Reduction of Excess Fan Power due to Economizer Damper Pressure Losses

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ABSTRACT

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An air conditioning system economizer, investigated in this thesis, consisted of three dampers, an outdoor air, a recirculation, and a discharge damper. The purpose of an economizer is to control the ratio of the outdoor air flow rate to the recirculation air flow rate. The dampers were opposed blade type and had equal sizes. They were coupled through a control system.

The flow rate through a damper is related to the head loss across the damper by a flow coefficient. The study showed that the flow coefficient is directly proportional to the damper throat area. It was shown that excess power consumption due to the economizer dampers is dependent on the pressure differential across the recirculation damper. It was also shown that the pressure differential may be reduced by using linear dampers and by the control strategy.

The kinematic and dynamic studies indicated that it is possible to linearize damper characteristic by geometry selection when designing the damper and by software modification of the control signal. The throat area as a function of the control signal, i.e., blade angles, may be varied by the selection of the length of the coupling link and the driven axis. Software linearization of the throat area

versus the control signal may be accomplished by 'compensating' the ideal signal before it is sent to the motor. The ideal signal is that required by a feedback control algorithm. The compensated signal is that sent to the motor. The compensated signal is related to the ideal signal by an algorithm based on the throat area as a function of the ideal control signal.

Linearization of throat area versus control signal will reduce excess power consumption and decrease variation in fan control signals required to maintain constant pressures. Reducing fan speed variations extends the operating range of a fan.

Two control strategies were investigated. In the first strategy, the three economizer dampers were coupled so that the outdoor air and discharge damper open as the recirculation damper closes. In a second strategy, the discharge damper was always open during operation and the outdoor air and recirculation dampers were coupled so that the outdoor air damper opens as the recirculation damper closes. The two coupled dampers control strategy will require slightly less power than three coupled damper strategy.

The study showed that the linearization of throat area versus control signal is the most important factor for reducing power, whether the three coupled or two coupled dampers control strategy is used. linearization provides good control of the air flow rates, from 0% to 95%. Whereas linearization may be accomplished by means of software, power savings may be realized without capital expenditure.

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NOMENCLATURE

A_d	face area of damper
A_t	throat (free flow) area of damper
a	offset right length as shown in Figures 3.12 and 3.13
b	offset left length as shown in Figures 3.12 and 3.13
c	coupler length
C_v	flow coefficient
C_d	damper discharge coefficient
f	correlation factor
n	number of damper blades
P	Pressure
Q	air flow rate
r	crank arm length
s	axis to axis distance
SG	control signal
TA	damper throat area
W	fan power
α	angle between the positive x-axis and α crank arm
β	angle between the positive x-axis and β crank arm
γ	angle between the positive x-axis and the coupler
Γ	torque
ρ	air density

Subscripts

atm	atmospheric pressure
D	drive
DIS	discharge
i	ideal
m	motor
MAX	maximum
MIN	minimum
ODA	outdoor air damper
R	resistance
RTN	return
REC	recirculation
SUP	supply
T	total
α	α crank arm
β	β crank arm
0,1,2,.....	key points for calculations

CHAPTER 1

INTRODUCTION

A heating ventilating and air conditioning (HVAC) system economizer may be used with both constant volume variable temperature (CVVT) and variable air volume (VAV) HVAC systems. If the HVAC system is a CVVT system, the supply air flow rate to a space is constant while its temperature varies to meet the desired space temperature. If the HVAC system is a VAV system, the supply air temperature is constant while its flow rate varies to meet the desired space temperature. The temperature of the supply air depends on heating and cooling seasons. In heating season warm air is supplied and in cooling season chilled air is supplied.

Two fans are used in most large HVAC systems, a supply fan and a return fan. The supply fan draws mixed air, outdoor air and recirculation air, and supplies it to the space. Return fan draws return air from the space. Some of the return air is exhausted and the rest is recirculated. The power consumed by these fans is a part of the total energy consumption of the HVAC system. An economizer can reduce the system power consumption by varying the ratio of the outdoor air to recirculated air. The ratio of outdoor air to recirculated air may be varied according to energy or indoor air quality consideration. The economizer ensures a minimum amount of outdoor air to maintain the indoor air quality at an acceptable level. A conditioned space should be provided with a minimum amount of fresh air to maintain the carbon dioxide (CO_2) concentration below 1000 ppm at all times.

The primary objective of the current investigation was to develop and validate means of reducing excess fan power consumption due to economizer damper pressure losses. This involved a study of damper design and control strategies.

An economizer has three dampers - an outdoor air damper, a recirculation damper, and a discharge damper. The three dampers are coupled by means of a control system. The air flows through each damper can be varied by changing the position of the damper from a minimum position up to a maximum position. The dampers should be sized and selected so that an increase in outdoor air flow is matched by an equal decrease in the recirculation air flow.

The dampers may be controlled under energy - air enthalpy or temperature - considerations or under indoor air equality considerations. If the temperature of the outdoor air is less than the desired temperature of the supply air, the quantity of outdoor air is varied so that the combination of the recirculation air and the outdoor air yield a supply air at the desired delivery air temperature. A 100% of outdoor air is used if the outdoor air enthalpy is less than the return air enthalpy. A minimum outdoor air is required if the outdoor air enthalpy is greater than the return air enthalpy. The minimum quantity of outdoor air must ensure that the concentration of CO_2 in the space at an acceptable level.

Two different types of dampers may be used in economizers to modulate the air flow. These are opposed blade dampers and parallel blade dampers. Adjacent blades of opposed blade damper rotate in opposite direction. All blades of parallel blade damper rotate in the same direction. The damper characteristic depends on the damper type. The system characteristics depend on the damper characteristics and the resistances of all the components (coils, filters, ducts, etc.) in the system. Generally, neither opposed blade nor parallel blade dampers have linear characteristics. The non-linear characteristic of parallel blade damper leads to poor control and causes instability. The opposed blade dampers are mostly used to provide better control. Properly sizing and selecting dampers is an important factor in designing a good economizer. The dampers characteristic should compliment each other so that the total mixing flow rate will be constant. It is therefore necessary to study a damper geometry to design a linear damper.

The basic concept of this work was investigated by Zhao [12]. It was shown that the control strategy effects system power consumption. Using an approximate linearization algorithm, Zhao also showed that linearized damper characteristics reduce system power consumption. The approximate linearization algorithm considered non-overlapping, flat blades with zero thickness. The economizer tested was limited to control signals between 20% and 85%.

A secondary objective of this investigation was to develop and validate accurate linearization methods - both hardware and software. In the current investigation, the damper geometry has been studied in order to design a damper with a linear characteristic. Methods to linearize damper characteristics by software were also developed. Two control strategy have been tested - a three coupled damper strategy, and a two coupled damper strategy. In three coupled dampers control strategy, all the dampers are coupled. In the two coupled damper control strategy, only outdoor air and recirculation dampers are coupled while the discharge damper remains open during operating time.

CHAPTER 2

LITERATURE SURVEY

Two types of dampers are described in ASHRAE Fundamentals [1] and ASHRAE Systems Handbooks [2], parallel blade dampers and opposed blade dampers. In a parallel blade damper, all blades rotate in the same directions. In an opposed blade damper, adjacent blades rotate in opposite directions. These two types of dampers have different inherent flow characteristics. Opposed blade dampers do not change the flow direction but parallel blade dampers do. The opposed blade damper give a very slow increase in the flow when the damper begins to open. The opposed blade damper are preferred to control the air flow rather than the parallel blade dampers. Both dampers have non-linear characteristic. This non-linear performance may lead to poor control and may cause instability when controlled by a feedback control system.

Alley [3] indicated in his paper that the mid-point of the control range of the mixing dampers with same size may offer some unexpected surprises. For parallel blade dampers, 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. Alley did a series of tests for coupled dampers with a combination of opposed and parallel blades. The mid-point volume, mid-point pressure drop, can by varied by selecting a damper with combination of opposed and parallel blades. A damper with proper characteristics may be selected as a damper with a mix of parallel and opposed blades in order to match the system requirement.

Avery [4] did a series of tests for coupled dampers with a combination of opposed and parallel blades. 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.

Avery [5], based on the data available for selecting and sizing dampers, recommended in his paper that the damper flow characteristics must be specified as well as maximum pressure drop and face velocity. An equivalent data on single frame combination blade dampers and on double frame, one opposed blade section and one parallel blade section, combination dampers linked to operate in unison must be provided by the manufacturer. Damper actuators must be positive positioning and large enough to properly position the dampers.

Belimo Damper Application Guide 1 [6] describes opposed blade dampers and parallel blade dampers. They have different inherent flow characteristics. Neither parallel blade nor opposed blade dampers have linear characteristics. The damper inherent flow characteristic changes when the damper installed in the system. This is because of the effect of the other system components. The percentage of the flow resistance of the open damper relative to the total system flow resistance is called "damper authority". The damper authority should be selected so that the system has a close approximation to a linear relationship between the flow rate ratio and the blade angle. The dampers are usually operated in unison, so as the outdoor air and discharge dampers open, the recirculation damper closes. This control strategy is called the three coupled damper strategy. The resistant to the flow and the total supply air flow should be constant, regardless of the mixing ratio between outdoor air and recirculation air flows, so the pressure in the mixing plenum will stay constant. This is very hard to accomplish unless the dampers have linear characteristics.

Dickson [7] indicated that an improper selection of dampers will lead to a poor ventilation control and pressure fluctuations in the mixed air section. An appropriate selection can ensure comfort at the lowest possible energy cost. He also indicated that if proper size is not possible in a design or if an existing system contains oversized dampers, the objective can be achieved by the linkage adjustments and by good selection of actuators.

Krakow [8] indicated that hysteresis in motorized dampers and valves might result in poor repeatability of experimental data. It also might result in the deviation of a response of a proportional integral controlled system from its target response and in hunting. The amount of the hysteresis was dependent on the motor, and the linkage mechanism. He presented a software method to reduce the effect of hysteresis.

Robinson [9] did many tests on parallel blade dampers to determine the control response and mixing effectiveness of a combination mixing/filter box in a constant-volume air-handling unit. He divided the control response into three regions, fully closed to 15°, 15° to 60°, and the last one from 60° to fully open.

He indicate that these three-region does not appear to agree with the published control curves for the dampers. The use of parallel blade dampers provided little mixing in this system. He concluded that there are many factors affect the performance of a mixing box including the type of dampers used.

Seem, House and Klaassen [10] in their article, used a new control system instead of the traditional control for an air handling unit. They used a volume matching control strategy to control the return fan. The traditional control system links the position of exhaust air damper, recirculation air damper and outdoor air damper. The exhaust and outdoor air dampers are normally closed and the recirculation air damper is normally open. In the new control strategy, the outdoor air damper was fully open, and the position of the exhaust and recirculation dampers were linked during occupied times. They indicated that this method could prevent air from entering the air handling unit though the exhaust air outlet in special situations where recirculation is possible.

Van Becelaere [11] tested three typically shaped blade dampers used in manufactured air-handling units. Each of the three dampers has five configurations. He concluded that parallel blade dampers are optimal for factory-built air-handling units based on the result of a drastic increase in pressure on the opposed blade dampers. A 3.5 inch airfoil blade had the most constant static pressure for all flow rates. The parallel blade configuration of the dampers with interacting throw the air toward one another is the best arrangement from the air mixing standpoint.

Zhao [12] developed a numerical model of an HVAC system, with an economizer, for a single room ventilation. This numerical model enabled analysis and simulation system performance. The numerical model was validated experimentally in the laboratory. Only outdoor air damper and recirculation damper were coupled while the discharge damper remained open during operation. This strategy is called two coupled damper control strategy. The dampers characteristics curve as function of control signal were linearized by means of a software algorithm. The linearization algorithm was based on thin, flat plate, non-overlapping blades. The compensated signal, the control signal sent to the motor, was determined as a function of the ideal signal. The ideal signal is the signal used by the PI algorithm. The area compensation method reduces the fan power requirement. The simulation and experimental results of this new control strategy showed an approximate 10% saving of fan energy compared to conventional three coupled damper control strategy. Whereas the linearization method was based on a simplified damper model, the control range was limited to control signals between 20% and 85%.

CHAPTER 3

THEORETICAL ANALYSIS WITH ILLUSTRATIONS

The primary objective of the theoretical analysis was to develop a method to 'linearize' the damper characteristics by either design of linkage or software algorithm. A linearized damper characteristic is one that has a flow coefficient linearly proportional to the control signal. A secondary objective was to relate the excess power required by a system of dampers, i.e., an economizer, due to the damper characteristics. To achieve these objectives, it was required to relate the geometric design of a damper linkage to the fluid dynamic characteristics of the damper. Analyses of the aerodynamics, kinematics and dynamics were therefore required.

3.1 Economizer

The economizer shown in Figure 3.1 consists of three dampers, an outdoor air damper (ODA), a recirculation air damper (REC), and a discharge air damper (DIS). The three dampers are of the same size and type. The dampers are operated by electric motors (actuators) which have a 90 degree rotation range. This range allows the dampers to be controlled from fully closed to fully open and in any specific position in between according to the control signal that the motors receive from the controller. During the economizer control the ODA and DIS dampers are rotated in the same direction while the REC damper is rotated in the opposite direction. In other words, the REC air damper rotated to its closed position while the ODA and DIS air dampers rotated to their open positions and vice versa.

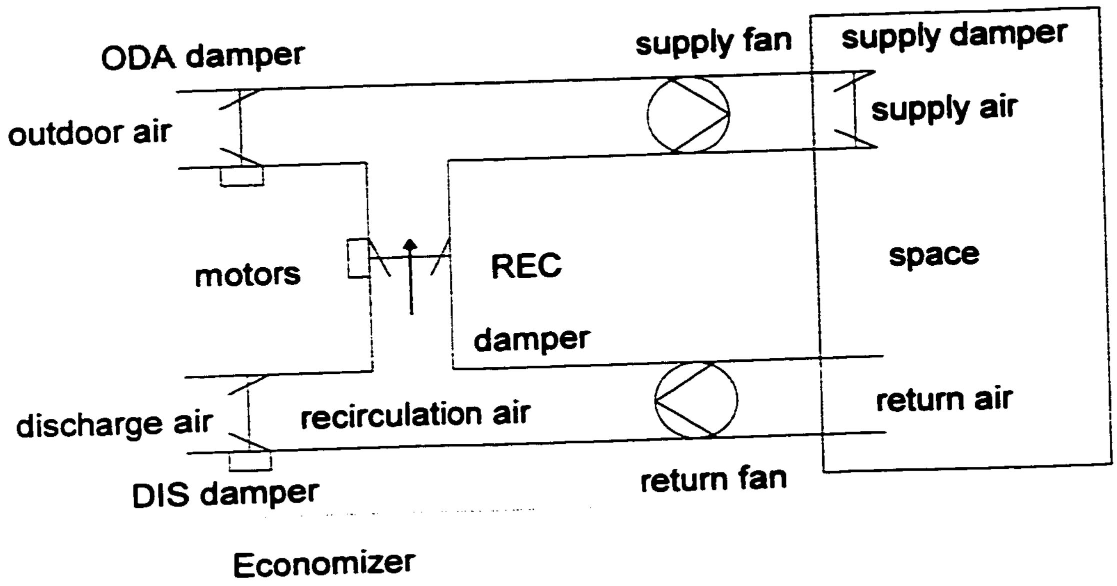


Figure 3.1 Air Distribution System with an Economizer.

The main function of an economizer is to provide the conditioned space with a sufficient quantity of outdoor air, i.e., fresh air, at all times to assure good indoor air quality. The economizer controls the ratio of the outdoor air to the recirculation air. Furthermore, by using the economizer the space can be pressurized to prevent infiltration of unconditioned outdoor air.

The economizer may be controlled under (1) energy considerations during air conditioning, or (2) indoor air quality (IAQ) considerations. For example, if a recirculation air enthalpy is greater than the outdoor air enthalpy, 100% outdoor air is used. If space CO_2 concentration is too high, more outdoor air is supplied.

3.2 Geometry of the Damper

Mainly, dampers are of two types: opposed blade dampers where the adjacent blades rotate in opposite directions and parallel blade dampers where all blades rotate in the same direction. In this study all the dampers are opposed blade dampers. A cross-section of a damper is shown in Figure 3.2.

The damper consists of the following components.

- The frame: The frame is constructed of extruded aluminum.
- Blades: The blades are extruded aluminum.
- Crank arms: A crank arm is fixed to a blade axis at one end and to a pivot connecting it to the connector link at the other end. The crank arm is fixed at 45° with respect to the blade center line. There are two kinds of crank arms:
  1. α crank arms - clockwise rotation opens the damper (viewed from linkage side), and
  2. β crank arms - counter clockwise rotation opens the damper (viewed from linkage side).
- Connector links: A connector link connects the pivot ends of the crank arms. At least two crank arms (blades) must be connect to each connector link so that it moves parallel to the blade axes. There are two connector links:

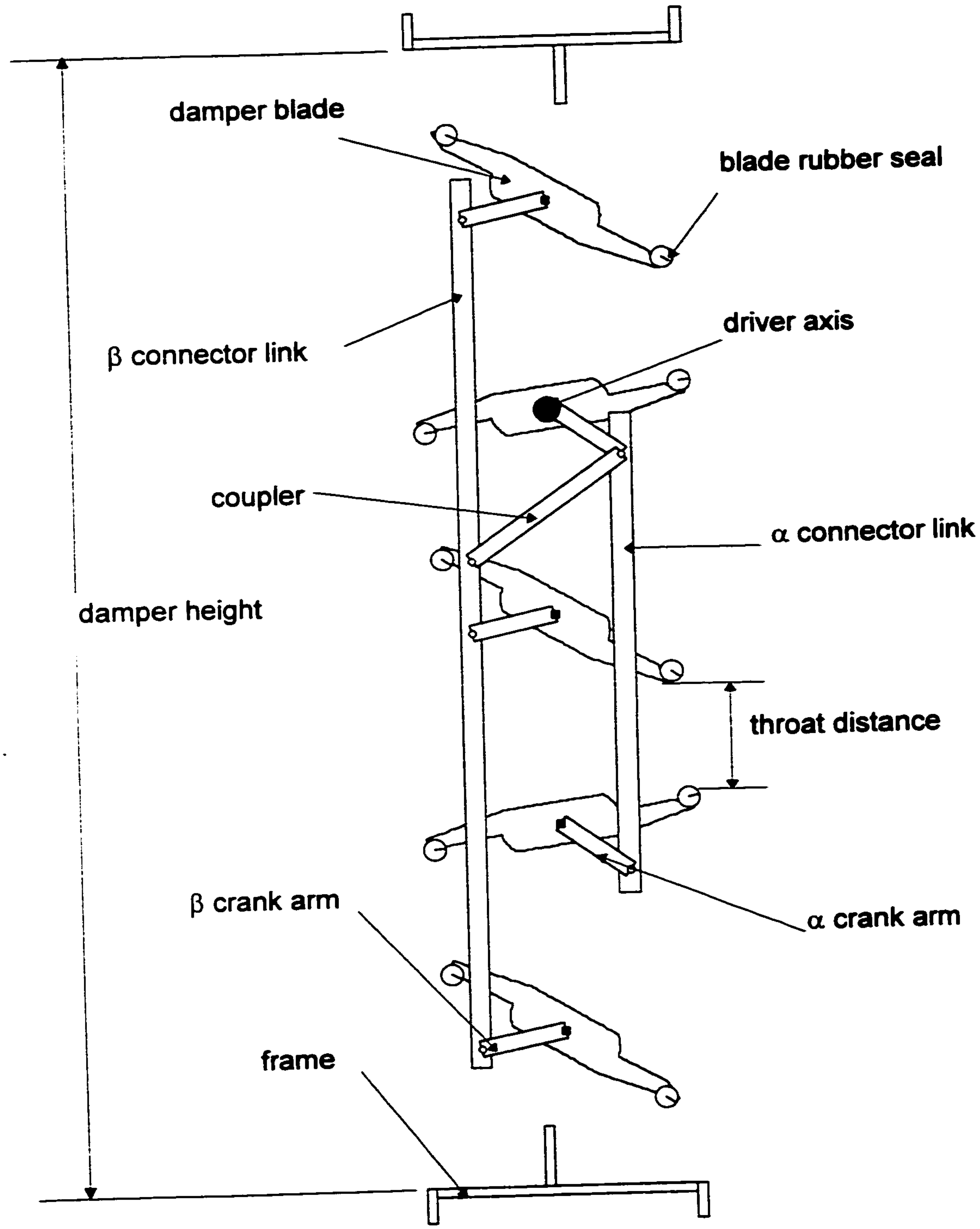


Figure 3.2 Damper Geometry.

1. an α connector link - connects α crank arms pivots, and
2. a β connector link - connects β crank arms pivots.

- Coupler: The coupler connects the α connector link and the β connector link to transmit motion from the driver to the driven crank arms.
- Air leakage seals: Rubber seals are mounted on both tips of each blade, and the right and left sides of the frame. The seals prevent air leakage while the damper is closed.

Table 3.1 shows the dimensions of the original damper as supplied by the manufacturer.

Table 3.1 Dimensions of Damper Components.

Actual damper component dimensions			
component	length (in)	width (in)	thickness, depth or diameter (in)
frame	18	18	4 depth
blade	18	4	0.5 thickness (middle)
crank arm	1.25	0.25	0.25 thickness
connector links	6.5 & 13	---	0.313 diameter
coupler	2.75	0.75	0.25 thickness

Dampers with this specific design were installed in the experimental HVAC system as an economizer. Each damper was controlled to allow 0% to 100% of the supply air flow rate. The throat area characteristics of these dampers were not a linear function of the control signal. This fact has a great effect on the system control. The amount of air flows through the damper is primarily a function of the damper throat area. It will be shown that if the damper design is improved, based on a linear damper throat area characteristic, the system control will be improved and the HVAC system will be more effective and more economical.

3.3 Aerodynamic Analysis

In the economizer system shown in Figure 3.1, the supply fan is used to provide the space with the conditioned air. During the economizer operation the supplied air is either completely outdoor or mixed (outdoor and recirculation) air depending on the amount of outdoor air required for comfort in a space. The supply fan draws the outdoor air into the system through the outdoor air damper (ODA). The return fan is used to draw air back from the conditioned space and discharge it into the economizer section. Depending on the conditions of the outdoor air and return air, the return air can be either returned to the system through the recirculation air damper (REC), or exhausted outdoor through the discharge air damper (DIS).

3.3.1 Flow Coefficient

The air flow rate through a damper can be related to the pressure drop across the damper using a flow coefficient, C_v, by

$$Q = C_v \times \sqrt{\Delta P} \tag{3.1}$$

In the current investigation the flow coefficient was calculated theoretically and obtained experimentally. The two flow coefficients were compared to develop a correlation coefficient between the theoretical and actual (experimental) values. Because of the complexity of the air flow rate across a damper, it is customary to make simplifying assumptions. The damper shown in Figure 3.3 may be considered as an orifice because of its geometry. The schematic of the equivalent orifice for the damper is shown in Figure 3.4.

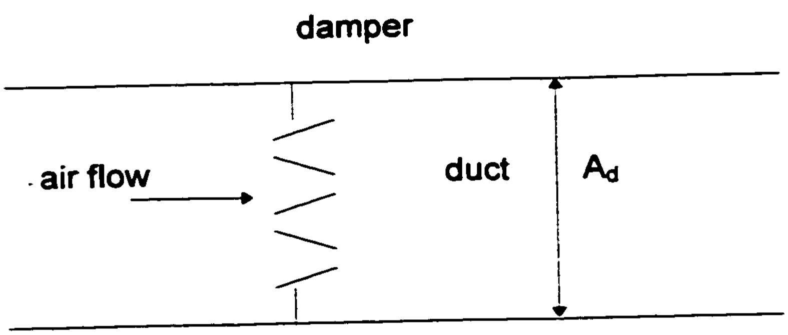


Figure 3.3 Damper in Duct Schematic.

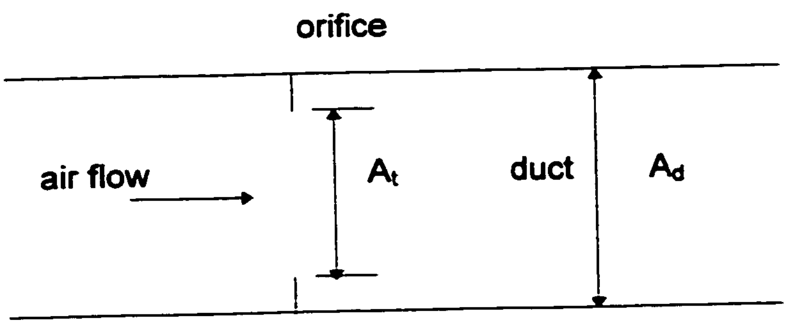


Figure 3.4 The Equivalent Orifice for the Damper.

The theoretical relation used to determine the air flow rate in terms of the pressure drop across the orifice can be found in several fluid mechanics text books. The relation proved by White [13] is

$$Q = \sqrt{\frac{2}{\rho}} \times \frac{C_d}{\left(1 - \frac{A_t^2}{A_d^2}\right)^{\frac{1}{2}}} \times A_t \times \sqrt{\Delta P} \tag{3.2}$$

By comparing Equation 3.1 and Equation 3.2 the flow coefficient C_V can be written as

$$C_V = \sqrt{\frac{2}{\rho}} \times \frac{C_d}{\left(1 - \frac{A_t^2}{A_d^2}\right)^{0.5}} \times A_t \tag{3.3}$$

where C_d is a discharge coefficient that accounts for the viscous and turbulent effects. The flow coefficient C_V can be presented as a function of the throat area, A_t. Equation 3.3 can be reduced to an equivalent form

$$C_V = f \times \sqrt{\frac{2}{\rho}} \times A_t \tag{3.4}$$

where

$$f = \frac{C_d}{\left(1 - \frac{A_t^2}{A_d^2}\right)^{1/2}} \tag{3.5}$$

The correlation coefficient, f, is a dimensionless coefficient used to correlate the theoretical relation with experimental data. It will be obtained by comparing the experimental and theoretical graphs of the C_V versus the damper control signal. The constant part $\sqrt{2/\rho}$ of Equation 3.4 is calculated based on the British unit system as 4007 ft/(min·inWC$^{1/2}$) for an air density.

The damper throat area is equal to the product of the number of damper blades, the throat distance between adjacent blades, and the damper width. The effects due to the damper frame are neglected. The damper throat area is changed by varying the angular position of the damper blades. The throat distance between adjacent blades varies from 0% to 100%, depending on the control signal. For example, If the control signal increases, the throat distance increases. Therefore, the damper throat area is a function of the throat distance between adjacent blades which, in turn, is a function of the control signal.

3.3.1.1 Illustration of the Correlation of Flow Coefficient and Throat Distance

The correlation of flow coefficient and throat area may be illustrated using analytical and actual experimental data. The correlation factor was obtained by the comparison of theoretical and actual flow coefficients. The theoretical flow coefficients were obtained from Equation 3.4 using throat areas. The throat distance was measured in two ways, using a software program called Working Model, and manually. In each way, the throat distance was determined by visual inspection.

Working Model 2D is a software program for graphic kinematic analysis of mechanisms [14]. The damper is simulated for both geometry and control by this software. The damper geometry was shown in Figure 3.2. A motor is connected to the α crank arm. The motor position may be varied over a range of the rotation angel between 0° and 90°. The rotation angle is linearly proportional to the control signal. For example, 0° corresponds to a 0% signal, 45° corresponds to a 50% signal, and 90° corresponds to a 100% signal. Figure 3.5 shows some of the damper simulations at various control signals. At each specific control angle, the throat distance was measured and then plotted as a curve in Figure 3.6. This figure shows the relation between the throat distance between adjacent blades and the control signal.

The throat distance was measured manually from the ODA damper of the experimental HVAC system. The experimental technique used to measure the throat distance is explained in detail in Chapter 4, Section 4.1. The damper was controlled from 0% (fully closed) to 100% (fully open) by a control signal to the damper motor. The control signal was increased by 5% steps. When the damper reached its fully open position, the control signal was decreased by 5% steps until the damper fully closed. At each signal the throat distance was measured.

The measured data, Figure 3.7, showed that the throat distance at the same signal for opening and closing positions are not the same. That is because of the hysteresis caused by the motor and damper mechanism. The motor had one position when the control signal increased and would not have the same position for the same control signal when the signal decreases. The effect of hysteresis is reduced by a software method presented by Krakow [8]. Figure 3.7 shows the relation between the damper control signal and the throat distance after the control signal was corrected for hysteresis equal to 3 %.

Theoretical flow coefficients may be calculated from Equation 3.4 by

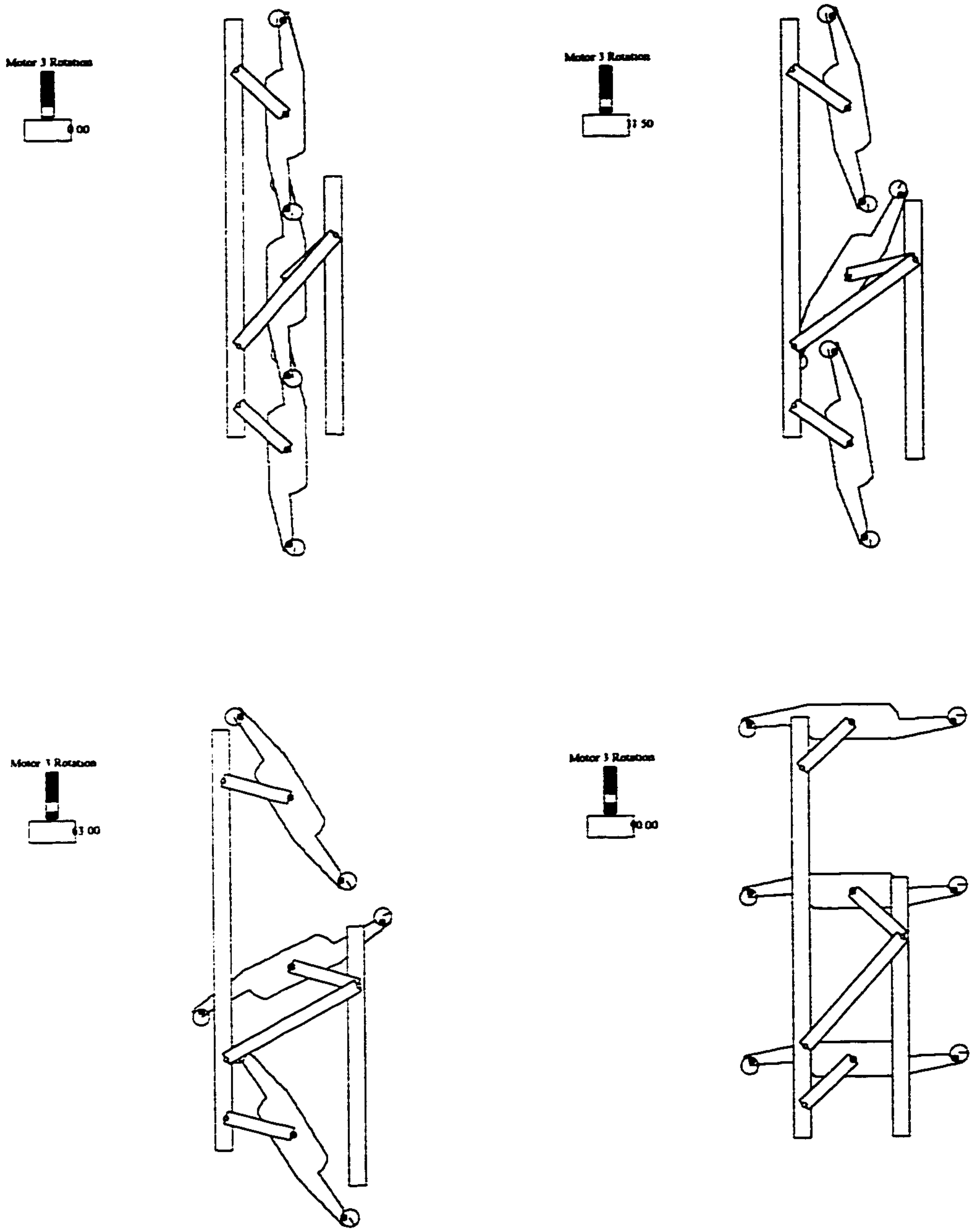


Figure 3.5 The Damper as Simulated by Working Model software. Top left — 0% signal (fully closed), Top right — 35% signal, Bottom left — 70% signal, Bottom right — 100% signal (fully open).

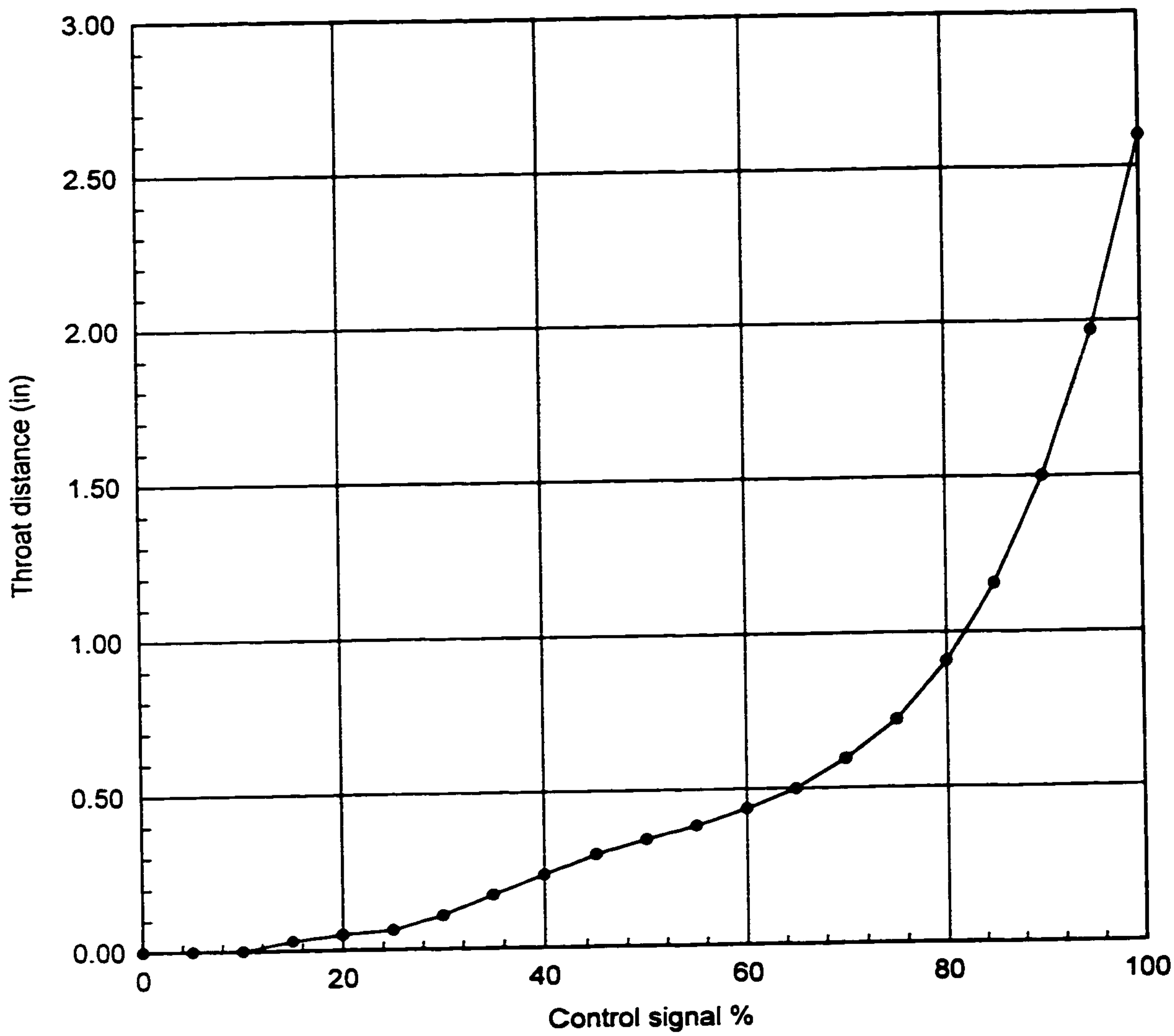


Figure 3.6 Throat Distance Versus Control Signal for the Normally Closed Damper Based on Working Model Measurements.

assuming the value of the correlation coefficient is equal to unity (f = 1.0). Theoretical flow coefficients versus the control signal are illustrated in Figures 3.8 and 3.9. The throat distance was based on the two ways of measurement as shown in Figures 3.6 and 3.7.

Actual experimental flow coefficients for the damper were obtained using Equation 3.1 and experimental data were used to obtain the correlation factor. The actual flow coefficients account for the influences of viscosity, turbulence and the dimensional effects. The HVAC system used to obtain the data is described in Chapter 4, Section 4.2. Figure 3.10 shows the relationship between the flow coefficient and the damper control signal for a normally closed damper. By comparing Figure 3.10 to Figure 3.9 (lab measured data), the correlation coefficient f is ~ 0.75. By comparing Figure 3.10 to Figure 3.8 (Working Model data), f is ~ 1.0. The Working Model simulation does not include hysteresis effects. Hysteresis effects are most evident at high values of the control signal. In either case, it may be assumed that the flow coefficient is primarily dependent on the throat distance.

