Operating Strategies for Economizer Control in an Air Conditioning System

Feng Zhao

A Thesis
in
The department
of
Mechanical Engineering

Presented in Partial Fulfilment of the Requirements
for the Degree of Master of Applied Science at
Concordia University
Montreal, Quebec,Canada

December 1998

0-612-39108-6

ABSTRACT

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A typical economizer in a air conditioning system is investigated in this paper. New operating strategies of the economizer are proposed. An economizer consists of three coupled dampers - an outdoor air, a recirculation, and a discharge damper - controlled so that the amount of outdoor air may be varied from a minimum to 100% of the supply air. Traditionally the three dampers are coupled. In a new strategies only the outdoor and recirculation dampers are coupled and the discharge damper is always fully open during operation. An additional strategy involves linearizing the damper area as a function of the control signal by means of a software algorithm. Fan energy requirements for the new strategies are compared to those for the traditional strategy. The amount of fresh air may be controlled by heating and cooling considerations or by indoor air quality considerations. Carbon dioxide (CO_2) concentration is an indication of indoor air quality. The objectives of the investigation were to determine the best strategy to minimize fan energy consumption, and to control ventilation by demand as indicated by room CO_2 concentration. Numerical system models were constructed to analyze system performance. An experimental system was built and tested to verify the models and simulations proposed in this paper.

A numerical model of a system with an economizer for a single room ventilation was developed for the analysis and simulation system performance. The numerical system

model included damper models and fan models. A simulation program based on this model was written in FORTRAN code. Various control strategies were able to be simulated by the program. The results show that two coupled damper system were more economic than traditional three coupled damper systems. Linearizing the damper area as a function of the control signal by means of a software algorithm is shown to reduce energy consumption and to improve the system performance. The model also predicts that two coupled damper systems with ideal linear damper characteristics would be the most economic control strategy for economizer control. Experimental investigations indicated that the model could represent the system performance quite accurately. The simulated and experimental results indicate that a two coupled damper system with a software algorithm for area linearization may save up to 10% fan energy compared to a traditional three coupled damper system.

The two coupled damper control systems with a software algorithm for area linearization was applied to demand control ventilation using feedback control for the economizer. A numerical model of the feedback controlled system based on the system model was constructed to simulate the indoor CO_2 concentration control process. The simulation results indicated that the two dampers system with area compensation is suitable for the CO_2 demand control ventilation. The system showed satisfactory performance with the feedback controller when the indoor CO_2 generation rate was varied over a large range. The experimental results substantiated the results predicted by the numerical models.

ACKNOWLEDGMENTS

The author is greatly indebted to his supervisor Professor K.I.Krakow for his valuable guidance and support throughout the research and preparation of this thesis.

The financial support provided by the Natural Sciences and Engineering Research Council of Canada (grant number OGP0001656) is gratefully acknowledged.

Finally, the author wishes to thank his wife and family for their encouragement and understanding throughout his university years.

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NOMENCLATURE

A	duct section area
C	CO_2 concentration
C_{SP}	CO_2 concentration set point
Cv	damper resistance coefficient
c_p	specific heat at constant pressure of the air
D	duct equivalent diameter
E	total energy consumption
f	friction coefficient
K_p	proportional coefficient
K_i	integral coefficient
L	equivalent length of the damper
met	physical activity
N	fan speed
n	number of the persons in the room
P	pressure
ΔP	pressure differential across the fan
Q	air flow rate
Q_g	occupation CO_2 generation rate
q_g	CO_2 generation rate per person
T	temperature

t	Time
V	air velocity
V_{Rm}	room volume
v	specify volume of air
W	fan power
ρ	air density
σ	sensitivity

Subscripts

Oda	outdoor air
Out	outside
Rm	room
Rtn	return air
Sup	supply air

1. INTRODUCTION

The general function of a heating, ventilating and air conditioning (HVAC) system is to make an indoor environment comfortable through control of temperature, moisture content, cleanliness, odor and air circulation, as required by occupants. Proper control of a HVAC system may often be a difficult problem for engineers, because the HVAC systems appear to be complex, cover large areas, use sophisticated equipment, and involve a several of manipulated control variables. In recent years, energy shortages have become more serious requiring HVAC systems to operate more efficiently. This gives engineers a new challenge.

The general objective of the current investigation is to analyze ways to reduce energy consumption in HVAC systems. In this paper, the basic types of system are discussed and their energy consumptions are analyzed by both theoretical and experimental methods.

1.1 Basic Systems

There are two basic types of HVAC systems used in large buildings, constant volume variable temperature (CVVT) systems and variable air volume (VAV) systems. In a CVVT system, the flow rate of conditioned air supplied to each room (the conditioned space) is constant, but its temperature is varied by heating coil and cooling coils to balance the room heating and cooling loads. In a VAV system, the temperature of the conditioned air supplied is constant, but its flow rate to each room is varied to balance the room heating and cooling loads. Warm air is supplied during the heating season, and chilled air is supplied during the cooling season.

A CVVT system has a cooling and heating coil for each room. The cooling coils are chilled water coils. A typical CVVT system is shown in Figure 1.1.

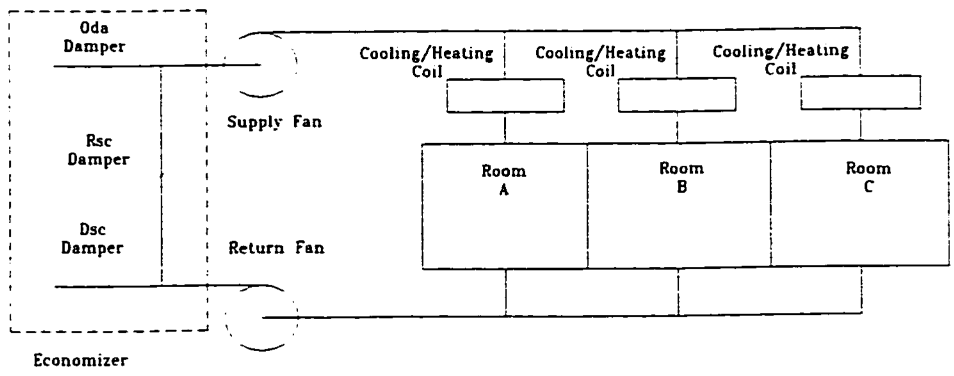

Figure 1.1: A schematic diagram of a CVVT system.

A typical VAV system has a single cooling coil and a single heating coil for all rooms. The cooling coil is a direct expansion coil. A typical VAV system is shown in Figure 1.2

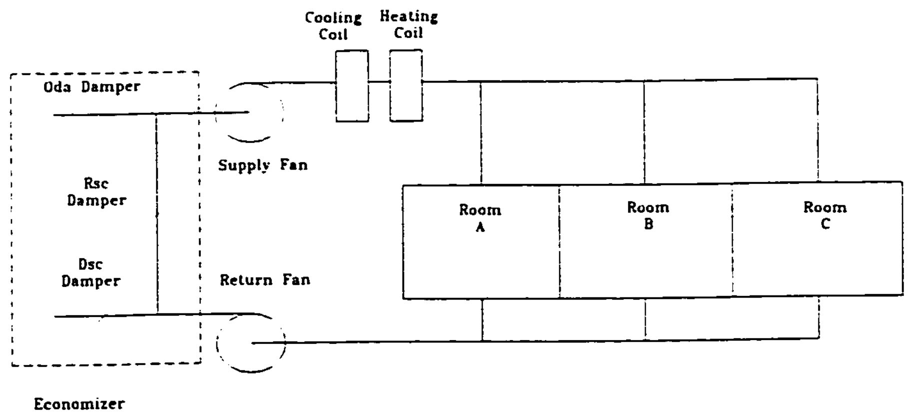

Figure 1.2: A schematic diagram of a VAV system.

Both CVVT and VAV systems have a supply fan and a return fan. In a CVVT system, both fans are constant speed. In a VAV system, both fans are variable speed. The supply fan is to provide a specified static pressure in the supply air duct. This specified static pressure is that needed in the longest duct run. For a variable speed supply fan, the return fan must also have variable speed. The return fan is to maintain indoor room pressure at a desired level, usually slightly above atmospheric. It is desired to maintain the indoor pressure slightly higher than the outdoor pressure to eliminate infiltration. However, the indoor pressure should not be so high that excessive force is required to open doors. The return fan flow rate must be proportional to the supply fan flow rate. Alternatively, the flow rate may be controlled by the supply fan flow rate. In small systems the return fan is optional.

VAV systems are more commonly used in the practice because of their lower capital and installation costs as well as the lower energy consumption than CVVT systems. However, CVVT systems yield better temperature control for individual room. In large HVAC systems, both VAV and CVVT system, the outdoor air flow rate can be controlled by an arrangement of dampers known as an economizer (sometimes called an economy cycle). The economizer consists of three coupled dampers: an outdoor air (Oda) damper, a recirculation air (Rec) damper, and a discharge air (Dsc) damper. The dampers are coupled by means of a control system. As the damper positions change, the mixing ratio of outdoor and recirculated air change. Air flow rates may also change. Traditionally, the three dampers have same size and configuration.

1.2 Outdoor Air Requirements

The amount of outdoor air required is determined by temperature considerations and by indoor air quality considerations. Heating and cooling loads are due to transmission through the exterior envelope (walls, windows, and roof) and due to generation by internal sources. People generate heat, moisture, and carbon dioxide. Outdoor air accounts for a significant proportion of the heating load and the sensible and latent cooling loads. The amount of outside air introduced into the room also must provide the ventilation necessary to maintain acceptable indoor air quality. Inadequate ventilation may cause sick building syndrome.

In operating the economizer, the dampers are controlled by the air enthalpy (or temperature) consideration. During cooling season, if the enthalpy of the return air is less than that of the outdoor air, the outdoor air damper is set at a minimum position and consequently the outdoor air is at minimum flow rate. If the enthalpy of the outdoor air is less than that of the return air, 100% outdoor air is used. If the temperature of the outdoor air is less than the required supply air temperature, the ratio of outdoor air to recirculation air is varied by the controlling the position of the dampers in the economizer. Therefore, the economizer may decrease the system cooling load. During heating season, the outdoor air damper is always set at the minimum position and consequently the outdoor air flow is at the minimum flow rate.

It is always necessary and important to provide the required minimum outside air flow under all conditions of the system. This minimum ventilation volume is determined by the consideration of the indoor air quality. In some systems, the amount of minimum ventilation volume is also determined by the controlled environment pressure consideration of keeping a positive internal pressure to prevent infiltration from the surrounding areas or a negative pressure to prevent the exfiltration.

The simplest method is to set a minimum position for outside air damper to ensure the outdoor air is greater than the minimum required fresh air at all times. This method will work well with CVVT systems. In CVVT systems, once the minimum outdoor air damper position is properly set, the necessary quantity of outside air would always be maintained regardless of the conditions under which the system is operating because the supply fan always provides the same amount of air. In a VAV system, a design requirement is that the system provide the minimum of outdoor air when the outside air damper is operating at the minimum position and the system is operating at part load. If a VAV system is not properly designed, the outdoor air flow may be lower than the minimum required ventilation rate when the outdoor air damper is at the minimum position and the demand for supply air is low. Therefore accurately controlling the minimum amount of the outdoor air will be more difficult in VAV system than in a CVVT system.

The minimum outdoor air flow rate requirement may be a fixed quantity based on the maximum anticipated occupancy. Alternatively, the minimum outdoor air may be controlled by the actual occupancy as determined by the CO_2 concentration. This alternate method is knows as demand control ventilation. Demand control ventilation can be used in both VAV and CVVT systems. An ASHRAE standard recommends that the minimum ventilation outdoor air flow be controlled based on a measure of the indoor air quality. In many situations, the occupant-generated CO_2 can serve as a suitable surrogate measure of indoor air quality.Whereas the outdoor air flow rate varies according to the actual occupancy, less outdoor air would be required by using demand control ventilation, thereby reducing energy requirement.

In the current investigation, the demand control was used to determine the outdoor air flow. If the CO_2 concentration level of the return air is greater than a specified acceptable level (approximately 1000 ppm CO_2 used in this paper), the ratio of outdoor air (approximately 350 ppm CO_2 used in this paper) is increased. Using the demand control, the outdoor air flow can be varied from a very low volume when the number of the occupants are reduced after peak occupancies office hours to 100% outdoor air for cooling purpose as determined by enthalpy considerations. A damper control system should be able to handle a wide range of the outdoor air flow rate with a good performance at any range. A computer is generally used with current control techniques. Digital controllers have become more accurate, flexible and reliable than analog controllers. The dampers, the main component of the control system in the economizer, are controlled with digital computers programmed to implement a feedback control algorithm. The damper system characteristics will directly affect the performance of the entire control system as it attempts to accurately maintain the required flow rate of outdoor air. Properly sizing and selecting dampers is an essential factor in designing a good air handling control system. Properly sizing and selecting dampers could minimize the interaction between the economizer control loop and the fan pressure control loops and reduced the fan power consumption.

An air ventilation system must maintain adequate ventilation for indoor air quality as well as control space static pressure. The economizer will also vary the outdoor air flow for temperature control purpose. Therefore, fan energy requirements depend on the control strategy of the economizer and the control algorithm.

1.3 Objectives

The objectives of the current investigation are:

- to investigate various control strategies and algorithms for energy reduction,
- to validate the control strategies and algorithms for demand control ventilation by using feedback control,
- to develop, code, and validate a numerical model of an economizer system, and
- to develop, code, and validate a numerical model of demand control ventilation.

The following control strategies are investigated:

- a three coupled damper economizer system,
- a two coupled damper economizer system (discharge damper remains at 100% open), and
- a two coupled damper economizer system with software linearized damper area versus control signal characteristics.

This last control strategy will be referred to as the "area compensated" strategy.

2. LITERATURE SURVEY

An economizer of HAVC system is not only responsible for saving the operating energy but also for controlling indoor air quality. Many papers related to this area concern proper damper selection and control strategies required to insure good indoor air quality and minimum energy consumption. A literature survey was performed to assess the state of the economizer design in recent years. The relevant topics are:

- sizing and selecting dampers,
- strategies for controlling dampers,
- minimum ventilation control for indoor air quality (IAQ), and
- proportional integral (PI) feedback control systems.

2.1 Sizing and Selecting Dampers

There are three dampers in an economizer system. They are outdoor air damper, recirculation air damper and discharge air damper. Traditionally, they normally are all the same size and are coupled.

As the description in the ASHRAE Fundamentals Handbook [1], there are two different types of damper commonly used in HVAC system. They are opposed blade dampers and parallel blade dampers. In an opposed blade damper, adjacent blades rotate in opposite directions. In a parallel blade damper, all blades rotate in the same directions. The opposed blade damper does not change the flow direction. The parallel blade damper changes the flow direction as the blades close. These two types of dampers have different inherent flow characteristics. The opposed blade damper give a very slow increase in the flow when the damper begins to open. Compared to the opposed blade, the parallel blade dampers have a rapidly increase in the flow when the damper begin to open.

As defined in *The Belimo Damper Application Guide* [2], a damper installed flow characteristic is also depended on "damper authority" or "characteristic ratio". The damper authority is defined as a percentage based on the flow resistance of the open damper relative to the total system flow resistance.

The flow resistance of a damper used to control a flow rate should be a significant proportion of the total system flow resistance for the proper control. If the flow resistance of the damper is too low, it will control only when it is nearly closed. Therefore, a damper generally should have a linear installed characteristic in an air handling system. If it is oversized it will have a small authority and its control characteristic will become on-off or two position control in nature. To obtain a near linear installed characteristic, the damper authority should be selected so that it has a close approximation to a linear relationship between the flow rate ratio and the blade angle.

R.L. Alley [3] did a series of tests for coupled dampers. For parallel blade dampers he found that when both the outdoor and recirculation air dampers (dampers of equal size are assumed) are at the mid-point, the total open area is 120% to 140% of what it would be of one damper is wide open and the other is fully closed. For opposed dampers, he found that the open area is only 40% to 60% of what it would be of one damper is wide open and the other is fully closed. He also did a series of similar air flow performance tests for coupled dampers with a combination of opposed and parallel blades. He found the coupled dampers with a combination of opposed and parallel blades can be designed to have a constant total flow performance with equal pressure drop across both dampers, regardless of the coupled dampers' blade position.

G. Avery [4] did tests similar to those by Alley [3]. He concluded that a damper with a combination of opposed and parallel blades would be the best choice to achieve the optimum performance curve of coupled outdoor and recirculation air damper system.

D.K.Dickson [5] also indicated in his paper that selecting the appropriate damper with its proper size can ensure comfort at the lowest possible energy cost. He indicated that an improper choice will lead to poor ventilation control and pressure fluctuations in the mixed air section which, in turn, will cause variations in the total volume of air supplied by the fan. If a proper damper sizing is not possible in a design, or if an existing system contains oversized dampers, the linkage adjustments outlined in the paper will help achieve the same objectives.

Krakow [6] presented a method to linearize damper blade opening area as a function of the control signal. The control signal to a damper may be compensated by software so that the area is a linear function of the signal. The equations for signal compensation are based on thin, flat plate, non-overlapping blades. The combined flow coefficient for coupled outdoor air and recirculation air dampers may be increased if the signals to dampers are compensated. A higher flow coefficient indicated a lower pressure loss. The area compensation method will also reduce the fan power requirement.

Robert Van Becelaere [7] tested dampers' performance in a manufactured mixing box application which covered three typically shaped blade dampers with five different configurations. He concluded that parallel blade dampers with parallel air flow configuration is the best arrangement from the air mixing standpoint. A 3.5 inch airfoil blade had the most constant static pressure for all flow rates. In all cases, the opposed blade dampers had substantially higher pressure increases at the midpoint blade position than opposed blade dampers.

2.2 Control of Damper Strategies

The Belimo Damper Application Guide [2] states that in a system, the dampers are usually operated in unison; so that as the outdoor air and discharge air dampers close, the recirculation air damper opens. The dampers should be sized so that they compliment each other, i.e., an increase in the outdoor air flow is matched by an equal decrease in the recirculation air flow. Ideally the combined supply flow should remain constant, regardless of the outdoor/recirculation damper mixing ratio.

J.E Seem, J.M.House and C.J. Klaassen [8] demonstrated that air could enter an air handling unit through the exhaust air outlet of a traditional volume matching control system. The traditional control system links the position of the outdoor air damper, exhaust air damper, and recirculation air damper. In their article, they used a new control system for an air handling unit that used a volume matching control strategy to control the return fan. During occupied times, the outdoor air damper was fully open, and the position of the exhaust and recirculation dampers were linked. They indicated that this method could prevent air from entering the air handling unit though the exhaust air outlet.

2.3 Minimum Ventilation Control

ASHRAE Standard 62-1989 [9] establishes criteria for determining the amount of outside air ventilation necessary to achieve acceptable indoor air quality. That amount is the minimum outside air quality. With a constant volume system, it is easy to maintain the minimum amount of outdoor air whenever the outdoor air damper is in the minimum position because the supply air flow rate is always constant. A variable air volume (VAV) system presents the design engineer with the problem of ensuring that the minimum amount of outdoor air is introduced into the building at all air flow rates. When a VAV system is operating at its maximum capacity and the outdoor damper is in minimum position, the amount of the outdoor air will be adequate. However when a VAV system is operating at its minimum capacity and the outdoor damper is in minimum position, the amount of the outdoor air not be inadequate.

J.Levenhagen [10] and D.C.Hittle [11] presented a way to ensure the minimum ventilation to use a fan tracking system, called the "volumetric tracking" system. The volumetric fan tracking system refers to the method that attempts to measure the supply air volume and resets the return fan volume to track the supply flow with a fixed differential. The fixed differential is the minimum ventilation volume that the supply fan draws from the outdoor air damper. The key components are the air flow measuring stations for the supply air and return air. They measure the velocity of the air flow, which can be converted into air volume. Some practitioners raise the issue of the inherent inaccuracy of using the difference between two large measurements as the measure of a small quantity. Air flow measurement may have an extremely low error of approximately ±5%. If the system is trying to provide 60,000 cfm of the supply air with 45,000 cfm return air to achieve the minimum outdoor air volume of 15,000 cfm and if the system takes account the minus 5% error, the minimum different volume of the supply and return air flow can be reduced to 9,750 cfm. This is 35% below the required minimum of the 15,000cfm. Therefore the fan tracking systems ignores the realities of the control accuracy. Combining control inaccuracy with a system that measure two indirect variables to directly control a third is unacceptable.

D.M. Elovitz [12] presents another method to control the minimum outdoor air flow. The most obvious approach to maintaining a constant amount of the outdoor air ventilation would be to measure the amount of outdoor air being brought into the system and control the outdoor air quantity directly. Although this method works fine in theory, it, too, fails in practice. To accurately measure outdoor air, the system must first overcome the turbulence from air entering and changing directions through the outdoor air louvers. Multiple-point, pitot-tube averaging probes works well with VAV boxes; but it must be remembered that the inlet velocity to VAV boxes at design flow is low. To achieve a high level of velocity pressure at an outdoor air measuring station, the outdoor air damper or flow station must be smaller to increase the outdoor velocity to a controllable level. The velocity must then be reduced immediately beyond the measuring station because the velocity through a filter section cannot exceed 500 fpm. The air pressure dropping through this restricted section must increase the supply fan energy. An electronic hot-wire sensor is suitable to measure the low velocity air flow. However, it cannot get a steady and accurate reading even if it is temperature-compensated. Even though the direct measurement of the outdoor air quantity is the only control approach that deals directly with how much air is being brought in from outdoors, it is still inaccurate.

Mumma and Wong [13] first proposed a plenum pressure control method to control the minimum outdoor air flow when the outdoor air damper position is fixed. If the minimum position of the outside air damper is set, the flow through these dampers will be proportional to the square root of the pressure drop across the damper. As the supply fan air flow is reduced, the differential pressure drops. The control system holds the outdoor air damper at a fixed position and modulates the recirculation damper toward the closed position to maintain a constant pressure across the outdoor air damper so that the required flow though the outdoor air damper will remain constant. This ensures that the minimum ventilation air volume is always supplied. The advantages of the method are that it provides the readable pressure drop signal, can control the minimum outdoor air flow very well, can be used in the new and existing system with or without return fans, and its controls are very simple. There is some energy penalty involved because the drop in pressure across the outdoor air damper has to be large enough to be measured and controlled with accuracy. If the drop in pressure is not enough, it is impossible to control it effectively.

John P. Kettler [14] presented an injection fan method to maintain the minimum outdoor air supply. The control system using a separate injection fan, which is parallel with the outdoor air duct, ensures that the minimum outdoor air is supplied. Its ductwork is sized so that velocity pressures are easily readable. This method overcomes disadvantages of the outdoor air measuring station and fan-tracking methods. Because it uses a parallel air flow path, it controls the minimum outdoor air flow directly and a small control error will result in only a small air flow error. The injection fan method improves minimum air flow control and eliminates the energy penalty imposed on the supply fan. Unfortunately, this method may require extra ductwork on new jobs, and can be difficult to retrofit to existing system.

G.J. Janu, J.D. Wenger and C.G. Nesler[15] propose a new method in their paper. The demand-controlled ventilation based on the a tracer gas concentration balance. They use carbon dioxide (CO_2) as a tracer gas. As an alternative to prescribed ventilation rates, ASHRAE standard 62-1989 [9] allows ventilation air to be controlled based on a measure of indoor air quality. In many situations, occupant-generated CO_2 can serve as a suitable surrogate measure of indoor air quality. Their article focuses on central VAV system controls and illustrates the multiple requirements that need to be considered in implementing the controls for outdoor air flow. The CO_2-based ventilation control affords a means to reduce energy consumption while maintaining the satisfactory indoor air quality. It is easy to implement as a retrofit on existing system.

2.4 PI Control

Krakow et al. [17, 18, 19, 20, 21] applied proportional integral (PI) control systems into the HVAC systems. They showed that PI control can be used in many thermo-fuild control systems. They developed a novel theory for determining coefficients for the proportional integral (PI) control of heating, ventilating, air-conditioning, and refrigeration system from the perspective of thermo-fuild engineering, not the perspective of control system engineering. The theory focused on the digital systems that use relatively long sampling intervals. The theory is applicable to systems with and without storage, i.e., second and first order systems. The two coefficients PI control system can be evaluated using a system property defined as "process sensitivity" and a sampling interval. The process sensitivity is based on open loop system operating characteristic.

3. THE SYSTEM MODEL

A typical air distribution system is discussed in this paper. The air distribution system investigated consists of a supply fan, a return fan, an economizer, and a conditioned space as shown in Figure 3.1.

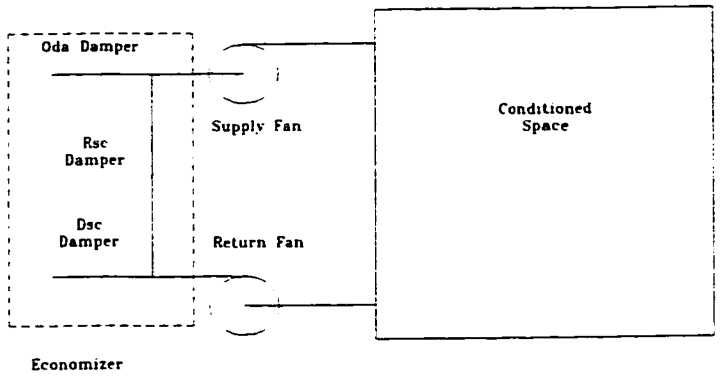

Figure 3.1: A typical air distribution system.

The economizer consist of three coupled dampers: an outdoor air (Oda) damper, a recirculation (Rec) damper, and a discharge (Dsc) damper. The purpose of the economizer is to control the ratio of the outdoor air flow rate to the recirculated air flow rate. The three dampers are coupled electronically, i.e., they were controlled by related control signals. A PI controller is commonly used to control the three coupled dampers. The control point may be the temperature of supply duct air flow, the temperature of the conditioned space, or the carbon dioxide (CO_2) concentration of a conditioned space. When the amount of fresh air introduced into a conditioned space is controlled by the CO_2 concentration, the system is operating in the demand control mode. The three dampers and the PI controller constitute a closed loop control system.

The supply fan and return fan in the air circulation system provide the necessary air flow rate to a conditioned space and also maintained the conditioned space at a constant pressure level. The fans' speed may be varied by means of a variable frequency drive. The speed is generally controlled by a PI controller to maintain some fixed duct air flow or duct static pressure at control point.The fans and PI controller with necessary sensors constitute additional closed loop control systems. The variable frequency drive is the most efficient method to achieve variation of a fan speed. The device maintains motor efficiency regardless of the motor speed or power output and the motor does not lose its power factor. Therefore, the method is commonly used in HVAC system in recent year. It enables reduction of fan energy consumption when the fan operates below the design fan speed.

As previously mentioned, the three dampers in the economizer may be controlled under temperature consideration or under CO_2 considerations. When controlled under CO_2 considerations the system is considered to operate in a demand control mode. In the economizer, the damper blades would open from a minimum position to a full open position. The minimum position of the dampers is variable if the demand control mode is in operation. Therefore, the variable range of damper blade may be from a preset minimum to a preset maximum. In an actual damper, when the damper blades are operating at small opening positions the air flow can not be controlled properly because flow rate versus control signal may not be a single valued function due to the blade overlap. Similarly, when the damper blades are operating at near wide open positions, the air flow rate will not change with control signal variation due to the damper blade profile. Considering that damper performance at small opening positions and near wide open positions is poorly controllable, the damper opening range was limited between the 20% to 85% opening in this study.

When the damper blade position was modulated by the damper controller, the air flow in the duct and duct static pressure would change correspondingly. As a result, this change would affect the fan control loops and the fan speeds would also be adjusted by their controllers. Conversely, the fan control loops may also affect the damper loop. These loops were interactive. Therefore, the fan energy consumption would depend on the control strategy of the economizer and the control algorithm.

3.1 The Air Distribution System Model

In order to study the control strategies of an economizer and to simplify the problem, a single room air distribution system is discussed in this paper. The performance of the air distribution system providing the ventilation of a single room is simulated using a model as shown in Figure 3.2. There are several assumptions made for this model:

- The conditioned room was well sealed. Infiltration or exfiltration is negligible. The supply fan and return fan maintained the static pressure of the conditioned room constant.
- In the experimental system, the total duct length was only a few meters and its resistance could be neglected. Comparing the damper resistance with duct and fitting resistances, the duct and fitting resistances were also small, and it could be included in the damper resistance coefficient.

