#4354

Building Serv. Eng. Res. Technol. 9(3) 89–98 (1988) Printed in Great Britain

Summary The paper describes a study of variable pitch and variable speed fans in temperature controlled and constant pressure controlled VAV systems using computer simulated building models and weather data.

Comparison of variable pitch fans and variable speed fans in a variable air volume system

C M JOHNSON

The Brian Warwicker Partnership, BWP House, The Street, Ardleigh, Colchester, CO7 7LD, UK†

Received 6 April 1987, in final form 11 January 1988‡

1 Introduction

One of the ways in which building services design has been changing is to make building systems more economic both in initial design and installation and in running costs. Both criteria must be met while still producing the highest standards of comfort within a space. The correct conditions within a space obviously depend upon the room load at any given time; over the years it has become apparent to most energy users that they can vary the flow of conditioned air and water and, to some extent, electricity into a space and vary the energy consumed, thus saving money.

Perhaps the most effective way to the economic and efficient control of a room space is to vary the volume of air supplied to that space according to its heating or cooling loads. These loads may be machine, solar, human or even animal depending upon the space application. The mass flow of air has an effect on the main plant both through initial sizing and through running times, so that as the air flow is varied, not only is the fan using different amounts of electricity, but so is the main plant. This has the beneficial effect of saving energy in all forms and reducing the chemical dosage costs in water systems.

The type of system which varies air flow with load is known as a variable air volume system or VAV for short. In this system the fans are the prime movers and they fall into two distinct categories, variable pitch and variable speed fans. It is the aim of this paper to identify the suitability of each type of variable volume fan in a VAV system.

2 Space simulation

In a study of this type, it is obviously a good idea to use an existing system and replace the fans when each stage of the observation is complete. This is always a difficult task because it is not always easy to persuade a user to allow one to 'mess around' with his building over a two-year period while readings are taken and fans changed. It is also expensive in both time and equipment costs.

Space simulation by computer model is the quickest and most economical solution and after approaching several companies we collaborated with a well-known company which is part of a prominent building services consultancy. The company provided a space which could be studied in all respects through their special building services simulation program. This program allows changes in main plant without changing the overall system and shows the effect of the changes on the space both in air change rate and running cost, allowing a designer to optimise his design before it is installed. Energy management systems may also be simulated and their effects studied. For this investigation the company provided one floor of an office block which was already on computer file.

There are indications of a trend towards sublet floors of large office blocks and other buildings having their own self-contained and metered main plant systems, removing the requirement for large central plant. To achieve a 'typical' floor simulation, we decided that a mid floor should be used to focus the effect of the fans and to remove extra roof gains and floor losses. The floor dimensions are given in Figure 1.

The area was sub-divided into five zones: Zone 1—Central Zone—30 m long × 10 m wide; Zone 2—North Zone—5 m long × 20 m wide; Zone 3—East Zone—40 m long × 5 m wide; Zone 4—West Zone—40 m long × 5 m wide; Zone 5—South Zone—5 m long × 20 m wide. The building is located in the London area with a north–south orientation, although the floor in question is not affected by shading from other buildings.

According to the computer output the space is a standard building structure comprising concrete cladding external walls and 70% double glazing of the absorptive type. To separate the zones, internal partitions made of plasterboard with glass fibre insulation are used. The whole structure

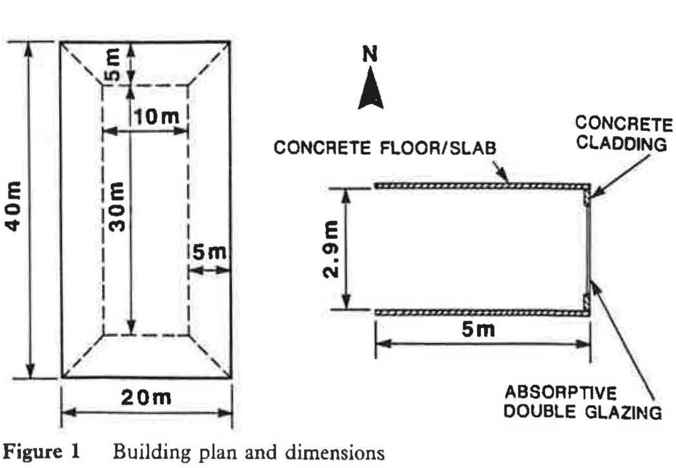


Figure 1 Building plan and dimensions

† The author submitted the paper while at Myson Fans Limited, Colchester.

‡ The paper was presented at the 1987 CIBSE Technical Conference

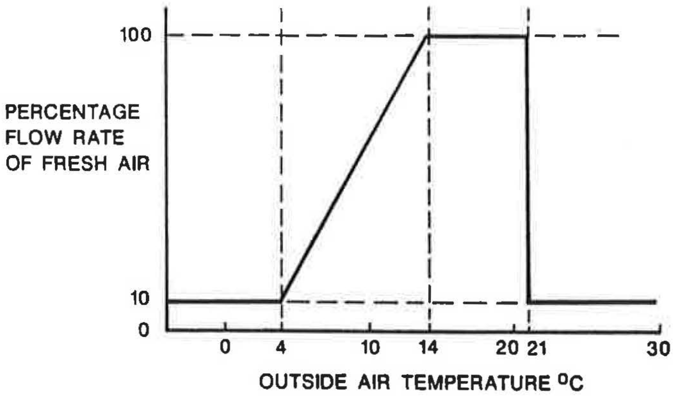


Figure 2 Recirculation control diagram

then conformed to the old GLC section 20 FF regulations. Internal design conditions were 21°C at 50% RH. Heat gain and heat loss calculations indicated that the maximum load of the system would be 70 kW.

Casual gains from lights were 20 Wm^{-2}, from machines 20 Wm^{-2}, from people 10 Wm^{-2} sensible and 4 Wm^{-2} latent. System on time was 0800; system off time was 1900; working 5 days per week.

The computer calculation for heat gains indicated that a peak air flow of 6.96 kg s^{-1} would be required at a constant 14°C with air density at 1.2 kg m^{-3}. This converted to 5.8 m^3s^{-1}. To minimise plant load, a standard recirculation system was employed using 90% recirculation air and 10% fresh air controlled by an optimiser to vary the percentage mix of the air according to outside air temperature as shown in Figure 2. To further reduce air plant size and to maintain a minimum temperature during unoccupied hours, a low-temperature hot water perimeter heating system was introduced to offset fabric heat losses.

From a duct sizing programme, the maximum static pressure for the fan to overcome was 1300 Pa. The fan duty was therefore 5.8 m^3 s^{-1} at 1300 Pa static pressure.

