Duct Loops and VAV Modelling and Control

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Abstracts

This paper summarises the results of a now completed three year research programme into the modelling and control of variable air volume air conditioning. Three common VAV boxes, used in standard VAV and displacement ventilation systems, were analysed on a rig and new, more detailed, models developed compared to the current HVACSIM+ model. These models were then used to analyse duct loop systems. Although not extensively used in Europe, duct loop systems are shown to save up to 35% in fan power compared to conventional radial duct systems. Conventional, static pressure sensor positioning is briefly discussed and the saving from different positions shown from models of synchronous and asynchronous VAV systems. Stability and box authorities are also briefly discussed.

1. Introduction

Variable Air Volume (VAV) systems are one of the most common air conditioning systems used in commercial buildings. The duct work which distributes the supply air from the air handling unit (AHU) to each individual zone has been traditionally based on a radial approach. This is where supply air goes through a main duct run and branch ducts will deliver the air to each zone.

In a large radial VAV duct network, approximately 60% of the total supply fan power required is used to deliver the supply air from the AHU outlet to each room diffuser. Hence the pressure losses through the duct distribution network are substantial. Duct looping has the potential to reduce the pressure requirements to deliver supply air to the individual zones. The cost of a duct loop installation may be cheaper than a radial system(1) as there are fewer fittings required, with more uniform duct size requirements. A typical example of a duct loop system can be seen in Figure 1.

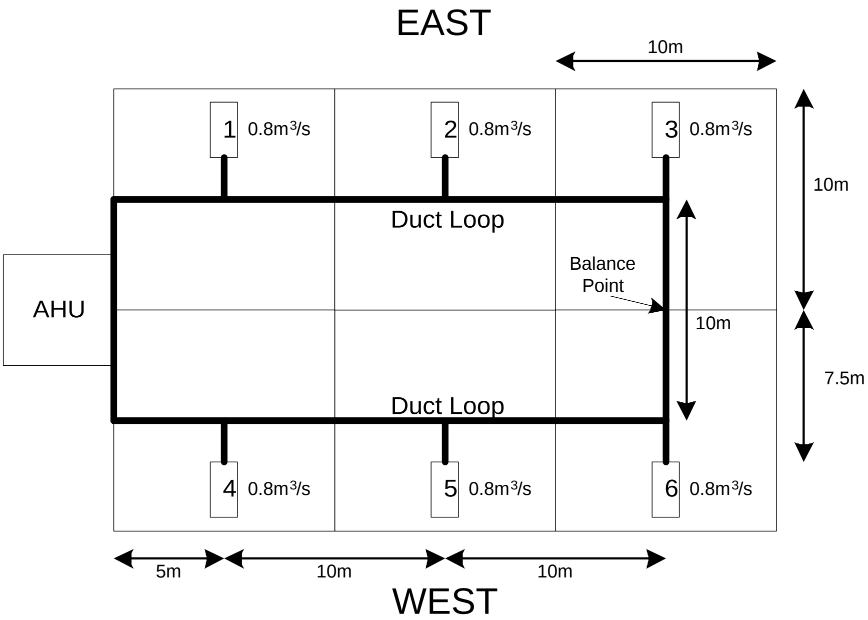

Figure 1: VAV Duct Network (Duct Loop)

2. Building Loads

The design and control of a VAV duct network is dependent in the distribution profile of the building's cooling load acting on the VAV system. The load distribution profile of a typical VAV system depends on the layout of the duct system to its supplying zones. The load distribution profile can be idealised into two categories, a synchronous and an asynchronous load distribution profile(2). This aids understanding and also the maximum savings in fan energy can then be determined.

A synchronous load distribution occurs when the VAV system has been designed to supply a corner of a building over multiple floors. The load on the VAV system varies together as shown in Figure 2. The fan flow rate and power varies widely. The VAV system in this case has to be designed for the sum of the peaks of the building load.