• The air density was considered constant because its temperature varied within a very small range.

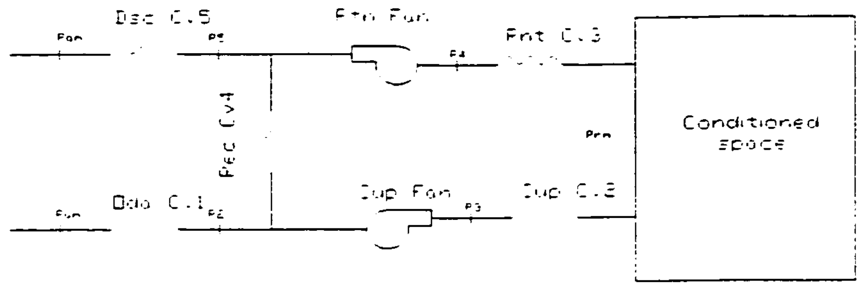

Figure 3.2: The air distribution system model.

When the system is in equilibrium, the supply air flow rate into the conditioned room would be equal the return air flow rate out of the conditioned room,

$$Q_{Sup} = Q_{Rtn} = Q_{Total} \tag{3.1}$$

The pressure drop across a length of the duct, a damper or a fitting can be calculated by

$$\Delta P = f \cdot \frac{L}{D} \cdot \frac{V^2}{2} \cdot \rho \tag{3.2}$$

The velocity, flow rate, and area are related by

$$V = \frac{Q}{A} \tag{3.3}$$

Combining Equation 3.2 and 3.3 and including all constants into a constant value of C_V yields:

$$Q = C_V \cdot \sqrt{\Delta P} \tag{3.4}$$

where

$$C_V = A \cdot \sqrt{\frac{2D}{fL\rho}} \tag{3.5}$$

C_V is simply called the damper resistance coefficient in this paper. The resistance coefficient, C_V, is a constant value when damper position is fixed. The air flow rate through a certain length of duct with fittings and dampers is depended only on the pressure difference between two ends of the duct times total damper resistance coefficient at fixed position.

The outdoor air (Oda), discharge (Dsc) and recirculation (Rsc) dampers are of the same type and size and are coupled in the system. The three dampers are controlled with equal operating signals. The only difference between Oda, Dsc, and Rsc dampers is that the Oda and Dsc dampers is normally open and Rsc damper is normally closed. The damper resistance coefficients of the three dampers may be assumed to be equal when they had same opening. All of the three damper resistance coefficients follow the same curve of resistance coefficient versus opening. In actual installations, the damper resistance coefficients may be affected by other devices in the system. In our model, the damper performance in the system was obtained by experimental tests. Figure 3.3 shows the variation of the damper resistance coefficient versus the control signal as a percentage. In the Figure 3.3, the damper resistance coefficient is a nonlinear function of the control signal. When damper control signal was changing from about 20% to 50%, the damper resistance coefficient had relative small value and increased slowly. After 50%, the damper resistance coefficient increased rapidly with the damper opening position. The typical damper resistance coefficient varies between the 400-4000 cfm/inWc$^{1/2}$ as the damper control signal varies from 20% -85% open. The pressure drop across a damper is measured in inch water column (inWc).

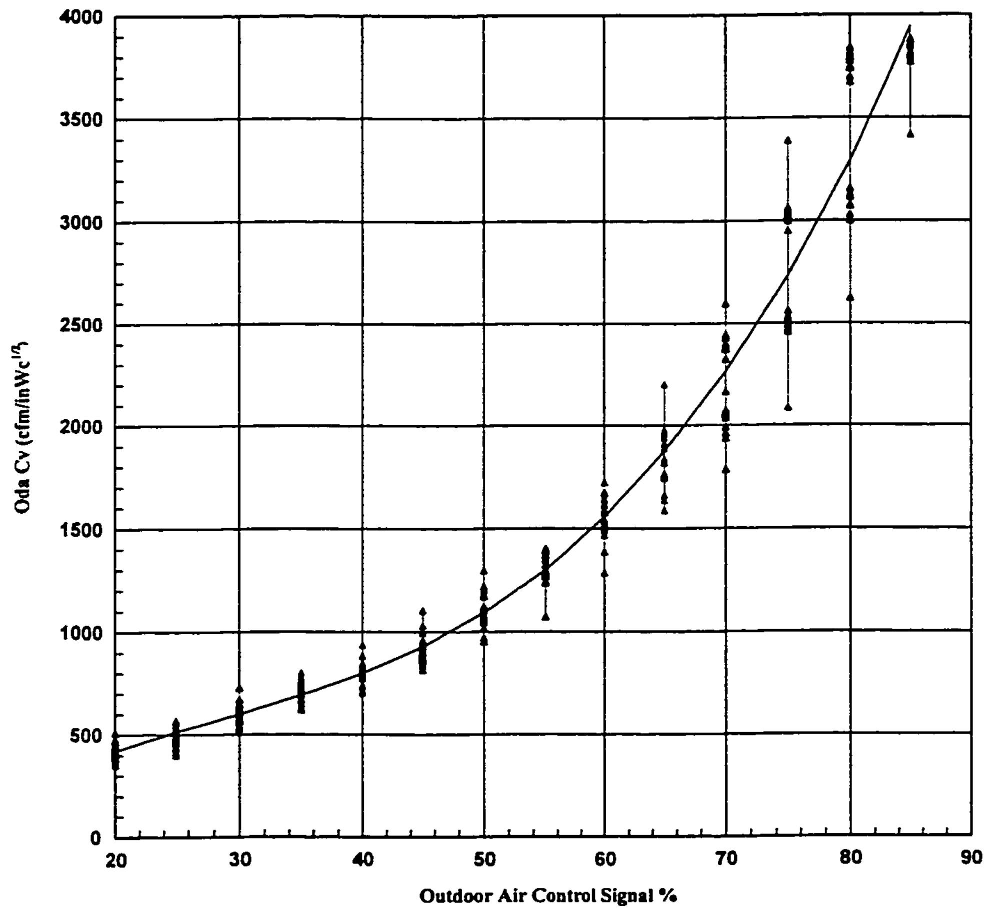

Figure 3.3: The relationship of damper resistance coefficient and control signal for a normally closed damper.

The damper characteristic represented in Figure 3.3 is modeled by the following third order equation, obtained by curve fitting techniques:

$$C_V = -165.077 + 44.52 \cdot S - 1.000459 \cdot S^2 + 0.0123513 \cdot S^3 \tag{3.6}$$

where the control signal is expressed as a percentage in range of 20% to 85%. In the simulating program, a function Cv(SG) related to the experimental damper resistance coefficient to the control signal as per Equation 3.6.

3.2 System Performance Simulation

System performance simulation involves the simultaneous solution of the equations governing the performance of the components.

3.2.1 Dampers

According to the assumptions above, the outdoor air (Oda), discharge (Dsc) and recirculation (Rsc) damper resistance coefficients follows the same characteristic curve as in Figure 3.2. A damper resistance coefficient of a normally closed damper may be represented as function of the damper control signal (%) as:

$$C_V = f \cdot (S) \tag{3.7}$$

The outdoor air (Oda) and discharge (Dsc) dampers are be normally closed. The outdoor air damper resistance coefficient is equal to the discharge one. Therefore:

$$C_{V1} = C_{V5} = f \cdot (S) \tag{3.8}$$

The recirculation (Rsc) damper is normal open. For the three damper system, the recirculation (Rsc) damper resistance coefficient will be:

$$C_{V4} = f \cdot (100 - S) \tag{3.9}$$

For two coupled damper system, the discharge damper always remains at maximum position during system operation. The discharge damper resistance coefficient is constant value:

$$C_{V4} = f(100) \tag{3.10}$$

In this simulated model, the supply air flow rate remains constant in all operating cases. The supply fan outlet pressure and conditioned room pressure are controlled pressures and are maintained at constant values. The supply and return dampers were set at the fixed positions in all operating cases. The resistance coefficients of supply and return ducts are therefore constant.

3.2.2 Air Flow Rates

Whereas constant air density is assumed, mass balance equations may be represented by

$$Q_{Sup} = Q_{Oda} + Q_{Rec} \tag{3.11}$$

and

$$Q_{Rtn} = Q_{Rec} + Q_{Dsc} \tag{3.12}$$

Because the infiltration or exfiltration is assumed negligible, and the controlled pressures remained constant, the supply air flow rate into the conditioned room would be equal the return air flow rate out of it, i.e.:

$$Q_{Sup} = Q_{Rtn} \tag{3.13}$$

Therefore, combining Equations 3.11 through 3.13,

$$Q_{Oda} = Q_{Dsc} \tag{3.14}$$

3.2.3 Pressures

The supply duct pressure (P_3) was the supply fan control point, which was set at 0.5 inWc. The other pressure control point was the room pressure (P_{rm}) which was controlled by the return fan PI controller at 0.0 inWc. In the simulated model, these two pressure are be treated as known specified pressures. By using Equation 3.4, the supply air flow rate is:

$$Q_{Sup} = C_{V2} \cdot \sqrt{P_3 - P_{rm}} \tag{3.15}$$

Because P_3 and P_{rm} are known pressures and the supply damper always remained at a fixed opening, i.e, the value of C_{V2} was a specified constant, the system total supply air flow rate (Q_{Sup}) could be calculated by Equation 3.15. This supply air flow rate would remain constant and be equal to the return air flow rate (Equation 3.13). The return air flow rate would be:

$$Q_{Rtn} = C_{V3} \cdot \sqrt{(P_{rm} - P_4)} \tag{3.16}$$

and

$$P_4 = P_{rm} - Q_{Rtn}^2 / C_{V3}^2 \tag{3.17}$$

Equation 3.17 indicates that the pressure P_4 is indirectly controlled by the return fan control. The return fan maintains the room pressure constant. Therefore, P_4 would also remain at a constant value at all operating conditions.

Combining Equations (3.4) with (3.11) or (3.12) the supply air flow rate and return air flow rate can also be shown as:

$$Q_{Sup} = C_{V4} \cdot \sqrt{P_5 - P_2} + C_{V1} \cdot \sqrt{P_{out} - P_2} \tag{3.18}$$

and

$$Q_{Rtn} = C_{V5} \cdot \sqrt{P_5 - P_{out}} + C_{V4} \cdot \sqrt{P_5 - P_2} \tag{3.19}$$

respectively, where the P_{out} is the outdoor, (i.e., atmosphere pressure) and equals 0.0 inWc. Since the supply flow rate equals the return flow rate, Equations 3.18 and 3.19 can be written as following:

$$C_{V5} \cdot \sqrt{P_5 - P_{out}} = C_{V1} \cdot \sqrt{P_{out} - P_2} \tag{3.20}$$

where

$$Q_{Dsc} = C_{V5} \cdot \sqrt{P_5 - P_{out}} \tag{3.21}$$

$$Q_{oda} = C_{V1} \cdot \sqrt{P_{out} - P_2} \tag{3.22}$$

Because the outdoor air flow is equal the discharge air flow, combining the Equations 3.21 and 3.22 and setting that P_{out} equal to 0.0 inWc, the pressure at point 5 (P_5) becomes:

$$P_5 = -\left(\frac{C_{V1}}{C_{V5}}\right)^2 \cdot P_2 \tag{3.23}$$

For three coupled damper system, outdoor damper resistance (C_{V2}) is equal the discharge damper resistance coefficient (C_{V5}). Hence, the supply fan inlet pressure (P_2) is equal to the negative of the return fan outlet pressure (P_5), i.e.:

$$P_5 = -P_2 \tag{3.24}$$

The difference between the three coupled damper and two coupled damper systems is that the discharge damper did not couple with the other outdoor air and recirculation dampers. For the two coupled damper system, the discharge damper resistance coefficient would not always equal outdoor damper one and the magnitude of pressure at point 2 (P_2) would never be same magnitude of the pressure at point 5 (P_5). Combining Equations 3.18 and 3.23, the supply flow rate can be written as:

$$Q_{Sup} = \left(C_{V4} \cdot \sqrt{\frac{C^2_{V2}}{C^2_{V5}} + 1} + C_{V1} \right) \cdot \sqrt{-P_2} \tag{3.25}$$

The supply fan inlet pressure can be calculated as:

$$P_2 = -\left(\frac{Q_{Sup}}{C_{V4} \cdot \sqrt{\frac{C_{V2}^2}{C_{V5}^2} + 1} + C_{V1}} \right)^2 \tag{3.26}$$

and return fan outlet pressure P_5 may be calculated from P_2 using Equation 3.23 or 3.24 depending on the system configuration. The supply flow rate is known and remains constant in current study of this paper. The fan power can be obtained by using the fan performance program after the system pressures and flow rates are known.

3.2.4 Area Compensation

As previously noted, the damper resistance coefficient is not a linear function with the control signal. The area versus control signal characteristic may be linearized by "area compensation". If the damper blades are considered as a zero thickness with no overlay, the compensated control signal may be determined as a function of the ideal signal as per Krakow [6]. For normally open opposed blade dampers;

$$S_{AC} = 100 \cdot \frac{\operatorname{asin}\left(\frac{S}{100}\right)}{\frac{\pi}{2}} \tag{3.27}$$

For normal closed opposed blade dampers;

$$S_{AC} = 100 \cdot \frac{\mathrm{acos}\left(1 - \frac{S}{100}\right)}{\frac{\pi}{2}} \tag{3.28}$$

The area compensation is accomplished using the Equations 3.27 and 3.28 in the damper control program. A linearized function between the damper resistance coefficient and the control signal is achieved after using the area compensation. The damper resistance coefficient are increased if the signals are compensated. A higher coefficient results in the lower pressure loss for a specified air flow rate as in Equation 3.4. A subroutine SGAV_SG(SGAC,SG) was used to convert the ideal signal to area compensation signal in the simulation program.

3.2.5 Linearized Damper Resistance Coefficient

Ideally, the damper resistance coefficient is linearily proportional to the control signal in the operating range (in this simulation, the signal range was from 20% to 85%. For normally open opposed blade dampers, it has:

$$C_V = k \cdot S/100 + 400 \tag{3.29}$$

For normally closed opposed blade dampers

$$C_V = k \cdot (1 - S/100) + 400 \tag{3.30}$$

where k is the proportional constant. The linear damper resistance coefficient characteristic is given in the Figure 3.4. The damper performance as indicated in Figure 3.4 shows that the minimum and maximum resistance coefficients, corresponding to the minimum and maximum control signal of 20% and 85% are 400 and 4000 cfm/inWc$^{1/2}$, respectively. Therefore:

$$k = \frac{C_{V,max} - C_{V,min}}{S_{max} - S_{min}} = \frac{4000 - 400}{85 - 20} = 55.4 \tag{3.31}$$

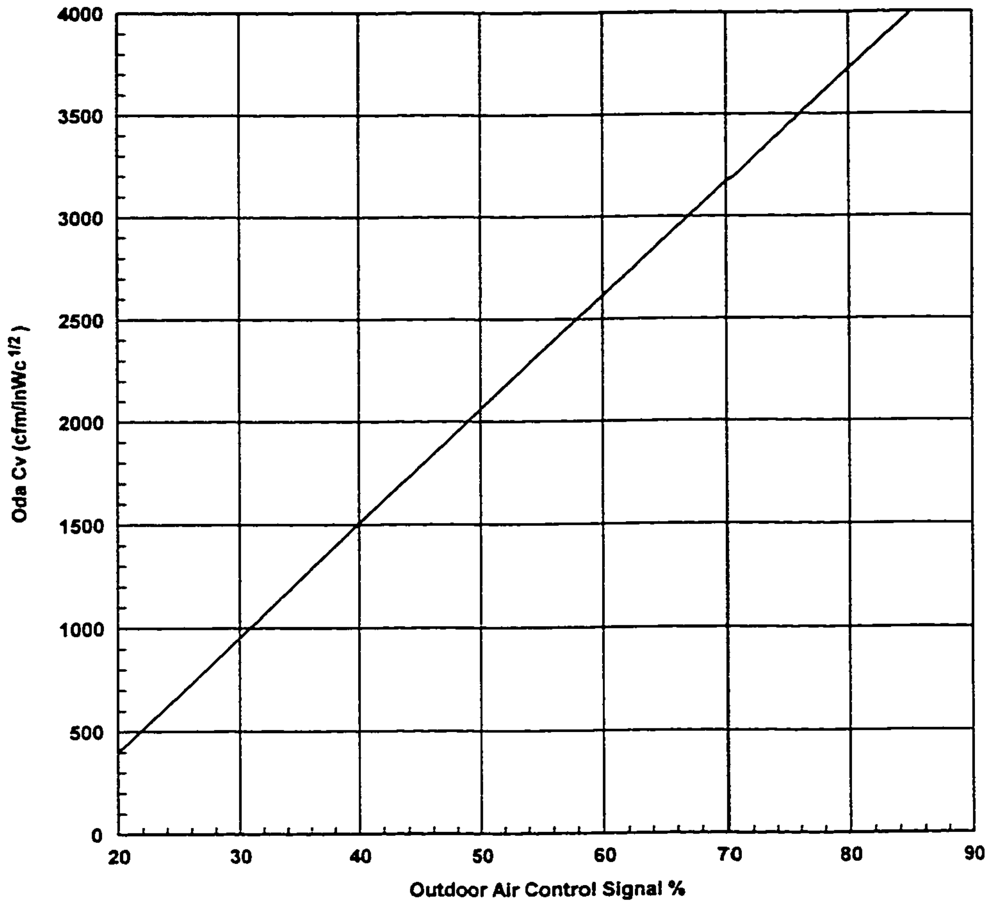

Figure 3.4: The linearized damper resistance coefficient for a normally closed damper.

Simulating system performance using the linearized damper coefficient relationship provides a potential minimum baseline fan energy consumption. This baseline energy consumption may be used as a reference for real system performance analysis. The linearizing damper resistance coefficients relationship is given in the simulation program by the function CVL(SG).

3.2.6 Fans

The fans used in the system were 13 inch wheel, single inlet, backward inclined blade fans manufactured by Delhi Fans. The fans are designated as model BI-13. According to the fan performance chart provided by the manufacturer, the fan power, speed, pressure and air flow can be obtained if any two of the following 4 parameters are known: flow rate, pressure differential, speed, and power. In order to simulate the fan performance in the model, a pressure versus flow rate characteristic line based on the fan speed of 1800 rpm as fourth-order equation, obtained by the curve fitting techniques, is

$$\Delta P = 2.20469 - 0.00021215 \cdot Q + 6.6248\times10^{-7} \cdot Q^2 - 5.78207\times10^{3} \cdot Q^3 + 8.81625\times10^{-14} \cdot Q^4 \tag{3.32}$$

This fan characteristic equation is given by the function of PRS(V) in the simulation program. The fan rated power versus air flow rate, based on the 1800 rpm line, is

$$W = 3.19851 - 0.001466 \cdot Q + 8.5122\times10^{-7} \cdot Q^2 - 4.63054\times10^{-10} \cdot Q^3 + 6.30554\times10^{-14} \cdot Q^4 \tag{3.33}$$

The function of WPV(VX) is used in the simulation program to relate fan power and air flow rate.

In the model, the characteristics of any fan operating point can be calculated by fan laws referenced the basic Equations 3.32 and 3.33. The fan law are:

$$Q_1 = Q_2 \cdot \frac{N_1}{N_2} \tag{3.34}$$

$$P_1 = P_2 \cdot \left(\frac{N_1}{N_2}\right)^2 \tag{3.35}$$

$$W_1 = W_2\left(\frac{N_1}{N_2}\right)^3 \tag{3.36}$$

$$W_1 = W_2\left(\frac{Q_1}{Q_2}\right)^3 \tag{3.37}$$

A fan speed subroutine called FAN_N(V,X,P), consisting of the function of PRS(V) representing Equation 3.32 and the fan law representing Equation 3.35, is used to calculate fan speed at an operating point when the fan air flow rate and pressure differential are known. A fan power subroutine called POWER(P,V,W), consisting by function of the WPV(VX) representing Equation 3.33 and the fan law representing Equations 3.36 and 3.37, is used to calculate the fan power at the fan operating point when the fan flow rate and pressure differential are known.

The fan fluid power rate is the part of the fan power used to increase the air stream enthalpy and kinetic energy. The fluid power does not include electric motor inefficiency, drive losses due to belt flexing and slippage. The fluid power was determined experimentally because the electric power measurements in the laboratory were inaccurate. At low flow Mach numbers, the Bernoulli equation for incompressible flow applies. The fan fluid power is

$$\frac{W}{\rho \cdot Q} = w = \frac{P_2 - P_1}{\rho} + \frac{V_2^2 - V_1^2}{2} \tag{3.38}$$

Neglecting the upstream air velocity, i.e., $V_1 \approx 0$, the total fan fluid power is:

$$W = Q \cdot (P_2 - P_1) + \rho \cdot Q \cdot V_2^2/2 \tag{3.39}$$

In our system, the supply fan flow rate was constant at 2600 cfm and the minimum outlet area of the fan was 1.04 ft.^2. Substituting these values into the Equation 3.39, the fan fluid power equation can be simplified as:

$$W = 306.0 \cdot \Delta P + 118.6 \tag{3.40}$$

The supply fan and the return fan fluid power can be calculated by the simulating program when the fan inlet and outlet pressure differential is known. The total fan fluid power consumption for both supply and return fans is presented in the system performance analysis.

3.3 Numerical System Model

Based on previous assumptions and theoretical analysis, a numerical model of the air distribution system was constructed. A simulation program, based on the model, was written in Fortran code. The program listing, and sample calculations, are presented in Appendix I. The simulation program has following steps:

(1) Input values of constants from an input data file. These constants are:

- the room pressure,
- the outdoor ambient pressure,
- the supply fan outlet pressure,
- the supply damper resistance coefficient,
- the return damper resistance coefficient,
- the damper configuration (two or three coupled dampers), and
- the damper characteristic option (experimental, area compensated, or linearized).

(2) Initialization

Preset the variables used in the program to zero.

(3) Calculate the supply fan flow rate from known pressures and supply damper resistance coefficient (Equation 3.15),

(4) Signal selection. The signal is varying from 20% to 85% by 14 steps in interval of 5%.

(5) Select the damper characteristic option. There are three options available in the program:

- experimental damper characteristic,
- area compensated characteristic, and
- linearized damper characteristic.

(6) Calculate the dampers resistance coefficients corresponding case of damper characteristic option selected. Calculate the outdoor air, discharge, and recirculation damper resistance coefficients for the control signal by using Equation 3.6 or 3.29.

(7) Calculate supply fan inlet pressure using Equation 3.26.

(8) Select the damper configuration, two or three coupled dampers.

(9) Calculate return fan outlet pressure.

(10) Calculate the pressure different across the supply fan,

(11) Calculate the supply fan speed and rated power. The fan speed and power is obtained by the subroutines of the FAN_N(CFM,N,P) and POWER(P,CFM,W) from the known flow rate and pressure differential across the fan.

(12) Calculate the pressure different across the return fan.

(13) Calculate the return fan speed and rated power using Equation 3.36.

(14) Calculate outdoor air flow rate and discharge air flow using Equations 3.28 and 3.29.

(15) Calculate fluid power for supply and return fans. The fan speed and power is obtained by the subroutines of the FAN_N(CFM,N,P) and POWER(P,CFM,W) from the known specified the fan flow rate and pressure different across the fan.

(16) Save and print to an output file the results at current signal selection.

(17) Precede to the next signal and repeat from Step 4 until 85% is obtained.

(18) Totalize the fan rated power and fluid power.

(19) Save and print to an output file the final results.

(20) End

Figure 3.5 represents a flow chart of the simulation program.

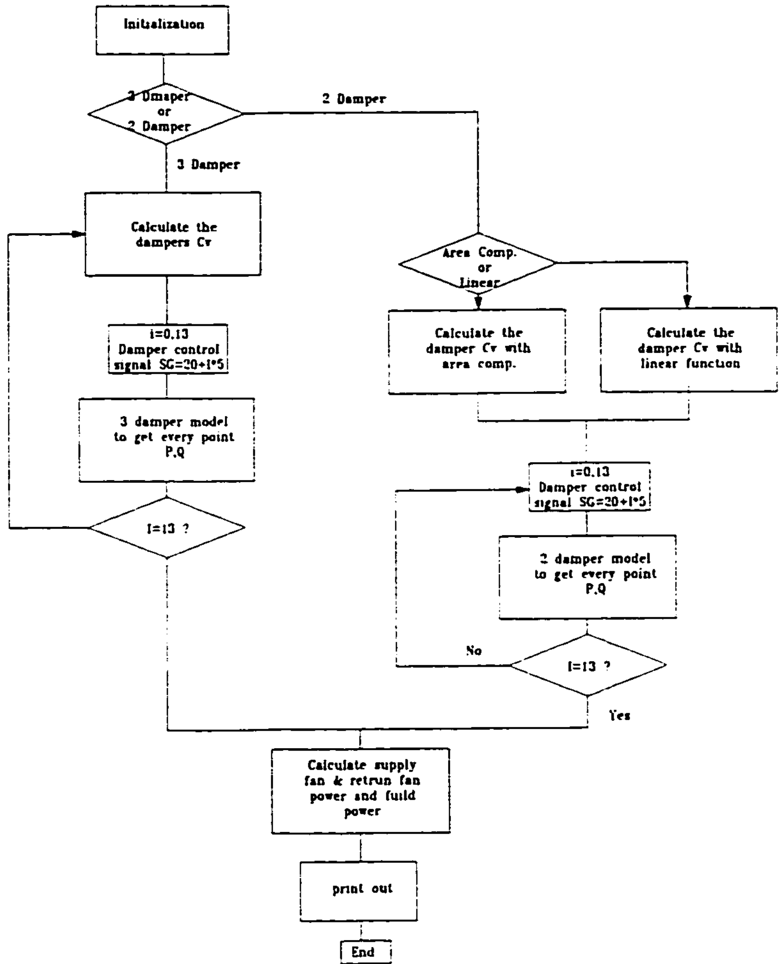

Figure 3.5: The flow chart of the simulation program.

3.4 Simulated Results and Discussion

The model can successfully simulate system performance - the key point pressures, flow rates, the fan rated powers, and fluid powers - for various control strategies. The following control were simulated:

- three coupled dampers (with experimental damper resistance coefficients),
- two coupled dampers (with experimental damper resistance coefficients),
- three coupled dampers with area compensated damper resistance coefficients,
- two coupled dampers with area compensated damper resistance coefficients, and
- two coupled dampers with linearized damper resistance coefficients.

Performance was simulated for outdoor air damper control signal ranging from 20% to 85%. Table 3.1, below, details figures showing the results, outdoor air flow rate, pressure, and power.

Table 3.1: Simulation Result Figures

	Strategy	Oda flow rate and pressure	Power
1	three coupled dampers	Figure 3.6	Figure 3.11
2	two coupled dampers	Figure 3.7	Figure 3.12
3	three coupled dampers, area compensated	Figure 3.8	Figure 3.13
4	two coupled dampers, area compensated	Figure 3.9	Figure 3.14
5	two coupled dampers, linearized	Figure 3.10	Figure 3.15

The average fan powers was determined by averaging powers for each control signal. Tables 3.2 and 3.3 show the simulated average fluid and rated power requirements.

Table 3.2: Average Fluid Power Simulation Results

	Strategy	**Supply Fan W**	**Return Fan W**	**Total W**	**% Saved compared with three dampers**
1	three coupled dampers	457.4	346.3	803.8	0%
2	two coupled dampers	530.8	181.5	712.4	11.4%
3	three coupled dampers, area compensated	412.5	301.3	713.8	11.2%
4	two coupled dampers, area compensated	484.4	166.9	651.3	19.0%
5	two coupled dampers, linearized	353.1	184.8	537.8	33.1%

From the fan fluid power simulation results, it is obviously that the two coupled damper system is more economical than the corresponding three coupled damper system. It will save about 11.4% of the energy. The three coupled damper system with area compensation is close in value of fan fluid power with two coupled damper system without area compensation. If the two coupled damper system with the area compensation was used, it would save more energy 19.0% compared to the three coupled damper system without area compensation. The two coupled damper system with a linearized damper resistance coefficient obtained the maximum energy saving of 33.1%. It is shown that the area compensation is effective and can improve the system's performance.

