OPTIMUM DUCT DESIGN FOR VARIABLE AIR VOLUME SYSTEMS

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"I will lift up my eyes to the hills-From whence comes my help?

My help comes form the LORD, Who made heaven and earth." (Psalms 120:1,2)

I give thanks to the LORD, my strength and my deliverer.

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TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

Figure Page

CHAPTER 1

INTRODUCTION

1.1 Introduction

The design of duct systems is an important factor for effective, energy-efficient, and comfortable heating, ventilating and air conditioning (HVAC) systems. Commonly utilized duct design procedures have been developed for constant air volume (CAV) systems and are based on peak load design conditions, for which the flow rates are assumed to be constant for the entire year. Yet, the most common system type for commercial office buildings is the variable air volume (VAV) system. VAV duct systems are commonly designed using maximum airflows to zones as if they are CAV systems. However, the VAV system spends much of the time at off-peak load conditions, providing less than peak-flow for many hours of the year. Conventional duct design methods do not account for the actual zone load profile. Consequently, VAV duct systems may not be designed optimally using current design methods. For this reason, duct design methods should be reconsidered for VAV systems.

Three duct design methods are presented in the 1997 ASHRAE Handbook-Fundamentals: equal friction, static regain, and the T-method. Equal friction and static regain methods were developed as expedient procedures and do not address optimization. Of the three, the T-method is the only optimization-based method and was introduced by

Tsal et al. (1988). Duct system optimization gives one the opportunity to save energy and reduce initial cost. The T-method finds optimal duct sizes and fan size by optimum distribution of pressure throughout the system in order to minimize system life-cycle cost. The system life-cycle cost includes the initial ductwork cost based on optimum duct sizes and the year-round electrical energy cost of the fan. The initial cost of the fan is not included. The calculation procedure of the T-method consists of three main steps: system condensing, fan selection, and system expansion. In the first step, the entire duct system is condensed into a single straight duct with multiple sections for finding the ratios of optimal pressure losses using sectional hydraulic characteristics. An optimal system pressure loss is found in the second step. In the third step, the system pressure is distributed throughout the system sections.

The T-method's calculations are based on a fixed amount of airflow throughout the year to determine duct sizes, overall system pressure drop and fan energy cost. However, in VAV systems, the airflow rate varies continuously through a year's operation, therefore the fan power changes with varying airflow. Fan power is also influenced if static pressure at the end of the longest duct line is controlled. Practically speaking, the fan speed controller is regulated by the static pressure at the end of the longest duct line, which is required to be held to a pressure ensuring adequate flow at the zone. Thus, optimization requires accurate modeling of VAV systems based on the actual varying amounts of airflow.

Spitler et al. (1986) investigated fan energy consumption for VAV systems and found, for some buildings, that a large number of hours may be spent at a minimum flow fraction. As an example, for an office building in Colorado Springs, CO, out of a total of

2,520 operating hours, 1,212 are spent at the minimum fraction for a 20% oversized system. Obviously, spending a large number of hours at the minimum fraction makes a significant impact on the fan electricity consumption. System life-cycle cost defined in the T-method does not account for these varying airflows of VAV systems and thus, the T-method may give nonoptimal values for VAV system optimization.

In this study, the system life cycle cost accounts for the impact of varying airflow rates on fan energy consumption. The system life cycle cost is minimized to find the optimal duct sizes and to select a fan. For comparison purposes, several example VAV systems are optimized using the T-method by selecting maximum airflows as design air volumes and then they are optimized again using an optimization procedure that accounts for varying airflow rates. Different from the T-method, duct sizes are selected as explicit design variables that have discrete values, part-load fan characteristics are considered to find fan efficiencies for different airflows, and duct static pressure control is incorporated into the operating cost calculation. As a preliminary step to find a VAV duct design procedure, the problem domain of VAV duct systems is analyzed in terms of duct sizes. The analysis will reveal which type of optimization is required, local or global optimization and consequently, suggest a VAV optimization technique. Tsal and Behls (1986) analyzed a two-dimensional hypothetical CAV duct system using a scalar field technique, which is the graphical representation of the objective function in terms of pressure losses of duct sections. They found a global minimum and the contour map has a convex shape that has a steep slop at the low pressure drop side and a gentle slop at the high pressure drop side.

After the problem domain analysis, the suggested VAV optimization procedure is refined to find discrete optimum duct sizes in a constrained duct design problem. Design constraints for VAV duct systems are added as penalty terms to the objective function for any violation of the constraints. Duct fitting loss coefficients for different design conditions are sought using the duct-fitting data base program as described in ASHRAE (1993). A direct search method is applied to search for a continuous design solution for the constrained duct design problem and a penalty approach for integer programming is employed to impose penalties of discrete violation on the objective function to enforce the search to converge to nominal duct sizes since the duct sizes take their values from a given discrete set. Several methods are in use for discrete/integer optimization. Using a modified branch and bound method, Hager and Balling (1988) sought a discrete optimum in the neighborhood of the continuous optimum. Fu et al. (1991) developed an algorithm which imposes penalties of integer or discrete violations on the objective function. Rajeev and Krishnamoorthy (1992) presented genetic algorithms for discrete optimization of structural design problems.

The objective of duct design is to meet the economic criteria of minimizing initial cost and operating energy cost. The VAV optimization procedure is applied to several VAV duct systems under different design conditions, such as different electric rate structures, different duct work costs, and different system operating schemes. The optimized results are compared to those derived from equal friction, static regain, and the T-method. The impact of varying airflow rates to the sizing of duct systems is investigated and the savings of the VAV optimization procedure are revealed.

1.2 Background and Literature Review

The purpose of this study is to optimize duct design of VAV systems. First, the VAV system is introduced by comparing with a constant air volume (CAV) system in order to further the understanding of the study. A comprehensive review of duct design methods presented in the ASHRAE 1993 handbook is also discussed in this chapter. The T-method is the only optimization-based method, so its objective function and optimization procedures are described in detail since the objective function is further developed from Tsal's definition. System simulation methodology, in the form of the AIRNET (airflow network) program developed by Walton (1988) is described. For the analysis of airflows, Walton developed element models such as fans, ducts, doorways, and construction cracks. The branch and bound method and the penalty approach for integer/discrete nonlinear optimization are discussed for optimum discrete duct sizing. A survey on the current trends of the HVAC duct system design was performed in order to determine which duct design methods are most commonly used for VAV systems and how fans for VAV systems are selected and controlled. The questions and responds of the survey are listed in this section.

1.2.1 Variable Air Volume Systems

VAV systems are described in several HVAC system reference books (Chen and Demster, 1996; Wendes, 1994; Kreider and Rabl, 1994; McQuiston and Parker 1994). From the preceding references, the VAV system is summarized as follows. Most of the HVAC systems in the past were CAV systems that varied the temperature of the delivered air to maintain space conditions. Typical examples are residential or small commercial systems delivering, for instance, 1500 CFM with the burner or air-conditioner going on and off, changing the air temperature to meet the heating or cooling load conditions. Examples of large commercial systems are reheat, dual duct, and multizone systems. In reheat systems, constant conditioned air is supplied from a central unit at a fixed cold air temperature designed to offset the maximum cooling load in the space. The reheat unit is activated when the temperature falls below the upper limit of the controlling instrument's setting. Dual duct systems have two sets of ducts. The central station equipment supplies warm air through one duct run and cold air through the other. The temperature in an individual space is controlled by mixing the warm and cool air in proper proportions. Multizone systems provide a single supply duct for each zone and obtain zone control by mixing hot and cold air at the central unit in response to room or zone thermostats. The CAV systems have significant inefficiencies and energy waste at part load. The air handlers are also expensive to operate since airflow rates cannot be reduced at part-load conditions.

One method of simplifying this problem is to reduce the airflow at part-load conditions. The variable air volume system is a commonly used design that significantly reduces energy consumption as the load is decreased. The basic concept of a VAV system is to reduce system airflow from full load levels whenever loads are less than peak loads. Since flow is reduced, energy transfer at the air handler coil as well as fan power is also markedly reduced. Figure 2.1 shows a typical VAV system with an optional reheat system. The basic system is a cooling-only system that modulates system airflow in response to cooling loads as sensed by a dry-bulb thermostat. As a separate subsystem, an optional reheat system is needed for zones with heating loads. Under peak cooling load conditions, the VAV system operates identically to a fixed volume system with the air handler operating at maximum flow and maximum cooling coil capacity. However, at reduced cooling loads, the system airflow is reduced by the combined action of closing zonal VAV box dampers and the fan speed controller.

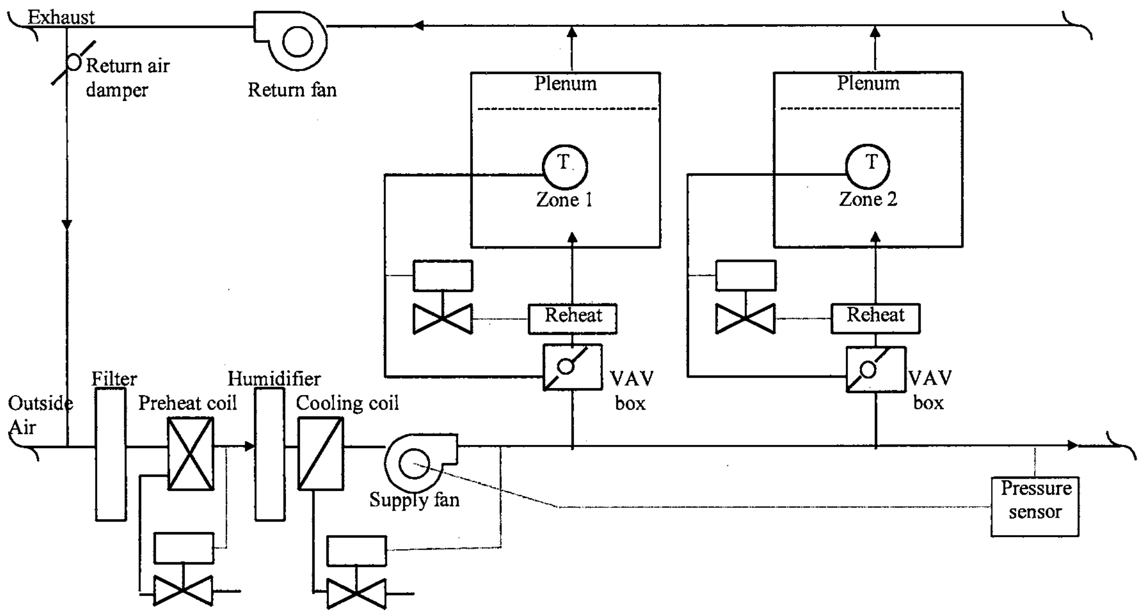


Figure 1.1 Variable Air Volume System with Optional Reheat (Kreider, 1994)

A general feature of the VAV system control is that one must ensure adequate flow at the zone most remote from the air handler. This is traditionally accomplished by controlling the supply fan speed with a pressure signal measured near the end of the duct as shown in Fig 1.1. The actual airflow to each zone is controlled by the thermostat's control of the damper position.

<u>VAV Terminal Box</u>

For comfortable air distribution within a zone, the VAV system is often modified to provide constant airflow by mixing varying conditioned air (called *primary air*) with room air (called *secondary air*) within a VAV box. According to the method for combining primary and secondary air, the VAV boxes have two types: (1) induction VAV boxes, and (2) fan-powered VAV boxes. In the induction method, primary air entrains secondary air that is induced through induction dampers of a VAV box. Fan-powered VAV boxes use a small fan to mix primary air and secondary air. The amount of primary air is controlled by the primary air damper that is controlled by the room thermostat. Fan-powered VAV boxes are either of parallel or series design. The schematic diagrams of each type are shown in Figure 1.2 and 1.3.

Another flow characteristic of VAV boxes is the dependence of flow on supply duct pressure. VAV boxes that are designed to supply constant airflow to the zone for a given thermostat signal despite varying pressures in the ductwork upstream of the VAV box are called *pressure-independent.* VAV boxes that are sensitive to supply duct pressure are called *pressure-dependent.*

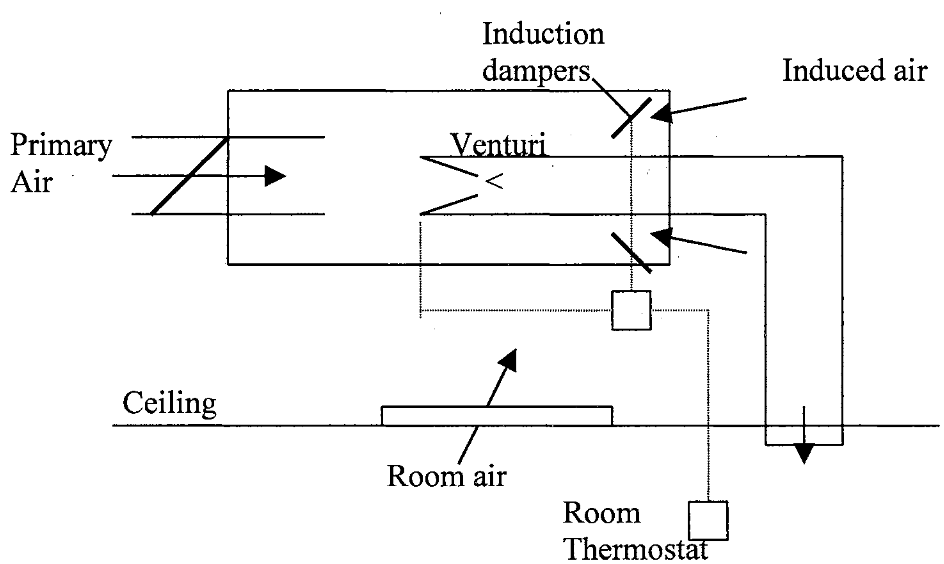


Figure 1.2 Schematic Diagram of Induction VAV Box

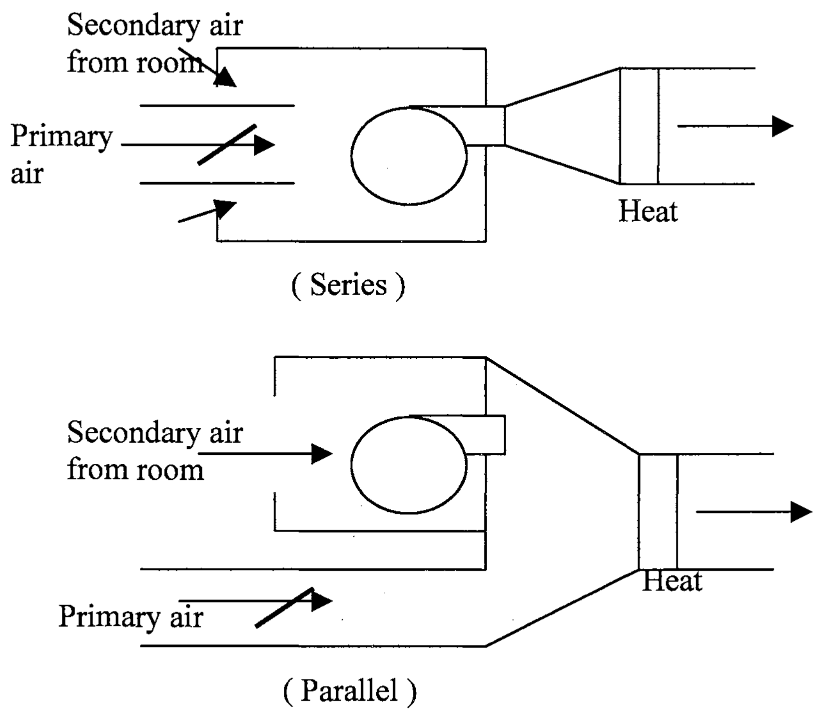


Figure 1.3 Schematic Diagrams of Series and Parallel Fan-powered VAV Box.

Krajnovich and Hittle (1986) tested the performance of pressure-independent VAV boxes to measure pressure independence, linearity, and hysteresis of boxes produced by three different manufacturers. Tests showed that all boxes required approximately one inch water gauge static pressure to operate properly. In this study, pressure-independent VAV boxes will be assumed for model simplification.

Fan Volume Control

Fan volume control methods in the VAV system are an important factor for fan electricity consumption, so they are briefly introduced here. There are several methods of controlling both the pressure and the delivery volume of fans for VAV applications (Chen and Demster, 1995; Haines and Wilson, 1994; Kreider and Rabl, 1994).

- A modulating damper at the fan discharge. A simple damper in the outlet can close to increase resistance and decrease the flow. This makes the fan ride up its curve and saves little fan energy.
- A bypass from supply to return, with a modulating damper. This means that the fan is working at constant volume at all times while system volume varies. Good control is obtained but there are no energy savings.
- Inlet vane dampers. A pre-rotated and limited amount of air enters the fan scroll. As these dampers modulate, they change the operating characteristics of the fan and energy is saved. Inlet vanes pose an energy penalty through added resistance to airflow.
- Electronic speed control. The fan speed is regulated using an electronic speed control called a variable frequency drive (VFD), variable speed drive (VSD), or inverter. This device modulates the power going to the AC induction electric motor so that the motor speed changes in response to the changing frequency of the power produced by the drive electronics. This controlling method saves energy significantly since energy use varies as the cube of the speed, although there are some losses in the electric circuits.

Spitler et al. (1986) compared the energy performance of three modulation methods for a centrifugal fan: *discharge damper, inlet vanes, and motor speed control.* The results show that the fan with an AC inverter offers great savings in fan electricity over the fan with inlet vanes and discharge dampers. The fan with inlet vanes consumes more than twice as much electricity as the fan with an AC inverter at the minimum flow fraction. The discharge damper method was not competitive with the other two methods.

Englander and Norford (1988) quantified the additional savings in fan energy that can be achieved with VSDs as a means of controlling supply duct pressure and return airflow. They stated the additional annual savings due to lowering duct static pressure is two-third of the savings resulting from the variable-speed drive alone, while lowering duct static pressure in a variable inlet vane system has little benefit. They suggested two methods of controlling the supply fan to minimize static pressure: modified PI control algorithm and heuristic algorithm. Both methods regulate either static pressure or fan speed directly, using an error signal derived in some fashion from the primary flow error signal from one or more zones. Thus, VSD modulation method is recommended to use for fan energy savings.

The basic components and their important features of VAV systems are described as follows.

1) VAV terminal box: varies the volume of air flowing through it, based on zone heating/cooling requirements.
2) Fan flow modulation device: variable speed drive on motor, an inlet vane damper, discharge damper, etc.
3) Static pressure sensor: used to sense and measure the terminal duct pressure near the end of the duct in order to ensure adequate flow at the zone. Maintenance of the static pressure is accomplished by controlling the fan flow modulation device.
4) Air distribution ducts: includes the main supply duct, branches and fittings for duct connection.
5) Automatic or manual dampers: controls outside air, recirculation air, return air and mixed air.

1.2.2 Duct Design Methods

Duct design methods are presented in the 1997 ASHRAE Handbook-Fundamentals. Three methods are presented: equal friction, static regain, and the T-method. As non-optimization-based methods, the equal friction method is widely used for low-pressure system in most buildings and the static regain method is used for very large, high-velocity systems (Kreider and Rabl, 1994; Mcquiston and Parker, 1994). The T-method, introduced by Tsal et al. (1988), is the only optimization-based method, described in the 1997 ASHRAE Handbook- Fundamentals*.

Equal Friction Method

The principle of the equal friction method is to produce a constant pressure loss per unit length for the entire system. The usual procedure is to select the velocity in the main duct adjacent to the fan and then the known airflow rate determines the duct size and the lost pressure per unit length. The same pressure loss per unit length is then used throughout the system. After initial sizing, the total pressure loss of the longest run is calculated including the dynamic pressure loss of all fittings and transitions.

As a hybrid of the equal friction method, the balanced equal friction method is introduced in the 1997 ASHRAE Handbook- Fundamentals. After the total pressure loss is decided, duct sections at the branch are resized to balance pressure losses at each junction. A well-balanced design can be produced with this approach if all runs from fan to diffuser are about the same length. However, most duct systems have a variety of duct runs ranging from long to short. The short runs will have to be dampered, which can

* Asiedu et al. (2000) introduced a genetic algorithm approach to design HAVC air duct systems incorporating sizes, variable time-of-day operating conditions and variable time-of-day utility rates.

cause considerable noise. When energy cost is high and installed ductwork cost is low, a low friction rate design is more economical than a high friction rate.

Static Regain Method

The static regain method is based on the requirement that the system static pressure remain about the same throughout the system. Specifically, ducts are sized so that the increase in static pressure in one section of duct exactly balances the pressure loss in the next duct section. The procedure is to first select a velocity for the duct attached to the fan. With the airflow capacity, the size of this main duct is decided. The duct run which has the largest flow resistance is then designed, using the most efficient fittings and layout possible. A velocity is assumed for the next section in the run and the static pressure regain is used to overcome pressure friction losses for that section. This method is suitable for high-velocity, constant-volume systems having long runs of duct with many takeoffs. The main disadvantages of this method are the very low velocities and large duct sizes that may result at the end of long runs.

T-method

The T-method is an optimization-based method that minimizes a life-cycle cost (Tsal et al. 1988). This method is based on the same tee-staging idea as dynamic programming (Bellman 1957, Tsal and Chechik 1968). It has been shown that duct systems optimized using the T-method can result in 12.2% to 53.4% lower life-cycle costs over a system designed using other methods (Tsal and Behls 1986). The goal of duct optimization is to determine duct sizes according to the optimal pressure losses and select a fan according to the optimal fan pressure that minimizes owning and operating costs. Information about owning and operating costs for the HVAC system is described in the 1995 ASHRAE Handbook-HVAC Applications (ASHRAE 1995). The calculation of annual owning costs is comprised of initial cost, analysis period, interest rate, and other periodic costs. The operating cost includes energy cost, maintenance cost, operation labor, and cost escalation. Owning and operating costs are coupled together to develop an economic analysis. The purpose of duct system optimization is to compare system life-cycle cost for different duct sizes and fan total pressure. Accordingly, many of the above constant elements can be excluded from the system cost and only initial cost, energy cost, time period, escalation rate and interest rate are considered for optimization (Tsal and Behls 1990).

Life-cycle cost is given by

$$E = E_p\,(PWEF) + E_S \qquad (1.1)$$

where E = life cycle cost, \$

E_p = annual energy cost, \$

E_s = initial cost, \$

PWEF = present worth escalation factor, dimensionless.

Electrical energy cost is determined by

$$E_p = Q_{fan} \frac{E_c Y + E_d}{10^3 g_f g_e} P_{fan} \qquad (1.2)$$

where Q_{fan} = fan airflow rate, m^3/s(cfm) (*note: constant flow rate throughout the year)

P_{fan} = fan total pressure, Pa (in. wg)

E_c = unit energy cost, \$/kWh

Y = system operating time, h/yr

E_d = energy demand cost, \$/kW

g_e = motor-drive efficiency, dimensionless.

g_f = fan total efficiency, dimensionless.

10^3= dimensional constant, 10^{-3} kW / [(m^3/s)·(N/s)]

The electric energy demand cost, E_d is assumed to be constant for simplification.

The present worth escalation factor is

$$\mathrm{PWEF} = \frac{\left[\frac{(1+\mathrm{AER})}{(1+\mathrm{AIR})}\right]^a - 1}{1-\left[\frac{(1+\mathrm{AER})}{(1+\mathrm{AIR})}\right]} \tag{1.3}$$

where AER = annual escalation rate, dimensionless.

AIR = annual interest rate, dimensionless.

a = amortization period, years

The initial cost is presented as the duct cost, which is a function of the cost per unit area of duct surface. For a round duct, the cost is given by

$$E_s = S_d\, \pi\, D\, L \tag{1.4a}$$

where S_d = unit duct work cost, including material and labor, \$/m^2(\$/ft^2)

D = duct diameter, m (in.)

L = duct length, m(in.)

For a rectangular duct, the cost is

$$E_s = 2\, S_d\, (H + W)\, L \tag{1.4b}$$

where H = duct height, m(in.)

W = duct width, m(in.)

Next, constraints necessary for duct optimization are described. A detailed explanation of each constraint can be found in Tsal and Adler (1987).

- Mass balancing. For each node, the flow in is equal to the flow out.
- Pressure balancing. The total pressure loss in each path must be equal to the fan total pressure.
- Nominal duct sizes. Each diameter of a round duct, or height and width of a rectangular duct, is rounded to the nearest lower or upper nominal size. Nominal duct size normally depends on the manufacturer's standard increment. Such increments may be 1 in. for sizes up to 20 inch then 2 inch increments.
- Air velocity restriction. This is an acoustic or particle conveyance limitation.
- Preselected sizes. Duct diameters, heights and/or widths can be preselected.
- Construction restrictions. Architectural space limitations may restrict duct sizes.
- Equipment. Central air-handling units and duct-mounted equipment are selected from the set produced by industry.

The T-method considers the duct system as a tree structure and is comprised of the following three major procedures.

- System condensing. The branches and roots of the tree are systematically condensed into a single imaginary duct section with identical hydraulic characteristics and the same owning cost as the entire system.

By substituting equation (1.2) and (1.4) into equation (1.1), life-cycle cost becomes

$$E = Z_1 (P_{fna}) + S_d \pi D L, \quad \text{for round duct} \tag{1.5a}$$

$$E = Z_1 (P_{fna}) + S_d 2 (H + W) L, \quad \text{for rectangular duct} \tag{1.5b}$$

where $Z_1 = Q_{fan} \dfrac{E_c Y + E_d}{10^3 g_f g_e}$

The Darcy-Weisbach equation for round and rectangular duct is

$$\Delta P = \left(\frac{fL}{D} + \sum C\right)\frac{V^2 \rho}{2g_c} \tag{1.6}$$

Introduce the coefficient r, $r = fL + \Sigma CD$, substitute r into equation (1.6), and then rearrangement yields, $D = 0.959\,(r\rho)^{0.2} Q^{0.4} (g_c \Delta P)^{-0.2}$

Substituting D into equation (1.4a) yields the initial cost as follows

$$E_s = Z_2\, K\, (\Delta P)^{-0.2} \tag{1.7}$$

where $Z_2 = 0.959\,\pi\,(\rho / g_c)^{0.2}\, S_d$

$K = n\, r^{0.2}\, Q^{0.4}\, L$, characteristic coefficient of a duct section

$n = 1$ for round duct, $n = 1.128$ for square duct

$n = \dfrac{1 + H/W}{\sqrt{\pi \cdot H/W}}$ for rectangular duct

Finally, system life-cycle cost becomes

$$E = Z_1\,(P_{fna}) + Z_2\, K\, (\Delta P)^{-0.2} \tag{1.8}$$

The system life-cycle cost for two duct sections is

$$\begin{aligned} E_{1\text{-}2} &= E_1 + E_2 \\ &= Z_1\,(\Delta P_1 + \Delta P_2) + Z_2\,[K_1\,(\Delta P_1)^{-0.2} + K_2\,(\Delta P_2)^{-0.2}] \end{aligned} \tag{1.9}$$

By taking the partial derivatives of equation (1.9) with respect to ΔP_1, ΔP_2, setting to zero, solving for pressure losses yield the optimum pressure ratio as follows

$$\frac{\Delta P_1}{\Delta P_2} = \left(\frac{K_1}{K_2}\right)^{0.833} \tag{1.10}$$

When two duct sections are connected in ***series***,

$$\Delta P_{1\text{-}2} = \Delta P_1 + \Delta P_2 \tag{1.11}$$

Using equation (1.10) and (1.11), equation (1.9) becomes

$$\begin{aligned} E_{1\text{-}2} &= Z_1 (\Delta P_1 + \Delta P_2) + Z_2 [K_1 (\Delta P_1)^{-0.2} + K_2 (\Delta P_2)^{-0.2}] \\ &= Z_1 \Delta P_{1\text{-}2} + Z_2 (K_1^{0.833} + K_2^{0.833})^{1.2} (\Delta P_{1\text{-}2})^{-0.2} \\ &= Z_1 \Delta P_{1\text{-}2} + Z_2 \; K_{1\text{-}2} (\Delta P_{1\text{-}2})^{-0.2} \end{aligned} \tag{1.12}$$

Thus, the characteristic coefficient of a condensed duct section, which is connected in *series*, is described by

$$K_{1\text{-}2} = (K_1^{0.833} + K_2^{0.833})^{1.2} \tag{1.13}$$

When two sections are connected in parallel,

$$\Delta P_{1\text{-}2} = \Delta P_1 = \Delta P_2 \tag{1.14}$$

Equation (1.9) becomes

$$E_{1\text{-}2} = Z_1 \Delta P_{1\text{-}2} + Z_2 (K_1 + K_2) (\Delta P_{1\text{-}2})^{-0.2}$$

Thus, the characteristic coefficient of a condensed duct section, which is connected in *parallel*, is described by

$$K_{1\text{-}2} = K_1 + K_2 \tag{1.15}$$

Tsal's equations (1.13) and (1.15) are applied from junction to junction in the direction of the root section so as to condense the entire system into one section.

- Fan selection. From the condensed system, the ideal optimum fan total pressure is calculated and used to select a fan. If a fan with a different pressure is selected, its pressure is considered optimum. By taking the derivative of equation (1.8) with respect to ΔP, setting to zero, and solve for pressure loss, the optimum fan pressure becomes

$$\Delta P = 0.26\ (Z_2/Z_1 \cdot K)^{0.833} + \Delta P_x \qquad (1.16)$$

where K is the characteristic coefficient of condensed root section.

ΔP_x is an additional pressure loss.

- System expansion. The imaginary duct section is expanded into the original system by distributing the optimized fan pressure. Duct pressure loss at section i is

$$\Delta P_i = P_i \cdot T_i \qquad (1.17)$$

where $P_i = \Delta P_{1\text{-}i}$, remaining pressure from duct section i to terminal duct section 1

$T_i = (K_i / K_{1\text{-}i})^{0.833}$, T-factor at duct section i.

Unlike the condensing procedure, the expansion procedure starts at the root section and continues in the direction of the terminals. A detailed explanation of each major procedure can be found in Tsal et al. (1988). Many parameters are unknown at the beginning and have to be defined during the iterative process, such as the C-coefficients for junctions and transitions since they depend on duct size. Also, the fan cannot be selected until the system K-coefficient is known. Usually, three iterations are enough to obtain accurate optimum solution (Tsal et al., 1988).

Size rounding to select a lower or an upper nominal duct size is also an optimization concern. If the lower nominal size is selected, the initial cost decreases, but the pressure loss increases and may exceed the fan pressure. If the upper nominal size is selected, the initial cost increases but the section pressure loss decreases. This saved pressure can be used to select a lower nominal size for the following duct section. The T-method has the procedures that predict how lower and an upper nominal duct sizes influence the initial cost for the rounding duct section and the remaining duct sections. The nominal size that produces a lower initial cost is selected as the rounded diameter.

The pressure loss subtracted from the rounded duct section is used as the upper value for rounding the children sections. The rounding procedure starts at the root section and continues in the direction of the terminals.

1.2.3 Simulation of VAV Systems

In order to simulate the operation of a building using a VAV system, it is necessary to determine the quantity of air required to meet the load, which can be done with any load calculation program. For this project, the Building Loads Analysis and System Thermodynamics program (BLAST 1986) was used. By considering the airflow rates into and out of the zones by the ductwork and the exhaust requirements, the building pressures can be calculated.

The program called AIRNET (Walton 1989) for building airflow network modeling, which was developed by the National Institute of Standards and Technology (NISTIR 89-4072), provides a method to estimate airflows and pressures in buildings. A building airflow network consists of a set of nodes, elements, and linkages. The various zones in buildings, the connection points in ductwork, and the ambient environment are points where the airflow and pressures are of interest. These points are represented as nodes in a network. The airflow elements represent passages between nodes, such as the ducts, fans, dampers, cracks, doors, etc. All nodes are connected by one or more airflow elements. The linkages describe how the nodes and elements are connected. A set of such linkages makes up a complete building HVAC network.

The program modules for airflow analysis in AIRNET are as follows:

1. a process for establishing an initial set of values to start the iterative solution process.

2. a solution method for nonlinear equations using Newton's method and a skyline solution process of sparse matrix methods (Dhatt and Touzot, 1984) to reduce both the storage and execution time, combined with Steffensen iteration (Conte and de Boor, 1972) to accelerate convergence.
3. airflow element subroutines that compute the flow rate and derivative of the flow with respect to pressure difference needed to form the Jacobian matrix.
4. a separate process for transferring the data into the Jacobian matrix .
5. solution of the simultaneous linear equations involving the Jacobian matrix.

Mass balance equations are the basic equations governing flows in an airflow network.

$$\sum_i \dot{m}_{in} = \sum_i \dot{m}_{out} \tag{1.18}$$

where, i = node number.

The relationship between the mass flow rate and the pressures can be described as:

$$\dot{m} = f(\Delta P) \tag{1.19}$$

Since modeling different elements involves nonlinear relationships, iteration needs to be done to arrive at the solution.

The following types of flow elements are simulated in the AIRNET program.

- Resistance. The mass flow rate of air across any restriction follows the empirical relationship of the form:

$$\dot{m} = C\sqrt{\rho}\Delta P^x \tag{1.20}$$

where C = flow coefficient

ρ = air density, kg/m^3

ΔP = total pressure loss across the element, Pa

$x =$ the flow exponent.

- Ducts. The pressure loss due to friction in a section of a duct or pipe is given by

$$\Delta P_f = \frac{fL}{D}\frac{\rho V^2}{2} \tag{1.21}$$

where f = frictional coefficient

L = duct length, m

D = hydraulic diameter, m

V = velocity of air, m/s

- Duct fittings. The dynamic pressure losses due to fittings are given by

$$\Delta P_d = C_0 \frac{\rho V^2}{2} \tag{1.22}$$

where C_0 = dynamic loss coefficient.

The total pressure loss can then be calculated as

$$\Delta P = \Delta P_f + \sum \Delta P_d \tag{1.23}$$

From the above equations, the flow rate is calculated using the relation

$$\dot{m} = \sqrt{2\rho A^2 \big/ (fL / D + \Sigma C_0)}\sqrt{\Delta P} \tag{1.24}$$

- Fans. In order to accurately simulate the performance of a fan, the fan performance curves have been fitted to a polynomial of the form

$$P = a_0 + a_1 \dot{m} + a_2 \dot{m}^2 + a_3 \dot{m}^3 + \ldots \tag{1.25}$$

where a_0, a_1, a_2, a_3, ... are the coefficients of the polynomial that fits the fan performance curve at a rated speed.