Having sized all the main plant using standard CIBSE methods, the weather data for the full year were introduced to the program and the system model input as shown in Figure 3 by systems programmers. Each number indicates a node or reporting point which has significance in the program.

To give accurate energy use figures, it is necessary to program the total unit efficiency of a fan rather than just its fan efficiency published on the performance graphs. To do this requires a detailed knowledge of motor performance under varying loads. Therefore, when the fans were selected, the motor manufacturer provided this information and it was programmed into the computer. At the end of its run, the program normally gives full psychrometric data, node data against time and a summary of energy used by the main plant. For the purposes of this investigation, the supply and extract fans were isolated and reports modified to show their energy consumption.

3 Fan descriptions and control

Both variable pitch and variable speed fans reduce their power consumption with reduced air flow, as can be seen from their performance envelopes (Figure 4). Other methods of varying air flow such as variable inlet guide vanes, damper bypass control or disc control are not included as they generally reduce volume flow by increasing the resistance on the fan and increasing power consumption or, in the case of damper control, bleed conditioned air into the outside atmosphere; this is wasteful.

3.1 Variable pitch fans

A variable pitch fan is an axial fan which is capable of changing its pitch angle while it is rotating at standard speeds. This change in geometry allows the fan to vary its characteristic constantly to suit system requirements.

The variable-pitch-in-motion fan (Figure 5) achieves this change by a central beam mechanism connected to an actuator mounted on the fan casing.

