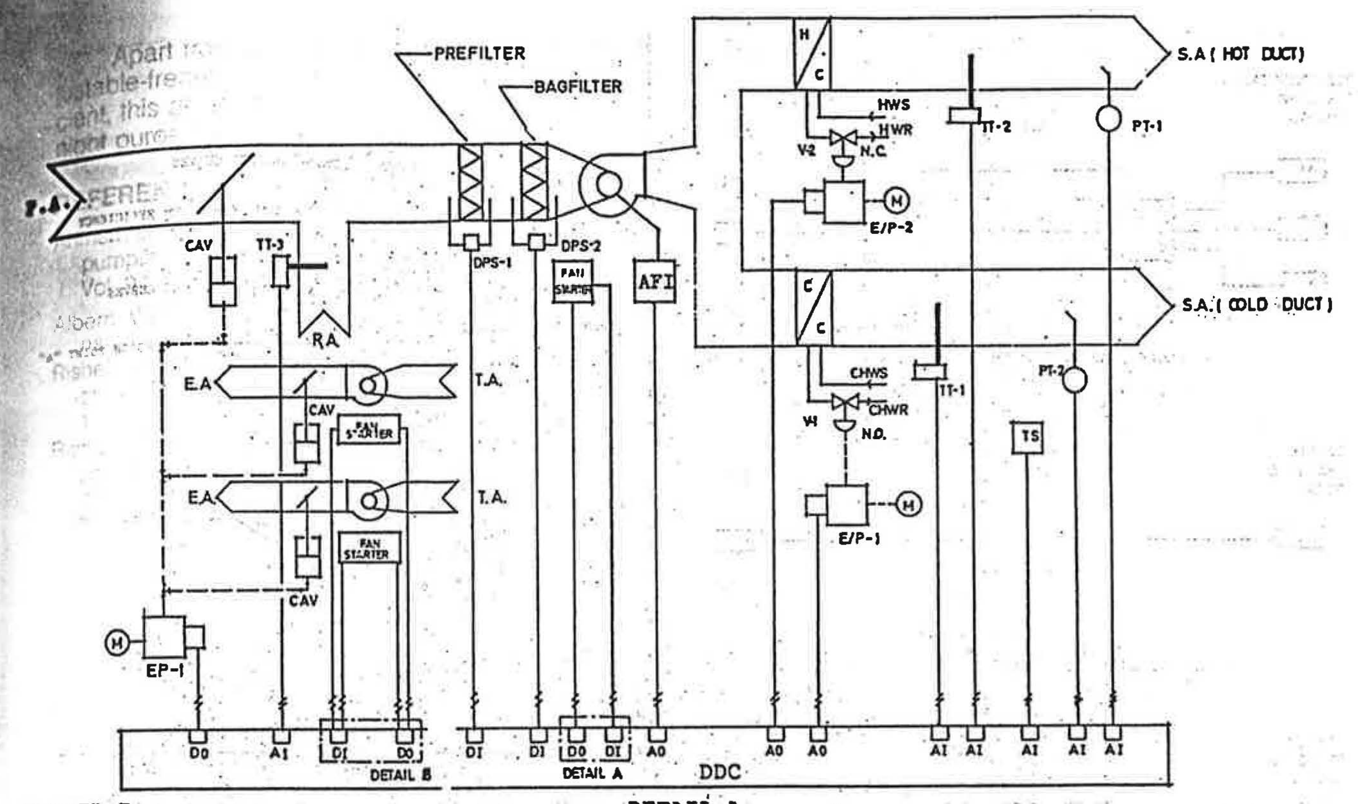
During night purge, the DDC algorithm operates the PAU, AHU, and exhaust air fans in the usual way to maintain the predetermined system pressures with chilled water and heating water valves closed.

Fan and Pump Motor Efficiency

All motors used for the above HVAC system are high efficiency, and the minimum efficiencies are shown in Table 4.

