November 8, 1973

To the Graduate Council:

I am submitting herewith a thesis written by George Franklin Pierce entitled "Static Modeling of Coal Pulverizer Systems at Bull Run Steam Plant." I recommend that it be accepted for nine quarter hours of credit in partial fulfillment of the requirements for the degree of Master of Science, with a major in Electrical Engineering.

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STATIC MODELING OF COAL PULVERIZER SYSTEMS AT BULL RUN STEAM PLANT

A Thesis

Presented to

the Graduate Council of

The University of Tennessee

In Partial Fulfillment

of the Requirements for the Degree

Master of Science

---

by

George Franklin Pierce

December 1973

ACKNOWLEDGEMENTS

I am deeply indebted and would like to express my sincere gratitude and appreciation to:

Dr. J. M. Bailey and Dr. J. M. Googe for directing the topic of study, and for guidance and encouragement;

Frederick Hebron and other personnel at Bull Run Steam Plant for their help and suggestions;

My wife for her patience and encouragement.

Finally, I owe a debt of gratitude to the Department of Electrical Engineering and the Tennessee Valley Authority without which this research could not have been done.

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ABSTRACT

The purpose of this study is to develop a model for the air and fuel flow in a pulverizer system at Bull Run Steam Plant. The physical system is described and mathematical models of the elements in the system are developed. A model is developed which describes the operation of the system. Methods of optimization are applied to the model and the results of this optimization are compared to the current operation of the system.

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CHAPTER I

INTRODUCTION

In this thesis a model for the air and fuel flow in the pulverizer system is developed and methods of optimization are applied to the model. This model was developed for several reasons. First, the capacity of the pulverizers has often been a problem at Bull Run Steam Plant under the adverse operating conditions of high moisture and low quality coal. A previous consideration of this problem by TVA resulted in the addition of equipment to the system which aids operation under these adverse operating conditions. The development of a model of this type can help identify possible areas for improving the operation of the pulverizer system. Second, there has been a gradual trend toward lower quality coal at Bull Run Steam Plant. Usage of this lower caloric value coal has resulted from both economic and environmental factors. The net result of this trend is that the rate of coal processing must be increased in order to maintain the same unit load. Finally, the development of a model and the optimization of a model is an interesting problem to consider from a control system approach and is a useful technique to apply to a modern steam plant.

A great deal of industrial experience has been accumulated in pulverizer design and operation. Development of pulverized coal firing began in 1880 and the first central station usage of pulverized firing was in 1920. Development of this process has attracted the attention of many engineers from these early developments to the present time.

The design, development, and operation of coal pulverizers has largely been an emperical art. This is because the variables involved are difficult to measure, and because the cause and effect relationships are difficult to account for analytically. However, the results of efforts made in pulverizer operation have been excellent. It is not the goal of this thesis to make any great contributions in pulverizer design. However, it is hoped that the model of the pulverizer system will explain the basic principles involved in pulverizer operation so that the results of optimization of the model may be clearly understood.

In the steam plant, the pulverizer system is part of the auxiliary equipment for the boiler, and the pulverized coal produced in the pulverizers supplies the fuel for combustion and thus the heat for steam generation. The pulverizer system performs several functions. First, the pulverizer with its classifier and exhauster supplies coal to the furnace in the form of a fine powder. Second, the pulverizer is the place where the coal is dried by a heat exchange process with the primary air. Finally, the pulverizers create uniform and favorable combustion conditions.

There are many parameters which influence the ability of the pulverizer to perform these functions. In order to analyze the pulverizer system, it is first necessary to identify the parameters which influence its operation. A general exploration of the factors influencing pulverizer operation will be undertaken in the remainder of this chapter.

Figure 1-1 is a diagram showing the input and output variables

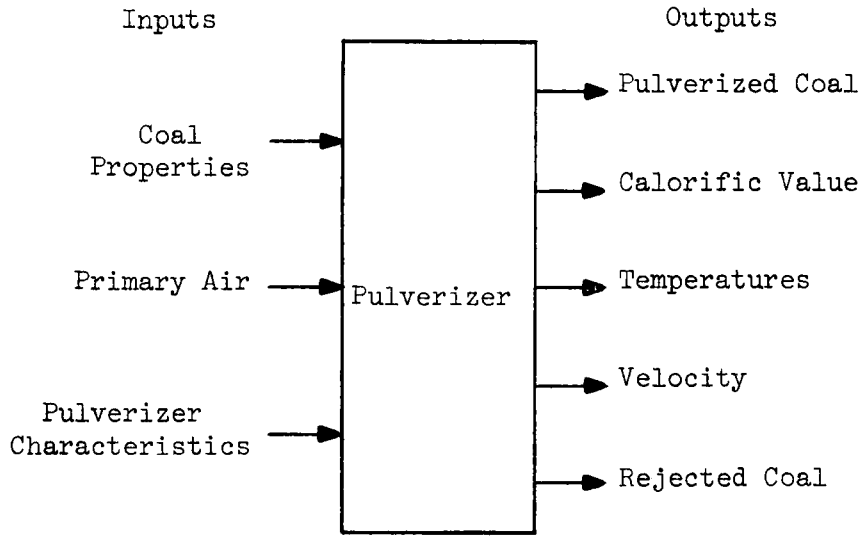


Figure 1-1

Pulverizer Block Diagram

which must be considered when analyzing the pulverizer. The purpose of the development of a model is to relate the important cause and effects shown in this diagram. Once these are obtained, a study of optimum performance is possible.

The properties of the coal fed into the pulverizer have an influence on the pulverizer operation. Table I shows the important properties of coal used at Bull Run Steam Plant. Both design and values and typical values are shown. Other properties of the coal which are important to pulverizer operation are feed rate, grindability, and temperature.

The primary air is important for drying and transport so its important properties are flow rate and temperature. Also, pulverizer characteristics, such as the condition of the bowl and rolls and the exhauster blades have an important influence on pulverizer operation.

The output variables which are a measure of pulverizer performance under the input constraints are shown as outputs in Figure 1-1. The size of the pulverized coal particles is a major concern. Design specifications for Bull Run's pulverizers are shown in Table II. The temperature, velocity, and calorific value of the coal-air mixture as it leaves the pulverizer are other output variables which are important operating considerations.

With this as an introduction to the pulverizer system, a clear statement of the objectives of this report is in order. One goal is to apply the techniques of modern system analysis to the coal pulverizer system at Bull Run Steam Plant. The system will be modeled as shown in Figure 1-2. Variables will be classified as inputs, disturbances, and

Table I

Coal Analysis - As Received

Design	Property	Jan. 1973
12,800	BTU/pound	11,000
46.5%	Fixed Carbon	44.1%
33.5%	Voliatle Matter	30.5%
7.0%	Moisture	9.0%
13.0%	Ash	16.4%
2.8%	Sulfur	1.0%

Table II

Pulverized Coal Sizing Specifications

Size	Percent Passing
200 Mesh	70
50 Mesh	98

Rated Capacity of Pulverizer 86,000 lb./hr.

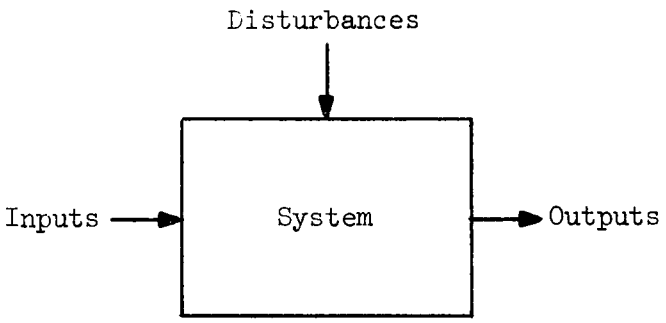


Figure 1-2

Pulverizer System Model

outputs.

The approach that will be used is shown in Figure 1-3. The pulverizer system is the physical system shown here. The conceptual model and mathematical model will be developed from engineering principles and measurements of variables in the system. Finally, a comparison of the predicted system behavior and the measured system behavior will be used to adjust the model.

Another goal is to use this model to examine the effects of changes in the control and disturbance variables on the performance of the system. This will be done by developing a cost function for performance of the system and optimizing it.

Finally, another goal is to identify problem areas in the system operation which could be improved if the cost of improvement were justifiable.

In the next chapter, the physical system will be discussed and individual parts of this system will be modeled.