The actuator receives a signal from the system controls and

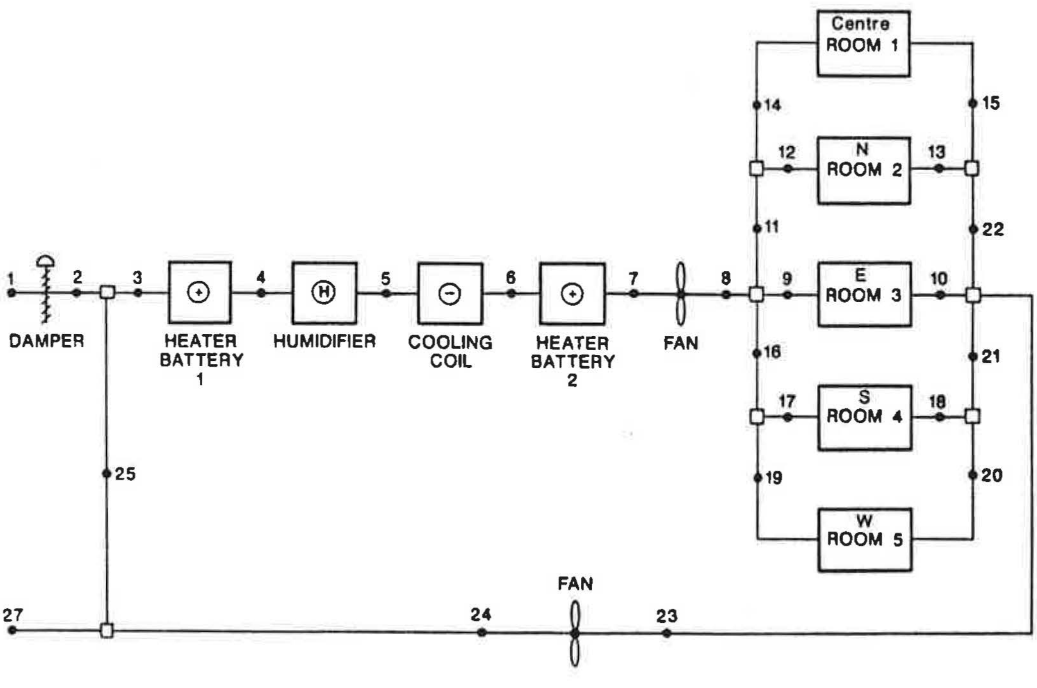


Figure 3 System schematic

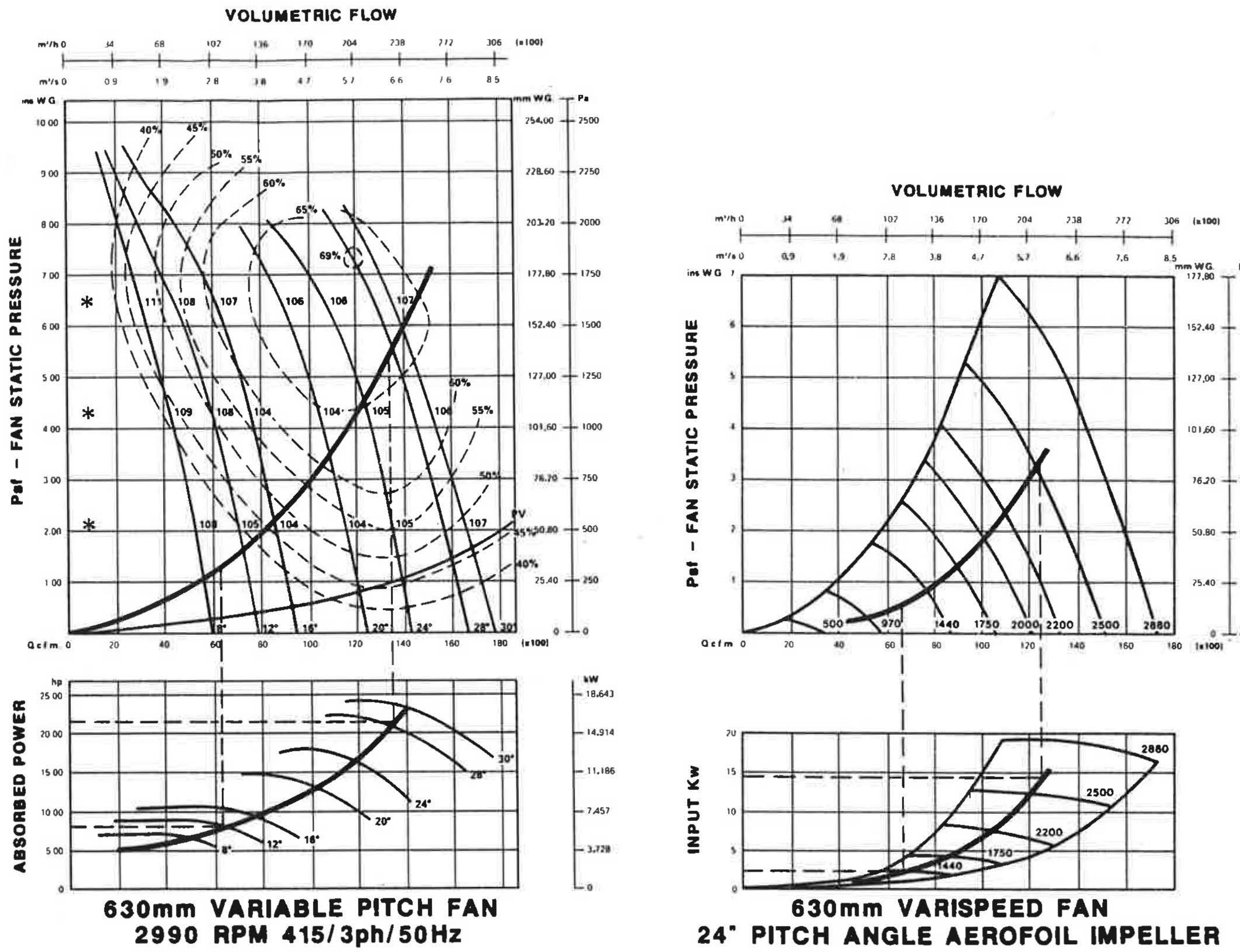


Figure 4 Typical performance envelopes for variable pitch and variable speed fans showing power reduction with flow reduction

moves the beam either backwards or forwards depending on the signal. The beam is pivoted at the bottom of the casing and is attached to the hub or impeller by a mechanical link on the fan centre line.

The link is attached to a control disc via a push-pull bearing. The control disc slides along a specially coated low-friction shaft and has a groove around its maximum circumference into which the arms of carrier units are connected.

These carrier units are fixed to the wing roots to turn the linear movement of the disc into a rotary motion at the wing, so that as the beam moves foward, the pitch angle of the blades increases, increasing air flow. The forces on the controller are reduced when a lower pitch angle is required because the centrifugal and aerodynamic forces on the wings assist it to reduce pitch angle naturally.

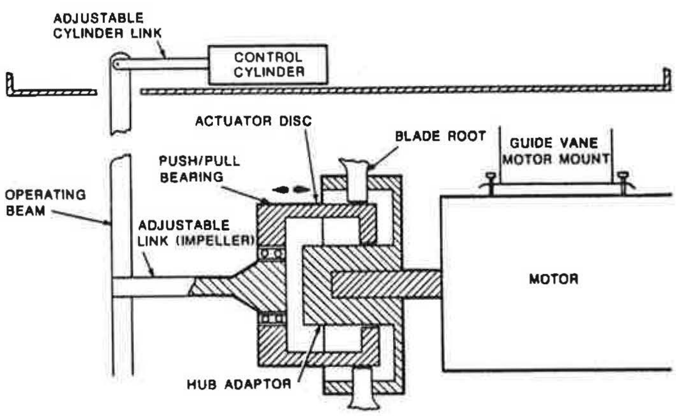


Figure 5 Layout of variable pitch fan (schematic)

The motor is mounted directly to the impeller and is cooled by the airstream in the duct flowing over it. It is also mounted using downstream guide vanes which increase the fan's pressure development and straighten the air flow after the impeller.

The fan chosen for this simulation was a 630 mm diameter, nine-wing unit operating at 2990 rpm and using a 160 frame size air stream rated motor.

As can be seen from the performance curve (Figure 6) this fan has an optimum efficiency point where the wing design, pitch angle, speed of rotation and motor size are at their peak performance. Thereafter, the fan efficiency reduces at all volumes and pressures. To input the fan characteristics into the computer, the motor efficiency versus load characteristic was required. The motor manufacturer provided this information and Figure 7 was plotted and applied to the fan performance.

3.2 Variable speed fans

There are several methods of varying fan speed:

Inverter drive

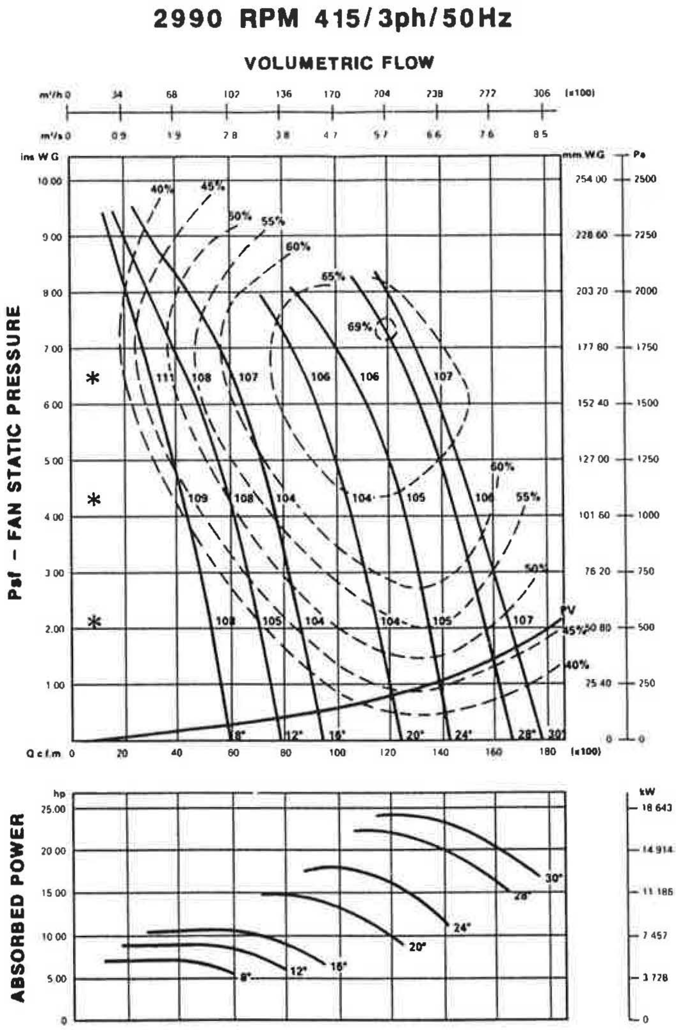


Figure 6 Performance chart for fan type IAF063-21A

Eddy current drive

Reluctance drive

Variable pulley and belt drive

Thermistor drive.

The constant speed motor driving variable ratio pulleys (Figure 8) was chosen as having the best compromise between drive efficiency and capital cost. This type of drive has been shown to be 93% efficient over most of its range. The fan has a standard fixed-pitch impeller in a casing driven by the variable pulley drive. The bottom pulley is spring loaded to maintain belt tension and change its diameter. The top pulley is driven by an electric actuator which moves its two halves closer or further apart depending on the signal from the system. As the top pulley halve moved closer together, the diameter increases. This tensions the belt which in turn forces the two halves of the bottom pulley apart against the spring, reducing its diameter. In this way the speed of the impeller is increased.

The unit chosen was the 630 mm diameter standard aerofoil impeller unit operating at 2990 rpm using a 160 frame size motor, totally enclosed and fan cooled, mounted externally on the fan casing.