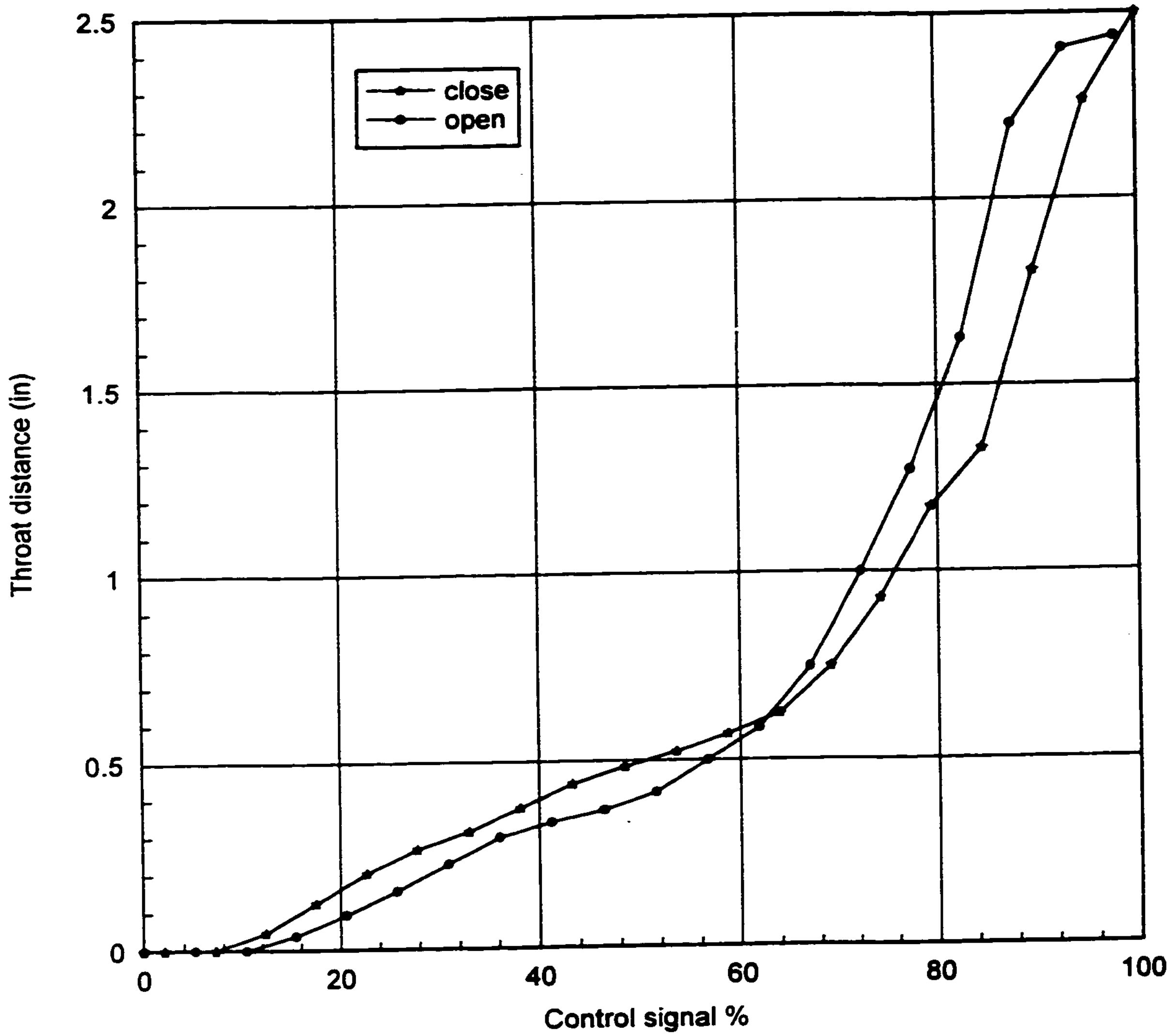


Figure 3.7 Throat Distance Versus Control Signal for the Normally Closed Damper Based on Lab System Measurements.

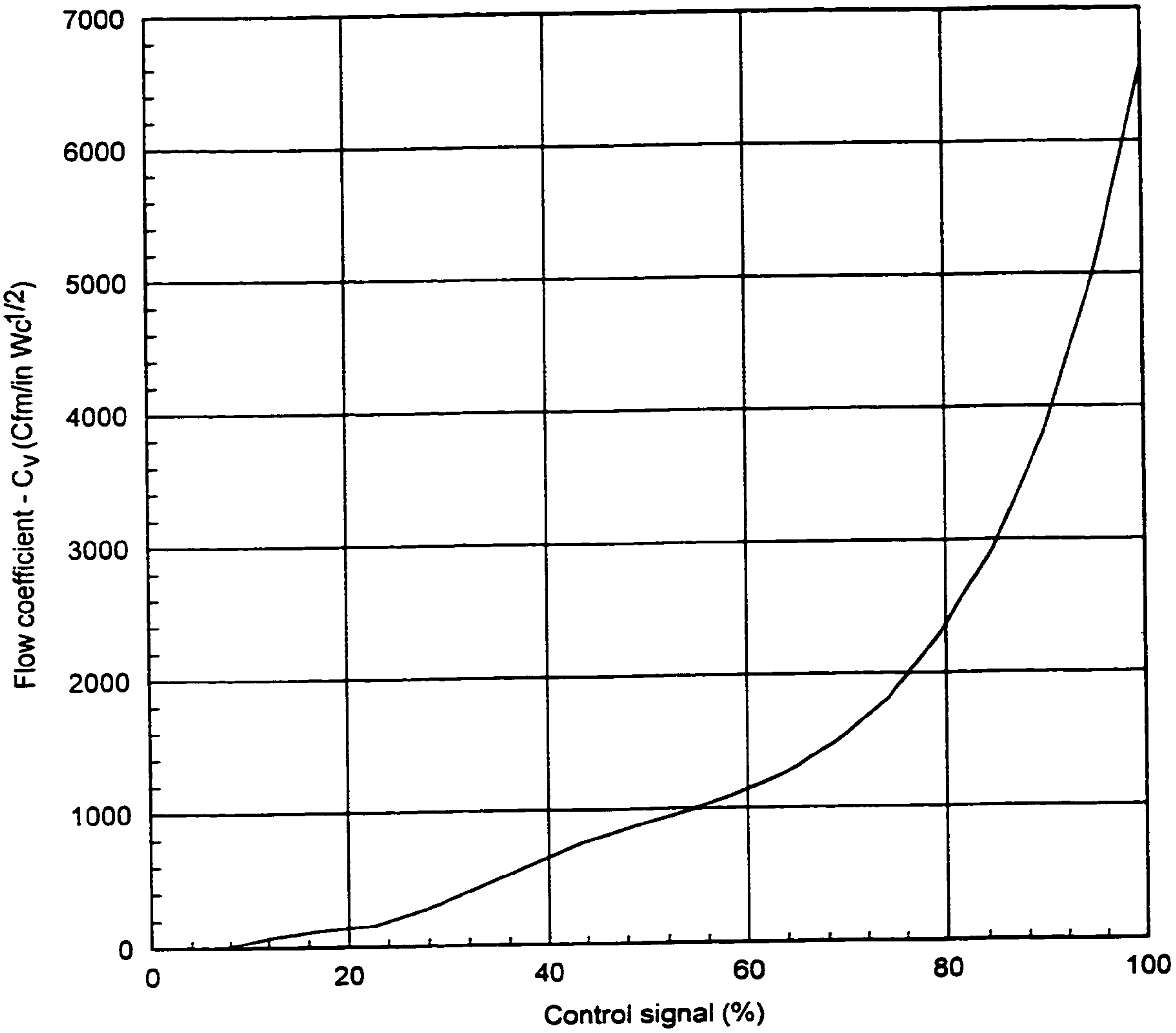


Figure 3.8 Theoretical Flow Coefficients Versus Control Signal for the Normally Closed Damper Based on Working Model Measurements, Equation 3.4, with f = 1.0.

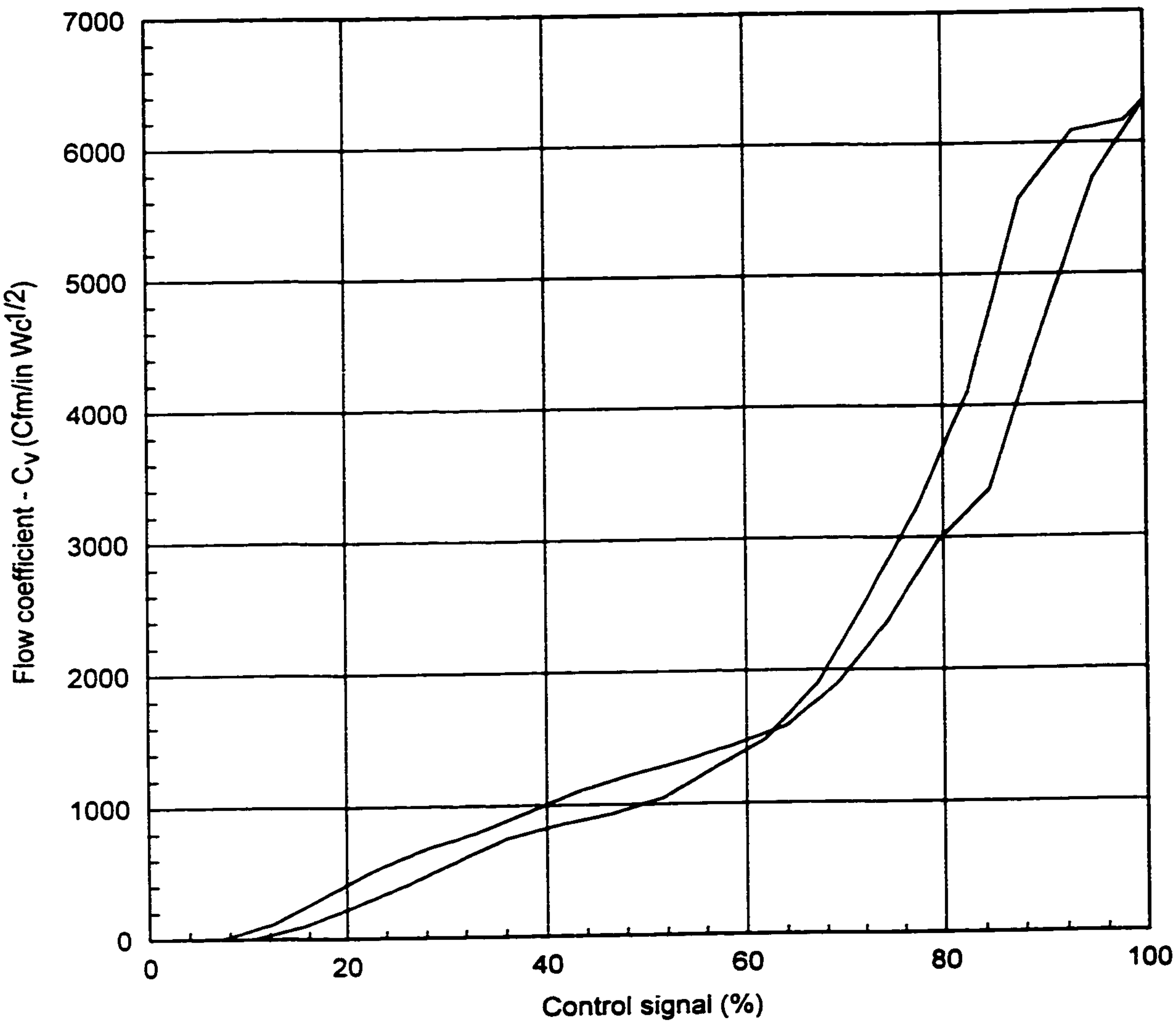


Figure 3.9 Theoretical Flow Coefficients Versus Control Signal for the Normally Closed Damper Based on Lab System Measurements, Equation 3.4, with f = 1.0.

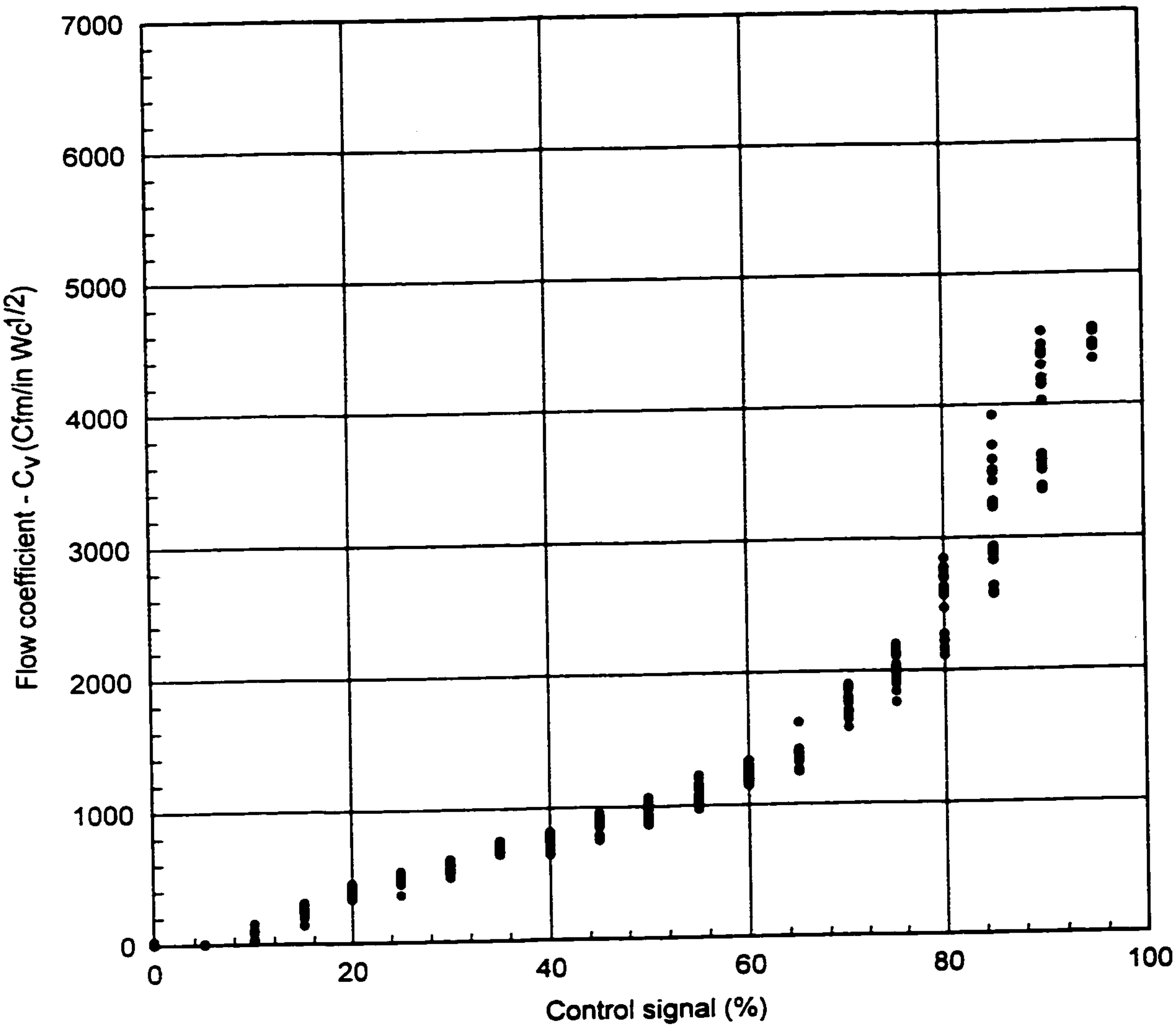


Figure 3.10 Experimental Flow Coefficient for the Normally Closed Damper.

3.3.2 Fan Power Consideration

The purpose of fans in an HVAC system is to propel air at an adequate velocity to provide the space with the required quantity of air for ventilation and for heating or cooling. The HVAC fans can be classified under two categories axial and centrifugal fans. The centrifugal fans are the most widely used in HVAC systems. In centrifugal fans, air enters from one or both sides at the center and propelled outward to the outlet where the air discharged to the air duct system.

The HVAC system for this study has two fans, both of them are centrifugal fans, a supply air fan and a return air fan. Each fan was driven by a variable speed electrical motor. The motor was operated and controlled by an adjustable frequency control system. The air flow quantity supplied to the space, and returned to the economizer could be increased or decreased depending on the fan rotational speed.

The fan power required to distribute the air based on the pressure rise is

$$W = Q \times \Delta P \tag{3.6}$$

Noting, the designation of the key points in Figure 3.11 the power for supply fan is

$$W_{Sup} = Q \times (P_2 - P_1) \tag{3.7}$$

and for return fan is

$$W_{Rtn} = Q \times (P_5 - P_4) \tag{3.8}$$

The total power is the algebraic sum of the supply and return fan powers and is given by Equation 3.9 where the supply air flow rate and the return air flow rate are equal.

$$W_T = Q \times \left[(P_2 - P_1) + (P_5 - P_4)\right] \tag{3.9}$$

Equation 3.9 can be rearranged to give the total fan power as

$$W_T = Q \times \left[(P_5 - P_1) + (P_2 - P_4)\right] \tag{3.10}$$

where

$$P_2 - P_4 = (P_2 - P_3) + (P_3 - P_4) \tag{3.11}$$

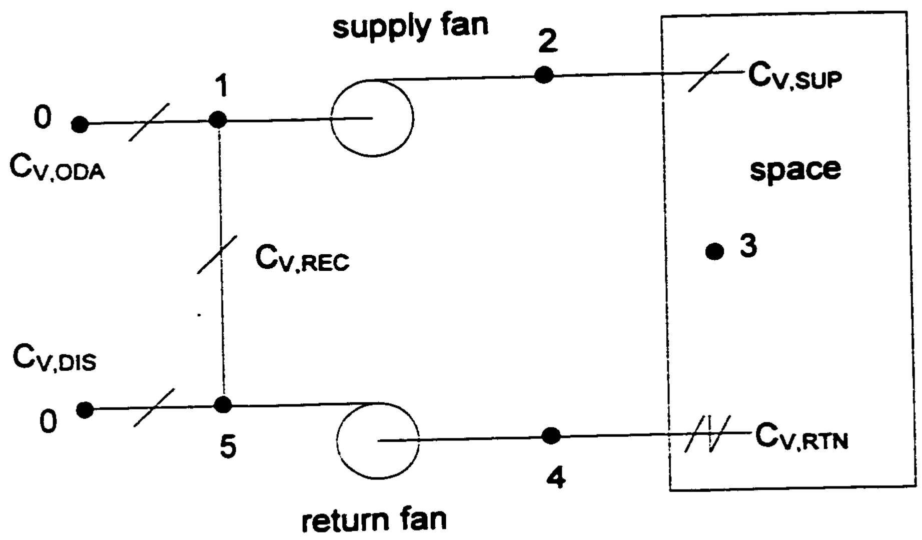


Figure 3.11 Schematic Diagram Showing the Keys points for System Pressures.

The space pressure was controlled to be equal to the atmospheric pressure (zero gauge pressure) and was called room pressure, P_3 . Substituting Equation 3.11 into 3.10 yields

$$W_T = Q \times \left[(P_5 - P_1) + (P_2 - P_3) + (P_3 - P_4)\right] \tag{3.12}$$

The minimum power requirement for the system is the power due to system components, excluding the economizer.

Since

$$P_2 - P_3 = \frac{Q^2}{C_{V,SUP}^{\;2}} \tag{3.13}$$

$$P_3 - P_4 = \frac{Q^2}{C_{V,RTN}^{\;2}} \tag{3.14}$$

Equation 3.12 can be rewritten as

$$W_T = Q \times (P_5 - P_1) + W_{MIN} \tag{3.15}$$

where the minimum power in terms of the flow coefficients and flow rate is

$$W_{Min} = Q^3 \times \left[\frac{1}{C_{V,SUP}^{\;2}} + \frac{1}{C_{V,RTN}^{\;2}}\right] \tag{3.16}$$

and the subscripts SUP and RTN refer to supply and return ducts respectively.

In an ideal economizer there would be no pressure drop across dampers. The dampers would serve to direct the flow, not to throttle the flow. Therefore, for an ideal economizer system

$$P_1 = P_5 = P_{atm} \tag{3.17}$$

and the minimum fan power in terms of pressures and flow rate is

$$W_{MIN} = W_T\big|_{MIN} = Q \times \left((P_2 - P_3) + (P_3 - P_4)\right) \tag{3.18}$$

Therefore, the power required due to economizer dampers is represented by

$$W_{Excess} = W_T - W_{MIN} = Q \times (P_5 - P_1) \tag{3.19}$$

where $(P_5 - P_1)$ is the pressure drop across the recirculation damper.

If the economizer control is optimized, the pressure across the recirculation damper is reduced, and a saving in the fans power consumption can be achieved.

3.4 Kinematic and Dynamic Analyses

The damper linkage, as shown in Figure 3.2, is located on one side of the damper. The other side has only bearings. The damper frame, 2 α crank arms and the α connector constitute a four-bar linkage. The frame, 2 β crank arms and the β connector constitute another four-bar linkage. The angle of the α crank arm, when the damper is closed is 45° with respect to an X-axis (angle α), while the β crank arm makes a 135° angle with respect to an X-axis (angle β), as shown in Figure 3.12. Figure 3.13 shows the damper linkage when the damper is at its open position where angle $\alpha = -45^{\circ}$ and angle $\beta = 225^{\circ}$.

The damper can be controlled in two ways.

1- An α crank arm is the driver. The damper as supplied by the manufacturer was controlled in this way.

2- A β crank arm is the driver. A modified damper was controlled in this way.

3.4.1 Analysis with Working Model Software

The coupler length of the damper supplied by the manufacturer was (driven from α) was 2.75 inches. The throat area characteristic of this damper was not linear. To modify the existing damper, only the coupler and the motor

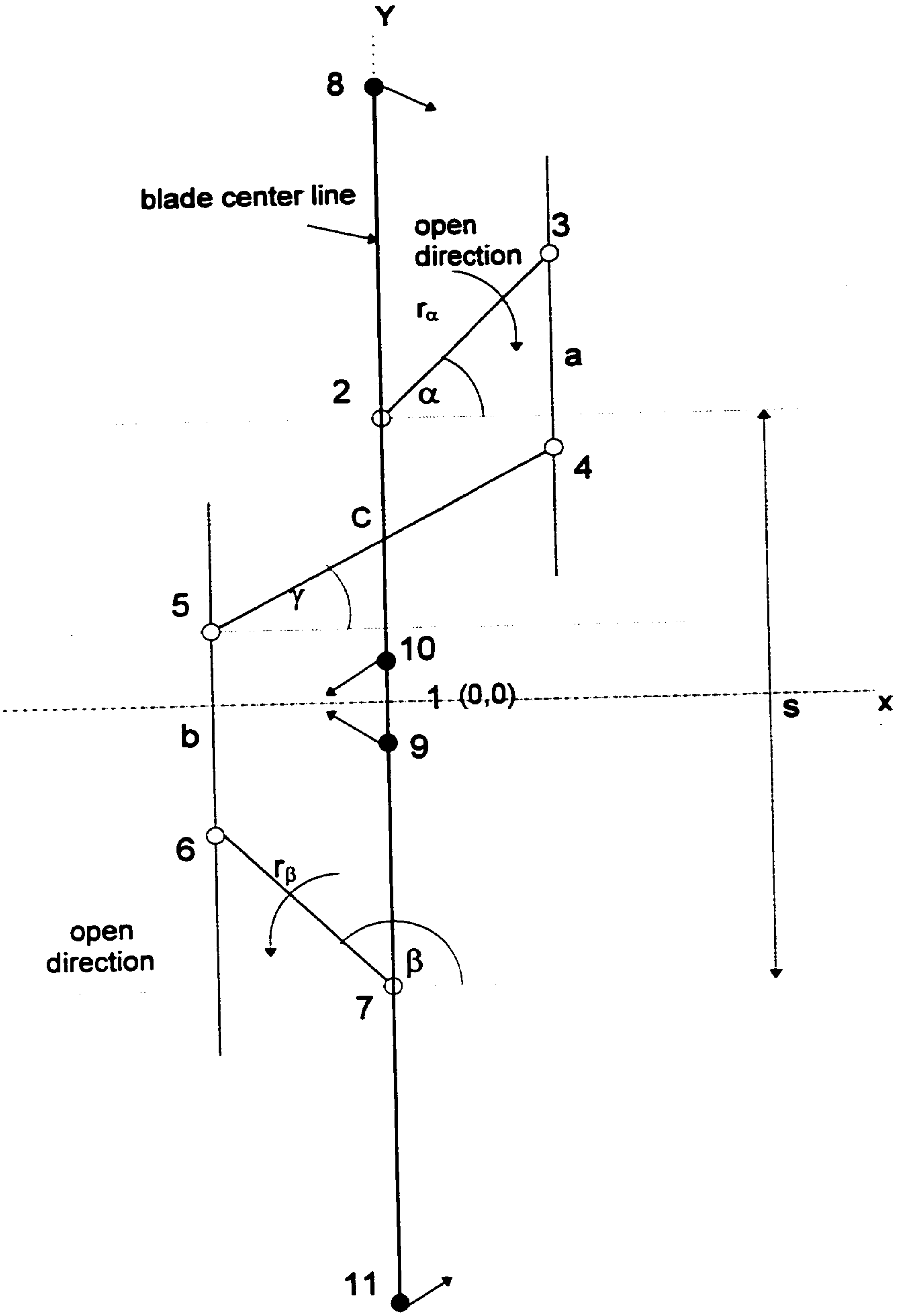


Figure 3.12 Damper Linkage Mechanism Showing a Damper in a Closed Position. Lines 8-9 and 10-11 represent the blades.

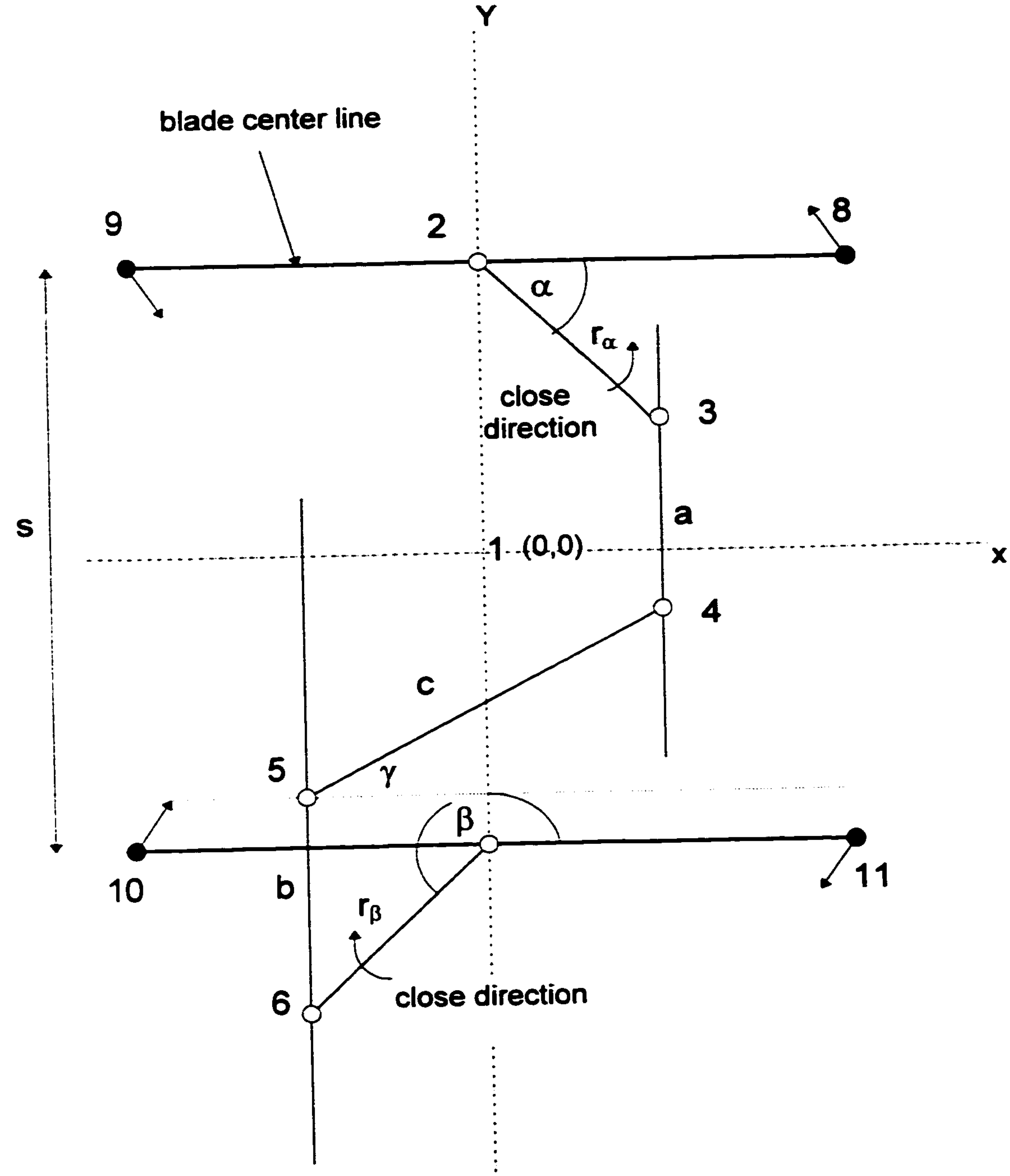
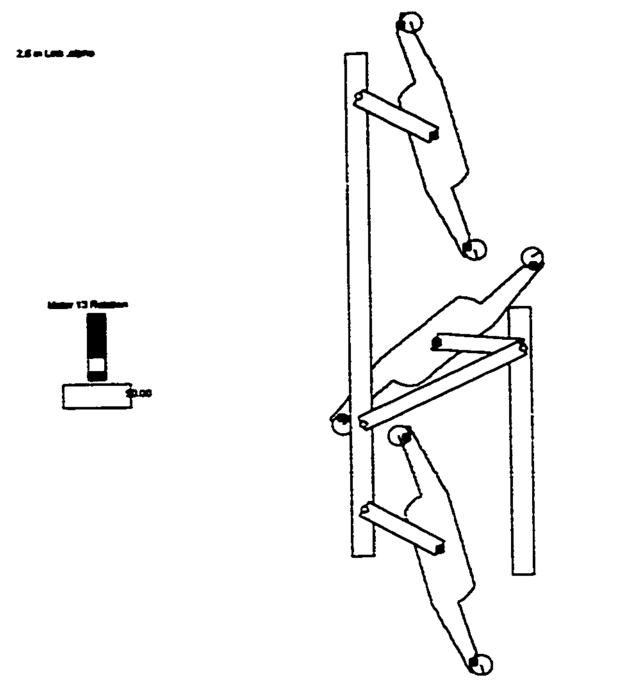
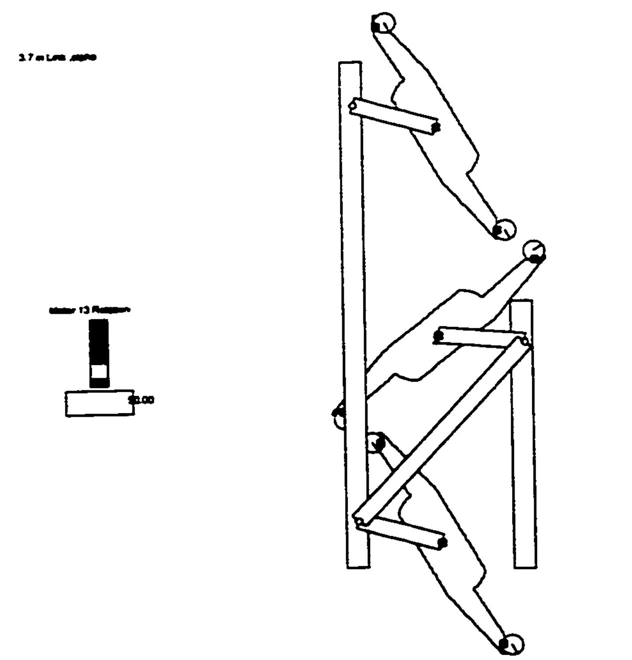


Figure 3.13 Damper Linkage Mechanism Showing a Damper in an Open Position. Lines 8-9 and 10-11 represent the blades.



(a) for a 2.6 inch coupler length



(b) for a 3.7 inch coupler length

Figure 3.14 Working Model Simulation for the Damper Driven from an α Axis.

location (the damper is driven from α or from β) could be changed. However, for a new damper the blade axis to axis distance can also be changed.

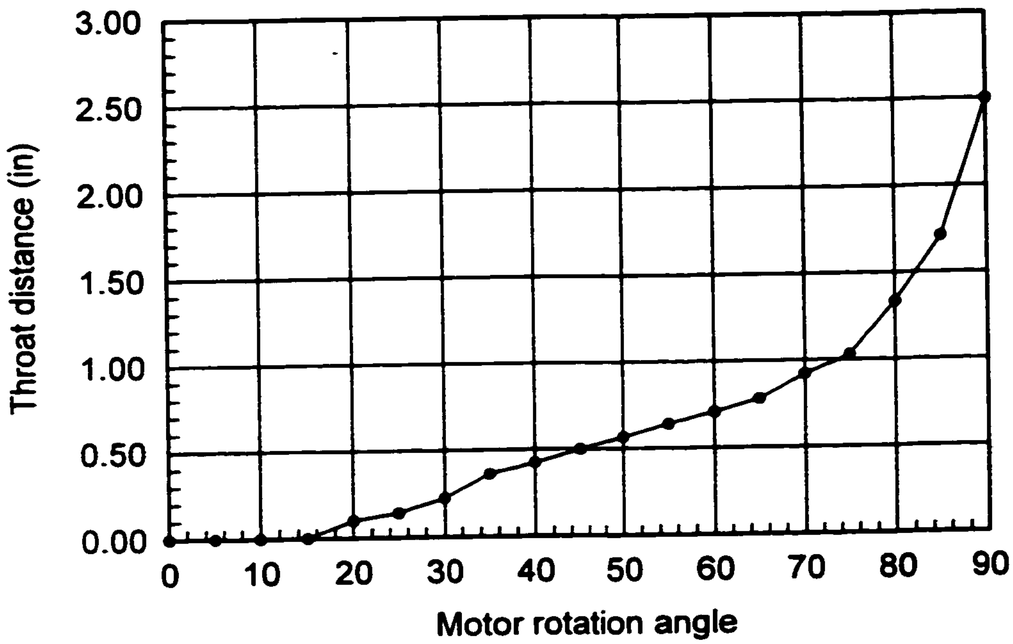
The damper was simulated by Working Model as shown in Figures 3.14 (driven from α) and 3.16 (driven from β) for 2.6 and 3.7 inches coupler lengths. The throat distance characteristic for each damper is illustrated in Figures 3.15 (driven from α) and 3.17(driven from β). The damper will not open fully if its coupler length is 2.6 inches and is driven from α. The throat distance is a linear function of motor rotation angle, as can be seen in Figure 3.17(a), when the damper is driven from β. Practically, a hardware 2.6 inches coupler was installed but it could not be closed tightly after it was opened.

A modified damper, used for final testing, was a damper driven from β and had a coupler length of 3.7 inches. This damper required less torque and closed completely. Figure 3.16(b) shows the Working Model simulation of this damper and Figure 3.17(b) shows its throat distance characteristic.

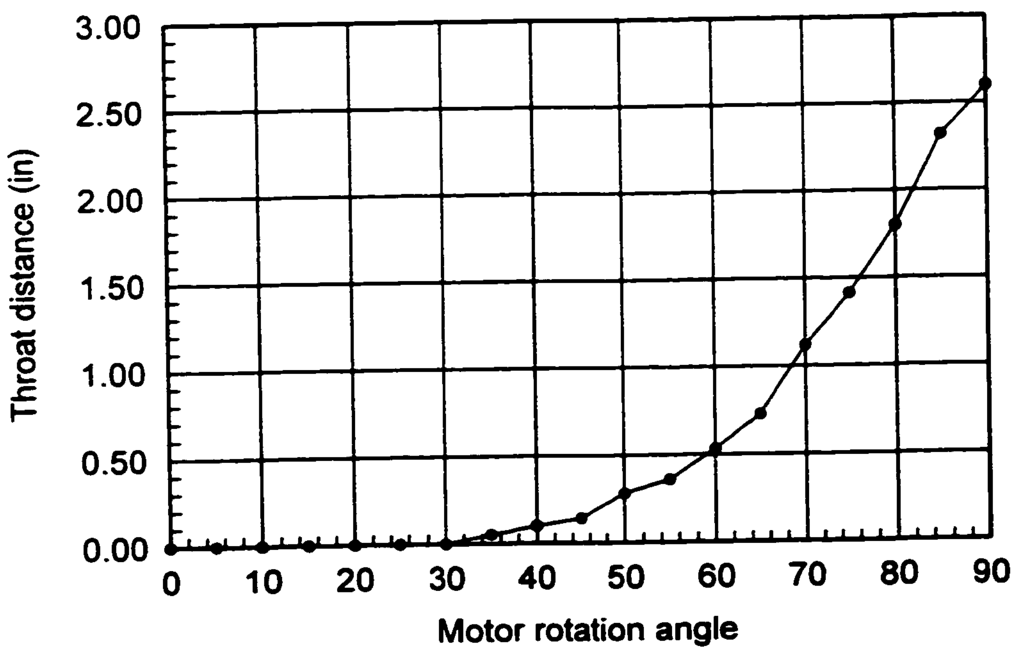
Working Model does not indicate problems related to torque. Practically, motors would not be able to close the linear damper tightly even though motors were rated for much larger dampers. A FORTRAN program was developed to further assist in analyzing the linkage from a dynamic perspective.