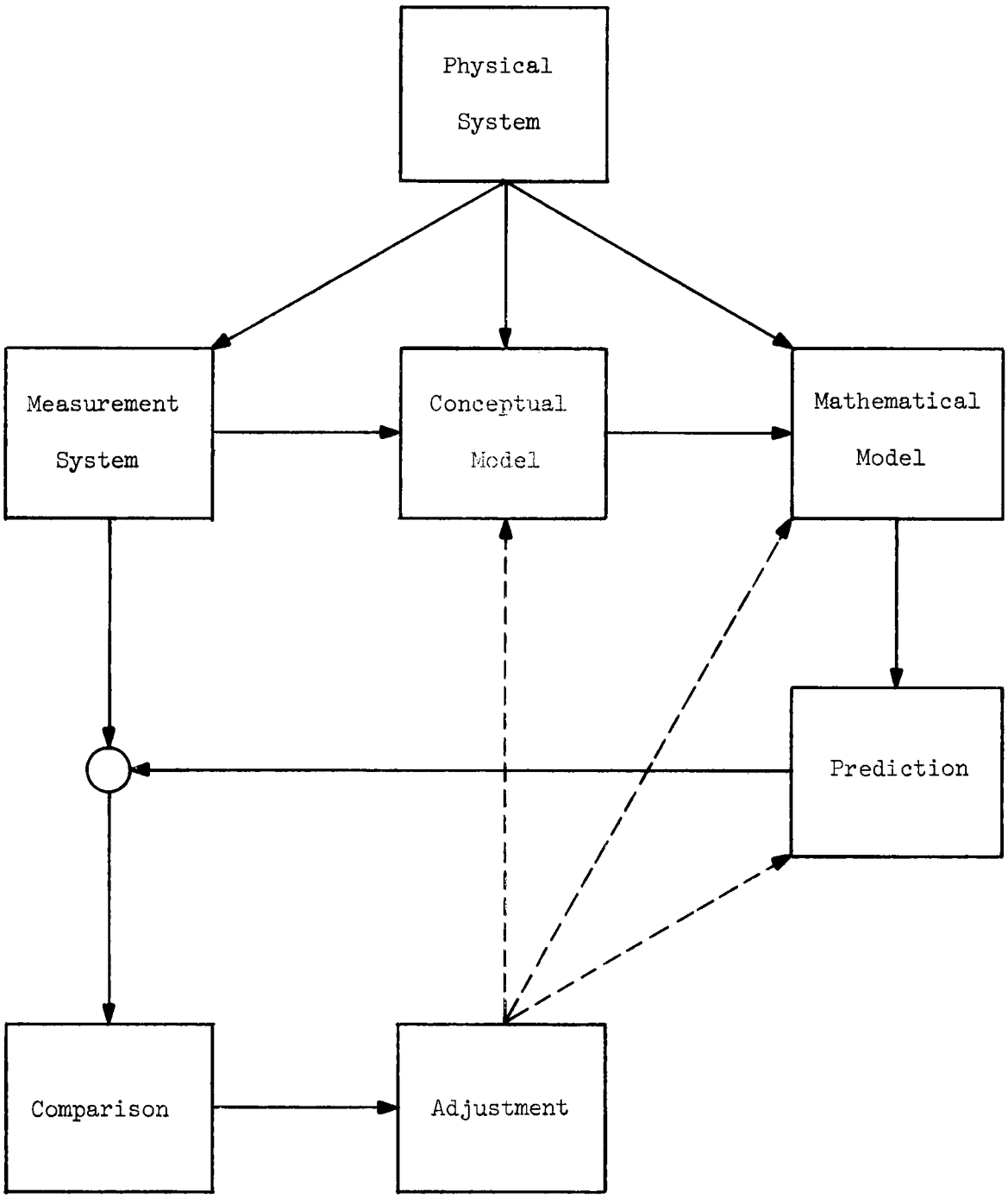


Figure 1-3

Modeling Process

CHAPTER II

DESCRIPTION OF THE PHYSICAL SYSTEM

The first step in the optimization problem is to formulate a model which describes the behavior of the system. However, before this can be done, it is necessary to have a complete understanding of the process under study, and it is important to determine all of the variables which influence the operation of the system. A schematic of the physical system is shown in Figure 2-1. The major elements shown in this diagram are discussed in the following material.

The steam generator at the Bull Run Steam Plant has twin divided furnaces, each of which is equipped with five elevations of burner nozzels. Figure 2-1 shows only one of these furnaces. An equivalent model for the air and fuel flow systems to only one furnace will be considered since the air and fuel flow systems for each furnace are isolated.

The combustion air for the furnaces is collected from the top elevations of the boiler building. This arrangement provides some preliminary heating to the combustion air from heat lost from the furnace walls. This air is drawn through air ducts by two Buffalo Forge centrifugal fans on each furnace. These fans operate in parallel and their outputs are equalized in the analog control system. The output of each fan is controlled by changing the hydraulic coupling provided by the American Standard Fluid Drive Units. The power for each drive unit is provided by a General Electric induction motor.

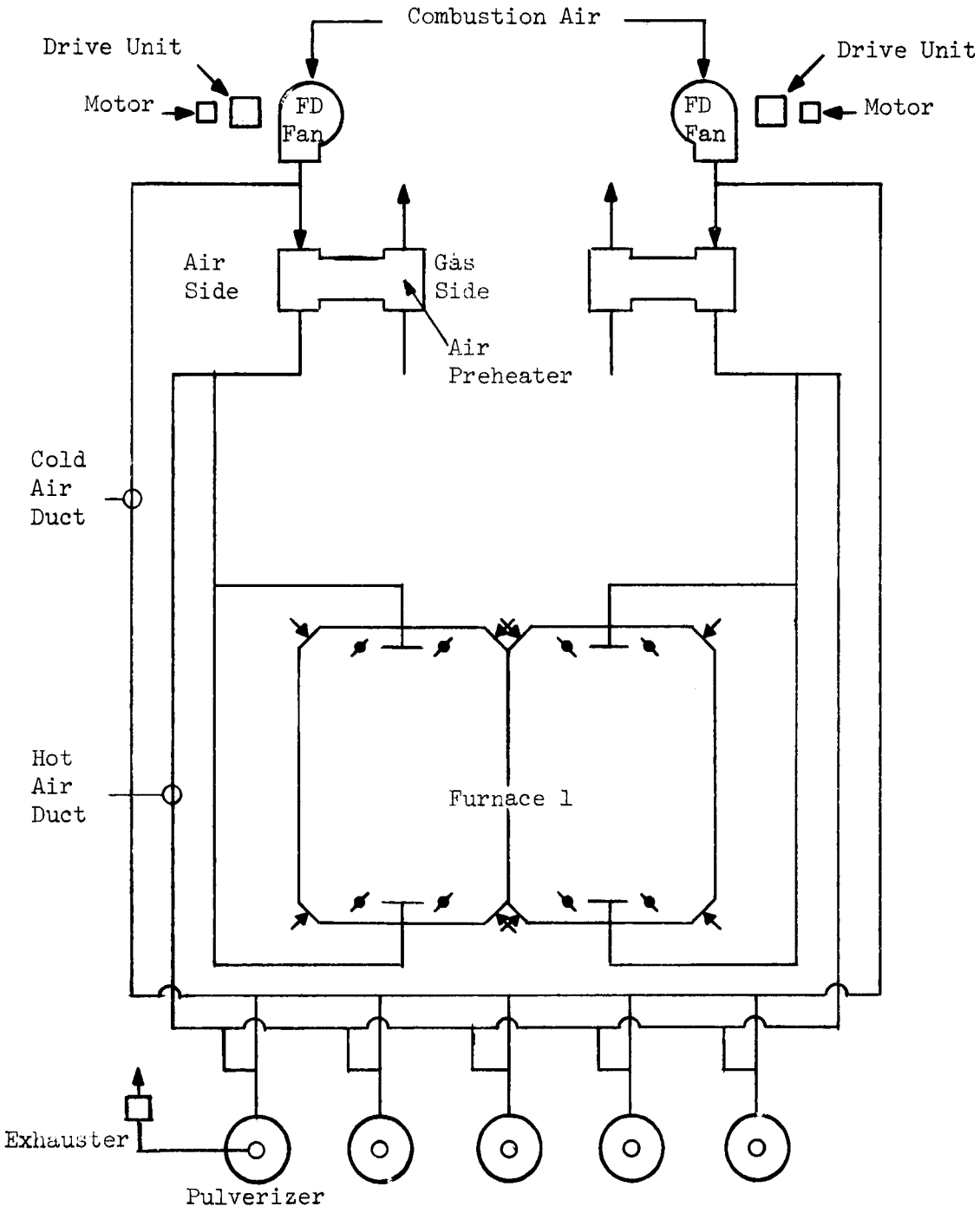


Figure 2-1

Physical System

The purpose of the forced draft fans is to move the combustion air at the required volume flow rate and to supply the total pressure loss. A complete understanding of the fans is important in order to understand the operation of the system. Figure 2-2 is the performance characteristic curves supplied by Buffalo Forge for these fans. Information about the effect of charges in fan operation can be obtained from these curves. Curve A is one of a set of operating curves giving total pressure developed by the fan as a function of static pressure and volume air flow. Curve B represents the system resistance seen by the fan as a function of volume air flow and static pressure. The intersection of these two curves determines the static operating point of the fan. Curve C relates fan speed and volume air flow. Curve D is the relationship between power required by the fan and volume air flow. Each of these relationships will be developed in equation form so they can be used in the model of the system.

The speed versus flow characteristic as supplied by Buffalo Forge has a linear relationship as shown in Figure 2-3. From this curve, the flow, Q, is

$$Q = K_q n \tag{2-1}$$

where K_q is the fan flow coefficient and n is the fan speed. K_q can be evaluated from the curve and for each fan the equation is