The principal attribute of the variable-speed drive is that fan efficiency on one point of the impeller performance curve is maintained throughout its speed range as it follows the fan laws.

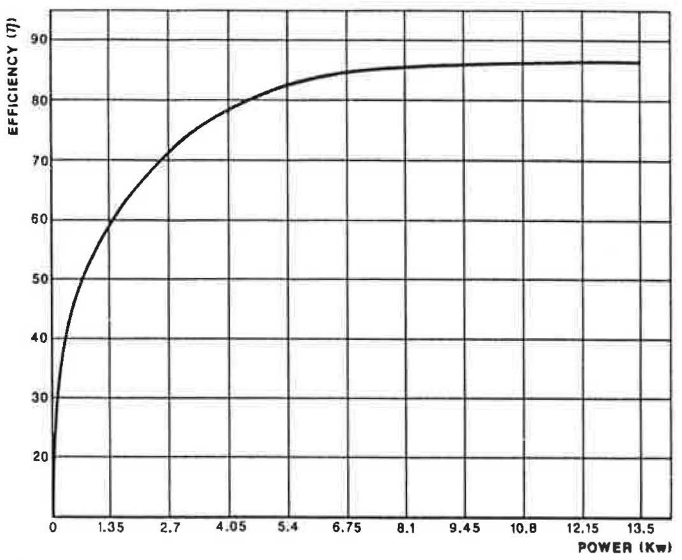


Figure 7 Motor efficiency versus power for variable pitch fan

The unique performance envelope is shown in Figure 9. The motor efficiency versus load graph to program total unit efficiency was plotted and applied to the fan performance (Figure 10).

4 System type and control

4.1 Direct temperature control

As the name implies, the fan is controlled using a thermostat mounted in the main return air duct (Figure 11).

Each zone has a temperature sensor mounted in its own branch return air duct which operates a damper in the supply air duct. As the heat gain in each zone increases, the damper in that zone opens, allowing more cooling into the zone. As the sun moves around the building, the dampers in each zone open or close in accordance with the room load. The sensor in the return air duct senses the average room temperature and controls the fan to supply enough cooling air to maintain that average temperature. To ensure that each zone is properly serviced, the sizing of ductwork is critical since each branch must have the same resistance as the index run. In this way, the system is self balancing, because the

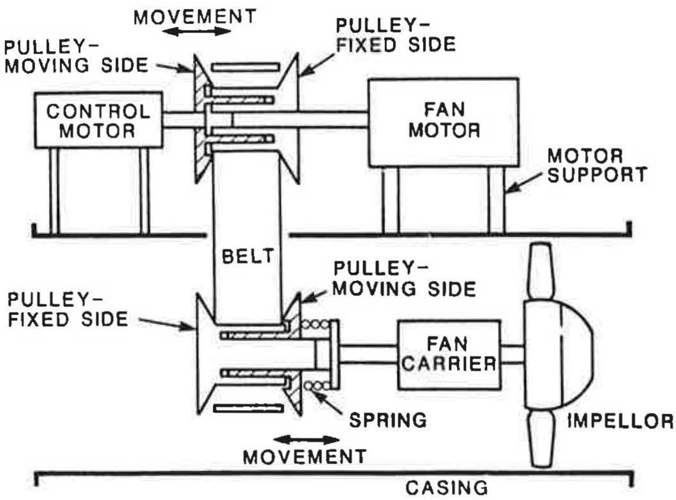


Figure 8 Layout of variable speed fan (schematic)