Table 3.3: Average Rated Power Simulation Results

	Strategy	Supply Fan W	Return Fan W	Total W	% Save compared with three damper
1	three coupled dampers	971.6	823.7	1795.3	0%
2	two coupled dampers	1073.7	612.8	1686.5	6.1%
3	three coupled dampers, area compensated	913.0	765.9	1679.7	6.4%
4	two coupled dampers, area compensated	1010.4	594.6	1605.0	10.6%
5	two coupled dampers, linearized	831.5	616.7	1448.3	19.3%

The fan rated power is the fan motor power, which includes the fan fluid power and other fan power consumption due the mechanical resistance, belts and fan efficient etc. This power must be higher in value than the fan fluid power. The simulated results in Table 3.3 show that fan rated power will be reduced about 6.1% if the two coupled damper control strategy was used rather than the three coupled damper control strategy. As with the fan fluid power, the fan rated power consumption of the three coupled damper system with area compensation is close to that of the two coupled damper system without area compensation. The two coupled damper system with area compensation will save up to 10.6% as compare with the three coupled dampers system.The two coupled damper system with a linearized damper resistance coefficient obtained the maximum energy saving of 19.3%. A comparison of the rated power and the fluid shows that the rated power savings will be similar but less than the fluid power saving.

The graphs of Figures 3.6 through 3.15 show outdoor air flow rate, supply fan inlet and return fan outlet pressures, and supply and return fan power as a function of the outdoor air control signal. The outdoor air flow rate for the two coupled damper system with the ideal linear damper characteristic versus the control signal is the most linear of the systems tested. The supply fan inlet and return fan outlet pressures for this system show the least variation. Whereas these pressure characteristics are the most constant, fan speeds modulation will be a minimum. This system requires the least power. This system is therefore the optimum. Using the foregoing criteria, the two damper system with area compensation is better than the two damper system without area compensation. Two damper systems are better than corresponding three damper systems.

For a three damper system, the maximum power consumption occurs at approximately 50% outdoor air flow. For the two coupled damper systems, the peak power consumption shifts toward the 100% outdoor air flow. The actual power savings would therefore depend on the operating time at each control signal range.

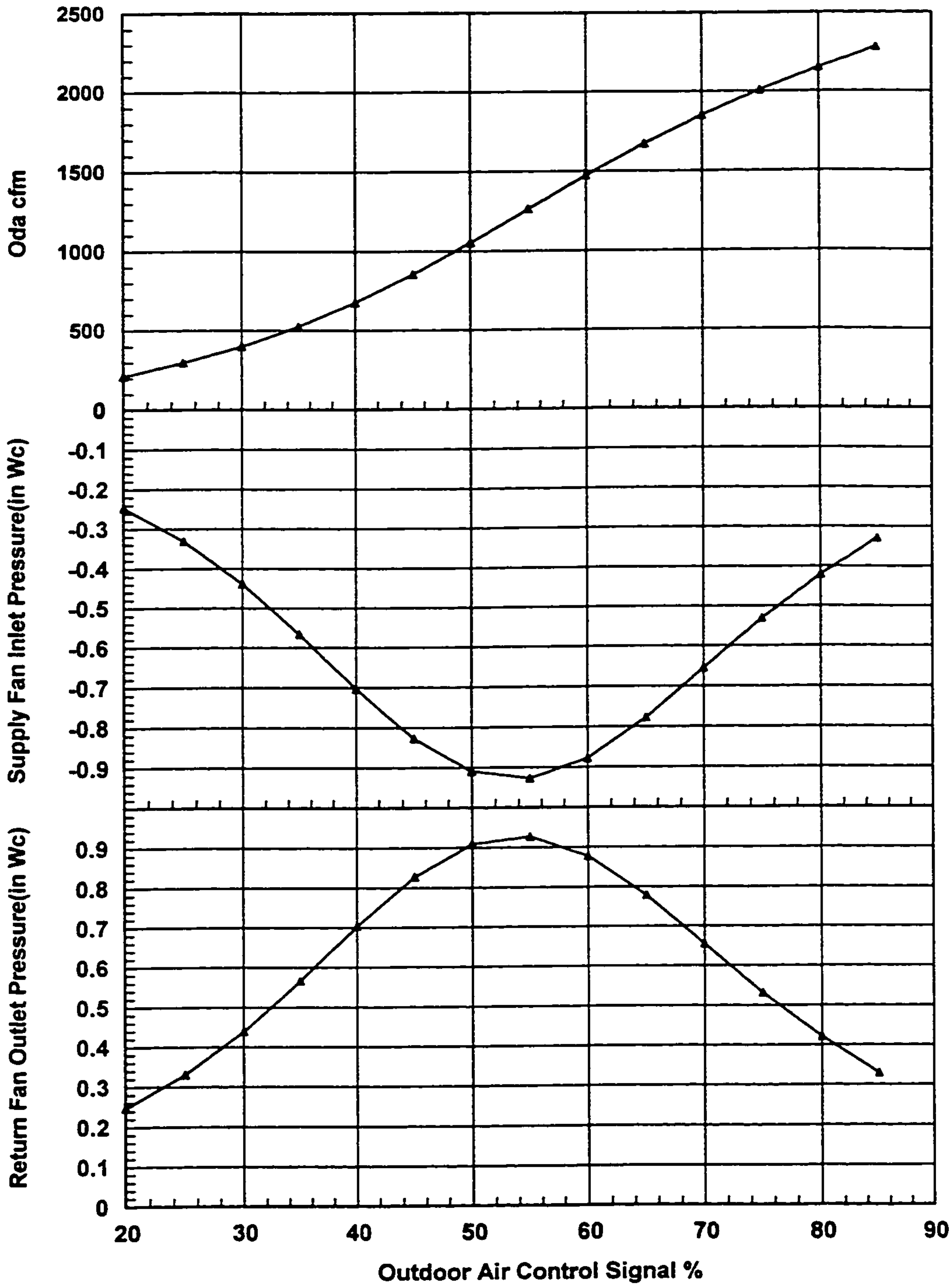

Figure 3.6: Flow rate and pressure characteristics for a three coupled damper system.

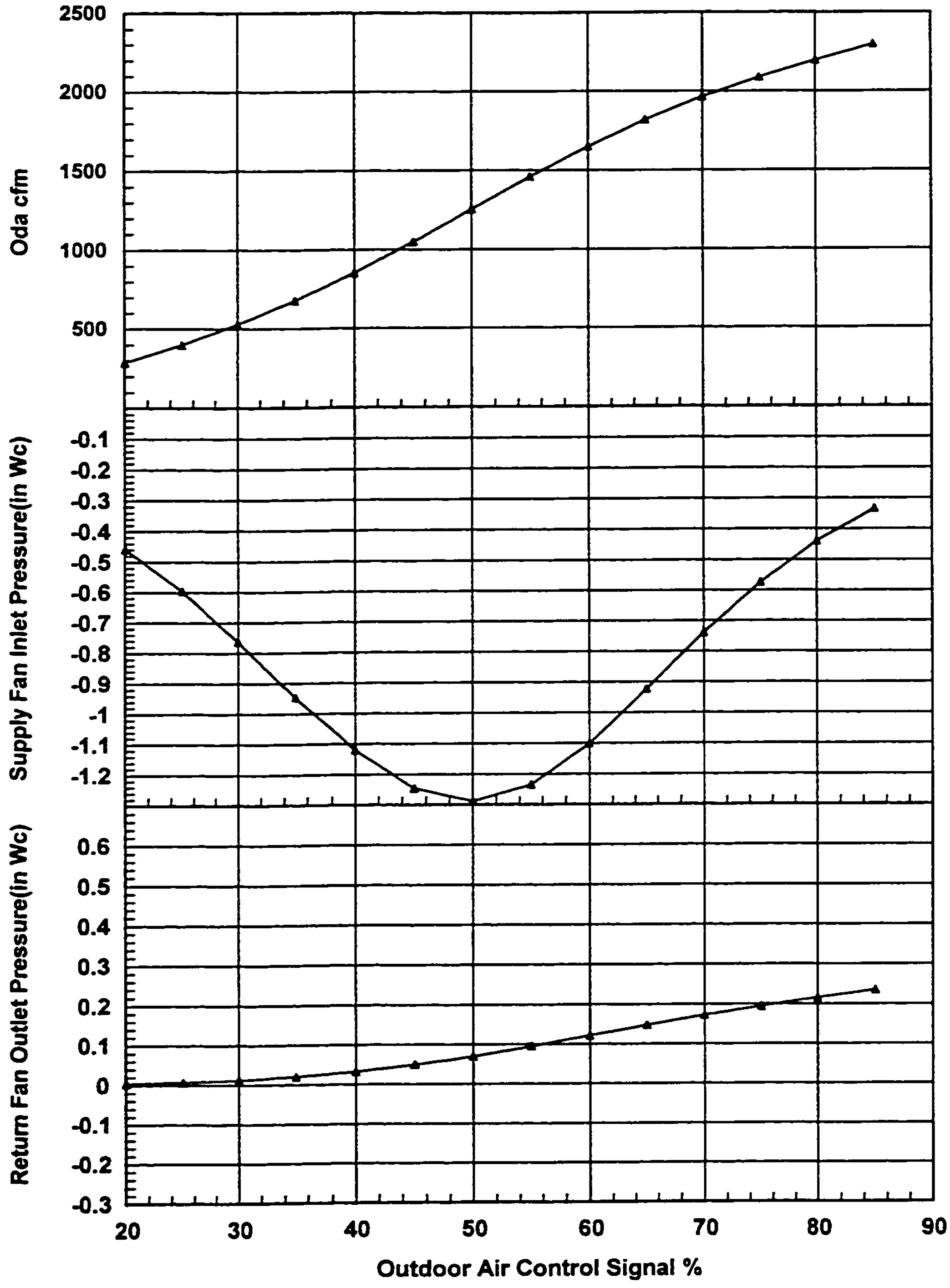

Figure 3.7: Flow rate and pressure characteristics for a two coupled damper system.

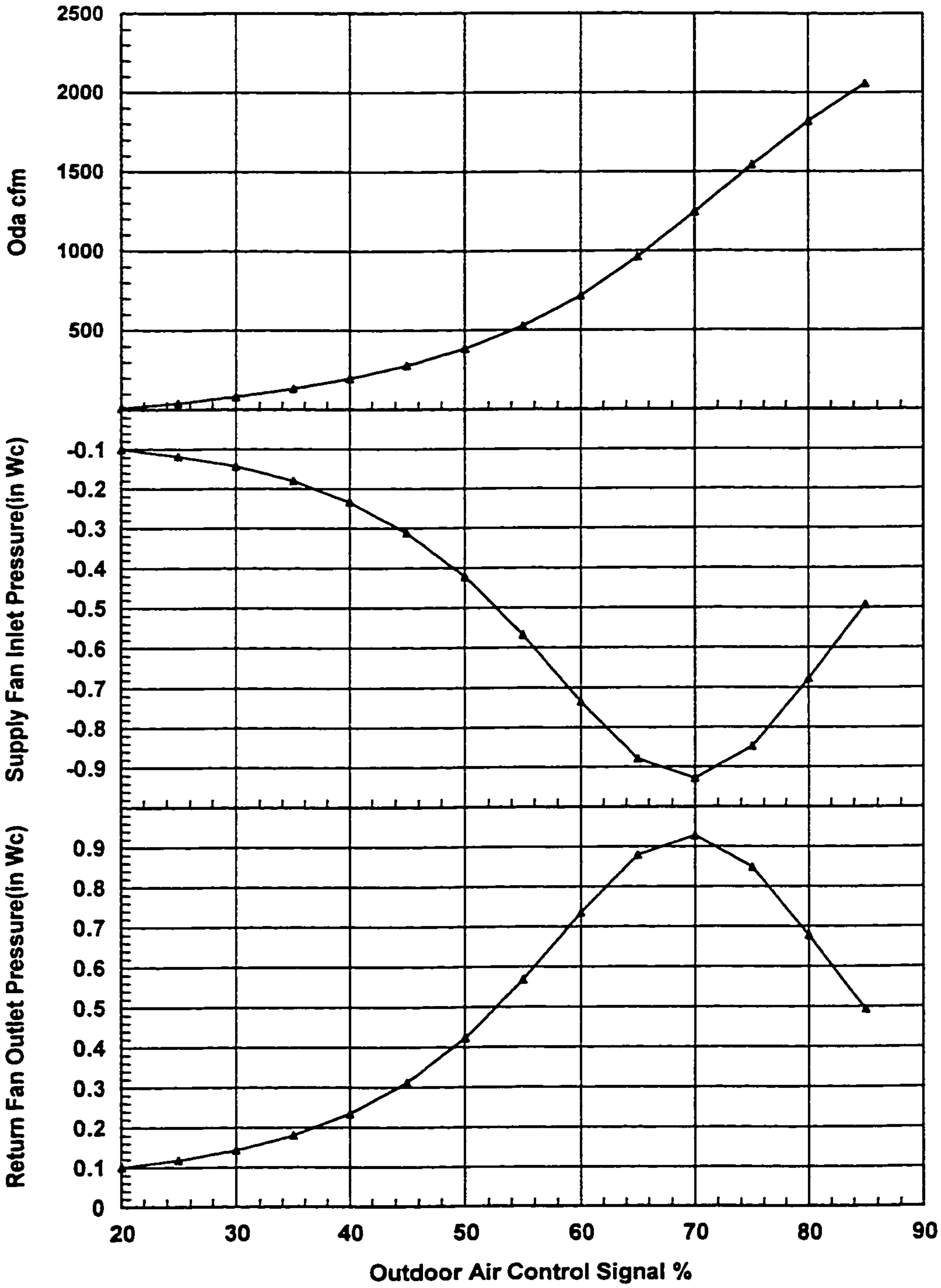

Figure 3.8: Flow rate and pressure characteristics for a three coupled damper system with area compensation.

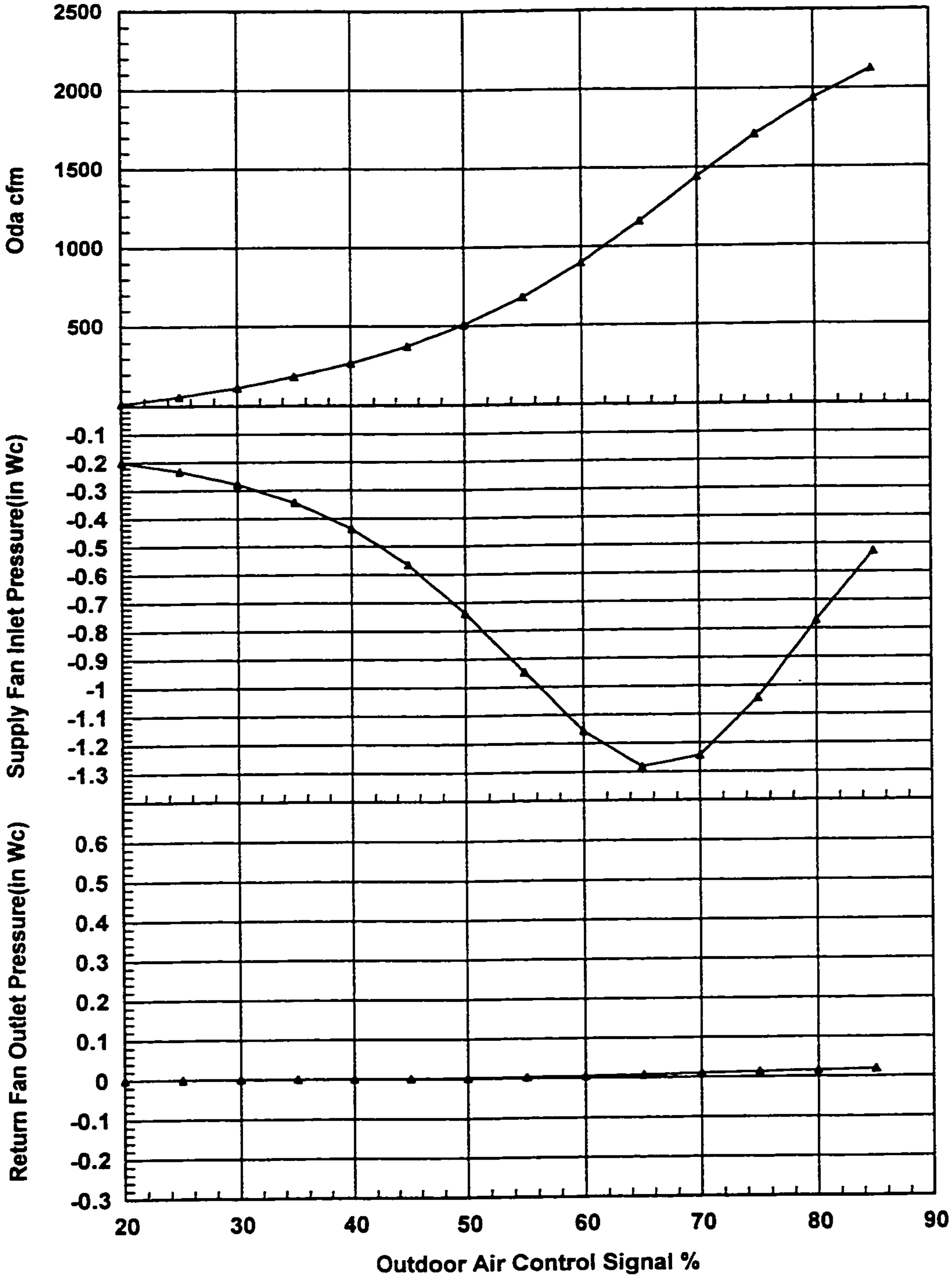

Figure 3.9: Flow rate and pressure characteristics for a two coupled damper system with area compensation.

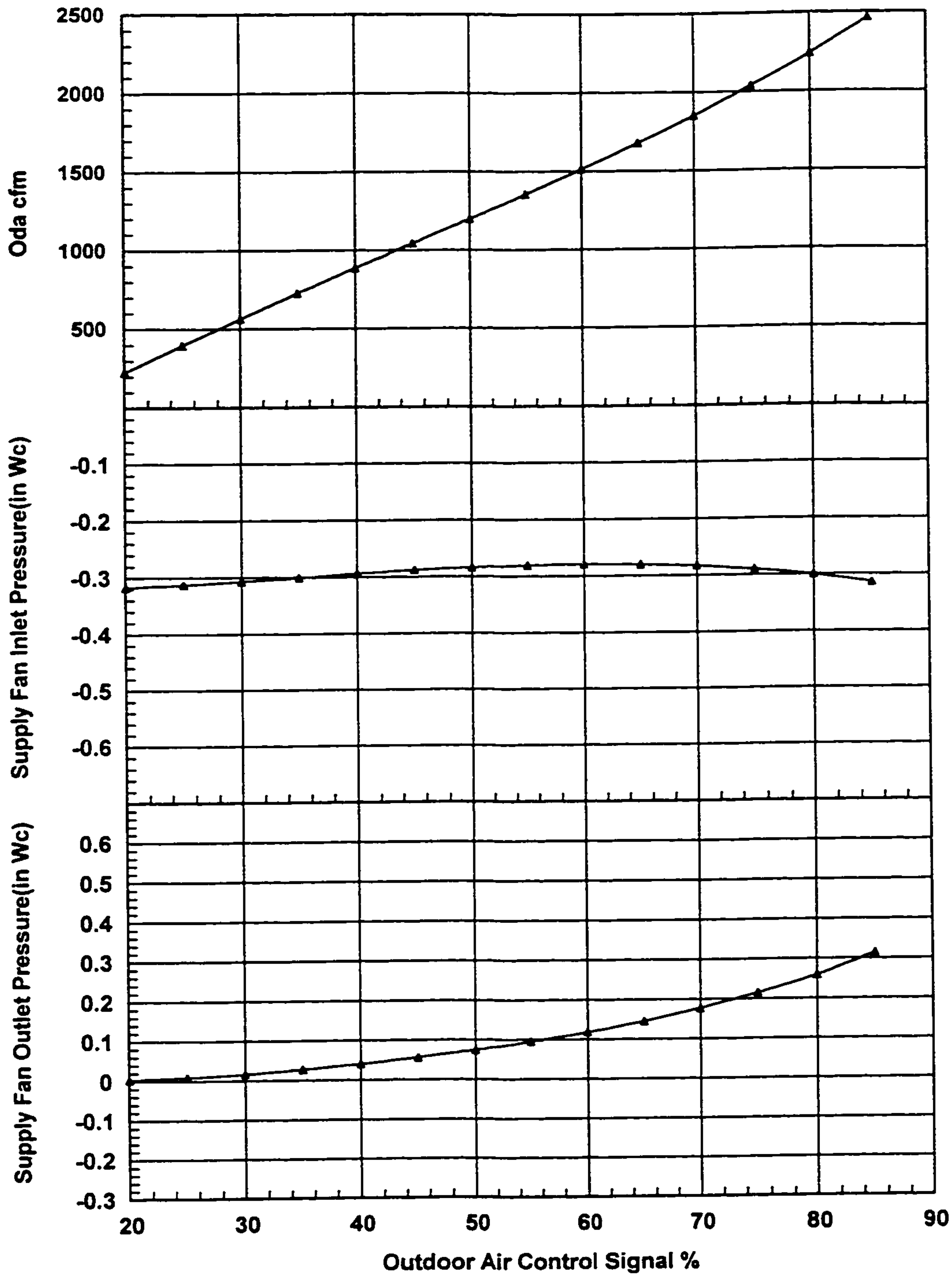

Figure 3.10: Flow rate and pressure characteristics for a two coupled damper system with ideal linear dampers.

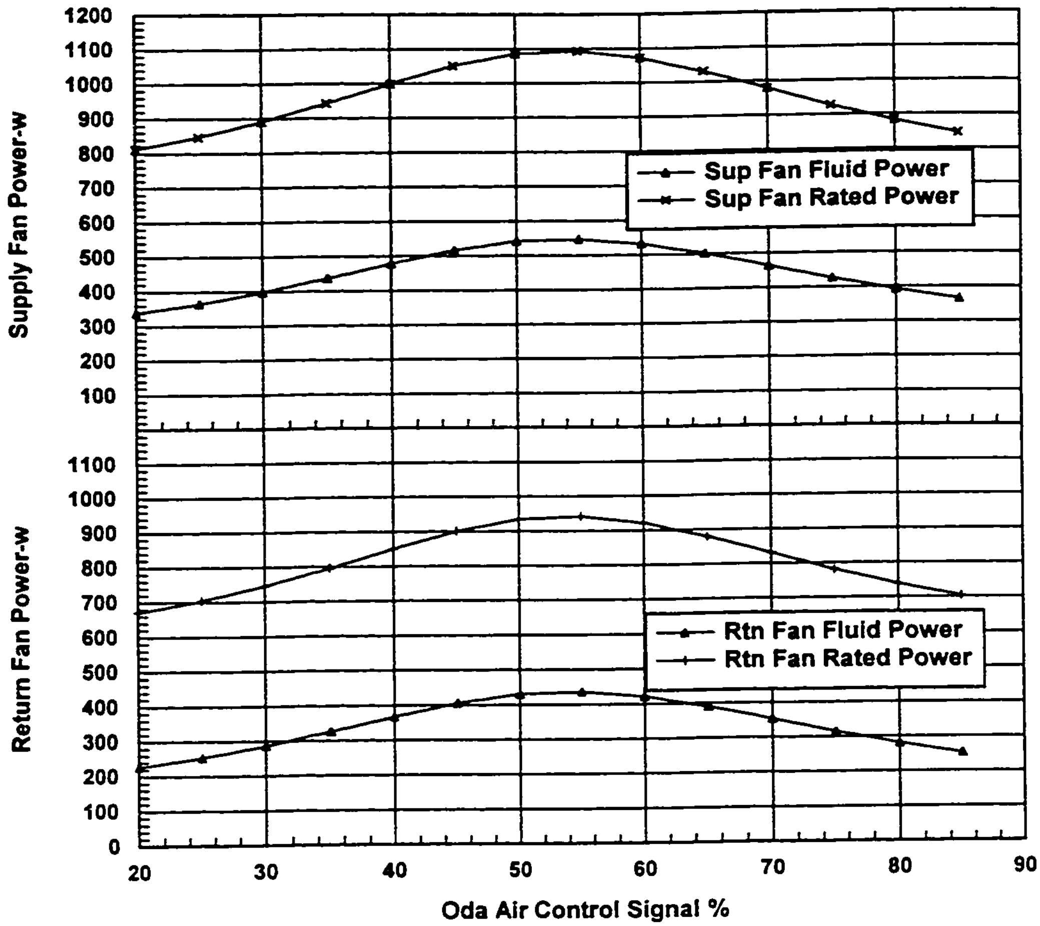

Figure 3.11: Power characteristics for a three coupled damper system.

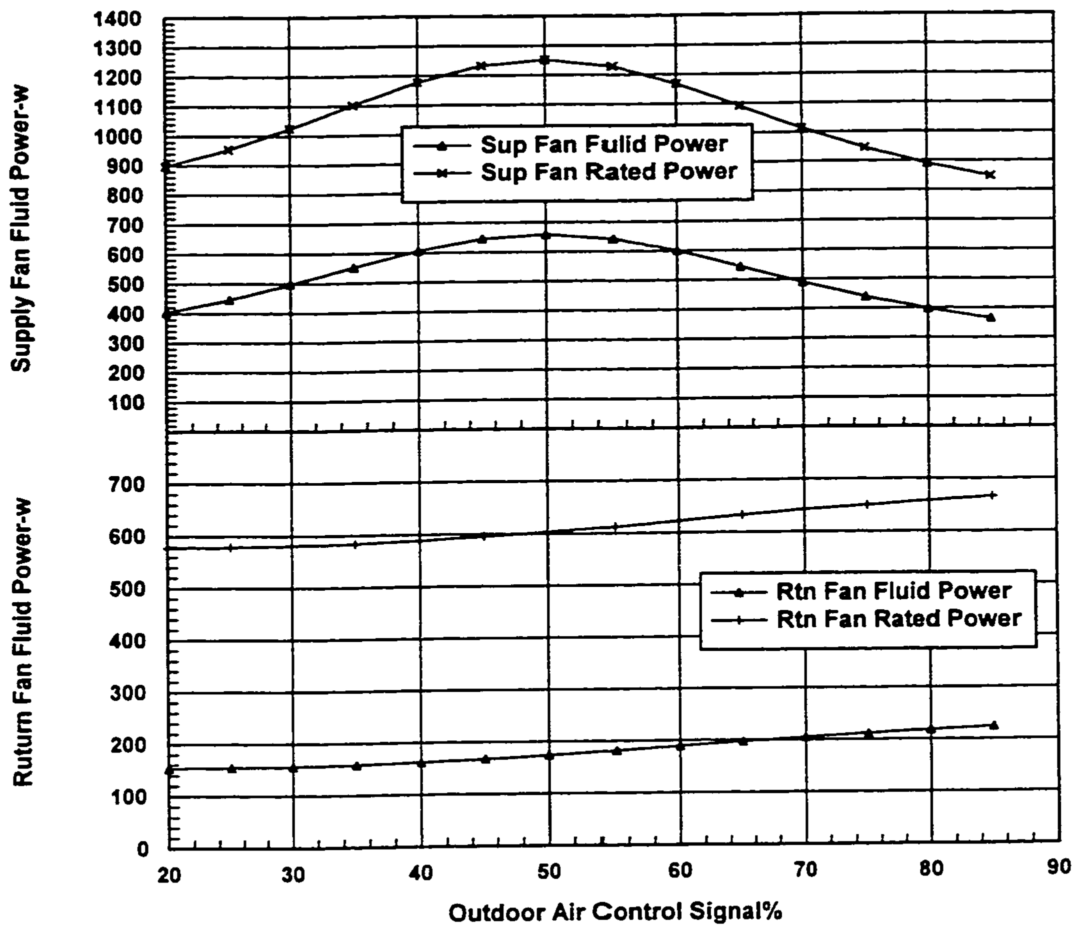

Figure 3.12: Power characteristics for a two coupled damper system.