3.4.2 Analysis with FORTRAN program

A sufficient torque should be applied to the damper to open it, or close it. Bearings friction and seal friction will resist this torque. The seal friction is greatest when the damper is closed. The torque considerations are based on

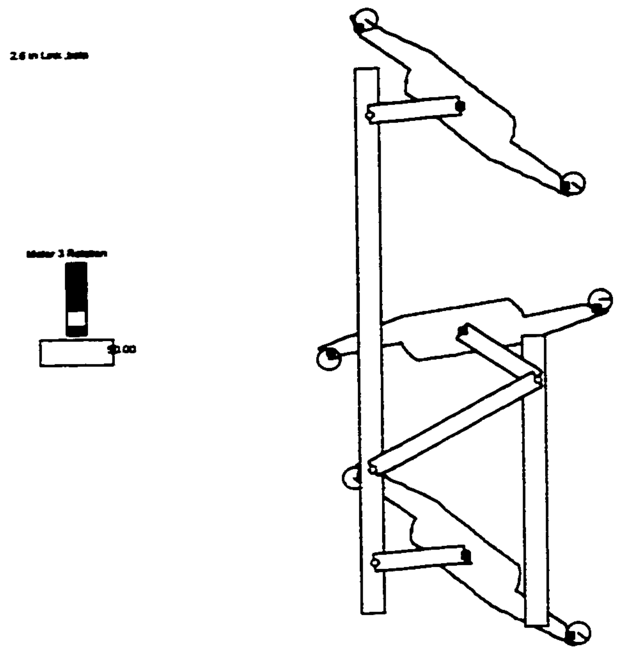


(a) for a 2.6 inch coupler length

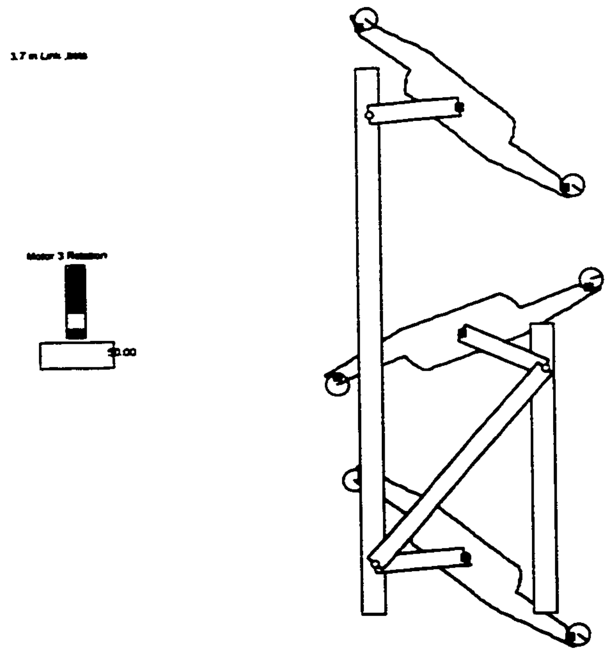


(b) for a 3.7inch coupler length

Figure 3.15 Throat Distance Characteristic for the Damper Driven from an α Axis.

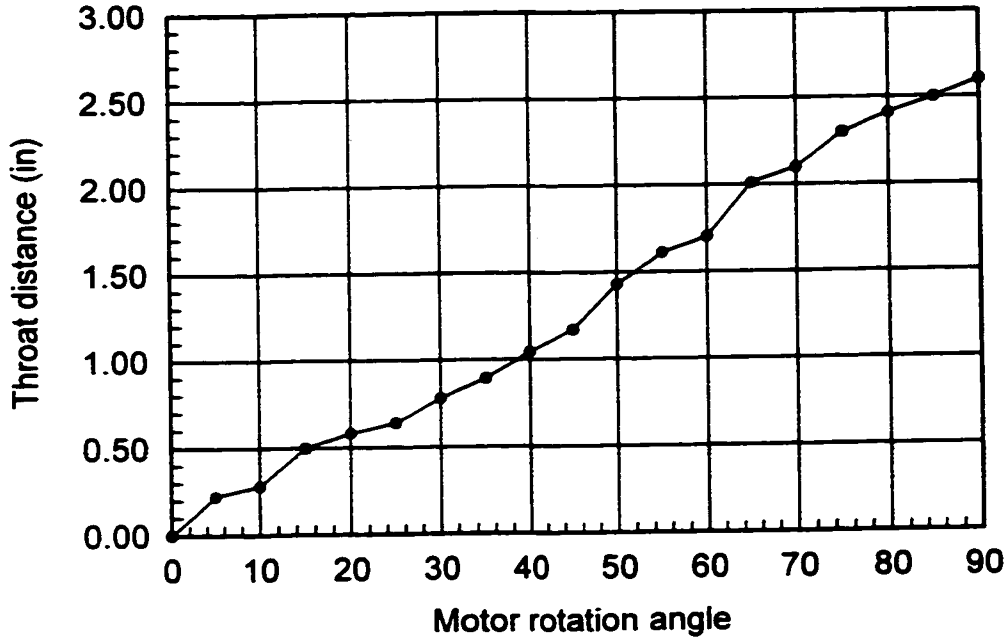


(a) for a 2.6 inch coupler length

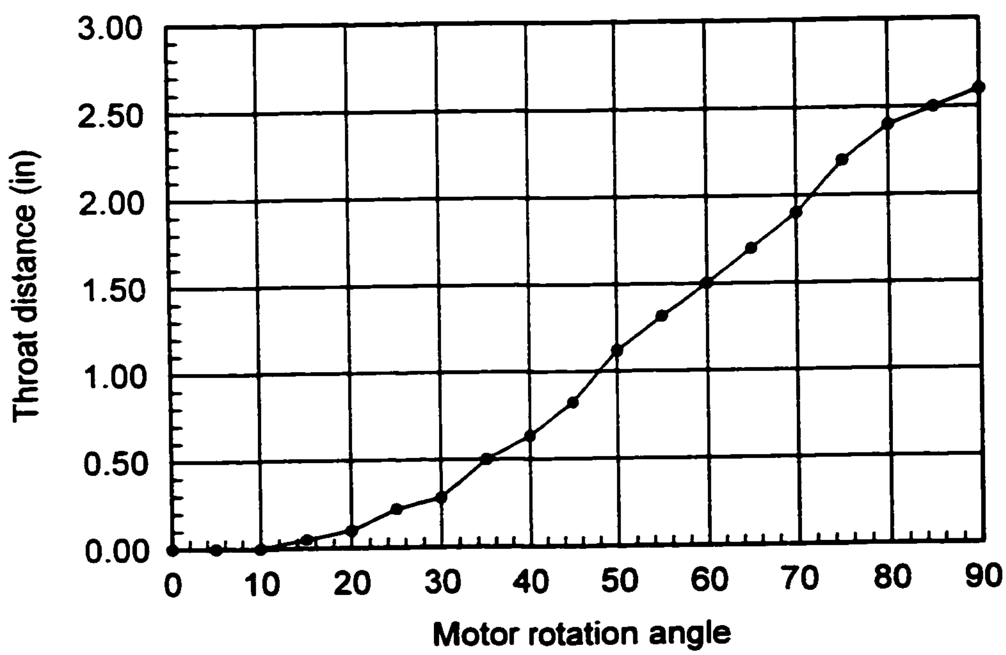


(b) for a 3.7 inch coupler length

Figure 3.16 Working Model Simulation for the Damper Driven from a β Axis.



(a) for a 2.6 inch coupler length



(b) for a 3.7inch coupler length

Figure 3.17 Throat Distance Characteristic for the Damper Driven from a β Axis.

$\Delta\beta/\Delta\alpha$ if the damper is driven from α, and on $\Delta\alpha/\Delta\beta$ if the damper is driven from β. The Method of Virtual work was applied to obtain a torque coefficient. The principle of virtual work states that "if a system of connected rigid bodies is in equilibrium under various external forces, and if the system is given an arbitrary displacement from that position of equilibrium, the total work done by the external forces during the displacement is zero" [15].

Referring to Figure 3.2 the damper has 2 blades connected by α connector and 3 blades connected by β connector. The bearing and seal friction is counted according to the number of blades. If the damper is driven from α, α crank arm is the driver and β crank arm is the follower. The direction of motion is as shown in Figures 3.12 and 3.13. The Method of virtual work, applied to the linkage, yields

$$\Gamma_D \times \Delta\alpha = \Sigma\Gamma_\alpha \times \Delta\alpha + \Sigma\Gamma_\beta \times \Delta\beta \tag{3.20}$$

where,

$$\Gamma_\alpha = \mathbf{n}_\alpha \times \Gamma_R \tag{3.21}$$

$$\Gamma_\beta = \mathbf{n}_\beta \times \Gamma_R \tag{3.22}$$

Substituting Equations 3.21 and 3.22 into Equation 3.20 yields

$$\Gamma_D = \Gamma_R \times \left(\mathbf{n}_\alpha + \mathbf{n}_\beta \times \left(\frac{\Delta\beta}{\Delta\alpha}\right)\right) \tag{3.23}$$

From Equation 3.23, the drive torque Γ_D is proportional to the torque coefficient $\frac{\Delta\beta}{\Delta\alpha}$. If $\frac{\Delta\beta}{\Delta\alpha}$ is high the drive torque Γ_D is high too, and if $\frac{\Delta\beta}{\Delta\alpha}$ is low then Γ_D is low. A high value of $\frac{\Delta\beta}{\Delta\alpha}$ when the damper starts closing or opening may cause problem. If the damper is driven from β Equation 3.23 becomes

$$\Gamma_D = \Gamma_R \times \left(n_\alpha \times \left(\frac{\Delta\alpha}{\Delta\beta} \right) + n_\beta \right) \quad (3.24)$$

where, the drive torque Γ_D is proportional to the torque coefficient $\frac{\Delta\alpha}{\Delta\beta}$.

The relationship between angle α and angle β may be calculated based on Figure 3.12. The horizontal distance (X-direction) between α connector link and β connector link is

$$c \times Cos(\gamma) = r \times Cos(\alpha) - r \times Cos(\beta) \quad (3.25)$$

The vertical distance (Y-direction) between point 3 and point 6 is

$$a + b + c \times Sin(\gamma) = s + r \times Sin(\alpha) - r \times Sin(\beta) \quad (3.26)$$

Equations 3.25 and 3.26 have 8 variables - r, c, s, a, b, α, β, γ. If 6 variables are specified - r, c, s, a, b, and α or β, then the equations may be solved for the remaining 2 variables α and γ or β and γ.

A FORTRAN program was written to solve these equations. The code is listed in Appendix I. The subroutine ANGL calculates β if α is specified, and vice versa. The program results are illustrated graphically in Figures 3.18 through 3.23.

Figure 3.18 shows β versus α, Figure 3.19 shows Δβ/Δα versus α, and Figure 3.20 shows (β - γ) versus α for coupler lengths of 2.6 and 3.7 inches for the damper driven from α. Figure 3.21 shows α versus β, Figure 3.22 shows

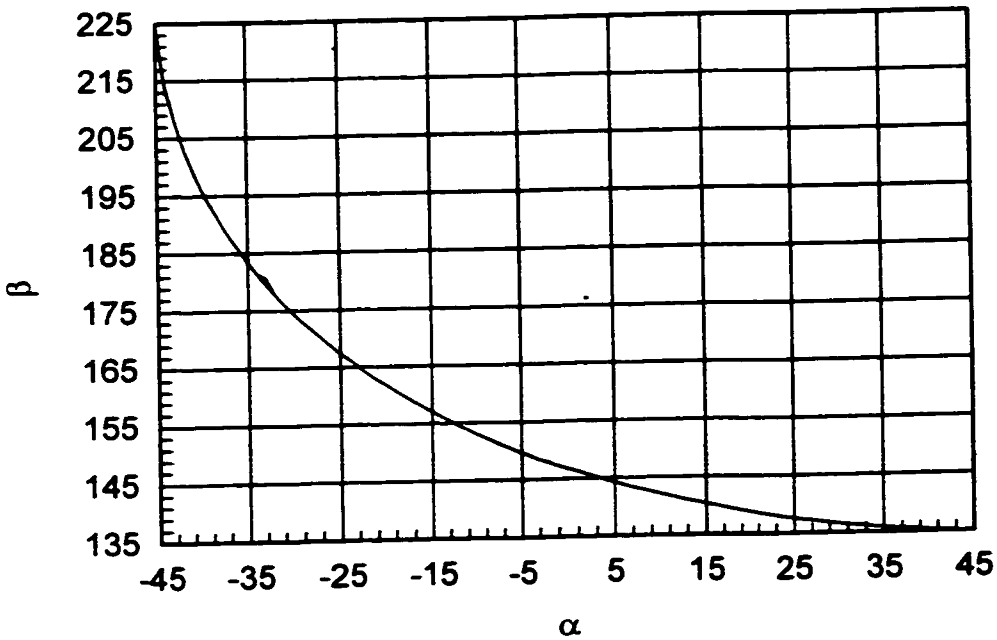
$\Delta\beta/\Delta\alpha$ versus β, and Figure 3.23 shows (γ - α) versus α for coupler lengths of 2.6 and 3.7 inches for the damper driven from β.

The study shows that for high torque coefficient the damper might bind. The damper driven from α will bind if the coupler and β crank arm are aligned. This case will exist if $\beta - \gamma = 180^\circ$. The damper driven from β will bind if the coupler and the α crank arm are aligned, $\gamma - \alpha = 0^\circ$. It may therefore be concluded that if the driven crank arm aligns with the coupler, the damper will bind.

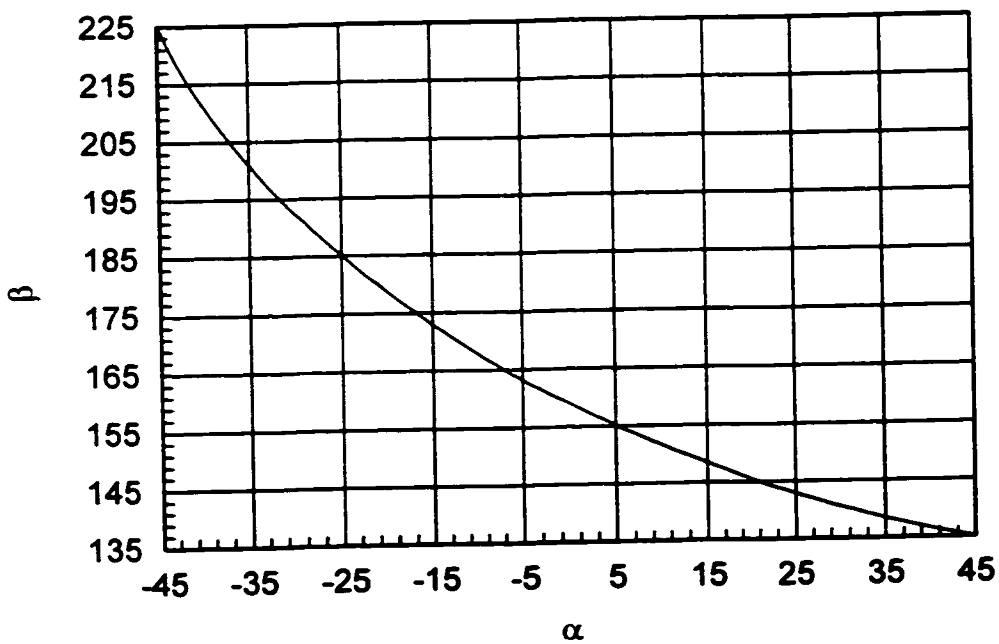
As mentioned in Section 3.4.1, the modified damper has a 3.7 inch coupler and is driven from β. Figure 3.22 (b) shows the torque coefficient is relatively low, i.e., it is not as high as if the coupler length is 2.6 inches.

Based on practical, Working Model, and FORTRAN program studies, some of potential problems are:

1. the damper may not close completely,
2. the damper may not be open completely,
3. excessive torque may be required, and
4. play in linkage may result in (hysteresis).

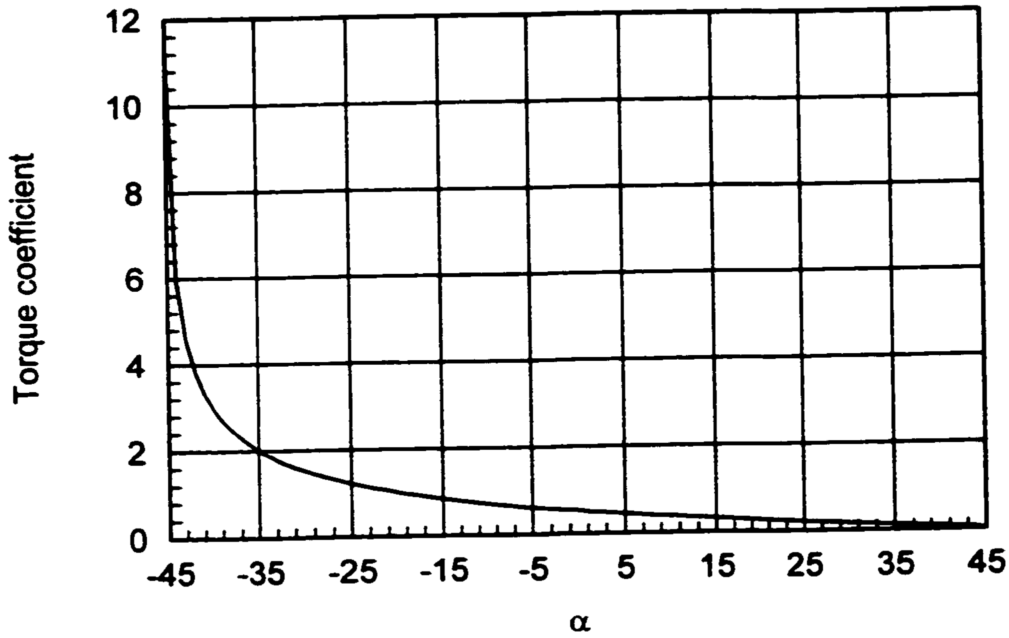


(a) for a 2.6 inch coupler length

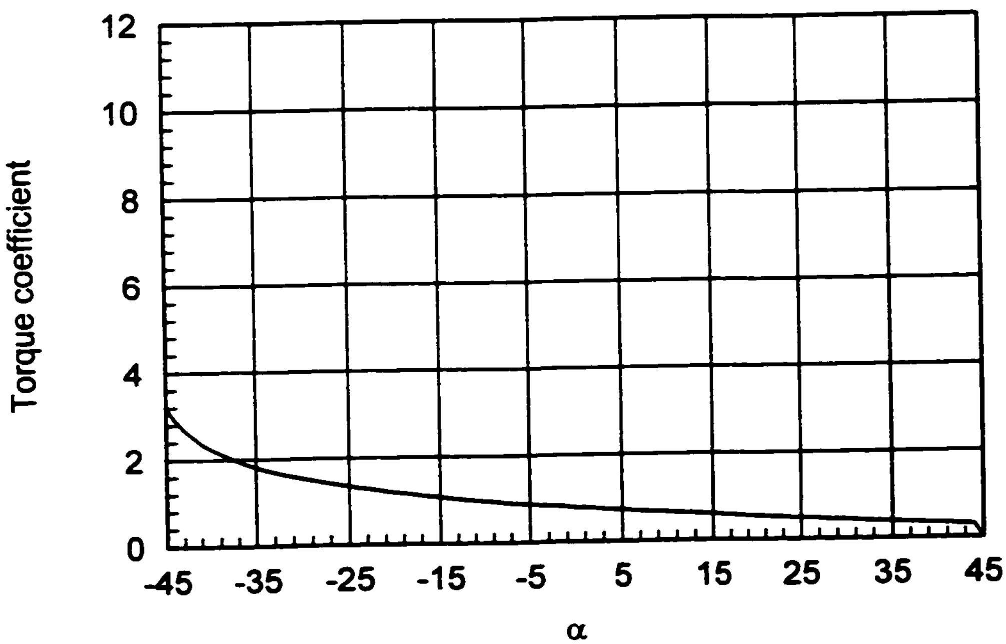


(b) for a 3.7 inch coupler length

Figure 3.18 β Versus α for the Damper Driven from an α Axis.

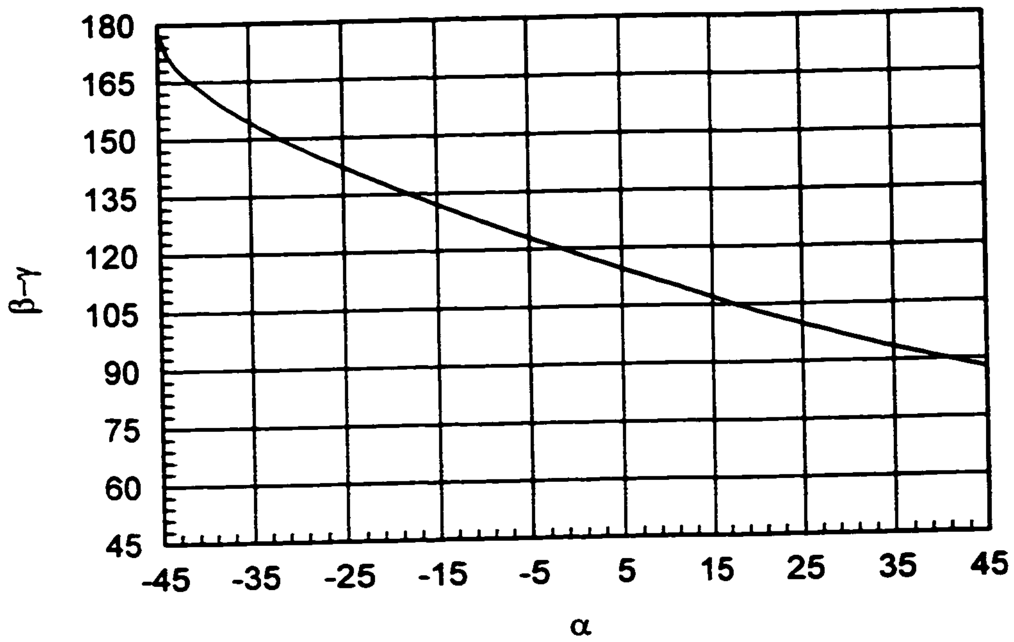


(a) for a 2.6 inch coupler length

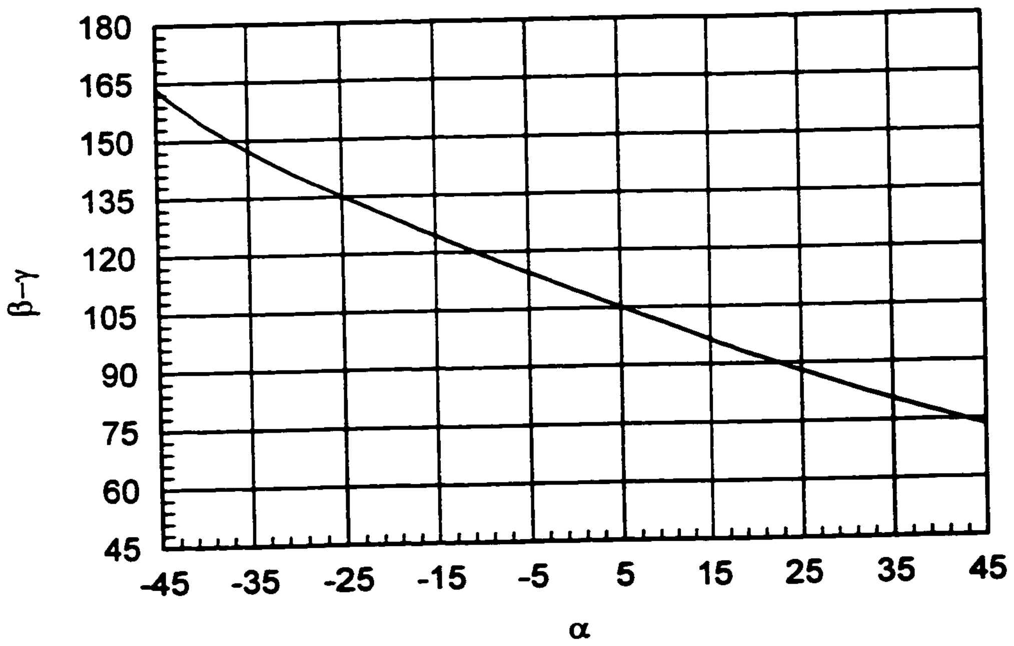


(b) for a 3.7 inch coupler length

Figure 3.19 Torque Coefficient ($\Delta\beta/\Delta\alpha$) Versus α for the Damper Driven from an α Axis.

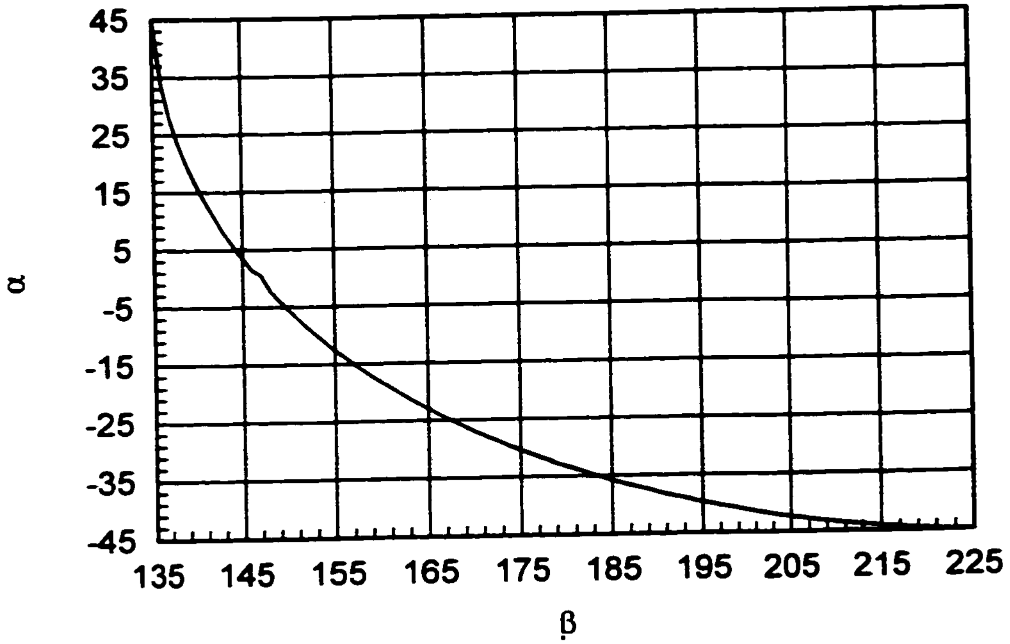


(a) for a 2.6 inch coupler length

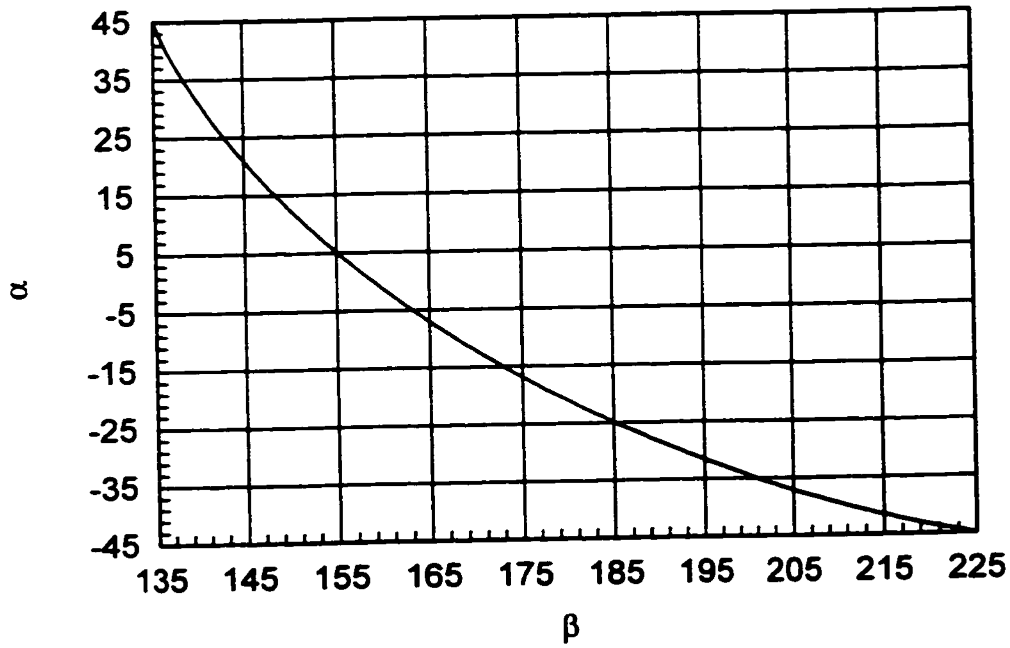


(b) for a 3.7 inch coupler length

Figure 3.20 (β - γ) Versus α for the Damper Driven from an α Axis.

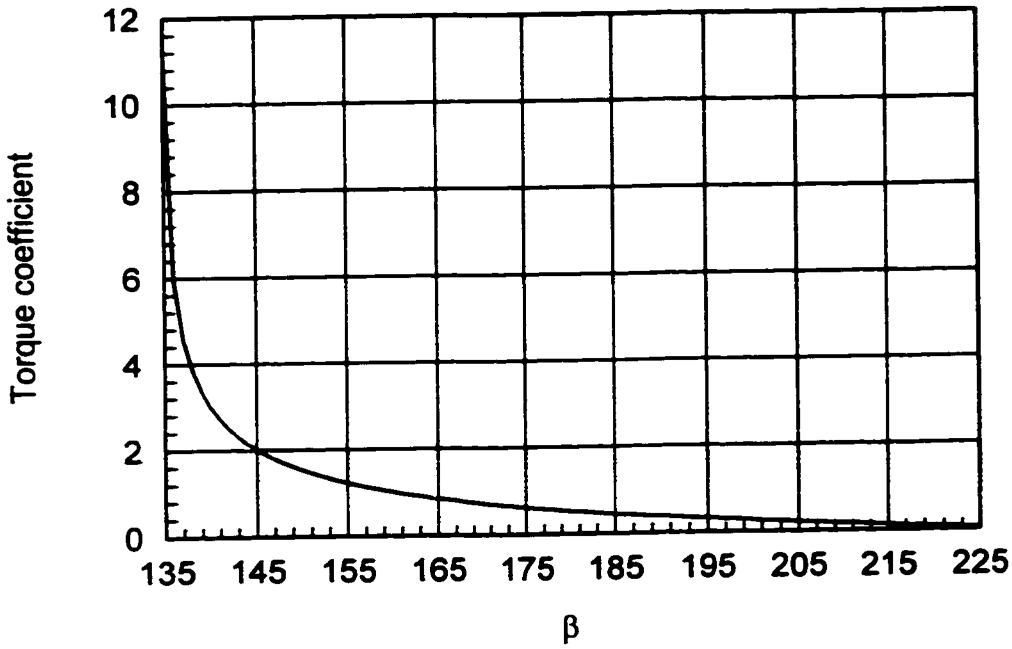


(a) for a 2.6 inch coupler length

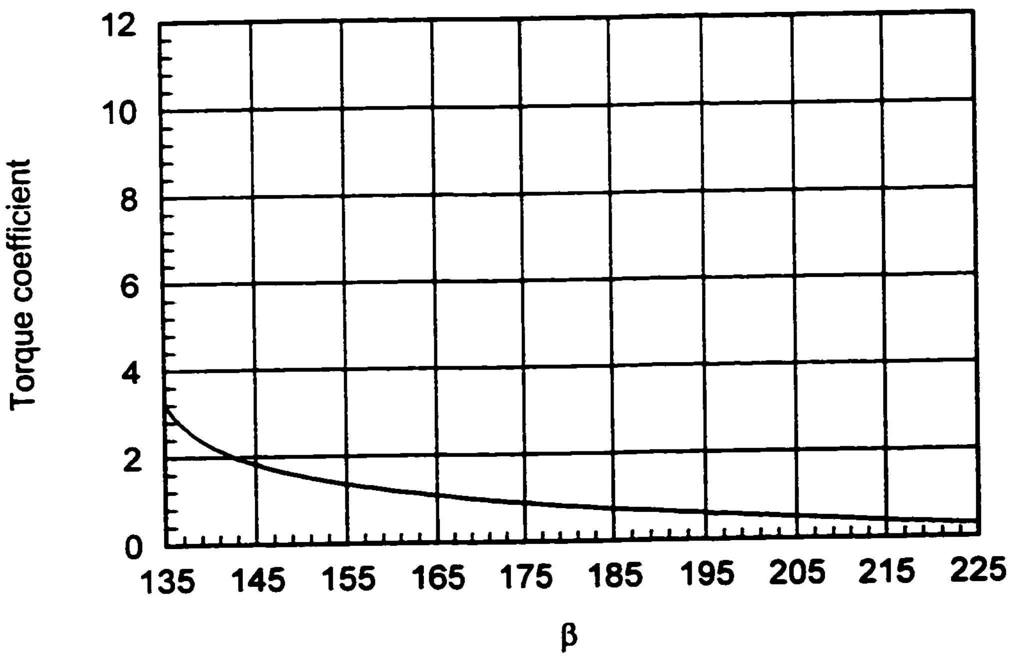


(b) for a 3.7 inch coupler length

Figure 3.21 α Versus β for the Damper Driven from a β Axis.

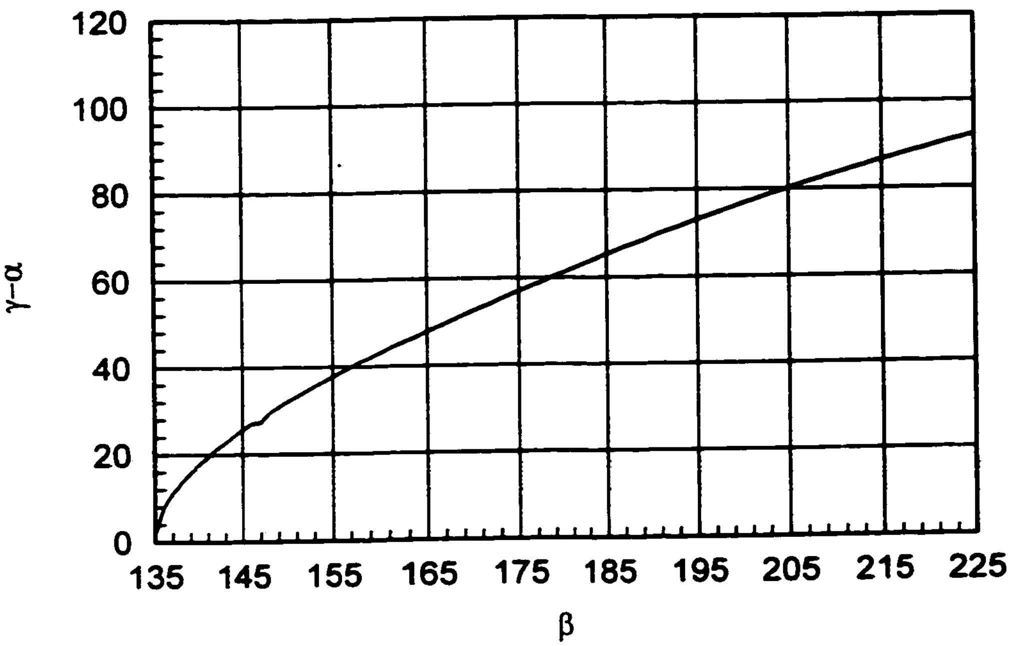


(a) for a 2.6 inch coupler length

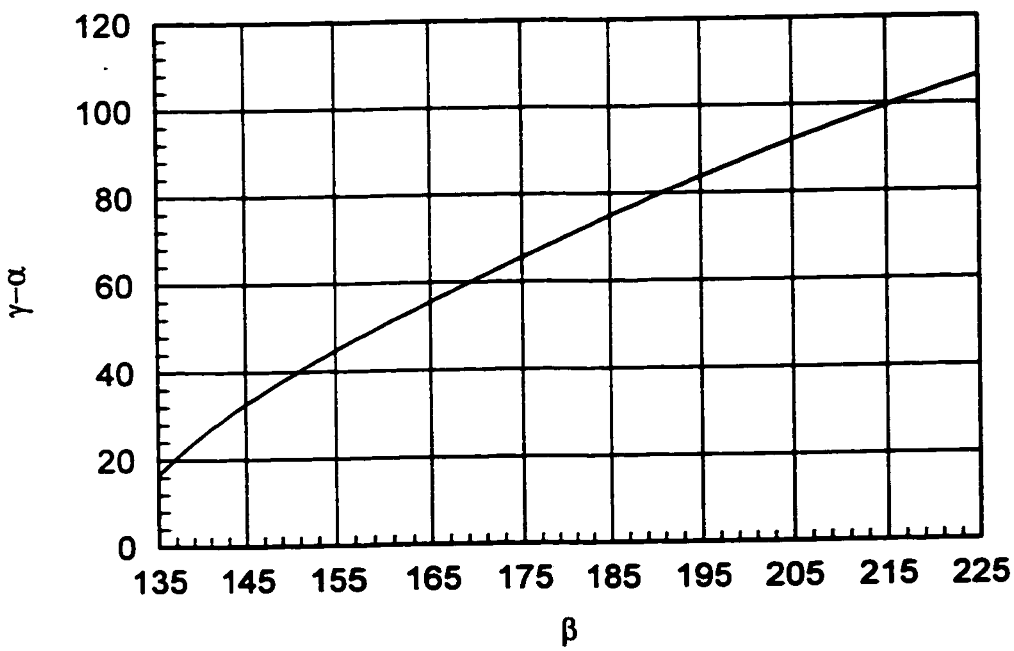


(b) for a 3.7 inch coupler length

Figure 3.22 Torque Coefficient ($\Delta\alpha/\Delta\beta$) Versus β for the damper Driven from a β Axis.