Using AIRNET, Delp et al. (1993) modelled seven basic types of VAV control systems for providing control of the minimum outside ventilation air and the pressurization of the building. All systems have a supply fan whose speed, and thus airflow, is controlled by a static pressure controller. The seven basic types are:

1. Return fan, with capacity control sequenced from supply air capacity control (static pressure) signal.
2. Return fan, with capacity control based on building pressure.
3. Return fan, with capacity control based on differential airflow between measured supply and return quantities.
4. Relief (exhaust) fan, with capacity control based on building pressure.
5. Relief (exhaust) fan, with capacity control based on outside air damper position.
6. Neither return nor relief fan, with building pressure controlled by relief dampers based on building pressure.
7. Relief dampers controlled by the pressure ratio across the return air damper coupled with the addition of an outside air injection fan.

All of these control systems should be able to do the following for proper operation:

- maintain duct static pressure so that the terminal units operate properly.
- maintain slightly positive pressure in the conditioned space to prevent infiltration of outside air into zones.
- supply minimum outside air to the conditioned space except during economizer operation.

In addition to the elements described in AIRNET, Delp et al. (1993) modelled dampers and VAV boxes using the relationship for resistance. If the damper is set at a particular angle, it would have a fixed resistance. To account for varying damper resistance, the following relation is given for dampers:

$$\dot{m} = C_v \cdot \sqrt{\rho_n} \cdot \Delta P^x \tag{1.26}$$

Modeling of the VAV boxes was done in the same manner as a damper.

For the system simulation, AIRNET was selected and a separate algorithm, which simulates the operation of VAV systems, was developed so that the zone loads could be satisfied. The control data file describing the desired operating conditions, such as desired duct static pressure ranges for supply and return fan control, desired outside air requirement, and desired building pressure is given as input for simulation.

1.2.4 Nonlinear Integer Optimization

Most optimization methods have been developed under the implicit assumption that the design variables have continuous values. In many practical situations, however, the design variables are chosen from a list of commonly available values, for example cross-section areas of trusses, thickness of plates, and membranes. Furthermore, in the optimum design of duct systems, ducts have discrete values that normally depend on the manufacturer's standard increment.

The branch and bound (B&B) method is a widely used algorithm for solving integer programming (IP) problems. However, the original B&B method is not suitable for solving a nonlinear integer programming (NIP) problem, primarily because the validity of the branching rules is tied with an assumption of linearity (Yokota et al. 1996). Lee (1983) solved several engineering nonlinear problems using the B&B method based on a nonlinear optimization code called BIAS, where the B&B procedure simply alters the upper and lower bounds on the variables. Hager and Balling (1988) sought a discrete optimum in the neighborhood of the continuous optimum using a modified B&B method.

Olsen and Vanderplaats (1989) presented a method of sequential linear discrete programming and converted the nonlinear discrete problem into a sequence of linear (0, 1) problems. The penalty approach for NIP problems is another popular method in structural optimization. Fu et al. (1991) developed an algorithm which imposes penalties of integer or discrete violation on the objective function to enforce the search to converge to discrete standard values. Lin and Hajela (1992) presented artificial genetics approach for global discrete optimization of structural design problems, which is a modified simple genetic algorithm proposed by Goldberg, based on natural genetics. Next, the B&B method and the penalty approach are discussed for nonlinear optimization with discrete design variables.

Branch and Bound Method

The branch and bound (B&B) method is based on converting the integer solution space to a continuous space by initially dropping the integer conditions. After obtaining the continuous optimum, the method forms new subproblems, called candidates. By branching, these candidates exclude the infeasible (non-discrete) region, and include all the feasible integer points of the problem. Bounds are used to rapidly discard many of the possible candidates by developing a bound on the optimum objective value of the integer problem. Any of the candidates whose value of the objective function falls outside the bound may be discarded as nonpromising.

The logic for the nonlinear branch and bound algorithm is that each of the individual nonlinear programming problems arising in the solution procedure is solved using an efficient nonlinear optimization method. Thus, the actual constraints are handled separately from the variable bounds. The basic solution procedure for the nonlinear programming problem may be summarized as follows:

- Step 1: Solve the original NLP problem, ignoring any integer restrictions.
- Step 2: Determination of the type of design variable. If the design variable is required to be integer, then go to step 3. If the design variable is required to be discrete, then go to step 4. If the design variable is required to be continuous real value, then go to step 5.
- Step 3: The design variable which is required to be an integer at the final solution, say X_i, is branched upon in the following manner: let the variable be P + Q where P is the integer part of X_i and Q is the fractional part. If P is defined as the largest integer not exceeding P+Q, then the region $P < X_i < P+1$ contains no feasible integer values, and two new branches can be created by imposing the restrictions $X_i \leq P$ and $X_i \geq P+1$ on the current problems.
- Step 4: The design variable which is required to be a discrete variable at the final solution, say X_j is branched upon in the following manner: let the variable be R. If the discrete value D_k is defined as the largest discrete variable not exceeding R, and D_{k+1} is defined as the smallest discrete variable exceeding R, then the region $D_k < R < D_{k+1}$ contains no feasible discrete value, and two new problems (branches) can be created by imposing the restrictions $X_j \leq D_k$ and $X_j \geq D_{k+1}$ on the current problem.
- Step 5: If an integer or discrete solution result becomes an upper bound on the final value of the objective function then all nodes with a value greater than this upper bound may be eliminated from the search (assuming unimodality).

- Step 6: If all nodes have been eliminated, the solution procedure is finished. If a node still exists, then the next required integer variable or discrete variable is branched upon, and step 2, 3 and 4 are repeated until all required integer or discrete variables have been branched upon, with the upper bound always being updated to the best integer or discrete variable solution found.
- Step 7: Once all required integer or discrete variables have been branched upon and a branching node still exists, then the required integer or discrete value is branched upon, and this step is repeated until all nodes have been eliminated.

In order to illustrate the application of the B&B method, the following example with two variables, previously discussed by Lee (1983), is introduced. The problem is as follows:

$$\text{Minimize } f(x) = \left[12 + x_1^2 + \frac{(1+x_2^2)}{x_1^2} + \frac{(x_1^2 x_2^2 + 100)}{(x_1^2 x_2^2)^4} \right] \Big/ 10$$

x_i is to be integer.

$1 \leq x_i \leq 3 \qquad i = 1,2$

The solution procedure of the problem is shown in Figure 2.4. The optimal solution without integer restriction is $x_1 = 1.855$, $x_2 = 1.834$ with $f(x) = 1.7542$ as shown in node 1. The integer solution procedure starts by dropping the integer restriction on x_1 and x_2. Since x_1 is required to be an integer, the range $1<x_1<2$ is deleted from the continuous solution space without deleting any feasible integer values. In other words, two constraints $x_1 \leq 1$ and $x_1 \geq 2$ are applied to node 1 to effect the deletion of the region $1<x_1<2$ from the continuous space. This results in the two nodes, node2 and 3. It says that two new branches are created by imposing the bounds $x_1 \leq 1$ and $x_1 \geq 2$. This branch and bound algorithm is continued until all integer solutions are attained. Each solution step is illustrated in Figure 1.4.

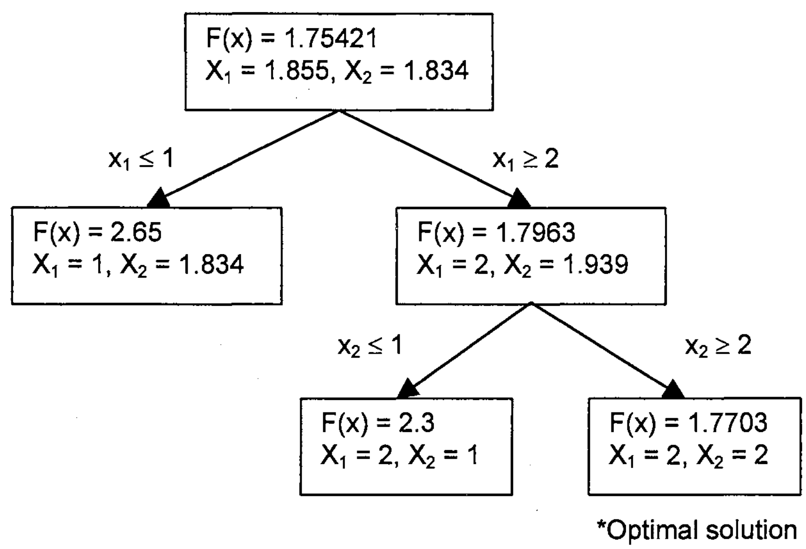


Figure 1.4 Solution Steps of Example Problem

<u>Penalty function method</u>

The penalty function method imposes penalties of integer or discrete violation on the objective function to effect the search in the way that the solution converges to discrete standard values, based on a commonly employed optimization algorithm. In duct systems, the diameter of a round duct, or the height and width of a rectangular duct is a discrete variable and the constraint of nominal duct sizes can be resolved using the penalty function approach. Any violation of the constraint is added to the life-cycle cost to enforce the search to converge to discrete duct sizes.

In general, a discrete optimization problem can be represented as a nonlinear mathematical programming problem of the following form:

$$\begin{aligned} &\text{Min} && f(X), && X \in E^n && (1.27)\\ &\text{Subject to:} && h_i(X) = 0 && i = 1, \ldots, m\\ &&& G_i(X) \geq 0 && i = m+1, \ldots, p\\ &&& l_i \leq x_i \leq u_i \end{aligned}$$

where $X = [x_1, x_2, \ldots, x_n]^T = [X^c, X^d]^T$

$X^c \in R^c$: feasible subset of continuous design variables

$X^d \in R^d$: feasible subset of discrete design variables

l_i and u_i: the lower and upper bounds for the design variables

The objective function may be expanded into a generalized augmented form to include penalty terms for the violation of the conditions for selecting specified discrete variable values

$$F(X) = f(X) + P(X^d) \tag{1.28}$$

where $P(X^d)$ is the penalty on specified discrete value violation.

The penalty function in this approach is defined as

$$P(X^d) = \gamma\, Q(X^d)^\beta \tag{1.29}$$

where $\quad Q(X^d) = \sum_{j\in d} 4q_j(1-q_j) \quad$ and $\quad q_j = \dfrac{(x_j - s_j^l)}{(s_j^u - s_j^l)}$

s_j^l and s_j^u are the nearest feasible lower and upper discrete values.

Cai and Thierauf (1993) discussed the proper choice of γ and β. For β, it is recommended to choose 1 or 2. A larger value of β makes the convergence to the discrete solution slower. The choice of the value of γ strongly influences the convergence of the objective function and the following estimating equation is suggested:

$$\gamma = \frac{F(X^m) - f(X^m)}{Q^\beta} \tag{1.30}$$

where $X^m = (S^l + S^u)/2$

and $S^l = [s^l_1, \dots s^l_n]$ and $S^u = [s^u_1, \dots s^u_n]$ are the nearest lower and upper discrete points of the starting point X^0.

In the solution process, an initial value γ is estimated from the equation. When the subsequent search is made iteratively, the factor γ is gradually increased as follows:

$$\gamma^{(k+1)} = c\gamma^{(k)} \tag{1.31}$$

where c is a constant value in the interval

$$1 < c < 2.$$

In order to illustrate the application of the penalty function method, the following 10 bar truss problem shown in Figure 1.5, previously discussed by Cai and Thierauf (1993), is introduced.

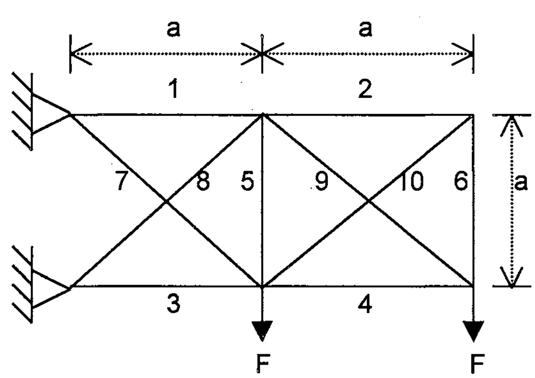


Figure 1.5 Ten-Bar Truss

The objective function of the problem is the weight of the structure. The design variables are the cross-sectional areas of the 10 members. The constraints are the member stresses and the vertical displacements of the nodes 2 and 4. The allowable displacement is limited 2 inch and the allowabl7e stress to ±25ksi. The design parameters are $E = 10^4$ ksi, $F = 100$ kips, $\rho = 0.1$ lb/in^3, and $a = 360$ inch. According to the American Institute of Steel Construction manual, a discretization for the cross-sectional areas is

S = (1.62, 1.80, 1.99, 2.13, 2.38, 2.62, 2.63, 2.88, 2.93, 3.09, 3.13, 3.88, 3.47, 3.55, 3.63, 3.84, 3.87, 3.88, 4.18, 4.22, 4.49, 4.59, 4.80, 4.97, 5.12, 5.74, 7.22, 7.97, 11.50, 13.50, 13.90, 14.20, 15.50, 16.00, 16.90, 18.80, 19.90, 22.00, 22.90, 26.50, 30.00, 33.50) (inch^2).

The value of factors c and β and the accuracy parameter ε were chosen as

$$c = 1.5,\quad \beta = 1.0,\quad \varepsilon = 0.005$$

and the initial value of the penalty factor was calculated with equation (2.30). In computation, the continuous solution was found after the first iteration using a sequential quadratic programming code NLPQL. Only one further iteration was processed to obtain the discrete solution. The minimum value is 5491.71 lb. The result is given in Table 1.1.

Table 1.1 Results for the ten-bar truss problem

	F(lb)	X_1	X_2	X_3	X_4	X_5	X_6	X_7	X_8	X_9	X_{10}
1*	5482.56	32.114	1.62	23.186	15.396	1.62	1.62	8.314	22.761	21.567	1.62
2*	5491.71	33.50	1.62	22.90	15.50	1.62	1.62	7.97	22.00	22.00	1.62

1*: continuous solution with NLPQL
2*: discrete solution with combination of penalty algorithms and NLPQL

1.2.5 Survey of HVAC duct system design

Equal friction, static regain, and the T-method, introduced in the 1997 ASHRAE Handbook - Fundamentals, have been developed for duct design of CAV systems. However, they are also used for VAV systems in commercial buildings. Since the design of VAV systems needs to consider varying airflows, part-load characteristics, and duct static pressure control, it would be useful to find out how they are taken into account in consulting practice.

For this purpose, a survey on the current trends of the HVAC duct system design was performed in order to answer the following two questions: (1) which duct design methods are most commonly used for VAV systems? and (2) how fans for VAV systems are selected and controlled? The subjects of the survey are duct design engineers who are involved in duct design of VAV systems. They were contacted by a letter and asked to fill in a self-administered, mail-in questionnaire (See Appendix A). Most of the participants are the Oklahoma ASHRAE chapter members or practicing engineers affiliated with the ASHRAE duct design technical committee. Fifty surveys were sent out and eight returned.

The responses to the questions in the survey are summarized as follows:

- *Duct design methods that are used to size ducts of VAV systems:* Most respondents answered that they use the equal friction method. Some of them answered that they use the static regain method for high-pressure duct systems or due to the convenience of computer programs. There was one response that the T-method was used for duct systems of nuclear facilities.

- *Incorporation of any diversity when sizing ducts or fan:* It was answered that diversity is considered in both the duct system and fan. Some answered that block and peak loads are used for sizing a fan.
- *Type of fan specified for VAV systems:* The most often used type of fan was the airfoil and secondly, the backward inclined fan. Some answered the forward curved type fan is used occasionally.
- *Fan selection method to avoid fan operation in the surge region:* It was answered that operating points for full and partial loads were checked as to whether they were in the proper region of the fan curve.
- *Any required fan pressure in addition to the pressure loss from ducts and fittings:* Most respondents answered that they considered the static pressure that should be maintained at the end of the longest duct line. Some answered that they considered a positive pressure at the zone to prevent air infiltration. Some specified the static pressure at the VAV terminal box and others said that they considered an additional 10% air pressure.
- *Any specific method for selecting a fan:* Some answered that they considered acoustic (noise) problems, avoidance of surge (checking operating points), and efficiency.
- *Duct static pressure at the end of the longest duct line:* The answers varied: 0.25 in. wg, 0.8 to 1.0 in. wg, and 1.5 in. wg.
- *Any positive air pressure in a zone:* The answer varied: 0.03 in. wg, 0.1 in. wg, 0.25 in. wg, "depending on the application", and "air volume of the terminal controller".

From the survey, it is considered that the equal friction and static regain methods are still preferable to duct design engineers. Some designers especially use the static regain method, which is packaged into software with an option to select a fan. For a moderately larger system, airfoil and backward inclined types of fans are specified for VAV systems. In fan selection, noise problems, efficiency, and operating points avoiding fan surge were the main concerns for the designer. The response to the static pressure requirement was that it is necessary, but the pressure values differed. A positive air pressure can be specified depending on the application.

CHAPTER 2

OPTIMUM DUCT DESIGN FOR VAV SYSTEMS

In this chapter, the optimum duct design problem is defined accounting for varying airflows, part-load fan characteristics, duct static pressure control, and discrete duct sizes. The initial and operating costs are explained at the problem definition in greater detail. Design constraints are explained qualitatively and their use in optimization problem is discussed. The scheme for the VAV optimization procedure is explained with operating cost calculation and integer/discrete programming technique.

2.1 Objective of the Study

The objective of optimum duct design is to find duct sizes and select a fan that minimizes system life-cycle cost for VAV systems. The VAV system is a commonly used design that significantly reduces energy consumption as the load is reduced. Current duct design methods for variable air volume (VAV) systems are based on the use of peak constant airflow. However, VAV systems operate much of the time at an off-peak load condition and the impact of varying airflow rates to the sizing of duct systems has not been considered.

This study introduces an optimum duct design procedure for VAV systems to see the importance of the varying airflows to the system design. Hourly airflow requirements, part-load fan characteristics, and duct static pressure control are incorporated into the problem formulation. Constraints such as discrete duct sizes, telescopic restriction, and velocity limitation are incorporated into the duct design procedure. Since the reduction of airflow results in decreasing fan speed, fans are modeled to give exact fan power consumption for different airflow.

In order to suggest an optimization procedure, the domain of a VAV optimization problem is analyzed to define the problem characteristics. Then the VAV duct design procedure is applied to several VAV duct systems and results are compared to those obtained using current duct design methods.

The benefits that might accrue as a result of the study are:

- The modeling of VAV duct systems is realized by considering varying airflows and fan pressure control instead of using constant airflow and no control scheme.
- The domain characteristics of VAV duct systems will be identified.
- The impact of varying airflow rates on fan energy consumption will be identified.
- The new duct design procedure will find better duct design and the performance of the current duct design methods for VAV systems will be evaluated.

2.2 Problem Definition

Optimum duct design of VAV systems is to find duct sizes and select a fan that minimizes system life-cycle cost. The system life-cycle cost is made up of annual owning and operating costs (ASHRAE 1995). The elements of annual owning costs are initial cost, analysis period, interest rate, and other periodic costs such as insurance, property taxes, etc. Operating costs includes the costs of energy, and maintenance. Since the purpose of optimization is to compare system costs for different fan total pressures, many of the above factors are constant and can be excluded from the objective function. Only initial cost, energy cost, time period, escalation rate, and interest rate are considered for optimization. VAV systems have varying airflows to meet the different loads in the zone and the fan is controlled to maintain the desirable static pressure at the end duct section of the longest duct line, the fan is modeled on an hourly basis to determine the fan electricity consumption and operating cost.

The VAV duct design problem is defined as:

$$\text{Minimize} \quad E = E_p \cdot PWEF + E_s \tag{2.1}$$

$$\text{Subject to} \quad D_p - D_c \geq 0$$

$$L_i \leq D_i \leq U_i$$

where

E = system life cycle cost, \$

E_p = first year energy cost, \$

E_s = initial cost, \$

PWEF = present worth escalation factor, dimensionless.

D_p = upstream duct section diameter, m (in.)

D_c = downstream duct section diameter, m (in.)

$D_i = [D_1, D_2, \ldots, D_n]^T = [D^d]^T$

$D^d \in R^d$: feasible subset of discrete duct sizes, m (in.)

n = the number of duct sections

L_i and U_i = the lower and upper bounds of duct section i, m (in.), due to velocity or geometric constraints.

2.2.1 Initial Cost

The initial cost includes the cost of installed ducts and HVAC equipment. The duct cost is determined as a function of the cost per unit area of duct surface. The cost of HVAC equipment, such as fittings, heating coil, and cooling coil are considered constant and are not included in the objective function, except the fan cost. The initial cost is

$$E_s = E_{duct} + E_{fan} \tag{2.2}$$

where

E_{duct} = duct cost, $

E_{fan} = fan cost, $

Duct Cost.

The duct size of each duct section is a discrete design variable selected from the nominal sizes limited to the manufacturer's standard increments. For a round duct, the duct cost is

$$E_{duct} = S_d \pi D L \tag{2.3}$$

where

S_d = unit duct work cost, including material and labor, $\$/m^2$ ($\$/ft^2$)

D = duct diameter, m (in.)

L = duct length, m (in.).

For a rectangular duct, the duct cost is

$$E_{duct} = 2S_d (H + W)L \tag{2.4}$$

where

H = duct height, m (in.)

W = duct width, m (in.)

Fan Selection and Cost.

Since duct optimization involves searching among different duct sizes, a fan that can work with each configuration must be selected and included in the initial cost calculation. When selecting a fan, the following factors govern the type of fan to be selected and its size:

- Peak airflow rate.
- Static pressure at a peak volumetric flow rate.
- Efficiency: select a fan that will deliver the required volume at the expected static pressure with minimum horsepower.

Among the above factors, the fan selection in the VAV duct optimization is mainly based on the system pressure requirement at a peak hour's design airflow that gives maximum annual airflow. The pressure drop for every duct path is calculated and the highest-pressure drop is selected as the one determining fan selection. Starting from the smallest fan size, the system design point that includes the system pressure requirement and peak airflow is checked as to whether that point is on the fan operating range. If the fan cannot operate at that design point, the next larger size fan is entered for selection. After a suitable fan is selected, fan efficiency is calculated for the airflow through a year's operation. The desired fan efficiency calculation is necessary to assure that the fan can deliver the required airflow at the desired static pressure within the fan operational range.

The fan selection process resolves itself into the following three steps:

1. Prepare fan performance data.
2. Select the smallest fan that satisfies the system design point.
3. Calculate fan efficiency for different airflows.

Centrifugal fans are used for the fan selection and the fan cost including motor and drive can be found from the RS Means Mechanical Cost Data (1998).

2.2.2 Operating Cost

The operating cost consists mainly of the electrical energy cost required by the fan and is represented by a present value. A multitude of electrical rate structures may be encountered in practice. Here, electrical energy cost is assumed to be based on a unit energy cost and a demand charge based on the annual peak electricity consumption. However, any rate structure could be incorporated, as energy consumption is calculated on an hourly basis. The first year energy cost is

$$E_p = \frac{1}{10^3 \eta_m} \left(\sum_{Y(hr)} \frac{Q_{fan} P_{fan} E_c}{\eta_f} + \sum_{Y(mo)} \frac{Q_{fan,peak} P_{fan,peak} E_d}{\eta_{f,peak}} \right) \tag{2.5}$$

where

Q_{fan} = fan airflow rate, m^3/s (cfm)

P_{fan} = fan total pressure, Pa (in.wg)

Y = system operating time, h/yr

E_c = unit energy cost, \$/kWh

E_d = energy demand cost, \$/kW

η_f = fan shaft efficiency, dimensionless

η_m = motor-drive efficiency, dimensionless

10^3= dimensional constant, 10^{-3} kW / [(m^3/s)·(N/s)] or 1.1741×10^{-4}kW / [cfm·in.wg].

The cost of hourly energy consumption is added across a year's operation according to varying fan airflow rates and fan total pressures. The shaft efficiency, η_f, is the function of fan speed and airflow rate. The hourly shaft efficiency is computed from a fan performance equation and it is used for computing the hourly energy cost. The motor efficiency, η_m, is assumed as a constant. The electrical energy demand cost, E_d, is based on the customer's peak kilowatt demand. The present worth escalation factor is

$$PWEF = \frac{\left[{(1+AER)}/{(1+AIR)} \right]^a - 1}{1 - \left[{(1+AIR)}/{(1+AER)} \right]} \tag{2.6}$$

where

AER = annual escalation rate, dimensionless

AIR = annual interest rate, dimensionless

a = amortization period, years.

<u>Duct Modeling</u>

The duct size is used to calculate the pressure loss in a duct section. The pressure loss of a duct section is calculated using the Darcy-Weisbach equation:

$$\Delta P = \left(\frac{fL}{D_h} + \sum C \right) \frac{V^2 \rho}{2 g_c} \tag{2.7}$$

where

f = friction factor, dimensionless

L = duct length, m(in.)

D_h = hydraulic diameter, m (in.)

ΣC = the summation of local loss coefficients within the duct section.

V = mean air velocity, m/s

ρ = air density, kg/m^3 (lb$_m$/cu ft)

g_c = dimensional constant, 1.0(kg.m)/(N.s^2), 32.2(lb$_m$ft)/(lb$_f^2$)

For a rectangular duct, the equivalent-by-friction diameter (hydraulic diameter) is

$$D_f = 2\frac{H \cdot W}{H + W} \tag{2.8}$$

Next, the equivalent length, L_e, is introduced

$$L_e = L + \frac{D_h}{f}\sum C \tag{2.9}$$

and substitute into the Darcy-Weisbach equation to yield

$$\Delta P = \frac{f}{D_h} L_e \frac{\rho V^2}{2g_c} \tag{2.10}$$

The friction factor in equation (10) is calculated from Altshul's equation:

$$f = 0.11\left(\frac{\varepsilon}{D_h} + \frac{68}{R_e}\right) \tag{2.11}$$

And the Reynolds number is given by

$$R_e = \frac{D_h V}{\nu} \tag{2.12}$$

where

ν = kinematic viscosity, m^2/s (ft^2/s)

Fan Modeling

The fan model was introduced for estimating airflows as a component of fluid flow networks (Clark 1985, Walton 1989). It uses fourth-order polynomial fits to the dimensionless head and efficiency to predict the pressure rise and power consumption. The fan similarity laws allow the dimensionless curves to be used to treat varying rotation speed and different diameters. The performance of a fan is characterized in terms of the pressure rise across the device and the shaft power requirements at a given fluid flow rate. These two characteristics are pressure head and efficiency. The two dimensionless performance curves that relate pressure head and efficiency to fluid flow rate are represented by polynomials with empirical coefficients that can be computed using manufacturer's data. These performance curves form the basis of the model. The mathematical description of fan model is described in greater detail in Chapter 3.

2.2.3 Design Constraints

The design specifications are introduced as constraints in the optimization problem and the design constraints define the viability of the design solution. Tsal and Adler (1987) defined design constraints necessary for duct optimization and they are shown at the following list (constraints 1 through 8). The constraint 9 is newly added for VAV systems.

1. Kirchhoff's first law. The summation of the flow at each node is zero.
2. Pressure balancing restriction. It is required that the pressure losses are the same for all the duct paths.
3. Nominal duct sizes. The manufacturer sets the standard increments of duct sizes. This study follows the 1-inch (0.025 mm) increment for duct sizes up to 20 inches (0.5m) and, then 2-inch (0.05mm) increment.
4. Air velocity limitation. This is for the limiting of duct noise.
5. Preselected sizes. Duct sizes for some sections may be predetermined.
6. Construction restriction. The allowable duct sizes can be restricted for architectural reasons.

7. Telescopic restriction. In some systems, the diameter of the upstream duct must not be less than the diameter of the downstream duct.
8. Standard equipment restrictions. Duct-mounted equipment is selected from the set produced by industry.
9. Duct static pressure control. For a VAV system with a variable speed fan, the fan speed is often controlled to maintain a minimum static pressure at the end of the longest duct line. A minimum static pressure is required on order that no terminal unit be starved for air. To save fan energy, it is desirable that this setpoint be as low as possible. Englander and Norford (1992) suggested setpoints of 1.5 in. wg (373 Pa) for September through May and 2.5 in. wg (622 Pa) throughout the summer. These setpoints were adopted for this study.

The duct static pressure control at the end of the longest duct line directly affects system pressure loss and, accordingly, the operating cost calculation. Assuming that the fan control system exactly maintains the specified duct static pressure at the end of the longest duct line, the total fan pressure is calculated by solving the system sequentially from the terminal duct section to the root of the system. Among the above 9 constraints, the constraints 1, 2 and 9 are enforced by the objective function and the constraints 5, 6, and 8 can be handled by either of two ways: (1) the constraint is added to the objective function as a penalty term to provide some penalty to limit constraint violations; (2) if the constraint states a predetermined duct size, then it can be assigned to the place of one of the variables and the number of dimensions is thereby decreased.

In this study, the duct size is used as an explicit design variable in the VAV optimization procedure. Thus, the design constraints that limit the design domain are the following:

- Nominal duct sizes;
- Air velocity limitation;
- Telescopic restriction.

The constraint of nominal duct sizes is treated in the integer/discrete programming technique and it is introduced in the following section. Air velocity limitation sets the boundaries for duct sizes. The recommended velocities for the control of noise generation are different depending on the application, however all categories fall within the range between 600 fpm (3 m/s) and 3000fpm (15 m/s) (Rowe, 1988). In this study, minimum and maximum air velocity limits are set to 600 fpm (3 m/s) to 3000fpm (15 m/s). Telescopic restriction limits the diameter of the downstream duct. Air velocity limitation and telescopic restriction are set as penalty terms of the transformed objective function.

2.3 VAV Optimization Procedure

In this section, the VAV optimization procedure is introduced. First, an overall scheme of the VAV optimization procedure is discussed. Second, the operating cost calculation, which is the important part of the objective function evaluation, is introduced in greater detail. Finally, an integer/discrete programming technique is discussed to find an optimum discrete solution from optimum continuous duct sizes.

2.3.1 Overall Scheme

As shown in Figure 2.1, the VAV optimization procedure is mainly comprised of preparation of airflow data, evaluation of the objective function, and generation of a design solution that includes continuous and discrete solutions. First, time-varying airflow rate data for VAV duct systems can be provided by an hourly building simulation program. Second, the evaluation of the objective function requires fan selection, initial cost calculation, search for fitting loss coefficients from duct fitting database, system pressure loss calculation and operating cost calculation. The fan selection algorithm is discussed in Chapter 3. The operating cost calculation was tested by modifying the AIRNET program and solving the duct system sequentially for duct paths. Both operating cost calculation methods are introduced later in this section. The detailed description of the objective function evaluation is given in Chapter 3 with the introduction of fan modeling. Third, an optimization method provides candidate design values for the estimation of system life-cycle cost. Before an optimization method is selected, the problem domain must be analyzed. The domain analysis is discussed in Chapter 4 in order to define the problem characteristics and suggest a proper optimization method. Duct sizes are explicit design variables and they have discrete values. Therefore, the program must handle discrete/integer programming problems. Discrete variables impose additional constraints on the design problem and the optimum cost function value is likely to increase when discrete values are assigned to variables. The objective function evaluation is comprised of selection of design variables with an optimization method, fan selection, system life cycle cost calculation. These essential constituents are iterated until a feasible and finally optimum design is attained.

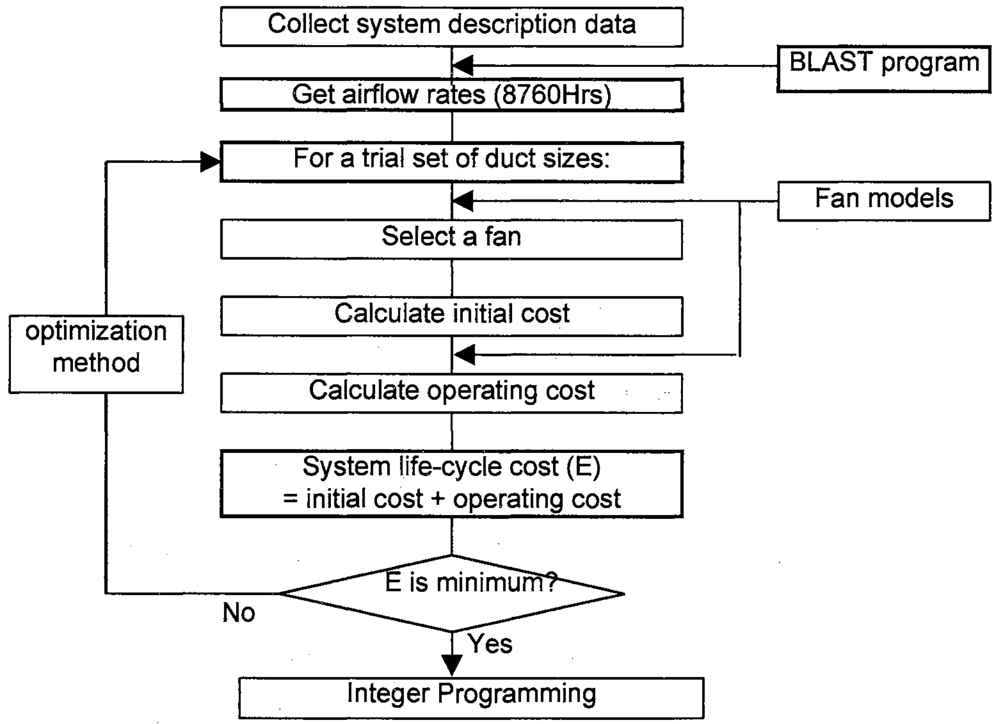


Figure 2.1 Optimum Duct Design Procedures

2.3.2 Operating Cost Calculation

The operating cost is fan energy consumption for a year's operation. It can be evaluated using two methods. First, build a VAV system as a network model and simulate the system with the modified AIRNET program. Second, assuming that the fan control system exactly maintains the terminal duct static pressure, find fan energy consumption by solving the system sequentially (sequential method) from the terminal duct section to the root of the system. Each of the methods applied to the system has pros and cons, however, the sequential method is adopted for this study since the computational time is so important. Both methods are discussed as follows.