$$Q = 467 \times n \frac{\text{cu.ft.}}{\text{min.}} \tag{2-2}$$

The mass flow rate is more useful in this application and it is

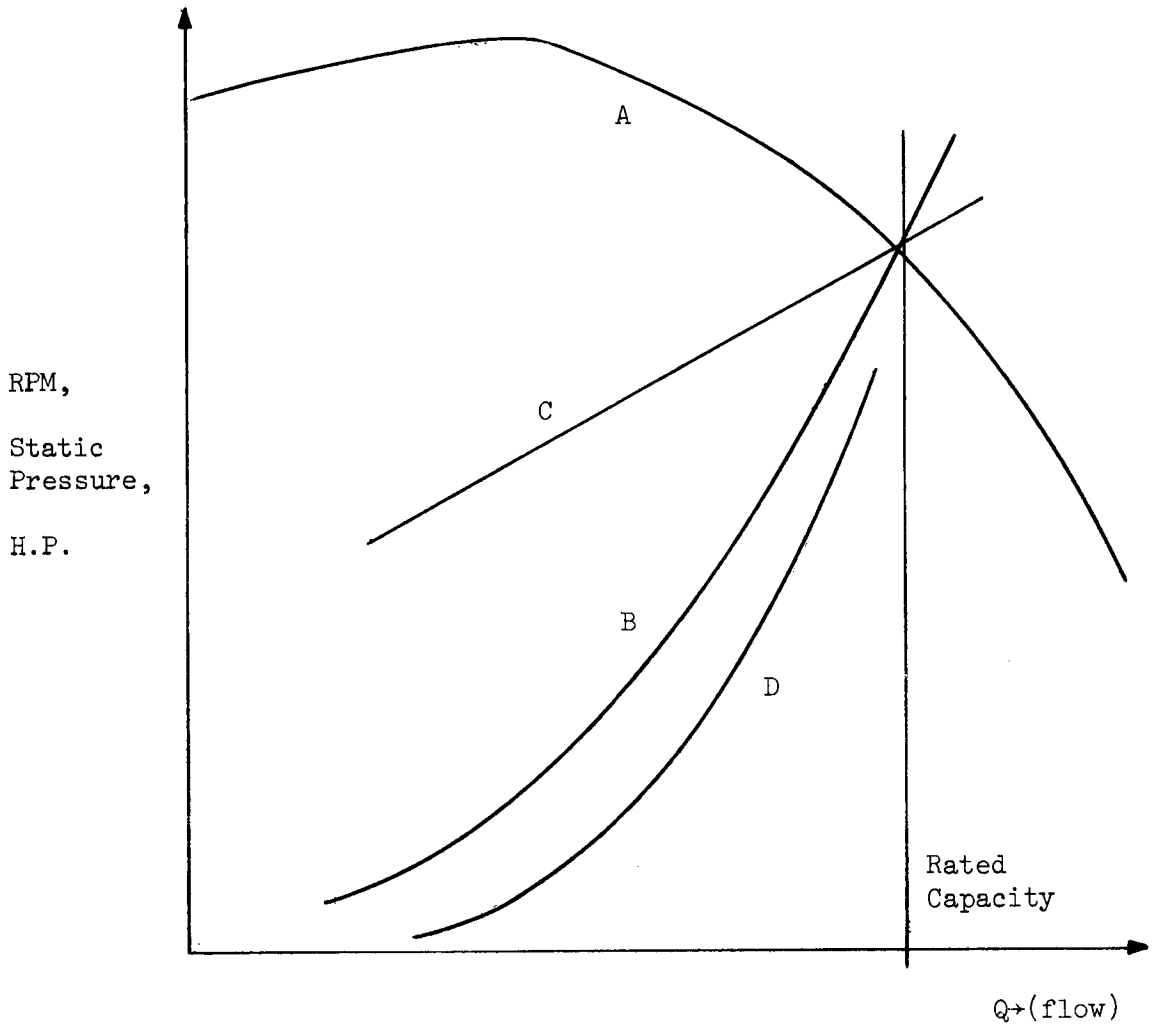


Figure 2-2

Fan Characteristics

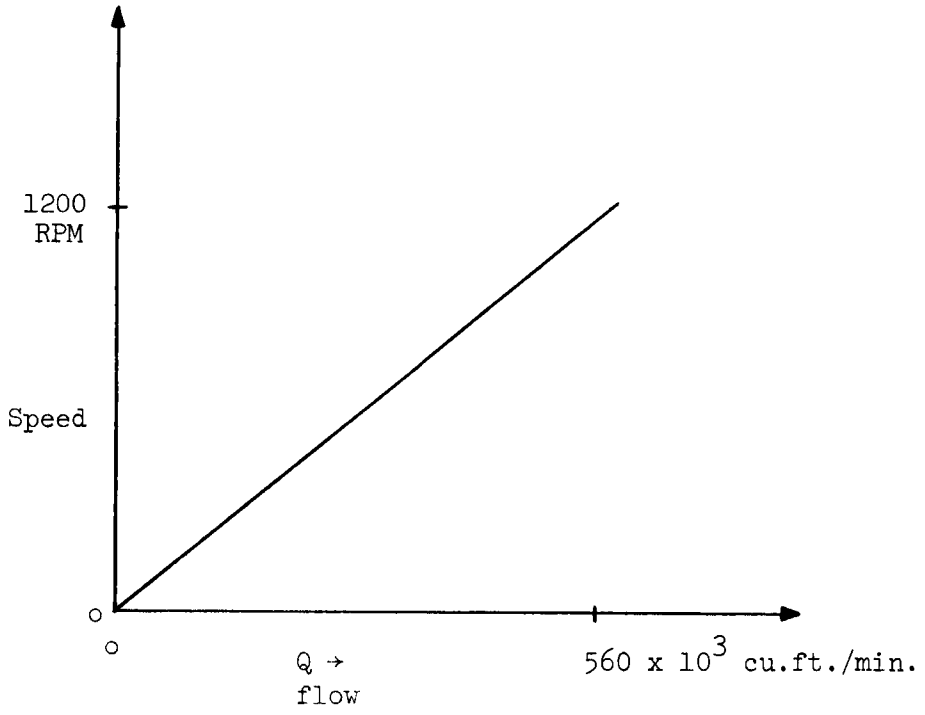


Figure 2-3

Speed vs. Flow Characteristics

directly proportional to density.

This flow is

$$Q = C_q \cdot n \cdot \rho_o \cdot \frac{T_o}{T_o + \Delta T} \quad \text{lb./min.} \tag{2-3}$$

where ρ_o is the density at a temperature T_o and ΔT is positive when the temperature is higher than T_o.

The relationship between the fan pressure and volume flow is also given by the fan characteristic curves. This relationship is shown in Figure 2-4. In this graph, system pressure, which is the sum of velocity pressure and static pressure, is plotted as a function of static pressure and flow.

Another fan law defining the relationship between system pressure loss and volume flow is

$$\Delta p \propto Q^2. \tag{2-4}$$

If this relationship is plotted on the fan characteristic fan pressure for the system being studied, the intersection of the two curves is the operating point for the fan as is shown for a typical case in Figure 2-5.

The fan power may also be found on the manufacturer's characteristics or from another fan law which is

$$P = K_p n^3, \tag{2-5}$$

where P is the fan input power and K_p is the fan power coefficient. This is shown in Figure 2-6.