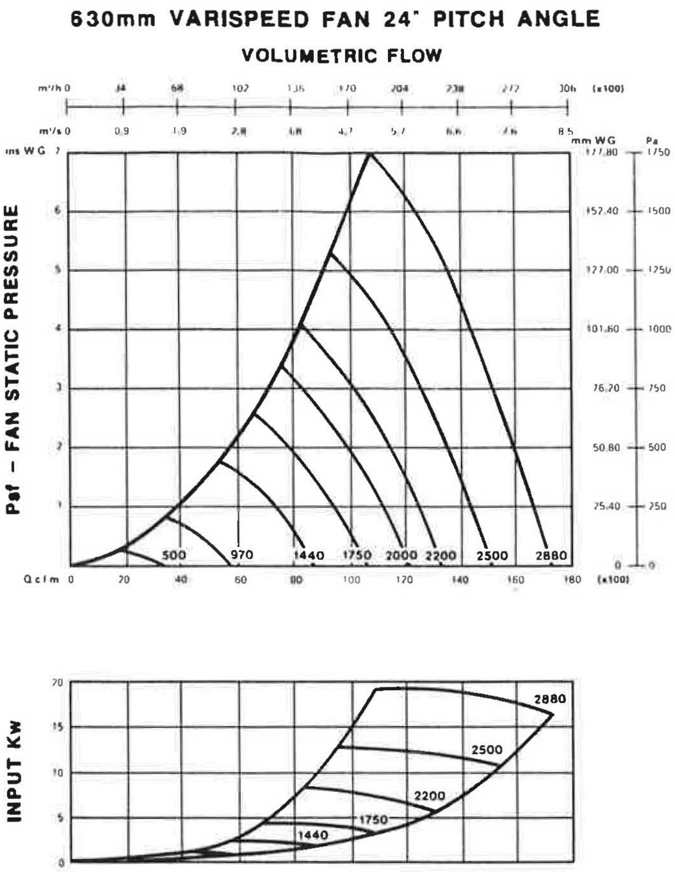


Figure 9 Performance chart for fan type 24FVVS

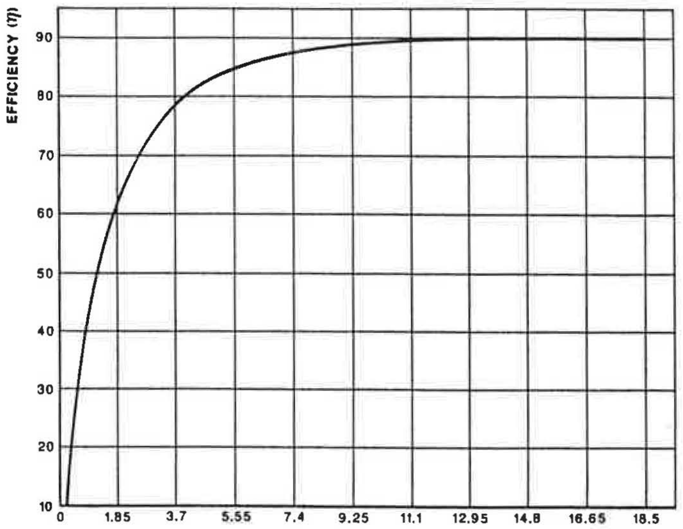


Figure 10 Motor efficiency versus power for fan type 24FVVS

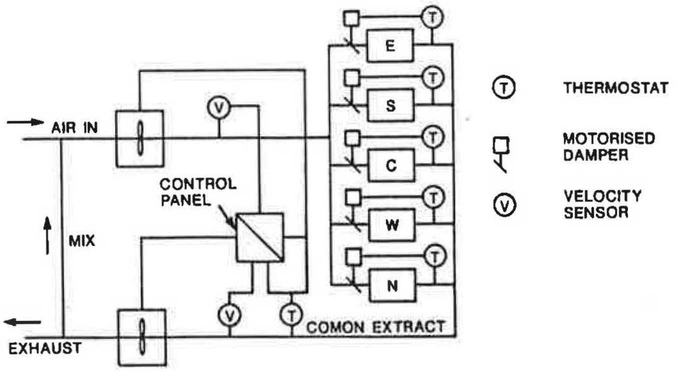


Figure 11 Fan control using return air thermostat

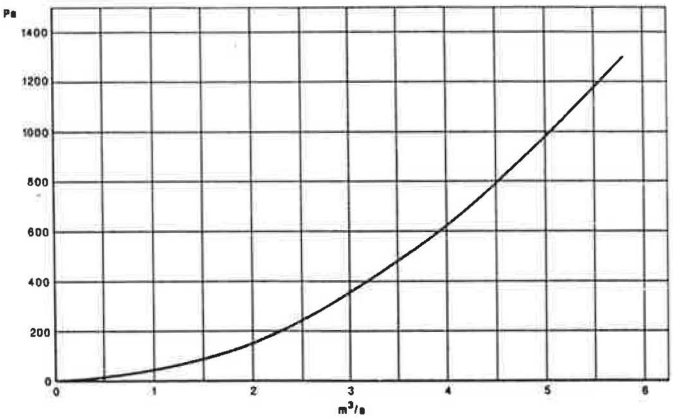


Figure 12 Typical temperature control system line

equivalent length of ducting remains constant for every 'damper open' combination. The system line then follows the curve $Q \propto p^2$ (Figure 12) because the varying volume of air passes through ducting of a constant equivalent length.

In general terms this is known as a 'pressure dependent' system because the dampers have no independent volume flow sensors to compare with the temperature signals emitted from the zone sensors and may only pass a proportion of the air supplied by the system which is based on average temperature.

4.2 Constant pressure control

Constant pressure is maintained in the supply air ductwork (Figure 13).

Each zone has its own volume control terminal which can sense air flow rate into the zone and compare the flow with room temperature to maintain the correct conditions.

The terminal unit shown in Figure 14 is typical of this type of unit. The individual control on each box may be preset to provide a maximum or minimum air supply into a space and may be adjusted on site to suit local conditions.

The ductwork system to the units is sized by standard methods and balanced for maximum air flow. It is also sized with the damper of the box at the index run fully open, since this will be the most economical way to run the system.

As the boxes close down, in accordance with zone loads, the

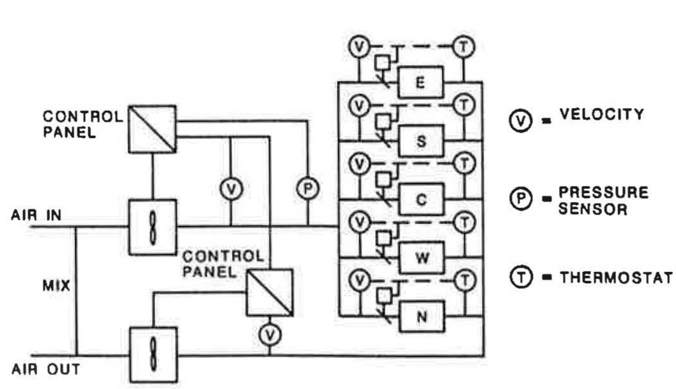


Figure 13 Fan control using constant pressure sensor

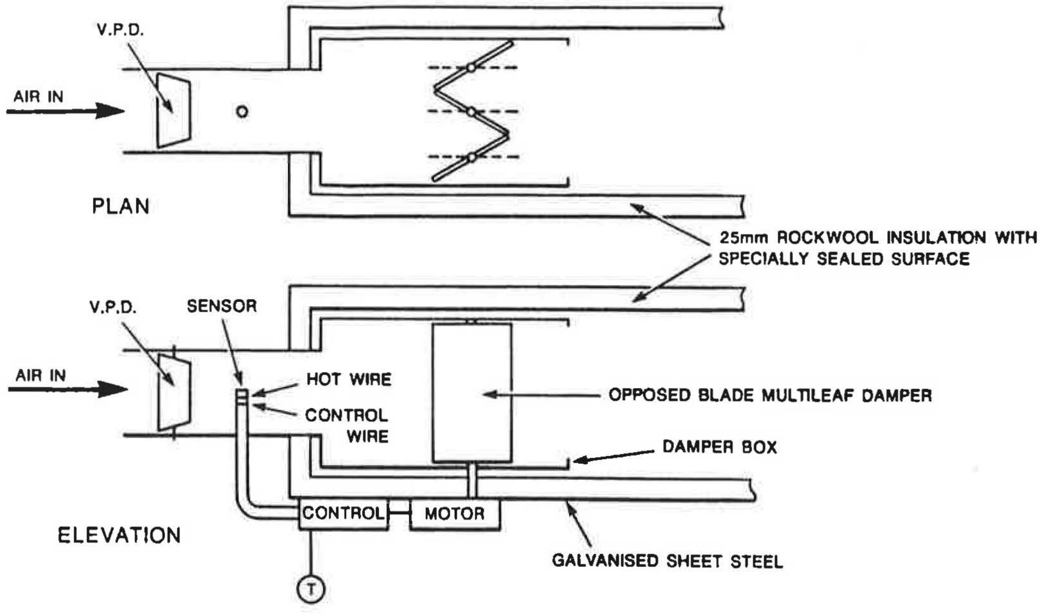


Figure 14 VAV terminal unit layout (schematic)

pressure in the system tends to increase. The pressure sensor in the system senses this and sends a signal to the fan to reduce air flow following the curve of Figure 15. As the flow reduces, the box dampers open until the system is operating at the balanced condition where the index box is fully open and the fan is producing the correct flow rate.

This system is also known as 'pressure independent' since each terminal unit senses and controls air flow independently.

5 Results

5.1 General

In each test run the building structure and layout remained unchanged as did the weather information used. The only component changed was the fan. Obviously direct drive fans, with their motors in the air stream, transfer more heat into the conditioned air than belt drive units. The effect of this difference was found to be negligible on the overall performance of the main plant as it produced a less than 1% change in air flow rate at any given time. The heater and cooler batteries were able to compensate for this and cancelled out the difference.