Using the modified AIRNET program

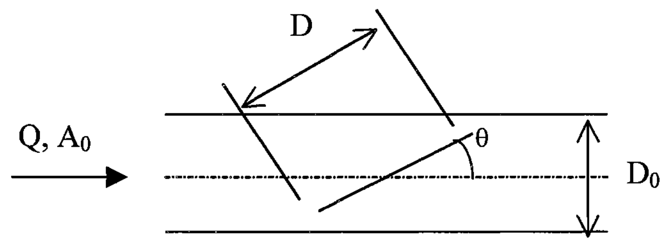
The program called AIRNET provides a methodology to estimate airflows and pressures in buildings. A detailed description is given in Section 1.2.3. In order to apply the program to VAV systems, the following two additional items must be considered; (1) modeling of VAV boxes, and (2) fan control to maintain the terminal duct static pressure constant.

As a simplification, VAV boxes are simulated as butterfly dampers and are considered duct fittings causing dynamic losses. Substituting $V = \dot{m} / \rho A$ into equation (1.22) and rearranging yield

$$C_0 = 2\rho A^2 \cdot \frac{\Delta P}{\dot{m}^2} \tag{2.13}$$

The airflow is known from the zone load calculation. As Krajnovich and Hittle (1986) discussed, pressure independent VAV boxes required one inch water gauge static pressure to operate properly. The fitting loss coefficient, C_0, can be calculated with the given airflow rate and pressure drop. Damper angles are determined by linear interpolation with the data of Table 2.1. Starting with the rated fan speed and fan airflow, the duct terminal static pressure is calculated by nonlinear solver. If the pressure is not in the range of setting point, then the fan speed is adjusted to satisfy the pressure setting point. Once the actual fan speed is found, the fan control parameter is calculated based on the rated fan speed and then the saved fan energy is found.

Table 2.1 C_0 Values of Butterfly Damper,
(1997 ASHRAE Fundamental Handbook, p. 32.32)



D/D_0	Angle θ											
	0	10	20	30	40	50	60	70	75	80	85	90
0.5	0.19	0.27	0.37	0.49	0.61	0.74	0.86	0.96	0.99	1.02	1.04	1.04
0.6	0.19	0.32	0.48	0.69	0.94	1.21	1.48	1.72	1.82	1.89	1.93	2.00
0.7	0.19	0.37	0.64	1.01	1.51	2.12	2.81	3.46	3.73	3.94	4.08	6.00
0.8	0.19	0.45	0.87	1.55	2.60	4.13	6.14	8.38	9.40	10.30	10.80	15.00
0.9	0.19	0.54	1.22	2.51	4.97	9.57	17.80	30.50	38.00	45.00	50.10	100.0
1.0	0.19	0.67	1.76	4.38	11.20	32.00	113.0	619.0	2010	10350	99999	99999

Figure 2.2 shows the program procedure. The mass balance equations at the nodes of a network model establish the simultaneous nonlinear equations. The nonlinear equations are then solved iteratively using a modified Newton's method with a skyline solution process until a convergent solution is obtained. The method using the modified AIRNET program for evaluation of objective function requires a substantial amount of computational time due to the iterations required to find a solution of the set of nonlinear equations. If the equations are repeated 8760 times for a year's operation and the system is complicated, the computational time is substantial. However, the modified AIRNET program gives all node pressures and element airflows, and controlled values of VAV boxes and fan speeds, such as damper angles and fan speed control parameter.

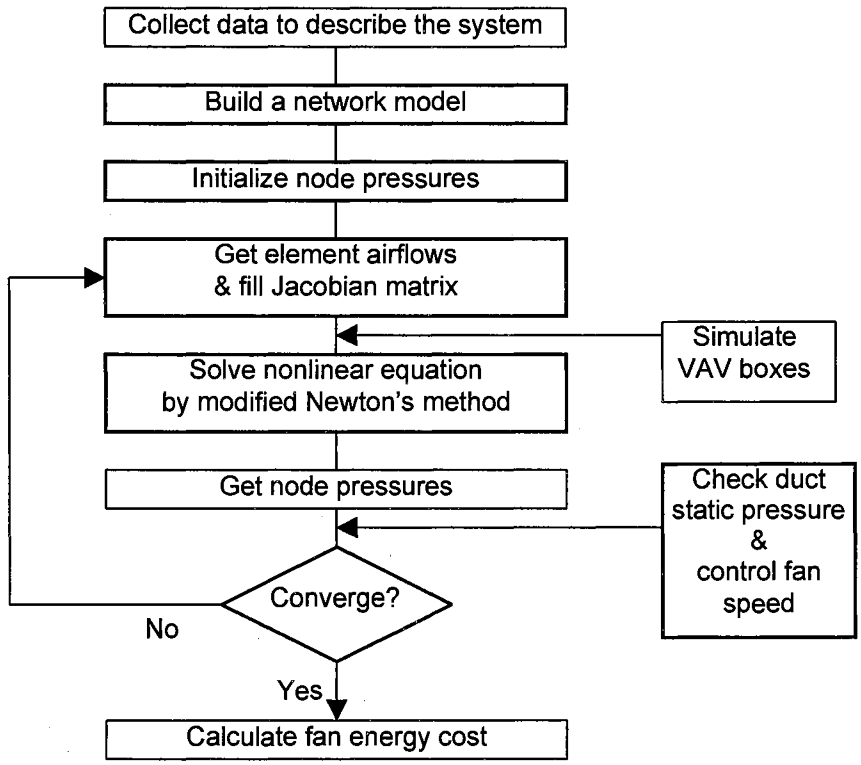


Figure 2.2 AIRNET Program Procedure.

Using the Sequential Method

If the VAV boxes are pressure-independent (refer to section 1.2.1), the system evaluations can be simplified. Assuming that the terminal duct static pressure is controlled exactly, the upstream node pressure can be calculated knowing the mass flow rate and element flow characteristics. Likewise, all pressures can be calculated from the terminal duct section to the root duct section sequentially. No iteration is required to solve the nonlinear equations. Substantial computational time is saved compared to the AIRNET method.

The programming for the sequential method can be done two ways: (1) using a linked list, (2) without a linked list. The sequential method with a linked list forms a binary tree structure with nodes. In order to get airflows and pressures, it is required to traverse the nodes of a binary tree several times. The sequential method without linked list does not form a binary tree. The equations of element flow characteristics are placed in order in the program from the terminal element to the root element. This method is fast in calculation, but the program must be changed for other systems. The first method can be applied to any duct system by changing an input file without modifying the program. However, it is slower than the latter method. In this study, first the sequential method without linked list is applied to several VAV duct systems. Then the program is generalized by providing information on duct structure in the input file. Figure 2.3 shows the program procedure of the sequential method.

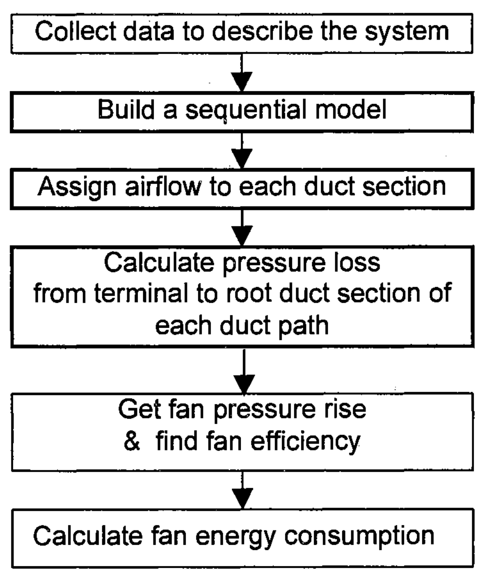


Figure 2.3 The Procedure of the Sequential Program

2.3.3 Integer/Discrete Programming

The constraint of nominal duct sizes restricts the diameter of a round duct, or the height and width of a rectangular duct to be rounded to the nearest lower or upper nominal size. This constraint will be resolved using discrete programming methods. The discrete programming method that was applied to the duct design problem is the penalty function method. Before the penalty method is introduced, the duct design problem is defined quantitatively.

The optimum duct design problem is defined as:

$$\text{Minimize} \quad E = E_p\,(PWEF) + E_s \tag{2.14}$$

$$\text{Subject to} \quad x_p - x_c \geq 0 \tag{2.15}$$

$$l_i \leq x_i \leq u_i \tag{2.16}$$

where E = system life cycle cost, \$

E_p = first year energy cost, \$

E_s = initial cost, \$

PWEF = present worth escalation factor, dimensionless.

x_p = parent duct section, inch

x_c = child duct section, inch

$x_i = [x_1, x_2, \ldots, x_n]^T = [X^d]^T$

$X^d \in R^d$: feasible subset of discrete duct sizes, inch

n = the number of duct sections

l_i and u_i = the lower and upper bounds of duct section i, inch

Equation (2.15) is due to telescopic restriction and Equation (2.16) is due to air velocity limitation and nominal duct sizes.

The penalty function method includes penalty terms for specified discrete value violation after obtaining the continuous optimum duct sizes. Then the penalty function is

$$F(X) = E + P(X^d) \tag{2.17}$$

The penalty term is defined as

$$P(X^d) = \gamma\, Q(X^d)^\beta \tag{2.18}$$

where $Q(X^d) = \sum_{j \in d} 4q_j(1-q_j)$ and $q_j = \dfrac{(x_j - s_j^l)}{(s_j^u - s_j^l)}$

s_j^l and s_j^u are the nearest feasible lower and upper discrete duct sizes.

As discussed in Chapter 1, it is recommended that β is 1 or 2, and the initial value of γ is computed from

$$\gamma = \frac{F(X^m) - f(X^m)}{Q^\beta} \tag{2.19}$$

where $X^m = (S^l + S^u)/2$

and $S^l = [s^l_1, \ldots s^l_n]$ and $S^u = [s^u_1, \ldots s^u_n]$ are the nearest lower and upper discrete duct sizes of the starting point X^0.

When the subsequent search is made iteratively in the solution process, the factor γ is gradually increased as follows

$$\gamma^{(k+1)} = c\gamma^{(k)} \tag{2.20}$$

where c is a constant value in the interval

$$1 < c < 2.$$

The solution process of the penalty function method is shown in Figure 2.4. When the optimization algorithm is carried out in order to find a continuous solution, the duct sizes are restricted by air velocity limitation. The continuous solution is then treated as a starting point in the search for a discrete solution to consider the nominal duct size constraint. From Equation (2.19), an initial value of the penalty factor γ is estimated and the search is made iteratively. In every iteration, a continuous optimization is performed with the Nelder and Mead downhill simplex method and the penalty factor γ is gradually increased as shown in Equation (2.20).

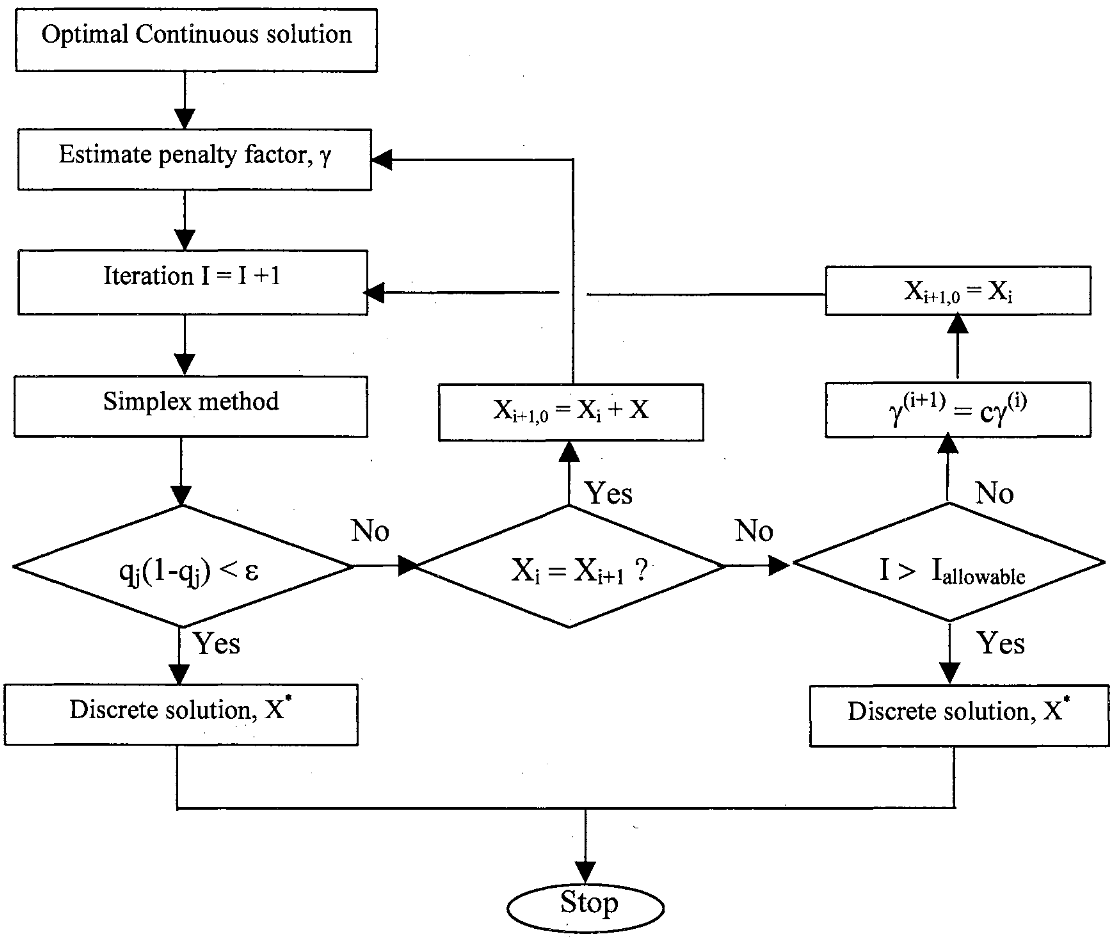


Figure 2.4 Solution Process of Discrete Programming

CHAPTER 3

FAN MODELING, SELECTION AND FAN POWER CALCULATION

Variable air volume systems require proper fan selection and efficient, stable operation over the entire airflow range. Once the system requirements for air volume and static pressure have been determined, a fan is selected with the consideration of initial cost versus operating cost, capacity control, available space, allowable noise level, and drive selection. Since VAV systems seldom operate at full design air volume, significant part load power savings at reduced air volume is realized through the proper fan operation.

Fans are controlled by placing a static pressure sensor in the downstream ductwork, typically two-thirds of the way down the longest duct path. This sensor is set at a static pressure that will ensure sufficient pressure to move the air from that point through the remaining ductwork. As VAV terminal units begin to close in response to a decreasing cooling load, static pressure in the ductwork increases. This causes the fan operating point to temporarily move upward to the left on a constant rpm curve as shown in Figure 3.1 (pt. A to pt. B). The static pressure sensor detects an increase in duct pressure and signals the fan to decrease speed until the static pressure is satisfied, moving the operation point to C. Fan operation is modulated to the point on the other fan curve of the decreased fan speed. Thus, for the evaluation of VAV systems, fans should be modeled to give an exact operating point according to the varying airflow requirement.

In addition to the change of fan operation, fan efficiency is also an important factor for the calculation of the fan operating cost. The required shaft power input is greater than the power input to the air because of inefficiencies. The ratio of the air power to the shaft power is the fan efficiency. The fan efficiency changes with pressure and airflow rate. For a given air volume and pressure requirement, a corresponding fan speed must be found to calculate the efficiency at that operating point. The performance of fans is generally given in the form of a graph showing pressure efficiency, and power as a function of capacity and mathematically represented by a polynomial regression. The fan model is validated and tested for the calculation of fan power consumption.

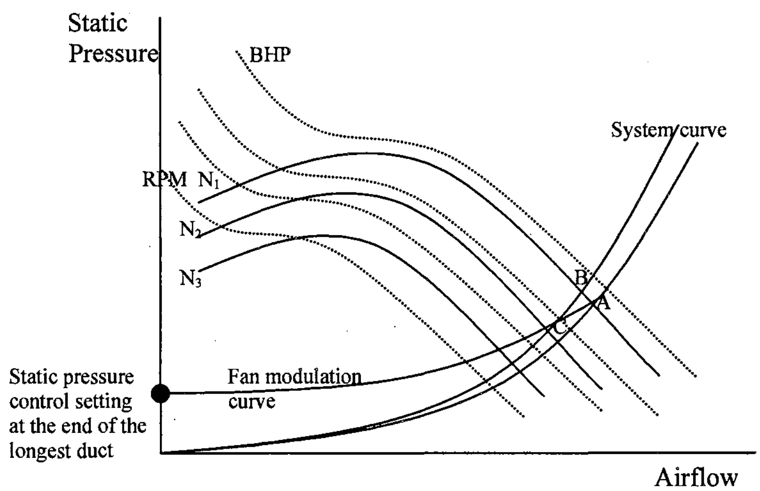


Figure 3.1 Fan Operation in VAV Systems

3.1 Fan Modeling and Validation

The fan model was introduced for estimating airflows as a component of fluid flow networks. The source of fan model can be found from Clark (1985) and Walton (1989). It uses fourth-order polynomial fits to the dimensionless head and efficiency to predict the pressure rise and power consumption. The fan similarity laws allow the dimensionless curves to be used to treat varying rotation speed and different diameters.

3.1.1 Mathematical Description

The performance of a fan is characterized in terms of the pressure rise across the device and the shaft power requirements at a given fluid flow rate. These two characteristics are pressure head and efficiency. The two dimensionless performance curves that relate pressure head and efficiency to fluid flow rate are represented by polynomials with empirical coefficients that can be computed using manufacturer's data. These performance curves form the basis of the model

In order to simplify and generalize the model, the dimensionless variables are defined using the fan similarity laws: flow coefficient and pressure head coefficient. The dimensionless flow coefficient is defined as

$$\phi = \frac{\dot{m}}{\rho N d^3} \tag{3.1}$$

where $\dot{m}$ = dry air mass flow rate, kg/s

ρ = entering moist air density, kg/m^3

N = fan speed, rpm

d = fan wheel diameter, m.

The dimensionless pressure head coefficient is defined as

$$\psi = \frac{\Delta P}{\rho N^2 d^2} \tag{3.2}$$

where ΔP = pressure rise across the fan, Pa.

It should be noticed that the use of dimensionless variables implicitly apply the fan laws for changes in speed, density, and diameter.

The fan efficiency is defined as

$$\eta_s = \frac{\dot{m} \cdot \Delta P}{\rho \cdot \dot{W}_s} \tag{3.3}$$

where $\dot{W}_s$ = shaft power, W.

The polynomial performance curves are

$$\psi = a_0 + a_1\phi + a_2\phi^2 + a_3\phi^3 + a_4\phi^4 \tag{3.4}$$

$$\eta_s = b_0 + b_1\phi + b_2\phi^2 + b_3\phi^3 + b_4\phi^4 \tag{3.5}$$

where $a_0, a_1, a_2, a_3, a_4, b_0, b_1, b_2, b_3, b_4$ are determined from the manufacturer's data.

The total system power that is used to calculate the operating cost is expressed in terms of the shaft power and the motor efficiency.

$$\dot{w}_t = \frac{\dot{w}_s}{\eta_m} \tag{3.6}$$

3.1.2 Example of Fan Modeling

A centrifugal fan is modeled as and example. Trane company fan model 12 BISW has the following specifications: wheel diameter = 12.25 inch, outlet area = 0.86 ft^2, blast area = 0.66 ft^2, max. rpm = 3900, max. static pressure = 8 in.wg, max. air volume = 4,200 cfm. In Table 3.1, air volume, pressure, and horsepower are obtained from the manufacturer's data as illustrated by the fan performance table of the centrifugal fan catalog. Given a nominal operating speed, entering fluid density, and the wheel diameter, the pressure and flow coordinates of the data can be converted to the dimensionless flow (ϕ) and head (ψ) variables.

Table 3.1 Fan Performance Data when Fan Speed is 3100 rpm

Air Volume Cfm	Outlet velocity ft/min	Static press. in.wg	Vel. press. in.wg	Total press. in.wg	Horse power hp	Flow coeff. . □	Press coeff. □	Efficiency □
3258.90	3789.32	0.125	0.895	1.020	2.188	0.9882	0.8177	0.2394
3236.75	3763.63	0.250	0.883	1.133	2.204	0.9815	0.9082	0.2622
3214.22	3737.50	0.375	0.871	1.246	2.219	0.9747	0.9986	0.2842
3192.39	3712.17	0.500	0.859	1.359	2.237	0.9680	1.0894	0.3055
3169.08	3685.13	0.625	0.847	1.472	2.253	0.9610	1.1796	0.3261
3146.46	3658.89	0.750	0.835	1.585	2.269	0.9541	1.2702	0.3461
3123.23	3631.95	0.875	0.822	1.697	2.285	0.9471	1.3605	0.3655
3098.87	3603.68	1.000	0.810	1.810	2.298	0.9397	1.4505	0.3844
3050.00	3546.60	1.250	0.784	2.034	2.328	0.9249	1.6305	0.4198
2999.53	3487.65	1.500	0.758	2.258	2.357	0.9096	1.8102	0.4527
2947.62	3427.24	1.750	0.732	2.482	2.392	0.8938	1.9897	0.4819
2894.90	3366.09	2.000	0.706	2.706	2.423	0.8778	2.1693	0.5095
2786.52	3240.36	2.500	0.655	3.155	2.484	0.8450	2.5286	0.5574
2657.69	3090.50	3.000	0.595	3.595	2.498	0.8059	2.8819	0.6026
2548.53	2963.29	3.500	0.547	4.047	2.573	0.7728	3.2442	0.6317
2393.10	2782.93	4.000	0.483	4.483	2.550	0.7257	3.5932	0.6628
2201.82	2560.11	4.500	0.409	4.909	2.493	0.6677	3.9345	0.6830
1845.83	2146.17	5.000	0.287	5.287	2.230	0.5597	4.2379	0.6893

- Air density = 0.075 lb_s/ft^3
- ϕ is obtained using equation (3.1)
- ψ is obtained using equation (3.2)
- η is obtained using equation (3.3).

The dimensionless performance is modeled by fitting the data to a polynomial. The relation between dimensionless head and dimensionless flow is represented as

$$\psi = -12.706 + 78.363\phi - 124.66\phi^2 + 81.339\phi^3 - 21.672\phi^4$$

The relation between efficiency and dimensionless flow is represented as

$$\eta_s = -3.1832 + 21.179\phi - 43.829\phi^2 + 41.372\phi^3 - 15.342\phi^4.$$

It should be noted that the above two equations should be used for $0.4 \le \phi \le 1$, which represents the performance in the stable operating range. For $\phi \le 0.4$, called the surge region of the fan performance curves, operation should not be attempted. In the surge region, backflow can occur in a cyclical fashion causing large intermittent forces on the fan blades with subsequent physical damage (Kreider and Rabl, 1994). The limiting value, $\phi = 0.4$ is obtained as shown in Table 3.2. The air volume for different fan speeds is found at the boundary of the stable operating range as given by the manufacturer in graphical form. The dimensionless flow coefficient ϕ is computed using equation (3.1).

Table 3.2 Dimensionless Flow Coefficient Value
at the Boundary of Stable Operating Range

Fan speed Rpm	Air volume cfm	Static press in.wg	Total press in.wg	Flow coeff ϕ
3900	1650	8.55	9.899	0.398
3700	1560	7.75	8.904	0.396
3400	1430	6.50	7.474	0.395
3100	1310	5.40	6.210	0.397
2800	1190	4.40	5.061	0.400
2500	1055	3.50	4.027	0.397
2200	920	2.65	3.058	0.393
1900	796	1.95	2.254	0.394
1600	670	1.36	1.576	0.394
1300	545	0.90	1.043	0.394

The dimensionless performance in the surge region is modeled as

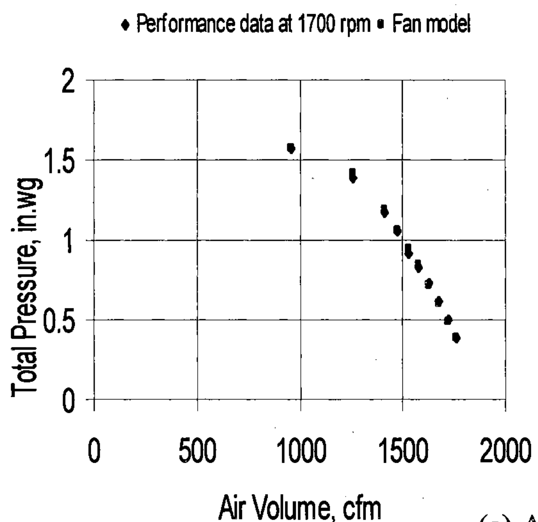
$$\psi = 3.8519 + 5.676\phi - 19.732\phi^2 + 5.4065\phi^3 + 45.444\phi^4$$

$$\eta_s = 0.0118 + 3.4203\phi - 11.056\phi^2 + 30.976\phi^3 - 38.474\phi^4$$

for $0 \le \phi \le 0.4$.

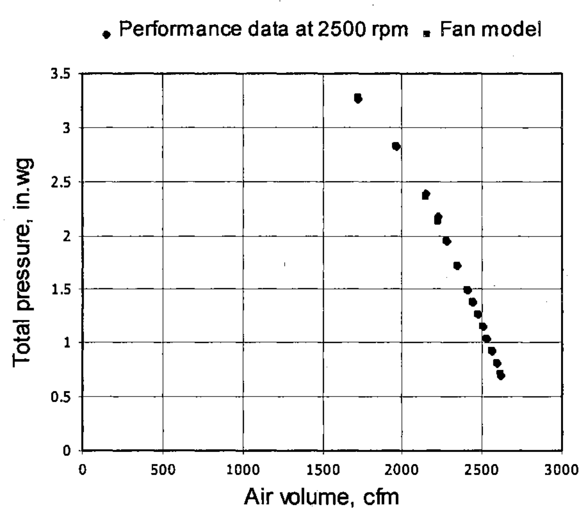
3.1.3 Validation of Fan Model

The fan model is validated by comparing the model to the value of performance table, for the pressure and efficiency at a given air volume and fan speed. Figure 3.2(a) shows the pressure comparison between the model and the performance data at 1700 rpm.



Air volume cfm	Press at 1700 rpm	Press from fan model	Error %
1758.70	0.3857	0.3869	0.29
1719.42	0.4992	0.4914	1.57
1675.69	0.6117	0.6035	1.33
1629.48	0.7237	0.7171	0.91
1577.27	0.8248	0.8394	1.77
1529.46	0.9182	0.9454	2.96
1475.00	1.0584	1.0590	0.06
1409.52	1.1675	1.1844	1.45
1257.99	1.3834	1.4212	2.74
955.47	1.5770	1.5779	0.06

(a) At 1700 rpm



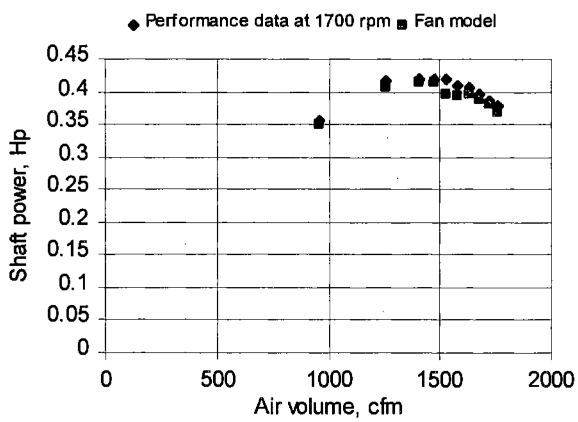
Air volume cfm	Press at 2500 rpm	Press from fan model	Error %
2618.16	0.7028	0.7087	0.84
2591.38	0.8159	0.8165	0.07
2563.04	0.9287	0.9287	0.00
2534.73	1.0416	1.0389	0.25
2505.13	1.1540	1.1521	0.16
2475.00	1.2664	1.2653	0.08
2444.91	1.3789	1.3762	0.20
2413.08	1.4909	1.4912	0.02
2348.79	1.7152	1.7161	0.06
2280.69	1.9385	1.9434	0.25
2222.40	2.1663	2.1287	1.74
2147.04	2.3886	2.3547	1.42
1968.33	2.8266	2.8235	0.11
1723.81	3.2505	3.2766	0.80

(b) At 2500 rpm

Figure 3.2 Fan Model Validation for Pressure

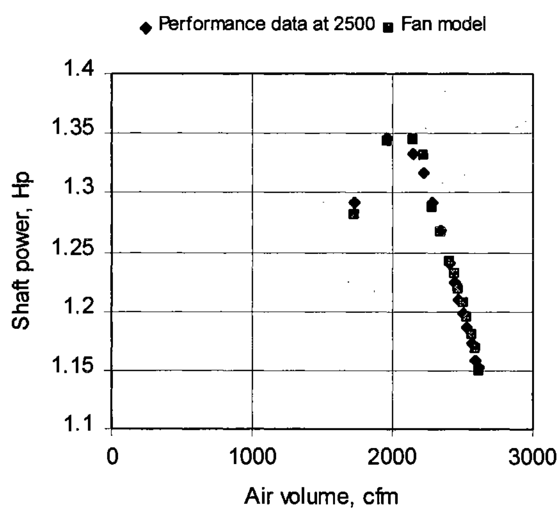
Figure 3.2(b) is at 2500 rpm. The data from the fan model shows a maximum 3% error at 1700 rpm and a maximum 1.74% error at 2500 rpm.

Next, the shaft power is compared between that performance data and the fan model. Figure 3.3 shows the shaft power comparison at 1700 rpm and 2500 rpm. The data from the fan model shows a maximum 5.89% error at 1700 rpm and 1.05% error at 2500 rpm.



Air volume cfm	Power at 1700 rpm	Power from fan model	Error %
1758.70	0.379	0.367	3.16
1719.42	0.387	0.382	1.36
1675.69	0.396	0.389	1.82
1629.48	0.407	0.396	2.57
1577.27	0.409	0.395	3.64
1529.46	0.420	0.397	5.89
1475.00	0.420	0.415	1.17
1409.52	0.419	0.414	1.02
1257.99	0.416	0.406	2.34
955.47	0.355	0.350	1.51

(a) At 1700 rpm



Air volume cfm	Power at 1700 rpm	Power from fan model	Error %
2618.16	1.153	1.150	0.29
2591.38	1.158	1.169	0.89
2563.04	1.173	1.181	0.67
2534.73	1.186	1.195	0.73
2505.13	1.199	1.206	0.65
2475.00	1.210	1.218	0.66
2444.91	1.224	1.232	0.65
2413.08	1.241	1.242	0.11
2348.79	1.269	1.266	0.18
2280.69	1.292	1.287	0.39
2222.40	1.317	1.331	1.05
2147.04	1.333	1.345	0.89
1968.33	1.345	1.343	0.19
1723.81	1.291	1.281	0.79

(b) At 2500 rpm

Figure 3.3 Fan Model Validation for Shaft Power

3.2 Fan Selection

When selecting a fan, the following information must be known. These factors govern the type of fan to be selected and its size:

- Peak airflow rate.
- Static pressure at a peak volumetric flow rate.
- Efficiency: select a fan that will deliver the required volume at the expected static pressure with minimum horsepower.

Among the above factors, the fan selection in the VAV duct optimization was mainly based on the system pressure requirement at a peak hour's design airflow that gives maximum airflow through a year's operation. The pressure drop at every duct path was calculated and the highest pressure drop was selected as the one determining fan selection. Starting from the smallest fan size, the system design point that includes the system pressure requirement and peak airflow was checked as to whether that point is on a fan characteristic curve at which the fan can operate. If the fan cannot support that design point, the next bigger size of fan is entered in the computation. After a suitable fan was selected, fan efficiency was calculated for the airflow through a year's operation. The desired fan efficiency calculation is necessary to assure that the fan can deliver the required airflow at the desired static pressure within the fan operational range.

When two sizes of fans are selected, a smaller size of fan may save initial cost, but the operating cost is higher than a larger fan with better efficiency for a given airflow. In a sense, the fan that saves initial cost might be a designer's choice, however when the VAV duct system is large and consumes much electrical power, the next bigger size of fan will likely be more cost effective. The fan selection process resolves itself into the following four points.

- Prepare fan performance data
- Find the system pressure at a peak hour's airflow.
- Select the smallest fan that satisfies the system design point.
- Calculate fan efficiency for different airflows

3.3 Fan Power Calculation

Using the fan model, fan power can be calculated for a given air flow and pressure rise. The dimensionless performance curve relating pressure head and air volume includes fan speed (equation 3.4). The equation is rearranged with respect to fan speed and the fan speed at a given air flow and pressure can be found using a numerical root-finding method. Once the fan speed is found, the shaft efficiency is obtained from equation 3.5. The shaft power is computed with the resulting efficiency. Consequently, the total power consumption is computed with motor efficiency. The algorithm to calculate the fan power is as follows:

(1) Compute the system pressure loss for a given airflow rate (Q, ΔP)

(2) Find the fan speed (RPM)

Rearrange equation (3.4) with respect to fan speed:

$$F(N) = a_0 + a_1\,\phi(N) + a_2\,\phi^2(N) + a_3\,\phi^3(N) + a_4\,\phi^4(N) - \psi(N) = 0,$$

Find the fan speed using the Newton-Raphson or bisection method.

(3) Compute the fan efficiency, η_s, using equation (3.5).

(4) Compute the shaft power : $$H_s = \frac{\Delta P \cdot Q}{\eta_s},$$

(5) Compute the total system power: $$H_m = \frac{H_s}{\eta_m}$$

The algorithm is applied to a duct system that has 3009 cfm design airflow. As a preliminary test for VAV systems, air volumes that are supplied to the systems are ranged from 20% to 100% of the design air volume. For different airflows, the algorithm for operating cost calculation is checked for the efficiency and accuracy. Table 3.3 shows a computation result and the result is compared to the performance data from a Trane Co. fan catalog. The algorithm reasonably calculates the fan shaft power in all airflow ranges.