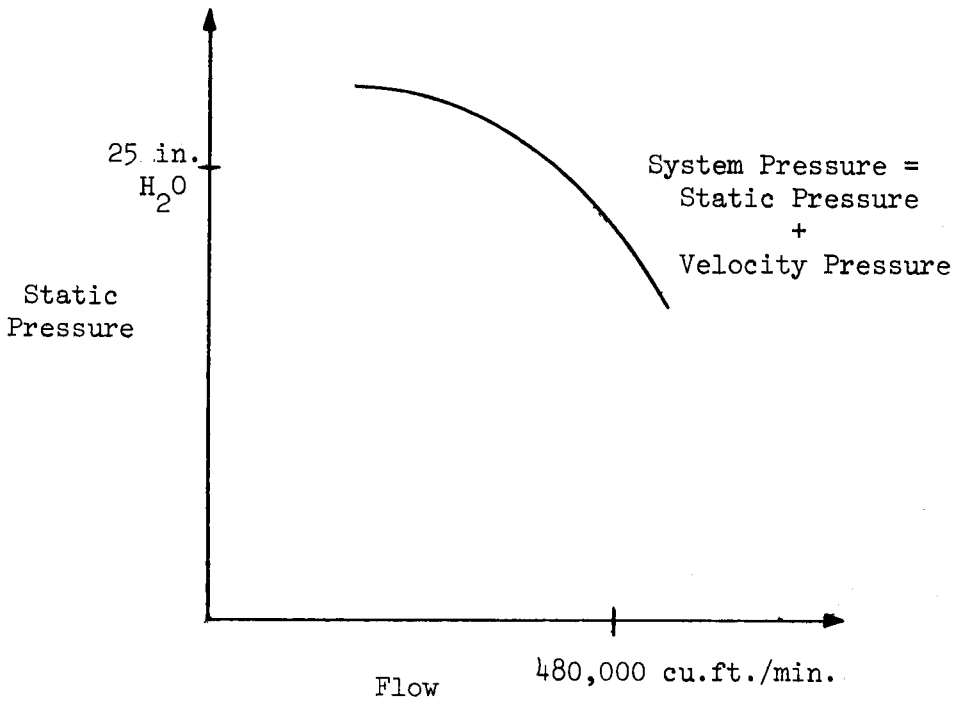


Figure 2-4

Operating Characteristic Curve

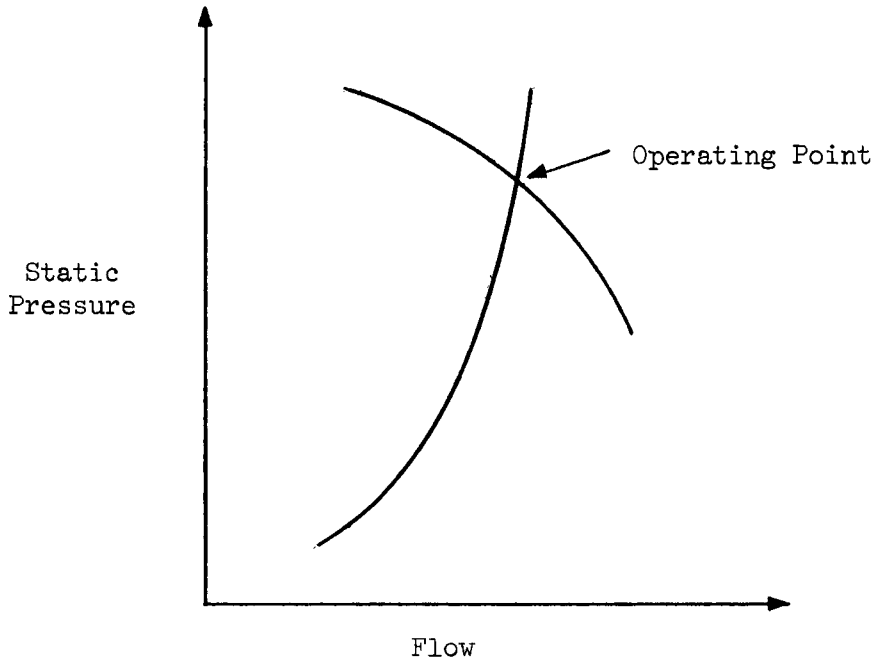


Figure 2-5

Intersection of Fan Characteristic
Curve and System Pressure Curve

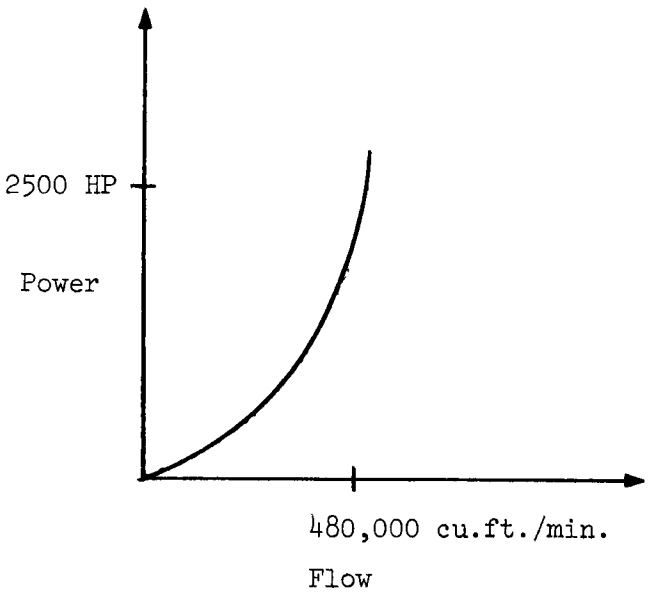


Figure 2-6

Fan Power Consumption

These three relationships provide the mathematical basis for a description of important variables related to the forced draft fans and are

$$\left.\begin{aligned} Q &= K_q n \\ H &= K_h n^2 \\ P &= K_p n^3 \end{aligned}\right\} \quad (2\text{-}6)$$

A single equivalent fan for the model will be derived from the two parallel fans. Two fans connected in parallel have the same total pressure difference from the common connection points downstream to the common connection points upstream. The total volume flow of the equivalent fan will be the sum of the two volumes flowing through each fan at the same effective total fan pressures. The forced draft fans at Bull Run Steam Plant are essentially identical; thus, the characteristic for the equivalent fan is constructed as shown in Figure 2-7.

The control system subloop which controls air flow to the furnace is also of interest in understanding the operation of this system. A block diagram of this control loop is shown in Figure 2-8. A computed flow demand signal is compared to a measured flow. The resultant error signal changes the hydraulic coupling between the forced draft fan and motor. This change in coupling is used as a feedback signal to stabilize the control.

On the discharge side of the fan, the air flow is split. Part of the air flow goes to the pulverizers as primary cold air through the cold air ducts. The remainder of the air flow goes through the air

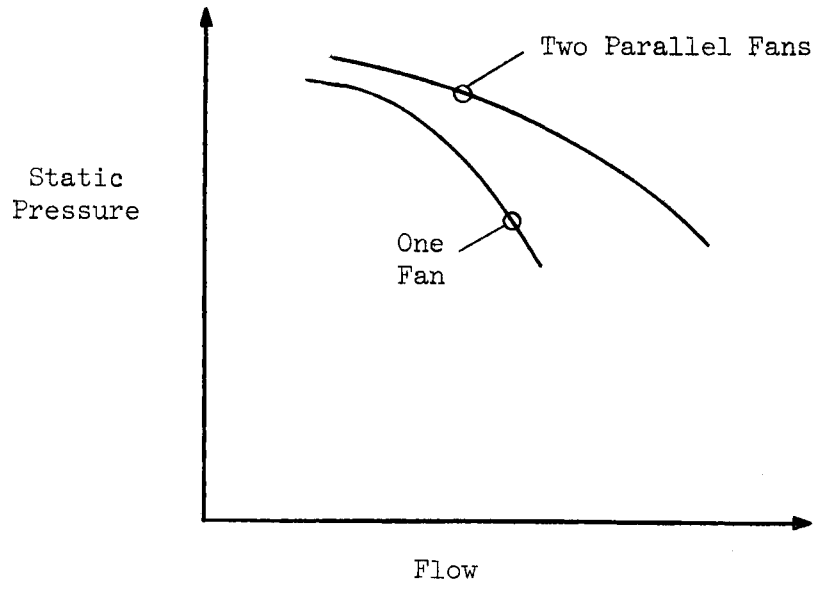


Figure 2-7

Parallel Fan Operation

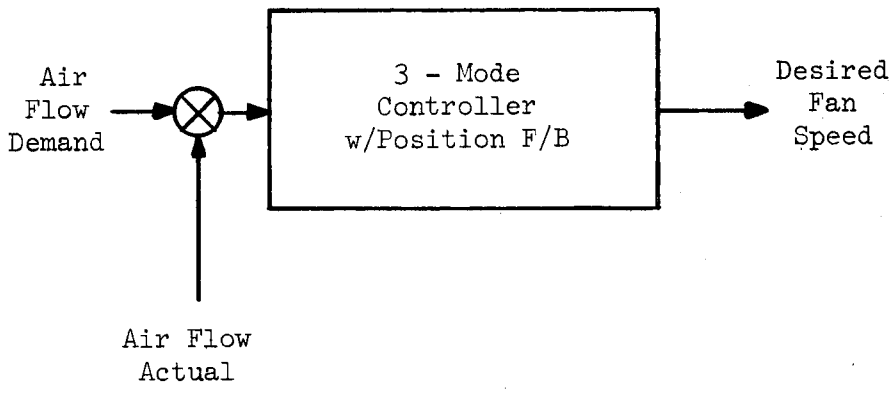


Figure 2-8

Air Flow Controller

side of the Jungstrom air preheater. The air preheater in the air flow system raises the temperature of the air from approximately 100°F to 520°F. The air preheater is a revolving basket type whose elements are heated in the exit flue gases from the furnace and cooled by the entering combustion air.

The air is preheated so that the primary air will have more heat content. This gives the primary air more drying capacity and raises the fuel temperature closer to the ignition temperature.

As the air is heated there is a reduction in its density. The pressure loss in a heat exchanger with a linear temperature gradient based on the air volume flow entering is

$$\Delta p = \Delta p_o \times \frac{T + T_o}{2T_o} , \qquad (2\text{-}7)$$

where T_o is the entering temperature, T is the exit temperature, and Δp_o is the pressure drop with no heating effects. This air flow is divided again on the discharge side of the air preheaters. Part of this flow goes to the pulverizer as primary hot air through the hot air ducts and the remainder enters the furnace as secondary air through the windbox at the coal burner nozzles. Note that the part of the air flow which enters the furnace through the pulverizers is called primary air and the part which enters the furnace directly through the windbox dampers is called secondary air. The primary air comprises 15%-20% of the total combustion air and serves as the transporting medium and drying medium for the pulverized fuel. The remaining 80%-85% of the total combustion air is secondary air. The flow of secondary air through the

windbox provide a mixing action to the burning coal which maximizes the completeness of combustion in the furnace and minimizes the amount of excess air required to maintain a stable flame in the furnace (excess O_2 is the percent of O_2 in the fuel gases). Control of the secondary air is maintained by an analog controller which maintains the differential pressure across the auxiliary air dampers in the windbox at a set point.

The hot and cold parts of the primary air are mixed prior to entering each pulverizer. The flow of primary air is controlled by a hot air damper and a cold air damper for each pulverizer. The basic flow equation for this type of valve is

$$Q = K \sqrt{\Delta p} \quad , \tag{2-8}$$

where Q is the flow rate and Δp is the pressure drop across the valve. Each cold air damper position is regulated in proportion to the volume of coal entering the corresponding pulverizer. This is shown in Figure 2-9. The position of each hot air damper is regulated to maintain a pulverizer temperature of approximately $180^{\circ}F$. Thus a heat balance is maintained on the materials entering each pulverizer. It is important that this heat balance is maintained. If the temperature becomes too high, pulverizer fires and explosions may result. If the temperature becomes too low, the pulverized coal is not adequately dried and it sticks together and burns poorly.

A schematic of a pulverizer is shown in Figure 2-10. The ten pulverizers at Bull Run Steam Plant are Raymond Bowl Mills. The coal which enters the pulverizer has already been sized in a coal breaker