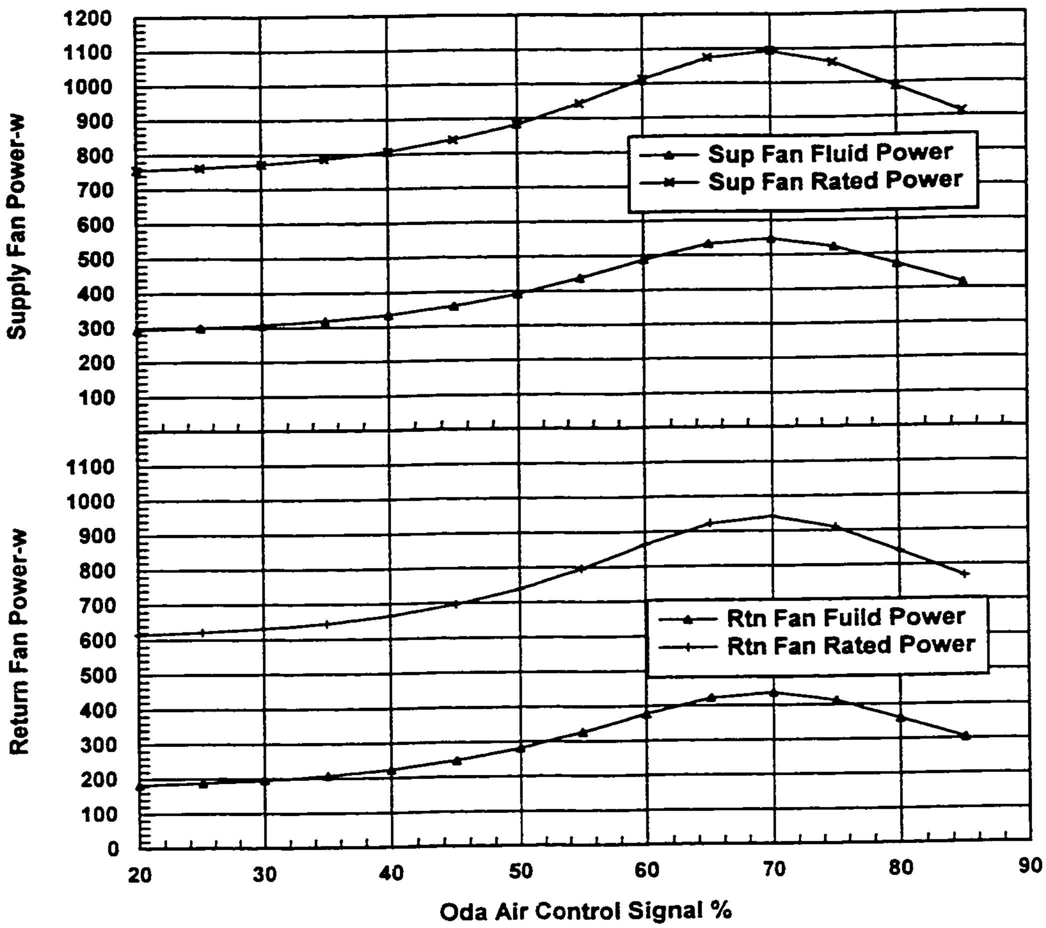

Figure 3.13: Power characteristics for a three coupled damper system with area compensation.

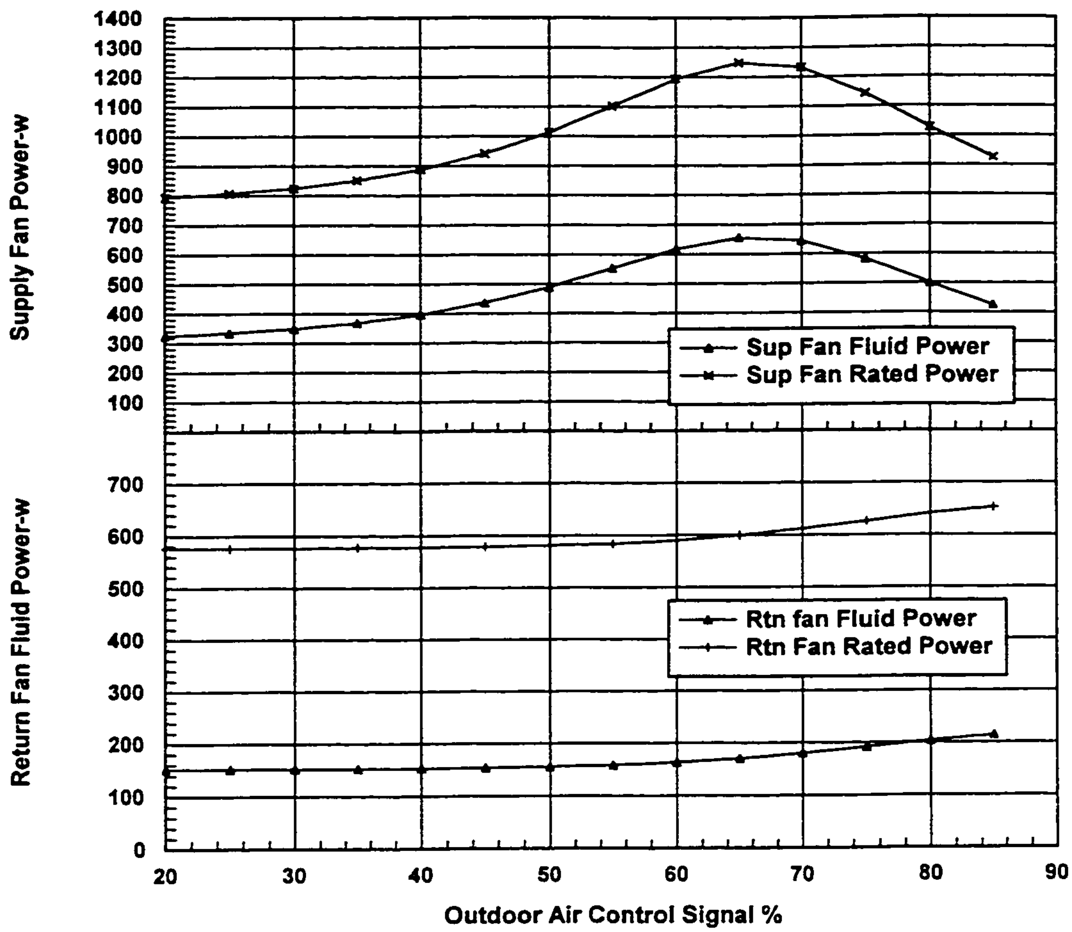

Figure 3.14: Power characteristics for a two coupled damper system with area compensation.

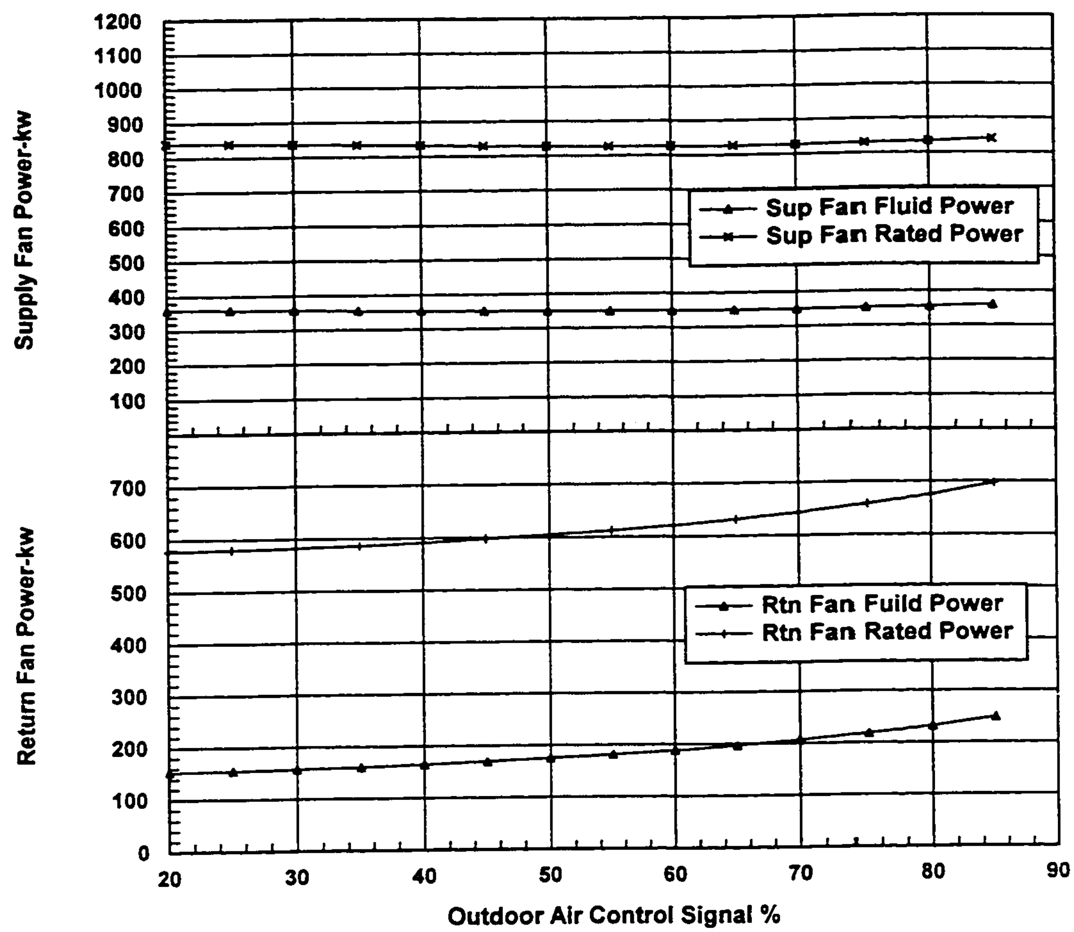

Figure 3.15: Power characteristics for a two coupled damper system with ideal linear dampers

4. AIR QUALITY CONTROL USING A CO_2 CONCENTRATION METHOD

Indoor air quality has become very important in a building's ventilation control. To avoid adverse health effects, an air ventilation control strategy should be considered in order to provide adequate indoor air quality with a minimum of energy used. Traditionally, a minimum fresh air flow rate is introduced into the building as the measure to conserve energy. ASHRAE Standard 62-1989 [9] allows the outdoor ventilation air to be controlled based on a measure of the indoor air quality. In many situations, the occupation generated CO_2 can serve as a suitable measure of the indoor air quality. Because the rate of indoor generated carbon dioxide is dependent on the number of occupants, it is proposed that the indoor air carbon dioxide concentration could be used to measure the indoor air quality control to determine the fresh outdoor air flow rate introducing into the building. In a air ventilation system, a carbon dioxide sensor can be used in the return duct to measure the carbon dioxide concentration in order to monitor the build air quality and provide a control feedback signal to a proportional integral (PI) control system. The PI control system will control the outdoor damper opening to ensure the minimum fresh outdoor air flow into the building. Such controlled ventilation provides a possibility for minimum energy consumption when number of the occupants change.

4.1 The Air quality Control Model

A air quality control model for a single room can be built by the mass balance method. Assuming that the controlled room is well sealed, the infiltration or the exfiltration can be neglected. The supply fan provides a constant air flow rate to the room. The concentration of CO_2 in the supply air varies and is controlled by the coupled outdoor air and recirculation dampers by mean of changing the proportion of the return air and outdoor fresh air. The return air CO_2 concentration is the control point, which is generally set at the 1000 ppm. The occupants generate CO_2 at a specified rate into the room. The model is shown schematically in Figure 4.1.

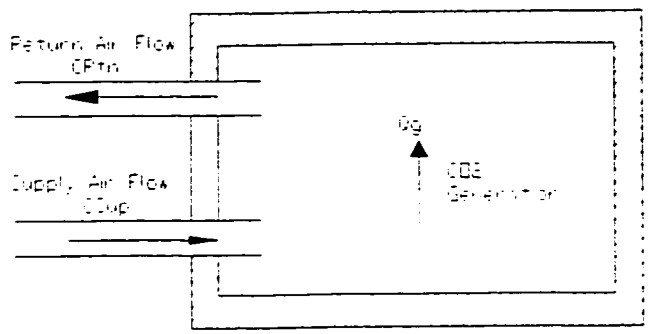

Figure 4.1: The conditioned room air quality model.

The steady state mass balance of CO_2 can be represented as the follows assuming constant densities:

$$Q_{Rtn} \cdot C_{Rtn} = Q_{Sup} \cdot C_{Sup} + Q_g \tag{4.1}$$

The room CO_2 concentration is generally controlled at a set point of 1000ppm. The supply air flow rate is equal to the return air flow rate and are constant, i.e.:

$$Q_{Rtn} = Q_{Sup} \tag{4.2}$$

Equation 4.1 may therefore be re-written as

$$(Q_{Dsc} + Q_{Rsc}) \cdot C_{Rtn} = (Q_{Oda} + Q_{Rsc}) \cdot C_{Sup} + Q_g \tag{4.3}$$

According to the CO_2 mass balance, the CO_2 mass flow rate of supply air can be presented as

$$(Q_{Oda} + Q_{Rsc}) \cdot C_{Sup} = Q_{Oda} \cdot C_{Oda} + Q_{Rsc} \cdot C_{Rtn} \tag{4.4}$$

The outdoor CO_2 concentration is constant. Therefore, Equation 4.3 becomes

$$(Q_{Dsc} + Q_{Rsc}) \cdot C_{Rtn} = Q_{Oda} \cdot C_{Oda} + Q_{Rsc} \cdot C_{Rtn} + Q_g \tag{4.5}$$

and (4.6)

$$Q_{Oda}(C_{Rtn} - C_{Oda}) = Q_g \tag{4.7}$$

4.2 CO_2 Generation

According to the ASHRAE standard 62-1989 [9], the indoor carbon dioxide concentration is depended on the occupation physical activity. The activity level is evaluated by the met unit (1.0 met=18.4 Btu/h.ft^2). The standard provides the met range for different activity level as shown in Table 4.1

Table 4.1: Physical Activity Levels

activity level	met	example
very light activity	0.5 -1.5	sleeping seated, quiet office work
light activity	1.5 - 3.5	walking 3 to 5 mph
moderate activity	3.5 -5.0	heavy work

The standard also gives the relationship between the occupation CO_2 generation rate and the physical activity level by the following:

$$q_g = 0.0088 \times met \tag{4.8}$$

The outdoor air flow rate per person, obtained from Equation 4.7, is

$$q_{out} = \frac{q_g}{(C_{Rtn} - C_{Oda})} \tag{4.9}$$

or

$$Q_g = n \times q_g \tag{4.10}$$

In this paper, the control set point of the room CO_2 concentration is set at 1000 ppm. The outdoor air CO_2 concentration is measured to be the constant at 350 ppm. Therefore, the required fresh outdoor air flow rate is a function of the occupation CO_2 generation flow rate and is proportional with the number of persons in the room when the system is controlled by the controller and is stable, i.e.:

$$Q_{Oda} = f(n) = k \cdot n \tag{4.11}$$

4.3 The CO_2 Control Mode

If the rate of CO_2 generation is known, the required outdoor air flow rate may be determined by the solution of the foregoing equations. However, if the rate of CO_2 generation is not known, a feedback control system is required to determine the required outdoor air. A proportional integral (PI) control system may be used to determine the required outdoor air flow rate.

4.3.1 The PI Control

As stated, the CO_2 generation of the occupants is a variable and depends on the number of people in the room. Therefore, the outdoor air flow needs to be modulated in order to maintain the CO_2 concentration of a room at a constant value. The CO_2 concentration of a room is considered the controlled variable and is measured by a sensor in the room. The outdoor flow rate is considered the system variable. A controller is required to alter the system variable by varying the damper position to meet the specification of the CO_2 concentration in the room. The function of the controller is to detect the error between a desired set point and the actual measured value of control variable and generate a control signal according to the error for the actuator to eliminate the error until the control variable is brought to the desired set point within the design limitations. This system is called feedback control.

A PI controller is widely used in HVAC systems. The PI control algorithm can be represented as following:

$$S = K_p \cdot E + K_I \cdot \Sigma E \tag{4.12}$$

where the error is

$$E = C_{SP} - C \tag{4.13}$$

The K_P is the propositional coefficient and the K_I is the integral coefficient. For the system PI controller, the control coefficients could be calculated, as per Krakow [21], by:

$$K_p = \frac{1}{\sigma \cdot \Delta t} \tag{4.14}$$

and

$$K_I = \frac{1}{4 \cdot \sigma \cdot \Delta t} \tag{4.15}$$

The sensitivity is

$$\sigma = \frac{d}{ds}\left(\frac{dC}{dt}\right)$$

The sensitivity is determined by:

$$\sigma = \frac{Q_{Sup}}{V_{Rm}} \cdot \frac{(C_{Sup} - C)_{Max} - (C_{Sup} - C)_{Min}}{S_{Max} - S_{Min}} \tag{4.16}$$

The value term of $(C_{Sup} - C)_{Min}$ can be considered as approximately equal to zero if 100% or the return air is recirculated. Therefore the equation can be written as:

$$\sigma = \frac{Q_{sup} \cdot (C_{Sup} - C)_{Max}}{V_{Rm} \cdot (S_{Max} - S_{Min})} \tag{4.17}$$

Because the sensitivity must be the maximum value, the sensitivity can be obtained by:

$$\sigma = \frac{Q_{sup} \cdot (C_{Oda} - C_{Sup})_{Max}}{V_{Rm} \cdot (S_{Max} - S_{Min})} \tag{4.18}$$

The maximum anticipated space CO_2 concentration is used to calculate the maximum sensitivity.

4.3.2 The Mathematical Control Model of the System

The CO_2 concentration in the room is a function of time. It depends on the amount of CO_2 brought into the room by the supply air and brought out by return air as well as the rate of CO_2 generation by occupants in the room. The function can be determined by applying law of conservation of mass to model in the Figure 4.1 and can be represented by a differential equation as follow:

$$\dot{m}_{(CO_2, Sup)} + \rho_{CO_2} \cdot Q_g = \rho_{CO_2} \cdot V_{Rm} \cdot \frac{dC}{dt} + \dot{m}_{(CO_2, Rtn)} \tag{4.19}$$

Noting that

$$\dot{m}_{(CO_2, Sup)} = \rho_{CO_2} \cdot Q_{Sup} \cdot C_{Sup} \qquad (4.20)$$

$$\dot{m}_{(CO_2, Rtn)} = \rho_{CO_2} \cdot Q_{Rtn} \cdot C_{Rtn} \qquad (4.21)$$

$$Q_{Sup} = Q_{Rtn} \qquad (4.22)$$

and

$$C_{Rtn} = C \qquad (4.23)$$

therefore

$$V_{Rm} \cdot \frac{dC}{dt} + Q_{Sup} \cdot (C - C_{Sup}) = Q_g \qquad (4.24)$$

Equation 4.24 can be rewritten as the form of finite element equation for small interval as:

$$V_{Rm} \cdot \frac{\Delta C}{\Delta t} + Q_{Sup} \cdot (C - C_{Sup}) = Q_g \qquad (4.25)$$

or

$$C_{t+\Delta t} = \left(\frac{(Q_g - Q_{Sup} \cdot (C - C_{Sup})) \cdot \Delta t}{V_{Rm}}\right) + C_t \qquad (4.26)$$

Equation 4.26 enables the calculation of the value of CO_2 concentration in the room at next time step. This equation will be used in the CO_2 control simulation program.

4.3.3 Mixing Dampers Equation

Applying the law of conservation of mass to the mixing dampers, it yields

$$C_{Oda} \cdot Q_{Oda} + C_{Rec} \cdot Q_{Rec} = C_{Sup} \cdot Q_{Sup} \qquad (4.27)$$

The CO_2 concentration in the supply air is, therefore,

$$C_{Sup} = C_{Oda} \cdot \frac{Q_{Oda}}{Q_{Sup}} + C \cdot \frac{Q_{Rec}}{Q_{Sup}} \qquad (4.28)$$

4.3.4 The Solution

Based on previous assumptions and theoretical analysis, a numerical model of the air CO_2 control system was built. A simulation program, based on the model, was written in Fortran code, is listed in Appendix II.

The simulation program has following steps:

(1) Input the values of constant from a data input data file. The constants are:

- the time interval,
- simulation time,
- initial CO_2 generation rate of occupancy,
- initial CO_2 concentration in the room in ppm,
- the outdoor air CO_2 concentration in ppm,
- the specified set point CO_2 concentration in room in ppm,
- the anticipated maximum CO_2 concentration in room in ppm for sensitivity calculation,
- size of room in ft^3,
- the supply air flow rate in cfm,
- the number of damper controlled, and
- the damper characteristic option

(4) Unit conversions.

Convert the CO_2 concentration in ppm to the percentage.

(5) Initialization.

Preset the variables used in the program to zero.

(6) Calculate system sensitivity by using Equation 4.18

(7) Calculate proportional and integral coefficients by using the Equations 4.14 and 4.15.

(8) Increase the CO_2 generation rate by step 5 cfh from initial value of 30 cfh to maximum 60 cfh and then back to 30cfh at 30 minute intervals.

(9) Calculate system outdoor air flow rate under specified control signal provided by the PI controller by using the system subroutine SYS(NSG,VODA,NDPR,NCR,VSUP), which is described in the Chapter 3.

(10) Calculate the CO_2 concentration in the supply air by using the Equation 4.29

(11) Calculate the CO_2 concentration in the room at the next time step by using the equation 4.27

(12) Calculate PI control signal for current error by using the PI subroutine PI(SP,PIKI,PIKP,PV,PISG,INI) by using Equation 4.12 and 4.13

(13) Save the results for current control signal to a output file.

(14) Increase the time by 1 minute steps until the difference of initial and final time is greater than 30 minutes, then go back Step 9

(15) Next CO_2 concentration rate.

(16) Check the CO_2 concentration rate is back to initial value or not. If not, go back step 9. If true, stop.

(17) End.

A listing of the program, and sample calculations, are given in Appendix II.

Figures 4.2 and 4.3 present simulated results for the system with two different initial condition: for the first the initial room CO_2 concentration is at 400 ppm, for the second the initial room CO_2 concentration is 1400 ppm. The CO_2 generation rate was increased and decrease in 5 cfh increments. The simulations were performed for a two damper system with area compensation. The results show that the control system will operate satisfactorily.

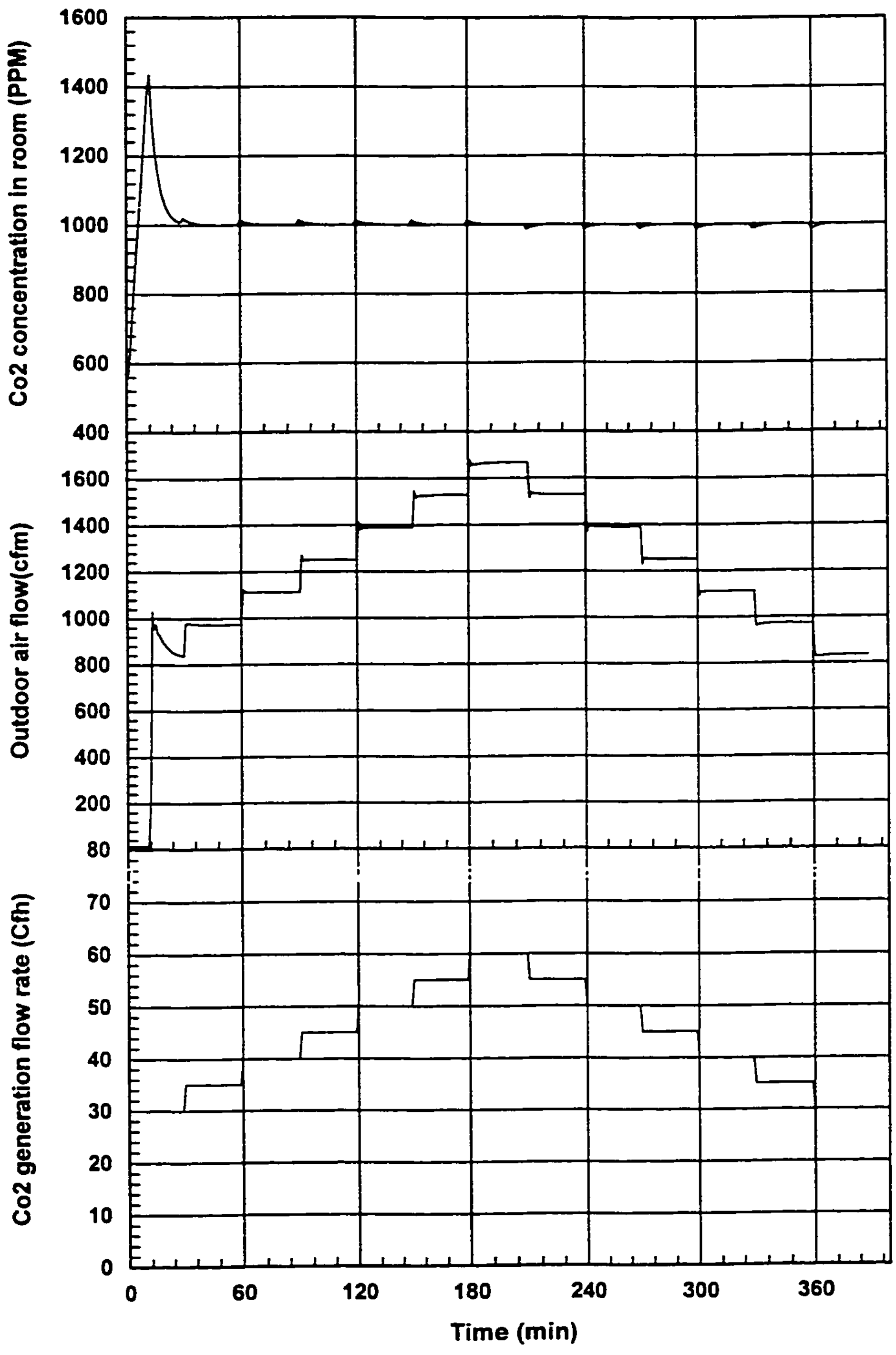

Figure 4.2: Demand control system simulation results for an initial room CO_2 concentration of 400 ppm.

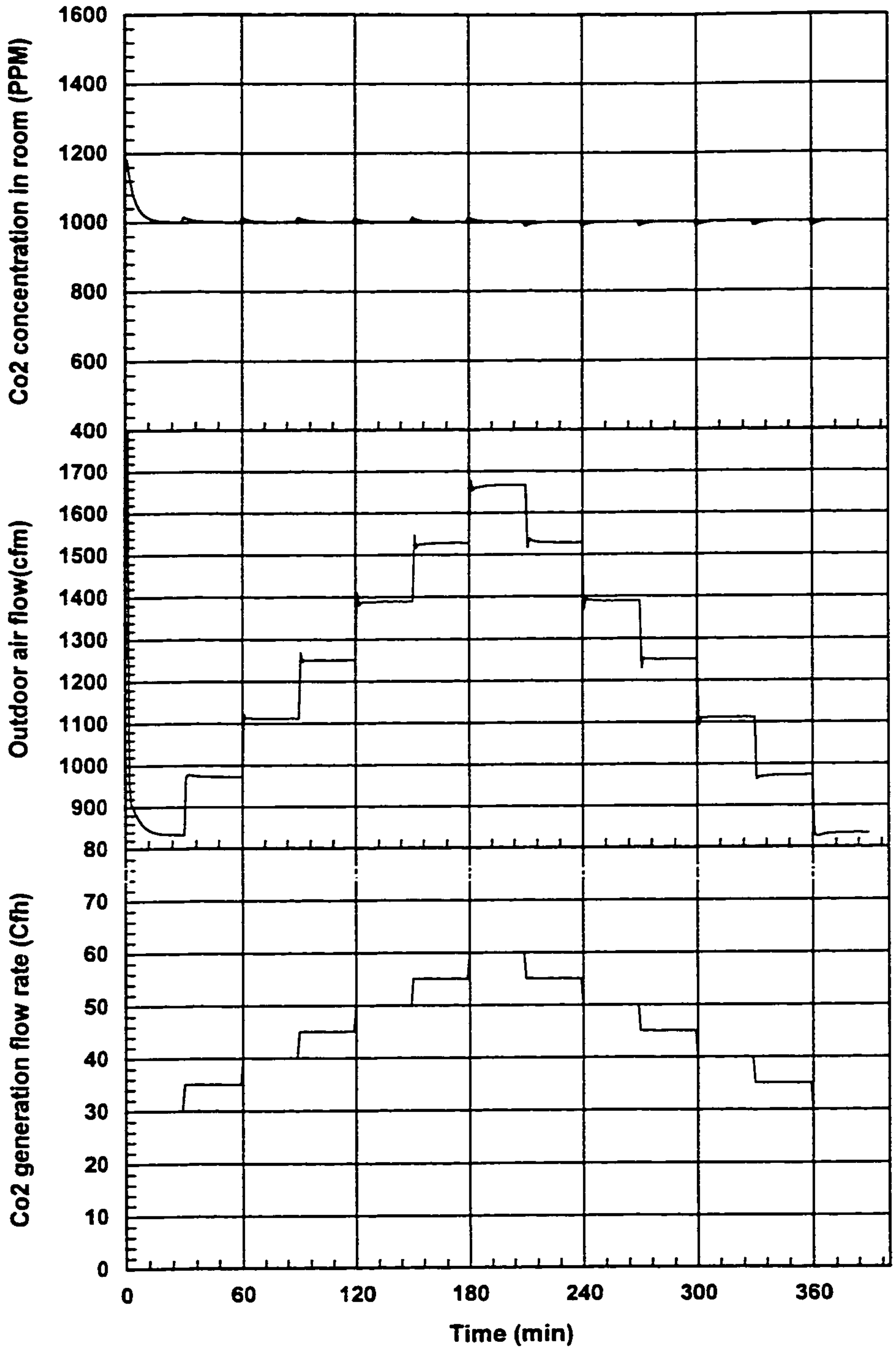

Figure 4.3: Demand control system simulation results for an initial room CO_2 concentration of 1400 ppm.