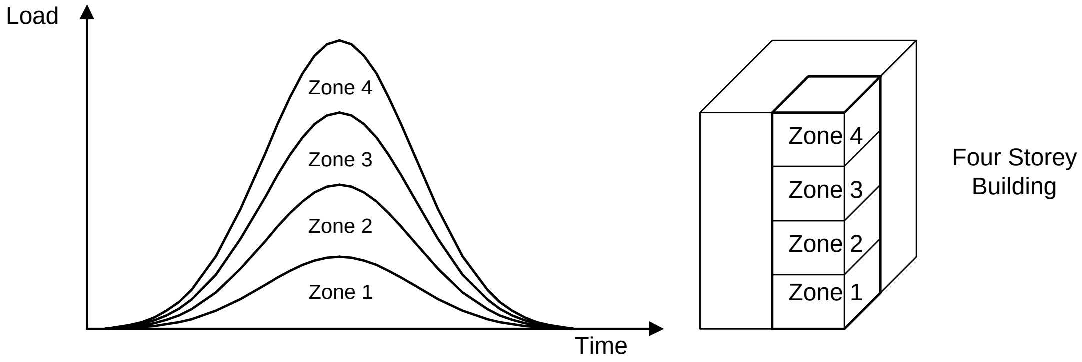

Figure 2: Synchronous load profile with 4 zones

An asynchronous load distribution would be a typical VAV system supplying the entire floor of a building. With perfect diversity, the total load on the AHU is virtually constant as illustrated in Figure 3. In such systems, the fan speed and power variations are small. The savings in such systems are made in the design, capital and running cost i.e. smaller fan and plant.

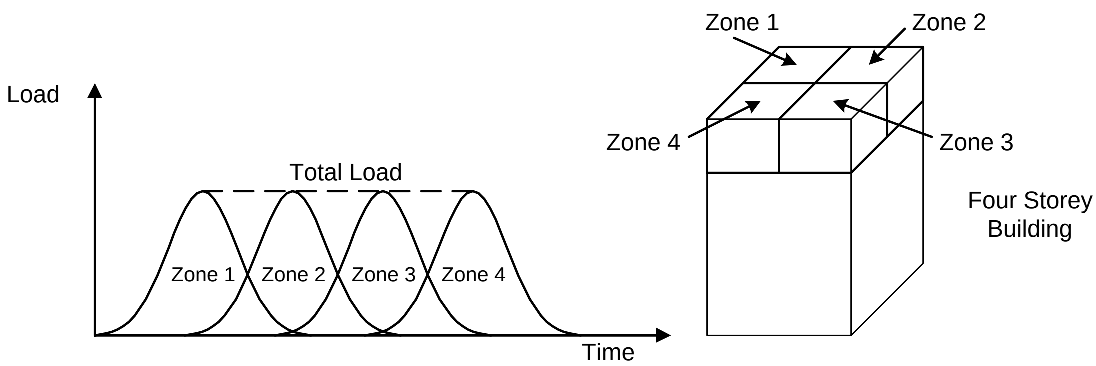

Figure 3: Asynchronous load profile with 4 zones on one floor

Most practical systems will fall between these two idealised profiles which would determine the diversity factor to undertake during the duct sizing design process.

3. VAV Box Models

Three steady state terminal unit models for use in simulation of VAV system have been derived from test rig experiments[3]. These models (one pressure dependent and two pressure independent terminal units) are different from the damper only models being used by well established simulation packages like HVACSIM+ or TRNSYS. The total pressure loss across the terminal unit can be expressed in terms of a pressure loss coefficient K as shown in Equation 1.

$$\Delta P_T = KP_V \qquad \text{Equation 1}$$

where

ΔP_T Total pressure loss across the terminal unit
K Pressure loss coefficient
P_V Velocity pressure at the inlet of the terminal unit