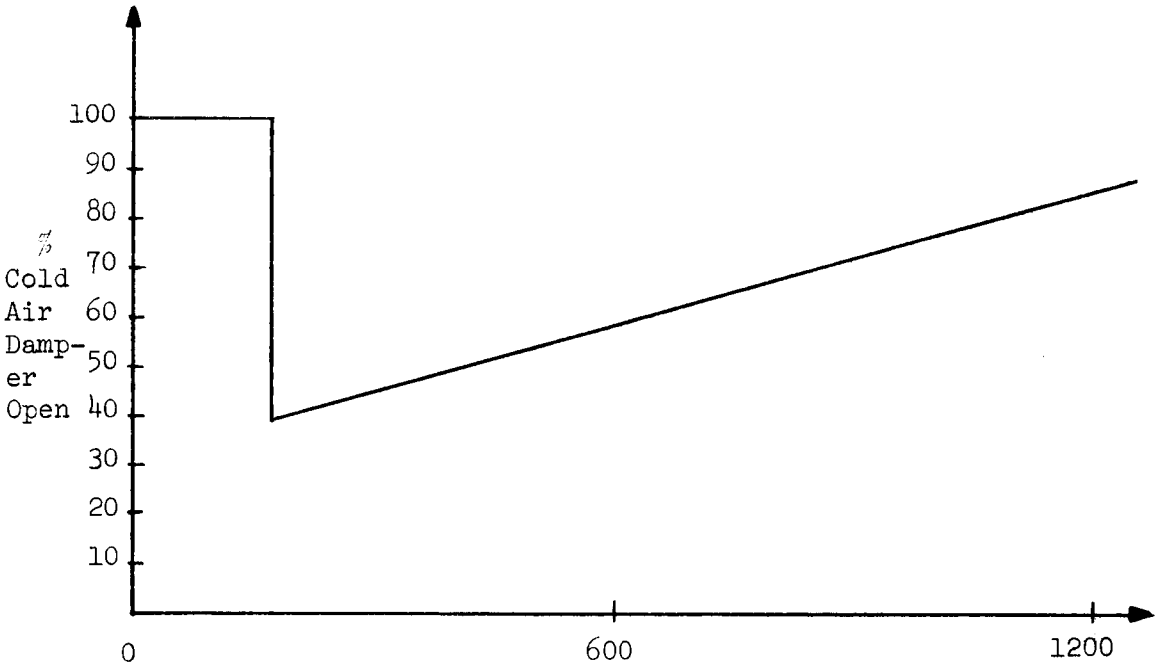


Figure 2-9

Cold Air Damper Programed to Feeder Speed

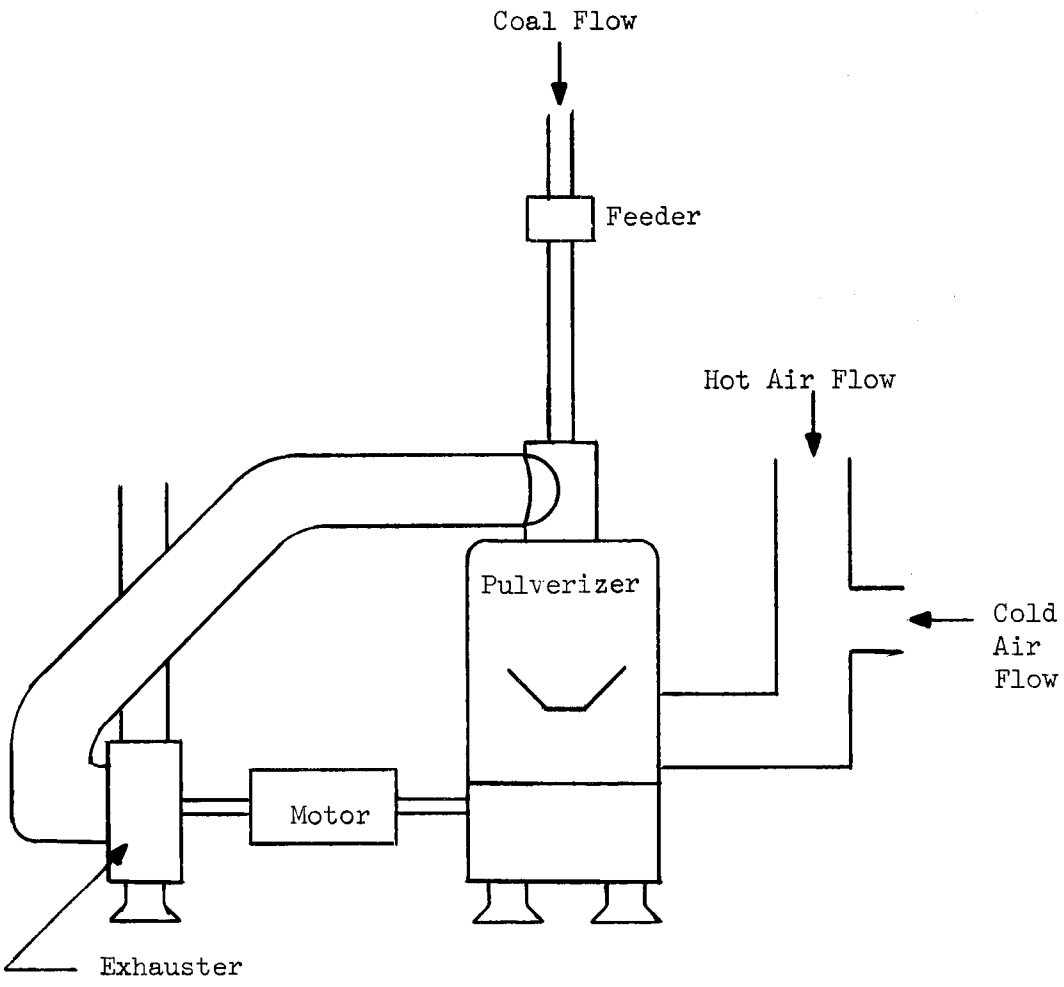


Figure 2-10

Coal Pulverizer

to a uniform size range--3/4" to 1 1/2". The coal is stored in a bunker above the pulverizers and is fed at a controlled rate to each pulverizer by separate volumetric feeders. The flow of coal is set by the firing rate demand computed by the analog control system. The coal enters the pulverizers from top through a coal pipe and falls onto the pulverizer bowl. The bowl is a large dish shaped platform which is turned by an induction motor through a worn gear reduction. Then centrifugal force from the rotating bowl throws the coal toward the outside edge of the bowl. This forces the coal into the region where it is pulverized between the bowl and the pulverizer rolls. The rolls are spring loaded crushers which are shaped like truncated cones.

The pulverized coal which reaches the outside edge of the bowl is picked up by this primary air as it enters the pulverizer from the base. The coal-air mixture flows upward due to the head developed at the forced draft fans as well as the suction developed by the exhauster fan at the discharge side of the pulverizer. The pulverized coal passes through a classifier at the top of the pulverizer. The classifiers have variable vane setting and are set to maintain pulverized coal which will be approximately 70% smaller than 200 Mesh and only 1% or less 50 Mesh or greater over as wide an operating range as is possible.

In order to understand the process more clearly a flow diagram of the process is shown in Figure 2-11. The important variables for the model to be developed in Chapter 3 are shown on the diagram. The combustion air enters the flow diagram at A. The measured variable are

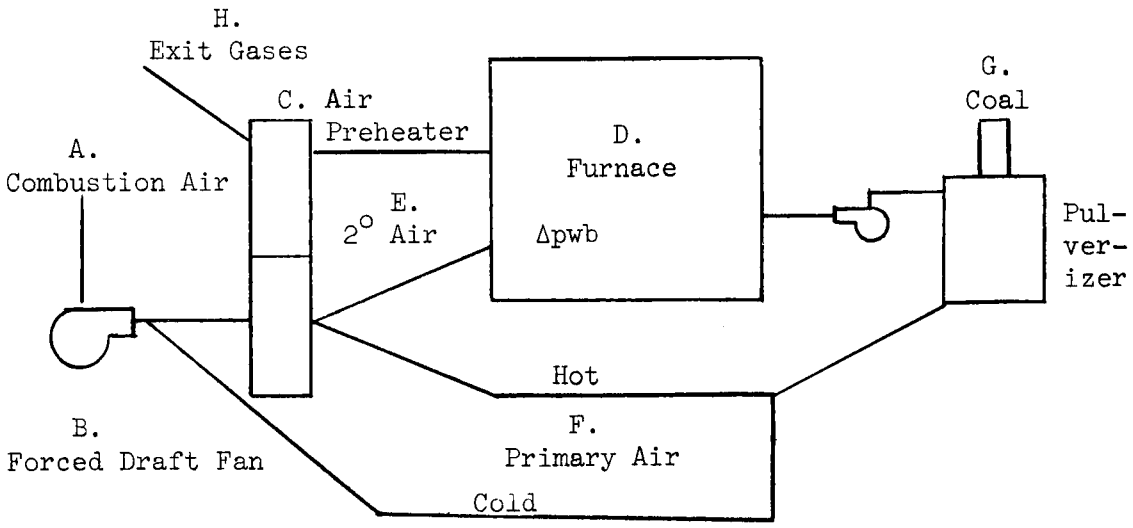


Figure 2-11

Flow Diagram

temperature and flow. The temperature of the combustion air is measured near the point where it enters the air ducts. A change in temperature causes a corresponding change in density which directly affects the process. Air flow is measured by modified venturi sections in the air ducts. The signal developed by this measurement is used to control the forced draft fans. At B, the air flow passes through the forced draft fans. The measured variable here is speed which is used to compute flow, system pressure and fan power consumption. Important variables at C, the air preheater, are entering and exiting temperatures on the gas and air sides. The load demand on the furnace at D determines the required air and coal flow. The temperature and flow rate are the important variables at E, the secondary air. Similarly, the temperatures and flows of primary air are the measured variables at F. The coal enters the pulverizer at G. The measured variable here is flow rate. Other important variables are calorie content, moisture content and temperature. The measured variable at H, the exit gases, are percent excess oxygen and combustible losses.

This chapter has presented a brief description of the elements to be included in the model and the mathematics which describe them. In the next chapter the system model will be formulated and its operation discussed.

CHAPTER III

DEVELOPMENT OF A STATIC MODEL

In order to develop a model, the pulverizer is approached as a system with variables categorized as controls, disturbances, and outputs as shown in Figure 3-1. A necessary step in system analysis is the identification of the variables so that the properties of the system may be examined. Consequently, the first problem that must be solved is the analysis problem as shown in Figure 3-2. The goal here is to derive the mathematical relationships which describe the pulverizer system. If properly derived, this model will compute the state variables or outputs when given measured values of the control and disturbance variables. This mathematical description forms the model of the physical system.

When the model is defined so that the behavior of the real system can be predicted accurately, the information in the model can be used in a control and optimization strategy using the techniques of modern system analysis. This control and optimization problem is illustrated in Figure 3-3. The control variables, the disturbance variables, and the state variables are measured in the pulverizer system. Then the control variables are modified by a computational routine that optimizes some chosen criteria.