(a) for a 2.6 inch coupler length



(b) for a 3.7 inch coupler length

Figure 3.23 (γ-α) Versus β for the Damper Driven from a β Axis.

3.5 Linearization of Damper Throat Area versus Control Signal

Based on the result of kinematic and dynamic analyses in the previous Sections, it may be concluded that the damper of a specific size may be linearized by changing the coupler length or drive axis (motor location). Therefore, the damper throat area should be linearized by using a software method. The given control signal to the damper motor may be corrected based on a linear damper throat area. Two subroutines were written in QBasic code, one for the normally closed dampers (ODA and DIS), and one for the normally open damper (REC). The subroutines are designed to convert the ideal signal to a compensated signal in the control program. The ideal signal (SG_i) is the specified signal in the control program. In the current studies, it was varied from 0% to 95% by increment of 5% and from 95% to 0% by decrement of 5%. In an actual feedback control system it would be the control signal determined by the feedback control algorithm. The compensated control signal (SG_m) is the signal which is sent to the motor. The compensated signal yields a throat distance linearly proportional to the ideal control signal. The throat distance characteristics of the dampers as supplied by the manufacturer and the modified dampers were linearized based on the measured data of the dampers throat distance in the laboratory.

For a normally closed damper, the throat distance should be a linear function of the ideal control signal, i.e.,

$$TA = \frac{TA_{MAX}}{100} \times SG_i \tag{3.27}$$

For a normally open damper

$$TA = TA_{MAX} - \left(\frac{TA_{MAX}}{100} \times SG_i\right) \quad (3.28)$$

Damper throat distance as a function of the actual control signal supplied to the motor may be modeled by polynomial equations, obtained by curve fitting techniques, i.e.,

$$TA = a_0 + a_1 \times SG_m + a_2 \times SG_m^2 + a_3 \times SG_m^3 + \ldots\ldots + a_n \times SG_m^n \quad (3.29)$$

If the control signal is not compensated the control signal sent to the motor equal the ideal control signal. To compensate the control signal Equation 3.29 must be solve simultaneously with Equation 3.27 or 3.28 for the specified ideal control signal, i.e.,

for a normally closed damper

$$\frac{TA_{MAX}}{100} \times SG_i = a_0 + a_1 \times SG_m + a_2 \times SG_m^2 + \ldots\ldots + a_n \times SG_m^n \quad (3.30)$$

for a normally open damper

$$TA_{MAX} - \left(\frac{TA_{MAX}}{100} \times SG_i\right) = a_0 + a_1 \times SG_m + a_2 \times SG_m^2 + \ldots\ldots + a_n \times SG_m^n$$

(3.31)

Equations 3.30 and 3.31 may be solved by applying Newton's Method,[16], Appendix II. Two subroutines were written in QBasic code, Appendix III. LINODA for a normally closed damper and LINRSC for a normally open damper. The value of the compensated control signal for each ideal signal was calculated in these subroutines and sent to the motors.

Another method of determining the compensated signal is to obtain an array of the ideal signal versus the compensated signal by solving Equations 3.30 and 3.31 for specified values of the compensated signal at fixed intervals, i.e., 0%, 5%, 10%,..........95%. This array is then curve fitted by a best fit polynomial of the format

$$SG_m = b_0 + b_1 \times SG_i + b_2 \times SG_i^2 + \ldots + b_n \times SG_i^n \qquad (3.32)$$

The compensated signal is controlled in the same range as the ideal signal:

$SG_m = 0$ if $SG_i = 0$

$SG_m = 100$ if $SG_i = 100$

3.6 Summary

1. The results of the aerodynamic analysis can be summarized as follows:

The flow coefficient is directly proportional to the damper throat area. The theoretical flow coefficient calculated from Equation 3.4 differs from the experimental flow coefficient. The conclusion based on comparing the theoretical flow coefficient, Figures 3.8 and 3.9, with experimental flow coefficient, Figure 3.10, is that the correlation coefficient f should be between 0.75 and 1.0.

The excess power is dependent on the pressure drop across the recirculation damper $(P_5 - P_1)$. Fan energy will be saved if the pressure drop across this damper is reduced.

2. The kinematic and dynamic analysis indicated that it is possible to linearize an existing damper by changing the coupler length and motor location. However,

several problems may be encountered. The damper may not open completely, it may not close completely, or it may bind, i.e., require excessive driving torque. Binding will occur if the coupler aligns with the driven crank arm.

Whereas the flow coefficient is related to the throat distance, damper characteristics may be linearized by software if the throat distance is a known function of the control signal. Throat distances as a function of control signal may be determined analytically or experimentally.

CHAPTER 4

EXPERIMENTAL WORK

The primary objective of the experimental work was to study the effects of control strategy on system characteristics. Some of the control strategies involved linearized damper characteristics. Linearization of damper characteristics required that throat distances are a linear function of the control signal. Therefore the flow coefficients are a linear function of the control signal. A preliminary objective was to develop an experimental method for measuring throat distance.

4.1 Damper Throat Distance Measurement

The ODA damper was modified to enable manual measurement of the throat distance. A damper blade having the same profile shape and the same dimensions as the damper blades used for this study was cut to three pieces. The two end pieces were 2.5 inch in length. The hollows in the two end pieces were filled with plaster and painted with black color. These two parts were connected to two successive blades outside the damper frame as shown in Figure 4.1.

A computer data logger system converted a digital input signal to an analog output signal. The motor position was determined by a direct current voltage ranging from 2 to 10 VDC.

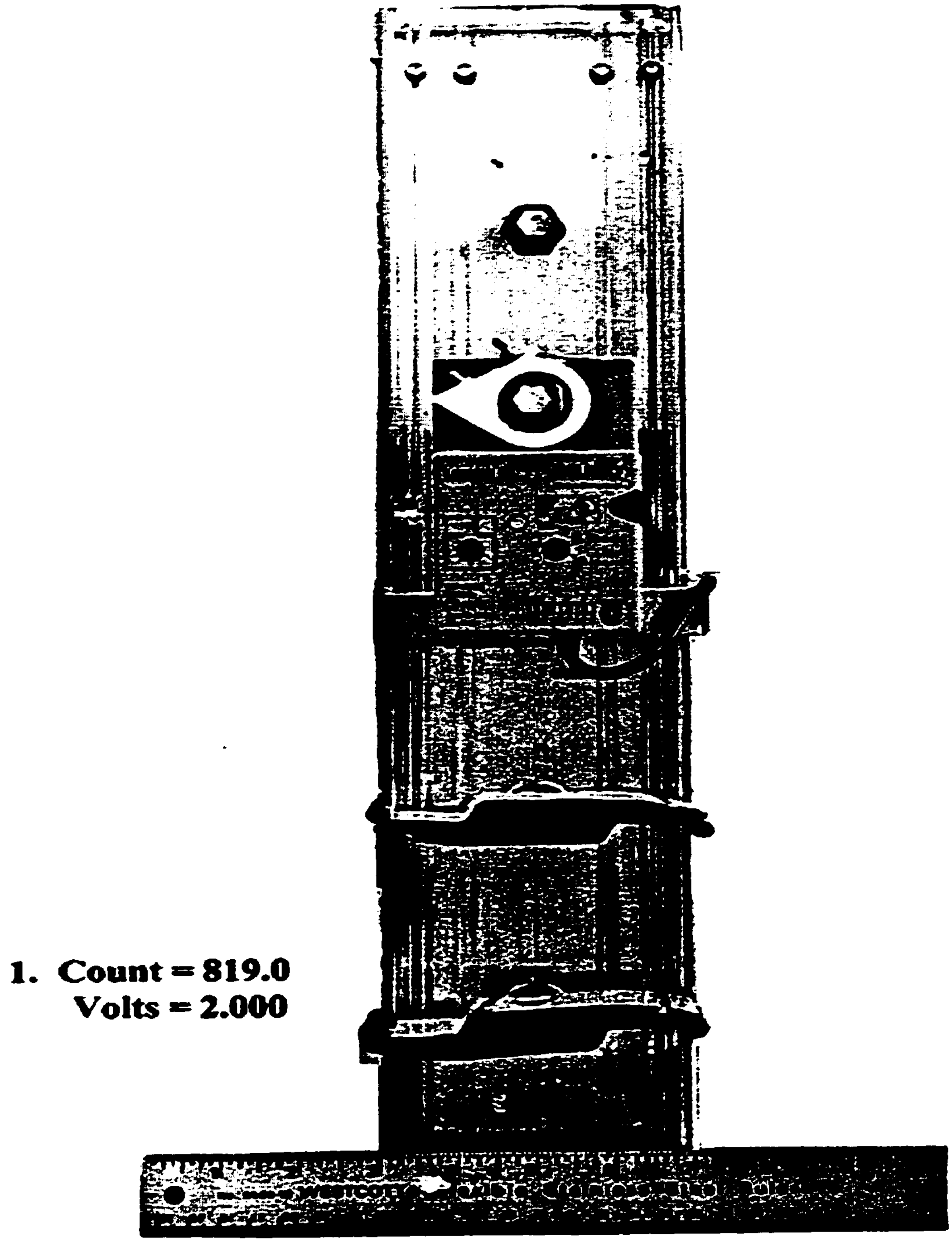


Figure 4.1 The Photograph Used to Measure the Throat Area.

A 100% signal corresponded to 4095 counts (2^{12} = 4096, the data logger used had a 12 - bit chip). which in turn corresponded to 10 VDC. A 0% signal corresponded to 819 counts which in turn corresponded to 2 VDC. The motor was controlled according to the given signal to position the damper blades from fully closed (0% signal) to fully open (100% signal) in signal steps of 5%, and then from fully open (100% signal) to fully closed (0% signal) in signal steps of 5%. The increasing and decreasing in the signal procedure was used in measurements because of the motor and linkage hysteresis.

The throat distance between adjacent blades can be measured from the outside blade sections in two ways, by measuring distance with a slide Vernier calipers directly from the damper, or by taking photographs and then measuring the distance from the photograph. Both ways were used and compared for accuracy result.

The control signal was corrected for 3% hysteresis by software program as per Krakow [8]. The throat distance characteristic of the damper as supplied by the manufacturer and as modified are shown in Figures 4.2 and 4.4, respectively. The curve shows that some hysteresis is still present. Coefficients representing best fit polynomials of the experimental data are shown in Table 4.1. Figures 4.3 and 4.5 represent the required linear characteristics. Figure 4.3 shows the required linear relationship between the ideal control signal and the normally closed dampers supplied by the manufacturer throat distance and Figure 4.5 shows the required linear relationship between the ideal control signal and the normally closed modified damper throat distance. For example, for the

Table 4.1 The Polynomial Equation Coefficients for the Damper Throat Distance.

coefficients	the damper as supplied by manufacturer	the modified damper
a_0	-0.0566	- 0.1401
a_1	0.0164	0.01144
a_2	-3.94×10^{-4}	1.95×10^{-4}
a_3	5.22×10^{-6}	0.0
a_4	-1.91×10^{-9}	0.0

normally closed damper as supplied by the manufacturer if the ideal signal is 60% the throat distance is 1.5 inches, (Figure 4.3). To get 1.5 inches the compensated signal should be 83%, (Figure 4.2). For normally closed modified damper if the ideal signal is 60% the throat distance is 1.58 inches, (Figure 4.5). To get 1.58 inches throat distance the compensated signal should be 64.5%, (Figure 4.4).

This experimental method of determining the throat distance as a function of the control signal provides data required for damper characteristic linearization using the methods detailed in the previous chapter.

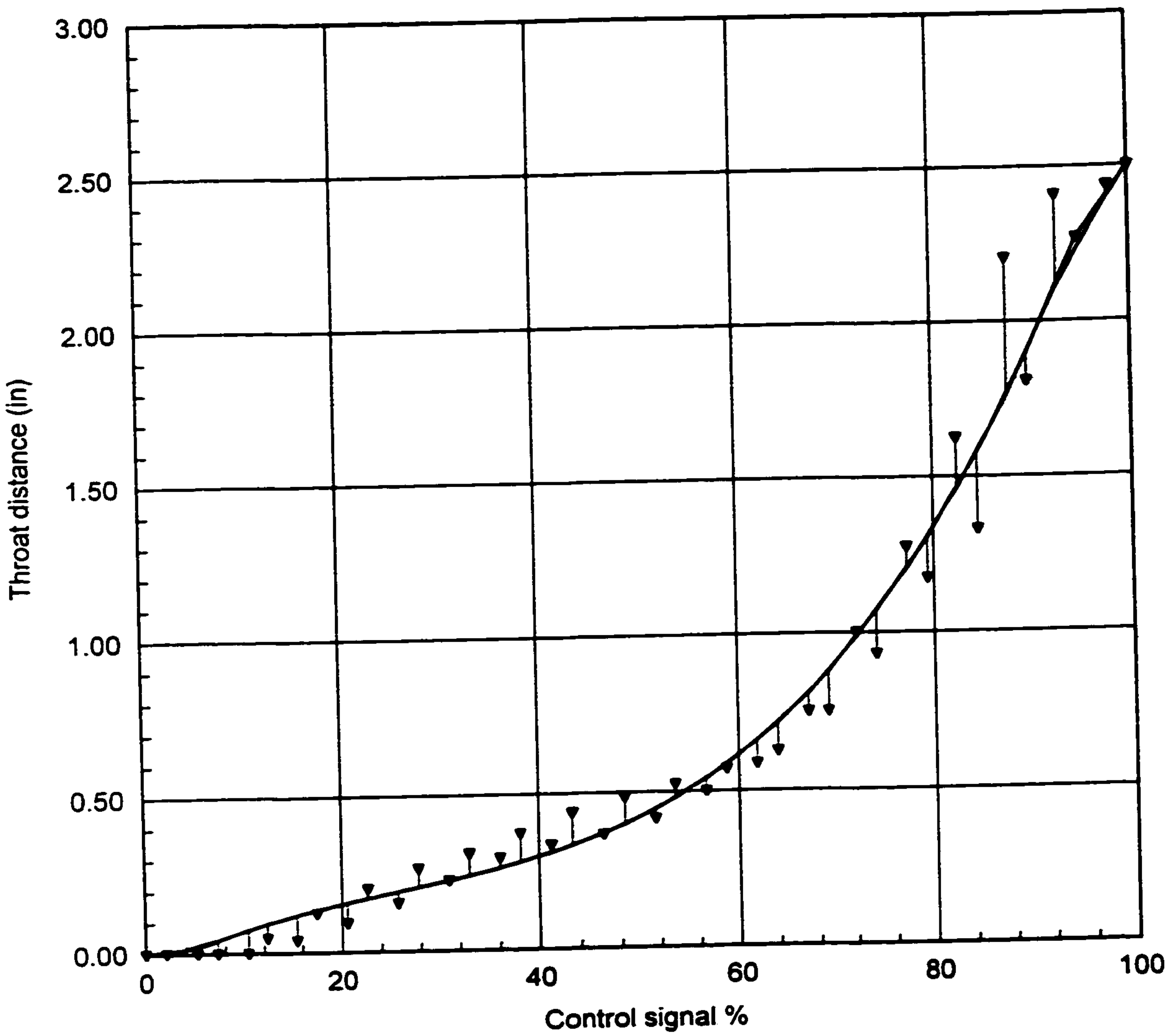


Figure 4.2 Best Fit Polynomial Curve of the Throat Distance of the Normally Closed Damper as Supplied by the Manufacturer.

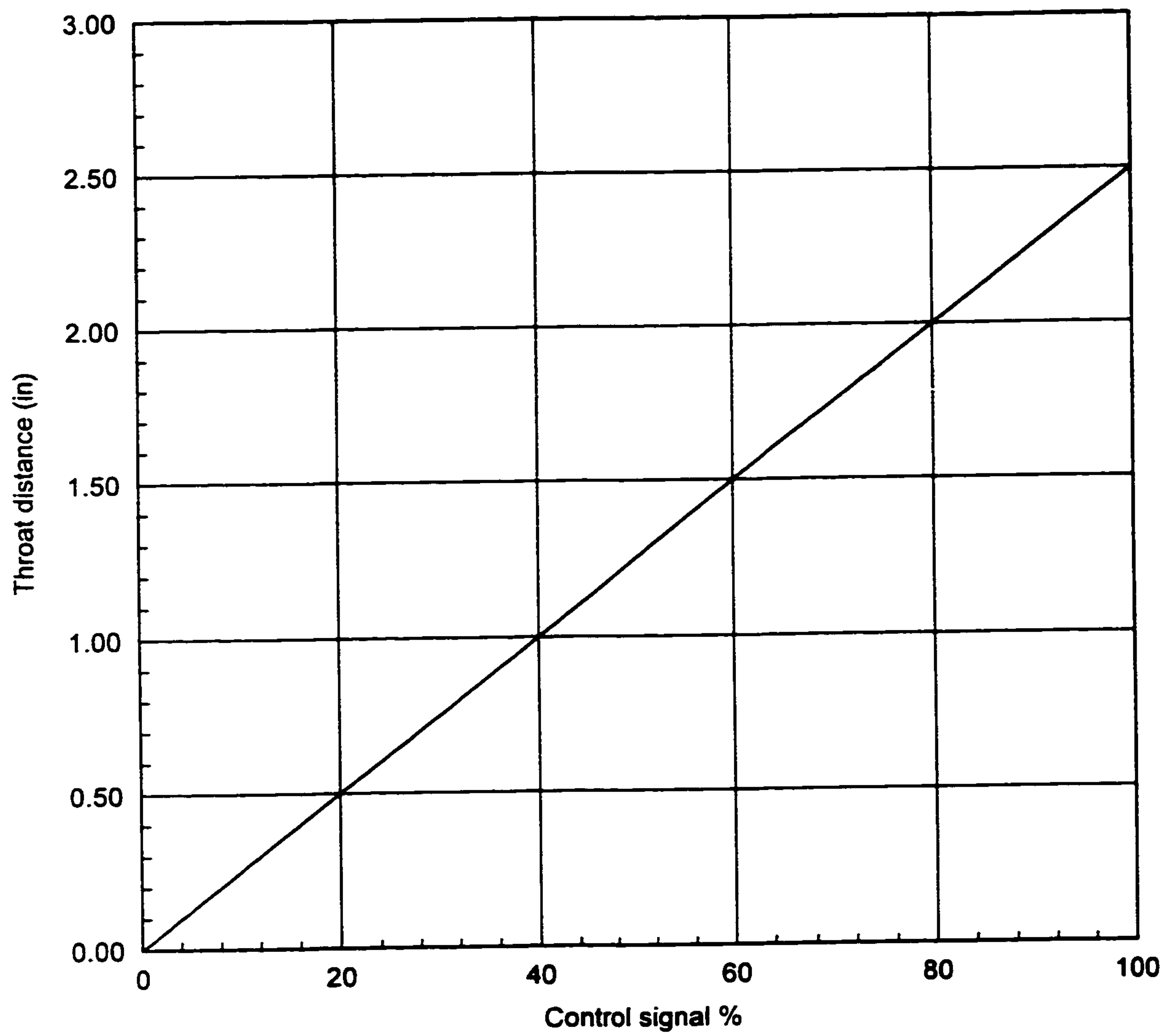


Figure 4.3 Linear Relationship Between the Throat Distance and the Ideal Control Signal for the Normally Closed Damper as Supplied by the Manufacturer.

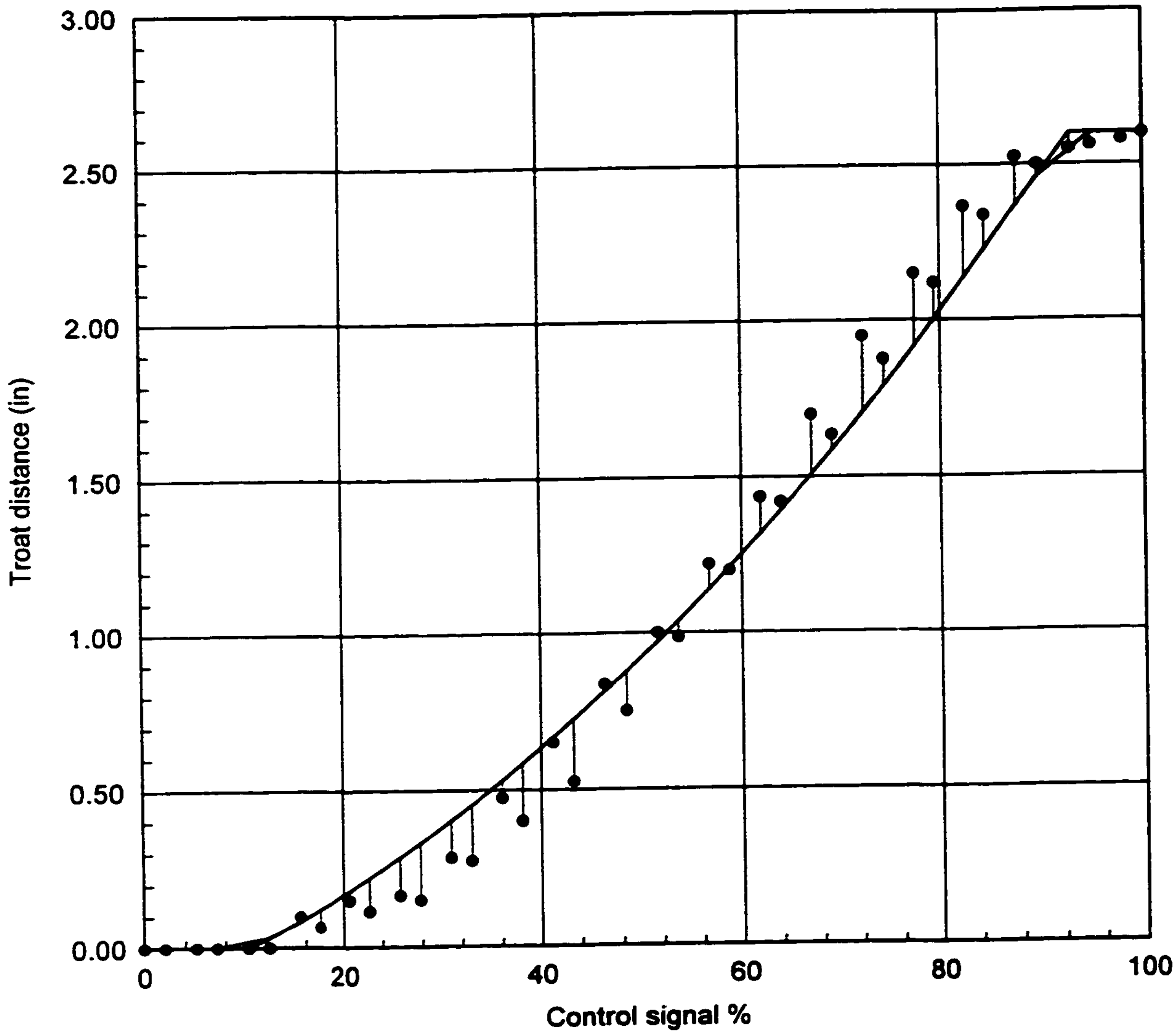


Figure 4.4 Best Fit Polynomial Curve of the Throat Distance of the Normally Closed Modified Damper.

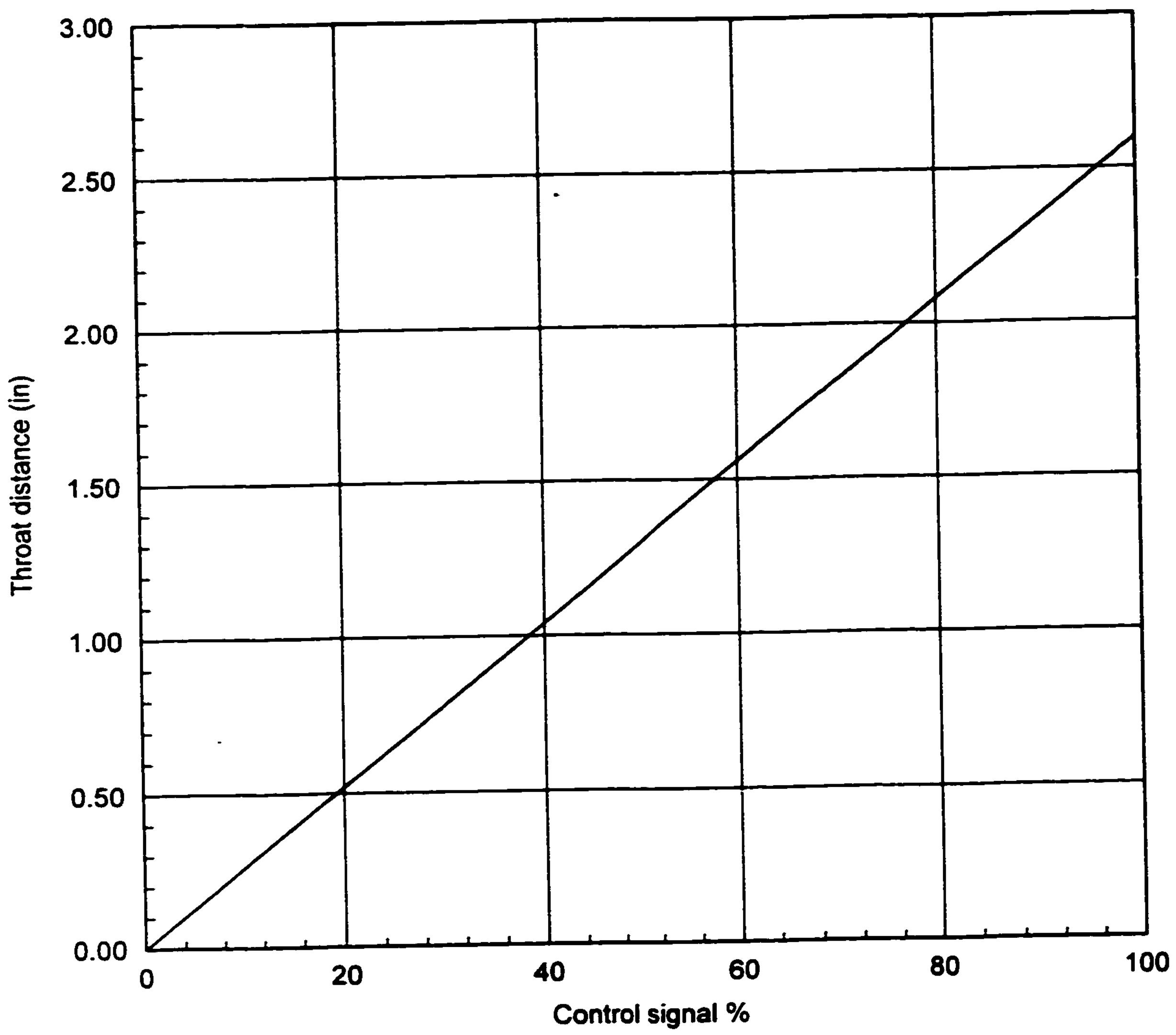


Figure 4.5 Linear Relationship Between the throat Distance and the Ideal Control Signal for the Normally Closed Modified Damper.

4.2 System Description

The HVAC system used for this experimental work was located in one of Mechanical Engineering laboratories at Concordia University's Loyola Campus. This laboratory consists of two rooms. A small room contained the HVAC system equipment. A large room was the conditioned space. The supply duct damper was installed at the end of the supply duct inside the conditioned room. The return air duct intake was connected to the return fan through the wall between the two rooms. The conditioned room housed a computer hardware and data logger. Figure 3.1 shows the schematic diagram of the HVAC system. The basic components of this HVAC system are:

- economizer (three dampers),
- supply and return fans,
- air distribution ducts,
- supply air damper,
- return air duct resistance, and
- control system.

4.2.1 Economizer

The economizer consists of three dampers, an outdoor air damper (ODA damper), a recirculation air damper (REC damper), and a discharge air damper (DIS damper). The three dampers had the same size (18 inch by 18 inch). Through the ODA damper, the space was provided with the required quantity of fresh air. The REC damper was controlled to allow some quantity of the drawn air from the space to be returned to it again. The rest of discharged air was exhausted through the DIS damper to the outside.

4.2.2 Supply and Return Fans

Two centrifugal fans were used in this system, a supply fan and a return fan. Both fans were single inlet fans with backward inclined blades and 13 inch wheels. The motors were variable speed (800 - 2200 rpm) rated at 1½ hp. The supply fan is used to provided the space with the desired amount of conditioned air. The return fan is used to draw the air from the conditioned space through the return air duct, where the drawn air can be recirculated or discharged. Supply and return fan speeds were controlled to maintain pressures in the supply duct and conditioned space at specified set point values.

4.2.3 Air Distribution Ducts

The outdoor air enters the HVAC system through the supply air duct and leaves it through the return air duct. The supply air duct consists of two parts. The first part, which connects the economizer to the inlet supply fan was 18 inch by 18 inch, and the second part between the supply fan outlet and the supply damper was 16 inch by 12 inch. The recirculation air duct, where the recirculation damper was installed was 18 x 18 inch.

4.2.4 Supply Air Damper

The supply air damper is connected to the end of the supply air duct in side the space. By controlling this damper the quantity of supplied air can be increases or decreases depending on the required supply air flow rate for the space. The damper size was 14 x 14 inch.

4.2.5 Return Air Duct Resistance

A plastic screen was used as a resistance. This resistance was installed in the intake section of the return air duct inside the conditioned space. The function of this resistance is to simulate a larger system by producing a greater pressure drop in that section.

4.2.6 Control System

The control system consisted of transducers, motors, and a controller.

Transducers: Pressure and velocity transducers were installed in the system. The pressure transducers were installed to measure the static pressure in five places of the HVAC system. The pressure transducers were strain guage types. The locations were in the conditioned space, and in the inlet and outlet air ducts of each fan. The air velocity of the outdoor air flow rate and the return air flow rate were monitored. The air velocity transducers were installed in the intakes of the supply fan before the ODA damper and return air ducts after the plastic screen. The velocity transducers were hot wire types.

Motors: The dampers were controlled by positional motors (actuators), which had the range of control from 0 to 90 degrees. The damper motors received the signal from the controller in order to control the dampers. Each fan was connected to a variable speed rotational motor, which was connected to adjustable frequency drives in order to control the fan speed.

Controller: A computer was connected to the system transducers and a data logger. The data logger converts the input analogue signals to digital data and the digital output data to analogue output signals. The control sequences are entered to the computer by the software controller. The software controller is a computer program designed to use input data for controlling. The program was modified to meet the objective of each experiment.

4.3 Control Strategies

The computer program was designed to use the input data and to calculate the other required data to control the HVAC system. The data logger converts the digital signals to analogue signals (from 2 to 10 volt), and supplies the analog signals to the dampers motors. The input signals from the transducers are converted by the data logger from analog to digital signals before supplied to the computer. The computer program calculates the actual results and prints them in the output files.

At the beginning of each test, all the economizer dampers, supply air damper and the fans were set at their original positions by the first signal supplied to the dampers motors and to the fans motors. The ODA and DIS dampers were normally closed, the REC damper and the supply air damper were normally open.

Proportional and integral (PI) controller subroutines were used by the software program to calculate the proper signal to control both supply and return fans. The supply fan speed was controlled by the static pressure in the supply duct. Two supply duct pressures were used 0.5 inWC and 0.6 inWC. The return fan speed was controlled based on the static pressure in the conditioned space, which was equal to the atmospheric pressure (0 inWC). The space was sealed completely so that the supply and return flow rates were equal.

The air velocity for the outdoor air (fresh air) and the return air were measured by the air velocity meters for each operating point. The computer program calculated the outdoor air flow rate and the return air flow rate.

The five pressure transducers measured the static pressure at the five key points as shown in Figure 3.11. Two transducers were used to measure the pressure in the supply duct: 1- the supply fan inlet pressure and the REC damper outlet pressure, and 2- in the outlet section of the supply fan. This outlet pressure is the pressure controlled by the supply fan. The other two transducers measured the pressure in the return air duct: 1- in the inlet return fan pressure, and 2- in the outlet section of the return fan. This pressure is the return fan outlet pressure and the REC damper inlet pressure. The fifth transducer was installed in the conditioned space to measure the space pressure. This pressure was controlled by the return fan.

Two types of control strategy were followed when performing each test: three coupled dampers and two coupled dampers.

4.3.1 Three Coupled Dampers

When a test was performed with the three coupled dampers control strategy all three dampers had equal control signals. The ODA and DIS dampers were controlled from fully closed to fully open by increment of 5% control signal. The REC damper was controlled from fully open to fully closed by reduction of 5% control signal. When the economizer dampers reached the last signal, (fully open ODA damper) they were controlled by the same strategy to return back to their original positions.

4.3.2 Two Coupled Dampers

The two coupled dampers control strategy differed from the control strategy for the three coupled dampers by the fact that the DIS damper was set at 100% signal, fully open, and remained fully open during entire time of the experiment.

4.4 Experimental System Characteristics

The effect of the linear throat area characteristic of the economizer dampers in the HVAC system can be clearly seen from the results of the following experimental tests. These tests were performed from May to August 1998. The system was tested as follows:

1- the dampers as supplied by manufacturer,

2- the dampers as supplied by manufacturer linearized by the software method,

3- the modified dampers, and

4- the modified dampers linearized by the software method.

Series of tests were performed for each one of the above damper classifications. The outdoor air flow rate, the recirculation damper inlet and outlet pressures and the fans signals were determined for a 0.5 inWC supply air duct static pressure set point. For confirmation, the same tests were repeated with a 0.6 inWC supply air duct static pressure. The supply air damper was positioned at 18% signal and remained at the same position during the test. The two types of control strategies mentioned in Section 4.2 were followed in performing the tests. Figures 4.6

through 4.29 show the results of the outdoor air flow rate, recirculation damper inlet and outlet pressures, and fans signals. Tables 4.2 and 4.3 list the Figures relating to the dampers as supplied by manufacturer and modified dampers, with and without software linearization respectively.

Table 4.2 Figures Numbers of Figures Showing Experimental Results of the Dampers as Supplied by Manufacturer.

Dampers as supplied by the manufacturer with and without software linearization				
control strategy	supply duct pressures	outdoor air flow rate	REC damper Pressures	fans signals
three coupled dampers	0.5 inWC	Figure 4.6	Figure 4.14	Figure 4.22
	0.6 inWC	Figure 4.7	Figure 4.15	Figure 4.23
two coupled dampers	0.5 inWC	Figure 4.8	Figure 4.16	Figure 4.24
	0.6 inWC	Figure 4.9	Figure 4.17	Figure 4.25

Table 4.3 Figures Numbers of Figures Showing Experimental Results of the Modified Dampers.

Modified dampers with and without software linearization				
control strategy	supply duct pressures	outdoor air flow rate	REC damper pressures	fans signals
three coupled dampers	0.5 inWC	Figure 4.10	Figure 4.18	Figure 4.26
	0.6 inWC	Figure 4.11	Figure 4.19	Figure 4.27
two coupled dampers	0.5 inWC	Figure 4.12	Figure 4.20	Figure 4.28
	0.6 inWC	Figure 4.13	Figure 4.21	Figure 4.29

4.4.1 Flow Rates

Figures 4.6 to 4.13 indicate that the ODA flow rates for dampers without software linearization can be controlled from 10% to 95% signal for the damper as supplied by manufacturer and from 10% to 85% signal for the modified damper. The linearization enables better control, from 0% to 95% signal. The outdoor air flow rate was a more linear function of the control signal for the linearized dampers. A more linear characteristic would yield better control. The maximum outdoor flow rate increased by approximately 10% when the supply pressure was increased from 0.5 inWC to 0.6 inWC. The supply flow rate is equal to the maximum outdoor air flow rate at all times since no change in the supply damper position.

4.4.2 Recirculation Damper Pressures

Based on the fans power analysis in Section 3.3.2, the excess power is proportional to the differential pressure across the REC damper. The pressures are shown in Figures 4.14 through 4.21. Figure 4.30 shows the excess power versus the recirculation damper differential pressure. The maximum REC damper pressure differential is 1.5 inWC and occurs at a 55% signal, Figure 4.14(a) which corresponded to 388 W, (Figure 4.30). The two coupled dampers for the dampers as supplied by manufacturer reduce power if compare to three coupled dampers, from 1.5 inWC to 1.37 inWC at a 55% signal, (Figure 4.14(a) and Figure 4.16(a)), without software linearization, and from 0.45 inWC to 0.39 inWC, (Figure 4.14(b) and 4.16(b)), with software linearization. There is no pressure differential reduction if the two and three coupled dampers for the modified dampers are compared, Figures 4.18 through 4.21. The modified dampers are better than the dampers as supplied by manufacturer, because of the less variation of the pressure across the damper. The linearized dampers reduce the pressure differential to average of 0.32 inWC which corresponded to 83 W. Therefore, the maximum reduction in excess power can be achieved with linearized dampers. The percentage of power saving depends on supply fan and return fan differential pressure which depends on the supply duct and return duct differential pressure. The same conclusion can be reached when increasing the supply pressure set point to 0.6 inWC .