5. EXPERIMENTAL INVESTIGATIONS

A experimental air distribution system with an economy cycle was built in our laboratory. The schematic diagram of the system is shown in the Figure 5.1.

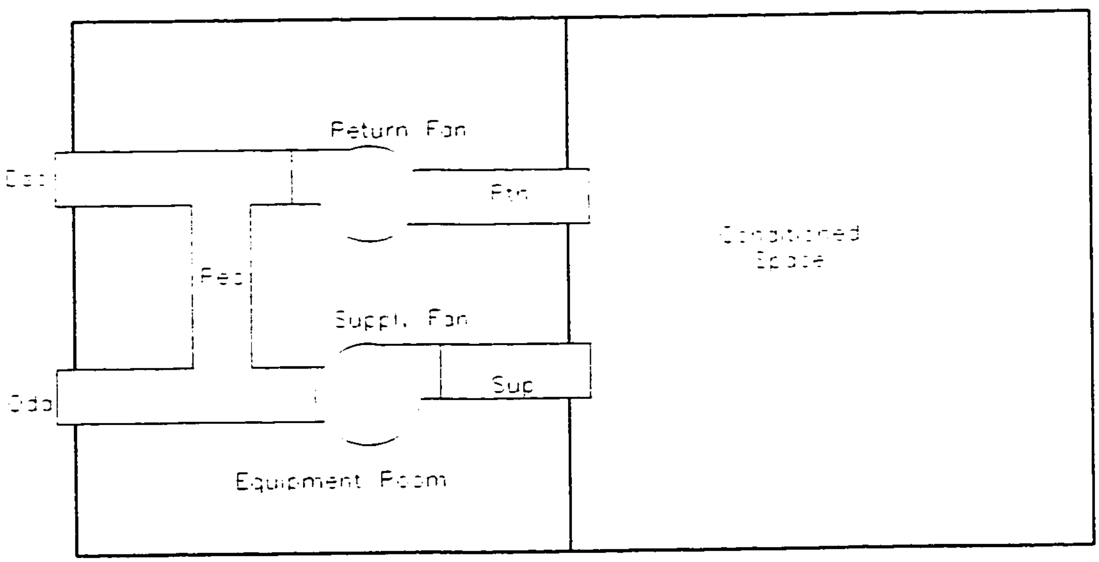

Figure 5.1: The air distribution system with a economizer

The experimental system in Figure 5.1 includes five dampers - the outdoor air (Oda) damper, the discharge (Dsc) damper, the recirculation (Rec) damper, the supply duct (Sup) damper, and the return duct (Rtn) damper - and two variable speed fans - the supply fan and return fan. The controlled environment space is a single room. The air distribution system is located adjacent to the controlled room. It is a simple air distribution system without heating or cooling coils. The system provided the necessary ventilation air flow into the room to maintain the room pressure at constant value (0.0 inWc). It controlled the room CO_2 concentration at a specification level of 1000 ppm CO_2 by varying the mixing ratio of fresh outdoor air and recirculation air. Various control strategies were studied with this system to achieve the minimum energy use during the operation.

All dampers are operated by positional motors (actuators) having a rotational position proportional to a control signal. The supply and return duct dampers are fixed at certain positions (i.e., percentage open) at all times. These two dampers provided additional resistance in the ducts to simulate resistance in an actual system with long ducts. The outdoor air damper, recirculation damper and discharge damper positions were variables and coupled in order to vary the mixing ratio of the fresh air and recirculated air flow. The outdoor damper and discharge damper were normally closed while the recirculation damper is normally open in the power-off condition.

Conventionally, the three dampers are controlled by same PI controller and varied in a couple of way, i.e., the outdoor damper, discharge damper and recirculation damper motors receive the equal control signals, but the recirculation damper is operated in a different direction with other two dampers. For example, if the signal is 30%, the outdoor air damper and discharger dampers are to be opened ~30% (i.e., ~70% closed) because they are normally open and the recirculation damper will open closed ~30% (i.e., opened ~70%) because it is normally closed. In this paper, this control strategy is called a three coupled damper system. The damper percent open and closed are approximate because all damper characteristics are not linear. Furthermore, dampers have some hysteresis, which means that there is one position curve for increasing signals, and a slightly different curve for decreasing signals.

In the current experiments, two other controlling strategies were applied on the system. In the first, the outdoor air damper and recirculation dampers were coupled while the discharger damper was kept ~100% open at all times. For example, if the outdoor air damper was open ~30% (i.e., ~70% closed), the recirculation damper would open ~70% (i.e., ~30% closed), and the discharger damper be open ~100%. In this paper, it is called the two coupled damper system. In a second control strategy, a area compensation algorithm implemented by using software was added in the system for the two dampers system. By using the area compensation algorithm, the damper open area was approximately linearly proportional to the control signal response.

Several series of experiments related to the different control strategies were performed. The power consumption between the different control strategies was compared in order to determine the most economical one. The most economical control strategy was then applied in room air quality control tests.

5.1 The Experimental System

The controlled environment room is 26.5ft x 21ft x 12.8ft (7123 ft.^{3}) with well-sealed windows and door. The air infiltration or exfiltration was reduced to minimum and could be considered negligible in this experiment. As the basic components of the experimental air distribution system, two single inlet, 13 inch wheel diameter, backward inclined blades fans were used. They were controlled to provide 2600 cfm circulation air flow. The fan outlet area was 1.04 ft.^{2}. The supply fan speed was controlled with a PI control algorithm and varied from 800 to 2200 rpm to maintain the supply duct pressure at a constant value. The return fan was also controlled with another PI control algorithm to maintain the room pressure at a constant value.

Between the supply fan and the controlled environment room, an opposed blade damper (16″ X 16″) with fixed opening was set in the duct line to provide a constant supply duct resistance. A return damper (17″ X 24″) was installed between the return fan and the controlled environment room to simulate a long return duct resistance. The outdoor damper, recirculation damper and discharge damper were all 18″ X 18″ opposed blade dampers. They were coupled and controlled by a PI controller.

The following five key point pressures in the system ducts were measured for the purposes of control and analysis:

supply fan inlet pressure (SfiPrs),

supply fan outlet pressure (SfoPrs),

return fan inlet pressure (RfiPrs),

return fan outlet pressure (RfoPrs), and

room pressure (RmPrs).

The above duct static pressure were measured by pitot tubes connected to the pressure transducers. The tube were 1/4 inch diameter and had 8 holes equally space to obtain average static pressure for a cross section of the duct. The supply duct pressure and room pressure were two controlled points having set points of 0.5inWc and 0.0 inWc, respectively.

Carbon dioxide meters were located in inlet of the outdoor air duct (upstream of the outdoor air damper), and in the inlet of return air duct.

There were three flow meters in the system. One air flow meters was located at upstream of the outdoor damper; The second one was located at upstream of supply fan inlet. The third one was located at downstream of the discharge damper. Unfortunately, in our experimental system, only the outdoor air flow meter was accurate because the measurement positions of the other two meters were too close the damper or T-connection, resulting in non-uniform flow profiles and excessive measurement error. Other position locations were not practical because of the short ducts. The flow measurement were obtained by the electronic air flow meters, a type of electric hot wire probe air meter. The heat transfer from a heated resistor (hot wire) is proportional to air velocity. The meter was a two-point sensor meter.

Although two fan motor power meters were used in the system, the system fan energy consumption is obtained by calculating the fluid power. The reason is that the two fan motors were operating at the bottom of the power meter ranges and accurate power value could not be obtain from the system power meters. Each power meter was rated at 20 kW. Neither fan power consumption exceed 1. kW. Furthermore, the fan motor power was dependent on the fan selection, e.g. 13″ or 15″ wheel fans in the system, fluid power is virtually independent on the fan selection. The fan fluid power was calculated with Equation 3.39.

All measurement signals were collected by a data logger and transferred to the central computer. The data logger contained analog-to-digital (A/D) and digital-to- analog (D/A) converter to exchange the signals from sensors to computer and to the actuators. All of PI algorithms were achieved by the computer software. The computer also served as a data collection and storage for saving all experimental data, and as system performance monitor.

5.2 The Control Program

Damper usually have some hystereses. This means that there may be two flow rates for a given control signal. Proper control requires that there is only flow rate for a given control signal. Hystereses effects may be reduced as detailed by Krakow [20]. This method was used in the experimental system.

The control program has several unique subroutines for area compensation, PI control, and hystereses reduction. Only these subroutines written in Basic are listed in Appendix III.

5.3 Experimental Results

All experiments were performed during the summer. Based on the controlling strategies discussed before, the experiments were divided into three group. Two series tests for each group were performed for the purposes of checking repeatability. Before each test, all dampers were preset by the software in order to confirm the dampers starting at same initial position.

During the tests, the two fans were operating under the control of individual PI control loops to maintain the stable, specified system operating conditions. The outdoor damper, recirculation damper and discharge damper would be varied in a coupled manner from an initial position to a final position, and then back to the initial one with series of steps. Between each step, the system was maintained 10 minutes to reach stable conditions. The measurement data was collected continually by 1 minute intervals.

The following three controlling strategies were tested:

- conventional three coupled dampers system,
- two coupled dampers system with the discharger damper 100% open,

- two coupled dampers system with an area compensation algorithm.

Two ranges of the damper control signal were investigated:

- 25% to 100% with variable (5% and 10%) signal increments, and
- 20% to 85% with constant (5%) signal increments.

Outdoor air flow rate, pressures, and fluid power were determined for each control test series. The test series were designated 3.1, 3.2, 2.1, 2.2, and 2.3. Three tests were performed for each series to check system repeatability. Series 3.1 and 3.2 yielded baseline reference data for the comparison of the various control strategies. The results obtained are shown Figures 5.2 through 5.16 as per Table 5.1.

Table 5.1: Summary of Tests

Test series	Strategy	Oda signal range	Oda flow rate	Pressure	Power
3.1	three coupled dampers	25% to 100%	Figure 5.2	Figure 5.7	Figure 5.12
3.2	three coupled dampers	20% to 85%	Figure 5.3	Figure 5.8	Figure 5.13
2.1	three coupled dampers	20% to 100%	Figure 5.4	Figure 5.9	Figure 5.14
2.2	two coupled dampers	20% to 85%	Figure 5.5	Figure 5.10	Figure 5.15
2.3	two coupled dampers, area compensated	20% to 85%	Figure 5.6	Figure 5.11	Figure 5.16

5.3.1 Flow rates

Figure 5.2 and 5.4 indicate that the flow rates will not be controlled properly with outdoor air damper control signals greater than 85%. Figures 5.3, 5.5, and 5.6 indicate that the control of flow rate is satisfactory with outdoor air damper control signals less than, or equal to, 85%. Figure 5.2 and 5.4 indicate that the maximum supply fan flow rate is 2600 cfm. This flow rate, obtained with the recirculation damper fully closed (100% signal), is considered the supply fan flow rate at all times since the pressure difference across the supply duct damper and the supply duct damper position are constant at all times. The variation of outdoor air flow rate at a given signal is due to random noise in the instrumentation system as well as the turbulent nature of the flow. A hysteresis correction algorithm was incorporated in the control algorithm.

As mentioned before, since the controlled room infiltration, or exfiltration is negligible, the system return duct air flow equaled the supply one with the recirculation damper closed (100% signal), being the 2600 cfm at all run.

5.3.2 Pressures

A comparison from Figures 5.7 to 5.12 shows that the major difference between the pressure characteristics of the different control strategies is the magnitude of the return fan outlet pressure. Figures 5.7 and 5.8 (series 3.1 and 3.2) indicate that the maximum outlet pressure of the return fan was 0.7 inWc, occurring at a 50% signal. The outlet pressure curves in Figure 5.8 to Figure 5.11 (series 2.1, 2.2 and 2.3) for the two damper coupled system show that the maximum pressure is approximately 0.1 inWC after the 80% signal. In addition, the pressure variation is just in the 0 to 0.1 inWc range, therefore, this system is better for control than the three coupled damper system and is more stable.

If the return fan outlet pressure would drop below 0.0 inWc in some case, the discharge damper may be closed by using the PI control to keep it at a set point of 0.01 inWc.

5.3.3 Powers

Figures 5.13 to Figure 5.17 indicate that the total fan flow power of the two coupled damper system is reduced comparing with the three coupled dampers system. The major difference between them is the magnitude of the return fan power. Because the return fan outlet pressure for the two coupled damper system is lower than the three coupled damper system (Figure 5.7 to 5.12), the result is that the less fan power is required. Moreover, most of the fan power reduction occurs at lower outdoor air flow rate range. This is very important for energy savings during heating season. The Figure 5.15 and 5.16 show that the fan power consumption has a significant reduction if the area compensation is applied in the two coupled damper system.

Tables 5.2 to 5.5 give the summary of the total energy consumption in a certain period for each system and provide the energy consumption percentage saving between of them. The energy is calculated by following equation:

$$E = \sum W_i \cdot \Delta t_i \tag{5.1}$$

Table 5.2: Fan Energy Consumption for Three and Two Damper Systems (Series 3.1 and 2.1)

Test	Three Coupled Dampers, kJ	Two Coupled Dampers, kJ	% for saving
1	3.27	3.13	4.5%
2	3.34	3.10	7.2%
3	3.39	3.06	9.8%
Avg	3.33	3.09	7.2%

Table 5.3: Fan Energy Consumption for Three and Two Damper Systems (Series 3.2 and 2.2)

Test	Three Coupled Dampers, kJ	Two Coupled Dampers, kJ	% for saving
1	3.97	3.65	8.0%
2	3.89	3.55	8.9%
3	4.04	3.54	12.4%
Avg	3.97	3.58	9.79%

Table 5.4: Fan Energy Consumption for Two Coupled Dampers with and without Area Compensation System (Series 2.2 and 2.3)

Test	Two Coupled Dampers, kJ	Two Coupled Dampers, area compensated, kJ	% for saving
1	3.65	2.77	24.1%
2	3.55	2.71	23.5%
3	3.54	2.82	20.4%
Avg	3.58	2.77	22.7%

Table 5.5: Fan Energy Consumption for Three Coupled Dampers and Two Coupled Dampers with Area Compensation System (series 3.2, 2.3)

Test	Three Coupled Dampers, KJ	Two Coupled Dampers, area compensated, KJ	% for saving
1	3.97	2.77	30.2%
2	3.89	2.71	30.3%
3	4.04	2.81	30.3%
AVG	3.97	2.77	30.3%

In the Tables 5.2 and 5.3, the result show that the 2 coupled damper system reduces average about 7.2% energy comparing with the three coupled damper system. The table 5.4 gives the energy consumption for the 2 coupled damper system with and without area compensation. It shown the system with area compensation save average about 22% power comparing the two damper system without area compensation. Moreover, in the Table 5.5, if the two coupled damper system added the area compensation, the energy consumption will reduce up to 30% comparing with three coupled damper system.

Whereas the two coupled damper system with area compensation is the most economical operating strategy, it was applied to an indoor air quality (IAQ) control system. Based on this result, in the IAQ control system investigated, only outdoor damper and recirculation dampers were used to modulate the ratio of fresh and recirculation air flow.

Figures 5.17 and 5.18 show the performance of experimental IAQ system, which was operating under the most economical strategy of two coupled damper with area conpemsation. The CO_2 concentration of the controlled environment room was controlled at a set point of 1000 ppm. The CO_2 was added to the room from a pressurized tank to simulate the CO_2 generation by occupants. The CO_2 emission rate to the room was varied from 0 cfh to 110 cfh and then back to 0 cfh in steps of 5 cfh at 30 minutes intervals, according to a manual flow meter. The CO_2 emission rate shown in Figures 5.12 and 5.13 was calculated from data recorded by the data logger assuming steady state conditions as per Equations 4.1 and 4.2. The CO_2 emission rate of 110 cfh is equivalent to the CO_2 produced by about 166 people. Initially, the minimum outdoor air was supplied because the CO_2 concentration was below the set point. After initial stabilization, the CO_2 concentration in the room was controlled by the PI controller within less than 50 ppm of the set point. The results in the Figures 5.17 and 5.18 show that the control strategy yielded satisfactory control result.

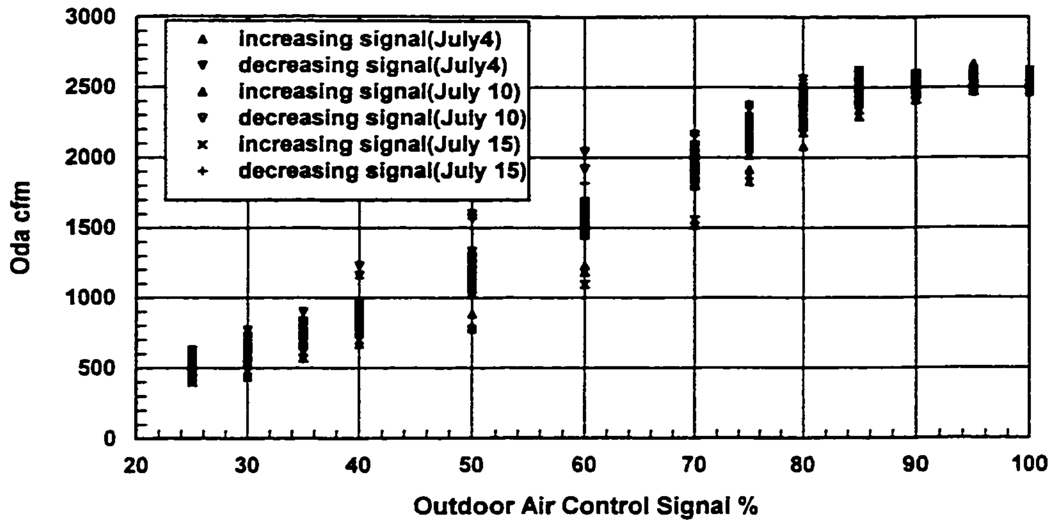

Figure 5.2: Experimental outdoor air flow rate characteristics for a three coupled damper system - Test Series 3.1.

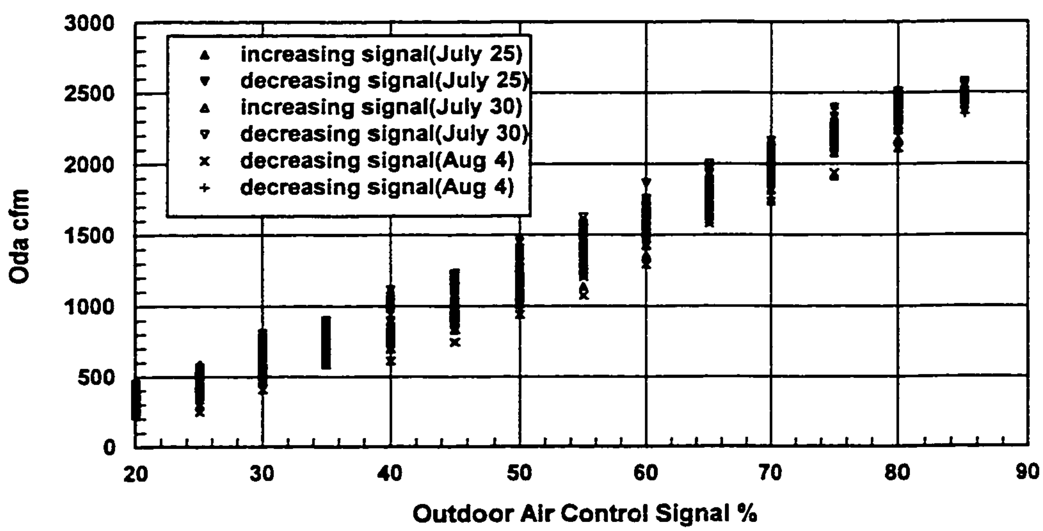

Figure 5.3: Experimental outdoor air flow rate characteristics for a three coupled damper system - Test Series 3.2.

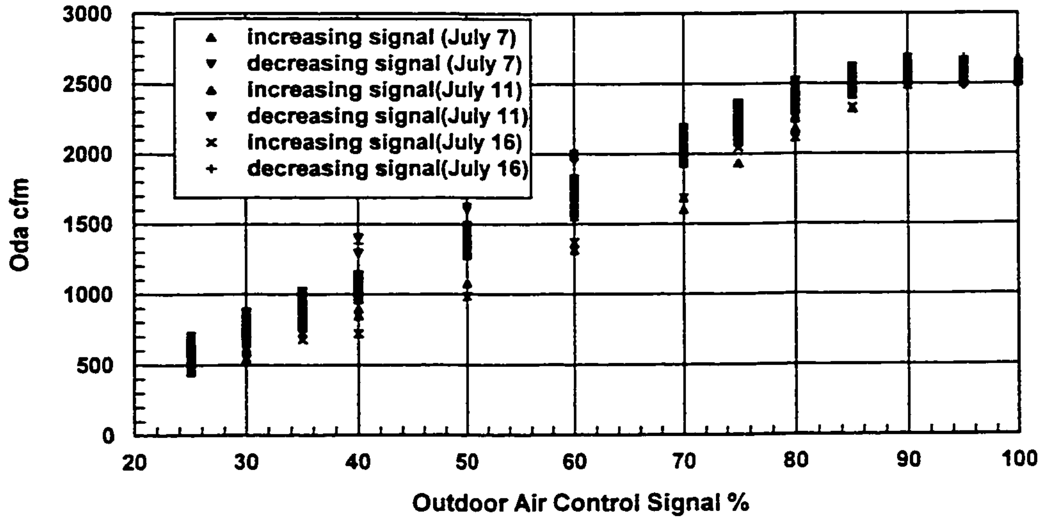

Figure 5.4: Experimental outdoor air flow rate characteristics for a two coupled damper system - Test Series 2.1.

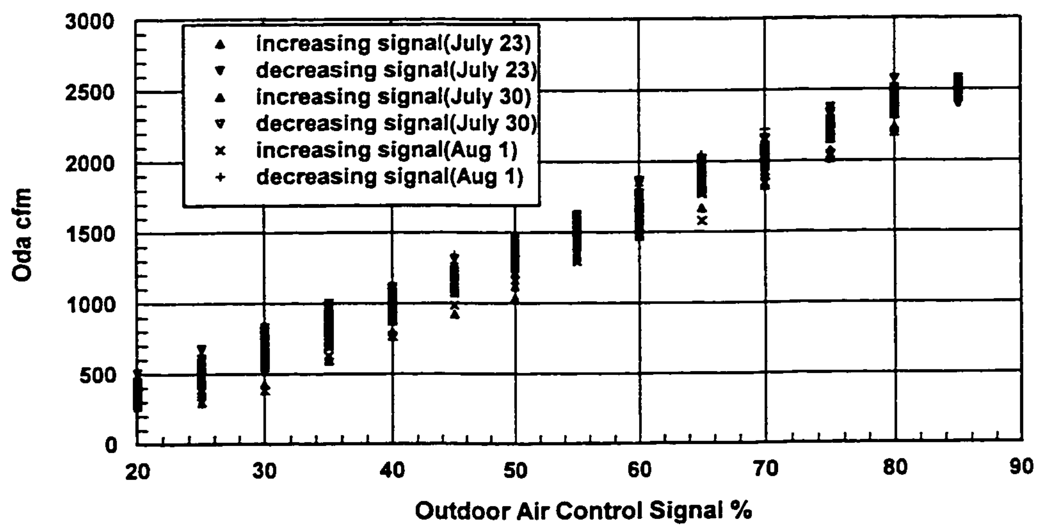

Figure 5.5: Experimental outdoor air flow rate characteristics for a two coupled damper system - Test Series 2.2.

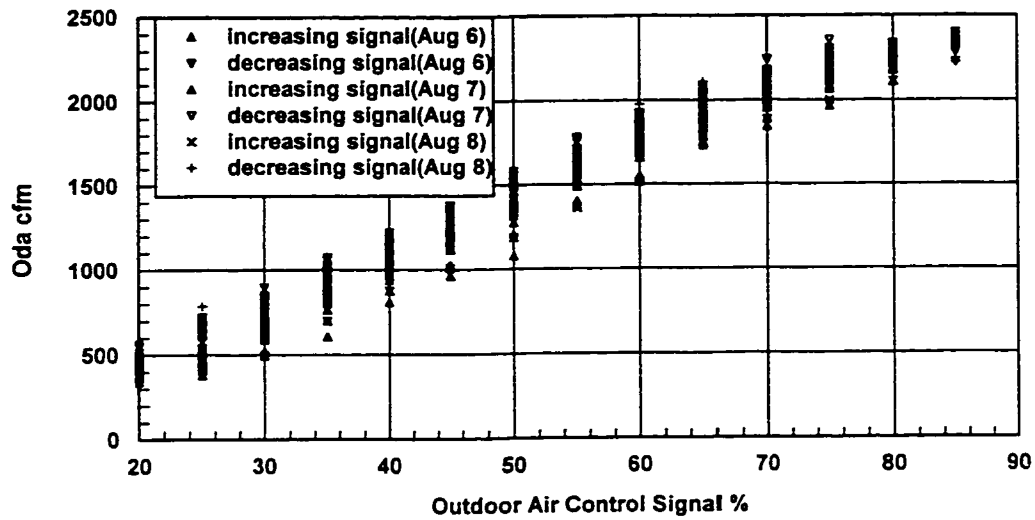

Figure 5.6: Experimental outdoor air flow rate characteristics for a two coupled damper with area compensation system - Test Series 2.3.

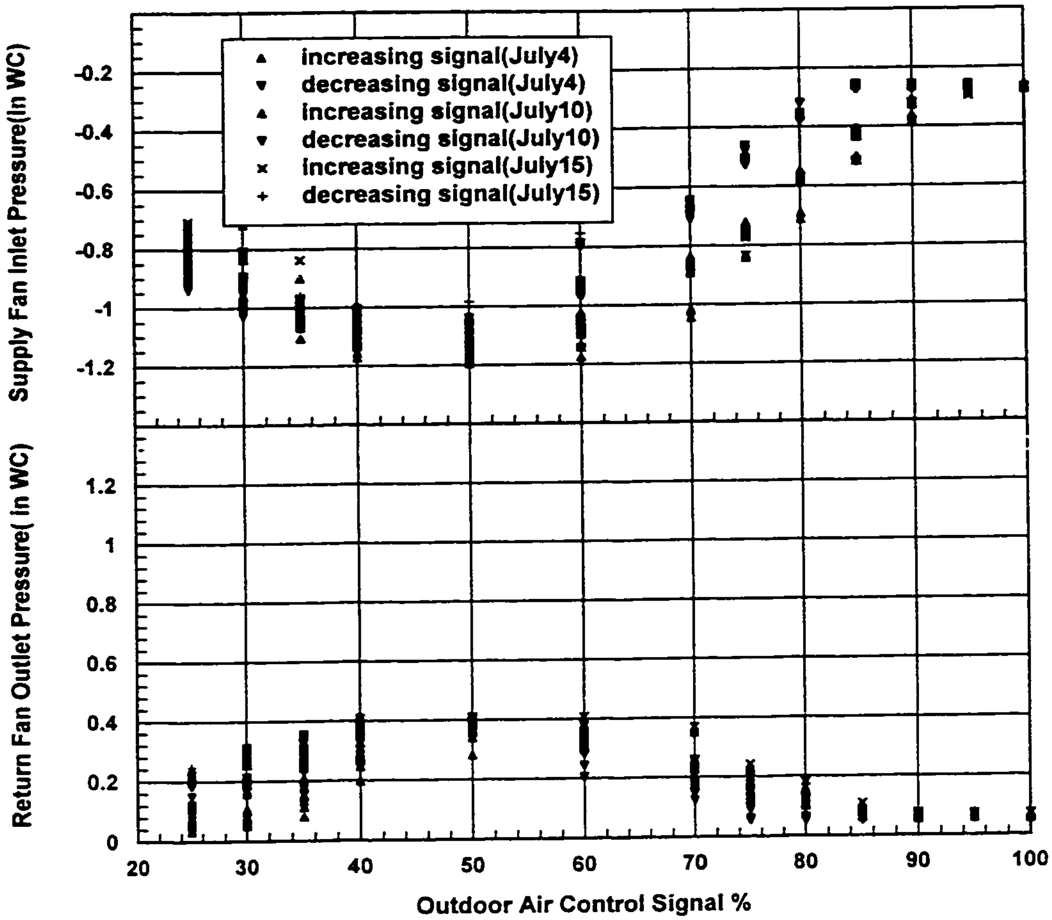

Figure 5.7: Experimental pressure characteristics for a three coupled damper system - Test Series 3.1.