In all other aspects, the psychrometric requirements at any given time were kept constant so that main plant size and running costs excluding the fans did not change. The average air flow requirements for each system followed the curve of Figure 16, with the peak requirement occurring at 1600 h on each test day. The trough was caused by the lunch hour and the afternoon peak by the building lag.

Once the flow rate was established, the input power for each flow rate was calculated using the fan and motor information previously described and a running cost determined by using a cost factor of £0.05 per kilowatt hour. In this way an annual running cost was produced for each type of fan and each system type.

5.2 Temperature control

As previously discussed, the system follows the curve $Q \propto p^2$. This was applied to the two fan performance curves previously shown and the power input graphs shown in Figure 17 were drawn taking account of motor and belt efficiency and Table 1 compiled.

5.3 Constant pressure control

In order to operate the system as previously described, the constant pressure maintained was required to be equal to the index run pressure. This was easily accepted by the variable pitch fan, but the variable speed fan was not able to meet the minimum air flow requirements and stalled at a very early stage (Figure 18).

A new variable speed fan was required to meet the duty and the only fan available was a mixed flow type.

The fan chosen was the 750 mm diameter unit with the same

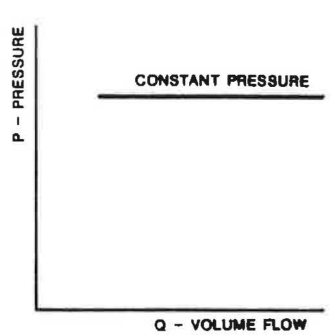


Figure 15 Typical constant pressure control system line

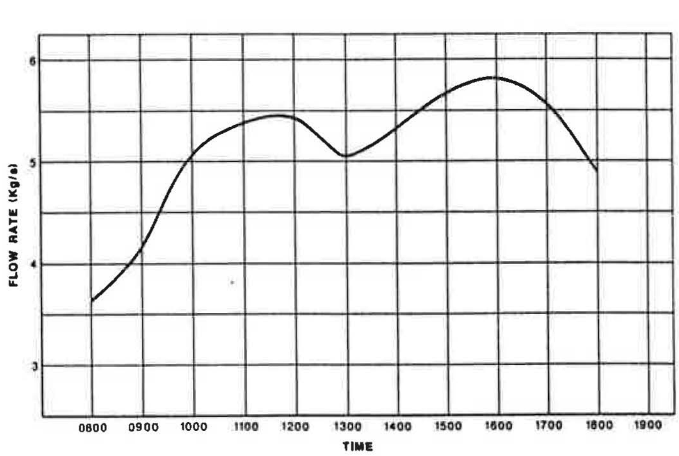


Figure 16 Typical time evolution of air mass flow rate

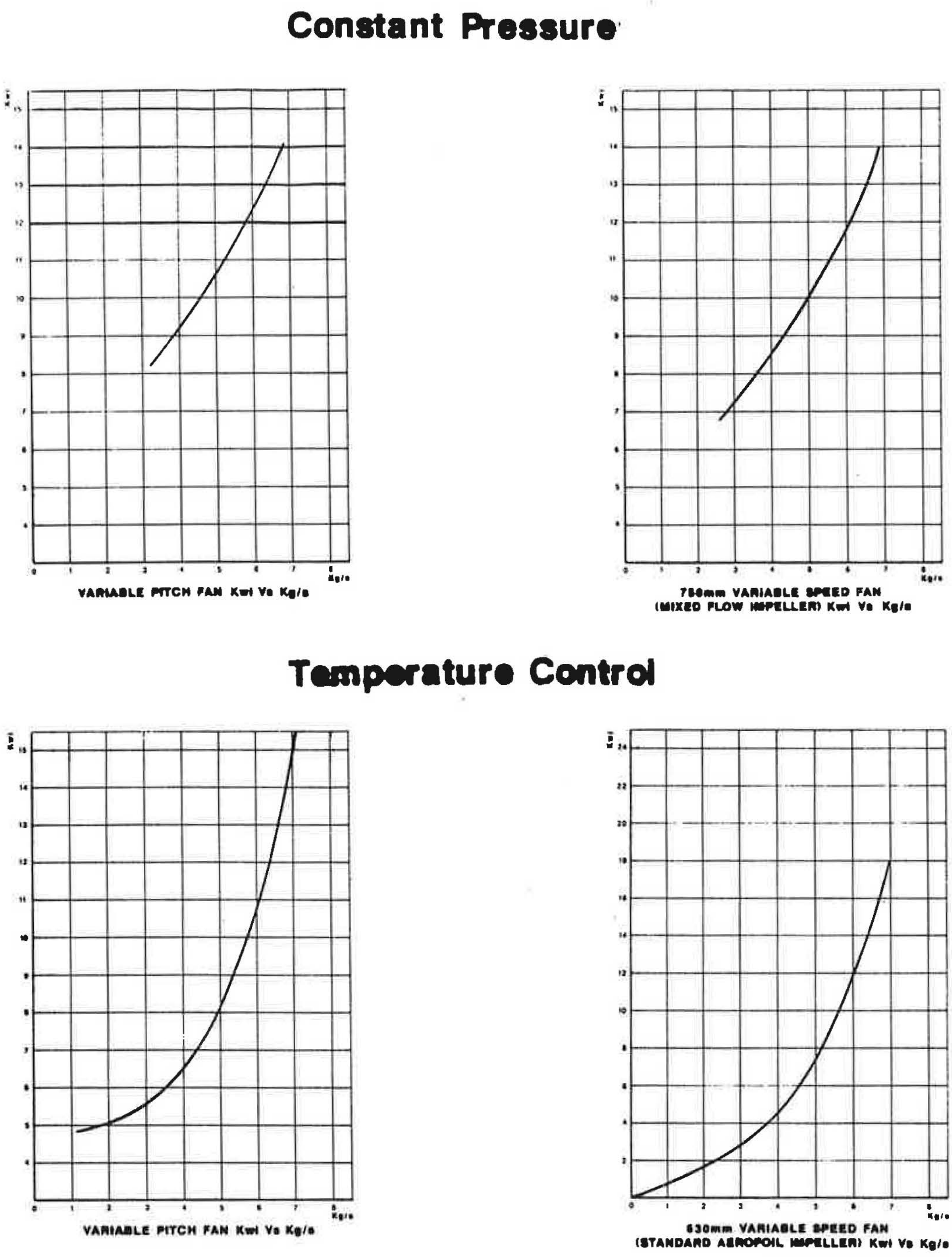


Figure 17 Power input versus flow rate for variable pitch and variable speed fans

type of belt drive and an 18.5 kW two pole 160 frame size motor. Its overall performance envelope is shown in Figure 19. A performance chart was drawn up and running costs established as before. The results for both system types and each fan are summarised in Table 2.