The system with linear dampers is the best for control irregardless of whether the three coupled or two coupled dampers strategy is used.

4.4.3 Fans Control Signals

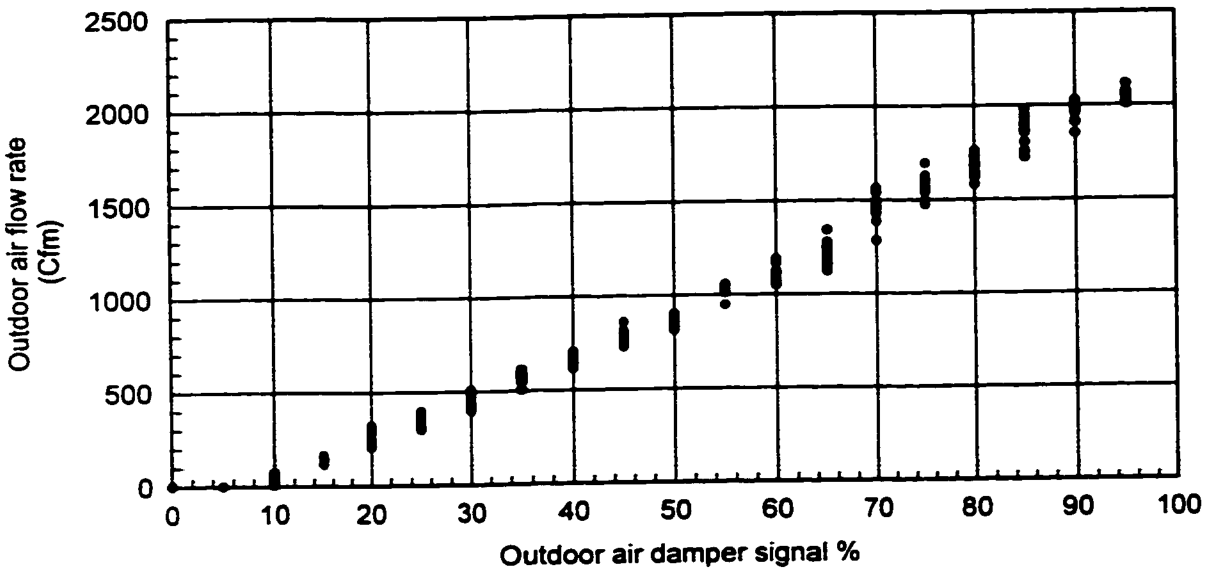
Figures 4.22 through 4.29 illustrated the variation in supply fan and return fan control signals. For dampers as supplied by manufacturer without linearization, Figures 4.22(a) and 4.24(a), three coupled dampers yield lower supply fan signal variation than two coupled dampers. However, two coupled dampers yield lower return fan signal variation than three coupled dampers.

When the supply fan control pressure was set to 0.6 inWC for both three and two coupled dampers without linearization, Figures 4.23(a) and 4.25(a) the supply fan signal reached 100% during operation indicating that the supply fan pressure less than set point pressure. Consequently, the supply flow rate is less than the design flow rate. With linearization, Figure 4.22(b), 4.23(b), 4.24(b), and 4.25(b) all fan control signals show less variation and the supply fan signal always less than 100%. The modified dampers, with and without linearization, Figures 4.26 through 4.29 show fairly constant fan control signals.

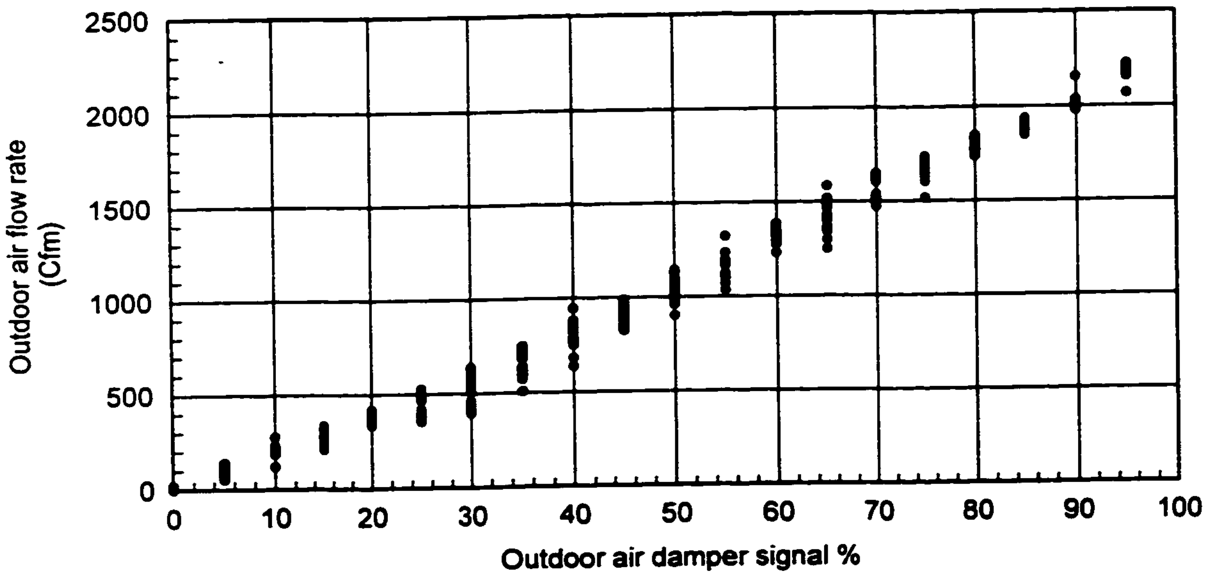
4.5 Summary

These results indicate the following.

1. Linearization will reduce excess power consumption and decrease variation in fan control signals to maintain constant pressure.
2. Two coupled dampers control strategy will require slight less power than three coupled dampers.
3. linearization may be accomplished by linkage geometry design or by software.

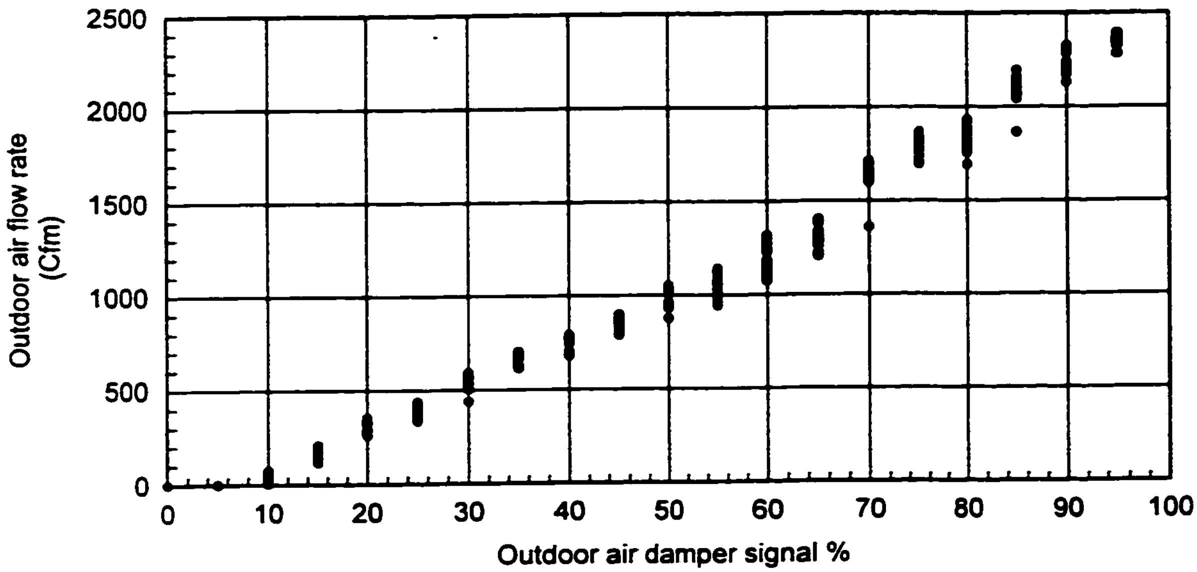


(a) the dampers as supplied by the manufacturer.

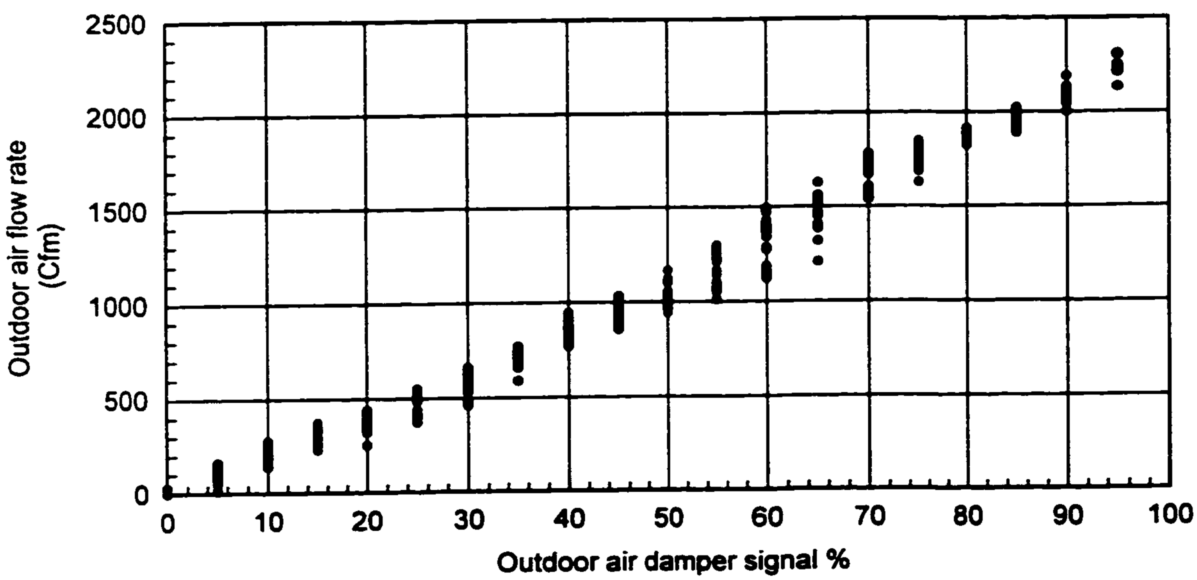


(b) the damper as supplied by the manufacturer with software linearization.

Figure 4.6 Outdoor Air Flow Rate, Three Coupled Dampers and 0.5 inWC Set Point.

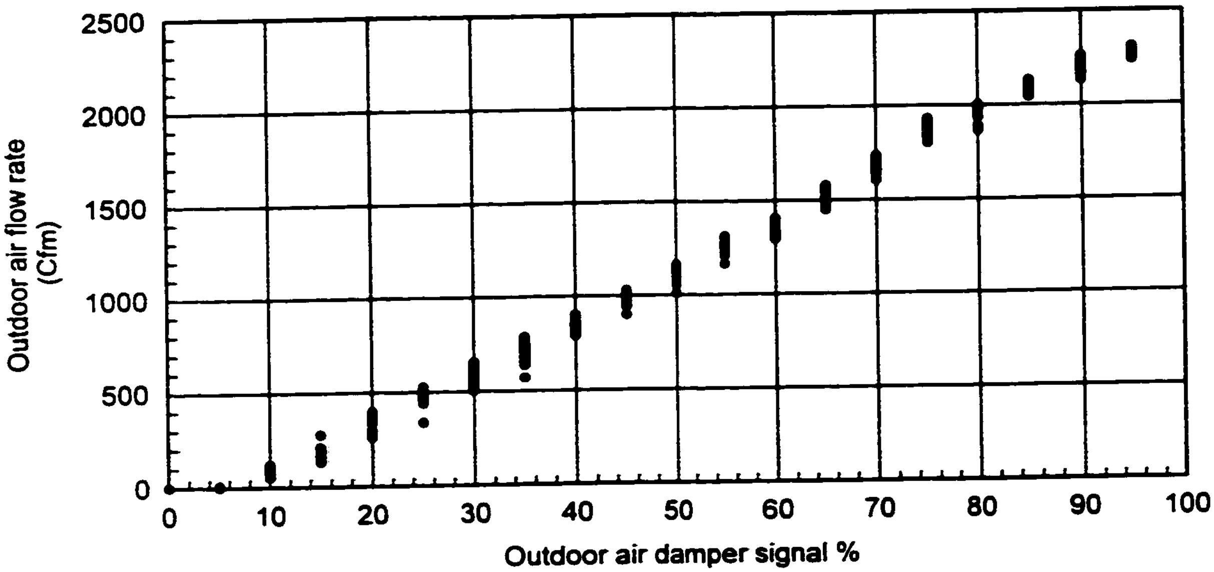


(a) the dampers as supplied by the manufacturer.

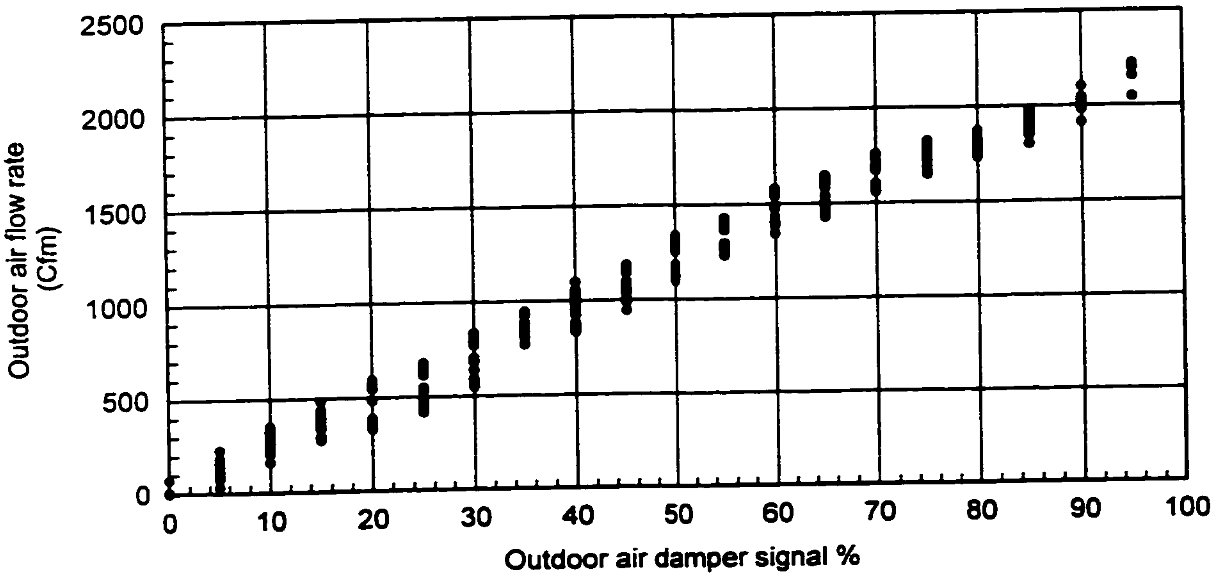


(b) the damper as supplied by the manufacturer with software linearization.

Figure 4.7 Outdoor Air Flow Rate, Three Coupled Dampers and 0.6 inWC Set Point.

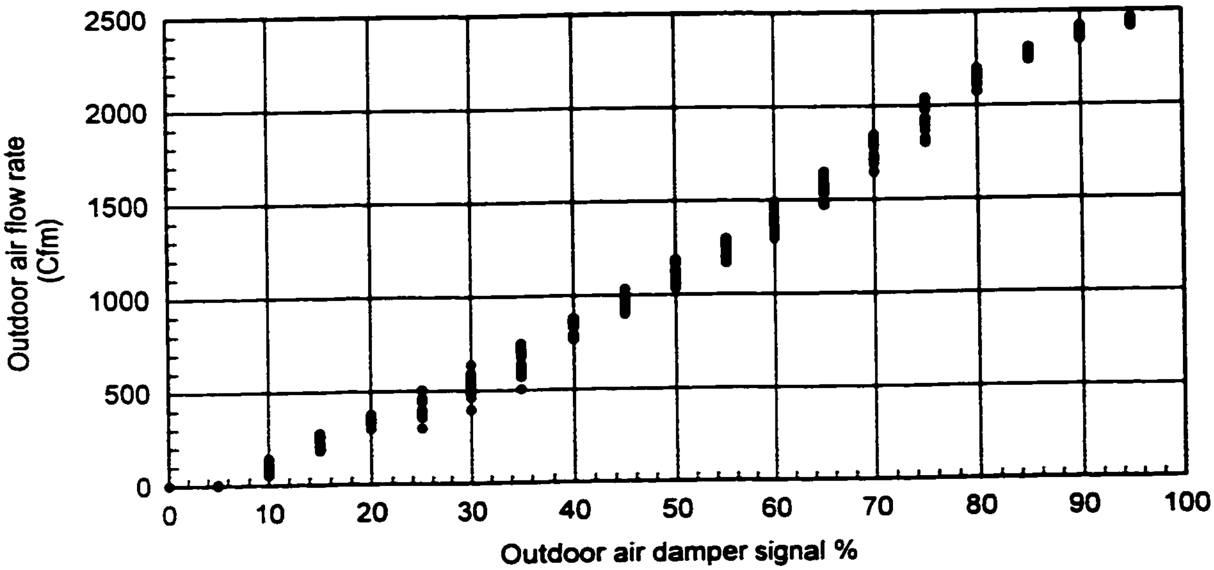


(a) the dampers as supplied by the manufacturer.

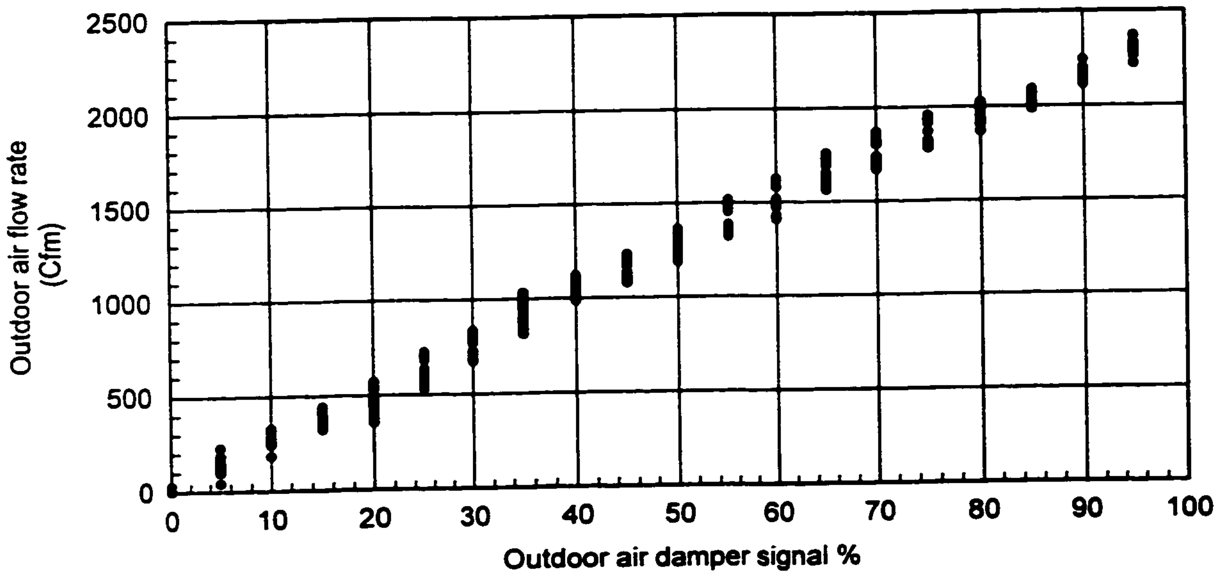


(b) the damper as supplied by the manufacturer with software linearization.

Figure 4.8 Outdoor Air Flow Rate, Two Coupled Dampers and 0.5 inWC Set Point.

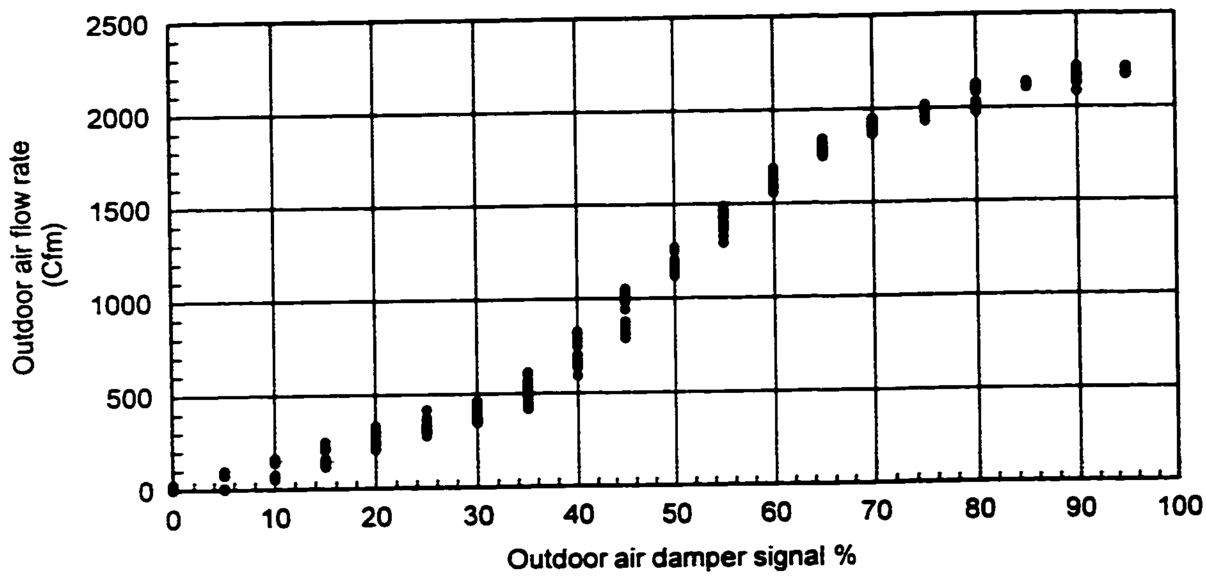


(a) the dampers as supplied by the manufacturer.

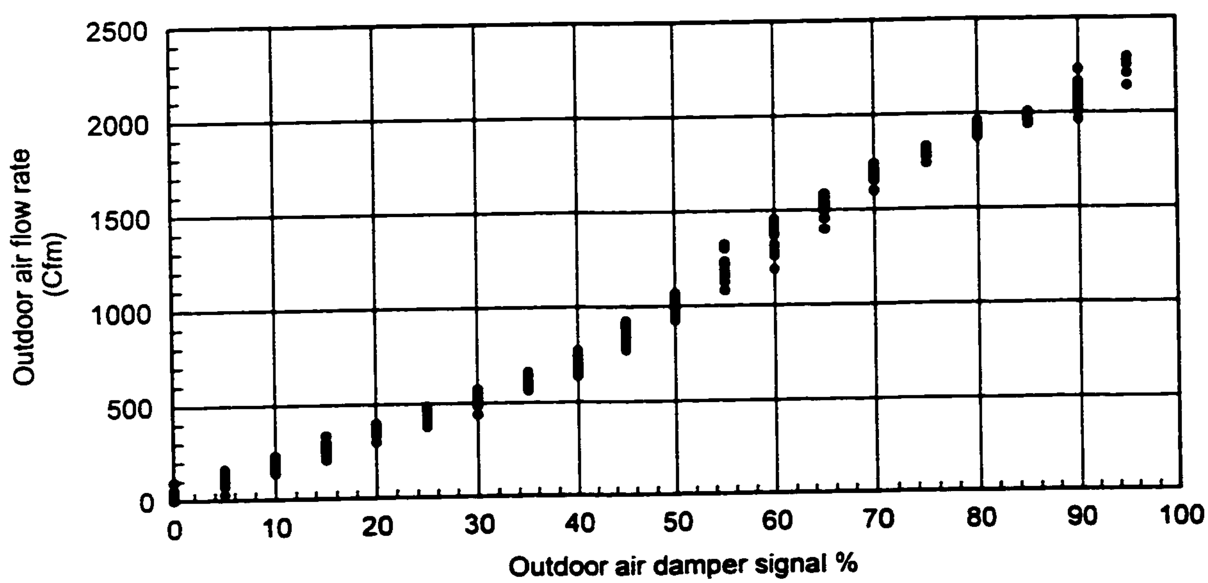


(b)) the damper as supplied by the manufacturer with software linearization.

Figure 4.9 Outdoor Air Flow Rate, Two Coupled Dampers and 0.6 inWC Set Point.

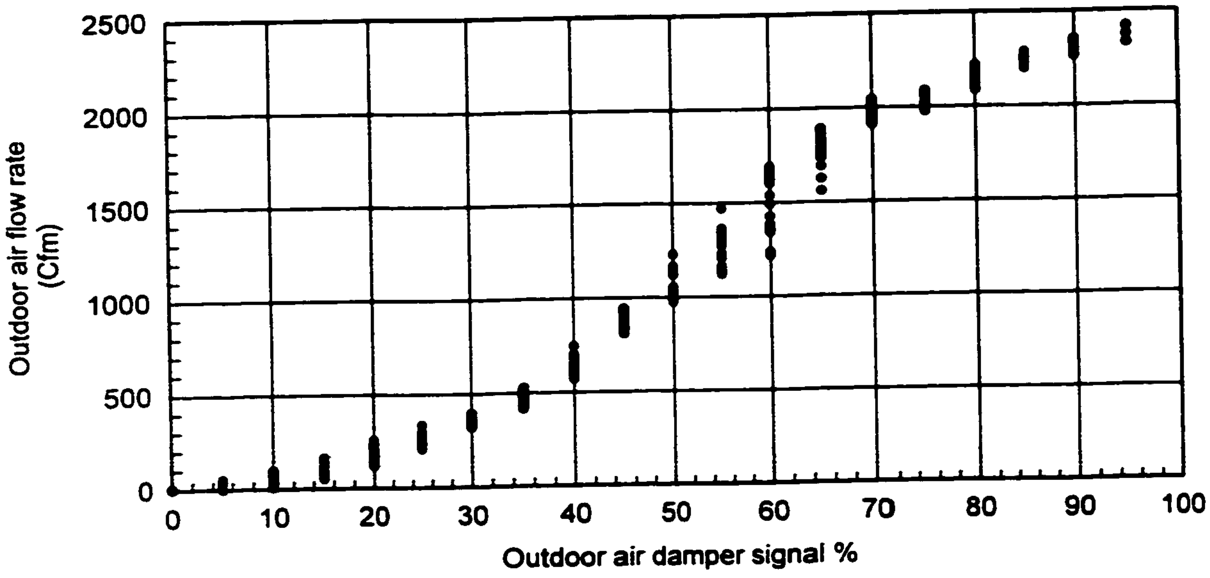


(a) the modified dampers.

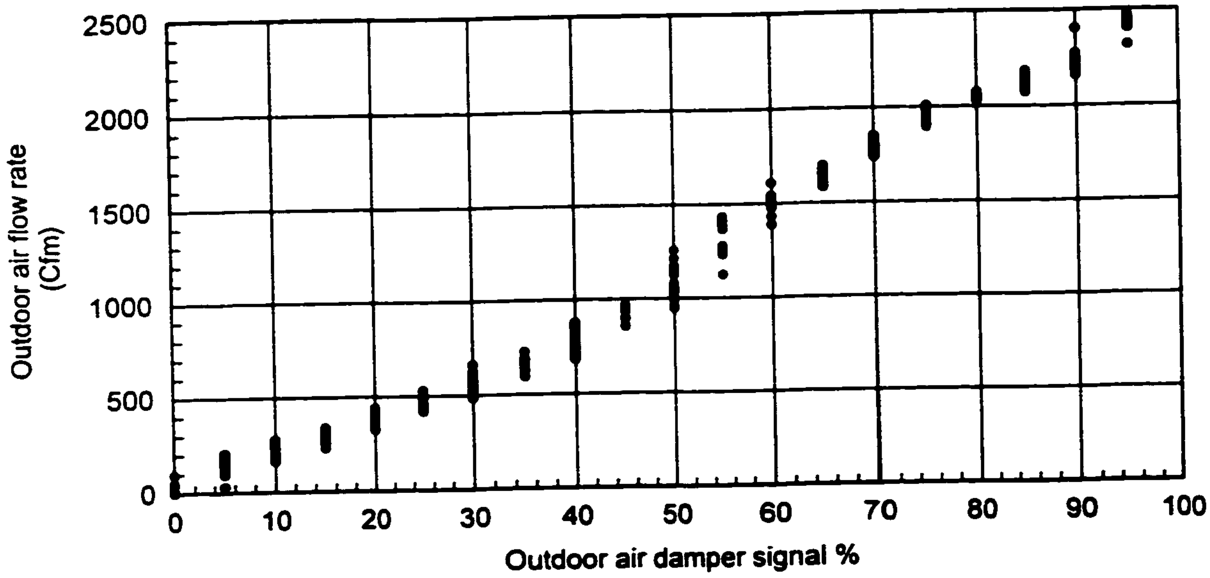


(b) the modified dampers with software linearization.

Figure 4.10 Outdoor Air Flow Rate, Three Coupled Dampers and 0.5 inWC Set Point.

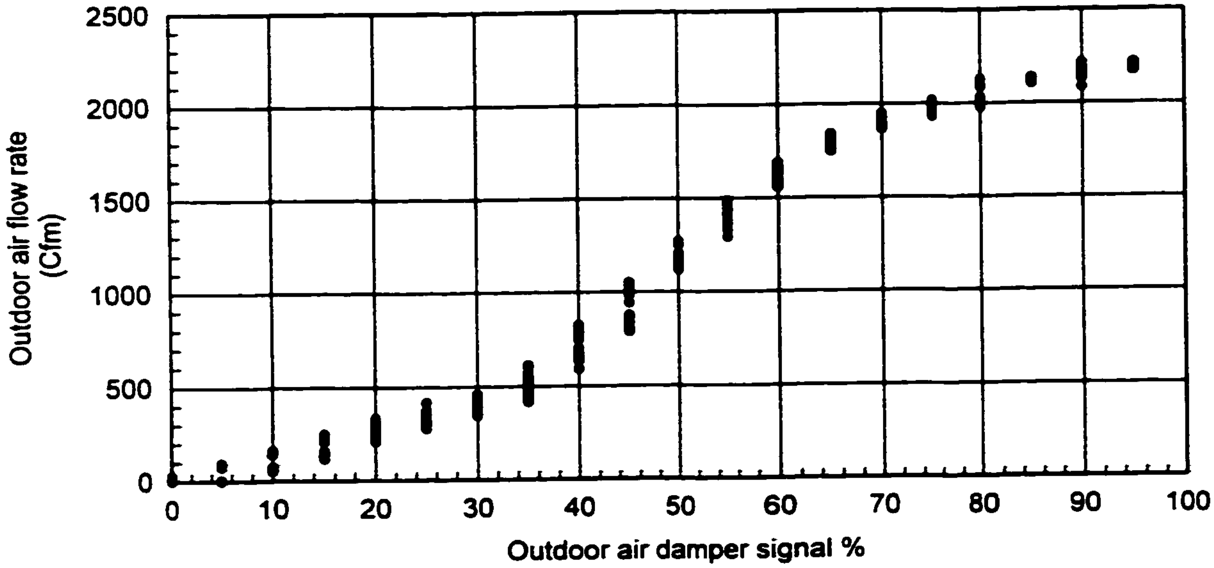


(a) the modified dampers.

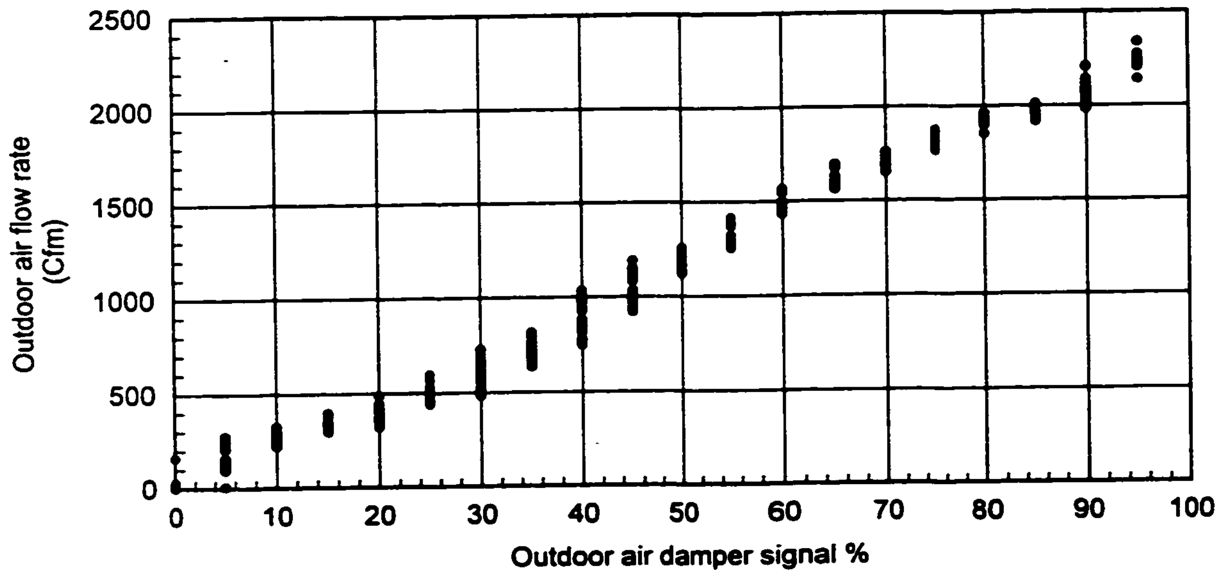


(b) the modified dampers with software linearization.

Figure 4.11 Outdoor Air Flow Rate, Three Coupled Dampers and 0.6 inWC Set Point.

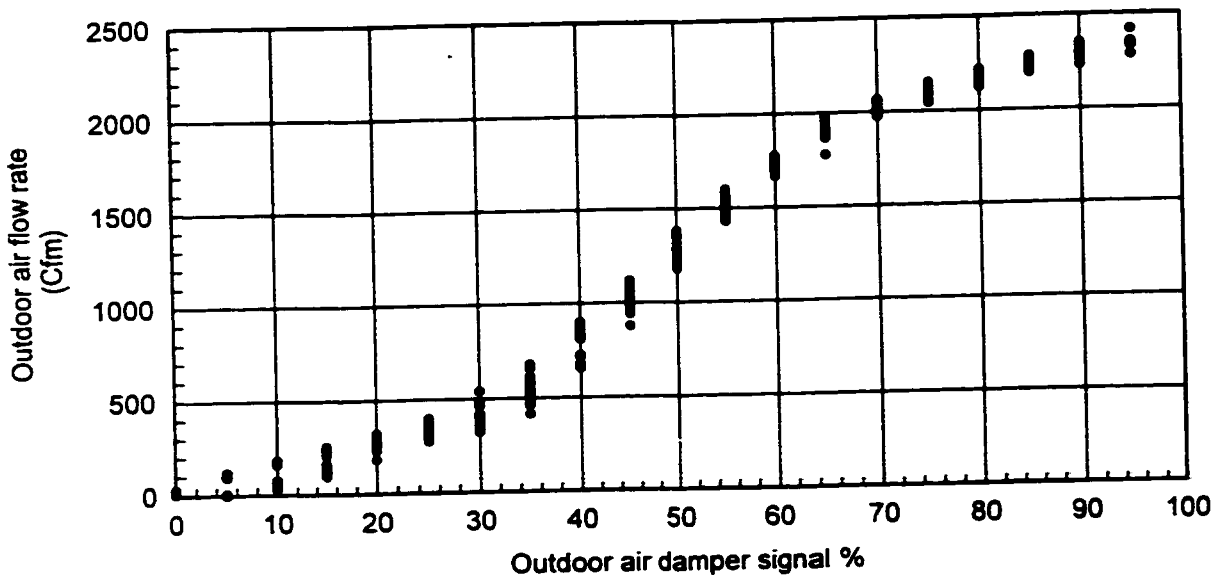


(a) the modified dampers.