Table 3.3 Fan Power Computation

Airflow Cfm	Pressure in.wg	From computation		From fan catalog		Difference in Hp-shaft, %
		RPM	Hp-shaft	RPM	Hp-shaft	
3009 (100%)	3.249	3276	2.891	3279	2.899	0.28
2257(75%)	3.147	2746	1.772	2752	1.776	0.23
1505 (50%)	2.561	2203	0.879	2219	0.891	1.35
1204 (40%)	2.374	2080	0.657	2100	0.66	0.45
903 (30%)	2.206	2121	0.591	1960	0.50	18.0
601 (20%)	2.083	1934	0.352	1930	0.36	2.22

CHAPTER 4

THE CHARACTERISTICS OF OPTIMUM DUCT DESIGN PROBLEMS

Selection of an optimum design method for duct systems is based on various preliminary design analyses investigating the characteristics of the duct design problem. With a combination of design values of duct sections, a duct system is evaluated to represent system life-cycle cost. The systematic analysis of the results illustrates the characteristics of the problem domain and clarifies whether the problem has local minima or only a minimum. It will also help suggest an optimum duct design method to find duct sizes and to select a fan that minimizes system life-cycle cost.

The optimum duct design problem has the objective function of system life-cycle cost and constraints, such as nominal duct sizes, air velocity limitation, and telescopic restriction as defined in Chapter 3. Tsal and Behls (1986) analyzed a 2-dimensional hypothetical CAV duct system using a scalar field technique, which is the graphical representation of the objective function. They found a global minimum and the contour map has a convex shape that has a steep slop at the low pressure drop side and a gentle slop at the high pressure drop side. The analysis is based on pressure losses of duct sections in relation to life-cycle cost to compare existing duct design methods. However, in this study, duct system is analyzed based on the hydraulic diameter of duct sections.

The problem domain is analyzed for both CAV and VAV duct systems. The approaches for the problem domain analysis are introduced first and then, the computation results are discussed for CAV systems and VAV systems. The problem characteristics are concluded through computation results. An example duct system is selected from Tsal et al, 1988 (part II) for the analysis of CAV duct systems, and was tested as the 2- and 4- dimensional cases. A hypothetical three-section duct system is created for the analysis of VAV duct systems.

4.1 Approaches for the Problem Domain Analysis

The duct size is assumed to be a continuous variable since the purpose of the study is to analyze the behavior of the problem in the feasible region. The objective function is not differentiable with respect to duct sizes, so two search methods are employed for analysis: (1) exhaustive search, and (2) Nelder and Mead's downhill simplex method. The data from exhaustive search are used for the graphical representation of the objective function.

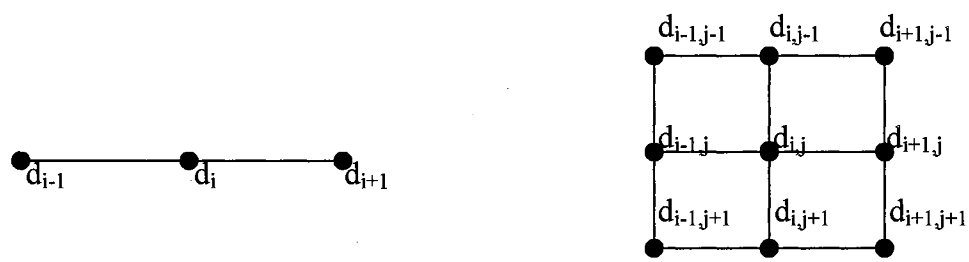
4.1.1 Exhaustive Search

An exhaustive search computes the function value at all discrete points in the problem domain and obtains any plausible local minima by comparing every point with its neighborhood. The neighborhood of N of the point $\mathbf{x}^*$ is mathematically defined as the set of points

$$N = \{ \mathbf{x} \mid \mathbf{x} \in S \text{ with } \| \mathbf{x} - \mathbf{x}^* \| < \delta \}$$

for some small $\delta > 0$.

In the exhaustive search, δ is set to a search discrete step size of Δd. Thus, the neighborhood is any point that any coordinate of a point has d ± Δd. In 1-dimensional space, the neighborhood of a point comprises 2 points as shown in Figure 4.1(a). In 2-dimensional space, there are 8 neighbor points as shown in Figure 4.1(b). Generally, n-dimension has 3^n-1 neighbors.



(a) 1-dimension: 2 neighbors (b) 2-dimension: 8 neighbors

Figure 4.1 Neighborhood in Exhaustive Search

If the exhaustive search finds several local minima with a small Δd, then those local minima should be tested as to whether they are truly local minima by searching each neighborhood with a smaller discrete step size. If the search shows that it has one global minimum, the minimum point can be accepted as an optimal solution with the interval of uncertainty of a discrete step size and the direct search method can be applied to the duct design problem. The evaluation and search procedure for a 5-section duct system is shown in Figure 4.2.

For a 2-dimensional case, the problem can be solved using the graphical method. The constraints identify the feasible region and the objective function contours are plotted through the feasible region. It is then possible to visually identify the shape of the objective function and an optimum design having the least cost. Once the graphical shape of the objective function is identified, a mathematically analogous test can be taken to the problem by deriving a function that gives a similar shape.

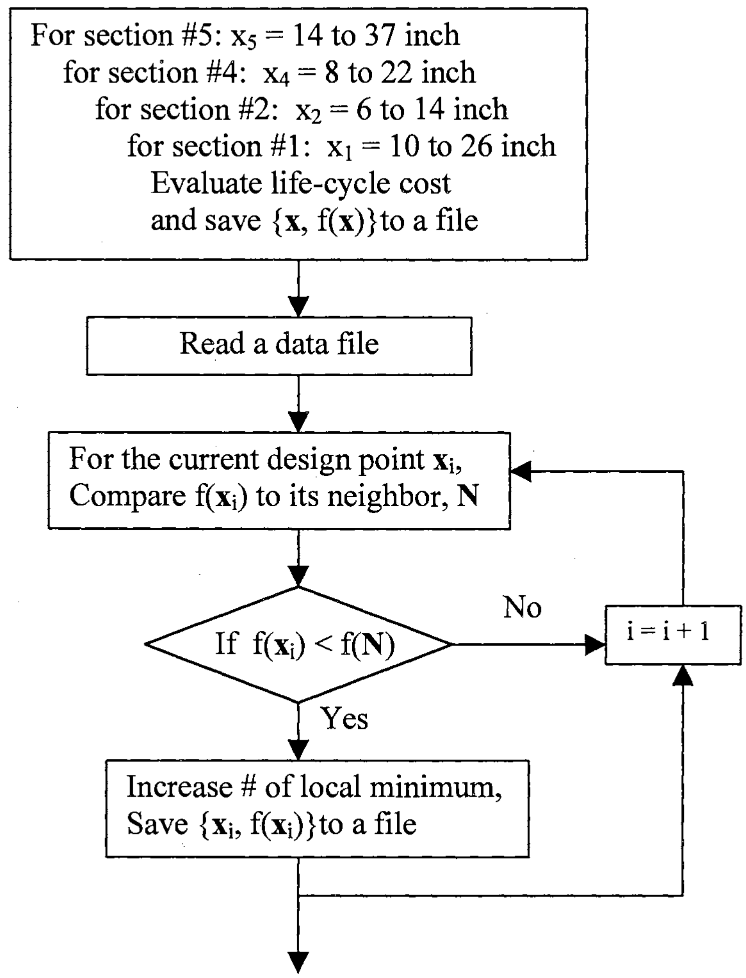


Figure 4.2 Evaluation and Search Procedure of 5-Section Duct System

4.1.2 Nelder and Mead's Downhill Simplex Method

Assuming one global minimum exists in the design domain, a direct optimization method can be applied to the duct design problem. A starting point is chosen at an extreme point, such as the corner point of the boundary feasible region and the search result is compared to the one obtained with different starting points. If the same optimum is found for a variety of starting points, it may be inferred that there is only a global minimum. The down hill simplex method (Nelder and Mead, 1965) is simple in calculations and uncomplicated in logic. The method is also effective when evaluation errors are significant because it operates on the worst rather than the best point (Reklaitis et al. 1983). Thus, in this study, the downhill simplex method is applied to obtain an optimal design point.

With a starting point $\mathbf{P}_0$, the initial simplex takes the other points using

$$\mathbf{P}_i = \mathbf{P}_0 + \lambda \mathbf{e}_i \tag{4.1}$$

where $\mathbf{e}_i$ is unit vectors

λ is a constant of the length scale

The steps that are taken in the simplex method are reflections, expansions, and contractions. The simplex is reflected away from the high point and if possible, it is expanded away on one or another direction to take larger steps. When it reaches a valley floor, the simplex is contracted in the transverse direction and oozes down the valley. The termination criterion used for this study is the rate of the difference to the sum of the minimum and maximum function values of the simplex. One thing to be noted is that the criteria might be fooled by a single anomalous step that failed to get anywhere, so it is recommended to restart the minimization routine at a point where the current step is at a minimum (Press et al. 1992).

4.2 Problem Domain Analysis for a CAV System

4.2.1 An Example Duct System for Analysis

An example is taken from Tsal et al. (1988b): It is a 4-dimensional 5-section duct system, in which one of the 5 duct sections has a fixed value. Figure 4.3 is a schematic diagram of the example system. The system parameters are summarized in Tables 4.1, 4.2, and 4.3, and they are economic, general, and sectional data respectively. In Table 4.3, the duct section 3 is preselected to 0.330m (13 inch), and airflows at terminal duct sections 1, 2, and 4 are 0.70, 0.22, 0.50 m^3/s respectively. It is defined to have constant airflows and to operate 4400 hrs throughout the year.

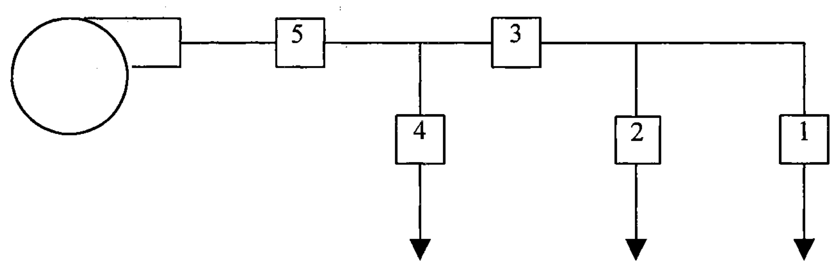


Figure 4.3 Five-Section Duct System

Table 4.1 Economic Data

Energy cost (E_p)	2.03 c/kwh
Energy demand cost	13 \$/kwh
Duct cost (S_d)	43.27 \$/m^2
System operation time (Y)	4400 h/yr
Present worth escalation factor (PWEF)	8.61

Table 4.2 General Data

Data	SI units	IP units
Air temperature (t)	22 degree C	71.6 degree F
Absolute roughness of aluminum duct (ε)	0.0003 m	0.00098 ft
Kinematic viscosity (ν)	1.54×10^{-5} m^2/s	1.66×10^{-4} ft^2/s
Air density (ρ)	1.2 kg/m^3	0.75 lb/ft^3
Fan efficiency (η_f): peak operating	0.85 0.75	0.85 0.75
Motor efficiency (η_e)	0.80	0.80
Total system airflow (Q_{fan})	1.42 m^3/s	3010 cfm

Table 4.3 Sectional Data of 5-Dection Duct System

Sections			Air flow, m^3/s	Duct length, m	Additional pressure loss, Pa	Duct size, m			C – coeffi-cient
Sec	Ch1	Ch2	Airflow	Length	DP_z	Height	Width	Dia	
1	0	0	0.70	14.00	25.0	0.254			0.80
2	0	0	0.22	12.00	37.5				0.65
3	1	2	0.92	8.00				0.330*	0.18
4	0	0	0.50	16.00					0.65
5	3	4	1.42	19.81	37.5				1.50

* Duct section 3 is fixed to 0.330 m (13 inch).

<u>Duct Section Diameter Bounded by Air Velocity Limitation</u>

The feasible region, which is referred to as a constraint set, is a collection of all feasible designs and it shrinks when constraints are added in the design model. The constraint that is applicable explicitly to duct size is air velocity limits for acoustic reasons. Minimum and maximum air velocity limits are set to 2 m/s and 16 m/s (Rowe, 1988). According to the airflow rates in Table 4.4, the duct size is bounded using equation (4.1). The bounded duct size of the 5-section duct system is shown in Table 4.4.

$$d = \sqrt{\frac{4}{\pi}\frac{Q}{V}} \tag{4.2}$$

Table 4.4. Bounded Duct Sizes of 5-Section Duct System

Duct section	Airflow, Q M^3/s	Duct size when V = 16m/s, inch	Duct size when V = 2m/s, inch
1	0.70	10 (0.24 m)	26 (0.67 m)
2	0.22	6 (0.13 m)	14 (0.37 m)
4	0.50	8 (0.20 m)	22 (0.56 m)
5	1.42	14 (0.34 m)	37 (0.95 m)

* Duct section 3 is preselected to 13 inches.

4.2.2 Computation Results and Discussion

The question of local or global optimum always arises in the optimum design of systems. One approach to answering the question requires showing the optimization problem is convex, since in that case any local minimum is also a global minimum. Mathematically, it is defined as follows: If $f(\mathbf{x}^*)$ is a local minimum for a convex function $f(\mathbf{x})$ on a convex set S, then it is also a global minimum. The convexity of the objective function is defined if and only if the Hessian matrix of the function is positive definite at all points in the convex set S (Arora 1989). However, in duct design problems, the Hessian matrix of the objective function cannot be derived since the function is not differentiable with respect to the duct size. The objective function is comprised of several procedures that must be evaluated consequently. Because the use of a convexity test is impossible, one has to use other approaches to analyze the duct design problem, such as exhaustive search and graphical representation. However, it may be impossible to definitively prove that there is a global minimum if the design space cannot be tested for convexity.

For the purpose of characterization and computation of local or global minima of the duct design problem, the analysis of the problem domain is as follows:

- Exhaustive search of the problem domain
- Graphical representation of the problem using a contour map
- Analysis by graphical analogy
- Employment of a direct optimization method, such as the Nelder and Mead downhill simplex method.

First, the example given in Section 4.2.1 is exhaustively searched as 4-and 2-dimensional cases in order to characterize local or global minima that will be found with the search. The function values in a 2-dimensional space can be used to draw contour maps to identify the surface shape of the objective function in the problem domain. In case that the exhaustive search cannot determine the structure of the objective function, a function with a similar shape is created and analyzed to find out whether it has the same characteristics as the given problem. Finally, if the above analyses show that the problem has a global minimum, the Nelder and Mead downhill simplex method is applied to the problem as a check on the global optimality of duct design problems.

<u>Exhaustive search</u>

When the exhaustive search was applied to the example system with a discrete step size of 1 inch, two design points are identified as those having neighboring points with higher functional values. These two points may be called apparent local minima: "apparent" because, while they appear to be local minima, it has not yet been established whether or not they are local minima. Their design values are shown in Table 4.5. It shows that one of the design values coincides with the rounding-off optimum value of Tsal et al, 1988 (Part II). The results from the exhaustive search have the same size in ducts 4 and 5 and have a 2 and 1 inch difference in ducts 1 and 2 respectively.

Table 4.5 Apparent Local Minima of 4-Dimension 5-Section Duct System when the Discrete Step Size is 1 inch

Optimization methods	Duct 1 inch	Duct 2 inch	Duct 4 Inch	Duct 5 Inch	Life-cycle cost(E) $
Exhaustive search	13	8	9	17	4481.668
	11	7	9	17	4482.365
T-method	11	7	9	17	4482.365

Since the purpose of the exhaustive search is the characterization and computation of local or global minima, the neighborhood of the two points from the exhaustive search is further subdivided with fine increments and exhaustively searched again. Ducts 4 and 5 have no change in the optimum integer value, so they are fixed to 9 and 17 inches respectively. Therefore, the 5-section duct system becomes a 2-dimensional problem that has two variables, namely duct 1 and 2. By changing the original problem into a 2-dimensional case, the analysis becomes simpler and the function values in the domain can be exploited for the graphical analysis of the problem. The step size is now further subdivided into 0.1 and 0.01 inch and applied only to the two duct sections of ducts 1 and 2. Table 4.6 shows apparent local minima when the discrete step size is 0.1 and 0.01 inch. The number of apparent local minima is increased with the decrement of the step size for the exhaustive search. The data in Table 4.6 are plotted on the graph for visual inspection as shown in Figure 4.4. From Figure 4.4, one can find that:

(1) The apparent local minima appear regularly, three steps on the duct 1 axis and two steps on the duct 2 axis.

(2) The two lines linking apparent local minima in each step size do not coincide.

(3) Those points are apparent local minima whose neighbors have higher function values, however their values decrease from both ends to the lowest local minimum (4456.56).

From the above observation, it is doubted whether those apparent local minima are true local minima. Thus, it is decided to search the neighbor of the apparent local minima with a series of 1-dimensional line searches. The line searches have been done on duct 2 with 0.0001-inch step size for every 0.01-inch of duct 1. Table 4.7 shows the local minimum data found with the line searches. The data in Table 4.7 is plotted in a 3-dimensional space for visual inspection as shown in Figure 4.5.

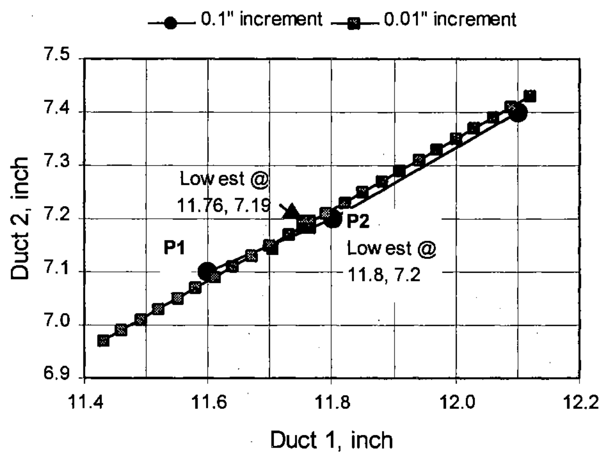
Table 4.6 Apparent Local Minima of the 2-Dimensional 5-Section Duct System

(a) when the discrete step size is 0.1 inch (* lowest value)
(b)

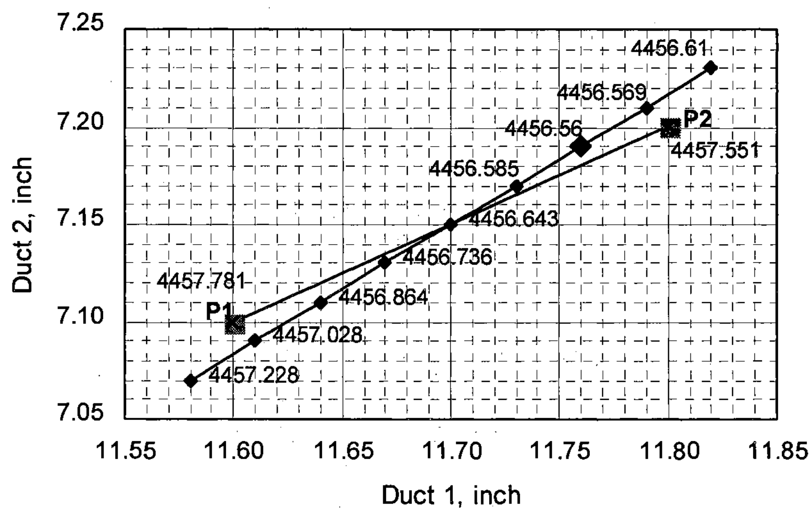
Duct 1 (10~26 in.)	Duct 2 (6~14 in.)	E-cost ($)
11.6	7.1	4457.781
11.8	7.2	4457.551*
12.1	7.4	4459.484

(b) when the discrete step size is 0.01 inch (* lowest value)

Duct 1 Inch	Duct 2 Inch	E-cost ($)	Duct 1 inch	Duct 2 inch	E-cost ($)	Duct 1 inch	Duct 2 inch	E-cost ($)
11.43	6.97	4458.798	11.67	7.13	4456.736	11.91	7.29	4456.926
11.46	6.99	4458.406	11.70	7.15	4456.643	11.94	7.31	4457.093
11.49	7.01	4458.054	11.73	7.17	4456.585	11.97	7.33	4457.290
11.52	7.03	4457.741	11.76	7.19	4456.560*	12.00	7.35	4457.517
11.55	7.05	4457.466	11.79	7.21	4456.569	12.03	7.37	4457.777
11.58	7.07	4457.228	11.82	7.23	4456.610	12.06	7.39	4458.102
11.61	7.09	4457.028	11.85	7.25	4456.684	12.09	7.41	4458.457
11.64	7.11	4456.864	11.88	7.27	4456.789	12.12	7.43	4458.843



(a) Apparent local minima found with 0.1 and 0.01 inch step sizes



(b) Detail view of data points between P1 and P2

Figure 4.4 Apparent Local Minima in the 2-Dimensional 5-Section Duct System

Table 4.7 Local Minimum Found with the Line Search on Duct 2

Duct 1 inch	Duct 2 inch	Cost $	Duct 1 Inch	Duct 2 inch	Cost $	Duct 1 inch	Duct 2 inch	Cost $
11.58	7.0684	4457.162	11.71	7.1547	4456.538	11.84	7.2417	4456.588
11.59	7.0750	4457.089	11.72	7.1614	4456.520	11.85	7.2484	4456.618
11.60	7.0816	4457.020	11.73	7.1680	4456.503	11.86	7.2551	4456.651
11.61	7.0882	4456.957	11.74	7.1747	4456.491	11.87	7.2618	4456.689
11.62	7.0949	4456.896	11.75	7.1814	4456.484	11.88	7.2686	4456.731
11.63	7.1015	4456.839	11.76	7.1881	4456.481	11.89	7.2753	4456.775
11.64	7.1081	4456.788	11.77	7.1947	4456.481*	11.90	7.2820	4456.824
11.65	7.1148	4456.740	11.78	7.2014	4456.484	11.91	7.2888	4456.876
11.66	7.1214	4456.695	11.79	7.2081	4456.492	11.92	7.2955	4456.931
11.67	7.1281	4456.657	11.80	7.2148	4456.503	11.93	7.3023	4456.991
11.68	7.1347	4456.620	11.81	7.2215	4456.518	11.94	7.3090	4457.054
11.69	7.1414	4456.590	11.82	7.2282	4456.538	11.95	7.3158	4457.120
11.70	7.1480	4456.562	11.83	7.2349	4456.562	11.96	7.3226	4457.191

*lowest function value

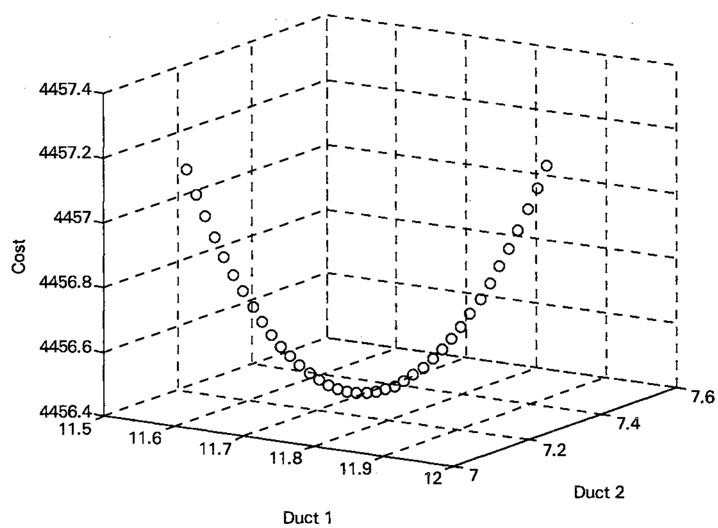


Figure 4.5 Local Minimum Points Searched along Duct 2

In order for the apparent local minima to be local minima, their neighbors cannot have a lower value. In the 2-dimensional exhaustive search, Table 4.6 and Figure 4.4 showed that the neighbors of the apparent local minima have higher values; the apparent local minima appeared every three steps on duct 1 and 2 steps on duct 2. However, Table 4.7 and Figure 4.5 shows that a lower value appeared in the neighboring boundary of apparent local minima and the minimum values are decreased in order toward the lowest one. Thus, the apparent local minima cannot be called as local minima any more. Figure 4.6 explains this situation. The figure has nine search nodes from point 1 to 9: point 5 was identified as an apparent local minimum in Table 4.6.

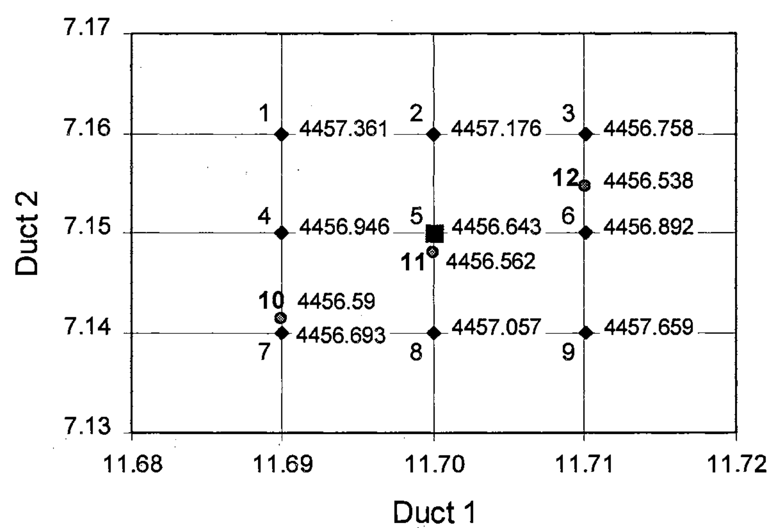
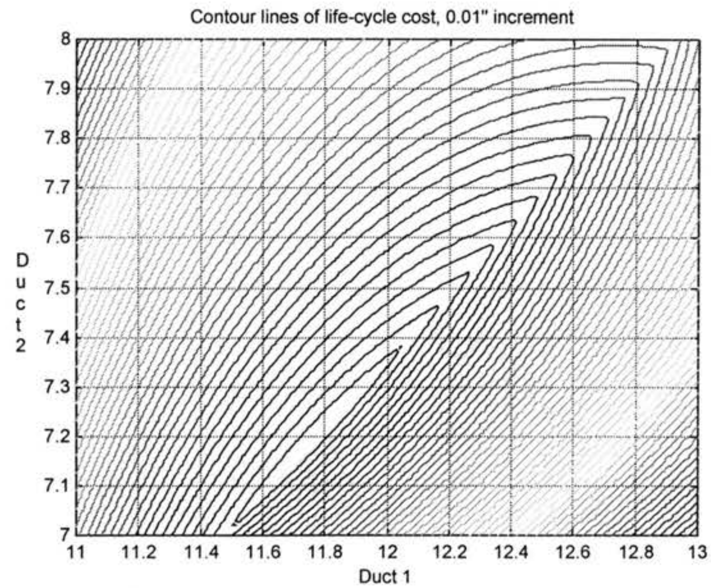


Figure 4.6 Neighbors of an Apparent Local Minimum

The very deep skewed valley of the objective function passes by point 10, 11, and 12. When the exhaustive search is employed, the node of points 5 is identified as an apparent local minimum since it is closest to the valley and surrounded by higher values. However, point 5 is not a local minimum since the lower function value is found at point 12 that is located in the neighboring boundary. This type of point arrangement repeatedly appears at every apparent local minimum. The lowest minimum from the line searches has the position like point 12 and it is the only point whose neighboring points have higher values including its boundary, so a very likely global minimum. Depending on the step size of the search, a slightly different location will be found as a global minimum. From this observation, it can be stated that a very likely global minimum is found at the lowest apparent local minimum with the exhaustive search or at the lowest minimum with the line searches.

Graphical representation

Some optimization problems can be solved by visually inspecting their graphical representation. The graphical representation also identifies the surface shape of the objective function. The objective function of a duct design problem was evaluated at every design point in the 2-dimensional region using the exhaustive search with 0.01-inch discrete step size. Those function values are now used for graphical representation to characterize the problem domain. For surface plots, contours are drawn on 2- and 3-dimensional spaces as shown in Figure 4.7. It can be seen that the objective function has a steep slope and a peaked hood-like convex shape. From the contour map, one can find the optimum solution lies between the two points of (11.5, 7.05) and (12.1, 7.35). The graphical representation also suggests that the problem very likely has only a global minimum.



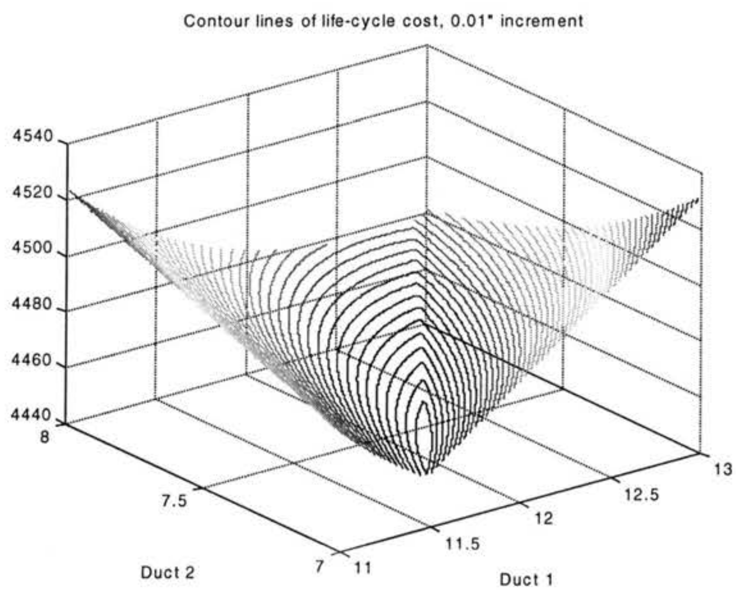


Figure 4.7 Contour Map of System Life-Cycle Cost

Analysis by graphical analogy

A question still arises in the duct optimization problem: why does the objective function show apparent local minima in the exhaustive search? In order to answer to this question, a function that has a similar shape is created for an analogical test. The function is

$$f(x,y) = -\sin(x+\pi/4)^{100} \, . \, \cos(y/2) + 2 \tag{4.3}$$

which has a minimum, f = 1.0 at x=1.34 and y = 0.75. This function is rotated 30° from the x-axis as shown in Figure 4.8.

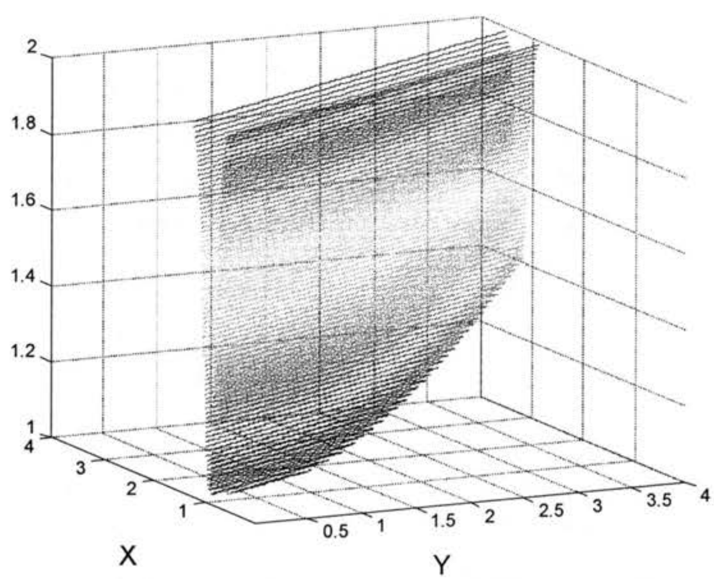


Figure 4.8 Contour Map of $-\sin(x+\pi/4)^{100} \, . \, \cos(y/2) + 2$

The function has only one local minimum. However, when it is exhaustively searched, it appears to have many local minima as shown in Figure 4.9. Fifty-one apparent local minima were found using the exhaustive search with 0.01 step size and they are arranged in descending order toward the lowest one. As shown in Figure 4.8, the function has very deep skewed valley. When the bottom of the deep valley is close to the search point, the point is brought into the apparent local minima as investigated in the duct design problem.

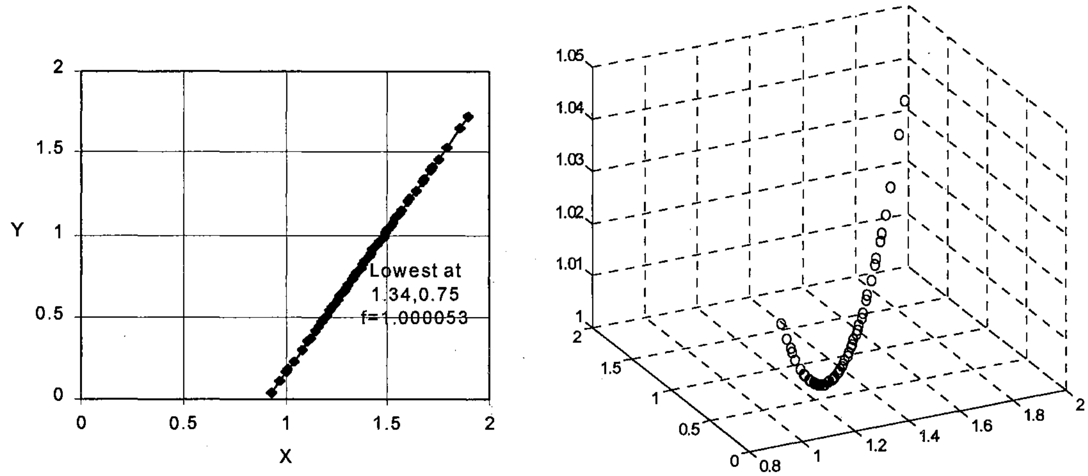


Figure 4.9 Local Minima of the Function Defined in Figure 4.8

The Nelder and Mead Downhill Simplex Method

In a 2-dimensional graphical representation, it is found that the problem domain has a global minimum. The downhill simplex method is now applied to the 2- and 4-dimensional duct systems. The initial starting point for a search is chosen at an extreme point, such as the corner point of the bounded region. The other N points of a simplex are defined by

$$\mathbf{P}_i = \mathbf{P}_0 + 1.0\ \mathbf{e}_i,$$

where the $\mathbf{e}_i$ is an N unit vector, and 1.0 is the problem's characteristic length scale. The fractional convergence tolerance (ftol) of the function value for a simplex routine is set to 1.0e-8. The function tolerance to stop routine iteration is set to 0.5. The result is shown at Table 4.8.