Before proceeding to the modeling of the pulverizer system, it will be useful to study the operation of the pulverizer and the variables that influence its operation. A schematic of a typical coal

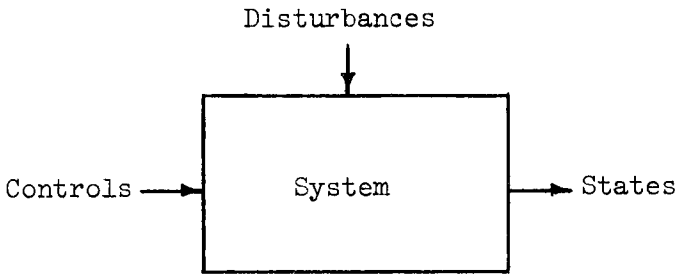


Figure 3-1

Application of Modern System Analysis
to Coal Pulverizer Systems

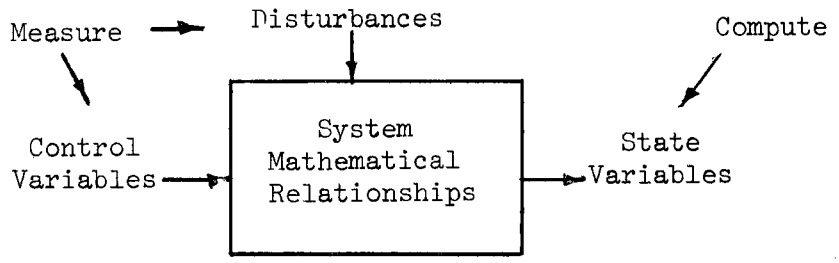


Figure 3-2

The Analysis Problem

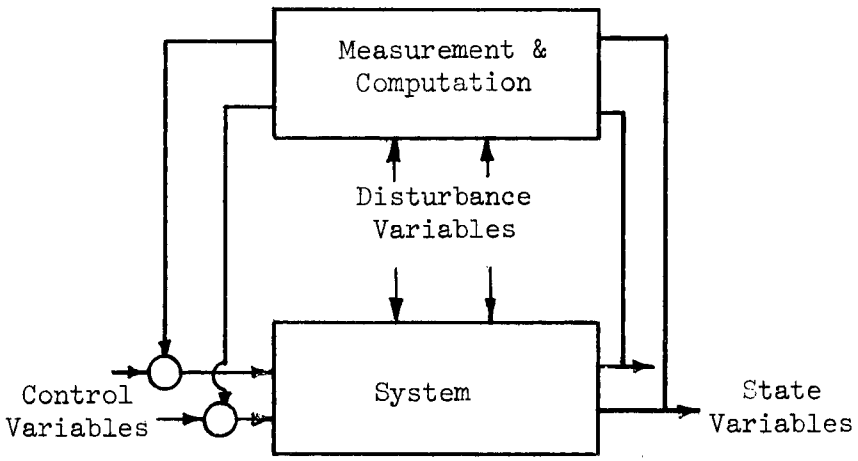


Figure 3-3

The Control and Optimization Problem

pulverizer showing the important processes involved is shown in Figure 3-4.

Under normal operating conditions, the coal is fed to the pulverizer from the top by a volumetric feeder. The coal then falls onto the revolving ring where it is crushed between the ring and rolls. Primary air enters from the bottom of the pulverizer and flows around the edge of the ring. The primary air is composed of hot air and cold air which mix at the base of the pulverizer. Each air flow is controlled by a separate damper. Pulverizer temperature is measured at the top of the pulverizer near the point where the pulverized fuel exits from the pulverizer. The exhauster is a fan which boosts the pulverized fuel and primary air into the furnace.

It will also be useful to view the overall system in order to understand the processes which influence the operation of the pulverizer. Figure 3-5 is a flow diagram of the overall system of which the pulverizer is a subsystem. This diagram shows the major equipment in the air and fuel flow system for the furnace. These elements are the forced draft fan, the air preheaters, the furnace, the pulverizers, the exhauster fans, and the ducts between this equipment. Important variables for the model are illustrated on this diagram.

The variables to be used for the model are shown in Figure 3-6. The categories of control, disturbance, and state variables are shown so that the model can be developed as illustrated in Figure 3-1. The control variables are chosen to conform to practical realization. For example the damper positions, θc and θ_h, are the most direct method for controlling the primary air flow. The windbox differential

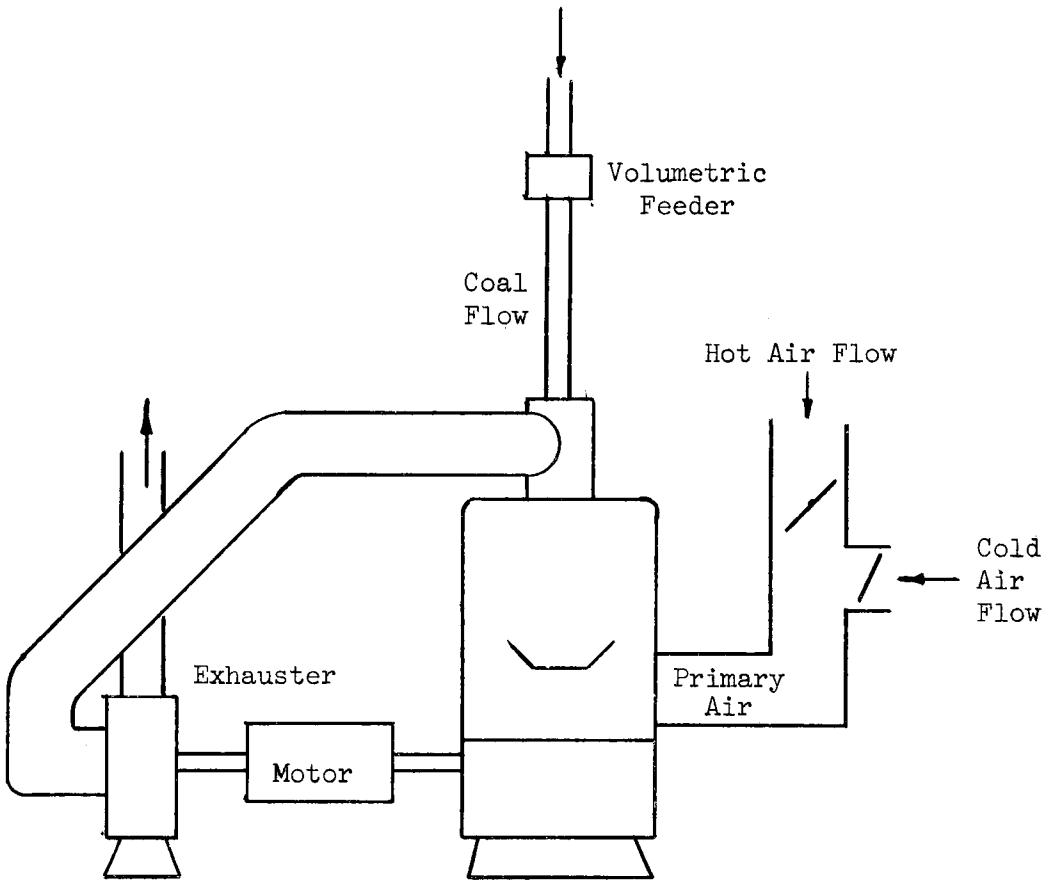


Figure 3-4

Measurement & Analysis on a Coal Pulverizer

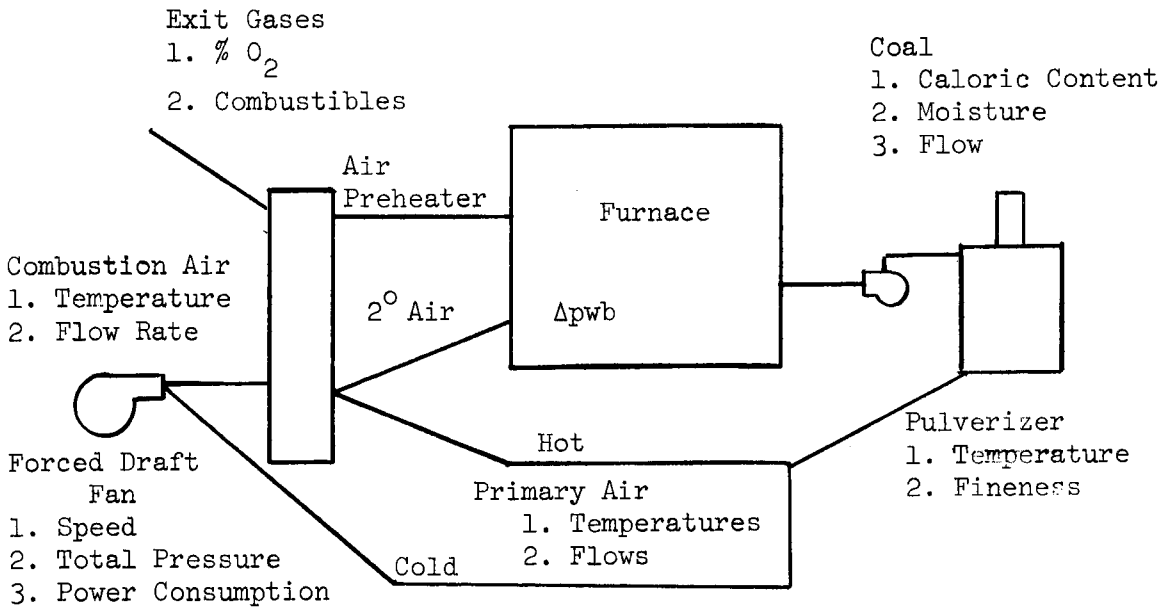


Figure 3-5

Flow Diagram Showing Important Variables

Disturbance Variables	Control Variables
T_c - Cold Air Temperature	θ_c - Cold Air Damper
T_h - Hot Air Temperature	θ_h - Hot Air Damper
P_B - Base Pressure	Δpwb - Windbox Damper Differential
T_F - Fuel Temperature	n - F.D. fan speed
M - Moisture Content of Coal	
R - Residual Moisture	

State Variables	Constraints
Q_c - Cold Air Flow	$160^{\circ} \leq T_m \leq 180^{\circ}F$
Q_h - Hot Air Flow	$0\% \leq \theta_c \leq 100\%$
T_m - Pulverizer Temperature	$0\% \leq \theta_h \leq 100\%$
	$4 \leq \Delta pwb \leq 8$
	$0 < n < 1173$

Figure 3-6

Variables for System Model

pressure controls the proportioning of the secondary air to the primary air and the fan speed controls the total amount of combustion air as illustrated in Figure 3-5. The state variables are chosen to conform to outputs of the model. Cold air flow and hot air flow to each mill combine to make the total primary air flow. The disturbance variables whose values influence the systems operation. The constraints indicated on Figure 3-6 are operating constraints and physical limitations.