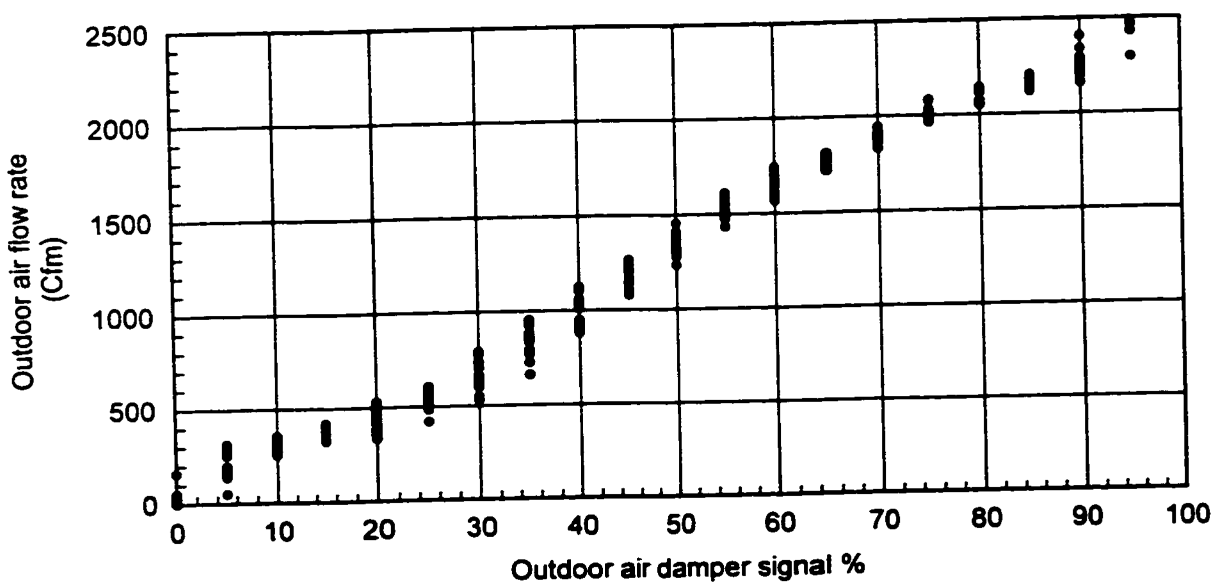


(b) the modified dampers with software linearization.

Figure 4.12 Outdoor Air Flow Rate, Two Coupled Dampers and 0.5 inWC Set Point.

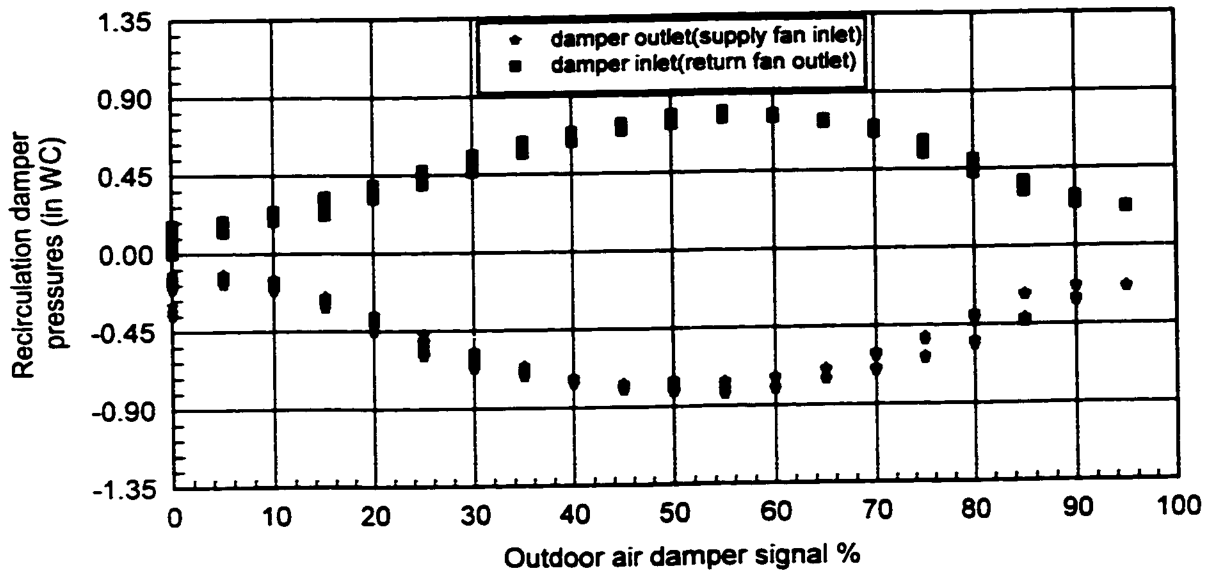


(a) the modified dampers.

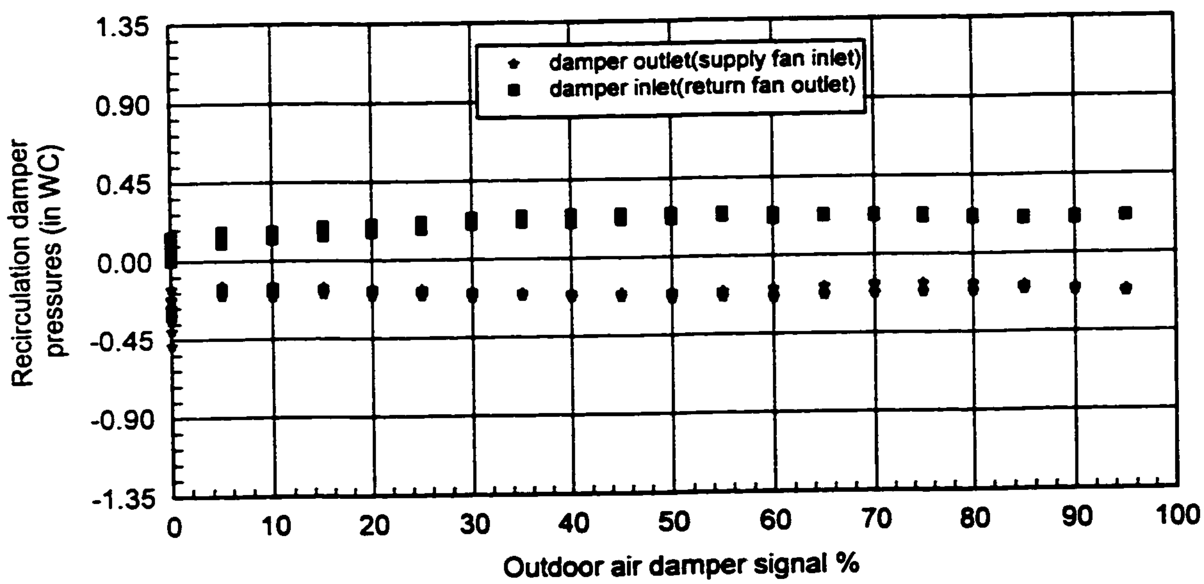


(b) the modified dampers with software linearization.

Figure 4.13 Outdoor Air Flow Rate, Two Coupled Dampers and 0.6 inWC Set Point.

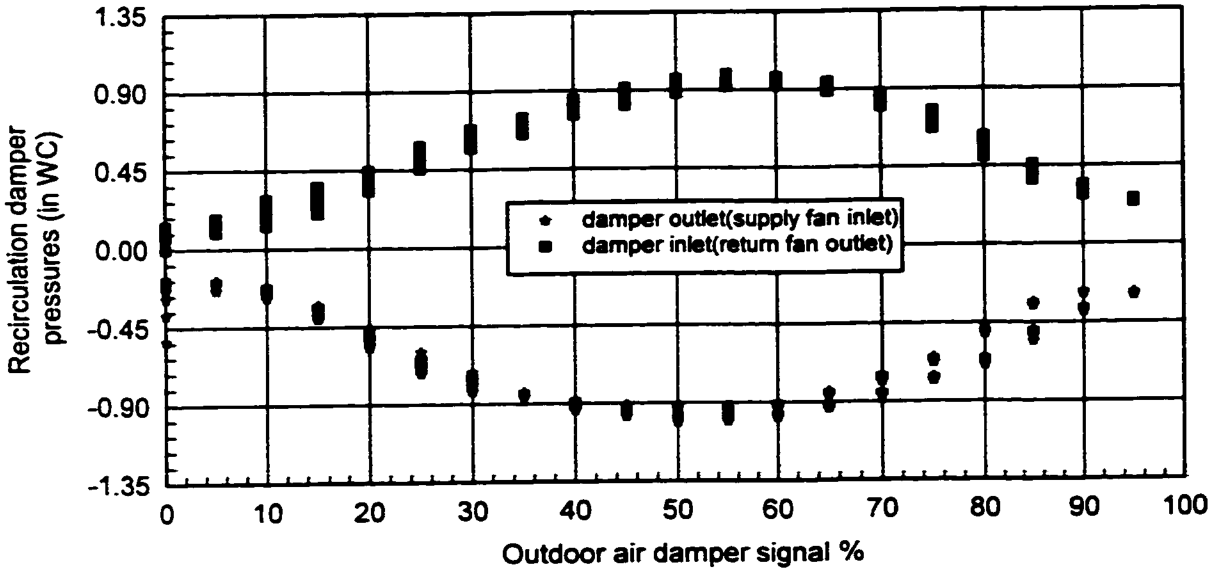


(a) the dampers as supplied by the manufacturer.

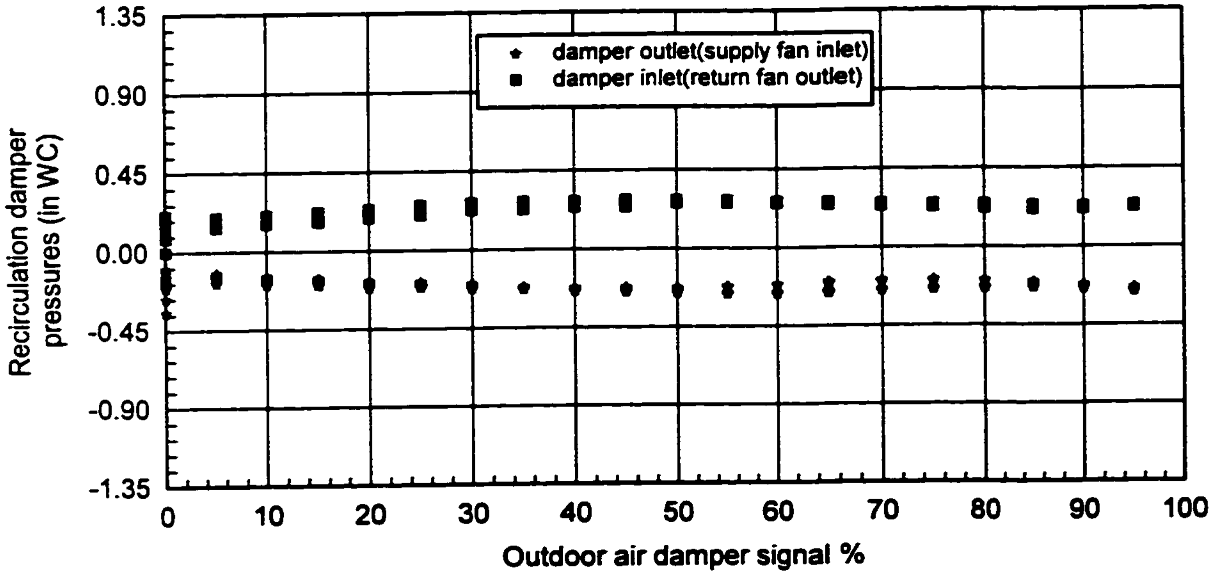


(b)) the dampers as supplied by the manufacturer with software linearization.

Figure 4.14 Recirculation Damper Pressures, Three Coupled Dampers and 0.5 inWC Set Point.

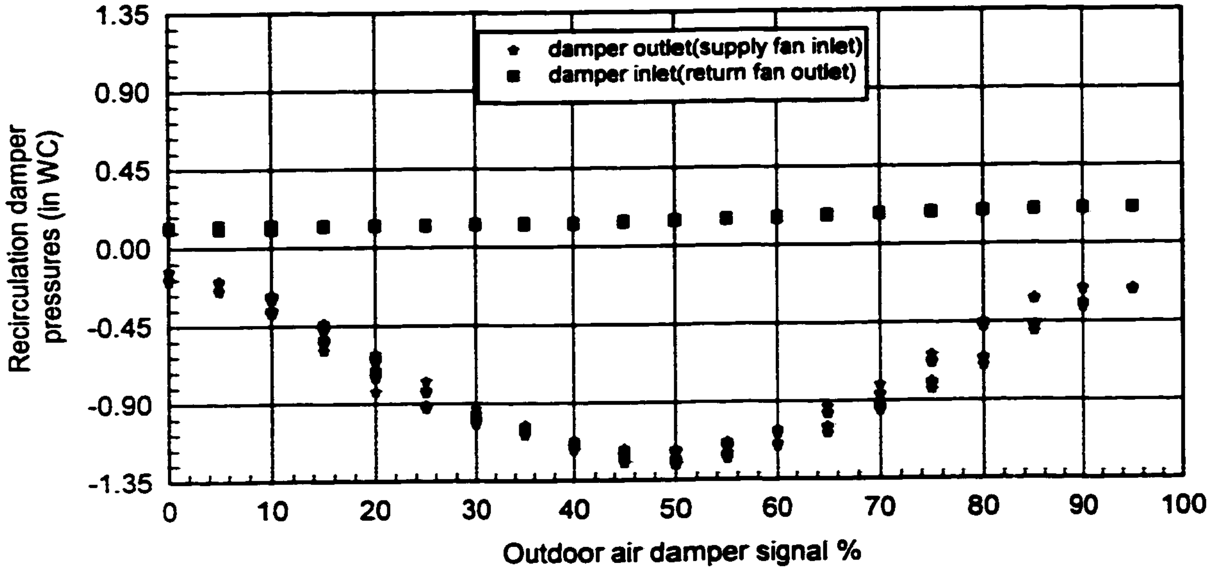


(a) the dampers as supplied by the manufacturer.

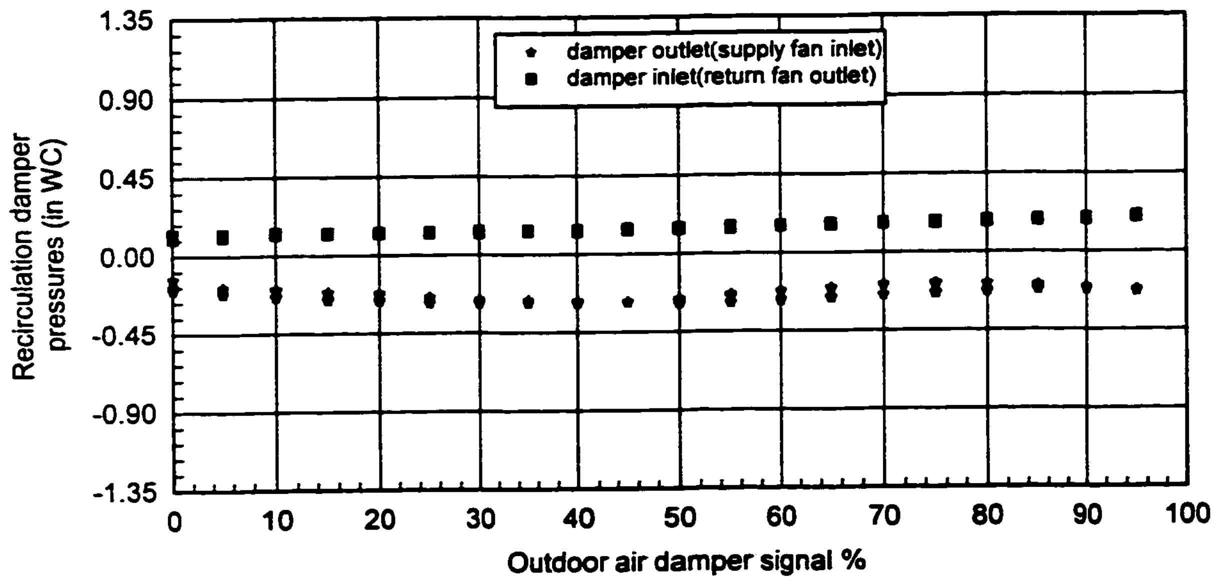


(b)) the dampers as supplied by the manufacturer with software linearization.

Figure 4.15 Recirculation Damper Pressures, Three Coupled Dampers and 0.6 inWC Set Point.

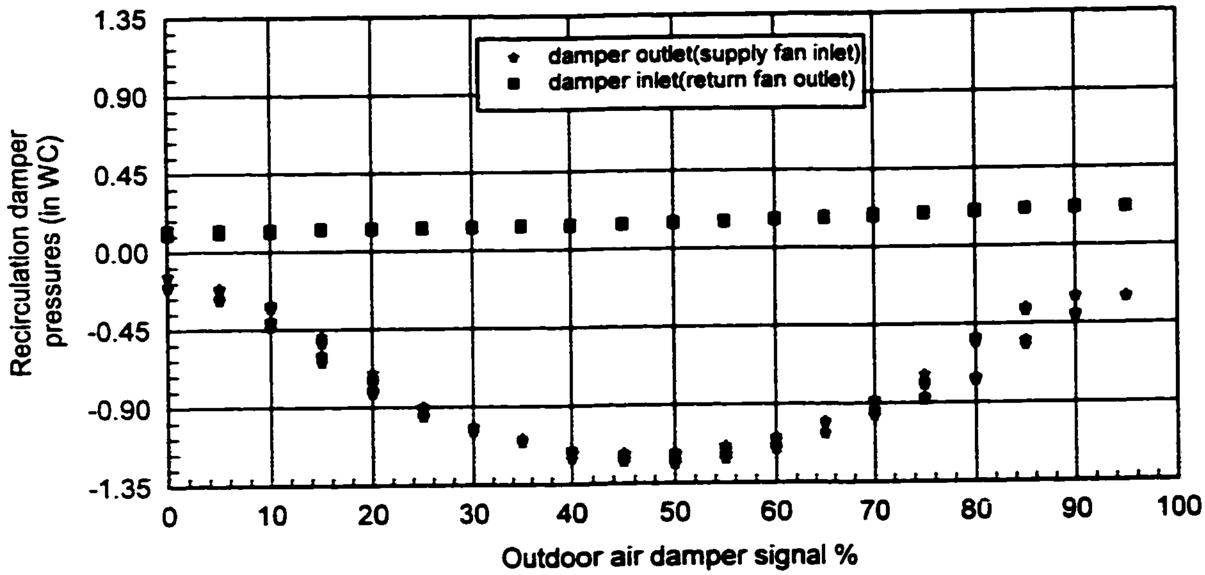


(a) the dampers as supplied by the manufacturer.

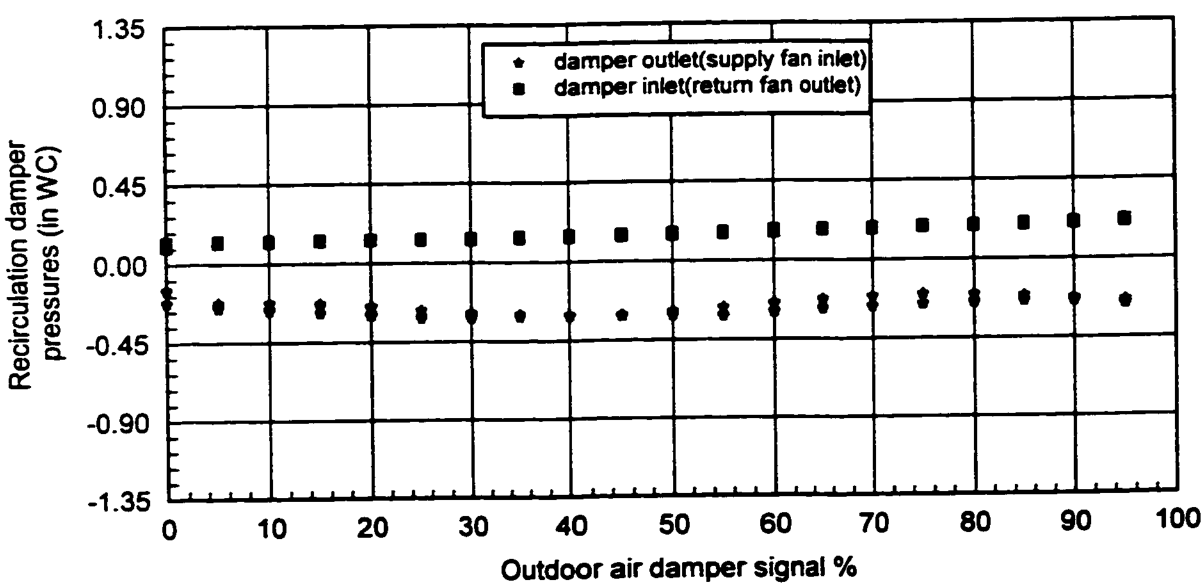


(b)) the dampers as supplied by the manufacturer with software linearization.

Figure 4.16 Recirculation Damper Pressures, Two Coupled Dampers and 0.5 inWC Set Point.

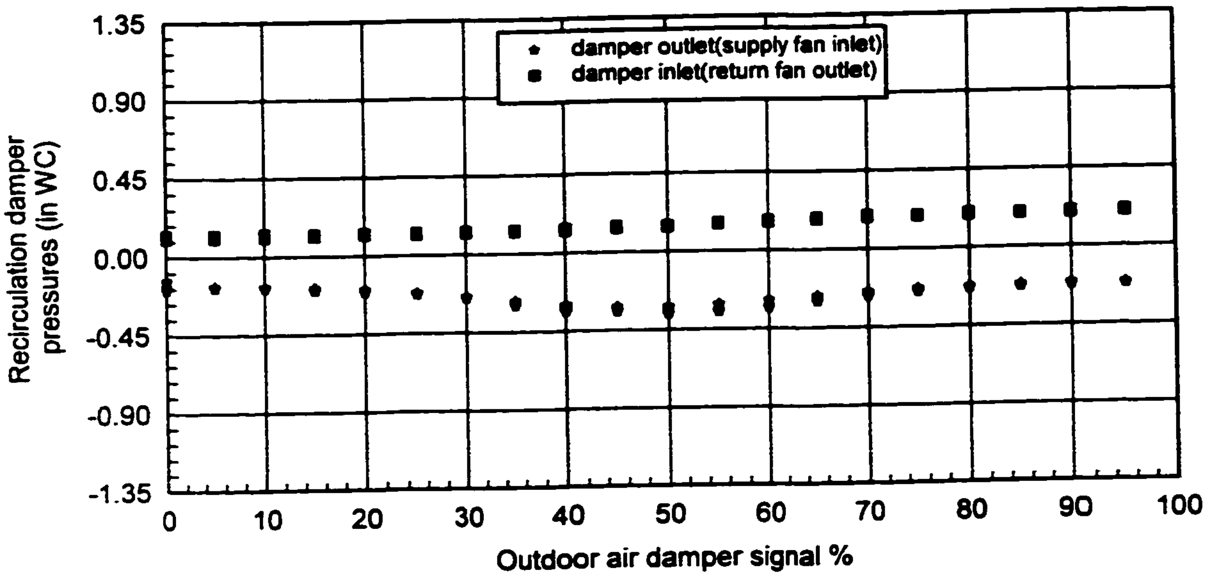


(a) the dampers as supplied by the manufacturer.

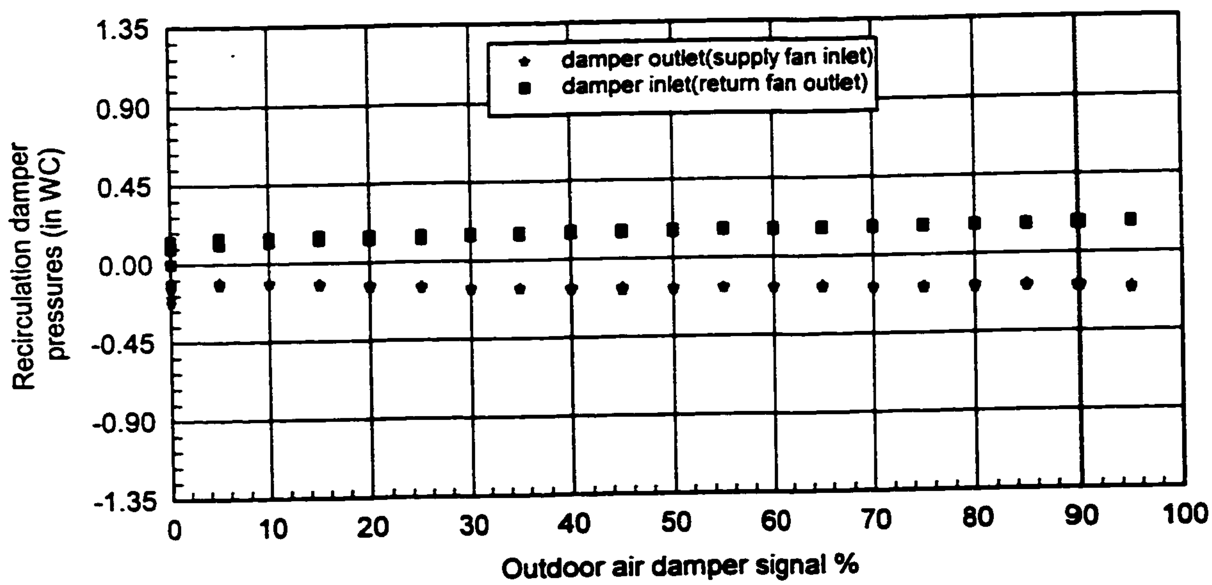


(b)) the dampers as supplied by the manufacturer with software linearization.

Figure 4.17 Recirculation Damper Pressures, Two Coupled Dampers and 0.6 inWC Set Point.

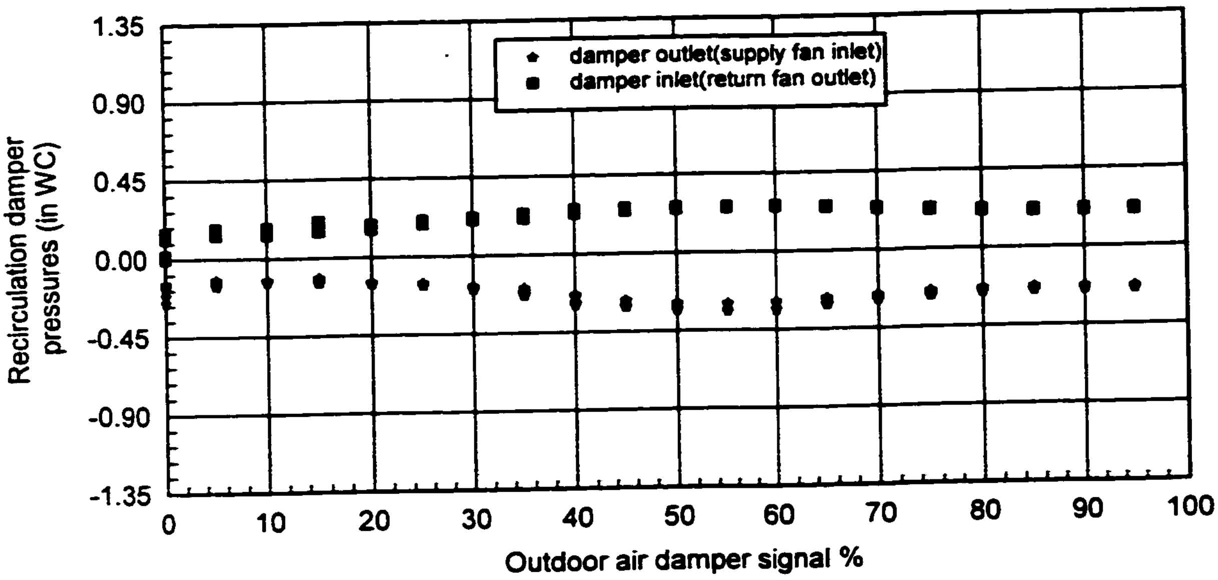


(a) the modified dampers.

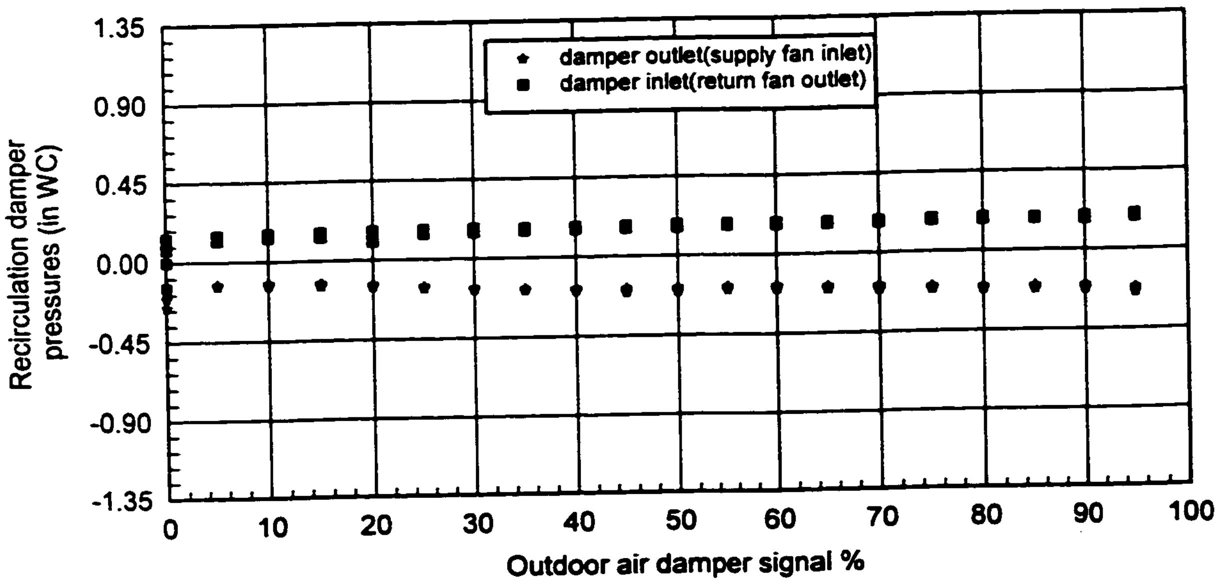


(b) the modified dampers with software linearization.

Figure 4.18 Recirculation Damper Pressures, Three Coupled Dampers and 0.5 inWC Set Point.

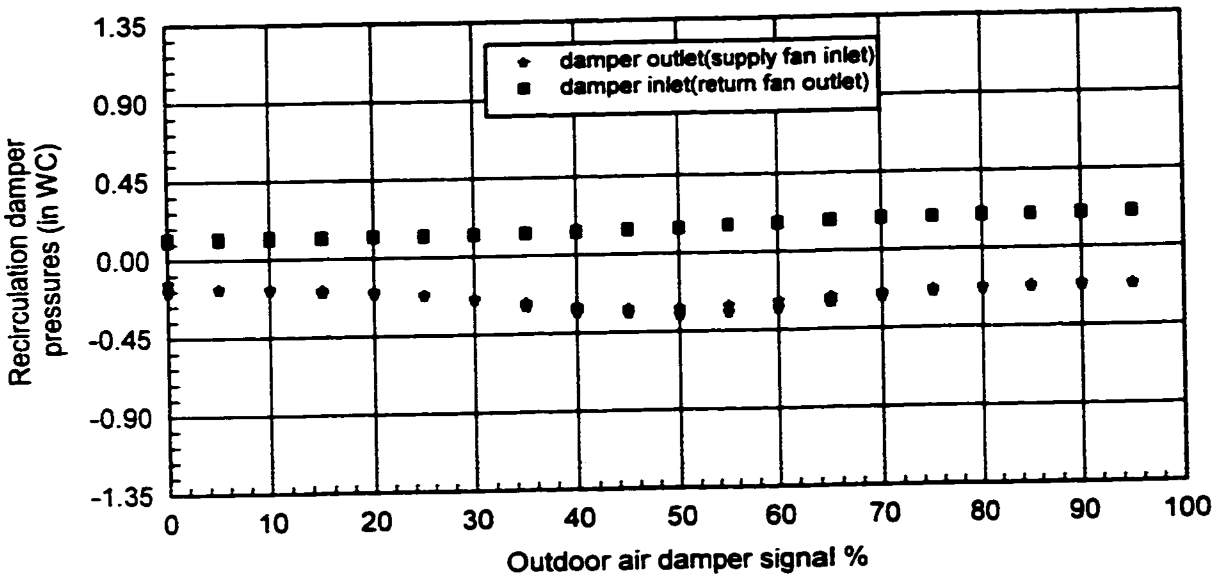


(a) the modified dampers.

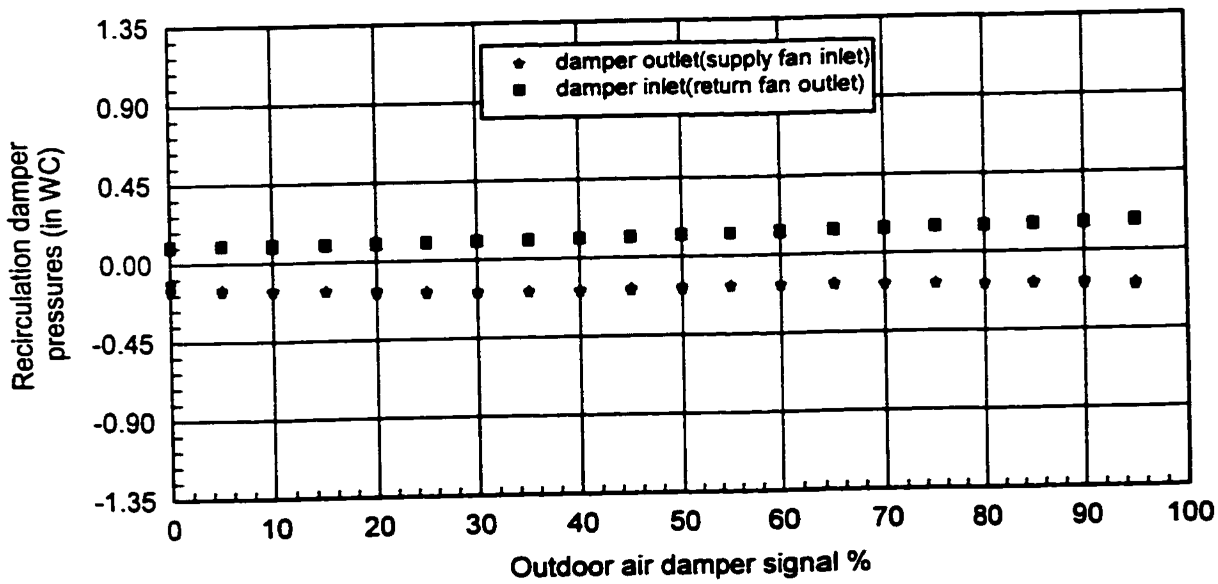


(b) the modified dampers with software linearization.

Figure 4.19 Recirculation Damper Pressures, Three Coupled Dampers and 0.6 inWC Set Point.

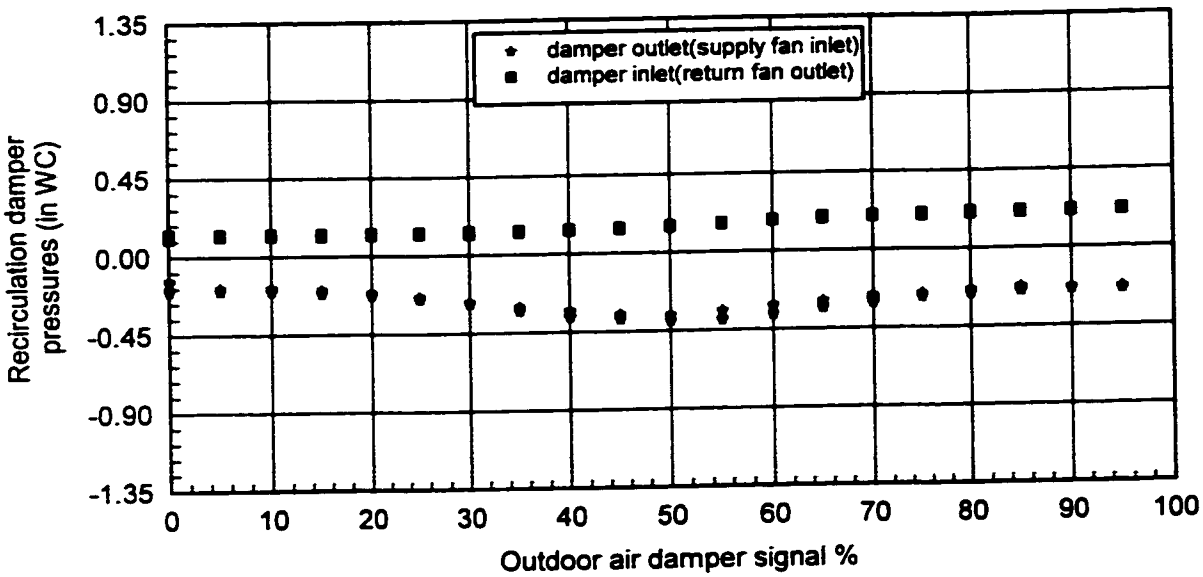


(a) the modified dampers.