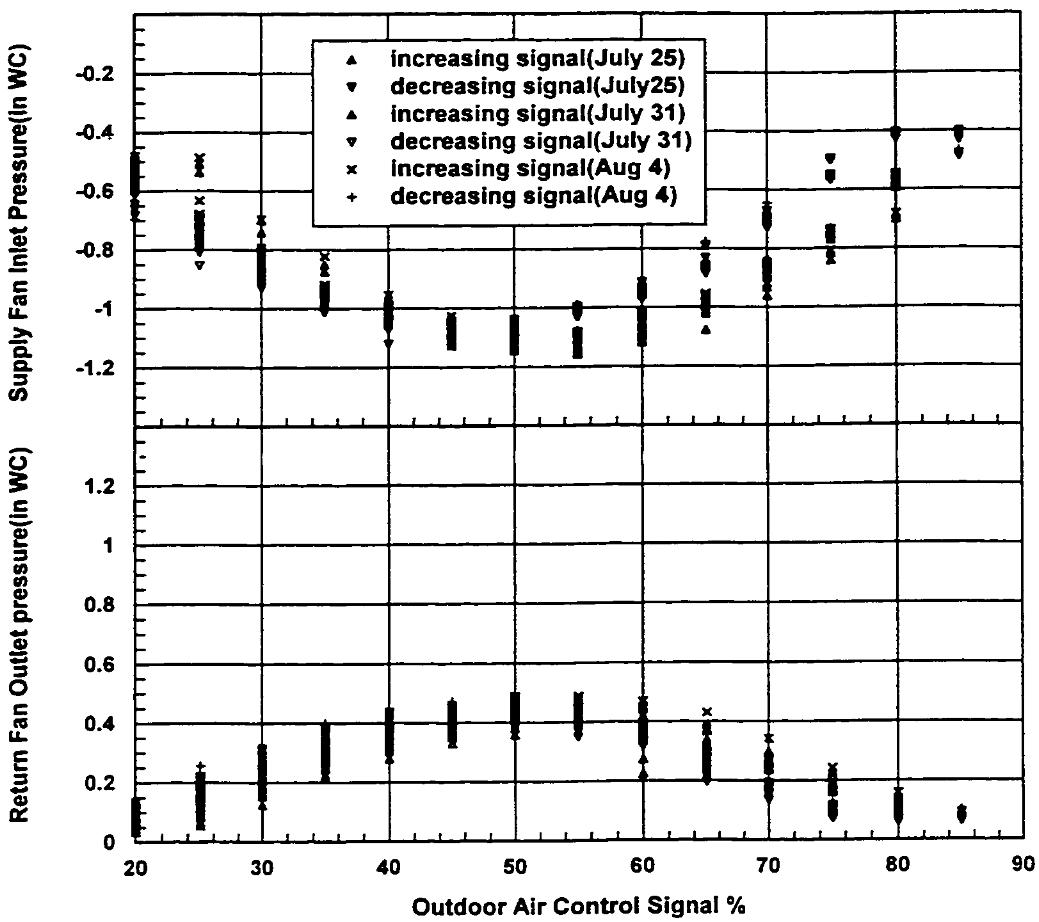

Figure 5.8: Experimental pressure characteristics for a three coupled damper system - Test Series 3.2.

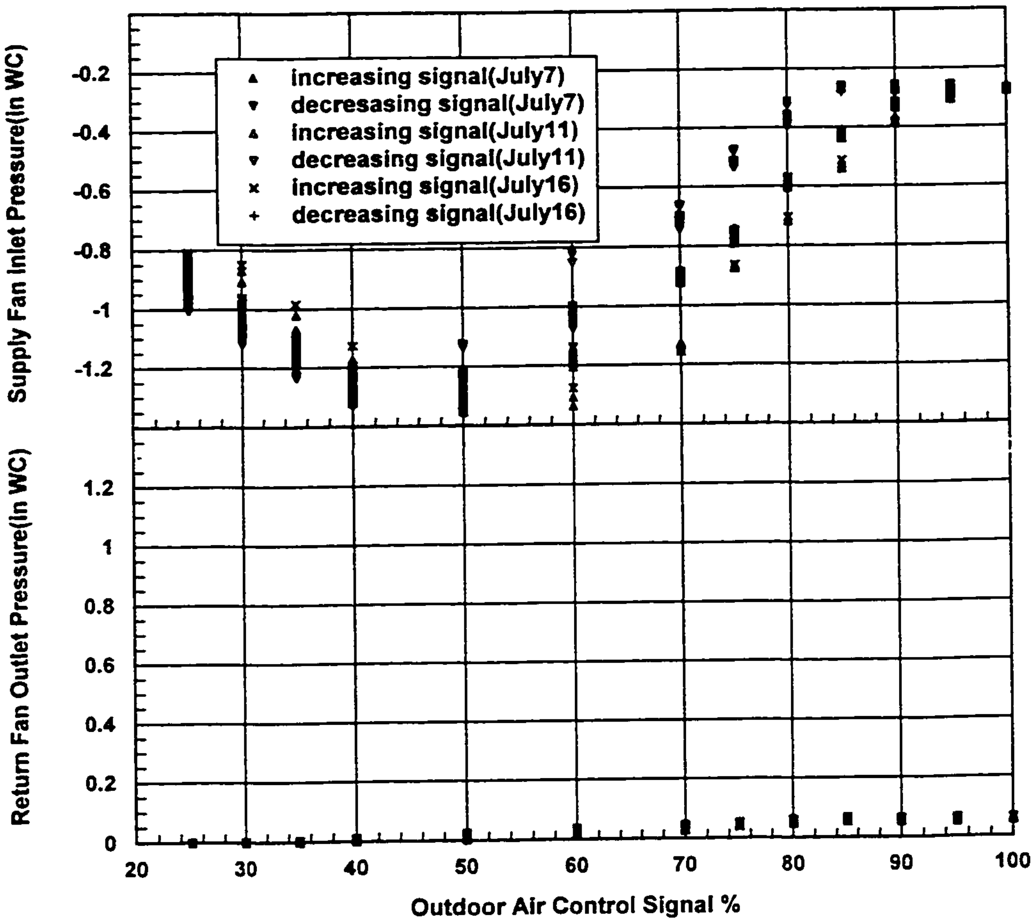

Figure 5.9: Experimental pressure characteristics for a two coupled damper system - Test Series 2.1.

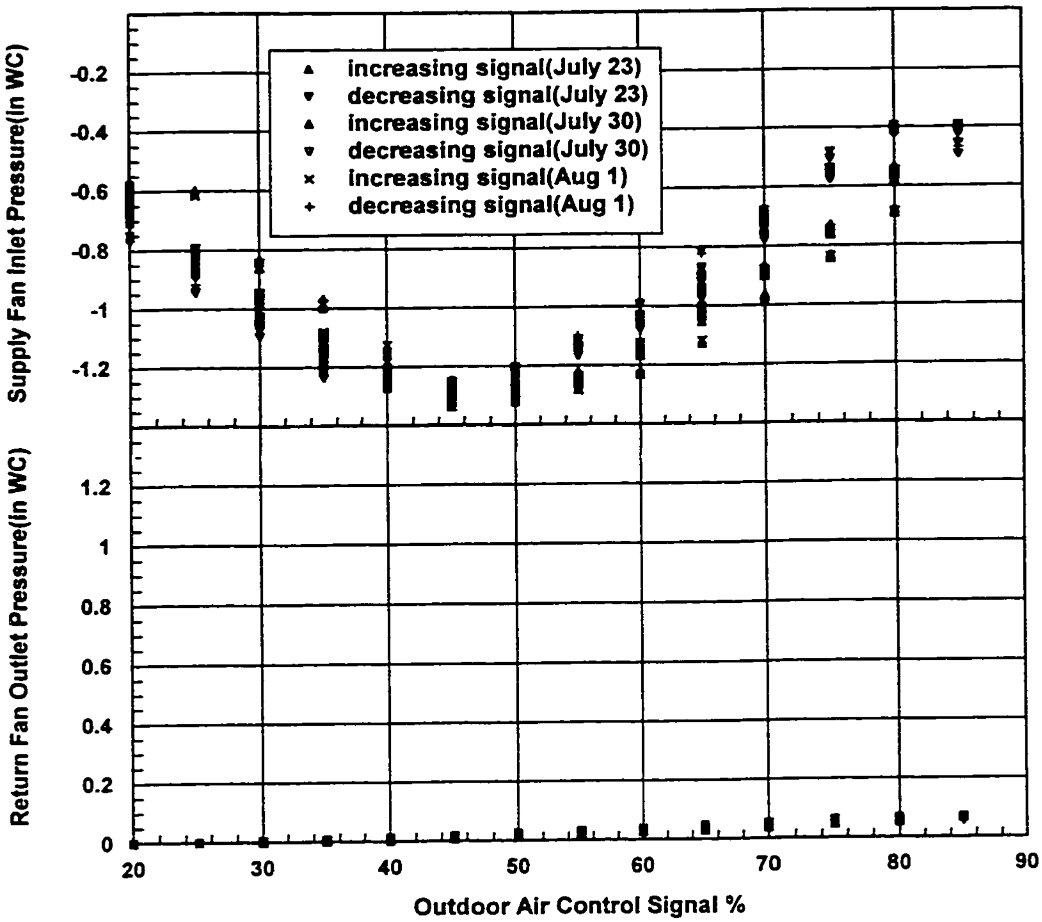

Figure 5.10: Experimental pressure characteristics for a two coupled damper system - Test Series 2.2.

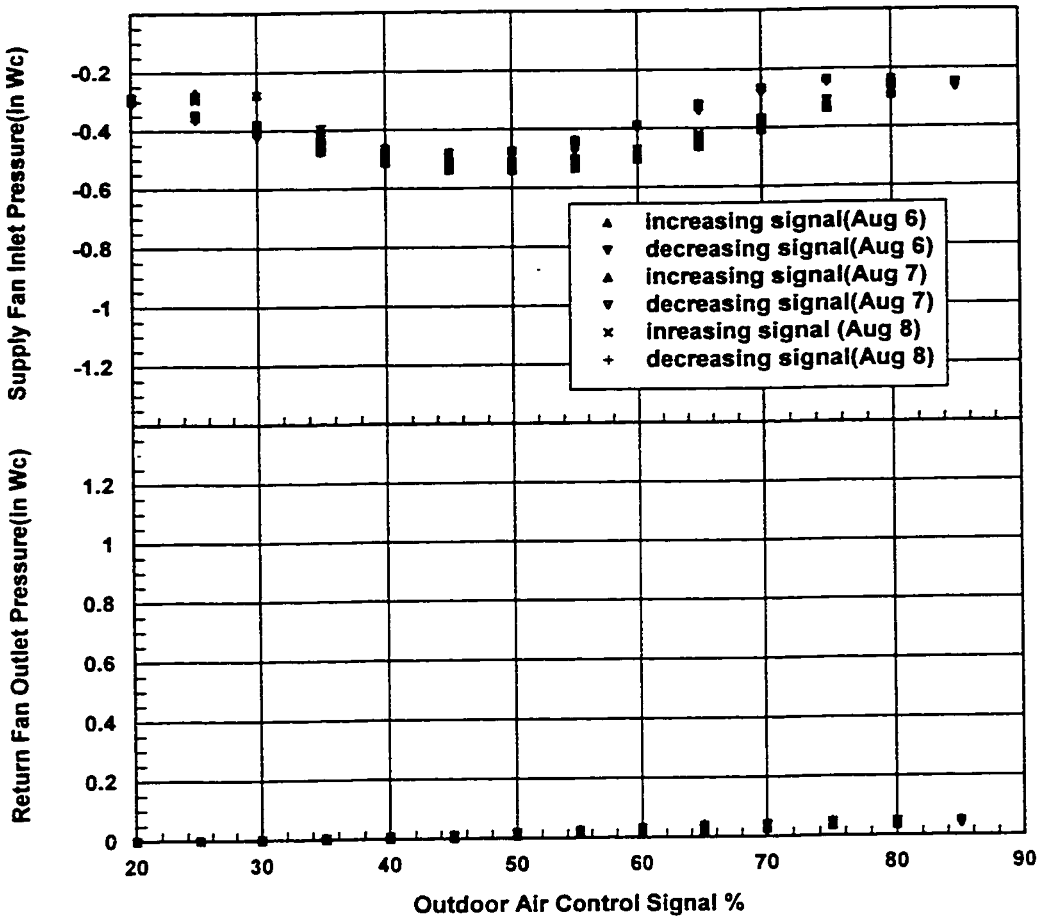

Figure 5.11: Experimental pressure characteristics for a two coupled damper system with area compensation - Test Series 2.3.

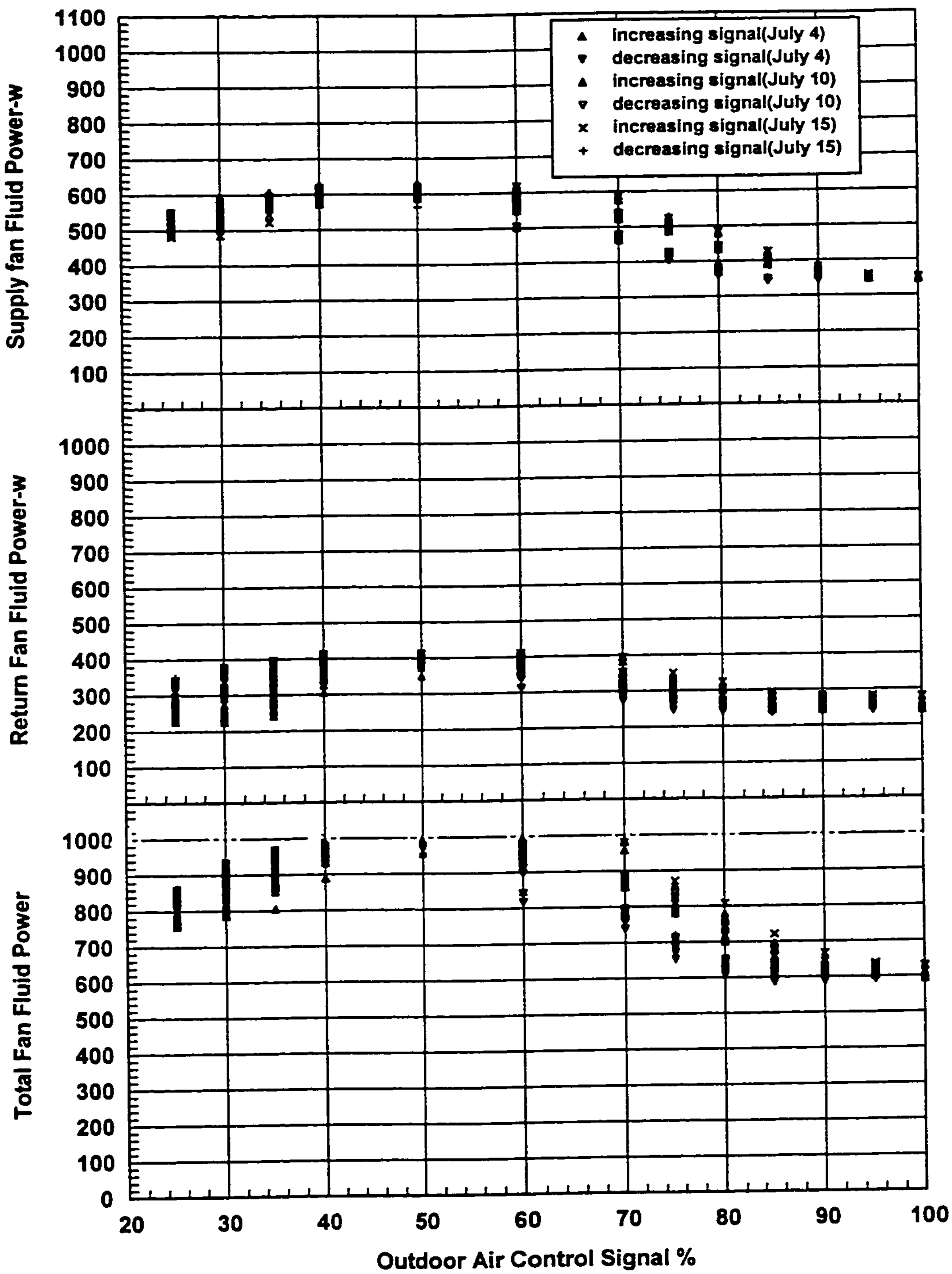

Figure 5.12: Experimental power characteristics for a three coupled damper system - Test Series 3.1.

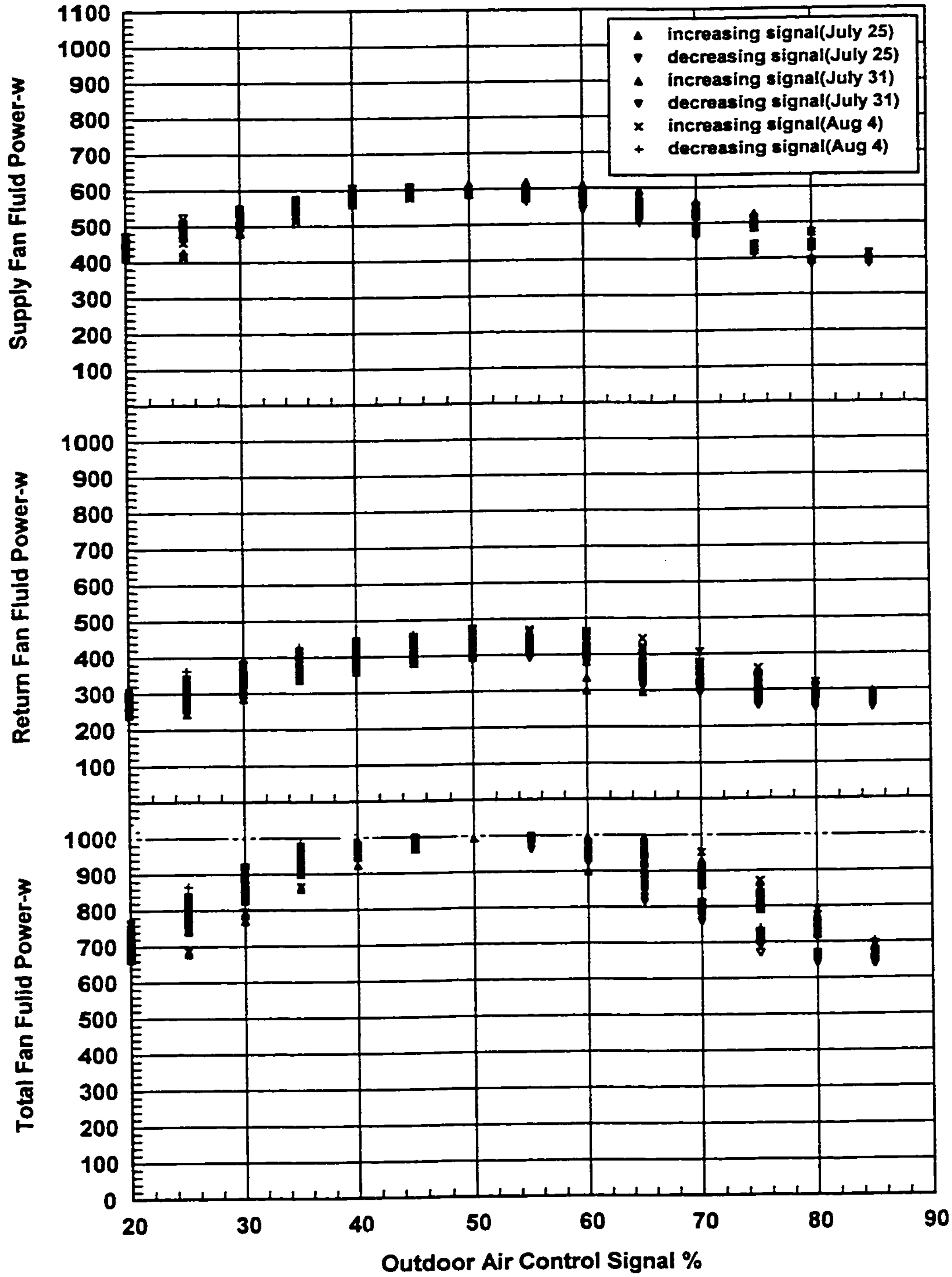

Figure 5.13: Experimental power characteristics for a three coupled damper system - Test Series 3.2.

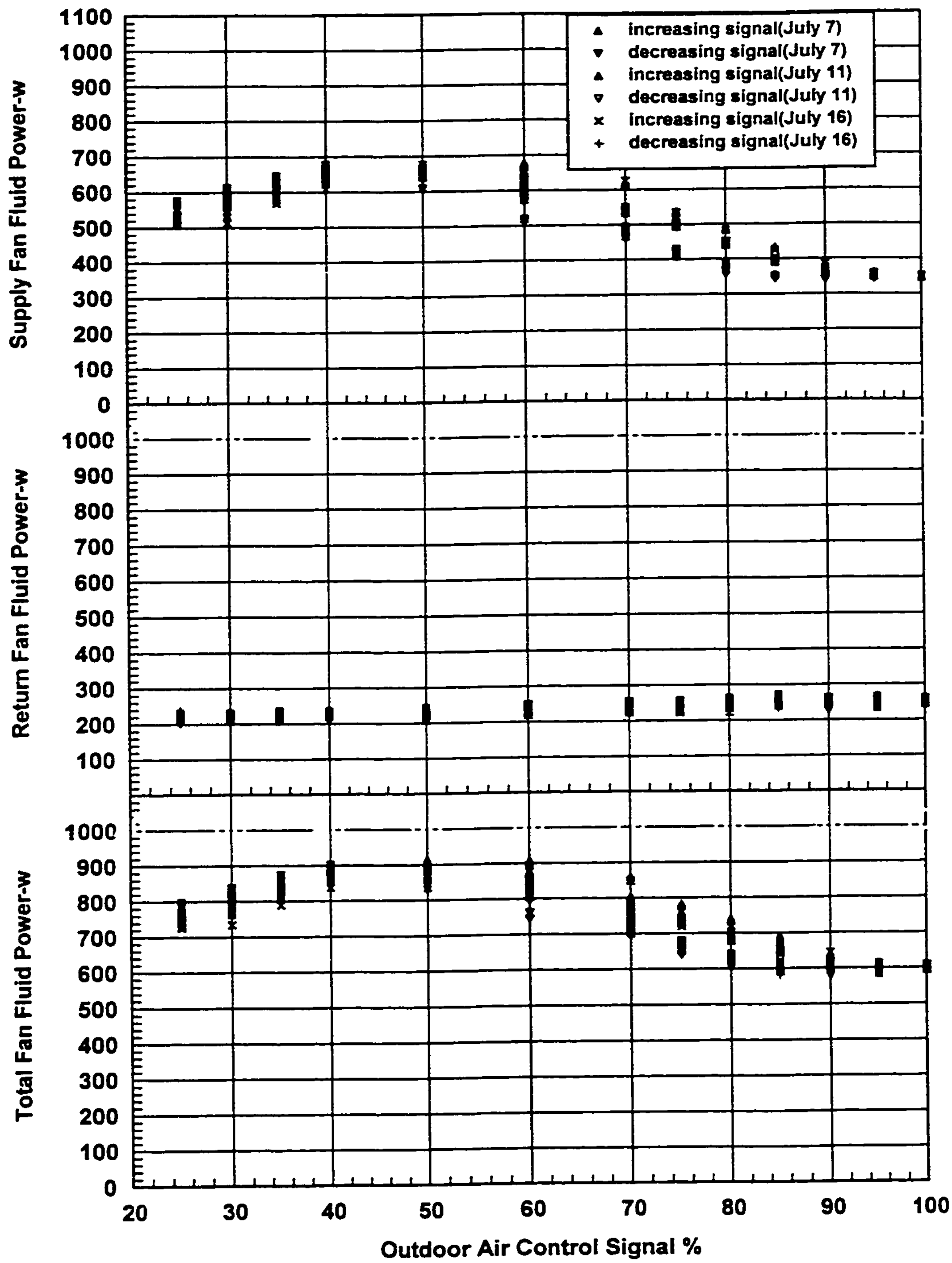

Figure 5.14: Experimental power characteristics for a two coupled damper system - Test Series 2.1.

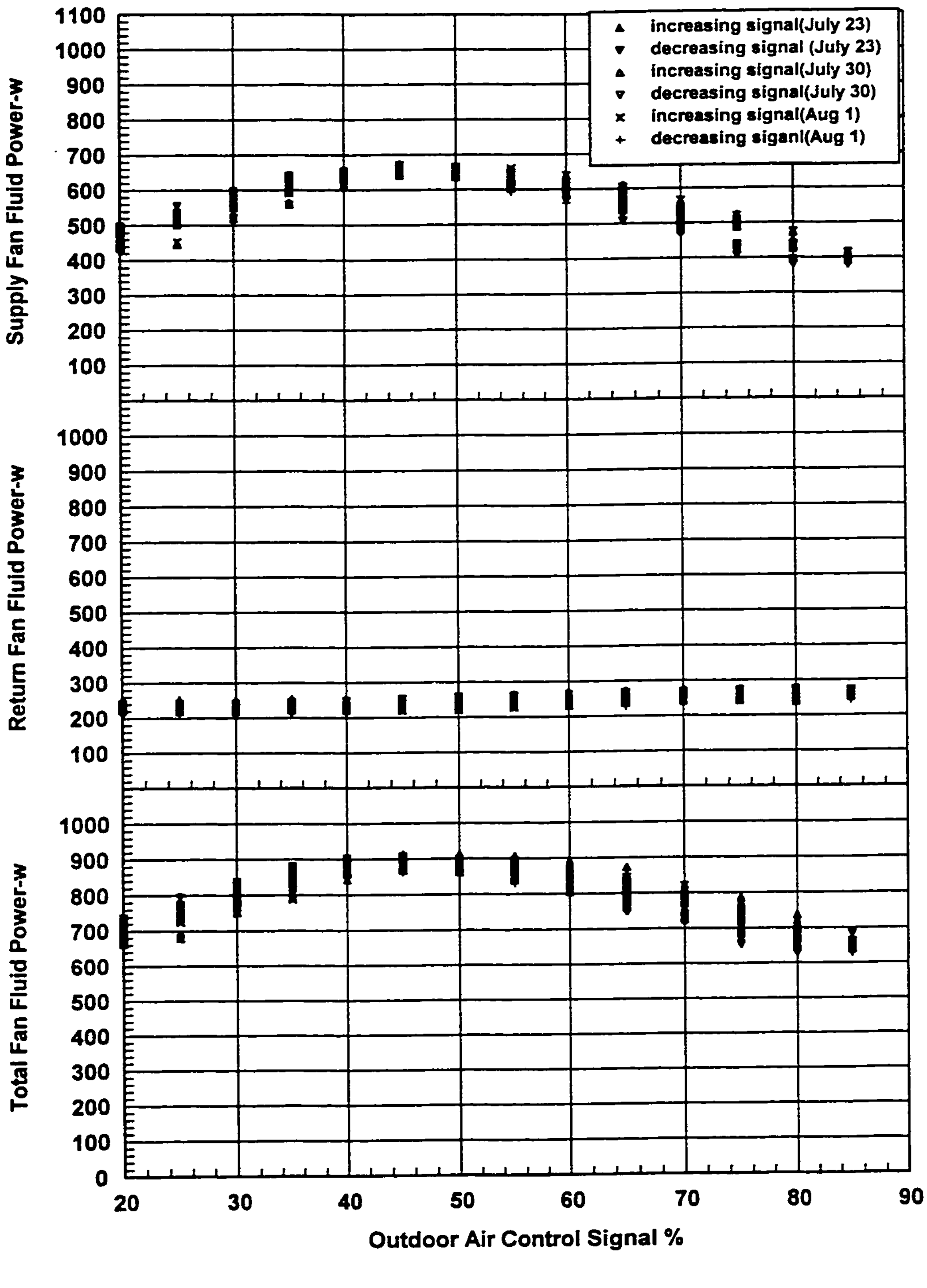

Figure 5.15: Experimental power characteristics for a two coupled damper system - Test Series 2.2.