Table 4.8 Global Minimum Found with the Downhill Simplex Method

(a) 2-dimensional duct systems (ftol: 10e-8)

Starting point (duct 1, duct 2)	Duct 1 inch	Duct 2 inch	Life-cycle cost, $	Function evaluations
10, 6	11.764	7.191	4456.478	147
10, 14	11.765	7.191	4456.478	163
26, 6	11.766	7.192	4456.479	195
26, 14	11.765	7.192	4456.479	168

(b) 4-dimensional duct system (ftol: 10e-8)

Starting point (duct 1, 2, 4, 5)	Duct 1 inch	Duct 2 inch	Duct 4 Inch	Duct 5 Inch	E-cost $	Func. Eval's
10, 6, 8, 14	11.371	6.931	8.353	16.769	4423.386	469
10, 6, 8, 37	11.370	6.930	8.353	16.753	4423.388	503
10, 6, 22, 14	11.364	6.926	8.351	16.767	4423.385	554
10, 6, 22, 37	11.365	6.927	8.352	16.765	4423.385	637
10, 14, 8, 14	11.363	6.926	8.351	16.763	4423.385	394
10, 14, 8, 37	11.365	6.927	8.352	16.766	4423.385	670
10, 14, 22, 14	11.364	6.927	8.351	16.765	4423.385	495
10, 14, 22, 37	11.365	6.927	8.352	16.767	4423.385	546
26, 6, 8, 14	11.365	6.927	8.352	16.766	4423.385	565
26, 6, 8, 37	11.364	6.927	8.351	16.767	4423.385	754
26, 6, 22, 14	11.364	6.926	8.351	16.765	4423.385	516
26, 6, 22, 37	11.365	6.927	8.352	16.765	4423.385	540
26, 14, 8, 14	11.361	6.925	8.350	16.764	4423.385	430
26, 14, 8, 37	11.364	6.927	8.351	16.767	4423.385	550
26, 14, 22, 14	11.370	6.930	8.353	16.772	4423.386	428
26, 14, 22, 37	11.359	6.923	8.350	16.773	4423.386	886

The downhill simplex method gives the same optimum around 11.765, 7.191 from four different starting points for the 2-dimensional case, and 11.365, 6.927, 8.352, 16.766 from 16 different starting points for the 4-dimensional case. Depending on the function tolerance, one can get more significant digits in the optimum value. However, the number of function evaluations is increased. Considering that the original optimum is discrete, it is probably be unnecessary to increase the ftol value. With the function tolerance of 1.0e-8, three significant digits are acquired in duct sizes and function values.

The problem's characteristic length scale, λ of Equation (4.1), also affects the number of function evaluations and the iterations. The appropriate values range from 1.0 to 1.5. When the length scale is smaller, the number of function evaluations is increased to more than 1000. It is also recommended to repeat the routine at a point that is found to be an optimum until the function value is not further improved. The reason for routine iteration is to avoid trapping at a point that is not a minimum due to anomalous steps in moving the simplex point. Occasionally, after a certain number of steps, the simplex takes a series of continuous contractions and the movement of the simplex is stopped at a certain point. When the program routine is repeated at a trapped point, the simplex easily gets out of the trapped point and settles down at a global minimum. In order to be certain that the trapped point is not a local minimum, its neighborhood is exhaustively searched with a 0.0001-inch step size. No point was found surrounded by higher function values. This proves that the trapped point is not a local minimum.

4.3 Problem Domain Analysis for a VAV System

The problem domain analysis has been done for a CAV system and the result shows that it very likely has only a global minimum. Now a three-section VAV duct system is created and tested for the domain analysis of a VAV system.

4.3.1 An Example Duct System for Analysis

A hypothetical duct system shown in Figure 4.10 was carefully selected to analyze the problem domain. It is a two-dimensional three-section duct system, in which duct section # 2 has a fixed value of 0.178m (7 inch). The system parameters are summarized in Tables 4.9, 4.10 and 4.11, and they are economic, general, and sectional data respectively. In Table 4.11, the peak airflows at terminal duct sections 1 and 2 are 0.70 and 0.50 m^3/s respectively. It is defined to have variable airflows and to operate a full year of 8760 hours. The minimum fraction of full flow is set to 0.4. Figure 4.11 shows the fractional flow distributions for this hypothetical duct system. The fractions of full flow are divided into increments of 0.05. Bin 1 corresponds to zero flow, bin 9 to minimum fraction and bin 20 to the full flow.

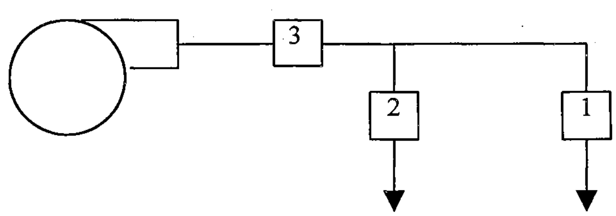


Figure 4.10 Three-Section Duct System

Table 4.9 Economic Data

Energy cost (E_p)	2.03 c/kwh
Energy demand cost	13 \$/kwh
Duct cost (S_d)	43.27 \$/m^2
System operation time (Y)	8760 h/yr
Present worth escalation factor (PWEF)	8.61

Table 4.10 General Data

Data	SI units	IP units
Air temperature (t)	22 degree C	71.6 degree F
Absolute roughness of aluminum duct (ε)	0.0003 m	0.00098 ft
Kinematic viscosity (ν)	1.54×10^{-5} m^2/s	1.66×10^{-4} ft^2/s
Air density (ρ)	1.2 kg/m^3	0.75 lb/ft^3
Motor efficiency (η_e)	0.75	0.75
Total system airflow (Q_{fan})	1.42 m^3/s	3010 cfm

Table 4.11 Sectional Data of Three-Section Duct System

Sections			Peak air flow, m^3/s	Duct length, m	Additional pressure loss, Pa	C – coefficient
Sec.	Ch1	Ch2	Airflow	Length	DP_z	
1	0	0	0.70	14.00	25.0	0.80
2	0	0	0.22	12.00	37.5	0.65
3	1	2	0.92	19.81	121.8	1.50

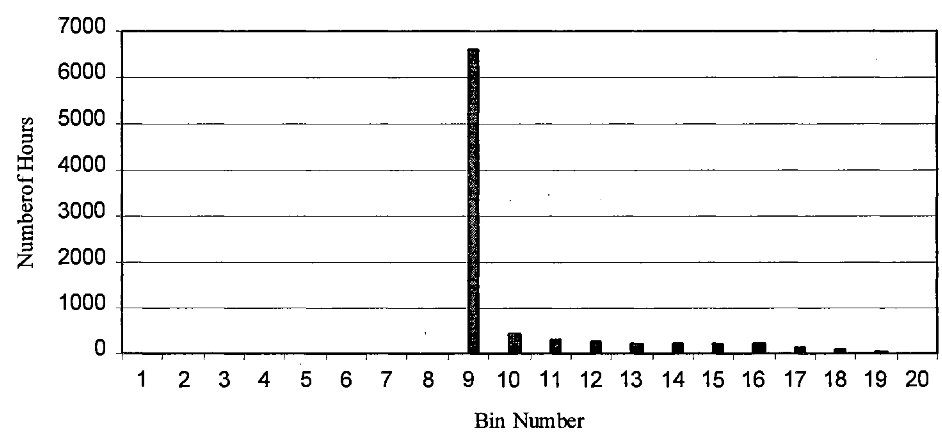


Figure 4.11 Annual Distribution of Fraction of Full Flow for the Hypothetical System

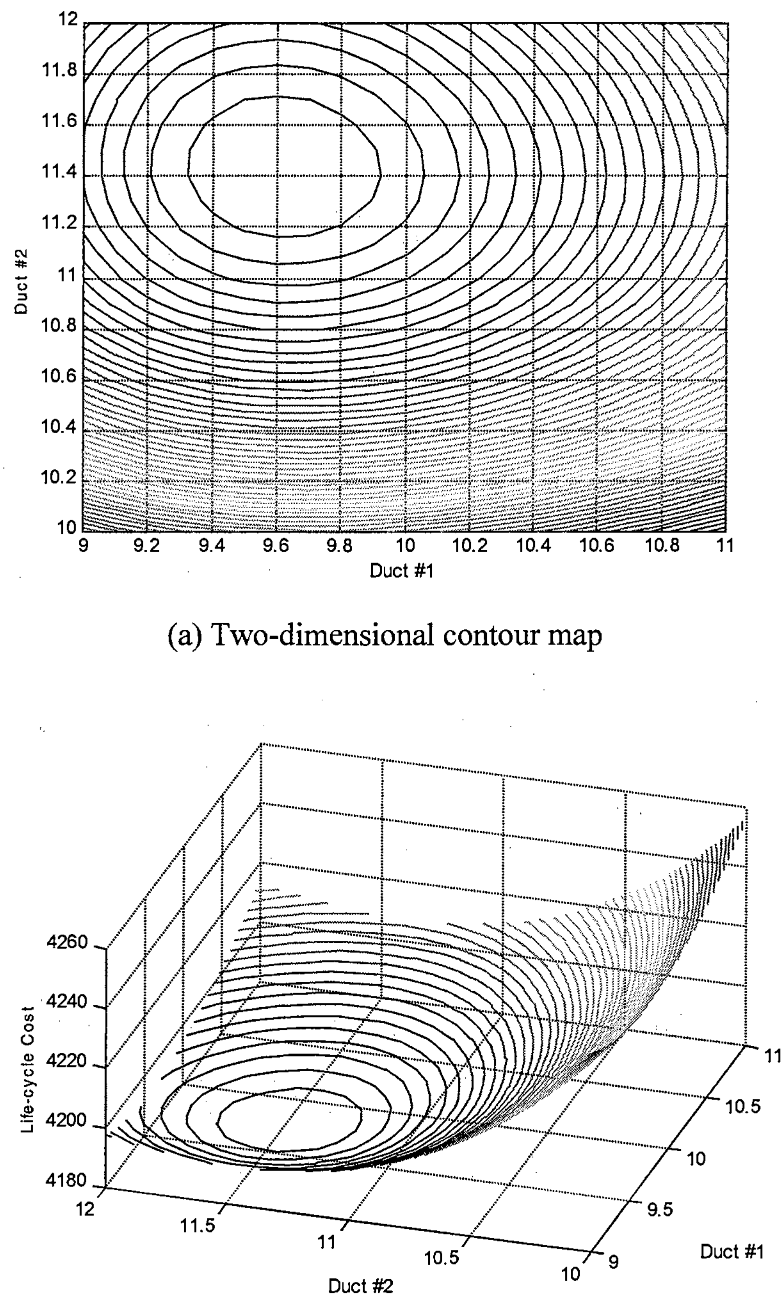
4.3.2 Computation Results and Discussion

Exhaustive search

When the exhaustive search was applied to the example system with a discrete step size of 1 inch, a design point is identified as that having neighboring points with higher functional values. This point may be called an apparent local minimum: "apparent" because, while it appears to be a local minimum, it has not yet been established whether or not it is a local minimum. Its design value is 10 and 11 inches in duct #1 and #3 respectively and system life-cycle cost is $4191.36. The problem domain is searched again with a discrete step size of 0.1 inch and a design point is identified as a local minimum. An apparent local minimum appeared at 9.6 and 11.4 inch in duct #1 and #3 respectively and system life-cycle cost is $4185.90. The step size is further subdivided into 0.01 inch and the domain has an apparent local minimum. It is 9.63 and 11.43 inches in duct #1 and #3 respectively and system life-cycle cost is $4185.88. A very likely global minimum is found at an apparent local minimum with exhaustive search.

Graphical representation

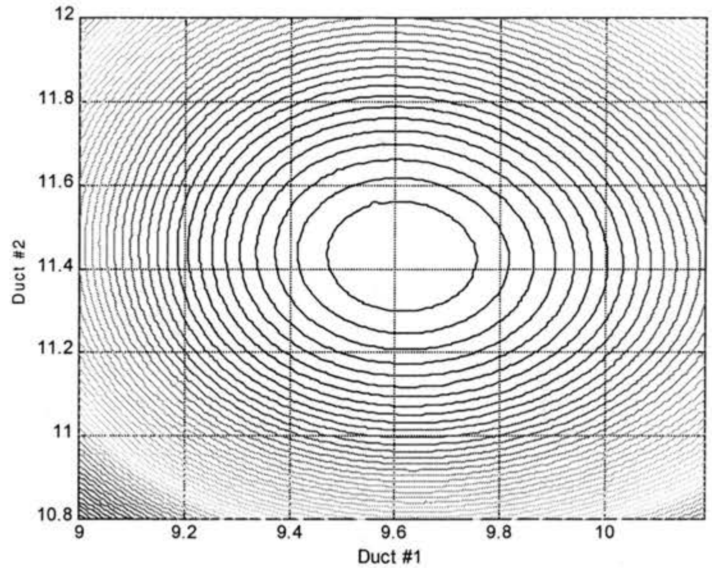
For a visual inspection of the problem domain, contour maps are drawn on two- and three- dimensional spaces for 0.1- and 0.01-inch grid searches as shown in Figure 4.12 and 4.13. It can be seen that the objective function has a typical shape of the convex function. The surface does not have a deep valley and a steep slop. The function slopes gradually down to the bottom. From the figure 4.12 and 4.13, one can easily find that the optimum solution lies around 9.6 and 11.4 inches in duct #1 and #3 respectively. The graphical representation also suggests that the problem very likely has only a global minimum.



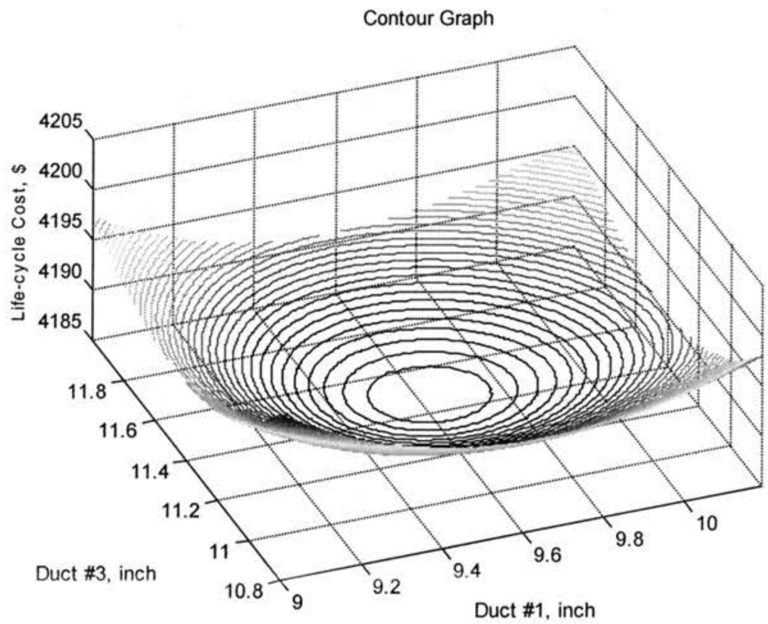
(a) Two-dimensional contour map

(b) Three-dimensional contour map

Figure 4.12 Contour Map of System Life-Cycle Cost
of the Hypothetical Duct System with 0.1 inch Exhaustive Search



(a) Two-dimensional contour map



(b) Three-dimensional contour map

Figure 4.13 Contour Map of System Life-Cycle Cost
of the Hypothetical Duct System with 0.01 inch Exhaustive Search

The Nelder and Mead Downhill simplex method

In a two-dimensional exhaustive search, it is found that the problem domain has a very likely global minimum. The downhill simplex method is now applied to the hypothetical system. The initial starting point for a search is chosen at an extreme point, such as the corner point of the bounded region. The other N points of a simplex are defined by

$$\mathbf{P}_i = \mathbf{P}_0 + 1.0\ \mathbf{e}_i,$$

where the $\mathbf{e}_i$ is an N unit vector, and 1.0 is the problem's characteristic length scale. The fractional convergence tolerance of the function value for a simplex routine is set to 1.0e-8. The function tolerance to stop routine iteration is set to 0.001. The result is shown at Table 4.12.

Table 4.12 Global Minimum Found with the Downhill Simplex Method

Starting point (duct #1, duct #3)	Duct 1 inch	Duct 3 Inch	Life-cycle cost, $	Number of function evaluation
Lower left (9, 10)	9.61	11.43	4185.883	88
Upper Right (21, 24)	9.62	11.44	4185.881	174
Upper Left (9, 24)	9.62	11.44	4185.881	184
Lower Right (21, 10)	9.61	11.43	4185.883	113

The downhill simplex method gives the same optimum around 9.61, 11.43 from four different starting points for the 2-dimensional hypothetical system. The information on the function tolerance, problem's characteristic length scale is similar to the one from the CAV system analysis.

4.4 Conclusion of the Problem Characteristics

Exhaustive search, graphical representation, analysis by graphical analogy, and the Nelder and Mead's downhill simplex method have been applied to CAV and VAV duct systems to define the characteristics of the problem domain. Consideration has been given to the verification of the local/global minimum when duct sizes are design variables.

CAV Duct System

The exhaustive search has found apparent local minima in the 2-dimensional problem domain. In order to see whether they are true local minima, the neighbors of the apparent local minima were searched with a smaller discrete step size along duct-2 lines. Lower function values were found in the neighbors and the minimum values on duct-2 lines were decreased in order toward the lowest one. In 2-dimensional duct systems, a contour map of the life-cycle cost shows that the domain has a very deep valley and a peaked hood-like convex shape. Analysis by graphical analogy was applied to the duct design problem by creating a function that has a similar shape. The created function has only one local minimum, however it also gives many apparent local minima when the problem domain is exhaustively searched. The Nelder and Mead's downhill simplex algorithm also found the same global minimum as the exhaustive search in both the 2- and 4-dimensional duct systems. Considering the results from the above four approaches, it can be concluded that the CAV duct problem very likely has only a global minimum.

VAV Duct System

The exhaustive search has been done with 1-, 0.1-, and 0.01-inch increments. It has found only one local minimum in the two-dimensional problem domain. Contour maps of system life-cycle cost shows that the domain does not have a deep valley. The function slopes gradually down to the bottom. The Nelder and Mead's downhill simplex algorithm also found a similar location of the global minimum as the exhaustive search in the two-dimensional duct systems. The test results also suggest the function tolerance of 1.0e-6 and the problem's characteristic length, λ, from 1.0 to 1.5.

Considering the results from the above three approaches, it is concluded that the VAV duct problem very likely has only a global minimum and the optimal point can be found using a direct search method. Now the duct design problem is to find a global minimum. A nonlinear, local direct search method will produce optimum duct sizes. The Nelder and Mead downhill simplex method is applied to several VAV duct systems to find optimum duct sizes and a fan. The results are compared to those from the T-method in Chapter 5.

CHAPTER 5

OVERVIEW OF PARAMETRIC STUDY

For a given building and duct topology, the main factors that influence optimization results are electricity cost, duct work cost, and the VAV system operating schedule. Since the life-cycle cost is composed of initial and operating costs, optimization is influenced by duct material chosen and cost of electricity supplied to an installation site. Also, depending on the VAV system operating schedule, including setback control or 8760 hour operation, airflow rates and operation time are different and thus influence the life-cycle cost. The influence of these factors is considered by optimizing duct systems for a given building with different duct installation costs, electricity rate structures, and operating schedules.

5.1 Duct Cost

The duct materials used for optimization are as follows:

- Aluminum duct: \$4.02 /ft^2 (\$43.27 /m^2), absolute roughness 0.0001 ft (0.00003 m) (ductwork unit price used by Tsal et al. (1988))
- Galvanized steel: \$5.16 /ft^2 (\$55.50 /m^2), absolute roughness 0.0003 ft (0.00009 m) (RS Means 2000)

5.2 Electric Energy Rates

As shown in Table 5.1, four different electric rate structures are used in this study:

- TSAL electric rate that was introduced by Tsal et al. (1988, Part II) for Seattle, Washington
- Tulsa, Oklahoma (PSC 2000)
- Minneapolis, Minnesota (NSP 2000)
- Binghamton, New York (NYSEG 2000)

TABLE 5.1 Electricity Rate Structures

Site	Customer Charge	Demand Charge		Energy Charge	
		On-peak	Off-peak	On-peak	Off-peak
Tsal et al. (1988, Part II)	-	Ed: $13/kW		Ec: $0.0203/kWh	
Oklahoma	$22.8/Mo	-	-	$0.0622/kWh, $0.0559/kWh, $0.0358/kWh (Jun-Oct)	$0.0373/kWh, $0.0303/kWh (Nov-May)
Minnesota	$21.65/Mo	$9.26/kW (Jun-Sep)	$6.61/kW (Oct-May)	$0.031/kWh (Jan-Dec)	
New York	$14.00/Mo	$11.35/kW (7am-10pm)	- (10pm-7am)	$0.08755/kWh (7am-10pm)	$0.05599/kWh (10pm-7am)

The electric rate in Tsal et al. (1988, Part II) is $0.023 /kWh for the energy charge and $13/kW for the energy demand charge without differentiating between on-peak periods and off-peak periods. In Oklahoma, the first electric rate is charged for kWh up to 150 multiplied by the current month maximum kW, the second electric rate is applied to the next 150 multiplied by the current month maximum kW, and the third electric rate is for all additional kWh used. In Minnesota, the energy charge is $0.031 /kWh all year long and energy demand charge is $9.26 /kW during June through September and $6.61

/kW during October through May. In New York, the on-peak period is 7 a.m. through 10 p.m., Monday through Friday and the off-peak period is 10 p.m. through 7 a.m., Monday through Friday, and all day Saturday, Sunday, and Holidays. The energy charge is $ 0.08755 /kWh for the on-peak period and $ 0.05599 /kWh for the off-peak period. The energy demand charge for on-peak service is $ 11.35 /kW.

5.3 Duct Design Methods

In order to investigate the saving of the VAV optimization procedure, the duct design methods implemented for VAV duct systems design are:

- Equal friction
- Static regain
- T-method (Tsal et al. 1988)
- VAV optimization procedure.

The first two methods are commonly utilized for VAV duct design. Equal friction, static regain and the T-method do not consider varying air volumes, so the peak airflow is used as the design air volume. The equal friction and static regain methods could generate many design solutions depending on pressure losses per foot of duct length and velocities for the duct attached to the fan, respectively. In this study, the friction rate or velocity was chosen to give the lowest life cycle cost for one of the candidate duct systems. This friction rate of velocity was then applied to all the candidate duct systems.

When the T-method is applied to duct sizing, the fan pressure is calculated using the following equation as given in Tsal el al. (1988, Part I).

$$\Delta P_{opt} = 0.26(\frac{z_2}{z_1}k)^{0.833} + \Delta P_x \tag{5.1}$$

where $$z_1 = Q_{fan}\frac{E_c Y + E_d}{10^3 \eta_f \eta_m} PWEF \tag{5.2}$$

$$z_2 = 0.959\pi(\frac{\rho}{g_c})^{0.2} S_d \tag{5.3}$$

As shown in the above equations, unit energy cost E_c, energy demand cost E_d, and unit duct work cost S_d are important factors that decide the fan pressure. The duct static pressure requirement at the end of the longest duct line is also considered in deciding fan pressure by adding that requirement to the equation. Now, based on the determined fan pressure, the T-method sizes ducts during the expansion procedure.

For comparison purpose, the duct systems designed with equal friction, static regain, and the T-method are evaluated under the VAV environment to seek the life-cycle cost. The calculated costs are then compared to the one from the VAV optimization procedure to investigate the importance of the varying airflows to the system design.

5.4 Example VAV Duct Systems

In order to investigate the importance of varying airflows for optimum duct design, three example duct systems are selected. The duct systems are (1) ASHRAE example, (2) a large office building in Oklahoma and (3) the same large office building in Minnesota.

5.4.1 ASHRAE Example

The ASHRAE example is a duct system given as example #3 of the 1997 ASHRAE Handbook-Fundamentals, Chapter 32 (ASHRAE 1997). It is a 19-section duct system that has 13 supply ducts (sections 7 through 19) and 6 return ducts (sections 1 through 6). This system has been taken as a typical example in many duct design studies. The ASHRAE example in its reference is assumed to be a CAV system and the peak airflow is given to every outlet and inlet. In order to supply the systems with time-varying airflows, the fraction of full flow of the large office building at Tulsa, Oklahoma was computed for a full year's operation using BLAST and was used as a baseline to create varying air volumes by multiplying constant air volumes by the fraction of full flow. A schematic diagram of the system is shown in Figure 5.1. The sectional data of the ASHRAE example are given in Table 5.2. Every duct section in this example is assumed to be a round duct.

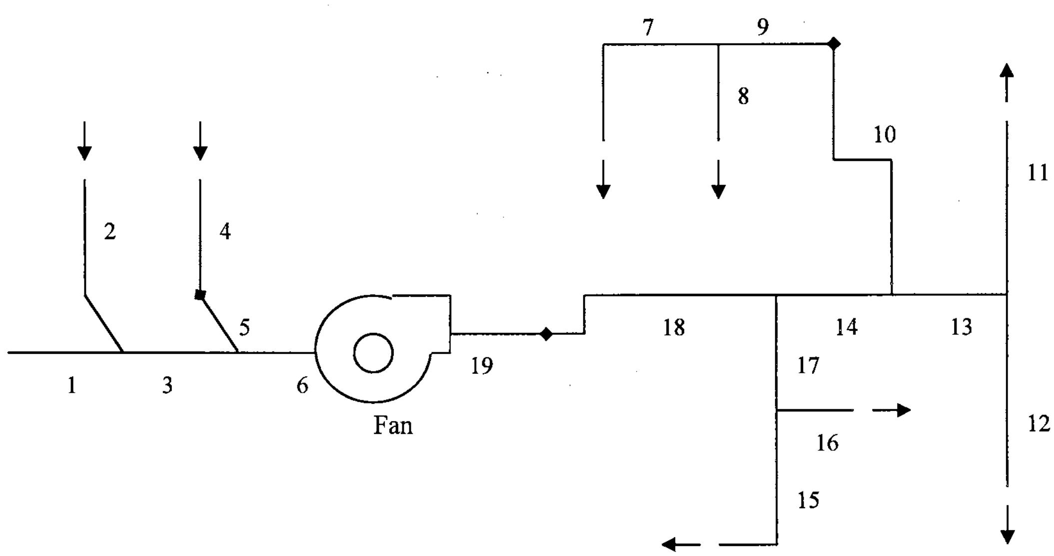


Figure 5.1 ASHRAE Example

TABLE 5.2 Sectional Data of ASHRAE Example

Sections No.	Sections Child	Peak airflow, cfm(m^3/sec)	Duct length, ft (m)	Additional Pres. loss, in.wg (Pa)	ASHRAE fitting No*
Return: 1	0, 0	1500 (0.71)	15 (4.57)	0	ED1-3, CD9-1, ED5-1
2	0, 0	500 (0.24)	60 (18.29)	0	ED1-1, CD6-1, CD3-6,CD9-1,ED5-1
3	1, 2	2000 (0.94)	20 (6.10)	0	CD9-1, ED5-2
4	0, 0	2000 (0.94)	5 (1.52)	0.1 (25)	CD9-4, ER4-3
5	4, 0	2000 (0.94)	55 (16.76)	0	CD3-17, CD9-1, ED5-2
6	3, 5	4000 (1.89)	30 (9.14)	0.22 (55)	CD9-3, CD3-9, ED7-2
Supply: 7	0, 0	600 (0.28)	14 (4.27)	0.1 (25)	CR3-3, CR9-1, SR5-13
8	0, 0	600 (0.28)	4 (1.22)	0.15 (37)	SR5-13, CR9-4
9	7, 8	1200 (0.57)	25 (7.62)	0	SR3-1
10	9, 0	1200 (0.57)	45 (13.72)	0	CR9-1, CR3-10, CR3-6,SR5-1
11	0, 0	1000 (0.47)	10 (3.05)	0	CR9-1, SR2-1, SR5-14
12	0, 0	1000 (0.47)	22 (6.71)	0	CR9-1, SR2-5, SR5-14
13	11, 12	2000 (0.94)	35 (10.67)	0	CR9-1, SR5-1
14	10, 13	3200 (1.51)	15 (4.57)	0	CR9-1, SR5-13
15	0, 0	400 (0.19)	40 (12.19)	0	CR3-1, SR2-6, CR9-1, SR5-1
16	0, 0	400 (0.19)	20 (6.10)	0	SR2-3, CR6-1, CR9-1, SR5-1
17	15, 16	800 (0.38)	22 (6.71)	0	CR9-1, SR5-13
18	14, 17	4000 (1.89)	23 (7.01)	0.04 (10)	CR6-4, SR4-1, CR3-17, CR9-6
Root: 19	18, 0	4000 (1.89)	12 (3.66)	0.05 (13)	SR7-17, CR9-4

* ASHRAE Duct Fitting Database (ASHRAE 1993)

5.4.2 Large Office Building in Oklahoma and Minnesota

The large office building in Oklahoma is a 34-section supply duct system of part of a single floor of the BOK building in Tulsa, Oklahoma. The building is a 52-story multipurpose office building located in Tulsa's downtown area. It measures about 160 feet (48.77 m) by 160 feet (48.77 m) and is about 1360 feet (414.53 m) in height. The building is oriented in a 20° northeast north direction and is not shaded by any other structures. It has a large area of glazing, about 65% of the exterior envelope. The building is described by Feng (1999) in greater detail. The example duct system from this building serves only part of a single floor – zones 18 through 22 as shown in Figure 5.2, approximately 13,200 ft^2 (1226 m^2) of floor space. A schematic diagram with section numbers is shown in Figure 5.3. The sectional data of the VAV duct system are given in Table 5.3.