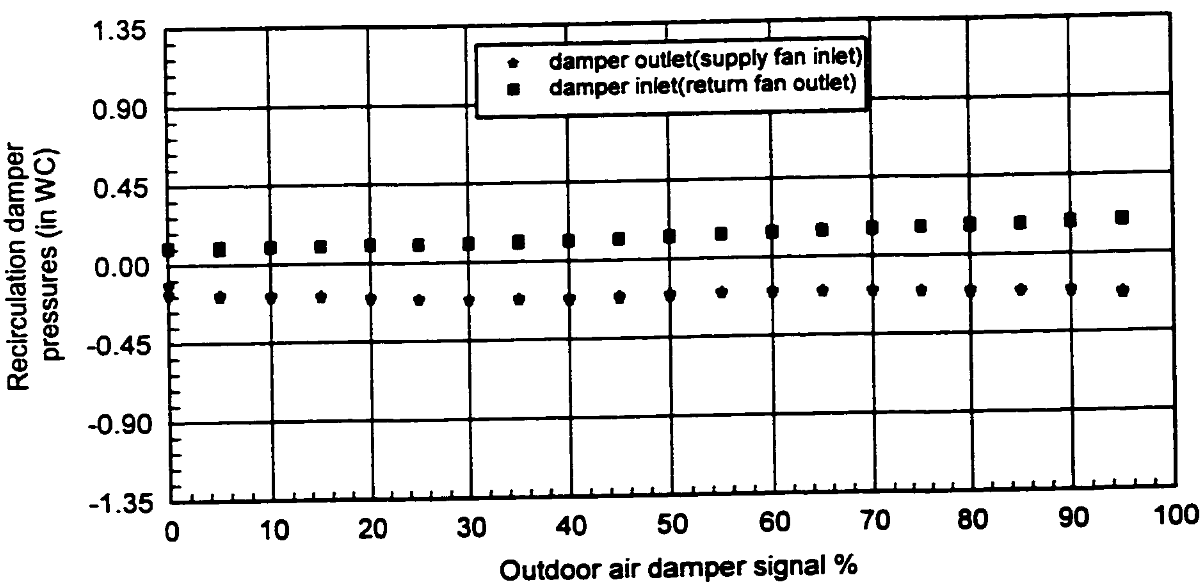


(b) the modified dampers with software linearization.

Figure 4.20 Recirculation Damper Pressures, Two Coupled Dampers and 0.5 inWC Set Point.

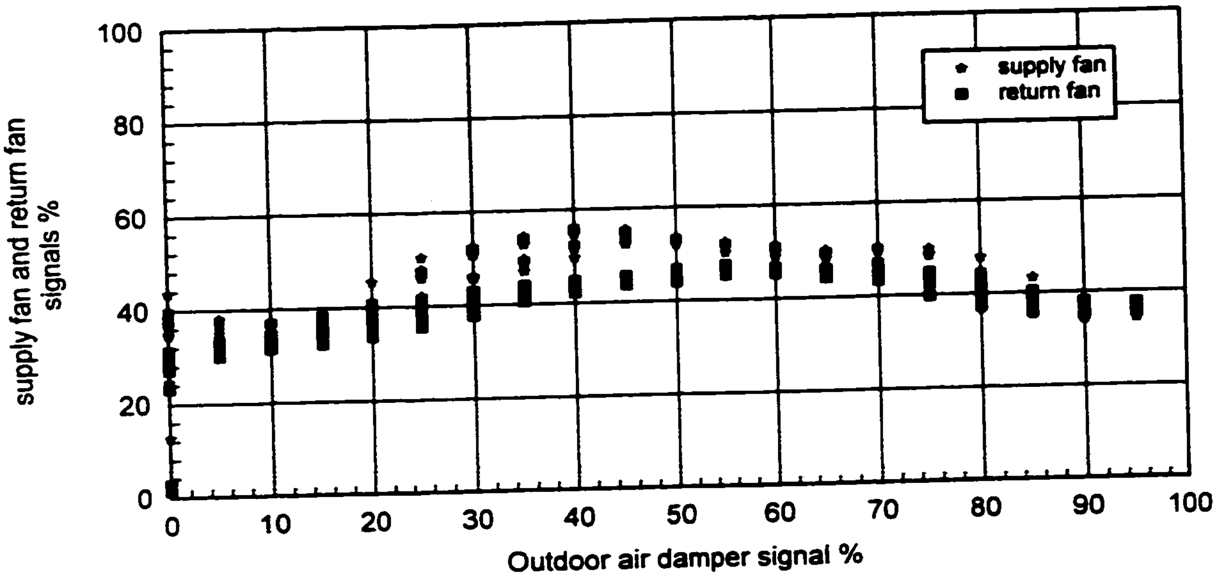


(a) the modified dampers.

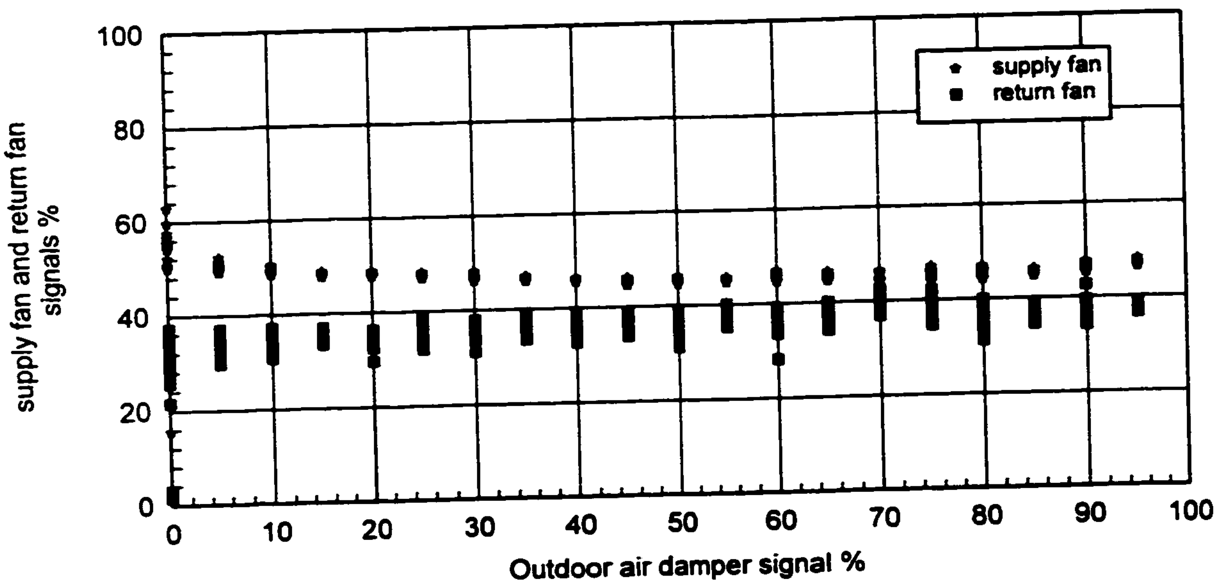


(b) the modified dampers with software linearization.

Figure 4.21 Recirculation Damper Pressures, Two Coupled Dampers and 0.6 inWC Set Point.

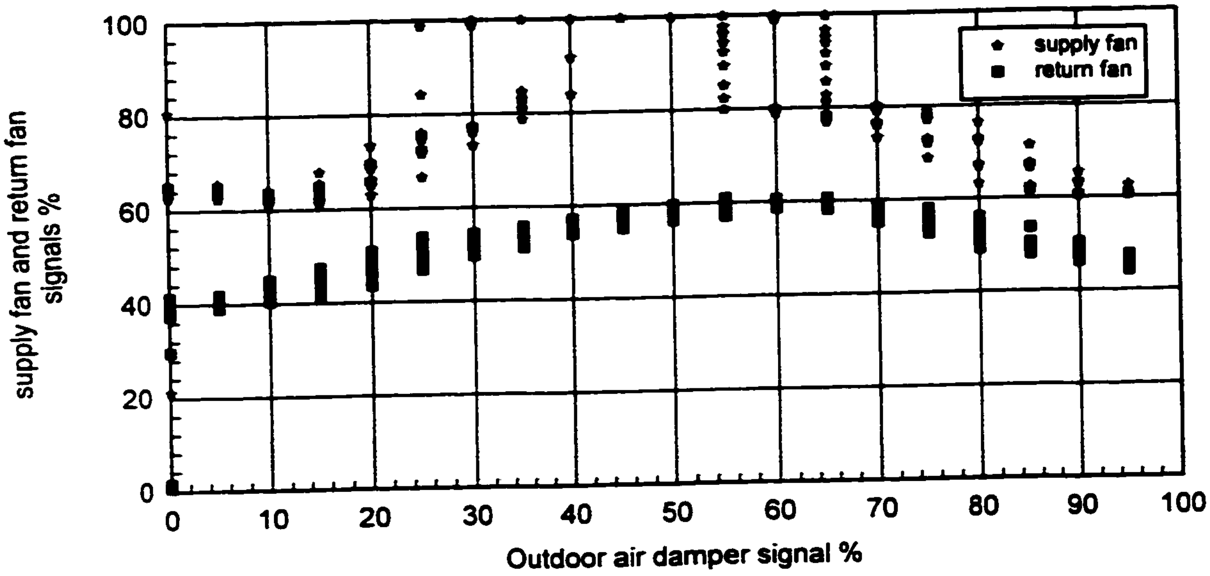


(a) the dampers as supplied by the manufacturer.

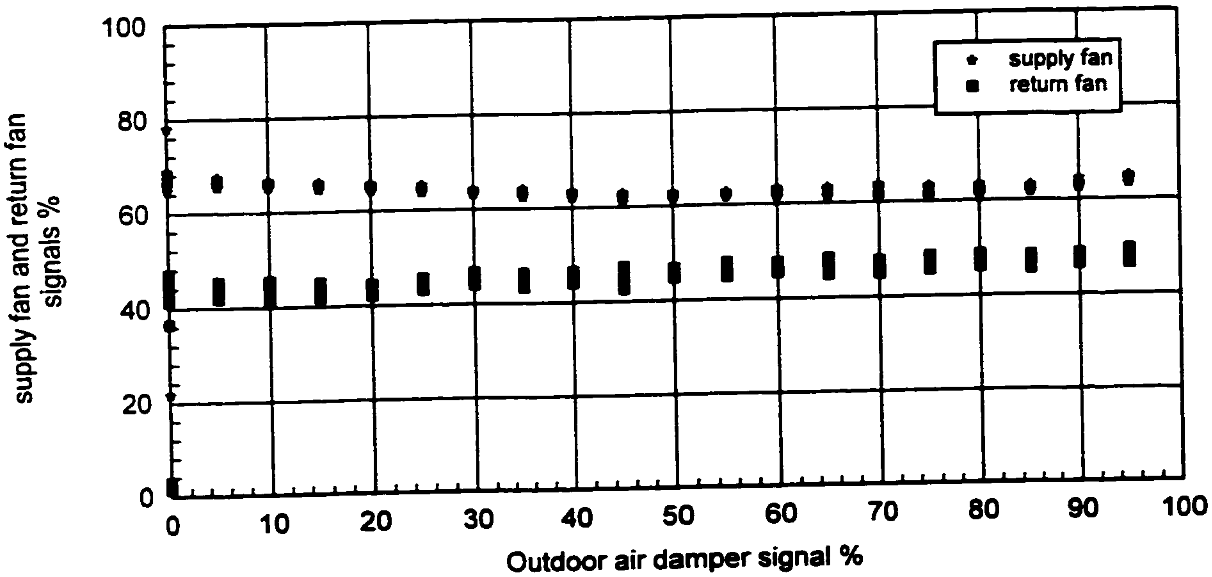


(b)) the dampers as supplied by the manufacturer with software linearization.

Figure 4.22 Fans Signals, Three Coupled Dampers and 0.5 inWC Set Point.

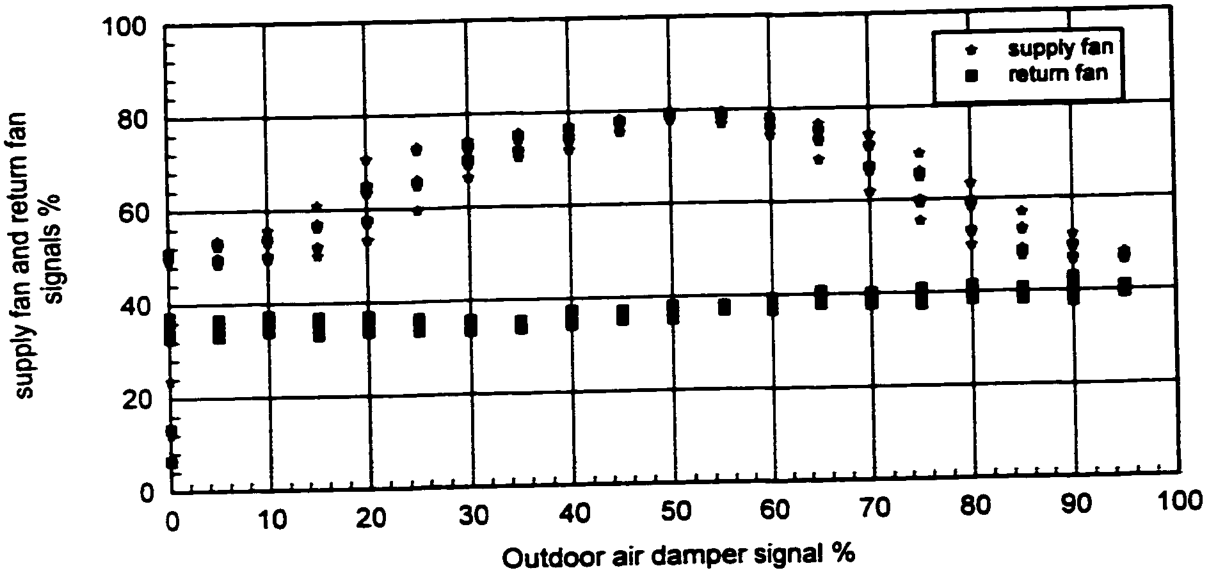


(a) the dampers as supplied by the manufacturer.

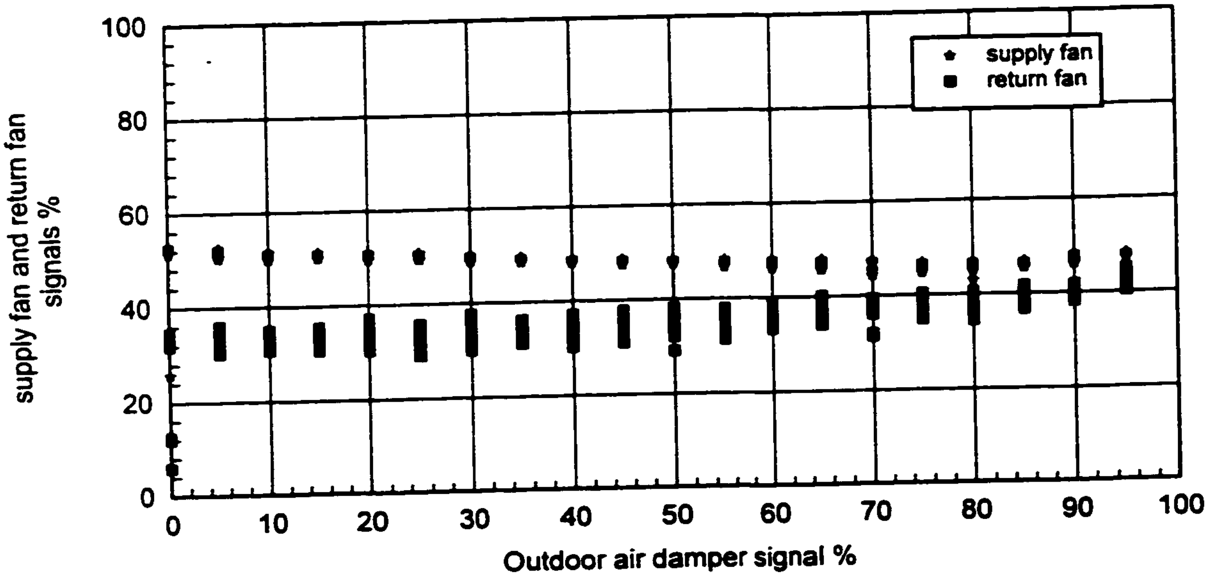


(b)) the dampers as supplied by the manufacturer with software linearization.

Figure 4.23 Fans Signals, Three Coupled Dampers and 0.6 inWC Set Point.

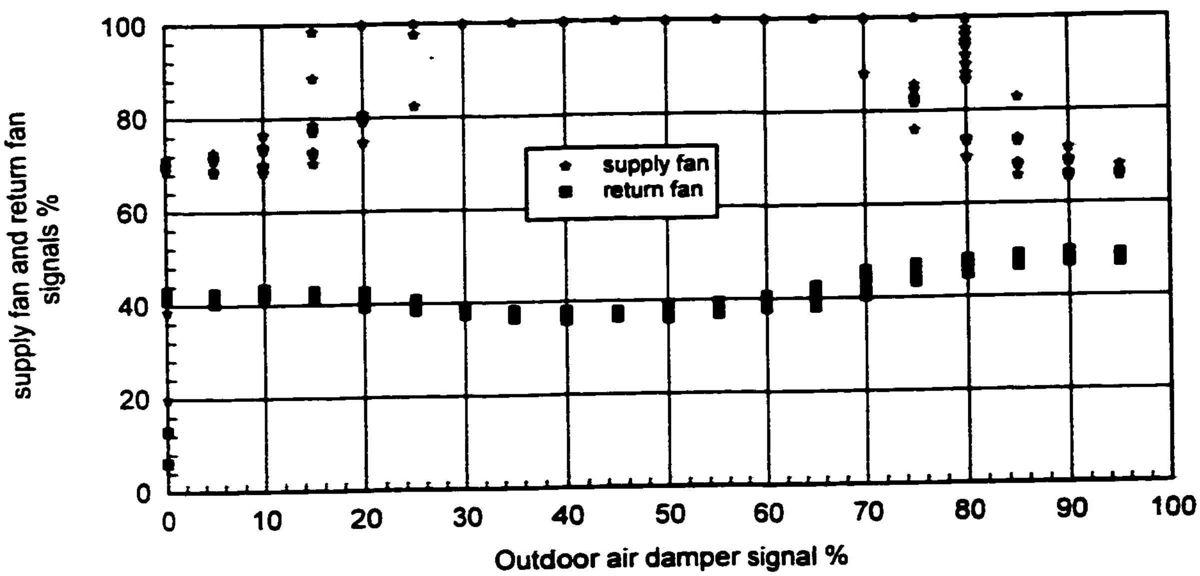


(a) the dampers as supplied by the manufacturer.

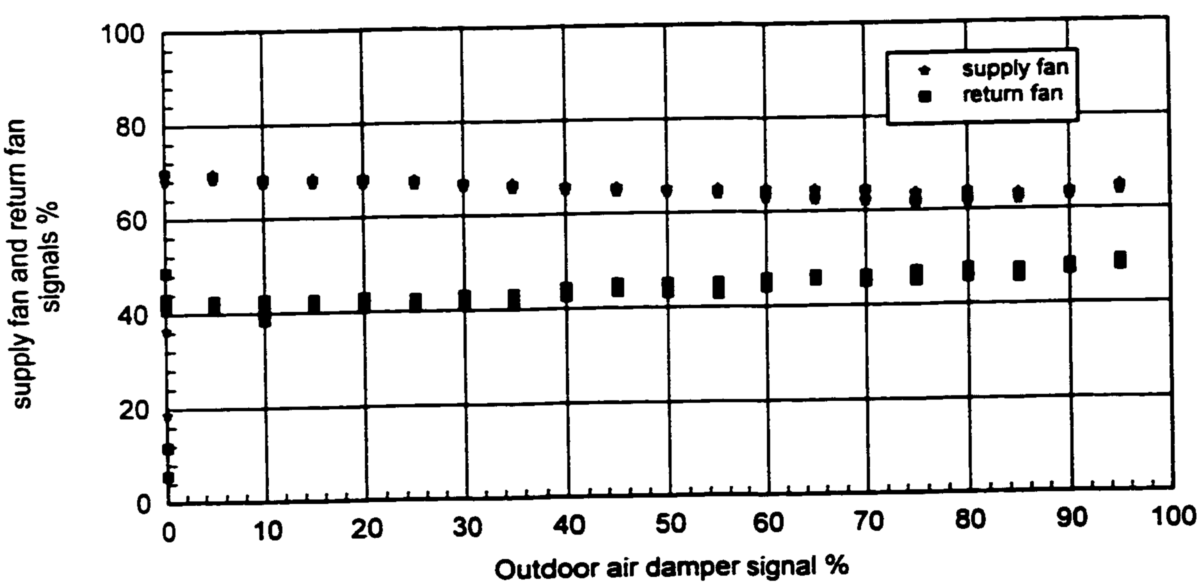


(b)) the dampers as supplied by the manufacturer with software linearization.

Figure 4.24 Fans Signals, Two Coupled Dampers and 0.5 inWC Set Point.

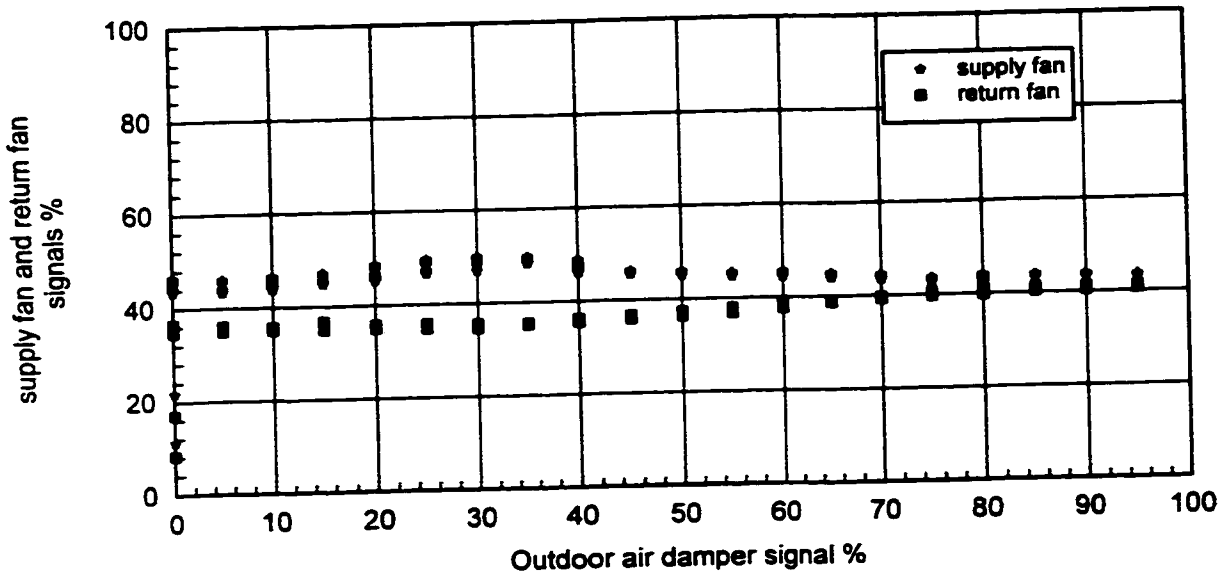


(a) the dampers as supplied by the manufacturer.

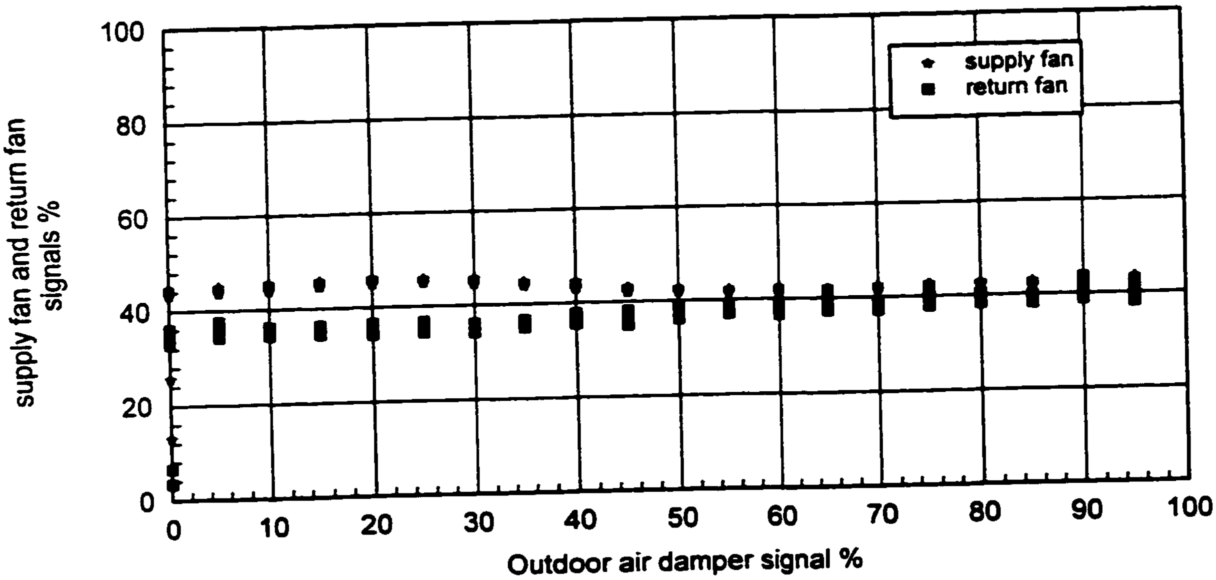


(b)) the dampers as supplied by the manufacturer with software linearization.

Figure 4.25 Fans Signals, Two Coupled Dampers and 0.6 inWC Set Point.

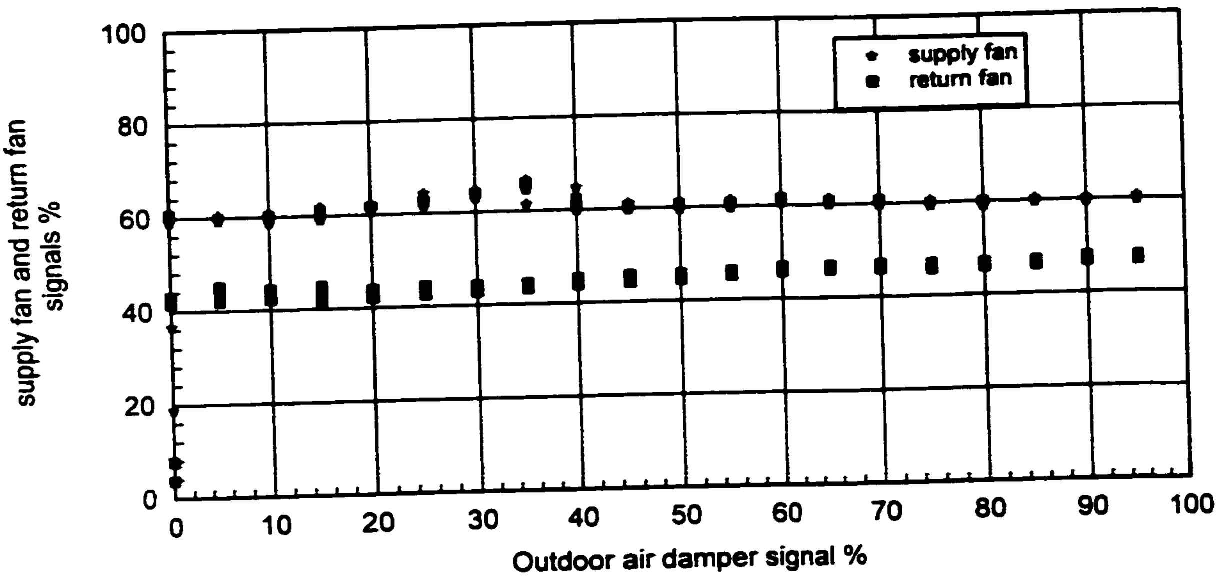


(a) the modified dampers.

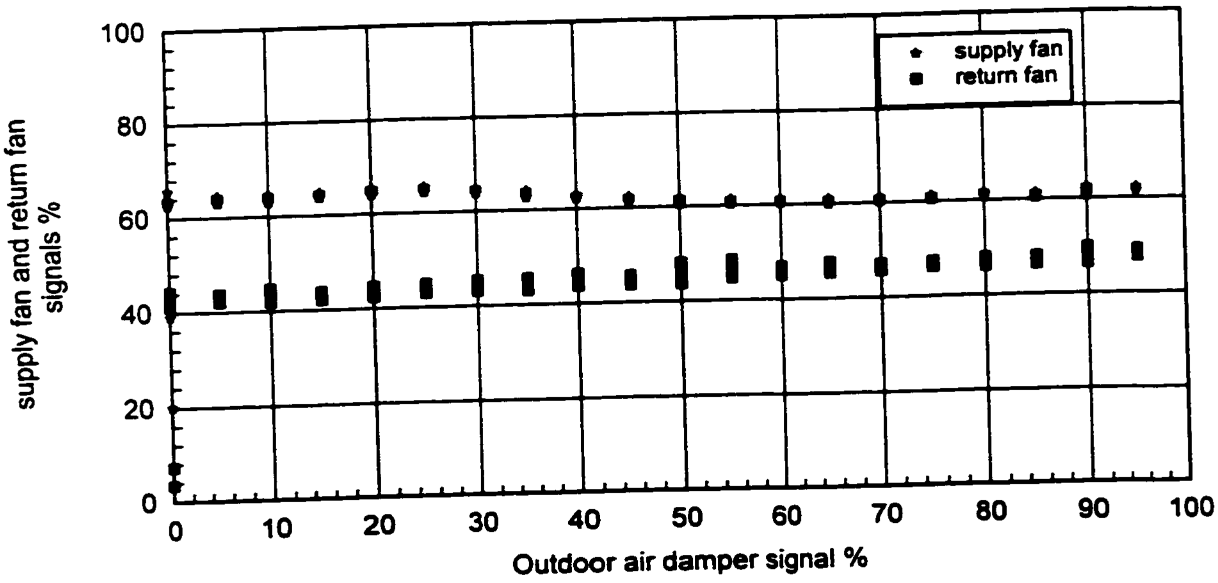


(b) the modified dampers with software linearization.

Figure 4.26 Fans Signals, Three Coupled Dampers and 0.5 inWC Set Point.

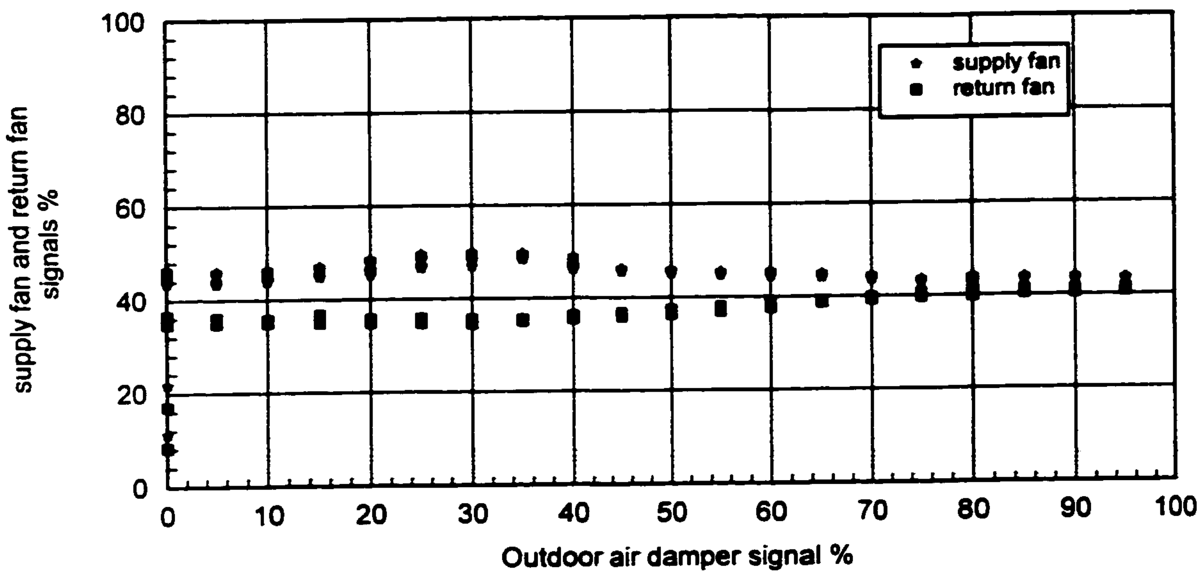


(a) the modified dampers.

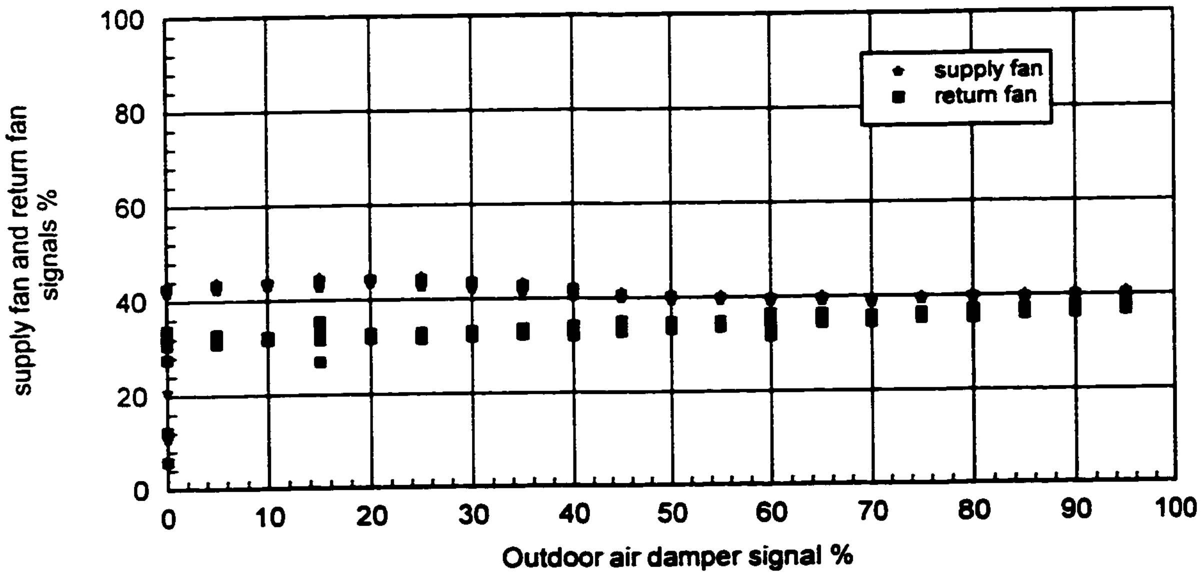


(b) the modified dampers with software linearization.

Figure 4.27 Fans Signals, Three Coupled Dampers and 0.6 inWC Set Point.

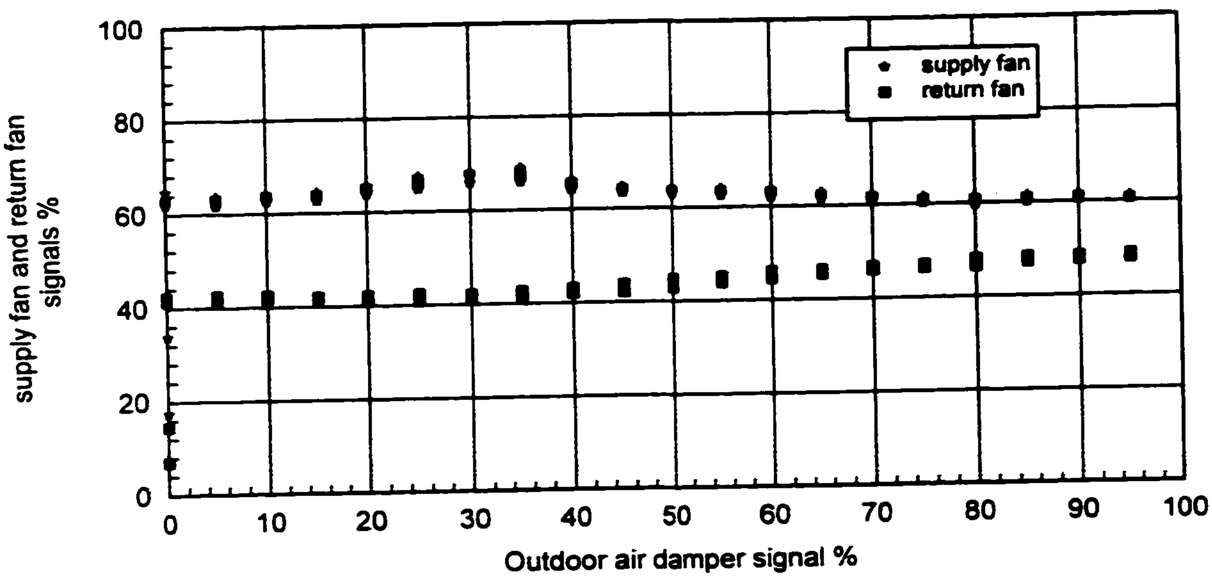


(a) the modified dampers.