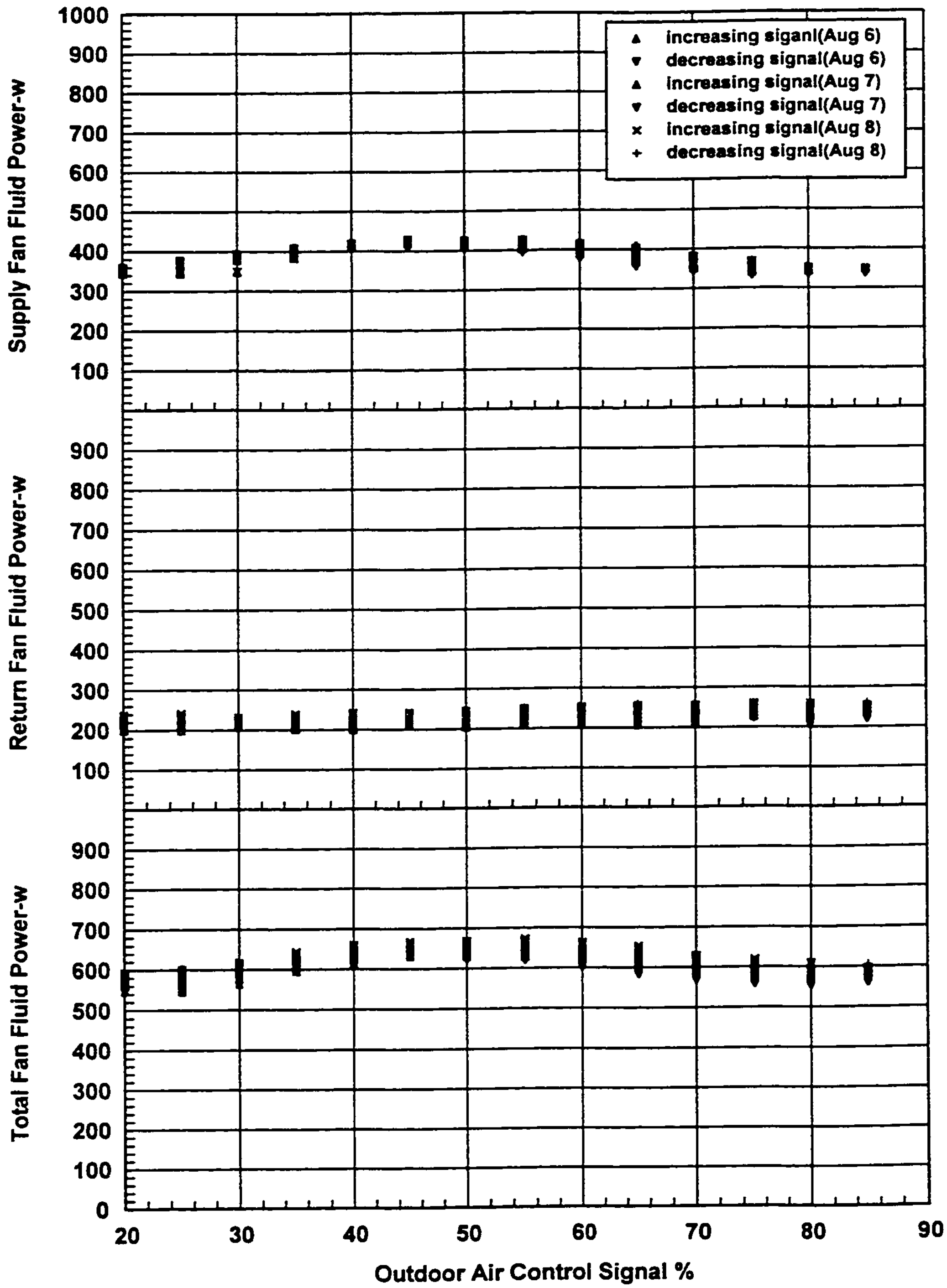

Figure 5.16: Experimental power characteristics for a two coupled damper system with area compensation - Test Series 2.3.

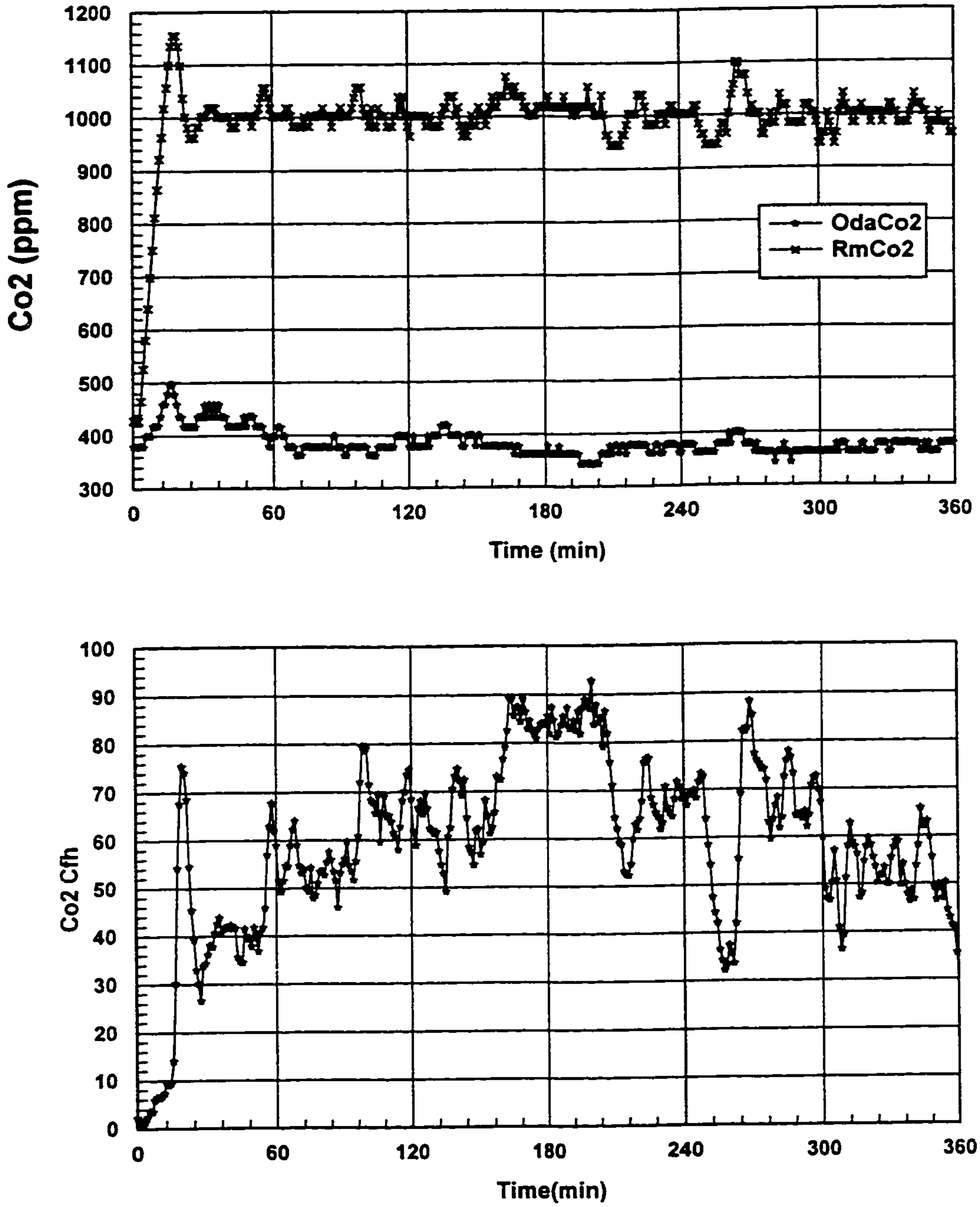

Figure 5.17: Experimental results for a demand control system for an initial room CO_2 concentration of 400 ppm.

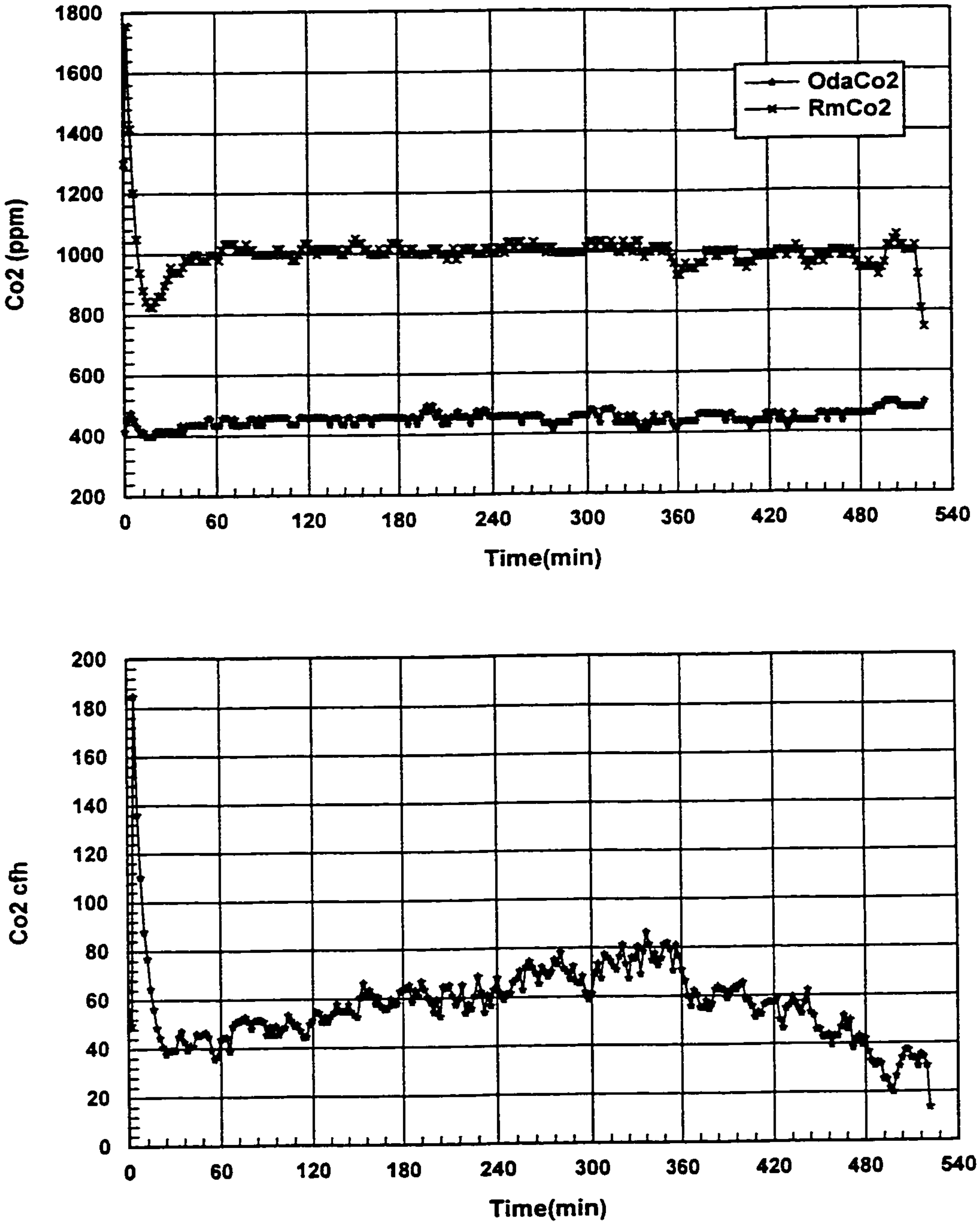

Figure 5.18: Experimental results for a demand control system for an initial room CO_2 concentration of 1400 ppm.

6. CONCLUSIONS

Control strategies for an economizer were investigated analytically and experimentally. The investigations showed that linearizing the damper response characteristic has a substantial positive effect on the fan power consumption. Power consumptions reductions of approximately 10% may be expected. The implementation of the linearization may be accomplished by means of software. Systems with two coupled dampers, the outdoor air and the recirculation dampers (leaving the discharge damper wide open during operation), have been shown to be more economical than the traditional three coupled damper systems. Therefore there is no capital equipment cost required to implement the energy saving strategy. Further energy savings may be implemented by using demand control. Proportional integral (PI) feedback control was shown to be suitable for such a control system. The amount of energy saved in a particular installation would depend on its occupancy schedule.

Linearized damper characteristics not only reduce fan power consumption by minimize supply fan inlet and return fan outlet pressure variations. This minimized the required fan speed variations to maintain the set point pressures constant and thereby contributes to good system control.

The numerical system model developed simulated system performance quiet accurately. It may easily be modified to apply to other systems by replacing the fan and damper models with those representing other components.

The control of the damper over a limited range - 20% to 85% - was sufficient to provide good control.

The following further investigations are indicated. The linearization algorithm used was based on thin, flat plate blades with no overlap. A more detailed linearization algorithm should be developed for optimum control. The system model should be incorporated with a building heating and cooling load model to enable estimation of energy saving for various occupancy schedules.

Proper control strategy can reduce both fan energy consumption as well as heating and cooling energy consumption. The implementation of a proper control strategy does not require capital equipment investment. It is therefore a desirable means of conserving energy.

7. REFERENCE

1. ASHRAE Handbook-Fundamentals 1993, I-P edition.
2. Belimo Aircontrols, Inc. "Damper Application Guide 1-Constant Volume Damper and Actuator Characteristic"
3. R.L. Alley. "Selecting and Sizing Outside and Return Air Dampers for VAV Economizer System", ASHRAE Transaction 1988, part 1, pp. 1457-1465.
4. G. Avery. "VAV Designing and Controlling an Outside Air Economizer Cycle", ASHRAE Journal, December 1986, pp. 26-30.
5. D.K.Dickson. "Control Dampers", ASHRAE Journal, June 1987, pp. 40-43.
6. K.I. Krakow. "Reduction of Power Loss Through Parallel Dampers" Internal report, Concordia University, 1997
7. Robert Van Bacelaere. "Mixing Box Damper Testing", ASHRAE Transaction 1998, V. 104, Part 2.
8. J.E Seem, J.M. House and C.J. Klaassen. "Leave the Outdoor Air Damper Wide Open", ASHRAE Journal, February 1998, pp. 58-60.
9. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.. "ASHRAE Standard 62-1989, Ventilation for Acceptable Indoor Air Quality", 1989.
10. J. Levenhagen. "Control System to Comply With ASHRAE Standard 62-1989", ASHRAE Journal, September, 1992, pp. 40-44.
11. D.C.Hittle. "Controlling Variable-Volune Systems", ASHRAE Journal, September 1997, pp. 50-52.

12. D.M. Elovitz. "Minimum Outside Air Control Methods for VAV System", ASHRAE Transaction, 1995, pp. 613-617.

13. S.A.Mumma and Y.M.Wong. "Analytical Evaluation of Outdoor Airflow Rate Variation vs. Supply Airflow Rate Variation in Variable-Air-Volume System When The Outdoor Air Damper Position Is Fixed", ASHRAE Transaction,1990,Volume 96,Part 1, pp. 1197-1208.

14. John P. Kettle. "Controlling Minimum Ventilation Volume In VAV Systems", ASHRAE Journal, May 1998, pp. 45-50.

15. G.J. Janu, J.D. Wenger and C.G. Nesler. "Outdoor Air Flow Control for VAV Systems" ASHRAE Journal April 1995, pp. 62-68.

16. G.J. Janu, J.D. Wenger and C.G. Nesler. "Strategies for Outdoor Airflow Control from A System Perspective" ASHRAE Transaction 1995, pp. 631-642.

17. K.I.Krakow, G.Hussein, S.Lin. "A Novel Theory for PI Control of HVAC&R Systems. Transaction of the CSME, Volume 22, No.2, 1998.

18. K.I.Krakow,S.Lin. "PI Control of Fan Speed to Maintain Constant Fan Discharge Pressure. ASHRAE Transaction,1995, Volume 101, Part 2, pp. 398-470.

19. K.I.Krakow. "Sample Interval Magnitude Relation to Digital PI System Response", ASHRAE Transaction, 1998, Volume 104, Part 2.

20. K.I.Krakow. "Reduction of Hysteresis in PI-Controlled System". ASHRAE Transaction,1998, Volume 101,Part 1.

21. K.I.Krakow. "Determination of PI Coefficients for Maximum Error Attenuation Rates", Internal report, Concordia University, 1998.

8. BIBLOGRAPHY

1. F.C. Mcquiston and J.D. Parker. "Heating, Ventilating, and Air Conditioning Analysis and Design, 1977. John Wiley &Sons. Inc.

2. M.B. Schell, S.C. Turner, R.O. Shim. "Application of Co_2-Based Demand-Controlled Ventilation Using ASHRAE Standard 62: Optimizing Energy Use and Ventilation. ASHRAE Transactions. 1998, Vol. 104, Part 2.

3. G.A. Johnson. "Retrofit of a Constant Volume Air System for Variable Speed Fan Control. ASHRAE Transactions. 1984. Vol. 90, Part 2B, pp. 201-212.

4. R.W. Haines. "Control System for Heating, Ventilating and Air conditioning". Second Edition, Van Nostrand Reinhold Company. 1977.

5. American Society of Heating, Refrigeration, and Air conditioning Engineers, "1993 ASHRAE Handbook of Fundamentals", 1993. Atlanta GA.

6. R.W. Haines. "Fan Energy - P vs. PI Control". Heating/Piping/Air Conditioning", August 1984, pp. 107-111.

7. G.V. Atkinson. "VAV System Volume Control Using Electronic Strategies". AHSRAE Transactions 1986, Vol. 92, Part 2B, pp. 46-57.

8. P.E. Wichman. "Improved Local Loop Control Systems". ASHRAE Transactions 1984, Vol. 90, Part 2B, pp.193-200.

9. H.P. Becker. "How Much Sense Do Room Occupancy Sensor Controls Make". ASHRAE Transactions 1986, Vol. 92, Part 2B, pp. 333-343.

10. T.E.Cappenllin. "VAV Systems-What Makes Them Succeed? What Makes Them Fail?". ASHRAE Transactions 1997, Vol. 103. Part 2.

11. M. Kasahara, T. Matsuba, Y. Hashimoto. "Optimal Preview Control for HVAC System". ASHRAE Transactions, 1998,Vol. 104, Part 1.

12. K.S. Maki, Z. Li, G. A. Chamberlin, L.L. Christianson. "VAV System Performance-Field Characterization of Airflow, System Diagnosis Tools, And Operational design implications". ASHRAE transactions, 1997, Vol. 103, Part 2.

13. J.P. Kettler. "Field problems Associated with Return Fans on VAV Systems". AHSRAE Transactions. 1988, Vol. 94, Part 1, pp. 1477-1482.

14. J.I. Levenhagen, D.H. Spethmann, "HVAC Controls and Systems". McGrraw-hill, Inc. 1993

15. Y.P. Ke, S.A. Mumma. "Using Carbon Dioxide Measurements to Determine Occupancy for Ventilation Controls", ASHRAE Transactions. 1997, Vol. 103, Part 2.

16. A.W. Mutammara, D.C. Hittle. "Energy Effects of Various control Strategies for Variable-Air -Volume Systems" ASHRAE Transactions. 1990, Vol. 96, Part 1, pp. 98-102.

17. D.K. Dickson. "Return Fans: Are They Necessary?", ASHRAE Journal. September 1997, pp. 30-33.

18. W.F. Chapman. "Back to Basic In HVAC Control", ASHRAE Transactions. 1984, Part 2B, pp. 213-215.

19. E. Ratnam, T. Campbell, R. Bradley. "Advanced Feedback Control of Indoor Air Quality Using Real-Time Computational Fluid Dynamics" ASHRAE Transactions. 1998, Vol. 104, Part 1.

20. Yu-Pei Ke, S.A.Mumma, D.Stanke. "Simulation Results and Analysis of Eight Ventilation Control Strategies in VAV systems", ASHRAE Transaction 1997, V103,Part 2.

21. K.Guttmann. "Proposed Method of Indoor Air Pollution Control By Dilution With Outdoor Air", Proceedings of the AHSRAE Conference, May 1987, Arlington, Virginia.

I. APPENDIX - SYSTEM MODEL

I.I Program Listing

```
C       FAN SIMULATION PROGRAM
C       FOR TWO and THREE DAMPER SYSTEMS
        REAL N1,N2
        INTEGER NCHAR
        CHARACTER*20 FNAME0,FNAME1,FNAME2,FNAME3

        WRITE(*,5)
      5 FORMAT(4X,'INPUT FILE NAME = '$)
        READ(*,10) FNAME0
     10 FORMAT(A20)
        OPEN(UNIT=3,FILE=FNAME0,STATUS='OLD')
        WRITE(*,15)
     15 FORMAT(4X,'INPUT RECORD   =   '$)
        READ(*,10) FNAME3
        OPEN(UNIT=4,FILE=FNAME3,STATUS='NEW')
        WRITE(*,20)
     20 FORMAT(4X,'OUTPUT FILE 1 =   '$)
        READ(*,10) FNAME1
        OPEN(UNIT=1,FILE=FNAME1,STATUS='NEW')
        WRITE(1,25)
     25 FORMAT(' SG     TCV       CV1      CV5       FSFPWR'
       #'       FRFPWR     CFMODA')
        WRITE(*,30)
     30 FORMAT(4X,'OUTPUT FILE 2 =   '$)
        READ(*,35) FNAME2
     35 FORMAT(A20)
        OPEN(UNIT=2,FILE=FNAME2,STATUS='NEW')
        WRITE(2,40)
     40 FORMAT('  SG     P2      P3     P4     P5    CFM'
       #'   N1     N2     W1     W2    TW ')

        READ(3,*) PRM
        READ(3,*) POUT
        READ(3,*) P3
        READ(3,*) CV2
        READ(3,*) CV3
        READ(3,*) NDAMPER
        READ(3,*) NCHAR

        WRITE(4,*) 'INPUT FILE     ', FNAME0
        WRITE(4,*) 'OUTPUT 1 FILE ', FNAME1
```

```
      WRITE(4,*) 'OUTPUT 2 FILE ', FNAME2
      WRITE(4,*) 'INPUT RECORD  ', FNAME3
      WRITE(4,*)
      WRITE(4,*) 'PRM     ', PRM
      WRITE(4,*) 'POUT    ', POUT
      WRITE(4,*) 'P3      ', P3
      WRITE(4,*) 'CV2     ', CV2
      WRITE(4,*) 'CV3     ', CV3
      WRITE(4,*) 'NDAMPER ', NDAMPER
      WRITE(4,*) 'NCHAR   ', NCHAR

C     P2 IS THE SUPPLY FAN INLET PRESSURE
C     P3 IS THE SUPPLY FAN OUTLET PRESSURE SET AT SETPOINT
C     P4 IS THE RETURN FAN INLET PRESSURE
C     P5 IS THE RETURN FAN OUTLET PRESSURE SET AT SETPOINT
C     PRM IS THE CONDITIONED ROOM PRESSURE
C     POUT IS THE (AMBIENT) OUTDOOR PRESSURE

C     NCHAR = 1 NO AREA CORRECTION
C     NCHAR = 2 WITH AREA CORRECTION
C     NCHAR = 3 LINEAR

C     INITILIZATION
      TW1=0.0
      TW2=0.0
      TTW=0.0
      TFSFP=0.0
      TFRFP=0.0

C     CALCULATE SUPPLY FAN FLOW RATE
      CFM=CV2*(P3-PRM)

C     SIGNAL SSELECTION FROM 20% TO 85% BY STEP 5%
      DO 65 I=0,13

C     SELECT DAMPER TYPE: EXPERIMENTAL,
C     WITH AREA CORRECTION, OR LINEAR

C     CALCULATE CV FOR THE DAMPER
      SELECT CASE (NCHAR)

      CASE (1)
C     EXPERIMENTAL
      SG=20+I*5
      CV1=CV(SG)
      CV5=CV(100-SG)
```

```
      TCV=CV1+CV5

      CASE (2)
C     WITH AREA COMPENSATION
      SGAC=20+I*5
      CALL SGAC_SG(SGAC,SG)
      CV1=CV(SG)
      CV5=CV(100-SG)
      TCV=CV1+CV5

      CASE (3)
C     LINEAR
      SG=20+I*5
      CV1=CVL(SG)
      CV5=CVL(100-SG)
      TCV=CV1+CV5

      END SELECT

C     CALCULATE SUPPLY FAN INLET PRESSURE
C     SELECT DAMPER CONFIGURATION, 3 OR 2
      SELECT CASE (NDAMPER)

      CASE (3)
      P2=-(CFM/(CV1+1.414*CV5))**2

      CASE (2)
      CV4=CV(90.0)
      P2=-(CFM/(CV1+(((CV1/CV4)**2+1)**0.5)*CV5))**2

      END SELECT
      WRITE(*,50) TCV,CV1,CV4,P2,CFM,NDAMPER
   50 FORMAT(5F10.4,I10)

C     CALCULTE THE DP ACROSS SUPPLY FAN
      DP1=P3-P2
C     SUPPLY FAN RPM CALCULATED FROM SPECIFIED CFM AND dP
      CALL FAN_N(CFM,N1,DP1)
C     SUPPLY FAN POWER CALCULATED FROM SPECIFIED CFM AND dP
      CALL POWER(N1,CFM,W1)

C     CALCULATE THE RETURN FAN PERFORMANCE
C     P4 IS THE RETURN FAN INLET PRESSURE
      P4=-(CFM/CV3)**2

C     CALCULTE THE P5
```

```
      SELECT CASE (NDAMPER)

      CASE (3)
      P5=-P2

      CASE (2)
      P5=-(CV1/CV4)**2*P2

      END SELECT

C     CALCULATE THE DP ACROSS THE RETURN FAN
      DP2=P5-P4

C     RETURN FAN RPM CALCULATED FROM SPECIFIED CFM AND dP
      CALL FAN_N(CFM,N2,DP2)
C     RETURN FAN POWER CALCULATED FROM SPECIFIED CFM AND dP
      CALL POWER(N2,CFM,W2)
      WRITE(*,111) W1,W2
  111 FORMAT(2X, 2F10.3)
C     PAUSE
C     CALCULATE OUTDOOR AIR FLOW RATE
      CFMODA=CV1*(-P2)**(0.5)

C     CALCULATE RATED POWER FOR SUPPLY AND RETURN FANS
      TW=W1+W2
      TW1=TW1+W1
      TW2=TW2+W2
C     TOTALIZE THE FAN RATED POWER
      TTW=TTW+TW

C     CALULATE THE FLUID POWER FOR SUPLLY AND RETURN FANS
      FSFPWR=0.0226*2600*(62.5/12*(P3-P2)
     #+0.5*0.074*2600*2600/32.17/(14.375*10.75/12/12)
     #/(14.375*10.75/12/12)/3600)
      TFSFP=TFSFP+FSFPWR
      FRFPWR=0.0226*2600*(62.5/12*(P5-P4)
     #+0.5*0.074*2600*2600/32.17/(14.375*10.75/12/12)
     #/(14.375*10.75/12/12)/3600)
      TFRFP=TFRFP+FRFPWR

C     SAVE AND PRINT OUT THE SIMULATION RESULTS
      WRITE(1,55) SG,TCV,CV1,CV5,FSFPWR,FRFPWR,CFMODA
   55 FORMAT(1X,F3.0,2X,F7.1,2X,F7.1,2X,F7.1,2X,F8.3,2X,
     #F8.3,2X,F8.3)

      WRITE(2,60) SG,P2,P3,P4,P5,CFM,N1,N2,W1,W2,TW
```

```
   60 FORMAT(1X,F3.0,2X,4F6.3,1X,F5.0,1X,F5.0,1X,F5.0,1X,
     #F6.1,1X,F6.1,1X,F6.1)
C     NEXT SIGNAL
   65 CONTINUE

C     TOTALIZE THE FAN FLUID POWER
      TTFPWR=TFSFP+TFRFP

C     SAVE AND PRINT OUT THE FAN POWER RESULT
      WRITE(4,70) TFSFP,TFRFP,TTFPWR
   70 FORMAT(/,4X,'     TFSFP     TFRFP    TTFPWR',/
     #,4X,3F10.3)
      WRITE(4,75) TW1,TW2,TTW
   75 FORMAT(/,4X,'       TW1       TW2       TTW',/
     #,4X,3F10.1)
      END


      FUNCTION CV(SG)
C     EXPERIMENTAL CV
C     CV FROM JULY 23i 1997 DATA
      CV=-165.077+44.52*SG1.000459*(SG**2)
     #+0.0123513*(SG**3)

      IF(CV.LT.0) CV=0
      IF(CV.GT.14000.0) CV=14000.0
      END


      FUNCTION CVC(SG)
C     CV FUNCTION WITH AREA CORRECTION
      CVC=813.527+15.1399*SG+1.13783*(SG**2)
     #-0.008666*(SG**3)
      IF(CVC.LT.0) CVC=0
      IF(CVC.GT.14000.0) CVC=14000.0
      END


      FUNCTION CVL(SG)
C     LINEAR CV
      CVL=400.0+4000.0/65*(SG-20)
      IF(CVL.LT.0) CVL=0
      IF(CVL.GT.14000.0) CVL=14000.0
      END
```

```
      FUNCTION PRS(V)
C     PRESSURE = f(FLOW RATE) @ 1800 RPM
      PRS=2.20469-0.000212152*V+6.6248E-7*(V**2)
     #-5.78207E-10*(V**3)+8.81625E-14*(V**4)
      END


      FUNCTION WPV(V18)
C     FAN POWER = f(FLOW RATE) @ 1800 RPM
      WPV=.38+.0006322*(v18)-2.61335E-7*(v18**2)
     #+5.31183E-11*(v18**3)-9.48011E-15*(v18**4)
      END


      SUBROUTINE SGAC_SG(SGAC1,SG1)
C     AREA CORRECTION:
C     SG - SIGNAL REQUIRED BY FEEDBACK CONTROL
C            CONSIDERATIONS
C     SGAC - SIGNAL TO ACTUATOR CORRECTED FOR AREA
C            LINEARIZATION
      REAL ALPHA
      SGAC2=0.0
      SG1=0.01
    5 CONTINUE
      IF(ABS(SGAC2-SGAC1).GT.0.01) THEN
         X=1-SG1/100
         Y=SQRT(1-X**2)
         Z=Y/X
         ALPHA=ATAN(Z)
         SGAC2=100*ALPHA/1.570796327
         SG1=SG1+0.001
         GOTO 5
      ENDIF
      RETURN
      END


      SUBROUTINE FAN_P(V,X,P)
C     FAN SUBROUTINE: KNOWN CFM & SPEED, CALCULATE PRESSURE
      V18=V/(X/1800.0)
      P=PRS(V18)*(X/1800.0)**2
      WRITE(*,5) V,X,P
    5 FORMAT(2X,F10.5,2X,F10.5,2X,F10.5)
      RETURN
      END
```

```
C       FAN SUBROUTINE (KNOWN CFM & PRESSURE TO SPEED)
        SUBROUTINE FAN_N(V,X,P)
C       CALCULATE FAN SPEED FROM SPECIFIED FLOW RATE (CFM)
C       AND PRESSURE dP
        REAL PP
        FLAG=3
        I =1.0
        X=1000.0
      5 V18=V/(X/1800)
        PP=PRS(V18)*(X/1800.0)**2
        IF (ABS(P-PP).LT.0.01) RETURN
        IF ((P-PP).GT.0) THEN
           IF (FLAG.EQ.2) THEN
               I=I*2.0
               X=X+1.0/I
               FLAG=1
           ELSE
               X=X+1.0/I
               FLAG=1
           ENDIF
        ELSE
           IF (FLAG.EQ.1) THEN
               I=I*2.0
               X=X-1.0/I
               FLAG=2
           ELSE
               X=X-1.0/I
               FLAG=2
           ENDIF
        ENDIF
        GOTO 5
        RETURN
        END


        SUBROUTINE POWER(SP,V,PW)
C       FAN POWER SUBROUTINE
C       CALCULATE THE FAN POWER,KNOWN TEH SPEED AND CFM
        vp=v/(sp/1800.0)
        pw18=WPV(vp)
        pw=745.7*pw18*(sp/1800)**3
        return
        end
```