A brief summary of the terminal unit models' pressure loss coefficients is explained here. Equation 2 shows a time-based pressure dependent terminal unit (TU1) model in which T, the travel time from its fully opened position, gives the relative position of the damper. The maximum travel time T for TU1 from fully opened to its minimum damper position is 39 seconds.

$$\ln K = 1.626 + (9.68 \times 10^{-6} T^{3.36}) \qquad \text{Equation 2}$$

Equation 3 and 4 shows the terminal unit models of two commercially popular pressure independent terminal units, TU2 and TU3 respectively. The damper angle θ range for terminal units TU2 and TU3 are 0° to 61.5° and 0° to 45° respectively. In Equation 3, two indices are used in the model to give a better curve fit to the measured pressure loss coefficient K characteristics. The low order index modelled the terminal unit (TU2) characteristic from 0° to 25° within an accuracy 10%. The higher order index extended the valid range of the terminal unit model up to 50°.

$$\ln K = 0.3575 + 4.37 \times 10^{-3} \theta^{1.77} + 1.74 \times 10^{-17} \theta^{9.95} \qquad \text{Equation 3}$$

$$\ln K = 0.332 + 3.68 \times 10^{-7} \theta^{4.5} \qquad \text{Equation 4}$$

Equation 3 and 4 may be compared with Equation 5[(4)] which shows the a typical opposed blade damper only model. Under similar authority and percentage damper angles, the installed characteristics of such damper only models could differ as much as 65% of total the volume flow. The damper range for Equation 5 is from 0° to 90°

$$\ln K = -1.5 + 0.105 \cdot \theta \qquad \text{Equation 5}$$

4. Radial and Duct Loop Designs

The basic approach to radial duct work design has been well established to size the ducts. Several duct sizing methods include equal friction, static regain and constant velocity. Design methods for duct loop have however not been well known and there are many design variants[(5)]. But the simplest approach is to design a duct loop, which is equally sized to half of the designed system volume flow rate. This approach will also yield greater stability and ease of supply fan control, as will be shown later.

5. System Comparison under Full Load Conditions

5.1 System Design

Twelve configurations of radial and duct loop systems have been looked at for both a small (6 terminal units) and large (24 terminal units) VAV system. A general summary of the layouts can be seen in Figure 4[6].