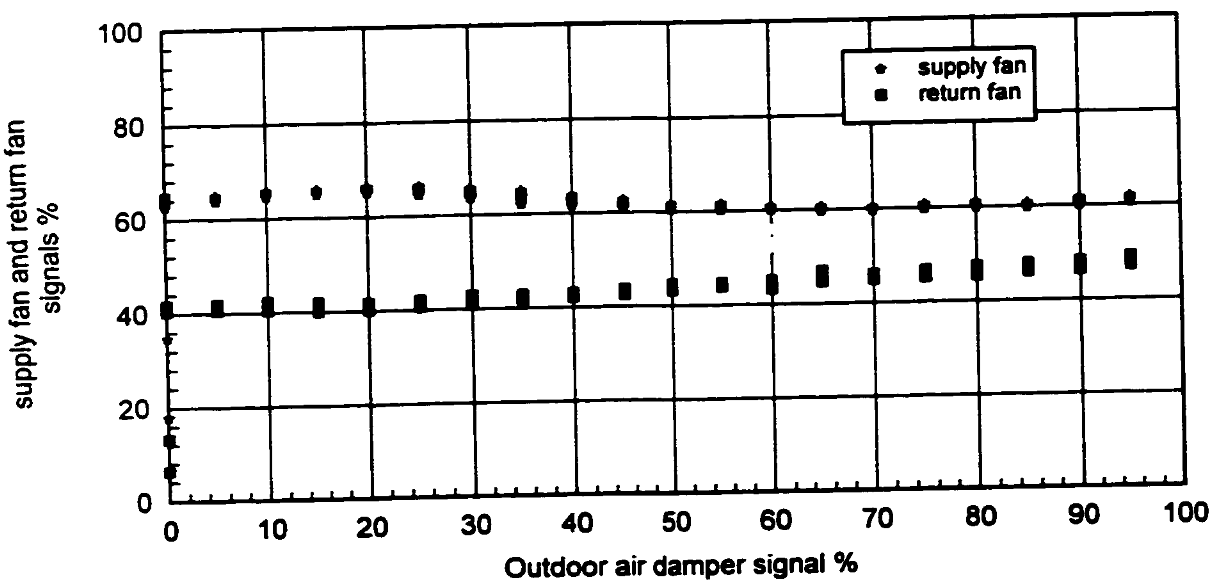


(b) the modified dampers with software linearization.

Figure 4.28 Fans Signals, Two Coupled Dampers and 0.5 inWC Set Point.



(a) the modified dampers.



(b) the modified dampers with software linearization.

Figure 4.29 Fans Signals, Two Coupled Dampers and 0.6 inWC Set Point.

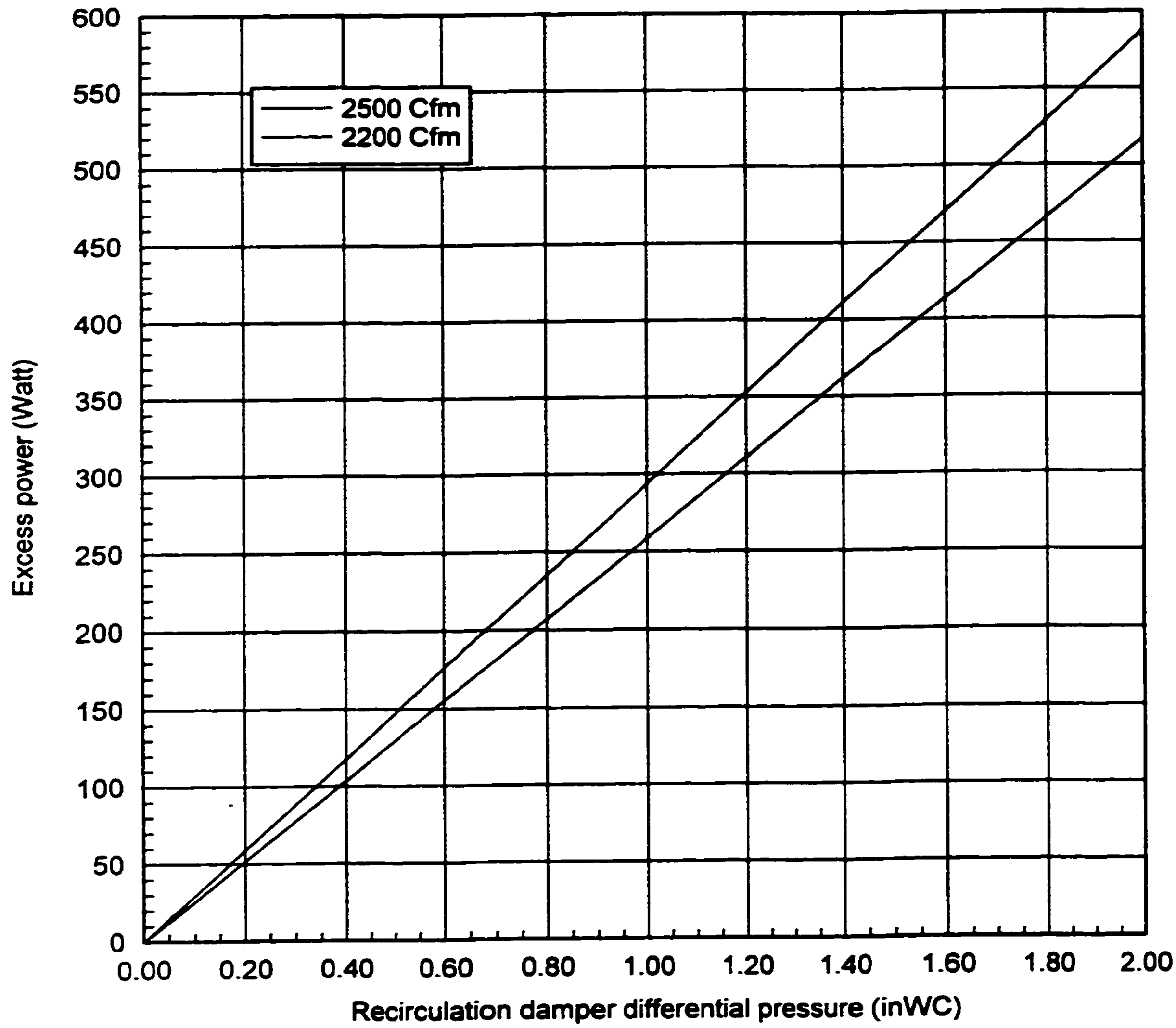


Figure 4.30 Excess Power Versus the Recirculation Damper Differential Pressure.

CHAPTER 5

CONCLUSIONS

The flow rate through a damper is related to the head loss across the damper by a flow coefficient. The head loss represents power consumption. An economizer consists of three dampers. It was shown that excess power consumption due to the economizer dampers is dependent on the pressure across the recirculation damper. It was also shown that the pressure across the recirculation damper may be reduced by using linear dampers and by control strategy.

The geometry of the opposed blade dampers, used for this investigation has been studied in order to design a damper with linear throat area characteristic. The kinematic and dynamic studies indicated that it is possible to linearize damper characteristic by geometry design - length of links - and by software modification of the control signal.

The study showed that the flow coefficient is directly proportional to the damper throat area. The throat area as a function of the control signal, i.e., blade angles, may be varied by the selection of the length of the coupling link and the driven axis. The linkage must be verified to ensure that the damper closes tightly, opens fully, and does not bind. It was shown, by kinematic and dynamic analyses, that binding may occur if the coupler link aligns with a driven crank arm.

Software linearization of the throat area versus the control signal can be accomplished by 'compensating' the ideal signal before it is sent to the motor. The signal is that required by a feedback control algorithm is the ideal signal. The signal is that sent to the motor is the compensated signal. The compensated signal is related to the ideal signal by an algorithm based on the throat area as a function of the ideal control signal.

Linearization of the throat area versus the control signal will reduce excess power consumption and decrease variation in fan control signals required to maintain constant pressures. Reducing fan speed variations extend the operating range of a fan.

Two control strategies were investigated. In the first strategy, the three economizer dampers are coupled so that the outdoor air and discharge damper open as the recirculation damper closes. In a second strategy, the discharge damper is always open during operation and the outdoor air and recirculation dampers are coupled so that the outdoor air damper open as the recirculation damper closes. The two coupled dampers control strategy will require slightly less power than three coupled damper strategy.

The study showed that the linearization is the most important factor, whether the three coupled or two coupled dampers control strategy is used. Linearization provides good control of the air flow rates, from 0% to 100%. Whereas linearization may be accomplished by means of software, power savings may be realized without capital expenditure.

Further investigations are required to evaluate power saving for actual systems considering damper control strategy, actual weather data, and occupancy schedule. The reduction of excess fan power depends on the control signal. The maximum reduction would occur at approximately 50% control signal. The control signal depends on weather data and occupancy.

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APPENDIX I

COMPUTER PROGRAM FOR DYNAMIC ANALYSIS

I.I Program Listing

```
C     ***************************************************************************
C     This program is designed to calculate the driven angle, the torque
C     coefficient based on the given drive angle and the blade geometry for
C     the opposed blade damper used for this study.
C     ***************************************************************************
C
      DIMENSION X(7),Y(7)
      CHARACTER*20 FILIN,OUTPUTF
C     INPUT FILE
      WRITE (*,*)'WRITE THE FILE NAME FOR INPUT DATA'
      READ (*,*) FILIN
      OPEN (UNIT=1,FILE=FILIN,STATUS='OLD')
      READ (1,*) OUTPUTF
C     OUTPUT FILE
      OPEN (UNIT=3,FILE=OUTPUTF,STATUS='OLD')
      WRITE (3,110)
      FORMAT (4X,'ALPHA'T14,'DALPHA',T24,'BETA',T34,'DBETA'
     * ,T44,'GAMA',T54,'TCOIFF',T65,'(B-G)or(G-A)')
C
C     CONVERTING THE ANGLES FROM DEGREES TO RADIANS
      PI = 2*ASIN(1)
      DEGRAD = PI/180
      ALPHA = 45
      BETA = 135
      ALPHAR = ALPHA * DEGRAD
      BETAR = BETA * DEGRAD
C     READ INPUT DATA
      READ (1,*)
      READ (1,*) AA,ARML,OSRT,COUPL
C     CALCULATING OSLT IF THE COUPLER LENGTH IS SPECIFIED
      IF (OSLT .EQ. 0) THEN
        CGAMAR=(ARML*ABS(COS(ALPHAR)) + ARML*ABS(COS(BETAR)))
     *  / COUPL
        GAMAR = ACOS(CGAMAR)
        GAMA = GAMAR/DEGRAD
        DY = COUPL * SIN(GAMAR)
```

```
      OSLT =AA+ ABS(ARML*SIN(ALPHAR)) - ABS(DY) - ABS(ARML*
     *  SIN(BETAR))
      END IF
C     CALCULATING COUPLER LENGTH IF THE OSLT IS SPECIFIED
      X(1) = 0
      Y(1) = 0
      X(2) = 0
      Y(2) = AA/2
      X(7) = 0
      Y(7) = - AA/2
      X(3) = ARML * COS(ALPHAR)
      Y(3) = ARML * SIN(ALPHAR) + Y(2)
      X(4) = X(3)
      Y(4) = Y(3) - OSRT
      X(6) = ARML *  COS(BETAR)
      Y(6) = ARML * SIN(BETAR) + Y(7)
      X(5) = X(6)
      Y(5) = Y(6) + OSLT
C
      COUPL = SQRT((X(4)-X(5))**2+(Y(4)-Y(5))**2)
C
      TEST = 0
      ARAD = ALPHAR
      BRAD = BETAR
C     READ INPUT DATA
      READ (1,*)
C     DA=0 (THE DAMPER IS DRIVEN FROM ANGLE ALPHA)
C     DB=0 (THE DAMPER IS DRIVEN FROM ANGLE BETA)
C     DEGR IS THE CHANGE IN THE DRIVE ANGLE.
      READ (1,*) DA,DB,DEGR
C
      NA = 2*ALPHA/DEGR
      NAT = 2*NA
C
      IF (DA .EQ. 0) THEN
         ALPHA = ALPHA + DEGR
      ELSE IF (DB .EQ. 0) THEN
         BETA = BETA - DEGR
      END IF
C
      DO 20 I=0,NAT
C     PY4&PY5 ARE DESIGNED TO CONTROL GAMA.
      PY4 = Y(4)
      PY5 = Y(5)
C     THE DAMPER IS DRIVEN FROM ALPHA
      IF (DA .EQ. 0) THEN
```

```
         IF (I .LE. NA)  THEN
           ALPHA = ALPHA - DEGR
         ELSE
           ALPHA = ALPHA + DEGR
         END IF
         CALL ANGLBE(AA,ARML,COUPL,OSLT,OSRT,ALPHA,BETA
     *   ,ALPHAR,BETAR,I,TEST,DA,DB,NA)
C       THE DAMPER IS DRIVEN FROM BETA
       ELSE IF (DB .EQ. 0) THEN
         IF (I .LE. NA)  THEN
           BETA = BETA + DEGR
         ELSE
           BETA = BETA - DEGR
         END IF
         CALL ANGLBE(AA,ARML,COUPL,OSLT,OSRT,BETA,ALPHA
     *   ,BETAR,ALPHAR,I,TEST,DA,DB,NA)
       END IF
C
C       ANGLE GAMA CALCULATION.
       IF (PY4 .GT. PY5) THEN
         GAMAR = ACOS((ARML/COUPL)*(COS(ALPHAR)-COS(BETAR)))
         GAMA = GAMAR/DEGRAD
       ELSE
         GAMAR = ACOS((-ARML/COUPL)*(COS(ALPHAR)-COS(BETAR)))
         GAMA = (GAMAR/DEGRAD) - 180
       END IF
C       TORQUE CALCOLATION.
C        RTA (RESESTANCE TORQUE WHEN THE DAMPER IS DRIVEN FROM
C       BETA)
C        RTB (RESESTANCE TORQUE WHEN THE DAMPER IS DRIVEN FROM
C       BETA)
       RST = 1.0
       BNA = 2
       BNB = 3
       DARAD = ALPHAR - ARAD
       DBRAD = BETAR - BRAD
       ARAD = ALPHAR
       BRAD = BETAR
C
       IF (DB .EQ. 0) GOTO 60
       IF (DA .EQ. 0) GOTO 70
C
 60    IF (DBRAD .EQ. 0) THEN
          TBCOEF = 0
       ELSE
         TBCOEF = ABS(DARAD/DBRAD)
```

```
        TMB = RST*(BNA*TBCOEF+BNB)
      END IF
        BGA = GAMA - ALPHA
      WRITE (3,130) ALPHA,DARAD,BETA,DBRAD,GAMA,TBCOEF,BGA
      GO TO 20
C
 70   IF (DARAD .EQ. 0) THEN
        TACOEF = 0
      ELSE
        TACOEF = ABS(DBRAD/DARAD)
        TMA = RST*(BNA+BNB*TACOEF)
      END IF
      BGA = BETA - GAMA
      WRITE (3,130) ALPHA,DARAD,BETA,DBRAD,GAMA,TACOEF,BGA
 130  FORMAT (7F10.4)
C
 20   CONTINUE
      END


      SUBROUTINE
     * ANGLBE(AA,ARML,COUPL,OSLT,OSRT,ANGL,BTA,ANGLR,BTAR,I,
     * TEST,DA,DB,NA)
C
      PI = 2*ASIN(1)
      DEGRAD = PI/180
      ANGLR = ANGL * DEGRAD
C
      IF (DB .EQ. 0) THEN
         A = - ARML * COS(ANGLR)
         B = AA - OSRT - OSLT - ARML* SIN(ANGLR)
      ELSE IF (DA .EQ. 0) THEN
        A =  ARML * COS(ANGLR)
        B = AA - OSRT - OSLT + ARML*SIN(ANGLR)
      END IF
C
      D  = (A**2 + B**2 + ARML**2 - COUPL**2)/(2*ARML)
      AP = A**2 + B**2
      BP = -2 * A * D
      CP = D**2 - B**2
      Z  = BP**2 - 4*AP*CP
C
      IF (Z .LT. 0) THEN
        WRITE (*,*) Z
        STOP
      ELSE IF (Z .EQ. 0) THEN
```

```
          BTAR = ACOS((-BP)/(2*AP))
          BTA = BTAR/DEGRAD
        ELSE
          ROOT = SQRT(Z)
          BTARP = ACOS((-BP+ROOT)/(2*AP))
          BTARN = ACOS((-BP-ROOT)/(2*AP))
        END IF
C
        IF (BTARP .GT. BTARN) THEN
          BTAT = BTARP
        ELSE
          BTAT = BTARN
        END IF
C
        IF (I .LE. NA) THEN
C       ANGLE  BETA  WHEN  THE  DAMPER  IS NORMALLYCLOSED &
C       ANGLE ALPHA WHEN THE DAMPER IS NORMALLY OPEN.
          IF ((TEST-BTAT) .LE. 0) THEN
            BTAR = BTAT
            BTA = BTAT / DEGRAD
          ELSE
            R = SIN(BTAT)
            BTAR = ASIN(-R)
            BTA = - ((BTAR/DEGRAD) - 180)
          END IF
        ELSE
C       ANGLE ALPHA WHEN THE DAMPER IS NORMALLY CLOSED &
C       AGLE BETA  WHEN THE DAMPER IS NORMALLY OPEN.
          IF ((TEST-BTAT).GT. 0)  THEN
            BTAR = BTAT
            BTA = BTAT / DEGRAD
          ELSE
            R = SIN(BTAT)
            BTAR = ASIN(-R)
            BTA = -((BTAR/DEGRAD) - 180)
          END IF
        END IF
        IF (DA .EQ. 0) BTA = BTA
        IF (DB .EQ. 0) BTA = 180 - BTA
        BTAR = BTA * DEGRAD
        TEST = BTAT
        RETURN
        END
```

I.II Sample Calculations

The input data and output data samples are for the damper as supplied by the manufacturer and driven from angle alpha.

I.II.I Input File

'OUTPUTF.DAT'			
AA.L	ARM.L	OSRT.L	COUPLER.L
3.25	1.25	0	2.6
D.ALPHA	D.BETA	DE.CHANGE	
0	1	1	

I.II.II Output File

ALPHA	DALPHA	BETA	DBETA	GAMA	TCOIFF	(B-G)or(G-A)
45.00	.0000	135.00	.0000	47.1634	.0000	87.8366
44.00	-.0175	135.04	.0007	46.6824	.0423	88.3599
43.00	-.0175	135.09	.0009	46.2016	.0513	88.8920
42.00	-.0175	135.15	.0011	45.7213	.0604	89.4327
41.00	-.0175	135.22	.0012	45.2415	.0694	89.9819
40.00	-.0175	135.30	.0014	44.7624	.0785	90.5395
39.00	-.0175	135.39	.0015	44.2841	.0876	91.1054
38.00	-.0175	135.49	.0017	43.8067	.0967	91.6794
37.00	-.0175	135.59	.0018	43.3304	.1058	92.2616
36.00	-.0175	135.71	.0020	42.8553	.1149	92.8516
35.00	-.0175	135.83	.0022	42.3815	.1241	93.4495
34.00	-.0175	135.96	.0023	41.9092	.1333	94.0551
33.00	-.0175	136.11	.0025	41.4385	.1426	94.6684
32.00	-.0175	136.26	.0027	40.9697	.1519	95.2891
31.00	-.0175	136.42	.0028	40.5027	.1613	95.9174
30.00	-.0175	136.59	.0030	40.0378	.1707	96.5529
29.00	-.0175	136.77	.0031	39.5752	.1802	97.1958
28.00	-.0175	136.96	.0033	39.1149	.1898	97.8458
27.00	-.0175	137.16	.0035	38.6573	.1995	98.5030
26.00	-.0175	137.37	.0037	38.2023	.2092	99.1671
25.00	-.0175	137.59	.0038	37.7503	.2190	99.8381
24.00	-.0175	137.82	.0040	37.3014	.2290	100.5161
23.00	-.0175	138.06	.0042	36.8557	.2390	101.2008
22.00	-.0175	138.31	.0044	36.4135	.2492	101.8923
21.00	-.0175	138.57	.0045	35.9749	.2595	102.5904
20.00	-.0175	138.84	.0047	35.5401	.2700	103.2952

19.00	-.0175	139.12	.0049	35.1095	.2806	104.0065
18.00	-.0175	139.41	.0051	34.6830	.2914	104.7243
17.00	-.0175	139.71	.0053	34.2610	.3023	105.4487
16.00	-.0175	140.02	.0055	33.8437	.3134	106.1794
15.00	-.0175	140.35	.0057	33.4313	.3247	106.9165
14.00	-.0175	140.68	.0059	33.0240	.3363	107.6601
13.00	-.0175	141.03	.0061	32.6221	.3480	108.4100
12.00	-.0175	141.39	.0063	32.2259	.3600	109.1661
11.00	-.0175	141.76	.0065	31.8355	.3721	109.9287
10.00	-.0175	142.15	.0067	31.4513	.3846	110.6975
9.00	-.0175	142.55	.0069	31.0735	.3974	111.4726
8.00	-.0175	142.96	.0072	30.7025	.4104	112.2541
7.00	-.0175	143.38	.0074	30.3384	.4237	113.0418
6.00	-.0175	143.82	.0076	29.9817	.4374	113.8359
5.00	-.0175	144.27	.0079	29.6327	.4514	114.6364
4.00	-.0175	144.73	.0081	29.2916	.4658	115.4433
3.00	-.0175	145.22	.0084	28.9589	.4806	116.2566
2.00	-.0175	145.71	.0087	28.6348	.4957	117.0764
1.00	-.0175	146.22	.0089	28.3199	.5114	117.9027
.00	-.0175	146.75	.0092	28.0143	.5275	118.7357
-1.00	-.0175	147.29	.0095	27.7187	.5440	119.5753
-2.00	-.0175	147.86	.0098	27.4334	.5611	120.4217
-3.00	-.0175	148.43	.0101	27.1589	.5788	121.2750
-4.00	-.0175	149.03	.0104	26.8957	.5971	122.1354
-5.00	-.0175	149.65	.0107	26.6441	.6159	123.0028
-6.00	-.0175	150.28	.0111	26.4049	.6355	123.8775
-7.00	-.0175	150.94	.0114	26.1785	.6557	124.7597
-8.00	-.0175	151.61	.0118	25.9655	.6768	125.6494
-9.00	-.0175	152.31	.0122	25.7665	.6986	126.5470
-10.00	-.0175	153.03	.0126	25.5822	.7213	127.4526
-11.00	-.0175	153.78	.0130	25.4133	.7449	128.3664
-12.00	-.0175	154.55	.0134	25.2605	.7696	129.2888
-13.00	-.0175	155.34	.0139	25.1246	.7953	130.2200
-14.00	-.0175	156.17	.0143	25.0064	.8222	131.1604
-15.00	-.0175	157.02	.0148	24.9068	.8503	132.1103
-16.00	-.0175	157.90	.0154	24.8267	.8798	133.0701
-17.00	-.0175	158.81	.0159	24.7672	.9107	134.0403
-18.00	-.0175	159.75	.0165	24.7293	.9431	135.0213
-19.00	-.0175	160.73	.0171	24.7142	.9774	136.0138
-20.00	-.0175	161.74	.0177	24.7232	1.0134	137.0183
-21.00	-.0175	162.79	.0184	24.7576	1.0516	138.0355
-22.00	-.0175	163.88	.0191	24.8188	1.0919	139.0661
-23.00	-.0175	165.02	.0198	24.9086	1.1348	140.1112
-24.00	-.0175	166.20	.0206	25.0286	1.1804	141.1716
-25.00	-.0175	167.43	.0215	25.1809	1.2291	142.2485
-26.00	-.0175	168.71	.0224	25.3674	1.2812	143.3431

-27.00	-.0175	170.05	.0233	25.5907	1.3371	144.4569
-28.00	-.0175	171.44	.0244	25.8533	1.3973	145.5916
-29.00	-.0175	172.91	.0255	26.1582	1.4625	146.7491
-30.00	-.0175	174.44	.0268	26.5088	1.5333	147.9318
-31.00	-.0175	176.05	.0281	26.9089	1.6107	149.1423
-32.00	-.0175	177.75	.0296	27.3629	1.6956	150.3839
-33.00	-.0175	179.54	.0312	27.8761	1.7897	151.6604
-34.00	-.0175	181.43	.0331	28.4546	1.8948	152.9768
-35.00	-.0175	183.44	.0351	29.1057	2.0130	154.3387
-36.00	-.0175	185.59	.0375	29.8385	2.1479	155.7538
-37.00	-.0175	187.90	.0402	30.6643	2.3037	157.2317
-38.00	-.0175	190.38	.0434	31.5977	2.4871	158.7853
-39.00	-.0175	193.09	.0473	32.6584	2.7077	160.4324
-40.00	-.0175	196.07	.0520	33.8737	2.9817	162.1987
-41.00	-.0175	199.41	.0582	35.2849	3.3361	164.1236
-42.00	-.0175	203.23	.0667	36.9590	3.8240	166.2735
-43.00	-.0175	207.80	.0797	39.0229	4.5669	168.7765
-44.00	-.0175	213.74	.1037	41.7883	5.9441	171.9552
-45.00	-.0175	225.00	.1965	47.1634	11.2564	177.8366
-44.00	.0175	213.74	-.1965	41.7883	11.2564	171.9552
-43.00	.0175	207.80	-.1037	39.0229	5.9441	168.7765
-42.00	.0175	203.23	-.0797	36.9590	4.5669	166.2735
-41.00	.0175	199.41	-.0667	35.2849	3.8240	164.1236
-40.00	.0175	196.07	-.0582	33.8737	3.3361	162.1987
-39.00	.0175	193.09	-.0520	32.6584	2.9817	160.4324
-38.00	.0175	190.38	-.0473	31.5977	2.7077	158.7853
-37.00	.0175	187.90	-.0434	30.6643	2.4871	157.2317
-36.00	.0175	185.59	-.0402	29.8385	2.3037	155.7538
-35.00	.0175	183.44	-.0375	29.1057	2.1479	154.3387
-34.00	.0175	181.43	-.0351	28.4546	2.0130	152.9768
-33.00	.0175	180.46	-.0169	27.8761	.9679	152.5873
-32.00	.0175	177.75	-.0474	27.3629	2.7166	150.3839
-31.00	.0175	176.05	-.0296	26.9089	1.6956	149.1423
-30.00	.0175	174.44	-.0281	26.5088	1.6107	147.9318
-29.00	.0175	172.91	-.0268	26.1582	1.5333	146.7491
-28.00	.0175	171.44	-.0255	25.8533	1.4625	145.5916
-27.00	.0175	170.05	-.0244	25.5907	1.3973	144.4569
-26.00	.0175	168.71	-.0233	25.3674	1.3371	143.3431
-25.00	.0175	167.43	-.0224	25.1809	1.2812	142.2485
-24.00	.0175	166.20	-.0215	25.0286	1.2291	141.1716
-23.00	.0175	165.02	-.0206	24.9086	1.1804	140.1112
-22.00	.0175	163.88	-.0198	24.8188	1.1348	139.0661
-21.00	.0175	162.79	-.0191	24.7576	1.0919	138.0355
-20.00	.0175	161.74	-.0184	24.7232	1.0516	137.0183
-19.00	.0175	160.73	-.0177	24.7142	1.0134	136.0138
-18.00	.0175	159.75	-.0171	24.7293	.9774	135.0213

-17.00	.0175	158.81	-.0165	24.7672	.9431	134.0403
-16.00	.0175	157.90	-.0159	24.8267	.9107	133.0701
-15.00	.0175	157.02	-.0154	24.9068	.8798	132.1103
-14.00	.0175	156.17	-.0148	25.0064	.8503	131.1604
-13.00	.0175	155.34	-.0143	25.1246	.8222	130.2200
-12.00	.0175	154.55	-.0139	25.2605	.7953	129.2888
-11.00	.0175	153.78	-.0134	25.4133	.7696	128.3664
-10.00	.0175	153.03	-.0130	25.5822	.7449	127.4526
-9.00	.0175	152.31	-.0126	25.7665	.7213	126.5470
-8.00	.0175	151.61	-.0122	25.9655	.6986	125.6494
-7.00	.0175	150.94	-.0118	26.1785	.6768	124.7597
-6.00	.0175	150.28	-.0114	26.4049	.6557	123.8775
-5.00	.0175	149.65	-.0111	26.6441	.6355	123.0028
-4.00	.0175	149.03	-.0107	26.8957	.6159	122.1354
-3.00	.0175	148.43	-.0104	27.1589	.5971	121.2750
-2.00	.0175	147.86	-.0101	27.4334	.5788	120.4217
-1.00	.0175	147.29	-.0098	27.7187	.5611	119.5753
.00	.0175	146.75	-.0095	28.0143	.5440	118.7357
1.00	.0175	146.22	-.0092	28.3199	.5275	117.9027
2.00	.0175	145.71	-.0089	28.6348	.5114	117.0764
3.00	.0175	145.22	-.0087	28.9589	.4957	116.2566
4.00	.0175	144.73	-.0084	29.2916	.4806	115.4433
5.00	.0175	144.27	-.0081	29.6327	.4658	114.6364
6.00	.0175	143.82	-.0079	29.9817	.4514	113.8359
7.00	.0175	143.38	-.0076	30.3384	.4374	113.0418
8.00	.0175	142.96	-.0074	30.7025	.4237	112.2541
9.00	.0175	142.55	-.0072	31.0735	.4104	111.4726
10.00	.0175	142.15	-.0069	31.4513	.3974	110.6975
11.00	.0175	141.76	-.0067	31.8355	.3846	109.9287
12.00	.0175	141.39	-.0065	32.2259	.3721	109.1661
13.00	.0175	141.03	-.0063	32.6221	.3600	108.4100
14.00	.0175	140.68	-.0061	33.0240	.3480	107.6601
15.00	.0175	140.35	-.0059	33.4313	.3363	106.9165
16.00	.0175	140.02	-.0057	33.8437	.3247	106.1794
17.00	.0175	139.71	-.0055	34.2610	.3134	105.4487
18.00	.0175	139.41	-.0053	34.6830	.3023	104.7243
19.00	.0175	139.12	-.0051	35.1095	.2914	104.0065
20.00	.0175	138.84	-.0049	35.5401	.2806	103.2952
21.00	.0175	138.57	-.0047	35.9749	.2700	102.5904
22.00	.0175	138.31	-.0045	36.4135	.2595	101.8923
23.00	.0175	138.06	-.0044	36.8557	.2492	101.2008
24.00	.0175	137.82	-.0042	37.3014	.2390	100.5161
25.00	.0175	137.59	-.0040	37.7503	.2290	99.8381
26.00	.0175	137.37	-.0038	38.2023	.2190	99.1671
27.00	.0175	137.16	-.0037	38.6573	.2092	98.5030
28.00	.0175	136.96	-.0035	39.1149	.1995	97.8458

29.00	.0175	136.77	-.0033	39.5752	.1898	97.1958
30.00	.0175	136.59	-.0031	40.0378	.1802	96.5529
31.00	.0175	136.42	-.0030	40.5027	.1707	95.9174
32.00	.0175	136.26	-.0028	40.9697	.1613	95.2891
33.00	.0175	136.11	-.0027	41.4385	.1519	94.6684
34.00	.0175	135.96	-.0025	41.9092	.1426	94.0551
35.00	.0175	135.83	-.0023	42.3815	.1333	93.4495
36.00	.0175	135.71	-.0022	42.8553	.1241	92.8516
37.00	.0175	135.59	-.0020	43.3304	.1149	92.2616
38.00	.0175	135.49	-.0018	43.8067	.1058	91.6794
39.00	.0175	135.39	-.0017	44.2841	.0967	91.1054
40.00	.0175	135.30	-.0015	44.7624	.0876	90.5395
41.00	.0175	135.22	-.0014	45.2415	.0785	89.9819
42.00	.0175	135.15	-.0012	45.7213	.0694	89.4327
43.00	.0175	135.09	-.0011	46.2016	.0604	88.8920
44.00	.0175	135.04	-.0009	46.6824	.0513	88.3599
45.00	.0175	135.00	-.0007	47.1634	.0423	87.8366

APPENDIX II

NEWTON'S METHOD

Newton's Method which can be summarized as: this method consists of drawing the tangent to the curve at any point and the intersection of the tangent with x - axes is used as the first approximation for finding the real root. Figure II.I shows the use of this method to solve the polynomial equations for a normally closed and a normally open dampers.

From Figure II.I,

$$\tan\Phi = \frac{f(X_1)}{X_1 - X_2} = f'(X_1) \qquad \text{(II.I)}$$

where the $f'(X_1)$ is the derivative of $f(X_1)$.

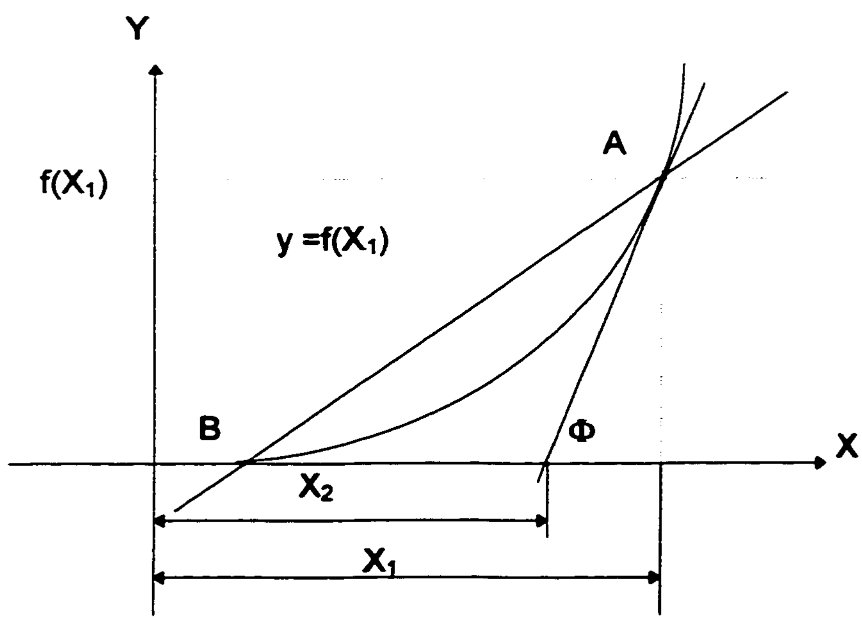

Figure II.I Newton's Method.

$$X_2 = X_1 - \frac{f(X_1)}{f'(X_1)} \quad \text{(II.II)}$$

Equation II.II can be rewritten in general form

$$X_{n+1} = X_n - \frac{f(X_n)}{f'(X_n)} \quad \text{(II.III)}$$

Equation II.II may be iterated untill the real root of point B obtained.

APPENDIX III

SUBROTINES USED TO LINEARIZE THE DAMPER CHARACTERISTICS

III.I The dampers as supplied by the manufacturer

1. For normally closed damper, outdoor air and discharge air dampers.

```
SUB LINODA (SG, SGC)
    SGC = 0
    XB = 0
    I = 0
    EP = 0.00001
    AMAX  = 2.5
    A0 = -0.0566
    A1 =  0.016
    A2 = -3.94E-4
    A3 =  5.22E-6
    A4 = -1.91E-9
    TA = SG * (AMAX / 100)
20   I = I + 1
    Y = -TA + A0 + A1 * SGC + A2 * SGC ^ 2+ A3 * SGC ^3+ A4 * SGC ^4
    YD = A1 + 2 * A2 * SGC +3 * A3 * SGC^2 +4 * A4 * SGC^3
    SGC = SGC - Y / YD
      IF ABS(SGC - XB) >= EP THEN
        XB = SGC
        GOTO 20
      END IF
    IF SG = 0  THEN  SGC  = 0
```

```
    IF SG = 100  THEN  SGC = 100
    SGC = SGC
END SUB
```

2. For normally open damper, recirculation air damper

```
SUB LINRSC (SG, SGC)
    SGC = 0
    XB = 0
    Z = 0
    EP = 0.00001
    AMAX  = 2.5
    A0 = -0.0566
    A1 =  0.016
    A2 = -3.94E-4
    A3 =  5.22E-6
    A4 = -1.91E-9A0 = -0.1401
    TA = SG * (AMAX / 100)
    TAC = AMAX - TA
40   Z = Z + 1
    Y = -TA + A0 + A1 * SGC + A2 * SGC ^ 2+ A3 * SGC ^3+ A4 * SGC ^4
    YD = A1 + 2 * A2 * SGC +3 * A3 * SGC^2 +4 * A4 * SGC^3
    SGC = SGC - Y / YD
      IF ABS(SGC - XB) >= EP THEN
        XB = SGC
        GOTO 40
      END IF
    IF SG = 0 THEN SGC = 100
    IF SG = 100 THEN SGC = 0
    SGC = 100 - SGC
END SUB
```

III.II The modified dampers

1. For normally closed damper, outdoor air and discharge air dampers.

```
SUB LINODA (SG, SGC)
    SGC = 0
    XB = 0
    I = 0
    EP = 0.00001
    AMAX  = 2.6
    A0 = -0.1401
    A1 =  0.01144
    A2 =  0.000195
    TA = SG * (AMAX / 100)
20   I = I + 1
    Y = -TA + A0 + A1 * SGC + A2 * SGC ^ 2
    YD = A1 + 2 * A2 * SGC
    SGC = SGC - Y / YD
      IF ABS(SGC - XB) >= EP THEN
        XB = SGC
        GOTO 20
      END IF
    IF SG = 0  THEN  SGC  = 0
    IF SG = 100  THEN  SGC = 100
    SGC = SGC
END SUB
```

2. For normally open damper, recirculation air damper

```
SUB LINRSC (SG, SGC)
    SGC = 0
```

```
    XB = 0
    Z = 0
    EP = 0.00001
    AMAX  = 2.6
    A0 = -0.1401
    A1 =  0.01144
    A2 =  0.000195
    TA = SG * (AMAX / 100)
    TAC = AMAX - TA
40   Z = Z + 1
    Y = -TAC + A0 + A1 * SGC + A2 * SGC ^ 2
    YD = A1 + 2 * A2 * SGC
    SGC = SGC - Y / YD
      IF ABS(SGC - XB) >= EP THEN
        XB = SGC
        GOTO 40
      END IF
    IF SG = 0 THEN SGC = 100
    IF SG = 100 THEN SGC = 0
    SGC = 100 - SGC
END SUB
```