Fan and Pump Speed Reduction

Primary Air Fan When the airflow rate is below the designed value, as in the extreme case when only one of the typical floors is occupied, the airflow rate is only 7.7% of the designed flow rate, as can be calculated from the numbers in Tables 1 and 2; some of the CAV terminal units admitting primary preconditioned air to the AHU rooms are closed. With reference to the typical fan curve diagram in Figure 9, if the fan speed is constant, the system design point (A) will move toward the left along the rpm curve, as shown by the dotted line (1). In this case, the adjustable-frequency inverter and the DDC slow down the primary



DETAIL B

DO1 START/STOP
DO2 FIRE TRIP
DI1 REMOTE/LOCAL STATUS
DI2 RUNNING STATUS
DI3 OVERLOAD TRIP ALARM

DETAIL A

DO1 START/STOP
DO2 FIRE TRIP
DI1 REMOTE/LOCAL STATUS
DI2 RUNNING STATUS
DI3 OVERLOAD TRIP ALARM
DI4 POWER ON STATUS
DI5 TRANSISTOR CIRCUIT FAULT FOR ADJUSTABLE-FREQUENCY INVERTER (AFI)

Figure 7 *Supply air system*

TABLE 3
Characteristics Of VAV Terminal Units

Airflow	Mimimum ΔP_s	Mimimum ΔP_t
400 cfm (189 L/s)	0.01 in. w.g. (2 Pa)	0.06 in. w.g. (15 Pa)
500 cfm (236 L/s)	0.01 in. w.g. (2 Pa)	0.09 in. w.g. (22 Pa)
600 cfm (283 L/s)	0.01 in. w.g. (2 Pa)	0.12 in. w.g. (30 Pa)
700 cfm (330 L/s)	0.01 in. w.g. (2 Pa)	0.16 in. w.g. (40 Pa)
800 cfm (378 L/s)	0.01 in. w.g. (2 Pa)	0.21 in. w.g. (52 Pa)

air fan, and the direction in which the system design point (A) moves depends on the position in which the air shaft pressure sensors are installed. If these sensors are installed close to the fan outlet, the system design point (A) will move toward the left horizontally, as shown by the dotted line (2). If there is no system change, the system design point (A) will move toward the left lower corner along the system curve, which is very similar to the total efficiency curves.

In this case, the sensors are installed two-thirds of the way downstream in the primary air shaft, at which point the static pressure is maintained at the predetermined value. Therefore, at reduced speed when the airflow rate is lower than the design rate, the static pressure is also lower; since the closing of CAV terminal units is a form of system change, the system design point (A) moves toward the left lower corner with a slope less steep than that of the system curve, as shown by the solid arrow (3).

It can be seen that the case represented by the solid arrow (3) is more economical and less likely to move the system design point (A) to an unstable range than the cases represented by the dotted lines (1) and (2).

Exhaust Air Fan Similar to primary air, when only one of the typical floors is occupied, the output flow rate of the exhaust air fans is only about 2.9% of the designed maximum flow rate, as can be calculated from the numbers in Tables 1 and 2. With reference to the fan curve in Figure 9, when the static pressure sensor in the exhaust air shaft reduces the fan motor speed through the DDC and adjustable-frequency inverter, both the airflow rate and static pressure at the fan outlet are reduced and the system design point (A) moves toward the left lower corner, as shown by the solid arrow (3).

Supply Air Fan At reduced airflow rates, the action of the supply air system control is similar to that

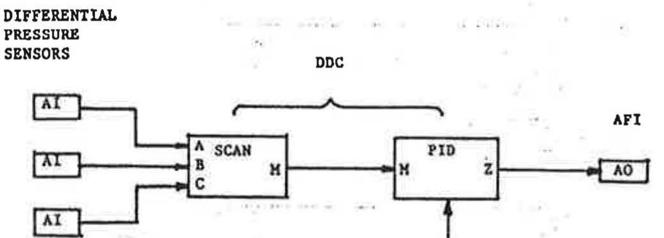


Figure 8 *Block diagrams for fan and pump speed control*

TABLE 4
Mimimum Efficiencies Of High-Efficiency Motors

Output	Efficiency
3 hp (2.2 kW)	85%
5 hp (3.7 kW)	87%
7.5 hp (5.6 kW)	88%
10 hp (7.5 kW)	90%
15 hp (11.2 kW)	90%
20 hp (14.9 kW)	91%
25 hp (18.7 kW)	92%
30 hp (22.4 kW)	92%
40 hp (29.8 kW)	93%
50 hp (37.3 kW) and higher	94%

of the primary and exhaust air systems described above, although the system change is due to the closing of VAV terminal units rather than CAV terminal units. With reference to the fan curve in Figure 9, when the differential pressure sensors in the cold and hot supply air ducts reduce the fan motor speed through the DDC and adjustable-frequency inverter, both the airflow rate and static pressure at the fan outlet are reduced and the system design point (A) moves toward the left lower corner, as shown by the solid arrow (3).