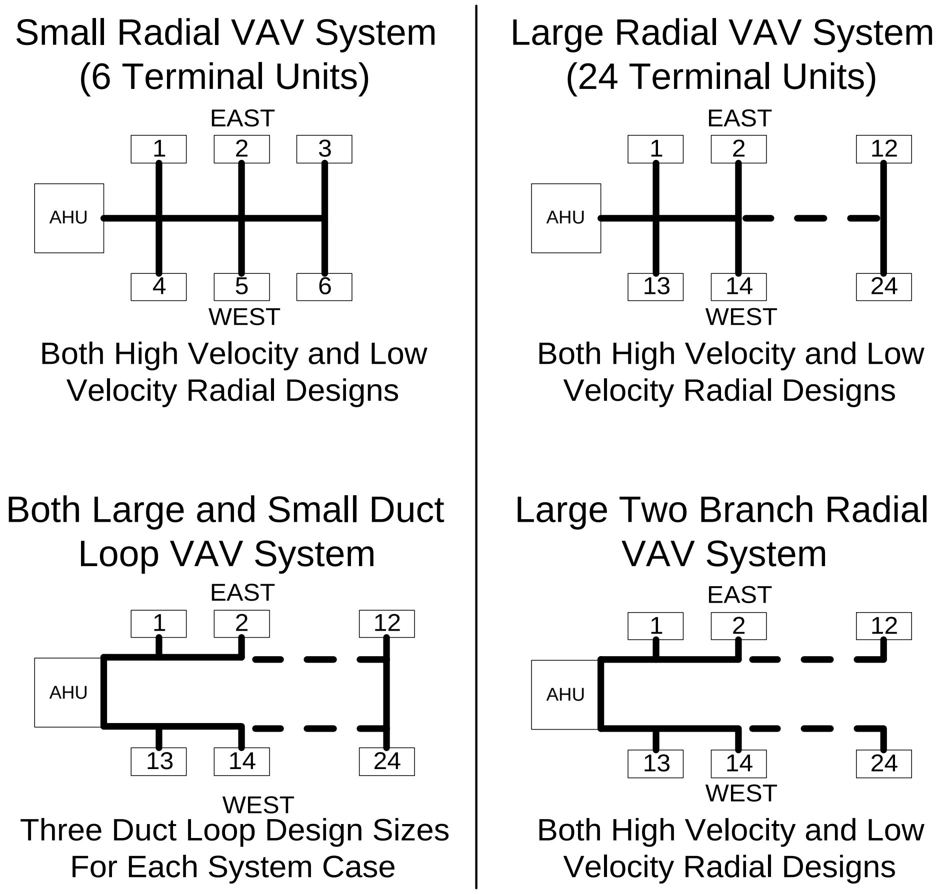

Figure 4: General summary of the various duct loop and radial design layouts

The two building layouts for a large and a small system have been based on a cooling load of 80 W/m^2, with each zone measuring 10 m by 10 m. Each VAV pressure independent terminal unit is sized to supply a maximum volume flow rate of 0.8 m^3/s. These larger boxes have been used rather than many smaller boxes to simplify the system without altering volume flow rates.

The inlet static pressure at maximum volume flow for a fully opened terminal unit was calculated to be 67 Pa, which included the pressure loss of the terminal unit (based on Equation 4), header box, 5m flexible ductwork and the slot diffusers. A maximum pressure loss of 250 Pa has been taken across the AHU due to the resistance of the filters and coils.

The high velocity designs were based on a constant velocity approach in which the ducts were sized to operate between 10 and 15 m/s air velocity during maximum cooling conditions. The low velocity designs were based on a constant pressure loss approach in which the design criterion was to size the main duct to have a pressure loss of 1 Pa/m.

The duct loop was designed similarly on 1 Pa/m and based on half the design volume flow rate. Two reduced sized duct loop designs were also included in the analysis.

5.2 Fan Savings due to design

The full load fan power consumption of each system configuration can be seen in Table 1[(6)]. It shows that duct loops can save up to 35% of fan power than an equivalent high velocity system under full load. The maximum duct diameter required can also reduce by up to 65%.

System Configuration	System Volume Flow Rate (m^3/s)	Required Fan Air Power (KW)	Fan Saving High Velocity (%)	Fan Saving Low Velocity (%)	Max. Duct Size (m) ***
1. Hi. Vel. Radial (Small)	4.8	2.73	0*	-27	0.71
2. Low Vel. Radial	4.8	2.16	21	0**	0.80
3. 500mm Duct Loop	4.8	2.17	21	-1	0.50
4. 560mm Duct Loop	4.8	2.07	24	4	0.56
5. 630mm Duct Loop	4.8	2.02	26	6	0.63
6. Hi. Vel. Radial (Large)	19.2	13.08	0*	-25	1.25
7. Low Vel. Radial	19.2	10.43	20	0**	1.40
8. 2 Br. Hi. Vel. Radial	19.2	14.41	-10	-38	1.00
9. 2 Br. Low Vel. Radial	19.2	9.91	24	5	1.12
10. 900mm Duct Loop	19.2	9.73	25	7	0.90
11. 1000mm Duct Loop	19.2	8.94	32	14	1.00
12. 1120mm Duct Loop	19.2	8.44	35	19	1.12

Table 1: Results of the Fan Air Power Required and Percentage Fan Savings Under Full Load Conditions

* Fan power saving being compared against a High Velocity Radial System.

** Fan power saving being compared with a Low Velocity Radial System.

*** Maximum duct size used in design for the distribution of air to each occupied zone.

6. Supply fan control and part-load savings of radial and duct loops systems

The part load savings and performance of the supply fan are largely dictated by its controls.