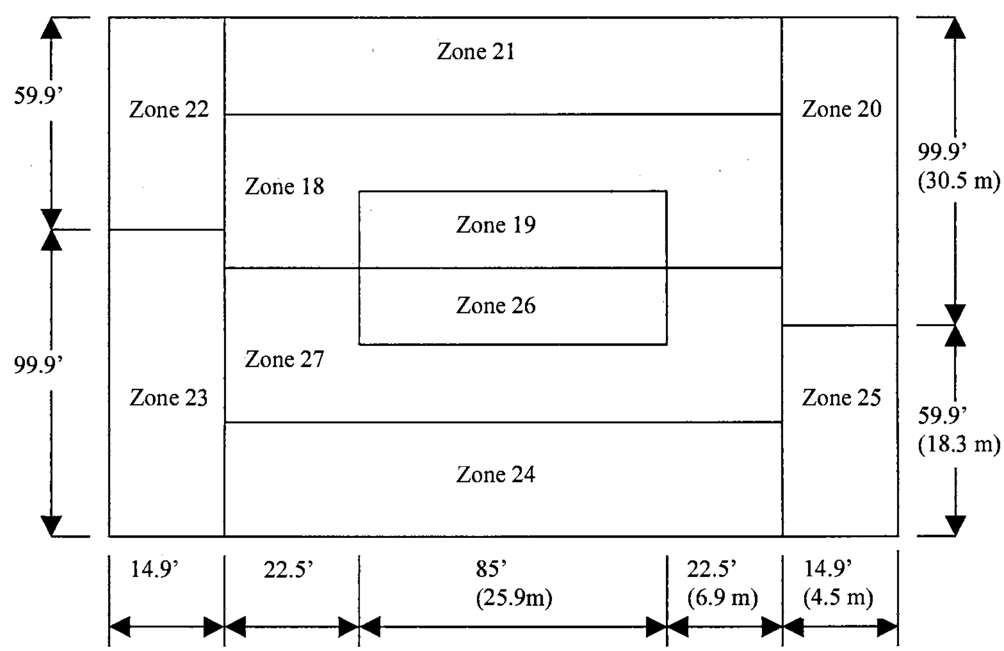


Figure 5.2 Zone Layout for Floor 8-24 of the Large Office Building

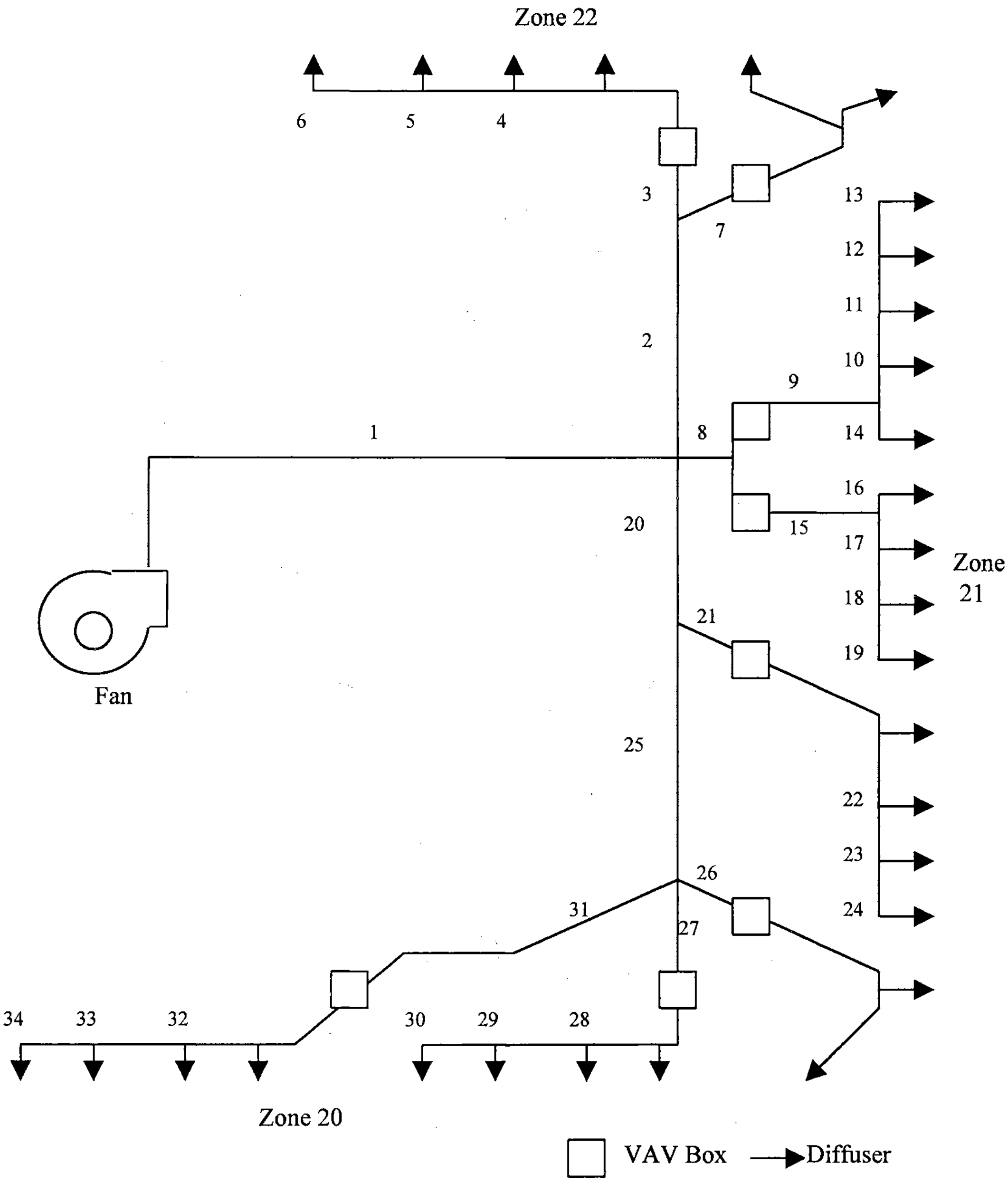


Figure 5.3 Schematic Diagram of the Duct System
of the Large Office Building

TABLE 5.3 Sectional Data of BOK at OK and BOK at MN

Sections		Peak air flow, cfm (m^3/sec)		Duct length, ft (m)	DPz, in.wg (Pa)	ASHRAE fitting No.*
No	Child	Tulsa, OK	Minneapolis MN			
1	2, 8, 19	8678 (4.095)	7584 (3.579)	50 (15.24)	0 (0)	SR7-17, SD4-2, CD3-9,
2	3, 7	2699 (1.274)	2481 (1.171)	35 (10.67)	0 (0)	SD5-26(b1),
3	4	1800 (0.849)	1654 (0.780)	25 (7.62)	0.2 (50)	SD5-2(S), CD3-8, CD3-8, CD9-1, CD3-9, SD4-1
4	5	1350 (0.637)	1240 (0.585)	10 (3.05)	0.2 (50)	SD4-1
5	6	900 (0.425)	827 (0.390)	10 (3.05)	0.2 (50)	SD4-1
6	-	450 (0.212)	413 (0.195)	10 (3.05)	0.2 (50)	-
7	-	900 (0.425)	827 (0.390)	20 (6.10)	0.4 (100)	SD5-2(b), CD9-1, CD3-14
8	9, 14	1637 (0.773)	1254 (0.592)	5 (1.52)	0 (0)	SD5-26(s)
9	10, 13	910 (0.429)	697 (0.329)	15 (4.57)	0 (0)	SD5-19, CD9-1, CD3-9
10	11	728 (0.343)	558 (0.263)	10 (3.05)	0.2 (50)	SD5-19(b1), SD4-1
11	12	546 (0.258)	418 (0.197)	10 (3.05)	0.2 (50)	SD4-1
12	13	364 (0.172)	279 (0.132)	10 (3.05)	0.2 (50)	SD4-1
13	-	182 (0.086)	139 (0.066)	10 (3.05)	0.2 (50)	-
14	-	182 (0.086)	139 (0.066)	10 (3.05)	0.2 (50)	SD5-19(b2)
15	16, 17	728 (0.343)	558 (0.263)	15 (4.57)	0 (0)	SD5-19, CD9-1,CD3-9
16	-	182 (0.086)	139 (0.066)	10 (3.05)	0.2 (50)	SD5-19(b1)
17	18	546 (0.258)	418 (0.197)	10 (3.05)	0.2 (50)	SD5-19(b1), SD4-1
18	19	364 (0.172)	279 (0.132)	10 (3.05)	0.2 (50)	SD4-1
19	-	182 (0.086)	139 (0.066)	10 (3.05)	0.2 (50)	-
20	21, 25	4341 (2.049)	3849 (1.816)	30 (9.14)	0 (0)	SD5-26(b2)
21	22	728 (0.343)	558 (0.263)	20 (6.10)	0.2 (50)	SD5-2(b), CD9-1, CD3-14, SD4-1
22	23	546 (0.258)	418 (0.197)	10 (3.05)	0.2 (50)	SD4-1
23	24	364 (0.172)	279 (0.132)	10 (3.05)	0.2 (50)	SD4-1
24	-	182 (0.086)	139 (0.066)	10 (3.05)	0.2 (50)	-
25	26,27, 31	3614 (1.705)	3291 (1.553)	40 (12.19)	0 (0)	SD5-2(s)
26	-	723 (0.341)	658 (0.311)	20 (6.10)	0.4 (100)	SD5-23(b1), CD9-1, CD3-14,
27	28	1445 (0.682)	1317 (0.621)	25 (7.62)	0.2 (50)	SD5-23(S), CD9-1, CD3-9
28	29	1084 (0.512)	987 (0.466)	10 (3.05)	0.2 (50)	SD4-1
29	30	723 (0.341)	658 (0.311)	10 (3.05)	0.2 (50)	SD4-1
30		361 (0.171)	329 (0.155)	10 (3.05)	0.2 (50)	-
31	32	1445 (0.682)	1317 (0.621)	60 (18.29)	0.2 (50)	SD5-23(b2), CD9-1, CD3-14, CD3-14, CD3-14,
32	33	1084 (0.512)	987 (0.466)	10 (3.05)	0.2 (50)	SD4-1
33	34	723 (0.341)	658 (0.311)	10 (3.05)	0.2 (50)	SD4-1
34	-	361 (0.171)	329 (0.155)	10 (3.05)	0.2 (50)	-

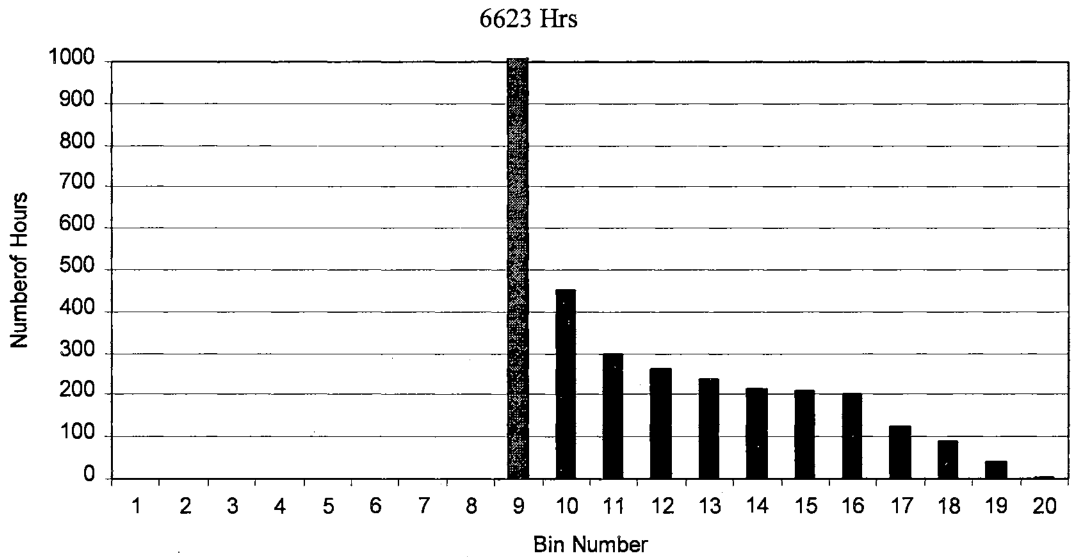
* From ASHRAE Duct Fitting Database (ASHRAE 1993)

The air-handling unit is located at Zone 19 and air is distributed to the perimeter zones 20, 21, and 22. Zone 20 on the east side has two terminal boxes and eight exits, zone 21 on the north side has four terminal boxes and 15 exits, and zone 22 has two terminal boxes and six exits. Every duct section is assumed to be a round duct.

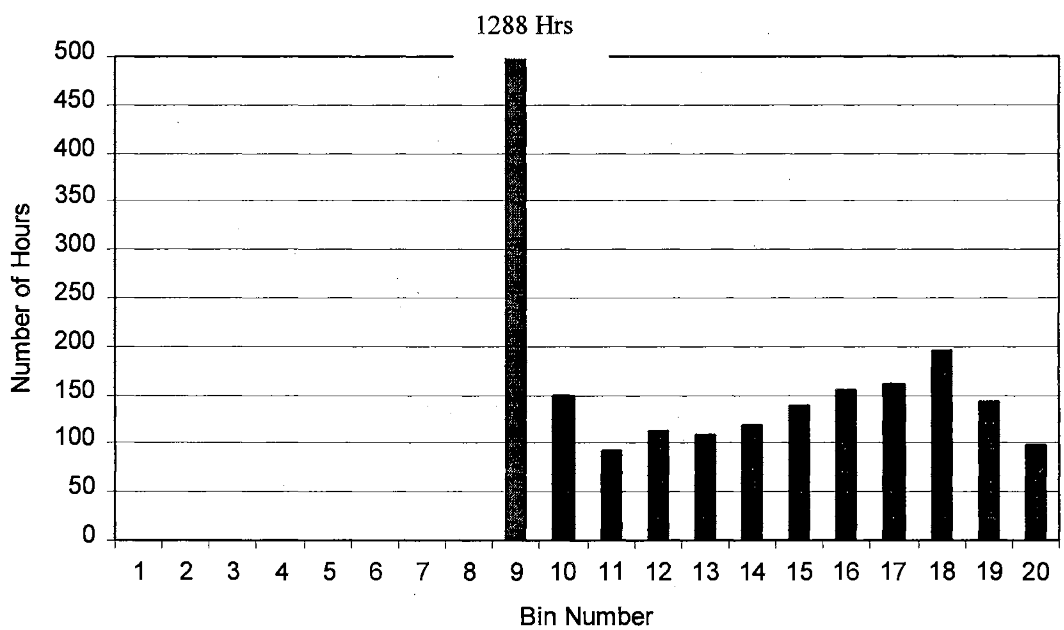
The HVAC system for this floor was originally a three-deck multizone system that featured hot and cold decks and separate mixing dampers for each zone. In this study it is assumed to have a VAV system, in which air flows through a main cooling coil at a design cold-deck temperature of 55 °F (12.78 °C). The air is then sent to each zone by modulating the amount of air with a VAV box. If the zone requires heating, the air is heated by use of auxiliary reheat. The system has a VAV control schedule that specifies the fraction of peak cooling or heating at a specific zone temperature for a VAV system. For purpose of this study the occupancy, lighting and equipment profiles for the building are assumed to have a weekday schedule of being fully on from 8 a.m. to 5 p.m., and the building is assumed to have no occupancy, lighting or equipment heat gain for nights, weekends and holidays. The system is simulated based on two different operating schedules: (1) 8760-hour schedule (always on), (2) setback controlled schedule. The 8760-hours schedule has VAV control for 24 hours a day all year long, while the setback controlled schedule has VAV control from 7a.m. to 5p.m., Monday through Friday and setback control from 5p.m. to 7a.m., Monday through Friday, all day Saturday, Sunday, and Holidays (See Appendix C for the BLAST input files). All the VAV boxes have minimum fractions of 0.4. It could be set to a lower minimum fraction for the VAV boxes. However, in this study when the duct system of the large office building was used with the BISW type of fan as introduced in Chapter 3, a lower minimum fraction than 0.4

caused fan operation in the surge region. Hence the minimum fraction was set to 0.4. Airflow data summed for all zones are represented using a histogram, which is a frequency distribution with the fraction of full flow as the abscissa and the number of hours at each increment as the ordinate in Figure 5.4. Bin 1 corresponds to 0~5% of full flow and bin 2 corresponds to 5~10% of full flow, etc. In Figure 5.4(a), for the large office building in Oklahoma bin 9 that corresponds to minimum fraction of full flow has 6,623 operating hours for the 8,760-hour schedule. In Figure 5.4(b), the setback controlled schedule results in 2,763 hours operation, of which 1,288 hours are at the of minimum fraction.

The large office building in Minnesota shares the same layout and sectional information with the one at Oklahoma. The building is simulated at Minneapolis, Minnesota, in order to investigate the effect of climate on optimum duct design with different weather conditions. All duct sections are again assumed to be round ducts. The histogram of airflow data of the building in Minnesota is shown at Figure 5.5. In Figure 5.5(a), bin 9 that corresponds to minimum fraction of full flow has 6788 operating hours for the 8,760-hour schedule. In Figure 5.5(b), the setback controlled schedule results in 4,269 hours operation and 3,177 hours of minimum fraction of full flow.

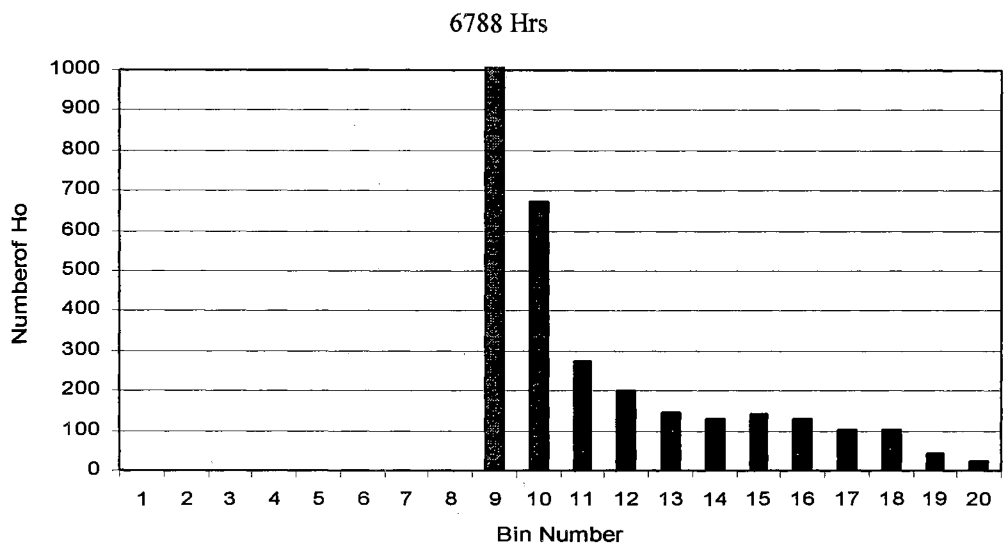


(a) On Schedule: 8760 hrs airflow

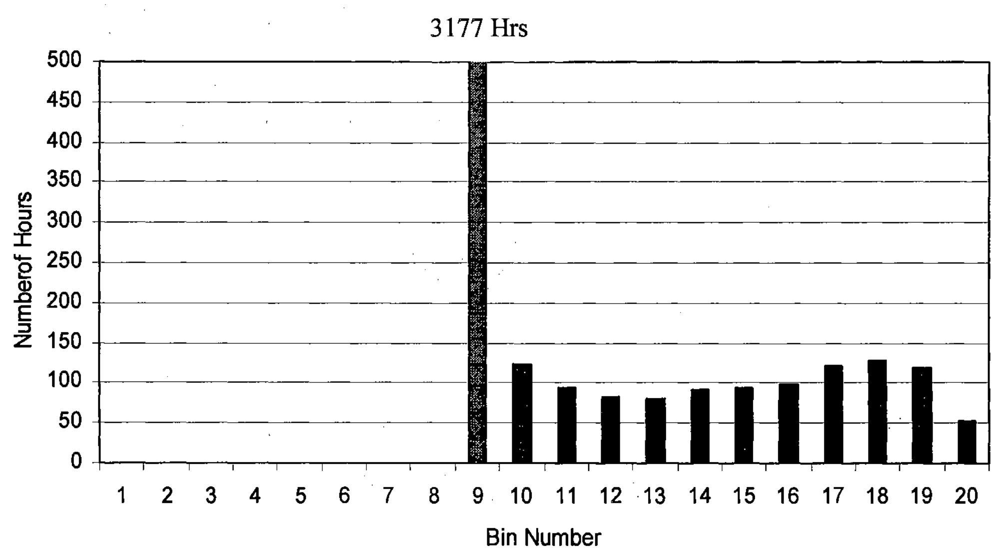


(b) Setback Controlled Schedule: 2763 hrs airflow

Figure 5.4. Annual Distribution of Fraction of Full Flow
of the large office building in Oklahoma



(a) On Schedule: 8760 hrs airflow



(b) Setback Controlled Schedule: 4269 hrs airflow

Figure 5.5. Annual Distribution of Fraction of Full Flow
of the Large Office Building in Minnesota

CHAPTER 6

PARAMETRIC STUDY

The three duct systems are optimized using four design methods for four different electric rates and two different operating schedules with aluminum ducts and galvanized steel ducts. Duct sizes from the equal friction method are obtained with the pressure loss per 100 ft that gives the lowest life cycle cost with the TSAL electric rate: 0.15 in. wg/100ft (1.22 Pa/m) for the ASHRAE example, 0.15 in. wg/100ft (1.22 Pa/m) for the office building in Oklahoma, and 0.1 in. wg/100ft (0.82 Pa/m) for the office building in Minnesota. Duct sizes from the static regain method are obtained with the velocity of the duct attached to the fan that gives the lowest life cycle cost with the TSAL electric rate: 2600 fpm (13 m/s) for the ASHRAE example, 2800 fpm (14 m/s) for the office building in Oklahoma, and 2700 fpm (13.5 m/s) for the office building in Minnesota. The duct systems designed with the equal friction and static regain methods are then simulated with different electric rates in order to see the economic effect under VAV operation. In the T-method, different electric rates establish different optimum duct sizes since the optimum fan pressure is changed. The VAV optimization procedure established optimum duct sizes with varying airflows through the selection of an efficient fan, finding duct fitting coefficients, calculating system pressure loss, and evaluating the life cycle cost. Typically, 20,000 ~ 25,000 objective function evaluations are utilized.

For duct size rounding or discrete programming, a 1-inch increment is used for duct sizes up to 20 inches and, a 2-inch increment is used for all others. The results are organized as follows:

- Comparison of Duct Design methods: (1) Life cycle cost Analysis and (2) Duct surface area.
- Influences on the Optimal Design: (1) Effect of electric rate on optimal design, (2) Effect of ductwork unit cost on optimal design, (3) Effect of topology on optimal design, (4) Effect of airflow schedules, (5) Unconstrained optimization results, and (6) Optimization Domain.

6.1 Comparison of Duct Design Methods

Three duct systems designed with four different design methods were evaluated under VAV operation for cost comparison purposes. The evaluation generated the initial, operating, and life cycle costs and they were compared to investigate the savings of the VAV optimization procedure. Furthermore, duct designs were compared for four different electric rate structures, although only the VAV optimization procedure and the T-method take the electricity rate into account in the duct design.

6.1.1 Life Cycle Cost Analysis

Table 6.1 shows the life cycle cost when the three duct systems are designed under four different electricity rate structures using aluminum ducts. The percent saved by the VAV optimization procedure compared to the other design methods is shown in parenthesis. When the life cycle cost of the VAV optimization procedure is compared with the other duct design methods as shown in Table 4, the VAV optimization procedure shows 6 ~ 19% savings for the equal friction method, 2 ~ 13% savings over the static regain method, and 1 ~ 4% savings over the T-method.

The VAV optimization procedure gives greater life cycle cost savings with lower electricity rates. For example, the ASHRAE example shows that the savings with the TSAL electric rate compared to the equal friction, static regain, and T-method are 14, 8, and 3%; and the savings with the NY electric rate are 6, 4, and 2%, respectively. The better savings with lower electric rates are explained below in the section, *Optimization Domain*. While the VAV optimization procedure gives larger savings for the large office building for all electricity rates, the savings are similarly lower for higher electricity rates.

TABLE 6.1
Life Cycle Cost and Savings of the VAV optimization procedure
(Aluminum Ducts)

Unit: $

Duct System	Duct Design Method	TSAL Electric Rate	OK Electric Rate	MN Electric Rate	NY Electric Rate
ASHRAE Example	Equal Friction	13263 (14.4%)	17293 (9.4%)	17664 (9.2%)	21424 (6.1%)
	ASHRAE Ans.	12125 (6.4%)	16231 (3.5%)	16640 (3.6%)	20560 (2.2%)
	Static Regain	12270 (7.5%)	16407 (4.5%)	16408 (2.2%)	20837 (3.5%)
	T-method	11665 (2.7%)	15960 (1.8%)	16377 (2.0%)	20547 (2.1%)
	VAV Opt	11348	15668	16044	20115
Building in OK	Equal Friction	14470 (17.3%)	19223 (12.2%)	20010 (11.5%)	25718 (7.8%)
	Static Regain	13303 (10.0%)	18165 (7.0%)	19025 (6.9%)	25060 (5.3%)
	T-method	12227 (2.1%)	17093 (1.2%)	17972 (1.5%)	24042 (1.3%)
	VAV Opt	11967	16887	17707	23720
Building in MN	Equal Friction	13419 (18.7%)	17777 (12.2%)	18536 (11.7%)	23246 (8.7%)
	Static Regain	12571 (13.3%)	16989 (8.1%)	17790 (8.0%)	22673 (6.4%)
	T-method	11370 (4.1 %)	15806 (1.2%)	16609 (1.4%)	21617 (1.8%)
	VAV Opt	10905	15617	16369	21231

* All costs are listed in Tables B4 to B7 of Appendix.

The VAV optimization procedure gives optimum duct sizes coincident or near to the minimum duct sizes for the large office building. This leads to the question as to how good or bad a system designed with all duct sizes at the minimum duct size constraint would be. The life cycle cost of the minimum duct size system compared to the VAV optimization-designed duct systems are 0.3 ~ 1.8% higher for the large office building in Oklahoma and 3.4 ~ 7.8% higher for the ASHRAE example. The life cycle cost of the minimum duct size system of the ASHRAE example that has less constrained optimal duct sizes is much higher than that of the optimal duct system.

Table 6.2 the life cycle cost and savings when galvanized steel, which has a slightly higher unit cost than the aluminum ducts, is used for ducts. The VAV optimization procedure yields the life cycle cost savings ranging 6.8 ~ 12.7% over the equal friction method, 4.8 ~ 7.7% over the static regain method, and 0.4 ~ 0.8% over the T-method. This is a slight decrease in savings as compared to aluminum ducts. The aluminum duct system designed with the VAV optimization procedure was nearly completely constrained by the lower limits on duct size, so that little reduction in duct size was possible with the more expensive galvanized steel ducts. Therefore, for this building, the VAV optimization procedure's performance, relative to the T-method decreases as duct unit cost increases.

TABLE 6.2
Life Cycle Cost and Savings of the VAV optimization procedure
(Galvanized Steel Ducts)

Unit: $

Duct System	Duct Design Method	OK Electric Rate	MN Electric Rate	NY Electric Rate
Building in OK	Equal Friction	21893 (12.7%)	22720 (11.0%)	28606 (6.8%)
	Static Regain	20712 (7.7%)	21641 (6.6%)	27996 (4.8%)
	T-method	19276 (0.8%)	20292 (0.4%)	26831(0.7%)
	VAV Opt	19123	20213	26654

* Optimum duct sizes and economic costs are shown in Tables B8 and B9 of Appendix.

6.1.2 Duct Surface Area

Table 6.3 gives a comparison of the duct surface area of three duct systems using different duct design methods. The VAV optimization procedure saved duct surface 23 ~ 31 % over the equal friction method, 13 ~ 22% over the static regain method, and 4 ~ 7% over the T-method.

A characteristic of the problem is that the duct system has an optimum solution near to the minimum sizes, depending on the electric rate. For the large office building in Oklahoma and Minnesota, the duct system is almost completely constrained to the lower limit duct sizes with the TSAL electric rate. However, when the higher electric rate is used, optimum duct sizes for most duct sections are higher than the lower bound. For the ASHRAE example, some optimum duct sizes are coincident and others are near to the lower limits with the TSAL electric rate. This differs from the nearly completely constrained large office building. The reason for this different optimal solution location depending on the problem is discussed later in the section, *Effect of Topology on Optimal Design.*

TABLE 6.3
Comparison of Duct Surfaces Using Different Duct Design Methods
(Aluminum ducts)

Duct System	Duct Surface, ft^2 (m^2)			
	Equal Friction	Static Regain	T-method	VAV Opt. Proc
ASHRAE Ex with TSAL E. Rate	2271 (211)	1996 (185)	1824 (169)	1740 (162)
Building at OK with OK E. Rate	2225 (207)	1897 (176)	1645 (153)	1585 (147)
Building at MN with MN E. Rate	2100 (195)	1868 (174)	1564 (145)	1458 (135)

* Optimum duct sizes are shown in Tables B1 to B3 of Appendix.

6.2 Influences on the Optimal Design

There are several factors that influence optimal duct design: electricity rate, ductwork unit cost, duct topology, airflow schedule, and constraints (air velocity/ duct diameter). The influences of these factors are investigated and discussed below.

6.2.1 Effect of Electricity Rate on Optimal Design

Table 6.4 shows the optimal duct surface area found with different electric rate structures using the VAV optimization procedure. The electricity cost of New York (\$0.08755 /kWh) is about 4 times higher than that of the TSAL electric rate (\$0.0203 /kWh), while having a similar demand charge. It might be expected that the higher electricity rates would cause the duct system diameters in New York to significantly increase in order to lower the operating cost. But, in fact, the duct surface with the NY electric rate is only 4.2 ~ 6.7% higher than that with the TSAL electric rate as shown in 6.4.

The increase of operating cost due to higher electricity rates should be offset by a optimal design that has larger duct sizes and hence larger initial cost, but also lower system pressure drop and lower operating cost. But, in fact, the increased duct diameters make a small impact on the average total system pressure drop because of the requirement to maintain a fixed static pressure at the end of the longest duct line. This static pressure requirement limits the potential reduction of the operating cost. For example, when the optimal duct diameters of the large office building in Oklahoma with the NY electric rate is doubled (duct surface area changed from 1614 ft^2 (150 m^2) to 3228 ft^2 (300 m^2)), it is expected that the system pressure drop should be reduced by a factor of

2^5 or the average pressure drop should only be about 3% of the original system. However, the average total pressure drop, **including the 1.5 in. static pressure requirement**, is only reduced by 12%. The average system pressure drop for 8760 hrs was 1.721 in. wg without doubled duct sizes and 1.514 in. wg with doubled duct sizes. The system pressure drop at minimum airflow is 1.646 in. wg without doubled duct sizes and 1.507 in. wg with doubled duct sizes. At full flow, the system pressure drop is 2.787 in. wg without doubled duct sizes and 1.533 in. wg with doubled duct sizes. At minimum airflow, the static pressure requirement at the end of the longest duct line dominates system pressure loss since the pressure losses in the ducts are very low. Considering the VAV system is operated much of time at lower airflows, the system pressure losses for a full year's operation do not change greatly with the change in duct sizes. Consequently, the change of operating costs becomes small, and does not force a significant change in duct sizes.

TABLE 6.4
Comparison of Duct Surfaces with Different Electric Rate Structures using the VAV Optimization Procedure (Aluminum ducts)

Duct System	Duct Surface, ft^2 (m^2)				Surface Increase % (TSAL to NY)
	TSAL Electric Rate	OK Electric Rate	MN Electric Rate	NY Electric Rate	
ASHRAE Ex	1740 (162)	1768 (164)	1773 (165)	1856 (172)	6.7
Building in OK	1546 (144)	1585 (147)	1551 (144)	1611 (150)	4.2
Building in MN	1410 (131)	1496 (139)	1458 (136)	1521 (141)	7.9

* Optimum duct sizes are shown in Tables B1 to B3 of Appendix.

6.2.2 Effect of Ductwork Unit Cost on Optimal Design

Table 6.5 shows the optimal duct surface area and operating cost found with the VAV optimization for the large office building in Oklahoma with the OK electric rate. The ductwork costs are \$4.02 /ft^2 (\$43.27 /m^2) for aluminum ducts, \$5.16 /ft^2 (\$55.50 /m^2) for galvanized steel ducts, and to investigate sensitivity a "double" ductwork unit cost of \$10.31 /ft^2 (\$111.00 /m^2) was set. It might be expected that the higher ductwork unit cost would cause the ducts to be sized smaller, which in turn would give a higher operating cost but a lower initial cost. However, the optimal duct surface area with a double ductwork unit cost is reduced only a little with a small increase of the operating cost as shown Table 6.5.

The small reduction of the optimal duct surface area results from the velocity/minimum duct size constraint. The optimization with a double ductwork unit cost could not make a further reduction in many duct sections since the diameters of the original duct system are coincident and near to the lower limits, which are constrained by the upper velocity limitation (See Table B8 in Appendix for the change of individual duct sizes of galvanized steel ducts).

TABLE 6.5.
Optimization for Different Ductwork Unit Costs
(Galvanized Steel Ducts)

Ductwork Unit Cost	Duct Surface, ft^2 (m^2)	Initial Cost, \$	Opr. Cost, \$	L.C. Cost, \$
Aluminum Ducts, \$4.02 /ft^2 (\$43.27 /m^2)	1585 (147.3)	8947	7940	16887
Galvanized Steel Ducts, \$5.16 /ft^2 (\$55.50 /m^2)	1543 (143.4)	10532	8591	19123
Doubled cost, \$10.31 /ft^2 (\$111.00 /m^2)	1530 (142.2)	18355	8674	27029

6.2.3 Effect of Topology on Optimal Design

In an earlier section, it was noted that the large office building example was almost completely constrained to the lower limit duct sizes with the TSAL electric rate while the ASHRAE example is constrained to the lower limit only for a few duct sections. They had the same electric rate, ductwork cost, and similar airflow distributions but the result was different. A possible explanation is the difference in duct topology. In order to investigate this difference, the large office building in Oklahoma with the TSAL electric rate is optimized again after increasing all duct lengths by 20% and 40%. The result was that the enlarged duct system moved duct sections off of the constraint. The original duct system had 5 of 34 duct sections not on the minimum size constraint. When the duct lengths were increased by 20%, 9 of 34 duct sections were not on the constraint. When the duct lengths were increased by 40 %, two duct diameters increased: one duct section was moved off the constraint and another duct section increased duct diameter by 2 in. This indicates that the VAV optimization procedure finds optimal solution near the lower limit duct sizes with the low electric rate, but the number of duct sections at the constraint depends on partly the duct topology.

6.2.4 Effect of Airflow Schedules

The climatic conditions and the system operation schedule both affect the hourly distribution of airflow rates. These have a relatively minor effect on the optimal duct sizes.

Climate condition: The large office building in Tulsa, Oklahoma was optimized under two different weather conditions: (1) Oklahoma and (2) Minnesota. They create two different airflow data sets, including two different system capacities and peak airflow rates, for the same duct topology. For the four different electric rates, the life cycle cost of the large office building duct system in Minnesota is 8 ~ 10% lower than that of the building in Oklahoma. The duct cost of the building in Minnesota is 6 ~ 10% lower than that of the building in Oklahoma. As expected, the duct system in a cold climate has a smaller duct system with a saving in the life-cycle cost.

System operation schedule: Table 6.6 shows the computation results for two different system operation schedules using the VAV optimization procedure. The setback controlled schedule results in 2763 hours operation for the large office building in Oklahoma and 4269 hours operation for the large office building in Minnesota. Comparing optimal duct areas, it is expected that the duct system with the setback controlled operation should have smaller ducts compared to the one with the 8760-hour operation as with a smaller number of operating hours the effect of the operating cost on the life cycle cost should be less. However, as shown in Table 6.6, the system with the setback controlled operation has 1.5 ~ 1.8% larger optimal duct surface area. The setback controlled operation requires a larger peak airflow than the 8760-hour operation due to morning start-up loads. The larger airflow results in higher system pressure drop and higher operating cost. Hence, the increase of operating cost is offset by an optimal design that has slightly larger duct sizes.

TABLE 6.6
Optimization for Different System Operation Schedules

Duct system	Operation Schedule	Duct Surface, ft^2 (m^2)	Initial Cost, \$	Opr. Cost, \$	L.C. Cost, \$
Building in OK with OK E Rate	8760-hr	1585 (147)	8947	7940	16887
	Setback control	1614 (150)	9063	5845	14908
Building in MN with MN E Rate	8760-hr	1458 (136)	8282	8087	16369
	Setback control	1480 (138)	8371	7182	15554

* Optimum duct sizes and economic costs for setback controlled schedule are shown in Tables B10 to B12 of Appendix.

6.2.5 Unconstrained Optimization Results

The comparison of duct design methods presented above showed that the VAV optimization procedure did not give significantly better results than the T-method. Further investigation of the effects of electricity rates, ductwork unit costs, topology and airflow schedules led to the observation that with the test buildings and electricity rates used;

- The problem tends to be constrained by the maximum velocity/minimum duct diameter.
- The minimum static pressure requirements lead to a rapidly diminishing point of return: operating costs can only be reduced up to a point by increasing the duct size.

In order to confirm these observations, a rather artificial comparison is performed. Table 6.7 shows an unconstrained optimization of the large office building duct system in Oklahoma with the NY electric rate and zero in. wg static pressure requirement.

Table 6.7
Unconstrained Optimization of the Large Office Building in Oklahoma
with No Velocity Limitation and Zero Static Pressure Requirement (Aluminum ducts)

Electric Rate		Duct Surface, ft^2 (m^2)	Init. Cost, $	Opr. Cost, $	L.C. Cost, $	Saving %
TSAL	VAV Opt. Proc.	1643 (152.6)	9179	2860	12039	-
	T-method	2127 (197.6)	11126	2341	13467	10.6
NY	VAV Opt. Proc.	1958 (181.9)	10447	11919	22366	-
	T-method	2775 (252.5)	13499	11019	24518	8.8

The life cycle cost savings of the VAV optimization procedure increased from 2.1% to 10.6% for the TSAL electric rate and 1.3% to 8.8% for the NY electric rate. The VAV optimization procedure gives much better savings with a lower electric rate, no size constraints and no static pressure requirement. This provides some confirmation for the above observations. The static pressure requirement and velocity constraints prevent the VAV optimization procedure from finding significantly better designs.

6.2.6 Optimization Domain.

Over the course of this investigation, it has been observed that the life cycle cost does not seem to be as sensitive to the duct design as originally expected. Although the optimization domain is not relatively flat when viewed as a function of individual duct sizes, there is a sense in which, if the individual ducts are correctly sized relative to one another, the domain is relatively "flat" (i.e. the life cycle cost is relatively insensitive to the total duct surface area). To help explain this, consider the following. If the T-method is utilized to size duct systems for the large office building in Oklahoma with no velocity limitation and no static pressure requirements, but with a range of electricity rates, a corresponding range of duct systems will result. The life cycle cost for these system are calculated, using a fixed electricity rate.