The analog diagram shown in Figure 3-7 was developed from the flow diagram. This diagram contains an equivalent fan for each pulverizer, a source of coal flow, the exhauster fan, damper resistances, and duct losses. The purpose of developing this analog diagram was to diagram the process for the development of the system mathematics.

In the analog model, current and air flow are analogs as are voltage and total pressure. The driving force of the system consists of two parallel centrifugal forced draft fans. In the electrical analog model these fans are represented by a current source whose output is proportional to fan speed. Fan speed is controlled in the physical system by an analog control loop utilizing excess oxygen and furnace firing rates as commands. Duct losses are represented in the model by resistances whose values depend on the flow through the ducts. The cold air dampers, hot air dampers, and windbox dampers are all louver types with typical non-linear characteristics and are represented by non-linear resistors in the model. The air preheater is represented by a resistance whose value depends upon restriction and temperature effects. The pulverizer has a restrictive orifice in the path of the primary air flow and is represented by a resistance. The exhauster is represented by a

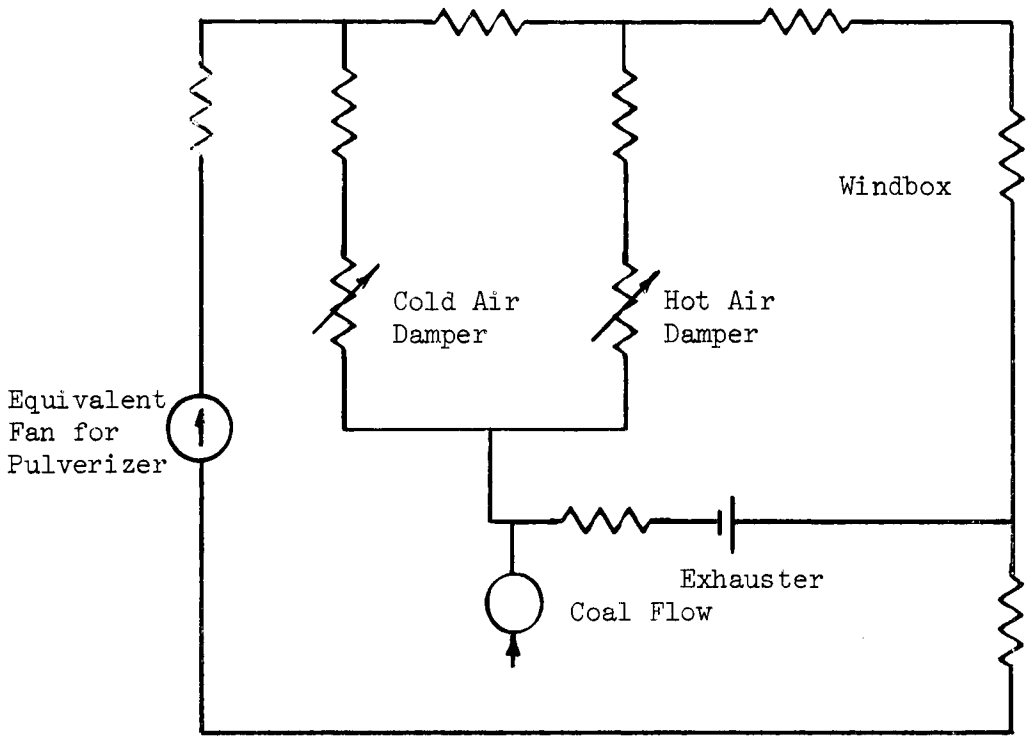


Figure 3-7

Analog Model for Pulverizer air and Fuel Flow

voltage source and the furnace by a lumped resistor. Model of these elements will be developed in the following pages and these will be used to develop the system mathematics.

The goal for this model is to write the state variables as functions of the control and disturbance variables. The principles used to develop these state equations are easily understood.

The cold and hot air dampers are modeled as variable restrictions to flow. A form of Bernouli's equation states that

$$\text{volume flow} = KA_2\sqrt{\frac{(\Delta P)(P_2)}{\rho(1 - A_2^2/A_1^2)}} \quad . \tag{3-1}$$

A_2 is the area of the region unrestricted by the damper as shown in Figure 3-8. P_2 is the downstream pressure and Δp is the drop in pressure across the damper. A_1 is the area of the unrestricted duct and ρ is the density of the air in the duct. The two state variables will be solved in terms of the control and disturbance variables when each term of Equation 3-1 is solved in terms of these variables.

The density is

$$\rho = \rho_o\left(\frac{T_o + 460}{T + 460}\right) \tag{3-2}$$

where ρ_o is the density of air at a temperature of T_o, and T is the temperature of the air in the duct. The temperatures are in degrees Farenheit, and the constant, 460, converts these temperatures to an absolute scale.

Next, A_2, the area of the duct restricted by the damper can be related to the control variable, θ. Figure 3-9 shows a damper in cross

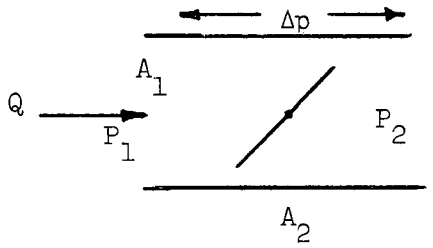


Figure 3-8

Viewing Damper as a Restriction

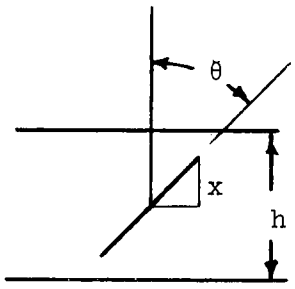


Figure 3-9

Damper

section perpendicular to the axis of rotation. The height, h, of the duct is arbitrarily assigned the value of unity for simplicity. Then, the area of the duct is

$$A_1 = w \tag{3-3}$$

where w is the width of the duct. The height of the damper is the same as that of the duct, and thus, the length from one end to the center is 1/2 as shown in Figure 3-9. On the triangle drawn in this figure the side x is assigned as shown. Then,

$$\cos \theta = 2x \tag{3-4}$$

describes the change in x as θ changes. Now, the height not restricted by the damper is unity minus 2x. Using Equation 3-4, the area opened to flow is

$$A_2 = (1 - \cos \theta)w \tag{3-5}$$

For the state equation, it is preferable to use percentage area open to flow. So dividing Equation 3-5 by Equation 3-3 and converting to percent gives

$$A_2 = 100(1 - \cos \theta) \ . \tag{3-6}$$

The ratio, A_2^2/A_1^2, can also be computed in terms of the control variable, θ. This ratio is

$$A_2^2/A_1^2 = (1 - \cos \theta)^2 \ . \tag{3-7}$$

Then the term, $1 - A_2^2/A_1^2$, can be computed. This term is

$$1 - A_2^2/A_1^2 = 2 \cos \theta - \cos^2 \theta \ . \qquad (3\text{-}8)$$

The variable θ in Equations 3-6 and 3-8 will be replaced by $.9\theta$ in the state equations since the damper positions in the actual system are normally given in percent from 0% to 100%.

The downstream pressure, P_2, can be approximated by the disturbance variable, P_B, due to the relative closeness of the dampers to the pulverizer base and to the fixed duct arrangement.

The upstream pressure on each damper depends upon the forced draft fan operation and system losses in front of the damper. Except for the windbox damper setting, the duct arrangement in front of the cold and hot air dampers is fixed. The losses depend upon the flow squared as shown in Equation 2-4. By the principle of continuity of flow, the velocity pressures on each side of the damper are equal. The static pressure on the upstream side of the cold air damper may be approximated by the forced draft fan static pressure minus the losses in the duct system. Then for the cold air damper, the pressure drop across the damper is

$$\Delta p = (K_1 + \Delta pwb)n^2 - P_B \ . \qquad (3\text{-}9)$$

There is an additional loss in the hot air duct system through the air preheater given by Equation 2-7. Thus the pressure drop across the hot air damper may be approximated by

$$\Delta p = (K_1 + \Delta pwb)n^2 - K_2 \left[\frac{T_h + 1505}{2030}\right] - P_B \ . \qquad (3\text{-}10)$$

The equations of flow can now be stated in terms of the control and disturbance variables. The cold air volume flow is

$$Qc = Kc\ [1-\cos(.9\theta_c)]\sqrt{\frac{[(K_1+\Delta pwb)n^2 - P_B]\ P_B}{\rho_c\ [2\cos(.9\theta c) - \cos^2(.9\theta c)]}}\ , \qquad (3\text{-}11)$$

and the hot air volume flow is

$$Qh = Kh\ [1-\cos(.9\theta_h)]\sqrt{\frac{[(K_1+\Delta pwb)n^2 - K_2\left[\frac{T_h + 1505}{2030}\right] - P_B]\ P_B}{\rho_h\ [2\cos(.9\theta_h) - \cos^2(.9\theta_h)]}}\ . \qquad (3\text{-}12)$$

A heat balance on the pulverizer gives the final state equation. The mill temperature is

$$Tm = \{.245(T_c Q_c \rho_c + T_h Q_h \rho_h) + (Q_{coal})(K_{motor}) + T_f(1-m)(.325)(Q_{coal}) + (Q_{coal})\left(\frac{r-m}{1-r}\right) K_{f_{T_m}} - K_{f_{T_f}} \} / \{.245(Q_c\rho_c + Q_h\rho_h) + (Q_{coal})(1-m)(.325) + K_R\}. \qquad (3\text{-}13)$$

This equation is derived in Appendix 1.