6.1 Fan Savings due to control

Simple network analysis indicates the potential of fan savings due to supply fan control applied to three of the examples, a high velocity, a low velocity and duct loop network design. The two control strategies examined are static pressure control and box polling.

6.1.1 Static pressure control

Static pressure control is a cheap and effective way of fan speed regulation in VAV. Static pressure control regulates the speed of the supply fan through a PI controller and maintains the static pressure in the main duct. This method is simple but does not give optimal fan savings.

For a radial system, the rule of thumb is to place the static pressure sensor two thirds[7] to three quarters[8] down the main duct run. Conservative control engineers would place the static pressure sensor close to the supply fan, this was seen in 4 out of 6 installations visited in the United Kingdom. The implications on the placement of the static pressure sensor depend on the systems load distribution profile. If the system load closely follows a synchronous load profile, more fan savings can be achieved with the placement of the static pressure sensor further down the main duct run, perhaps three quarters or more. While in an asynchronous system, a more conservative position perhaps half way down the main duct run is necessary to avoid starvation of any terminal units. Simple network analysis can ensure that terminal units are not starved during part load operation and decide the ideal static pressure sensor position.

In duct loop systems, two thirds static pressure sensor position rule cannot be applied. The placement of the static pressure sensor should be at the balance point of the duct loop. For an equally sized main duct loop design, this is the most stable position for the sensor as the static pressure changes due to dynamic flow fluctuations of adjacent terminal boxes is almost negligible. The static pressure set point should be set to the maximum total pressure required by terminal unit branches adjacent to the sensor to avoid starvation. Network analysis should be adopted to validate or increase the static pressure settings if any other upstream branches require more total pressure across the branch than the adjacent terminal unit branches.

6.1.2 Box polling

Box polling methods have the potential to yield optimum fan savings in a VAV system. Several box polling methods have been researched[9,10,11]. Terminal unit or box polling is based on the concept of requesting the status values of every terminal unit e.g. damper positions, volume flow demands, inlet pressures, etc. to regulate the speed of the supply fan. This method also aims to lessen the need for the control engineer to know the network details and system load profiles to identify the index terminal unit i.e. the terminal unit which requires the most total pressure.

In practical systems, such systems are still experimental and expensive to implement in small to medium installations. Such a control approach is highly dependent on the reliability of every terminal unit. A building energy management system with an outstation module on every terminal unit is also required to relay the terminal unit status to a central processor. Hence box polling methods can only be justified on very large VAV systems and where the load demands are highly diverse.

6.2 Comparison of control strategies under part load conditions

Table 2 shows a summary of the potential fan energy consumption for three system examples, high velocity radial, low velocity radial and a duct loop design with the two control strategies.

System Configuration	% Fan Power at 100% Design Load F/E/B*	% Fan Power at 75% Design Load F/E/B	% Fan Power at 50% Design Load F/E/B	% Fan Power at 25% Design Load F/E/B
1. Low Velocity	100/100/100	54/50/42	26/22/13	10/7/2
2. High Velocity	125/125/125	70/58/53	35/22/16	14/6/2
3. 1120mm Duct Loop	81/81**	43/34	20/10	8/1

Table 2: Results of the Percentage Fan Air Power Required under Synchronous Part-Load Conditions

* % Fan Power with Static pressure control and SP sensor near fan discharge (F) / SP sensor at end of main duct (E) / Box polling (B).

** % Fan Power with Static pressure control and SP sensor at the balance point / Box Polling.

The difference in fan power requirements between static pressure and box polling control is about 8% up to 19% (i.e. F minus B). The difference in potential fan energy consumption by positioning the static pressure sensor near the supply fan opposed to being near the end of the main duct run is in the order of 3% to 13% (i.e. F minus E) fan energy. As shown in Figure 5, static pressure control is an effective method to control the supply fan speed as a savings of up to 45% can be made over constant speed (No fan control/Ridding the fan curve) method. With box polling, the maximum potential fan energy saving is increased to 55%.