Figure 6.1 is a representation of the optimization domain for the large office building in Oklahoma with the NY electric rate. Economic costs are plotted in terms of total duct surface area. Each point represents a duct system that has been optimized with the T-method for an electric rate that is higher or lower than the actual NY electric rate. However, the operating costs are calculated with the actual NY electric rate. From the figure, the life cycle cost has a gently increasing slope, demonstrating that the life cycle cost is relatively insensitive to the total duct surface area. If the static pressure requirement is included in the duct design, the curve will be much flatter and the life cycle cost will be more insensitive to the total duct surface area. The life cycle cost savings of the VAV optimization procedure relative to the T-method will be further lowered. When the initial and operating costs' curves are compared, the initial cost curve is steeper than the operating cost curve.

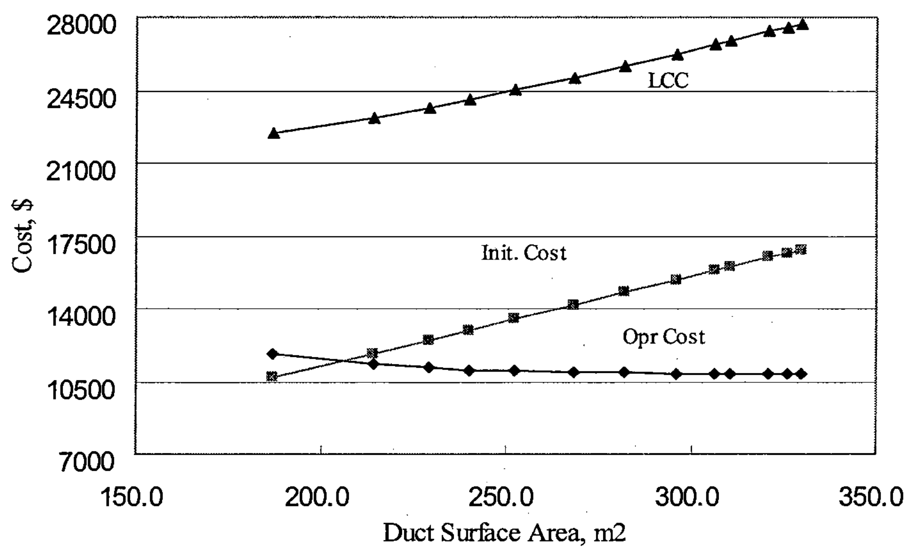


Figure 6.1 Optimization Domain of Duct Systems
of the Large Office Building in Oklahoma with the NY Electric Rate

Figure 6.2 is the plot of economic costs in terms of the electric rate multiplier. Both the kWh charge and the demand charge for New York were multiplied by the electric rate multiplier. Again, at each point the operating cost was calculated with the actual electric rate. When finding a design solution, the T-method uses the peak airflow and the VAV optimization uses a range of airflows, but they are dominated by the minimum airflows. This, in turn, results in the VAV optimization procedure calculating a lower operating cost for any given duct configuration. The savings in life cycle cost yielded by the VAV optimization procedure result from it being able to take advantage of the knowledge that the operating cost is lower than that calculated by the T-method. Although this is due to using the actual flow rates, it is analogous to having a lower electricity rate for the VAV optimization procedure. For any given building/duct topology/climate/etc. combination, the reduced operating cost is equivalent to a fixed percentage reduction in the electricity rate. As can be seen in Figure 9, the operating cost, initial cost, and life cycle cost all change more rapidly at lower electricity rates. Arguing by analogy provides an explanation for why the VAV optimization procedure (compared to the T-method) performs better at lower electricity rates. At lower electricity rates, the life cycle cost is more sensitive to a change in electricity rate or a change in operating cost caused by evaluating electricity consumption at low flow rates.

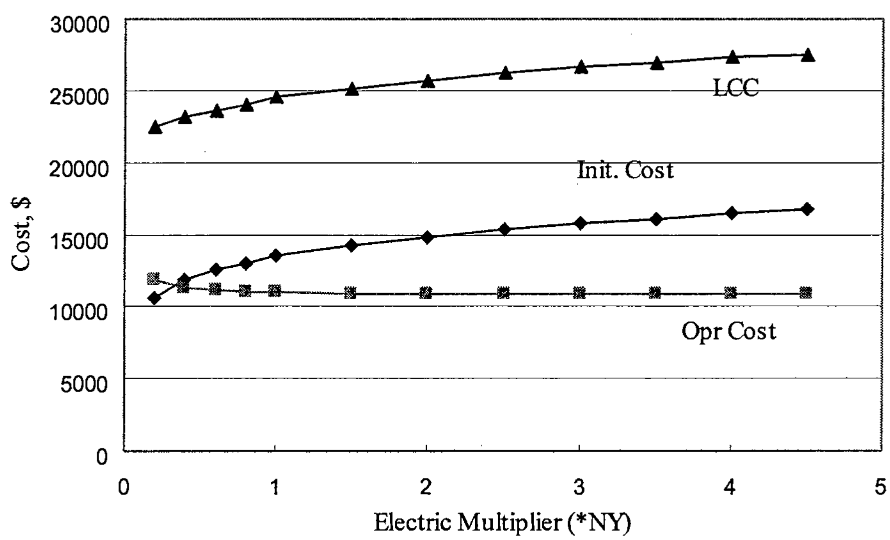


Figure 6.2 Plot of Costs in terms of Electric Rate of Duct Systems of the Large Office Building in Oklahoma with the NY Electric Rate

However, in actual practice, the change in performance is significantly damped by duct size constraints and static pressure requirements. Therefore, the VAV optimization procedure does not appear to offer significant enough savings to warrant its use in practice. (Since it requires a significant increase in the amount of input data and the computational time required.) Instead, the T-method seems to offer a good balance between results and ease-of-use, when implemented in a computer program.

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

The VAV optimization procedure was applied to the three VAV duct systems to investigate the impact of varying airflow rates on the sizing of duct systems. For comparison purposes, other duct design methods, such as equal friction, static regain, and the T-method were also applied to the duct systems. While the VAV optimization procedure uses varying airflows, the other methods use peak constant airflows for duct system design. The equal friction and static regain methods calculate system pressure loss after duct sizes are decided. The T-method calculated fan pressure using electric rate and ductwork unit cost and then optimized duct sizes. The VAV optimization procedure selects a fan by checking as to whether the system design point with the peak hour's airflow and other system operating points for varying airflows reside in the fan operating region. After fan selection, the downhill simplex method searches optimum duct sizes through evaluation of the life-cycle cost.

After optimum duct sizes are found, the duct systems resulting from four different duct design methods are simulated under operation for a typical meteorological year in order to investigate the performance of the methods. With respect to life cycle cost, the VAV optimization procedure showed 6 ~ 19% savings compared to the equal friction method, 2 ~ 13% savings over the static regain method, and 1 ~ 4% savings over the T-method. Compared to the T-method, the VAV optimization procedure gives a lower initial cost and a higher operating cost. The total duct surface, and hence the initial cost, using the VAV optimization procedure was significantly lower compared to other duct design method. The VAV optimization procedure saved duct surface 23 ~ 31 % over the equal friction method, 13 ~ 22% over the static regain method, and 4 ~ 7% over the T-method.

Trends that were identified include:

- The VAV optimization procedure allowed greater life cycle cost savings (compared to the T-method) with lower electricity rates.
- For the large office building, the life cycle cost savings of the VAV optimization procedure compared to the T-method decrease as duct unit cost increases.
- The duct topology influences the degree to which the optimal solution is at the duct size constraints. Longer duct lengths tended to reduce the number of duct sections at the minimum size constraint.
- While different climate conditions and operating schedules influenced the optimal design, they did not have a significant impact on the savings of the VAV optimization procedure compared to the T-method.

In general, the VAV optimization procedure yields significant life cycle cost savings compared with the equal friction and static regain methods. However, compared with the T-method, the life cycle cost savings of the VAV optimization procedure was not as great. This is partly due to two significant limitations that prevent the VAV optimization procedure from finding significantly better designs. First, optimal duct sizes are found near to the lower limits, which are constrained by the velocity limitation.

Second, the change of system pressure drop due to changing the duct surface area is smaller than expected because of the static pressure requirement at the longest duct line.

Even when these limitations are artificially eliminated, the optimization domain analysis showed that the life cycle cost is relatively insensitive to the total duct surface area, when the duct design has been arrived at by the T-method. This indicates that the T-method can be used favorably even in VAV system optimization. The T-method has great potential to save costs over the non-optimization-based methods, without the input data and computation time requirements of the VAV optimization method. Therefore, it is recommended that the T-method be utilized for duct design in VAV systems. Given the marginal improvement in life cycle cost yielded by the VAV optimization procedure compared to the T-method, further research is probably not warranted at this time. Nevertheless, if the situation arose where the procedure could be profitably applied, it would be useful to decrease the computational requirements. This might be achieved by modeling a statistical representation of the airflow data rather than all 8760 hours.

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APPENDIX

APPENDIX A--Survey on the HVAC duct System Design

Survey on the HVAC duct system design

Thank you for taking part in the survey on the current trends of the HVAC duct system design. The purpose of this survey is two-fold: first, to determine which duct design method are most commonly used for variable air volume (VAV) systems; and secondly, to determine how fans for VAV systems are selected and controlled.

The survey results will be referenced for my Ph.D. research work, which involves finding an optimum duct design method for VAV duct systems. All participants' specific information will be held strictly confidential and will not be used for any commercial purpose. Some comments from survey participants may be quoted in my final report, but they will be treated anonymously.

I appreciate your time and input in answering the following questions. This survey will take approximately 5 minutes. Please return the completed survey in the enclosed stamped and addressed envelope.

Thanks.

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1. In the last year, approximately how many duct systems for variable air volume systems have you designed?

2. Which design methods do you use to size ducts?

static regain   balanced capacity   equal friction

T-method   other______________________

3. Do you incorporate any diversity when sizing

the duct system?   the fan?

If so, how do you incorporate it?

4. What type of fans do you specify for VAV systems?

Type	Frequency				
	Never	Rarely	Occasiona lly	Often	Very Often
Airfoil Backward Inclined Forward Curved Other:____________					

5. How do you select a fan to avoid fan operation in the surge region?

6. When you select a fan for a duct system, how do you determine the required fan pressure? In addition to pressure loss from ducts and fittings, do you include:

static pressure that should be maintained at the end of the longest duct line.

a positive pressure at the zone to prevent air infiltration.

other____

7. Do you have any specific method or comments about how you select a fan?

8. If the duct static pressure at the end of the longest duct line is controlled,

If the pressure is constant year-round, what is it?

If it is not constant, how is the level scheduled?

9. If you specify a positive air pressure in a zone, what air pressure do you specify?

10. If you have any other comments related to this survey, please add them here.

APPENDIX B-- Tables for Parametric Study

TABLE B1.
Duct Sizes of the ASHRAE Example (8760-hour schedule, Aluminum ducts

Unit: inch (1 in.=0.025 m)

Duct Section	Max Airflow, cfm	Min. Duct Size	Max Duct Size	Equal Friction	ASHRA E Ans.	Static Regain	T-method				VAV optimization Procedure			
							TSAL E Rate	OK E Rate	MN E Rate	NY E Rate	TSAL E Rate	OK E Rate	MN E Rate	NY E Rate
Return: 1	1504	10	20	15	12.00	14	10	11	11	11	10	10	11	12
2	509	6	12	10	8.00	9	7	7	7	8	7	7	7	8
3	2013	11	24	17	12.00	14	13	14	14	14	13	14	14	13
4	1992	11	24	17	26.20	14	16	17	17	18	12	12	13	14
5	1992	11	24	17	15.00	14	12	13	13	13	12	13	13	14
6	4005	16	34	22	17.00	17	20	20	20	22	17	17	17	18
Supply:7	593	6	13	11	10.90	9	8	8	8	8	6	6	6	6
8	593	6	13	11	10.90	9	8	8	10	9	12	13	13	13
9	1187	9	19	12	15.20	12	10	10	10	11	12	13	13	13
10	1187	9	19	12	13.70	12	13	13	13	14	12	13	13	13
11	996	8	17	13	10.90	14	11	11	11	11	9	9	8	8
12	996	8	17	13	10.90	12	9	10	9	10	9	8	8	8
13	1992	11	24	17	12.90	16	12	12	12	13	12	12	11	11
14	3179	14	30	20	17.10	17	17	17	17	18	16	17	17	17
15	403	5	11	9	7.60	9	6	6	6	6	6	5	5	5
16	403	5	11	9	7.60	7	6	6	6	6	6	5	5	5
17	805	7	15	12	8.40	9	8	8	8	8	7	7	7	7
18	3984	16	34	22	18.80	17	26	26	26	28	24	24	24	26
Root: 19	3984	16	34	22	25.20	17	26	26	26	28	24	24	24	26

* Numbers in shaded cells indicate duct sizes at lower bound.

TABLE B2. Duct Sizes of the Large Office Building in OK (8760-hour Schedule, Aluminum ducts)

Unit: inch (1 in.=0.025 m)

Duct Section	Max Airflow, Cfm	Min. Size	Max. Size	Equal Friction	Static Regain	T-method				VAV optimization Procedure			
						TSAL E Rate	OK E Rate	MN E Rate	NY E Rate	TSAL E Rate	OK E Rate	MN E Rate	NY E Rate
1	8678	24	50	30	24	26	26	26	26	24	24	24	26
2	2699	13	28	18	14	13	13	13	14	13	13	13	13
3	1800	11	22	16	14	12	12	12	12	11	11	11	11
4	1350	9	20	14	14	9	9	9	10	9	9	9	11
5	900	8	16	12	12	8	8	8	9	8	8	8	9
6	450	6	11	10	9	7	7	7	7	6	7	7	7
7	900	8	16	12	9	8	8	8	8	8	8	8	8
8	1637	10	22	16	11	11	11	11	11	10	10	10	10
9	910	8	16	12	9	9	9	9	9	8	8	8	8
10	728	7	15	11	9	7	8	8	8	8	7	7	7
11	546	6	13	10	9	6	6	6	6	6	6	6	6
12	364	5	10	9	9	5	5	5	6	6	5	5	6
13	182	4	7	7	7	4	4	4	4	4	4	4	5
14	182	4	7	7	5	4	4	4	4	4	4	4	4
15	728	7	15	11	9	7	8	8	8	7	7	7	7
16	182	4	7	7	5	4	4	4	4	4	4	4	4
17	546	6	13	10	9	6	6	6	6	6	6	6	6
18	364	5	10	9	9	5	5	5	5	5	5	5	5
19	182	4	7	7	7	4	4	4	4	4	4	4	4
20	4341	16	36	22	18	19	19	19	20	16	20	17	20
21	728	7	15	11	9	7	7	7	8	7	7	7	7
22	546	6	13	10	9	6	6	6	6	7	6	6	7
23	364	5	10	9	9	5	5	5	6	5	5	6	5
24	182	4	7	7	7	4	4	4	4	4	4	5	4
25	3614	15	32	22	18	15	16	16	16	15	16	16	16
26	723	7	14	11	9	7	7	7	7	7	7	7	7
27	1445	10	20	15	14	10	10	10	11	10	10	10	10
28	1084	8	18	13	14	9	9	9	9	8	8	8	8
29	723	7	14	11	12	8	8	8	8	7	7	7	7
30	361	5	10	9	9	6	6	6	6	5	5	5	5
31	1445	10	20	15	14	12	12	12	13	11	11	10	10
32	1084	8	18	13	14	10	10	10	10	8	8	9	9
33	723	7	14	11	12	8	9	9	9	7	8	7	7
34	361	5	10	9	9	7	7	7	7	6	6	6	6

TABLE B3. Duct Sizes of the Large Office Building in MN (8760-hour schedule, Aluminum ducts)

Unit: inch (1 in.=0.025 m)

Duct Section	Max Airflow, Cfm	Min. Size	Max. Size	Equal Friction	Static Regain	T-method				VAV optimization Procedure			
						TSAL E Rate	OK E Rate	MN E Rate	NY E Rate	TSAL E Rate	OK E Rate	MN E Rate	NY E Rate
1	7584	22	48	28	22	24	24	24	26	22	24	22	24
2	2481	12	26	18	14	13	13	13	13	12	12	12	12
3	1654	10	22	15	14	11	11	11	12	10	11	11	11
4	1240	9	19	14	14	9	9	9	9	9	9	10	11
5	827	7	16	12	12	8	8	8	8	7	8	10	8
6	413	5	11	9	9	6	6	6	7	5	6	6	7
7	827	7	16	12	9	6	6	6	6	7	7	7	7
8	1254	9	19	14	9	10	10	10	10	9	9	9	9
9	697	7	14	11	9	8	8	8	8	7	8	7	7
10	558	6	13	10	9	7	7	7	7	7	6	6	7
11	418	5	11	9	9	6	6	6	6	5	5	6	6
12	279	5	9	8	9	5	5	5	5	5	5	5	5
13	139	3	6	6	6	4	4	4	4	4	4	4	4
14	139	3	6	6	5	3	3	3	3	3	3	3	3
15	558	6	13	10	7	7	7	7	7	6	6	6	6
16	139	3	6	6	4	3	3	3	3	3	3	3	3
17	418	5	11	9	7	6	6	6	6	5	5	5	5
18	279	5	9	8	7	5	5	5	5	5	5	5	5
19	139	3	6	6	5	4	4	4	4	3	4	3	4
20	3849	15	34	22	18	18	18	18	19	16	17	16	19
21	558	6	13	10	9	7	7	7	7	6	6	6	6
22	418	5	11	9	9	6	6	6	6	5	5	6	6
23	279	5	9	8	9	5	5	5	5	5	5	6	5
24	139	3	6	6	6	4	4	4	4	5	4	4	4
25	3291	14	30	20	18	15	15	15	16	14	14	16	17
26	658	7	14	11	9	7	7	7	7	7	7	7	7
27	1317	9	20	14	14	10	10	10	10	9	9	9	9
28	987	8	17	13	14	8	8	8	9	8	8	9	8
29	658	7	14	11	12	7	7	7	8	7	7	7	7
30	329	5	10	9	9	6	6	6	6	5	6	5	5
31	1317	9	20	14	14	12	12	12	12	9	11	9	9
32	987	8	17	13	14	9	9	9	10	8	9	9	9
33	658	7	14	11	12	8	8	8	9	7	8	7	8
34	329	6	10	9	9	7	7	7	7	6	6	6	6

TABLE B4. Cost Comparison between Different Duct Design Methods (TSAL electric rate)

Duct System	Duct Design Method	Duct Cost (Aluminum)	Fan Cost, $	Operating Cost,$	L.C. Cost, $	Saving % of VAV Opt
ASHRAE Example	Equal Friction	9131	2110	2022	13263	14.4
	ASHRAE Ans.	7914	2110	2101	12125	6.4
	Static Regain	8022	2110	2138	12270	7.5
	T-method	7331	2110	2224	11665	2.7
	VAV Opt	6995	2110	2243	11348	0.0
Building in OK	Equal Friction	8945	2575	2950	14470	17.3
	Static Regain	7625	2575	3104	13303	10.0
	T-method	6535	2575	3116	12227	2.1
	VAV Opt	6214	2575	3178	11967	0.0
Building in MN	Equal Friction	8440	2420	2559	13419	18.7
	Static Regain	7509	2420	2642	12571	13.3
	T-method	6288	2420	2662	11370	4.1
	VAV Opt	5667	2420	2817	10905	0.0

TABLE B5. Cost Comparison between Different Duct Design Methods (OK electric rate)

Duct System	Duct Design Method	Duct Cost (Aluminum)	Fan Cost, $	Operating Cost,$	L.C. Cost, $	Saving % of VAV Opt
ASHRAE Example	Equal Friction	9131	2110	6051	17293	9.4
	ASHRAE Ans.	7914	2110	6207	16231	3.5
	Static Regain	8022	2110	6276	16407	4.5
	T-method	7539	2110	6311	15960	1.8
	VAV Opt	7107	2110	6451	15668	0.0
Building in OK	Equal Friction	8945	2575	7702	19223	12.2
	Static Regain	7625	2575	7965	18165	7.0
	T-method	6614	2575	7904	17093	1.2
	VAV Opt	6372	2575	7940	16887	0.0
Building in MN	Equal Friction	8440	2420	6917	17777	12.2
	Static Regain	7509	2420	7060	16989	8.1
	T-method	6288	2420	7098	15806	1.2
	VAV Opt	6015	2420	7182	15617	0.0

TABLE B6. Cost Comparison between Different Duct Design Methods (MN electric)

Duct System	Duct Design Method	Duct Cost (Aluminum)	Fan Cost, $	Operating Cost,$	L.C. Cost, $	Saving % of VAV Opt
ASHRAE Example	Equal Friction	9131	2110	6423	17664	9.2
	ASHRAE Ans.	7914	2110	6616	16640	3.6
	Static Regain	8022	2110	6693	16408	2.2
	T-method	7524	2110	6742	16377	2.0
	VAV Opt	7128	2110	6806	16044	0.0
Building in OK	Equal Friction	8945	2575	8490	20010	11.5
	Static Regain	7625	2575	8825	19025	6.9
	T-method	6614	2575	8783	17972	1.5
	VAV Opt	6236	2575	8897	17707	0.0
Building in MN	Equal Friction	8440	2420	7676	18536	11.7
	Static Regain	7509	2420	7861	17790	8.0
	T-method	6288	2420	7901	16609	1.4
	VAV Opt	5862	2420	8087	16369	0.0

TABLE B7. Cost Comparison between Different Duct Design Methods (NY electric)

Duct System	Duct Design Method	Duct Cost (Aluminum)	Fan Cost, $	Operating Cost,$	L.C. Cost, $	Saving % of VAV Opt
ASHRAE Example	Equal Friction	9131	2110	10182	21424	6.1
	ASHRAE Ans.	7914	2110	10535	20560	2.2
	Static Regain	8022	2110	10705	20837	3.5
	T-method	7913	2110	10524	20547	2.1
	VAV Opt	7459	2110	10546	20115	0.0
Building in OK	Equal Friction	8945	2575	14197	25718	7.8
	Static Regain	7625	2575	14861	25060	5.3
	T-method	6835	2575	14631	24042	1.3
	VAV Opt	6478	2575	14668	23720	0.0
Building in MN	Equal Friction	8440	2420	12385	23246	8.7
	Static Regain	7509	2420	12745	22673	6.4
	T-method	6546	2420	12651	21617	1.8
	VAV Opt	6115	2420	12696	21231	0.0

TABLE B8. Duct Sizes of the Large Office Building in OK (8760-hr schedule, Ga. Steel ducts)

unit: inch (1 in.=0.025 m)

Duct Section	Max Airflow, cfm	Min. Size, in	Max. Size, in	Equal Friction	Static Regain	T-method			VAV Opt. Procedure			
						OK E Rate	MN E Rate	NY E Rate	OK E Rate	MN E Rate	NY E Rate	OK E Rate (doubled duct cost)
1	8678	24	50	30	24	26	26	26	24	24	26	24
2	2699	13	28	18	14	14	14	14	13	13	13	13
3	1800	11	22	16	14	12	12	12	11	11	12	11
4	1350	9	20	14	14	9	9	9	9	10	9	9
5	900	8	16	12	12	8	8	8	8	8	8	9
6	450	6	11	10	9	7	6	7	7	6	7	6
7	900	8	16	12	9	8	8	8	8	8	8	8
8	1637	10	22	16	11	11	11	11	10	10	10	10
9	910	8	16	12	9	9	9	9	8	8	8	8
10	728	7	15	11	9	7	7	7	7	7	8	7
11	546	6	13	10	9	6	6	6	6	6	6	6
12	364	5	10	9	9	5	5	5	5	5	5	5
13	182	4	7	7	7	4	4	4	4	4	4	4
14	182	4	7	7	5	4	4	4	4	4	4	4
15	728	7	15	11	9	7	7	8	7	7	7	7
16	182	4	7	7	5	4	4	4	4	4	4	4
17	546	6	13	10	9	6	6	6	7	6	6	6
18	364	5	10	9	9	5	5	5	5	6	5	5
19	182	4	7	7	7	4	4	4	4	4	4	4
20	4341	16	36	22	18	17	17	17	16	17	18	16
21	728	7	15	11	9	7	7	7	7	7	7	7
22	546	6	13	10	9	6	6	6	7	7	6	7
23	364	5	10	9	9	5	5	5	5	6	5	5
24	182	4	7	7	7	4	4	4	4	4	5	4
25	3614	15	32	22	18	15	15	16	16	16	18	15
26	723	7	14	11	9	7	7	7	7	7	7	7
27	1445	10	20	15	14	10	10	10	10	10	10	10
28	1084	8	18	13	14	8	8	9	8	9	8	8
29	723	7	14	11	12	7	7	8	7	7	7	8
30	361	5	10	9	9	6	6	6	5	5	5	5
31	1445	10	20	15	14	12	12	12	10	10	10	10
32	1084	8	18	13	14	9	9	10	9	9	9	9
33	723	7	14	11	12	8	8	9	7	7	8	7
34	361	5	10	9	9	7	7	7	6	6	6	5

TABLE B9. Cost Comparison between Different Duct Design Methods:
when the large office building in OK uses galvanized steel ducts

Unit: $

Electric Rate	Duct Design Method	Duct Cost (Ga. Steel)	Fan Cost	Operating Cost	L.C. Cost	Saving % of VAV Opt
OK Electric Rate	Equal Friction	11474	2575	7845	21893	12.7
	Static Regain	9915	2575	8223	20712	7.7
	T-method	8362	2575	8338	19276	0.8
	VAV Opt	7957	2575	8591	19123	0.0
MN Electric Rate	Equal Friction	11474	2575	8671	22720	11.0
	Static Regain	9915	2575	9151	21641	6.6
	T-method	8349	2575	9367	20292	0.4
	VAV Opt	8025	2575	9613	20213	0.0
NY Electric Rate	Equal Friction	11474	2575	14557	28606	6.8
	Static Regain	9915	2575	15506	27996	4.8
	T-method	8491	2575	15765	26831	0.7
	VAV Opt	8329	2575	15750	26654	0.0

TABLE B10. Duct Sizes of the Large Office Building in OK with Setback Control Operation
(Optimized with aluminum ducts and OK electric rate)

Unit: inch (1 in.=0.025 m)

Duct Section	Max Airflow Cfm	Min Size	Max. Size	Setback-control Schedule			8760-hr Schedule
				Eq. Friction	T-method	VAV Opt	VAV Opt
1	8789	24	50	30	26	26	24
2	2730	13	28	19	14	13	13
3	1820	11	22	16	12	11	11
4	1365	9	20	15	10	9	9
5	910	8	16	13	8	8	8
6	455	6	11	9	7	7	7
7	910	8	16	13	8	8	8
8	1639	10	22	16	11	11	10
9	911	8	16	13	9	9	8
10	729	7	15	11	8	7	7
11	546	6	13	10	6	6	6
12	364	5	10	9	6	5	5
13	182	4	7	7	4	4	4
14	182	4	7	7	4	4	4
15	729	7	15	11	8	7	7
16	182	4	7	7	4	4	4
17	546	6	13	10	6	7	6
18	364	5	10	9	5	5	5
19	182	4	7	7	4	4	4
20	4420	16	36	22	20	18	20
21	729	7	15	11	8	7	7
22	546	6	13	10	6	6	6
23	364	5	10	9	5	6	5
24	182	4	7	7	4	4	4
25	3691	15	32	22	16	16	16
26	738	7	14	11	7	7	7
27	1477	10	20	15	10	10	10
28	1107	8	18	13	9	8	8
29	738	7	14	11	8	7	7
30	369	5	10	9	6	6	5
31	1477	10	20	15	13	11	11
32	1107	8	18	13	10	9	8
33	738	7	14	11	9	8	8
34	369	5	10	9	7	7	6

TABLE B11. Duct Sizes of the Large Office Building in MN with Setback Control Operations
(Optimized with aluminum ducts and MN electric rate)

Unit: inch (1 in.=0.025 m)

Duct Section	Max Airflow cfm	Min Size	Max. Size	Setback-control schedule			8760-hr schedule
				Eq. Friction	T-method	VAV Opt	VAV Opt
1	7574	22	48	28	24	22	22
2	2498	12	26	18	13	13	12
3	1665	10	22	16	11	11	11
4	1249	9	19	14	9	9	10
5	833	7	16	12	8	8	10
6	416	5	11	9	6	6	6
7	833	7	16	12	7	7	7
8	1213	9	19	14	10	9	9
9	674	7	14	11	8	7	7
10	539	6	13	10	7	6	6
11	404	5	11	9	6	6	6
12	269	4	9	8	5	5	5
13	135	3	6	6	4	4	4
14	135	3	6	6	3	3	3
15	539	6	13	10	7	6	6
16	135	3	6	6	3	3	3
17	404	5	11	9	6	6	5
18	269	4	9	8	5	4	5
19	135	3	6	6	4	4	3
20	3864	15	34	22	18	16	16
21	539	6	13	10	7	6	6
22	404	5	11	9	6	6	6
23	269	4	9	8	5	5	6
24	135	3	6	6	4	5	4
25	3325	14	30	20	15	16	16
26	665	7	14	11	7	7	7
27	1330	9	20	14	10	9	9
28	997	8	17	13	8	8	9
29	665	7	14	11	7	7	7
30	332	5	10	9	6	6	5
31	1330	9	20	14	12	10	9
32	997	8	17	13	9	9	9
33	665	7	14	11	8	8	7
34	332	6	10	9	7	6	6

TABLE B12. Cost Comparison between Different Duct Design Methods when two different system operating schedules are used

Unit: $

Duct System	Duct Design Method	Duct Cost (Aluminum)	Fan Cost	Opr. Cost	L.C. Cost	Saving % of VAV Opt.
Building in OK with OK Electric Rate	Equal Friction Setback control	9029	2575	5649	17254	13.6
	T-method Setback control	6788	2575	5802	15165	1.7
	VAV Opt Setback control	6488	2575	5845	14908	0.0
	VAV Opt 8760-hr	6372	2575	7940	16887	11.7
Building in MN with MN Electric Rate	Equal Friction Setback control	8467	2420	6885	17771	12.5
	T-method Setback control	6288	2420	7110	15818	1.7
	VAV Opt Setback control	5951	2420	7182	15554	0.0
	VAV Opt 8760-hr	5862	2420	8087	16369	5.0

APPENDIX C-- BLAST Input Files

(1) Blast Input File for the Large Office Building in Oklahoma with On-Schedule

```
BEGIN INPUT;
 RUN CONTROL:
   NEW ZONES,
   NEW AIR SYSTEMS,
   PLANT,
   DESIGN SYSTEMS,
   REPORTS(96,plant loads),
   UNITS(IN=ENGLISH, OUT=ENGLISH);
   TEMPORARY MATERIALS:
    concrete1= (L=0.2917,K=1.7296,D=600.0,CP=0.837,ABS=0.65, TABS=0.900,ROUGH);
    aluminium1 =(L=0.0104,K=100.0000,D=171.0,CP=0.214,ABS=0.20,TABS=0.000,SMOOTH);
    bron2 = (R=0.070,SC=0.71,SMOOTH,GLASS);
    bron1= (R=0.980,SC=0.57,SMOOTH,GLASS);
   END;
   TEMPORARY WALLS:
    wall1= (aluminium1, IN81 - BOARD INSULATION, CB53 - 8 IN MW HOLLOW CBLK,
               IN82 - STANDARD BATT INSULATION);
   END;
   TEMPORARY FLOORS:
     cfloor1= (E8 - 5 / 8 IN PLASTER OR GYP BOARD, AIRSPACE - CEILING, concrete1 ,
        FINISH FLOORING - CARPET FIBROUS PAD);
     cfloor2= (E5 - ACOUSTIC TILE , AIRSPACE - CEILING , concrete1 ,
        FINISH FLOORING - CARPET FIBROUS PAD);
   END;
   TEMPORARY ROOFS:
     cceil1= (FINISH FLOORING - CARPET FIBROUS PAD , concrete1 ,
        AIRSPACE - CEILING , E8 - 5 / 8 IN PLASTER OR GYP BOARD);
     cceil2= (FINISH FLOORING - CARPET FIBROUS PAD , concrete1 ,
        AIRSPACE - CEILING , E5 - ACOUSTIC TILE );
   END;
   TEMPORARY WINDOWS:
    window2 = (bron2);
    window1 = (bron1);
   END;
   TEMPORARY SCHEDULE (people):
    MONDAY THRU FRIDAY=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,1.00,1.00,1.00,
        1.00,1.00,1.00,1.00,1.00,1.00,1.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
    SATURDAY=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
        0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
    SUNDAY=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
        0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
    HOLIDAY=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
        0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
    SPECIAL1=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
        0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
    SPECIAL2=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
        0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
    SPECIAL3=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
        0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
    SPECIAL4=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
        0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00);
   END;
   TEMPORARY SCHEDULE (lights):
    MONDAY THRU FRIDAY=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,1.00,1.00,1.00,
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      1.00,1.00,1.00,1.00,1.00,1.00,1.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   SATURDAY=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   SUNDAY=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   HOLIDAY=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   SPECIAL1=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   SPECIAL2=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   SPECIAL3=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   SPECIAL4=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00);

  END;
  TEMPORARY SCHEDULE (equipment):
   MONDAY THRU FRIDAY=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,1.00,1.00,1.00,
      1.00,1.00,1.00,1.00,1.00,1.00,1.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   SATURDAY=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   SUNDAY=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   HOLIDAY=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   SPECIAL1=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   SPECIAL2=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   SPECIAL3=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00),
   SPECIAL4=(0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
      0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00);
  END;

TEMPORARY CONTROLS (VAV):
   PROFILES:
    VAV=(1.0000 AT 55.00, 0.83683 AT 67.00, 0.0000 AT 69.4121,
           -0.17143 AT 70.0,-0.19429 AT 72.00,-0.6 AT 76,-1.0 AT 90);
   SCHEDULES:
      MONDAY THRU FRIDAY=(0 TO 24-VAV),
      SATURDAY=(0 TO 24-VAV),
      SUNDAY=(0 TO 24-VAV),
      HOLIDAY=(0 TO 24-VAV),
      SPECIAL1=(0 TO 24-VAV),
      SPECIAL2=(0 TO 24-VAV),
      SPECIAL3=(0 TO 24-VAV),
      SPECIAL4=(0 TO 24-VAV);
  END;
   PROJECT="Project of BOK simulation (HBLC) 2000 1151 ";
  LOCATION=TULSA ;
    DESIGN DAYS=TULSA SUMMER, TULSA WINTER ;
    WEATHER TAPE FROM 01JAN THRU 31DEC;
    REPORT FILE FROM 01JAN THRU 31DEC;
  GROUND TEMPERATURES=(55, 55, 55, 55, 55, 55, 55, 55, 55, 55, 55, 55);
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BEGIN BUILDING DESCRIPTION;
  BUILDING="BOK- Part #1 ";
   NORTH AXIS=340.00;
   SOLAR DISTRIBUTION=1;
   ZONE 18 "Zone 18 @ 850 204 0 T":
    ORIGIN:(143.20, 123.77, 0.00);
    NORTH AXIS=0.00;
    PARTITIONS :
     STARTING AT(0.00, 0.00, 0.00)
     FACING(180.00)
     TILTED(90.00)
     PARTITION02 (85.00 BY 12.80),
     STARTING AT(85.00, 0.00, 0.00)
     FACING(90.00)
     TILTED(90.00)
     PARTITION02 (20.40 BY 12.80),
     STARTING AT(85.00, 20.40, 0.00)
     FACING(0.00)
     TILTED(90.00)
     PARTITION02 (85.00 BY 12.80),
     STARTING AT(0.00, 20.40, 0.00)
     FACING(270.00)
     TILTED(90.00)
     PARTITION02 (20.40 BY 12.80);
    FLOORS :
     STARTING AT(0.00, 0.00, 0.00)
     FACING(90.00)
     TILTED(180.00)
     cfloor1 (20.40 BY 85.00);
    CEILINGS :
     STARTING AT(0.00, 0.00, 12.80)
     FACING(180.00)
     TILTED(0.00)
     cceil1 (85.00 BY 20.40);
    PEOPLE=15,people ,
     AT ACTIVITY LEVEL 0.45, 60.00 PERCENT RADIANT,
     FROM 01JAN THRU 31DEC;
    LIGHTS=7.10,lights ,
      0.00 PERCENT RETURN AIR, 40.00 PERCENT RADIANT,
      20.00 PERCENT VISIBLE, 0.00 PERCENT REPLACEABLE,
      FROM 01JAN THRU 31DEC;
    ELECTRIC EQUIPMENT=5.92,equipment ,
     30.00 PERCENT RADIANT, 0.00 PERCENT LATENT, 0.00 PERCENT LOST,
     FROM 01JAN THRU 31DEC;
       CROSS MIXING=500,CONSTANT ,
     FROM ZONE 19, 0.00 DEL TEMP,
     FROM 01JAN THRU 31DEC;
    CONTROLS=VAV, 2.587 HEATING, 18.863 COOLING,
     0.00 PERCENT MRT,
     FROM 01JAN THRU 31DEC;
   END ZONE;
   ZONE 19 "Zone 19 @ 1500 782 0 T":
    ORIGIN:(121.30, 123.87, 0.00);
    NORTH AXIS=0.00;
    PARTITIONS :
```