I.II Sample Calculations

I.II.I Input File for Three Damper without Area Compensation: o.dat

```
INPUT FILE    dtinput.dat
OUTPUT 1 FILE o1.dat
OUTPUT 2 FILE o2.dat
INPUT RECORD  o.dat

PRM        0.000000E+00
POUT       0.000000E+00
P2         5.000000E-01
CVSP        5080.000000
CVRN        6865.000000
NDAMPER               3
NCHAR                 1

      TFSFP      TFRFP     TTFPWR
   6298.540   4742.784  11041.320

        TW1        TW2        TTW
    13460.7    11394.3    24855.0
```

I.II.II Output File for Three Damper without Area Compensation: o1.dat

SG	TCV	CV1	CV2	FSFPWR	FRFPWR	CFMODA
20.	3741.4	423.9	3317.5	338.690	227.564	210.532
25.	3272.7	515.6	2757.0	364.554	253.429	296.706
30.	2889.2	603.6	2285.6	397.441	286.316	399.733
35.	2590.9	697.1	1893.8	436.569	325.443	524.665
40.	2377.8	805.5	1572.4	478.453	367.328	675.487
45.	2250.0	937.9	1312.1	516.291	405.166	852.889
50.	2207.4	1103.7	1103.7	541.369	430.243	1052.196
55.	2250.0	1312.1	937.9	546.883	435.757	1263.205
60.	2377.8	1572.4	805.5	531.812	420.686	1473.016
65.	2590.9	1893.8	697.1	501.349	390.224	1670.486
70.	2889.2	2285.6	603.6	463.596	352.471	1849.394
75.	3272.7	2757.0	515.6	425.682	314.557	2008.782
80.	3741.4	3317.5	423.9	391.912	280.787	2151.265
85.	4295.4	3976.0	319.3	363.939	252.814	2280.984

I.II.III Output File for Three Damper without Area Compensation: o2.dat

SG	P1	P2	P3	P4	CFM	N1	N2	W1	W2	TW
20.	.247	.500	-.137	.247	2540.	1972.	1871.	812.9	671.2	1484.1
25.	-.331	.500	-.137	.331	2540.	1995.	1895.	846.9	703.8	1550.6
30.	-.439	.500	-.137	.439	2540.	2024.	1925.	890.6	745.5	1636.0
35.	-.566	.500	-.137	.566	2540.	2058.	1960.	943.1	795.5	1738.5
40.	-.703	.500	-.137	.703	2540.	2093.	1997.	998.6	849.8	1848.4
45.	-.827	.500	-.137	.827	2540.	2125.	2030.	1050.7	899.7	1950.4
50.	-.909	.500	-.137	.909	2540.	2146.	2052.	1085.6	933.7	2019.3
55.	-.927	.500	-.137	.927	2540.	2151.	2057.	1094.0	941.5	2035.5
60.	-.878	.500	-.137	.878	2540.	2138.	2044.	1072.2	921.3	1993.5
65.	-.778	.500	-.137	.778	2540.	2113.	2017.	1031.0	879.9	1910.9
70.	-.655	.500	-.137	.655	2540.	2081.	1984.	979.4	830.6	1809.9
75.	-.531	.500	-.137	.531	2540.	2048.	1950.	927.5	781.0	1708.5
80.	-.421	.500	-.137	.421	2540.	2019.	1920.	883.0	738.5	1621.4
85.	-.329	.500	-.137	.329	2540.	1994.	1894.	845.4	702.4	1547.8

II. Appendix -Demand Control Model

II.I Program Listing

```
C       CO2 CONTROL SIMULATION

        PROGRAM CO2
        COMMON/PID/SGMIN,SGMAX
        CHARACTER*20 FNAME0,FNAME1,FNAME2
        REAL NSG,VODA,TS
        REAL KI,KP,KIPPM,KPPPM
        INTEGER NDPR,NCR,INI,I
        DATA SGMIN,SGMAX/20,85/
        WRITE(*,5)
      5 FORMAT(4X,'INPUT  FILE 0 =    '$)
        READ(*,15) FNAME0
     15 FORMAT(A20)
        OPEN(UNIT=1,FILE=FNAME0,STATUS='OLD')
        WRITE(*,10)
     10 FORMAT(4X,'OUTPUT FILE 1 =    '$)
        READ(*,20) FNAME1
     20 FORMAT(A20)
        OPEN(UNIT=2,FILE=FNAME1,STATUS='NEW')
        WRITE(*,30)
     30 FORMAT(4X,'OUTPUT FILE 2 =    '$)
        READ(*,20) FNAME2
        OPEN(UNIT=3,FILE=FNAME2,STATUS='NEW')
        WRITE(2,35) FNAME0,FNAME2
     35 FORMAT(2X'INPUT FILE=',A10,'OUTPUT FILE=',A10)
        WRITE(3,40)
     40 FORMAT(2X'   TIME     CFH      PPM              SG'
       #,'       VODA')
C       DATA INPUT
        READ(1,*) TDT
        READ(1,*) TSIM
        READ(1,*) G
        READ(1,*) CINI
        READ(1,*) CODA
        READ(1,*) CSP
        READ(1,*) CMAX
        READ(1,*) VRM
        READ(1,*) VSUP
        READ(1,*) NDPR
        READ(1,*) NCR

C       UNIT CONVERSION
```

```
      CINI=CINI/1000000.
      CODA=CODA/1000000.
      CSP=CSP/1000000.
      CMAX=CMAX/1000000.

C     INITIALIZATION
      GCFH=30.
      INI=0
      T=0.0
      C=CINI
      NSG=SGMIN
      I=0
      TS=30.0
      SENS=(CODA-CMAX)*VSUP/VRM/(SGMAX-SGMIN)
      SENSPPM=SENS*1000000.
      KP=1.0/SENS/TDT
      KI=1.0/4.0/SENS/TDT
      KPPPM=KP/1000000
      KIPPM=KI/1000000
      G=GCFH/60
      WRITE(2,50) KPPPM,KIPPM,SENSPP
   50 FORMAT(/2X,'      KPPPM     KIPPM   SENSPPM',/
     #,2X,3F10.5)

   60 IF(T.LT.TS) THEN
         CALL SYS(NSG,VODA,NDPR,NCR,VSUP)
         VREC=VSUP-VODA
         CSUP=VODA/VSUP*CODA+VREC/VSUP*C
         C=(G-VSUP*(C-CSUP))*TDT/VRM+C
         CALL PI(CSP,KI,KP,C,NSG,INI)
         T=T+TDT
         PPM=C*1000000.
         WRITE(3,70) T,GCFH,PPM,NSG,VODA
   70    FORMAT(4X,F6.0,2X,F6.2,2X,F10.4,2X,2F10.2)
      ELSE
         TS=TS+30.0
         I=I+1
         IF(I.LT.13) THEN
             IF (I.LE.6) THEN
                 G=0.5+5.0/60.0*I
                GCFH=G*60.0
             ELSE
                G=90.0/60.0-5.0/60.0*I
                GCFH=G*60.0
            END IF
```

```
          ELSE
             STOP
          END IF
        END IF
        WRITE(*,61) I
 61     FORMAT(2X,I5)
        GOTO 60

        END


        SUBROUTINE PI(SP,PIKI,PIKP,PV,PISG,INI)
C       PI CONTROL SUBROTINE
        COMMON/PID/SGMIN,SGMAX
C       DATA SGMIN,SGMAX/20,85/
        INTEGER INI
        IF(INI.EQ.0) THEN
          SEHI=SGMAX/PIKI
          SELO=SGMIN/PIKI
          SE=PISG/PIKI
          INI=1
        END IF
        E=SP-PV
        SE=SE+E
        PISG=PIKP*E+PIKI*SE
        IF(PISG.GT.SGMAX) PISG=SGMAX
        IF(PISG.LT.SGMIN) PISG=SGMIN
        IF(PIKI.GT.0) THEN
            IF (SE.LT.SEHI) THEN SE=SEHI
            IF (SE.GT.SELO) THEN SE=SELO
        ELSE
            IF(SE.GT.SEHI) THEN SE=SEHI
            IF (SE.LT.SELO) THEN SE=SELO
        END IF
        RETURN
        END

        SUBROUTINE SYS(SG,CFMODA,NDAMPER,NCHAR,CFM)
        REAL N1,N2,PRM,POUT,P2,CVRN,CVSP
        INTEGER NDAMPER,NCHAR
        N1=0
        N2=0
        PRM=0.0
        POUT=0.0
        P2=0.5
        CVRN=6865.0
```

```
      CVSP=5080.0

      CFM=CVSP*(P2-PRM)

C     SELECT DAMPER TYPE: EXPERIMENTAL, WITH AREA
C           CORRECTION, OR LINEAR
C     CALCULATE CV FOR THE DAMPER
      SELECT CASE (NCHAR)

      CASE (1)
C     EXPERIMENTAL
      CV1=CV(SG)
      CV2=CV(100-SG)
      TCV=CV1+CV2

      CASE (2)
C     WITH AREA CORRECTION

      SGAC=SG
      CALL SGAC_SG(SGAC,SG)
      CV1=CV(SG)
      CV2=CV(100-SG)
      TCV=CV1+CV2

      CASE (3)
C     LINEAR
      CV1=CVL(SG)
      CV2=CVL(100-SG)
      TCV=CV1+CV2

      END SELECT

C     CALCULATE SUPPLY FAN INLET PRESSURE
C     SELECT DAMPER CONFIGURATION, 3 OR 2
      SELECT CASE (NDAMPER)

      CASE (3)
      P1=-(CFM/(CV1+1.414*CV2))**2

      CASE (2)
      CV3=CV(90.0)
      P1=-(CFM/(CV1+(((CV1/CV3)**2+1)**0.5)*CV2))**2

      END SELECT

C     CALCULTE THE DP ACROSS SUPPLY FAN
```

```
      DP1=P2-P1

C     CALCULATE THE RETURN FAN PERFORMANCE
C     P3 IS THE RETURN FAN INLET PRESSURE
      P3=-(CFM/CVRN)**2

C     CALCULTE THE P4
      SELECT CASE (NDAMPER)

      CASE (3)
      P4=-P1

      CASE (2)
      P4=-(CV1/CV3)**2*P1

      END SELECT

      CFMODA=CV1*(-P1)**(0.5)
   10 CONTINUE
      RETURN
      END

      FUNCTION CV(SG)
C     EXPERIMENTAL CV
C     CV FROM JULY 23i 1997 DATA
      CV=-165.077+44.52*SG
     #-1.000459*(SG**2)+0.0123513*(SG**3)

      IF(CV.LT.0) CV=0.0
      IF(CV.GT.14000.0) CV=14000.0
      END


      FUNCTION CVC(SG)
C     CV FUNCTION WITH AREA CORRECTION
      CVC=813.527+15.1399*SG+1.13783*(SG**2)
     #-0.008666*(SG**3)
      IF(CVC.LT.0) CVC=0
      IF(CVC.GT.14000.0) CVC=14000.0
      END


      FUNCTION CVL(SG)
C     LINEAR CV
      CVL=400.0+4000.0/65*(SG-20)
      IF(CVL.LT.0) CVL=0
```

```
      IF(CVL.GT.14000.0) CVL=14000.0
      END


      FUNCTION PRS(V)
C     PRESSURE = f(FLOW RATE) @ 1800 RPM
      PRS=2.20469-0.000212152*V+6.6248E-7*(V**2)
     #-5.78207E-10*(V**3)+8.81625E-14*(V**4)
      END


      SUBROUTINE SGAC_SG(SGAC1,SG1)
C     AREA CORRECTION:
C     SG - SIGNAL REQUIRED BY FEEDBACK CONTROL
C            CONSIDERATIONS
C   SGAC - SIGNAL TO ACTUATOR CORRECTED FOR AREA
C            LINEARIZATION
      REAL ALPHA
      SGAC2=0.0
      SG1=0.01
   10 CONTINUE
      IF(ABS(SGAC2-SGAC1).GT.0.01) THEN
        X=1-SG1/100
        Y=SQRT(1-X**2)
        Z=Y/X
        ALPHA=ATAN(Z)
        SGAC2=100*ALPHA/1.570796327
        SG1=SG1+0.001
        GOTO 10
      ENDIF
      RETURN
      END


      SUBROUTINE FAN_P(V,X,P)
C     FAN SUBROUTINE: KNOWN CFM & SPEED, CALCULATE PRESSURE
      V18=V/(X/1800.0)
      P=PRS(V18)*(X/1800.0)**2
      WRITE(*,10) V,X,P
   10 FORMAT('V  X  P',2X,F10.5,2X,F10.5,2X,F10.5)
      RETURN
      END

C     FAN SUBROUTINE (KNOWN CFM& PRESSURE)
      SUBROUTINE FAN_N(V,X,P)
C     CALCULATE FAN SPEED FROM SPECIFIED FLOW RATE (CFM)
```

```
C              AND PRESSURE dP
       REAL PP
C      INTEGER FLAG
       FLAG=3
       I=1.0
       X=1000.0
    10 V18=V/(X/1800)
       PP=PRS(V18)*(X/1800.0)**2
       IF (ABS(P-PP).LT.0.01) RETURN
       IF ((P-PP).GT.0) THEN
          IF (FLAG.EQ.2) THEN
              I=I*2.0
              X=X+1.0/I
              FLAG=1
          ELSE
              X=X+1.0/I
              FLAG=1
          ENDIF
       ELSE
          IF (FLAG.EQ.1) THEN
              I=I*2.0
              X=X-1.0/I
              FLAG=2
          ELSE
              X=X-1.0/I
              FLAG=2
          ENDIF
       ENDIF
       GOTO 10
       RETURN
       END
```

II.II Sample Calculation

II.II.I Sample Input Date in Simulation of Two Damper with Area Compensation System for Low ppm in Room: o1.dat

```
TDT=          1.0
TSIM=         360.0
G=            10
CINI=         400
CODA=         400
CSP=          1000
CMAX=         1400
VRM=          6000.0
```

VSUP= 2400.0
NDPR= 2
NCR= 2

II.II.II Sample Data Output in Simulation of Two Damper with Area Compensation for Low ppm in Room: o1.dat

TIME	CFH	PPM	SG	VODA
1.	30.00	483.3333	20.00	13.50
2.	30.00	566.4791	20.00	13.50
3.	30.00	649.4377	20.00	13.50
4.	30.00	732.2097	20.00	13.50
5.	30.00	814.7953	20.00	13.50
6.	30.00	897.1951	20.00	13.50
7.	30.00	979.4093	20.00	13.50
8.	30.00	1061.4390	20.00	13.50
9.	30.00	1143.2830	20.00	13.50
10.	30.00	1224.9440	20.00	13.50
11.	30.00	1306.4200	20.00	13.50
12.	30.00	1387.7140	37.29	13.50
13.	30.00	1434.8380	62.61	219.96
14.	30.00	1340.5270	61.12	1029.98
15.	30.00	1274.4500	61.53	953.15
16.	30.00	1215.8320	60.78	973.99
17.	30.00	1171.9140	60.62	935.87
18.	30.00	1135.8220	60.28	928.28
19.	30.00	1107.4030	60.02	911.24
20.	30.00	1084.7640	59.79	898.83
21.	30.00	1066.8110	59.58	887.49
22.	30.00	1052.5910	59.41	877.79
23.	30.00	1041.3490	59.26	869.53
24.	30.00	1032.4800	59.14	862.58
25.	30.00	1025.4860	59.04	856.89
26.	30.00	1019.9820	58.96	852.17
27.	30.00	1015.6580	58.89	848.32
28.	30.00	1012.2630	58.84	845.22
29.	30.00	1009.5990	58.79	842.75
30.	30.00	1007.5110	58.76	840.76
31.	35.00	1019.7640	61.55	839.18
32.	35.00	1016.2730	61.65	975.02
33.	35.00	1012.8580	61.61	979.80
34.	35.00	1010.1700	61.59	978.14
35.	35.00	1008.0450	61.57	976.91
36.	35.00	1006.3580	61.56	976.01

37.	35.00	1005.0300	61.54	975.18
38.	35.00	1003.9760	61.53	974.59
39.	35.00	1003.1420	61.53	974.11
40.	35.00	1002.4860	61.52	973.68
41.	35.00	1001.9650	61.52	973.40
42.	35.00	1001.5520	61.51	973.17
43.	35.00	1001.2260	61.51	972.97
44.	35.00	1000.9720	61.51	972.77
45.	35.00	1000.7670	61.51	972.69
46.	35.00	1000.6080	61.50	972.57
47.	35.00	1000.4780	61.50	972.54
48.	35.00	1000.3770	61.50	972.46
49.	35.00	1000.3010	61.50	972.38
50.	35.00	1000.2360	61.50	972.38
51.	35.00	1000.1860	61.50	972.34
52.	35.00	1000.1490	61.50	972.30
53.	35.00	1000.1170	61.50	972.30
54.	35.00	1000.0900	61.50	972.30
55.	35.00	1000.0720	61.50	972.26
56.	35.00	1000.0570	61.50	972.26
57.	35.00	1000.0480	61.50	972.22
58.	35.00	1000.0360	61.50	972.26
59.	35.00	1000.0310	61.50	972.22
60.	35.00	1000.0220	61.50	972.26
61.	40.00	1013.9070	64.32	972.22
62.	40.00	1010.2450	64.14	1121.74
63.	40.00	1008.2620	64.15	1111.95
64.	40.00	1006.5720	64.15	1112.69
65.	40.00	1005.2370	64.14	1112.28
66.	40.00	1004.1740	64.14	1112.03
67.	40.00	1003.3250	64.14	1111.87
68.	40.00	1002.6500	64.13	1111.70
69.	40.00	1002.1080	64.13	1111.62
70.	40.00	1001.6830	64.13	1111.46
71.	40.00	1001.3410	64.13	1111.42
72.	40.00	1001.0660	64.13	1111.37
73.	40.00	1000.8510	64.13	1111.29
74.	40.00	1000.6750	64.13	1111.29
75.	40.00	1000.5400	64.13	1111.21
76.	40.00	1000.4300	64.13	1111.21
77.	40.00	1000.3410	64.13	1111.21
78.	40.00	1000.2720	64.13	1111.17
79.	40.00	1000.2200	64.13	1111.13
80.	40.00	1000.1730	64.13	1111.17
81.	40.00	1000.1400	64.13	1111.13
82.	40.00	1000.1080	64.13	1111.17

83.	40.00	1000.0860	64.13	1111.13
84.	40.00	1000.0680	64.13	1111.13
85.	40.00	1000.0580	64.13	1111.09
86.	40.00	1000.0460	64.13	1111.13
87.	40.00	1000.0360	64.13	1111.13
88.	40.00	1000.0270	64.13	1111.13
89.	40.00	1000.0210	64.12	1111.13
90.	40.00	1000.0190	64.13	1111.09
91.	45.00	1013.9030	66.95	1111.13
92.	45.00	1009.1040	66.54	1268.59
93.	45.00	1007.6790	66.62	1245.36
94.	45.00	1006.0860	66.61	1249.93
95.	45.00	1004.8920	66.61	1249.27
96.	45.00	1003.9220	66.61	1249.52
97.	45.00	1003.1410	66.61	1249.64
98.	45.00	1002.5220	66.61	1249.64
99.	45.00	1002.0200	66.61	1249.77
100.	45.00	1001.6190	66.62	1249.81
101.	45.00	1001.2970	66.62	1249.85
102.	45.00	1001.0420	66.62	1249.85

III Appendix -Selected Funtions Used for Experiemental Controls

III.I Function Listing (Basic)

```
FUNCTION HC (SGi, CH%, SGi0(), HYST(), PARA())
'HYSTERESIS CORRECTION
SGMNR = PARA(9, CH%)
SGMXR = PARA(10, CH%)
IF SGi >= SGi0(CH%) THEN
      SG = SGi + HYST(CH%) * (1 - SGi / 100)
ELSE
       SG = SGi - HYST(CH%) * SGi / 100
END IF
IF SG > SGMXR THEN SG = SGMXR
IF SG < SGMNR THEN SG = SGMNR
HC = SG
END FUNCTION
```

```
  FUNCTION SGACNC (SG)
' AREA COMPENSATION FOR NORMALLY CLOSED DAMPER
  X = 1 - SG / 100
  XX = SQR(1 - X ^ 2)
  IF X > 0 THEN
    ALPHA = ATN(XX / X)
  ELSE
    ALPHA = 1.570796327#
  END IF
  SGACNC = 100 * ALPHA / 1.570796327#
  END FUNCTION
```

```
  FUNCTION SGACNO (SG)
' AREA COMPENSATION FOR NORMALLY OPEN DAMPER
  X = SG / 100
  XX = SQR(1 - X ^ 2)
  IF XX > 0 THEN
    ALPHA = ATN(X / XX)
  ELSE
    ALPHA = 1.570796327#
  END IF
  SGACNO = 100 * ALPHA / 1.570796327#
  END FUNCTION
```

```
FUNCTION PI (PV!, CH%, PARA())
      SP=PARA(1,CH%)
      SCAN=PARA(2,CH%)
      SENS=PARA(3,CH%)
      EPS=PARA(4,CH%)
      DAMPR=PARA(5,CH%)
      DEC%=PARA(6,CH%)
      ORDER%=PARA(7,CH%)
      INI%=PARA(8,CH%)
      SGMNR=PARA(9,CH%)
      SGMXR=PARA(10,CH%)
      KP=PARA(11,CH%)
      KI=PARA(12,CH%)
      SE=PARA(13,CH%)
      'SGi=PARA(14,CH%)
      SEHI=PARA(15,CH%)
      SELO=PARA(16,CH%)
      'E=PARA(17,CH%)

      E = SP - PV!
      IF DEC% > -1 THEN E = RNDOFF(E,DEC%)
      PARA(17,CH%)=E

      IF INI% = 0 THEN
          SE = 0
          IF ORDER% = 1 THEN
              KP = 0
              KI = EPS / SENS
          ELSE
              KP = 2 * EPS / SCAN / SENS
              KI = EPS ^ 2 / SCAN / DAMPR ^ 2 / SENS
          END IF
          PARA(11,CH%)=KP
          PARA(12,CH%)=KI
          SEHI = SGMXR / KI
          SELO = SGMNR / KI
          PARA(15,CH%)=SEHI
          PARA(16,CH%)=SELO
          PRINT #3,
          PRINT #3, "  CH =",CH%
          PRINT #3, "  KP =",KP
          PRINT #3, "  KI =",KI
      ELSE
          SE = SE + E
      END IF
      PARA(13,CH%)=SE
```

```
        SGi = KP * E + KI * SE

'       WINDUP PREVENTION
        IF KI < 0 THEN
            IF SGi > SGMXR THEN
                SGi = SGMXR
                IF SE < SEHI THEN SE = SEHI
            END IF
            IF SGi < SGMNR THEN
                SGi = SGMNR
                IF SE > SELO THEN SE = SELO
            END IF
        ELSE
            IF SGi > SGMXR THEN
                SGi = SGMXR
                IF SE > SEHI THEN SE = SEHI
            END IF
            IF SGi < SGMNR THEN
                SGi = SGMNR
                IF SE < SGMNR THEN SE = SELO
            END IF
        END IF

        PI = SGi
        PARA(14,CH%)=SGi

        INI% = 1
        PARA(8,CH%) =INI%

'      FOR I% = 1 TO 17
'         PRINT #3, I%, PARA(I%,CH%)
'      NEXT I%

        END FUNCTION
```