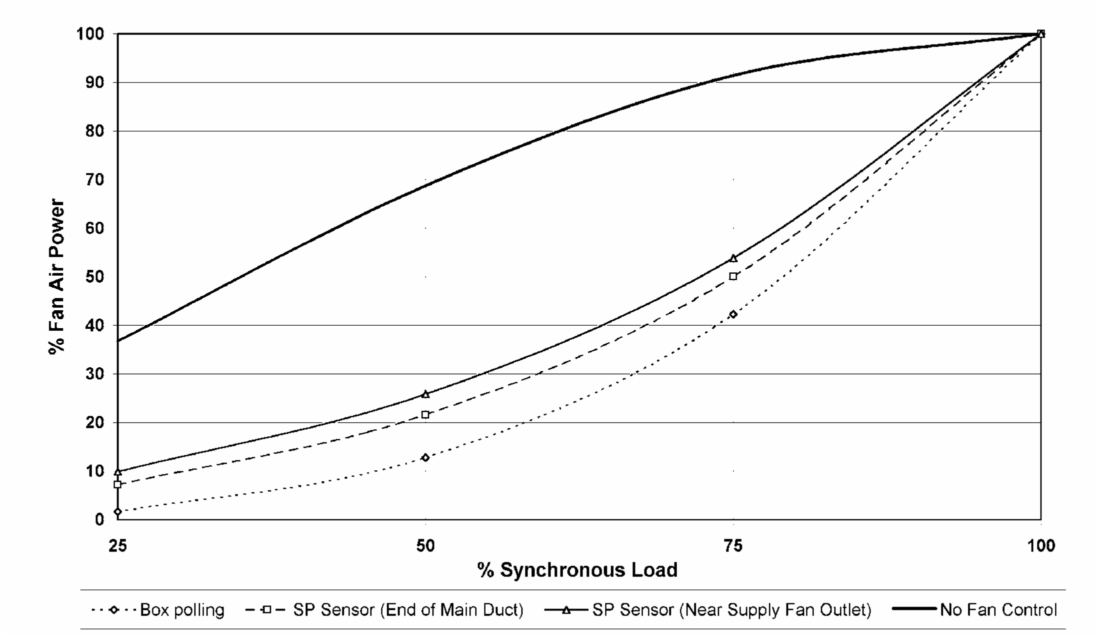

Figure 5: Comparison of the part-load fan power* requirements of a large low velocity radial VAV system with box polling, static pressure and no fan control. * Only Fan air power is considered. Fan, motor, coupling efficiencies are not included.

6.3 Stability Analysis

Box interaction is one of the causes of instability to the supply fan and other terminal units in a VAV system. Duct loops are more robust than radial systems as downstream boxes do not cause large static or dynamic pressure fluctuations due to zonal load changes to the main duct.

Pressure fluctuations due to changes in damper position in radial systems are more pronounced especially in high velocity systems. Hence box to box and box to fan interactions are more apparent in such systems. The influence of downstream terminal unit interaction with the supply fan increases as the static pressure sensor is positioned further down the main duct. Box authority and characteristics play an important role in the stability of the VAV system especially in radial design.

6.4 Starvation

Starvation occurs when a terminal unit is unable to satisfy the volume flow demanded when its dampers are fully opened. Hence all branch ducts must be designed to supply its maximum volume flow rate under minimum system flow conditions. Due to the small variation in static and total pressures around the duct loop, starvation is unlikely to occur during part-load operation. The static pressure set point would be set to the maximum total pressure required across any branch.

7. Conclusion

Duct loops are highly recommended in large systems. As the ease of control, and fan energy and stability is better than radial systems. Duct loops are also very flexible for expansion. The fan energy savings in large systems give an attractive approach for duct designers.

8. References

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