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 STARTING AT(0.00, 0.00, 0.00)
 FACING(180.00)
 TILTED(90.00)
 PARTITION23 (22.50 BY 12.80),
 STARTING AT(22.50, 0.00, 0.00)
 FACING(90.00)
 TILTED(90.00)
 PARTITION02 (20.40 BY 12.80),
 STARTING AT(22.50, 20.40, 0.00)
 FACING(180.00)
 TILTED(90.00)
 PARTITION02 (85.00 BY 12.80),
 STARTING AT(107.50, 20.40, 0.00)
 FACING(270.00)
 TILTED(90.00)
 PARTITION02 (20.40 BY 12.80),
 STARTING AT(107.50, 0.00, 0.00)
 FACING(180.00)
 TILTED(90.00)
 PARTITION23 (22.50 BY 12.80),
 STARTING AT(130.00, 0.00, 0.00)
 FACING(90.00)
 TILTED(90.00)
 PARTITION23 (78.20 BY 12.80),
 STARTING AT(130.00, 78.20, 0.00)
 FACING(0.00)
 TILTED(90.00)
 PARTITION23 (130.00 BY 12.80),
 STARTING AT(0.00, 78.20, 0.00)
 FACING(270.00)
 TILTED(90.00)
 PARTITION23 (78.20 BY 12.80);
FLOORS :
 STARTING AT(0.00, 0.00, 0.00)
 FACING(90.00)
 TILTED(180.00)
 cfloor1 (91.80 BY 91.90);
CEILINGS :
 STARTING AT(0.00, 0.00, 12.80)
 FACING(180.00)
 TILTED(0.00)
 cceil1 (91.90 BY 91.80);
PEOPLE=65,people ,
 AT ACTIVITY LEVEL 0.45, 60.00 PERCENT RADIANT,
 FROM 01JAN THRU 31DEC;
LIGHTS=34.54,lights ,
   0.00 PERCENT RETURN AIR, 40.00 PERCENT RADIANT,
  20.00 PERCENT VISIBLE, 0.00 PERCENT REPLACEABLE,
 FROM 01JAN THRU 31DEC;
ELECTRIC EQUIPMENT=28.78,equipment ,
  30.00 PERCENT RADIANT, 0.00 PERCENT LATENT, 0.00 PERCENT LOST,
 FROM 01JAN THRU 31DEC;
CROSS MIXING=500,CONSTANT ,
 FROM ZONE 18, 0.00 DEL TEMP,
 FROM 01JAN THRU 31DEC;
    CROSS MIXING=3628,CONSTANT ,
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    FROM ZONE 20, 0.00 DEL TEMP,
    FROM 01JAN THRU 31DEC;
      CROSS MIXING=2374,CONSTANT ,
    FROM ZONE 21, 0.00 DEL TEMP,
    FROM 01JAN THRU 31DEC;
      CROSS MIXING=2710,CONSTANT ,
    FROM ZONE 22, 0.00 DEL TEMP,
    FROM 01JAN THRU 31DEC;
  CONTROLS=VAV,
          11.988 HEATING, 87.410 COOLING,
   0.00 PERCENT MRT,
   FROM 01JAN THRU 31DEC;
END ZONE;
ZONE 20 "Zone 20 @ 200 999 0 T":
  ORIGIN:(251.39, 107.41, 0.00);
  NORTH AXIS=0.00;
  EXTERIOR WALLS :
    STARTING AT(14.90, 0.00, 0.00)
    FACING(90.00)
    TILTED(90.00)
    wall1 (99.90 BY 12.80)
      WITH WINDOWS OF TYPE
      window1 (99.00 BY 11.80)
        REVEAL(0.00)
        AT (0.60, 0.10),
    STARTING AT(14.90, 99.90, 0.00)
    FACING(0.00)
    TILTED(90.00)
    wall1 (14.90 BY 12.80)
      WITH WINDOWS OF TYPE
      window1 (14.00 BY 11.80)
        REVEAL(0.00)
        AT (0.40, 0.50);
  PARTITIONS :
    STARTING AT(0.00, 0.00, 0.00)
    FACING(180.00)
    TILTED(90.00)
    PARTITION23 (14.90 BY 12.80),
    STARTING AT(0.00, 99.90, 0.00)
    FACING(270.00)
    TILTED(90.00)
    PARTITION23 (99.90 BY 12.80);
  FLOORS :
    STARTING AT(0.00, 0.00, 0.00)
    FACING(90.00)
    TILTED(180.00)
    cfloor1 (99.90 BY 14.90);
  CEILINGS :
    STARTING AT(0.00, 0.00, 12.80)
    FACING(180.00)
    TILTED(0.00)
    cceil1 (14.90 BY 99.90);
  PEOPLE=10,people ,
    AT ACTIVITY LEVEL 0.45, 60.00 PERCENT RADIANT,
    FROM 01JAN THRU 31DEC;
  LIGHTS=6.10,lights ,
```

```
    0.00 PERCENT RETURN AIR, 40.00 PERCENT RADIANT,
    20.00 PERCENT VISIBLE, 0.00 PERCENT REPLACEABLE,
   FROM 01JAN THRU 31DEC;
  ELECTRIC EQUIPMENT=5.08,equipment ,
    30.00 PERCENT RADIANT, 0.00 PERCENT LATENT, 0.00 PERCENT LOST,
   FROM 01JAN THRU 31DEC;
  CROSS MIXING=3628,CONSTANT ,
   FROM ZONE 19, 0.00 DEL TEMP,
   FROM 01JAN THRU 31DEC;
  Infiltration=363.00,CONSTANT,
    WITH COEFFICIENTS (0.606000,0.020000,0.000598,0.000000),
    FROM 01JAN THRU 31DEC;
  CONTROLS=VAV, 86.444 HEATING, 137.118 COOLING,
   0.00 PERCENT MRT,
   FROM 01JAN THRU 31DEC;
 END ZONE;
 ZONE 21 "Zone 21 @ 1316 179 0 T":
  ORIGIN:(120.15, 201.65, 0.00);
  NORTH AXIS=0.00;
  EXTERIOR WALLS :
   STARTING AT(131.60, 4.90, 0.00)
   FACING(0.00)
   TILTED(90.00)
   wall1 (130.00 BY 12.80)
     WITH WINDOWS OF TYPE
     window1 (128.00 BY 11.80)
       REVEAL(0.00)
       AT (0.60, 0.50);
  PARTITIONS :
   STARTING AT(1.60, 0.00, 0.00)
   FACING(180.00)
   TILTED(90.00)
   PARTITION23 (130.00 BY 12.80),
   STARTING AT(131.60, 0.00, 0.00)
   FACING(90.00)
   TILTED(90.00)
   PARTITION23 (4.90 BY 12.80),
   STARTING AT(1.60, 4.90, 0.00)
   FACING(270.00)
   TILTED(90.00)
   PARTITION23 (4.90 BY 12.80);
  FLOORS :
   STARTING AT(0.00, 0.00, 0.00)
   FACING(90.00)
   TILTED(180.00)
   cfloor1 (4.90 BY 130.00);
  CEILINGS :
   STARTING AT(0.00, 0.00, 12.80)
   FACING(180.00)
   TILTED(0.00)
   cceil1 (130.00 BY 4.90);
  PEOPLE=5,people ,
   AT ACTIVITY LEVEL 0.45, 60.00 PERCENT RADIANT,
   FROM 01JAN THRU 31DEC;
  LIGHTS=2.61,lights ,
    0.00 PERCENT RETURN AIR, 40.00 PERCENT RADIANT,
```

```
    20.00 PERCENT VISIBLE, 0.00 PERCENT REPLACEABLE,
   FROM 01JAN THRU 31DEC;
  ELECTRIC EQUIPMENT=2.17,equipment ,
    30.00 PERCENT RADIANT, 0.00 PERCENT LATENT, 0.00 PERCENT LOST,
   FROM 01JAN THRU 31DEC;
  CROSS MIXING=2374,CONSTANT ,
   FROM ZONE 19, 0.00 DEL TEMP,
   FROM 01JAN THRU 31DEC;
  Infiltration=237.00,CONSTANT,
    WITH COEFFICIENTS (0.606000,0.020000,0.000598,0.000000),
    FROM 01JAN THRU 31DEC;
  CONTROLS=VAV, 75.416 HEATING, 89.728 COOLING,
    0.00 PERCENT MRT,
    FROM 01JAN THRU 31DEC;
 END ZONE;
 ZONE 22 "Zone 22 @ 200 598 0 T":
  ORIGIN:(106.14, 147.70, 0.00);
  NORTH AXIS=0.00;
  EXTERIOR WALLS :
   STARTING AT(14.90, 59.80, 0.00)
   FACING(0.00)
   TILTED(90.00)
   wall1 (14.90 BY 12.80)
    WITH WINDOWS OF TYPE
    window1 (14.00 BY 11.80)
     REVEAL(0.00)
     AT (0.50, 0.50),
   STARTING AT(0.00, 59.80, 0.00)
   FACING(270.00)
   TILTED(90.00)
   wall1 (59.80 BY 12.80)
    WITH WINDOWS OF TYPE
    window1 (59.00 BY 11.80)
     REVEAL(0.00)
     AT (0.60, 0.10);
  PARTITIONS :
   STARTING AT(0.00, 0.00, 0.00)
   FACING(180.00)
   TILTED(90.00)
   PARTITION23 (14.90 BY 12.80),
   STARTING AT(14.90, 0.00, 0.00)
   FACING(90.00)
   TILTED(90.00)
   PARTITION23 (59.80 BY 12.80);
  FLOORS :
   STARTING AT(0.00, 0.00, 0.00)
   FACING(90.00)
   TILTED(180.00)
   cfloor1 (59.80 BY 14.90);
  CEILINGS :
   STARTING AT(0.00, 0.00, 12.80)
   FACING(180.00)
   TILTED(0.00)
   cceil1 (14.90 BY 59.80);
  PEOPLE=5,people ,
   AT ACTIVITY LEVEL 0.45, 60.00 PERCENT RADIANT,
```

```
    FROM 01JAN THRU 31DEC;
   LIGHTS=3.65,lights ,
     0.00 PERCENT RETURN AIR, 40.00 PERCENT RADIANT,
    20.00 PERCENT VISIBLE, 0.00 PERCENT REPLACEABLE,
    FROM 01JAN THRU 31DEC;
   ELECTRIC EQUIPMENT=3.04,equipment ,
    30.00 PERCENT RADIANT, 0.00 PERCENT LATENT, 0.00 PERCENT LOST,
    FROM 01JAN THRU 31DEC;
   CROSS MIXING=2710,CONSTANT ,
    FROM ZONE 19, 0.00 DEL TEMP,
    FROM 01JAN THRU 31DEC;
   Infiltration=271.00,CONSTANT,
     WITH COEFFICIENTS (0.606000,0.020000,0.000598,0.000000),
     FROM 01JAN THRU 31DEC;
   CONTROLS=VAV, 65.203 HEATING, 102.428 COOLING,
     0.00 PERCENT MRT,
     FROM 01JAN THRU 31DEC;
  END ZONE;
 END BUILDING DESCRIPTION;

BEGIN FAN SYSTEM DESCRIPTION;
  VARIABLE VOLUME SYSTEM 1
  "vav 1 " SERVING ZONES
   18,19,20,21,22;
   FOR ZONE 18:
    SUPPLY AIR VOLUME=500;
    EXHAUST AIR VOLUME=0;
    MINIMUM AIR FRACTION=0.4;
    REHEAT CAPACITY=3412000;
    REHEAT ENERGY SUPPLY=HOT WATER;
    BASEBOARD HEAT CAPACITY=0.0;
    BASEBOARD HEAT ENERGY SUPPLY=HOT WATER;
    zone multiplier=1;
   END ZONE;
   FOR ZONE 19:
    SUPPLY AIR VOLUME=2312;
    EXHAUST AIR VOLUME=0;
    MINIMUM AIR FRACTION=0.4;
    REHEAT CAPACITY=3412000;
    REHEAT ENERGY SUPPLY=HOT WATER;
    BASEBOARD HEAT CAPACITY=0.0;
    BASEBOARD HEAT ENERGY SUPPLY=HOT WATER;
    zone multiplier=1;
   END ZONE;
   FOR ZONE 20:
    SUPPLY AIR VOLUME=3628;
    EXHAUST AIR VOLUME=0;
    MINIMUM AIR FRACTION=0.4;
    REHEAT CAPACITY=3412000;
    REHEAT ENERGY SUPPLY=HOT WATER;
    BASEBOARD HEAT CAPACITY=0.0;
    BASEBOARD HEAT ENERGY SUPPLY=HOT WATER;
    zone multiplier=1;
   report variables=(9);
   END ZONE;
   FOR ZONE 21:
```

```
    SUPPLY AIR VOLUME=2374;
    EXHAUST AIR VOLUME=0;
    MINIMUM AIR FRACTION=0.4;
    REHEAT CAPACITY=3412000;
    REHEAT ENERGY SUPPLY=HOT WATER;
    BASEBOARD HEAT CAPACITY=0.0;
    BASEBOARD HEAT ENERGY SUPPLY=HOT WATER;
    zone multiplier=1;
   report variables=(9);
  END ZONE;
  FOR ZONE 22:
    SUPPLY AIR VOLUME=2710;
    EXHAUST AIR VOLUME=0;
    MINIMUM AIR FRACTION=0.4;
    REHEAT CAPACITY=3412000;
    REHEAT ENERGY SUPPLY=HOT WATER;
    BASEBOARD HEAT CAPACITY=0.0;
    BASEBOARD HEAT ENERGY SUPPLY=HOT WATER;
    zone multiplier=1;
    report variables=(9);
  END ZONE;

OTHER SYSTEM PARAMETERS:
    SUPPLY FAN PRESSURE=2.48914;
    SUPPLY FAN EFFICIENCY=0.7;
    RETURN FAN PRESSURE=0.0;
    RETURN FAN EFFICIENCY=0.7;
    EXHAUST FAN PRESSURE=1.00396;
    EXHAUST FAN EFFICIENCY=0.7;
    COLD DECK CONTROL=FIXED SET POINT;
    COLD DECK TEMPERATURE=55.0;
    COLD DECK THROTTLING RANGE=0.0;
    COLD DECK CONTROL SCHEDULE=(57 AT 72, 53 AT 75.2);
    HEATING COIL ENERGY SUPPLY=HOT WATER;
    HEATING COIL CAPACITY=3412000;
    HOT DECK CONTROL=OUTSIDE AIR CONTROLLED;
    HOT DECK THROTTLING RANGE=7.2;
    hot deck control schedule=(120 at 0, 68 at 68);
    MIXED AIR CONTROL=FIXED PERCENT;
    DESIRED MIXED AIR TEMPERATURE=COLD DECK TEMPERATURE;
    OUTSIDE AIR VOLUME=0.0;
    PREHEAT COIL LOCATION=NONE;
    PREHEAT TEMPERATURE=46.4;
    PREHEAT ENERGY SUPPLY=HOT WATER;
    PREHEAT COIL CAPACITY=0;
    GAS BURNER EFFICIENCY=0.8;
    VAV VOLUME CONTROL TYPE=VARIABLE FAN SPEED;
    HUMIDIFIER TYPE=NONE;
    HUMIDISTAT LOCATION=18;
    HUMIDISTAT SET POINT=50;
    SYSTEM ELECTRICAL DEMAND=0.0;
  END OTHER SYSTEM PARAMETERS;
COOLING COIL DESIGN PARAMETERS:
    COIL TYPE=CHILLED WATER;
    AIR VOLUME FLOW RATE=20000;
    BAROMETRIC PRESSURE=406.8136;
```

```
      AIR FACE VELOCITY=490;
      ENTERING AIR DRY BULB TEMPERATURE=80.006;
      ENTERING AIR WET BULB TEMPERATURE=66.992;
      LEAVING AIR DRY BULB TEMPERATURE=60.404;
      LEAVING AIR WET BULB TEMPERATURE=54.0;
      ENTERING WATER TEMPERATURE=44.996;
      LEAVING WATER TEMPERATURE=54.644;
      WATER VOLUME FLOW RATE=0.5348;
      WATER VELOCITY=275;
     END COOLING COIL DESIGN PARAMETERS;
     HEAT RECOVERY PARAMETERS:
      HTREC1(0.85,0.0,0.0);
      HTREC2(0.0,0.0,0.0);
      HTREC3(0.0,0.0,0.0);
      HTREC4(0.0,0.0,0.0);
      HTREC5(0.0,0.0,0.0);
      HTREC6(0.0,0.0,0.0);
      HTPWR(0.0,0.0,0.0);
      HEAT RECOVERY CAPACITY=3412000;
     END HEAT RECOVERY PARAMETERS;
     EQUIPMENT SCHEDULES:
      SYSTEM OPERATION=OFF,FROM 01JAN THRU 31DEC;
      EXHAUST FAN OPERATION=ON,FROM 01JAN THRU 31DEC;
      PREHEAT COIL OPERATION=ON,FROM 01JAN THRU 31DEC;
      HEATING COIL OPERATION=ON,FROM 01JAN THRU 31DEC;
      COOLING COIL OPERATION=ON,FROM 01JAN THRU 31DEC;
      HUMIDIFIER OPERATION=ON,FROM 01JAN THRU 31DEC;
      TSTAT BASEBOARD HEAT OPERATION=ON,FROM 01JAN THRU 31DEC;
      HEAT RECOVERY OPERATION=OFF,FROM 01JAN THRU 31DEC;
      MINIMUM VENTILATION SCHEDULE=MINOA,FROM 01JAN THRU 31DEC;
      MAXIMUM VENTILATION SCHEDULE=MAXOA,FROM 01JAN THRU 31DEC;
      SYSTEM ELECTRICAL DEMAND SCHEDULE=ON,FROM 01JAN THRU 31DEC;
      VAV MINIMUM AIR FRACTION SCHEDULE=VAV MIN FRAC,FROM 01JAN THRU
31DEC;
     END EQUIPMENT SCHEDULES;
    END SYSTEM;
  END FAN SYSTEM DESCRIPTION;

  BEGIN CENTRAL PLANT DESCRIPTION;
    PLANT 1"PURCHASE_COOL" SERVING ALL SYSTEMS;
    EQUIPMENT SELECTION:
     PURCHASED cooling:
      1 OF SIZE 100000;
     END EQUIPMENT SELECTION;
     SCHEDULE:
      PLANT ELECTRICAL DEMAND=0.0,CONSTANT,FROM 01JAN THRU 31DEC;
      PROCESS WASTE HEAT=0.0, CONSTANT, FROM 01JAN THRU 31DEC, AT
     LEVEL 5;
     END SCHEDULE;
     FOR SYSTEM 1:
     SYSTEM MULTIPLIER=1;
     END SYSTEM;
     END PLANT;
  END CENTRAL PLANT DESCRIPTION;
 END INPUT;
```

(2) Part of the Blast Input File for the Large Office Building in Oklahoma with Setback Controlled Schedule.

The input file is the same as the one for the large office building in Oklahoma with the on schedule except the following:

```
TEMPORARY CONTROLS (VAV):
    PROFILES:
      VAV=(1.0000 AT 55.00, 0.809524 AT 67.00, 0.0000 AT 69.3182,
        -0.171429 AT 70.0,-0.194286 AT 72.00,-0.6 AT 76,-1.0 AT 90);
      setback=(1.0000 AT 54.00, 0.0000 AT 55.00, 0.00 at 99, -1.00 at 100);

    SCHEDULES:
      MONDAY THRU FRIDAY=(0 TO 7-setback, 7 TO 17-VAV, TO 24-setback),
      SATURDAY=(0 TO 24-setback),
      SUNDAY=(0 TO 24-setback),
      HOLIDAY=(0 TO 24-setback),
      SPECIAL1=(0 TO 24-setback),
      SPECIAL2=(0 TO 24-setback),
      SPECIAL3=(0 TO 24-setback),
      SPECIAL4=(0 TO 24-setback);
   END;
  BEGIN FAN SYSTEM DESCRIPTION;
   VARIABLE VOLUME SYSTEM 1
   "vav 1 " SERVING ZONES
    18,19,20,21,22;
    FOR ZONE 18:
      SUPPLY AIR VOLUME=514;
      EXHAUST AIR VOLUME=0;
      MINIMUM AIR FRACTION=0.4;
      REHEAT CAPACITY=3412000;
      REHEAT ENERGY SUPPLY=HOT WATER;
      BASEBOARD HEAT CAPACITY=0.0;
      BASEBOARD HEAT ENERGY SUPPLY=HOT WATER;
      zone multiplier=1;
    END ZONE;
    FOR ZONE 19:
      SUPPLY AIR VOLUME=2400;
      EXHAUST AIR VOLUME=0;
      MINIMUM AIR FRACTION=0.4;
      REHEAT CAPACITY=3412000;
      REHEAT ENERGY SUPPLY=HOT WATER;
      BASEBOARD HEAT CAPACITY=0.0;
      BASEBOARD HEAT ENERGY SUPPLY=HOT WATER;
      zone multiplier=1;
    END ZONE;
    FOR ZONE 20:
      SUPPLY AIR VOLUME=3706;
      EXHAUST AIR VOLUME=0;
      MINIMUM AIR FRACTION=0.4;
      REHEAT CAPACITY=3412000;
      REHEAT ENERGY SUPPLY=HOT WATER;
      BASEBOARD HEAT CAPACITY=0.0;
      BASEBOARD HEAT ENERGY SUPPLY=HOT WATER;
      zone multiplier=1;
      report variables=(9);
```

```
END ZONE;
FOR ZONE 21:
  SUPPLY AIR VOLUME=2377;
  EXHAUST AIR VOLUME=0;
  MINIMUM AIR FRACTION=0.4;
  REHEAT CAPACITY=3412000;
  REHEAT ENERGY SUPPLY=HOT WATER;
  BASEBOARD HEAT CAPACITY=0.0;
  BASEBOARD HEAT ENERGY SUPPLY=HOT WATER;
  zone multiplier=1;
  report variables=(9);
END ZONE;
FOR ZONE 22:
  SUPPLY AIR VOLUME=2741;
  EXHAUST AIR VOLUME=0;
  MINIMUM AIR FRACTION=0.4;
  REHEAT CAPACITY=3412000;
  REHEAT ENERGY SUPPLY=HOT WATER;
  BASEBOARD HEAT CAPACITY=0.0;
  BASEBOARD HEAT ENERGY SUPPLY=HOT WATER;
  zone multiplier=1;
  report variables=(9);
END ZONE;
```

(3) Part of Blast Input File for the Large Office Building in Minnesota with On Schedule.

The input file is the same as the one for the large office building in Oklahoma with the on schedule except the following:

```
TEMPORARY CONTROLS (VAV):
    PROFILES:
      VAV=(1.0000 AT 55.00, 0.91915 AT 67.00, 0.0 AT 69.70282,
        -0.171429 AT 70.00, -0.194286 at 72.00, -0.6 AT 76.00, -1.0 AT 90);
      SCHEDULES:
          MONDAY THRU FRIDAY=(0 TO 24-VAV),
          SATURDAY=(0 TO 24-VAV),
          SUNDAY=(0 TO 24-VAV),
          HOLIDAY=(0 TO 24-VAV),
          SPECIAL1=(0 TO 24-VAV),
          SPECIAL2=(0 TO 24-VAV),
          SPECIAL3=(0 TO 24-VAV),
          SPECIAL4=(0 TO 24-VAV);
    END;

BEGIN FAN SYSTEM DESCRIPTION;
   VARIABLE VOLUME SYSTEM 1
   "vav 1 " SERVING ZONES
    18,19,20,21,22;
    FOR ZONE 18:
      SUPPLY AIR VOLUME=498;
      EXHAUST AIR VOLUME=0;
      MINIMUM AIR FRACTION=0.4;
      REHEAT CAPACITY=3412000;
      REHEAT ENERGY SUPPLY=STEAM;
      BASEBOARD HEAT CAPACITY=0.0;
```

```
  BASEBOARD HEAT ENERGY SUPPLY=STEAM;
  zone multiplier=1;
END ZONE;
FOR ZONE 19:
  SUPPLY AIR VOLUME=2303;
  EXHAUST AIR VOLUME=0;
  MINIMUM AIR FRACTION=0.4;
  REHEAT CAPACITY=3412000;
  REHEAT ENERGY SUPPLY=STEAM;
  BASEBOARD HEAT CAPACITY=0.0;
  BASEBOARD HEAT ENERGY SUPPLY=STEAM;
  zone multiplier=1;
END ZONE;
FOR ZONE 20:
  SUPPLY AIR VOLUME=3304;
  EXHAUST AIR VOLUME=0;
  MINIMUM AIR FRACTION=0.4;
  REHEAT CAPACITY=3412000;
  REHEAT ENERGY SUPPLY=STEAM;
  BASEBOARD HEAT CAPACITY=0.0;
  BASEBOARD HEAT ENERGY SUPPLY=STEAM;
  zone multiplier=1;
  report variables=(9);
END ZONE;
FOR ZONE 21:
  SUPPLY AIR VOLUME=1853;
  EXHAUST AIR VOLUME=0;
  MINIMUM AIR FRACTION=0.4;
  REHEAT CAPACITY=3412000;
  REHEAT ENERGY SUPPLY=STEAM;
  BASEBOARD HEAT CAPACITY=0.0;
  BASEBOARD HEAT ENERGY SUPPLY=STEAM;
  zone multiplier=1;
  report variables=(9);
END ZONE;
FOR ZONE 22:
  SUPPLY AIR VOLUME=2504;
  EXHAUST AIR VOLUME=0;
  MINIMUM AIR FRACTION=0.4;
  REHEAT CAPACITY=3412000;
  REHEAT ENERGY SUPPLY=STEAM;
  BASEBOARD HEAT CAPACITY=0.0;
  BASEBOARD HEAT ENERGY SUPPLY=STEAM;
  zone multiplier=1;
  report variables=(9);
END ZONE;
```

(4) Part of Blast Input File for the Large Office Building in Minnesota with Setback Controlled Schedule.

The input file is the same as the one for the large office building in Oklahoma with the on schedule except the following:

```
TEMPORARY CONTROLS (VAV):
    PROFILES:
     VAV=(1.0000 AT 55.00, 0.91486 AT 67.00, 0.0 AT 69.6874,
        -0.17143 AT 70.00, -0.19429 at 72.00, -0.6 AT 76.00, -1.0 AT 90);
     setback=(1.0000 AT 54.00, 0.0000 AT 55.00, 0.00 at 99, -1.00 at 100);
    SCHEDULES:
     MONDAY THRU FRIDAY=(0 TO 7-setback, 7 TO 17-VAV, 17 TO 24-setback),
     SATURDAY=(0 TO 24-setback),
     SUNDAY=(0 TO 24-setback),
     HOLIDAY=(0 TO 24-setback),
     SPECIAL1=(0 TO 24-setback),
     SPECIAL2=(0 TO 24-setback),
     SPECIAL3=(0 TO 24-setback),
     SPECIAL4=(0 TO 24-setback);
   END;

  BEGIN FAN SYSTEM DESCRIPTION;
   VARIABLE VOLUME SYSTEM 1
   "vav 1 " SERVING ZONES
     18,19,20,21,22;
    FOR ZONE 18:
     SUPPLY AIR VOLUME=500;
     EXHAUST AIR VOLUME=0;
     MINIMUM AIR FRACTION=0.4;
     REHEAT CAPACITY=3412000;
     REHEAT ENERGY SUPPLY=STEAM;
     BASEBOARD HEAT CAPACITY=0.0;
     BASEBOARD HEAT ENERGY SUPPLY=STEAM;
     zone multiplier=1;
    END ZONE;
    FOR ZONE 19:
     SUPPLY AIR VOLUME=2321;
     EXHAUST AIR VOLUME=0;
     MINIMUM AIR FRACTION=0.4;
     REHEAT CAPACITY=3412000;
     REHEAT ENERGY SUPPLY=STEAM;
     BASEBOARD HEAT CAPACITY=0.0;
     BASEBOARD HEAT ENERGY SUPPLY=STEAM;
     zone multiplier=1;
    END ZONE;
    FOR ZONE 20:
     SUPPLY AIR VOLUME=3338;
     EXHAUST AIR VOLUME=0;
     MINIMUM AIR FRACTION=0.4;
     REHEAT CAPACITY=3412000;
     REHEAT ENERGY SUPPLY=STEAM;
     BASEBOARD HEAT CAPACITY=0.0;
     BASEBOARD HEAT ENERGY SUPPLY=STEAM;
     zone multiplier=1;
     report variables=(9);
```

```
END ZONE;
FOR ZONE 21:
 SUPPLY AIR VOLUME=1758;
 EXHAUST AIR VOLUME=0;
 MINIMUM AIR FRACTION=0.4;
 REHEAT CAPACITY=3412000;
 REHEAT ENERGY SUPPLY=STEAM;
 BASEBOARD HEAT CAPACITY=0.0;
 BASEBOARD HEAT ENERGY SUPPLY=STEAM;
 zone multiplier=1;
 report variables=(9);
END ZONE;
FOR ZONE 22:
 SUPPLY AIR VOLUME=2507;
 EXHAUST AIR VOLUME=0;
 MINIMUM AIR FRACTION=0.4;
 REHEAT CAPACITY=3412000;
 REHEAT ENERGY SUPPLY=STEAM;
 BASEBOARD HEAT CAPACITY=0.0;
 BASEBOARD HEAT ENERGY SUPPLY=STEAM;
 zone multiplier=1;
 report variables=(9);
END ZONE;
```

APPENDIX D--Institutional Review Board Approval

Oklahoma State University
Institutional Review Board

Protocol Expires: 5/31/01

Date : Wednesday, May 31, 2000 IRB Application No: EG004

Proposal Title: OPTIMUM DUCT DESIGN FOR VARIABLE AIR VOLUME SYSTEMS

Principal
Investigator(s) :

R.D. Delahoussaye
011 Engineering North
Stillwater, OK 74078

Taecheol Kim
011 Engineering North
Stillwater, OK 74078

J.D. Spitler
218 Engineering North
Stillwater, OK 74078

Reviewed and
Processed as: Exempt

Approval Status Recommended by Reviewer(s) : Approved

Signature :

Carol Olson by [illegible]

Carol Olson, Director of University Research Compliance

Wednesday, May 31, 2000
Date

Approvals are valid for one calendar year, after which time a request for continuation must be submitted. Any modifications to the research project approved by the IRB must be submitted for approval with the advisor's signature. The IRB office MUST be notified in writing when a project is complete. Approved projects are subject to monitoring by the IRB. Expedited and exempt projects may be reviewed by the full Institutional Review Board.

VITA

Taecheol Kim

Candidate for the Degree of

Doctor of Philosophy

Thesis: OPTIMUM DUCT DESIGN FOR VARIABLE AIR VOLUME SYSTEMS

Major Field: Mechanical Engineering

Biographical:

Personal Data: Born in Seoul, Korea, On November 4, 1959, the son of Jae-Sung Kim and Jong-Ye Han.

Education: Graduated from Han-Yung High School, Seoul, Korea in December 1977; received Bachelor of Science degree in Mechanical Engineering from Sung-Kyun-Kwan University, Seoul, Korea in December 1984; received Master of Science degree in Mechanical Engineering from Oklahoma State University, Stillwater, Oklahoma in December 1990. Completed the requirements for the degree of Doctor of Philosophy with a major in Mechanical Engineering from Oklahoma State University in May 2001.

Experience: Employed as a mechanical designer by Kolon Engineering Inc., Seoul Korea, 1985 to 1987; employed as a teaching and research assistant by Oklahoma State University, Department of Mechanical Engineering, 1989 to present.