6 Conclusions

As there is a negligible change in mass flow rate between fans, and each fan reacts in accordance with space demands, the major comparison is between capital and running costs.

The cheapest system to operate appears to be the temperature control system using the variable speed fan. However this is an extremely difficult system to build and commission accurately. This is a very slow-acting system which could allow local hot and cold spots to appear causing the fan to hunt around the set points. It is also a fixed system which cannot easily cope with changes in use or load. The variable pitch fan, although cheaper to buy, is more expensive to run. A projection of the capital and running costs of both fans would indicate that they are equal after about 4 years in service, thereafter the variable speed fan is cheaper.

The constant pressure system, although more expensive to operate, is more responsive and versatile, allowing changes in occupancy and use through its terminal box control. It is more tolerant of installation changes and can react quickly to system changes without hunting. Because of its versatility and ease of installation the constant pressure system is commoner.

The variable speed fan again appears to be the best solution for constant pressure systems. However, the high capital cost is prohibitive and over the same 4-year period the variable speed fan is still over £3500 more expensive than

Table 1 Typical daily power use results for system type CP. Day 10 of May (21 working days).

Time	Fan type			
	Variable pitch		Variable speed	
	Flow rate (kg s^{-1})	Power input (kW)	Flow rate (kg s^{-1})	Power input (kW)
0700	0		0	
0800	3.55	8.58	3.51	7.9
0900	4.09	9.3	4.08	8.65
1000	4.58	10.0	4.57	9.3
1100	4.80	10.3	4.76	9.6
1200	4.95	10.5	4.91	9.86
1300	4.57	9.98	4.53	9.2
1400	4.75	10.21	4.71	9.52
1500	5.22	11	5.17	10.27
1600	5.34	11.21	5.3	10.52
1700	5.11	10.79	5.05	10.1
1800	4.45	9.85	4.42	9.1
1900	0		0	
Total		111.72		104.02
Monthly average	4.68	10.16	4.64	9.46
Daily energy input (kWh) × £0.05 kWh^{-1}	£5.586		£5.201	
Monthly average power cost	£117.31		£109.22	

the variable pitch fan. When the capital and running costs of both fans are projected further, it becomes apparent that the variable pitch fan is still cheaper over 10 and 20 year periods. Costs are equalled after a 38 year period, which is an unrealistically long lifetime.

Since the variable pitch fan was able to cope with both systems it is obviously the more versatile. The variable speed fan is generally cheaper to run and is better suited to fixed systems with temperature control. Table 3 suggests the sort of systems suited to each fan.

Table 2 Summary of fan running costs (£)

Month	Control mode				Working days
	Temperature		Constant pressure		
	Variable pitch	Variable speed	Variable pitch	Variable speed	
Jan	74.26	49.56	105.42	97.69	21
Feb	72.43	50.80	102.46	95.21	20
March	84.86	60.58	119.05	110.59	23
April	75.94	56.70	105.64	98.00	20
May	87.36	71.35	117.31	109.22	21
June	97.46	91.20	125.29	116.81	21
July	96.81	85.35	127.05	107.70	22
Aug	102.29	96.63	131.23	122.58	22
Sept	86.32	66.73	118.72	110.29	22
Oct	86.80	63.71	121.10	112.44	23
Nov	81.52	58.58	114.42	106.42	22
Dec	66.78	47.43	93.38	86.79	18
Total	1012.83	798.32	1381.00	1273.74	255
Average daily cost	3.972	3.13	5.416	4.995	
Average power input (kW)	7.22	5.69	9.85	9.08	
Average flow rate (kg s^{-1})	4.52	4.50	4.50	4.45	
Capital cost	2850	3650	2850	6850	