Secondary Chilled Water and Heating Water Pumps With reference to Figures 3 and 4, respectively, for the secondary chilled water system and the secondary heating water system, differential pressure sensors DP-1, DP-2, and DP-3 are installed across the nearest load, the load two-thirds of the way downstream, and the last load, respectively. When the total load is below designed full load, these sensors transmit analog input signals to the DDC, which selects the lowest of them and in turn sends an analog output signal to the adjustable-frequency inverter to reduce the pump motor speed so that the secondary chilled/heating water system pressure is maintained at the predetermined value.

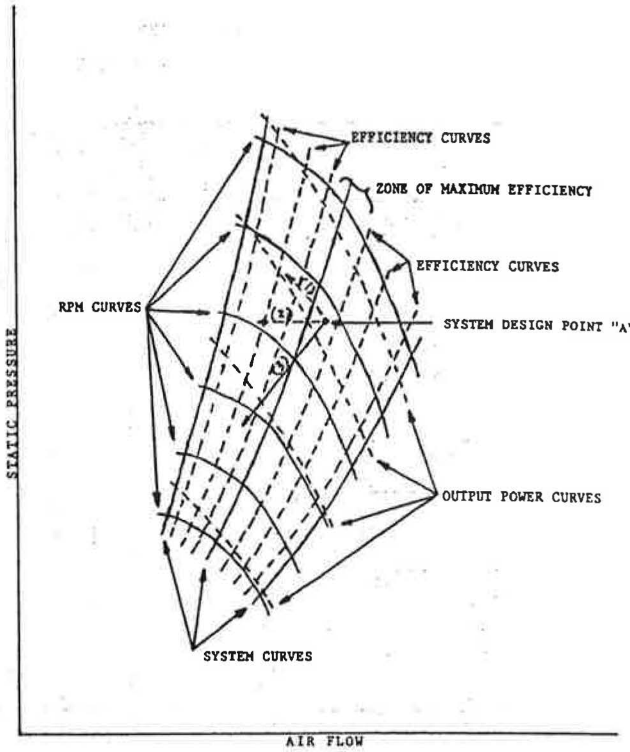


Figure 9 *Typical fan curve at reduced speed*

Various aspects of variable-speed pumps driven by adjustable-frequency inverters, including that of energy conservation, have been discussed in recent ASHRAE publications (Ahmed 1988; Albern 1986; Rishel 1985, 1988; Zell 1985). Therefore, the hydronic systems are not analyzed here. Anyway, at reduced pump speed, both the water flow and differential pressure across the pump inlet and outlet are reduced and the system design point moves in a way similar to that of the fans.

CONCLUSION

From the foregoing study of a current and practical case, it is seen that VAV systems with AHU—the fans of which are driven by "squirrel cage" motors controlled by adjustable-frequency inverters—are viable and economical alternatives for high-rise office buildings in tropical climates.

In this case, outside air volume is kept constant by means of primary air units (PAU) and constant-air-volume terminal units (CAV boxes), while building pressurization is controlled by the relative adjustment of the primary outside airflow rate and the exhaust airflow rate on each floor.

Apart from using high-efficiency motors and adjustable-frequency inverters, which are energy efficient, this design also uses outside air free cooling for night purge as an energy conservation measure.

REFERENCES

Ahmed, O. 1988. "Life-cycle cost analysis of variable-speed pumping for coils application." *ASHRAE Transactions*, Vol. 94, Pt. 1.

Albern, W.F. 1986. "Variable flow pumping?" *ASHRAE Journal*, June, pp. 34-36.

Rishel, J.B. 1985. "System analysis for a campus-type chilled water system." *ASHRAE Transactions*, Vol. 91, Pt. 1.

Rishel, J.B. 1988. "Twenty years' experience with variable speed pumps on hot and chilled water systems." *ASHRAE Transactions*, Vol. 94, Pt. 2.

Zell, B.P. 1985. "Design and evaluation of variable speed pumping systems." *ASHRAE Transactions*, Vol. 91, Pt. 1.

DISCUSSION

R. Sheng, Senior Engineer, University of Manitoba, Winnipeg, Canada: What is the minimum speed recommended for an inverter? Also, please comment on what kind of problems one encounters if the inverter is running at, say, 20 Hz, and comment on inverter "fault" and tripping.

L. Lo: Minimum recommended speed depends on application. Makers usually recommend 5% of full speed. For VAV applications, minimum speed is generally about 50% of full speed. There is no problem arising, particularly when running at low speed. To avoid inverter "fault" or tripping, one should follow the maker's operating instructions, particularly for various adjustments.