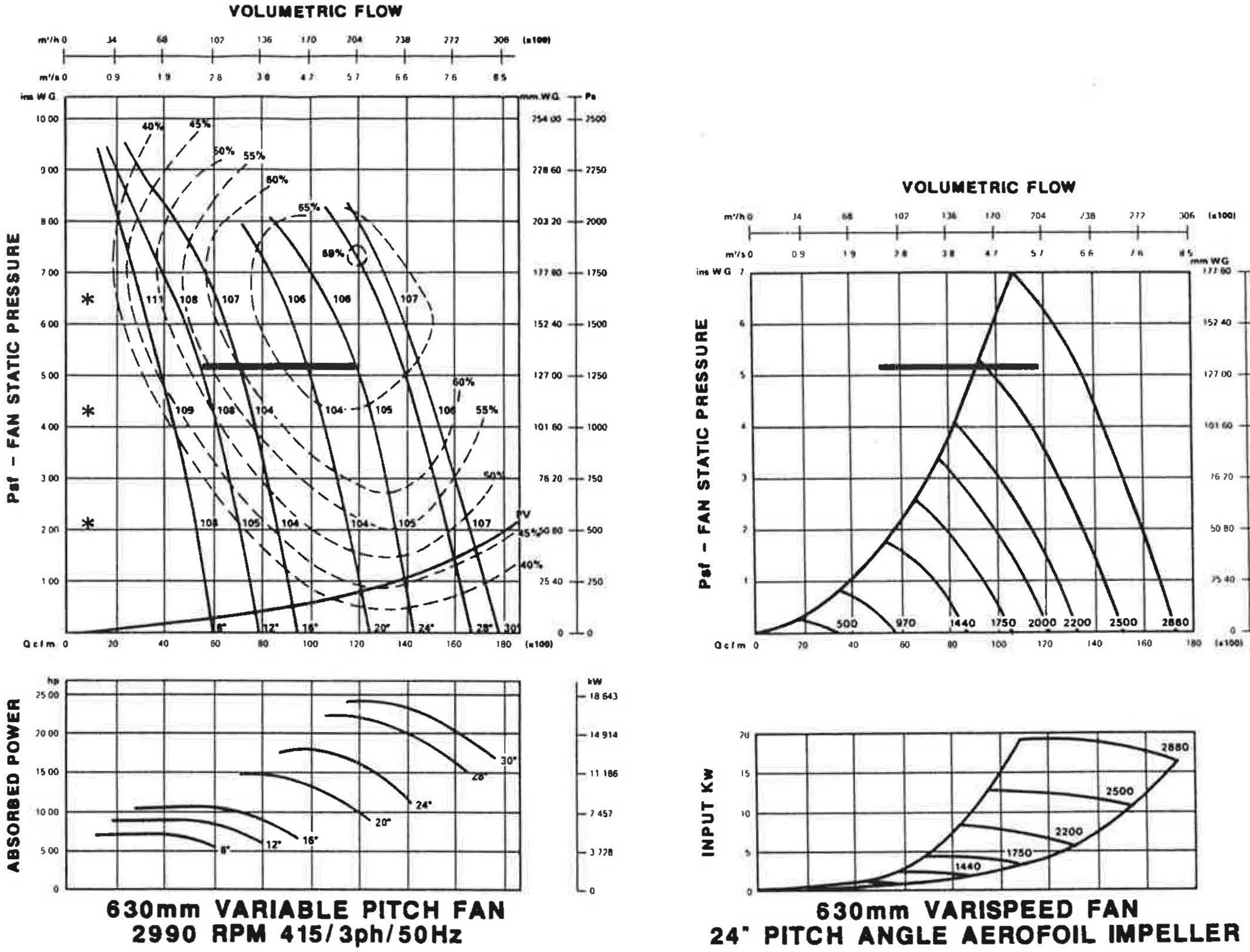


Figure 18 Constant pressure curve on variable pitch and variable speed fan performance charts

Table 3 Applications of variable volume fans

Variable speed	Variable pitch
Process drying	Office blocks
Process cooling	TV studios
Wind tunnels	Shopping centres
Plantroom ventilation	Computer suites
Swimming pool ventilation	Laboratory ventilation

7 Addendum on centrifugal fans

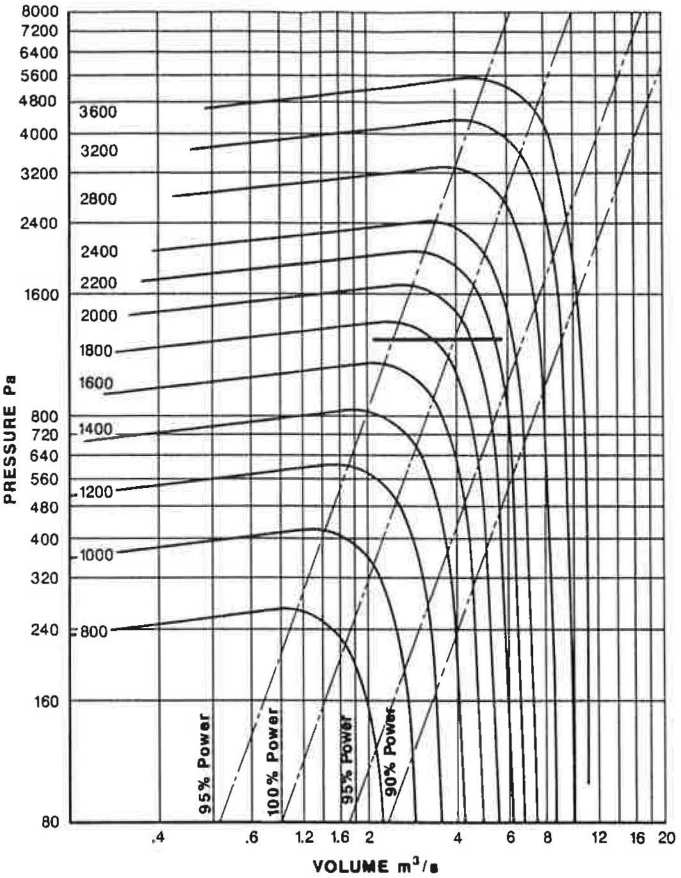
Centrifugal fans are more popular and more widely used than axial fans in supply air handling units.

7.1 Inlet guide vane control

Automatically variable guide vanes, controlled by system controls, are fitted to the inlet of the fan. These open or close to vary the volume of air entering the fan and hence

Table 4 Running cost summary for centrifugal fans

	Control			
	Temperature		Constant pressure	
	Inlet guide vane	Variable speed	Inlet guide vane	Variable speed
Capital cost (£)	3731	3997	3731	3997
Average flow rate (kg s^{-1})	4.52	4.50	4.50	4.45
Average power input (kW)	12.13	14.184	12.954	15.04
Average cost per day (£)	6.67	7.8	7.125	8.27
Running cost per year (£)	1700.85	1989.31	1816.8	2109.36



SPEED RPM	3600	3200	2800	2400	2200	2000	1800	1600	1400	1200	1000	800
Kw. ABS @ 100%	51.3	36.1	24.2	15.2	11.7	8.80	6.41	4.50	3.02	1.90	1.10	0.56
SOUND dBW	104	101	98	96	92	90	87	85	82	78	73	68

Figure 19 Variable speed mixed flow fan performance chart

the system characteristic. The fan motor is generally belted to the impeller and runs at a constant speed.

7.2 Variable speed

A standard centrifugal fan is belt driven, generally using an eddy current drive motor. The motor is again controlled with the rest of the system.

The fans chosen were:

- Inlet guide vane type 750 mm diameter SISW, backward inclined belt driven for operating at 1372 rpm with a BCP D160L 15 kW motor running at 1440 rpm.
- Variable speed controlled 750 mm diameter SISW backward inclined belt driven for operating between 1355 and 970 rpm with a TASC Drives 180 cd 4/42 18.5 kW motor.

7.3 Results for centrifugal fans

The same procedure was applied to the fan absorbed power as is previously described in sections 3 and 4 above, taking into account belt losses and motor efficiences to obtain kW input into the motor.

7.4 Conclusion on centrifugal fans

Centrifugal fans are widely used because of their high pressure capabilities and quietness as compared with axial fans.

From Table 4, it would seem that the inlet guide vane control on this particular fan performs better than the variable speed control using an eddy current type motor. However, the same impeller and fan size can be used on both temperature controlled and constant pressure controlled systems for both fan types.

In comparing Tables 2 and 4 it would appear that with the possible exception of the variable speed mixed flow fan, the axial fans are more economical.

Acknowledgements

The author thanks the following for their assistance.

Facet, St Albans (part of the Oscar Faber Group)

Messrs S Irvine and P Concannon for computer simulations using the APACHE programs.

Myson Fans, Colchester

Messrs R Sexton, G Godden, M Long and A Anglesea for information on fans as follows; variable pitch fan—IA4F063-2IA; variable speed fan—24FVVS 2-pole; variable speed mixed flow fan—750 mm Vary Conflow.

Myson Copperad, Milton Keynes

Messrs R Chabra and J Collin for information on system and terminal units utilising the Myson Buensod BTU range of units.

Brook Crompton Parkinson

Mr T Hall for information on motors as follows: variable pitch fan—DNV160LMAP; variable speed fan—D160Ma; variable speed mixed flow fan—D160L.

Matthews and Yates

Mr I Matts for information on 750 VCB centrifugal fans.

Tasc Drives

Mr S Young for information on eddy current motors.