Equation 3-11, 3-12, and 3-13 form the mathematical model for the system in terms of control, disturbance and state variables as illustrated in Figure 3-10. In Chapter 4, static optimum operation of the system will be discussed in terms of this model.

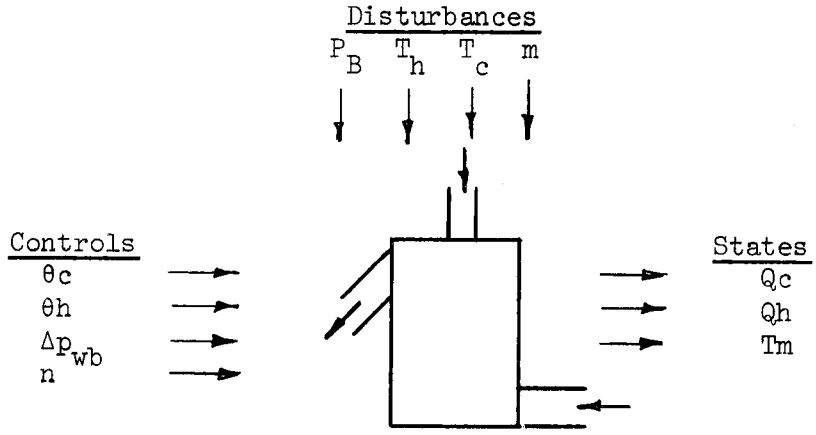


Figure 3-10

System Model

CHAPTER IV

ANALYSIS AND RESULTS

The objective of this chapter is to apply the techniques of modern system analysis to the model developed in Chapter 3. The utility of the mathematical model is that this model can be analyzed with straightforward and routine techniques. The solutions to problems solved in this way must then be related to the real physical world. It is important that the model closely represent the physical system, otherwise the results of the analysis are not useful.

Shown in Figure 1 is a graphical interpretation of the system state equations. The mass flow rates, Q_c and Q_h, are plotted on the axes in pounds per minute. The dashed lines show the maximum possible flows of Q_c and Q_h with the control and disturbance variables as shown. Equation 3-13 is solved for Q_c in terms of Q_h and is

$$Q_c = \{Q_{coal}(1-m)(.325)(T_m-T_f) + T_mR + Q_{coal}\left(\frac{m-r}{1-r}\right)(1120)-13Q_{coal} - Q_h(T_H-T_m)\}/(.245)(T_cT_m) \qquad (4-1)$$

This equation is then plotted for a particular value of pulverizer temperature, T_m. This line is shown in Figure 4-1.

If this graph is viewed as a linear programming problem where Q_c maximum, Q_h maximum, and T_m are the constants and the problem is to maximize a criteria such as total primary air flow, $Q_c + Q_h$, the solution would be at point A in Figure A-2. However, this point might not be a realistic place to operate the physical system because a disturbance driving the model past Q_h maximum on the line T_m is not

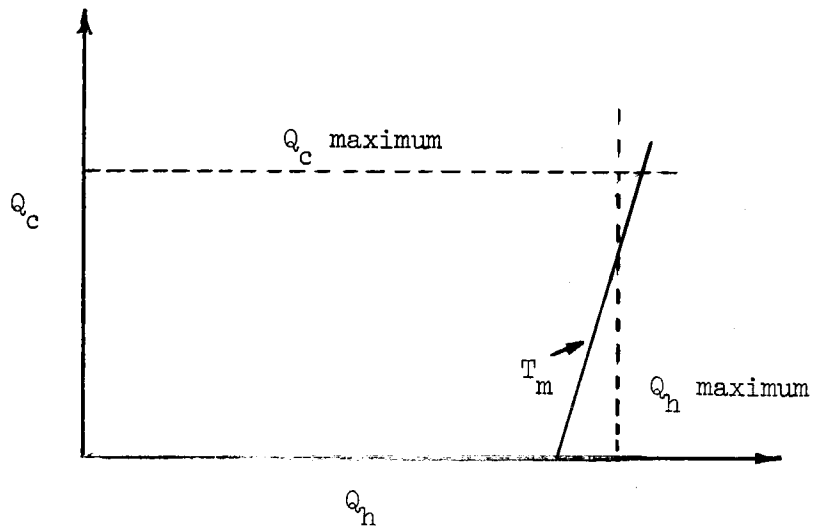


Figure 4-1

Static Map

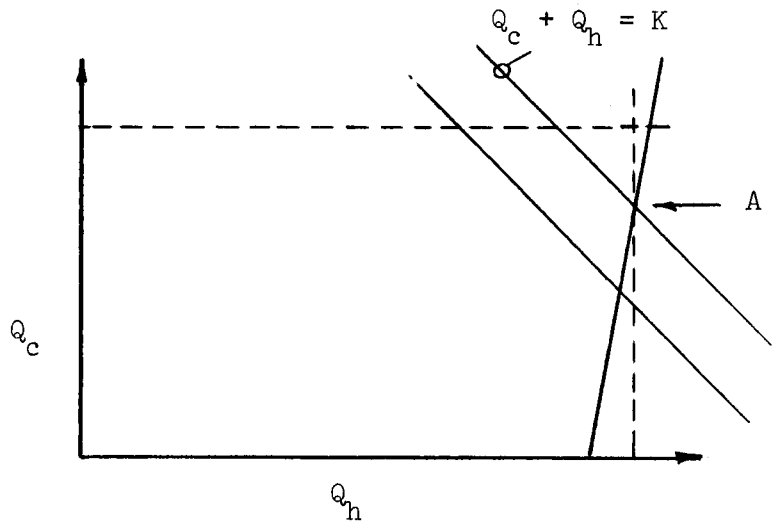


Figure 4-2

Linear Programming

realizable in the physical system.

Figure 4-3 shows the range of permissible operations points of Q_c and Q_h for the range of pulverizer temperatures from 160°F to 180°F. The operating region is shown in the shaded region and the operating conditions are shown.

It is illustrative to show the effects of changes in the disturbance variables on the operation of the system. Figure 4-4 shows the effect of a change in moisture content from 6 percent to 7 percent. In fact, increasing the moisture content to 8 percent drives the pulverizer temperature constraint line out of the operating region between Q_c maximum and Q_h maximum. Then operation with the conditions shown is no longer possible. Returning to the constraint line for a moisture content of 7 percent, consider how to improve operation in terms of the criteria of total pounds of primary air, $Q_c + Q_h$. Figure 4-5 shows that decreasing the rate of coal delivery moves the pulverizer temperature constraint line back to the left, where greater primary air flows are possible.

Next, consider the control and optimization problem illustrated in Figure 4-6. The optimization problem can be examined using these equations as the static model for the pulverizer system. Before an optimization method can be applied to the model, a cost function must be chosen. The cost function is the function that is to be optimized and it may be a function of state, control, and disturbance variables. For this example, we have chosen to maximize the primary air supply to the pulverizer given by $Q_c + Q_h$ where Q_c and Q_h are given in pounds per minute. The constraint on optimization is that the pulverizer

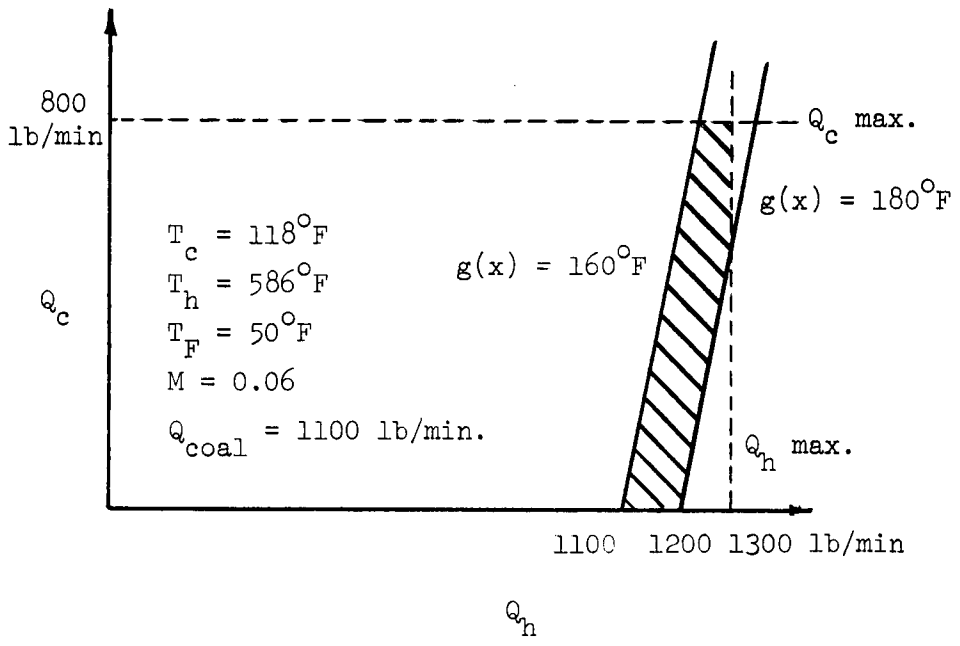


Figure 4-3

Static Map of Operation

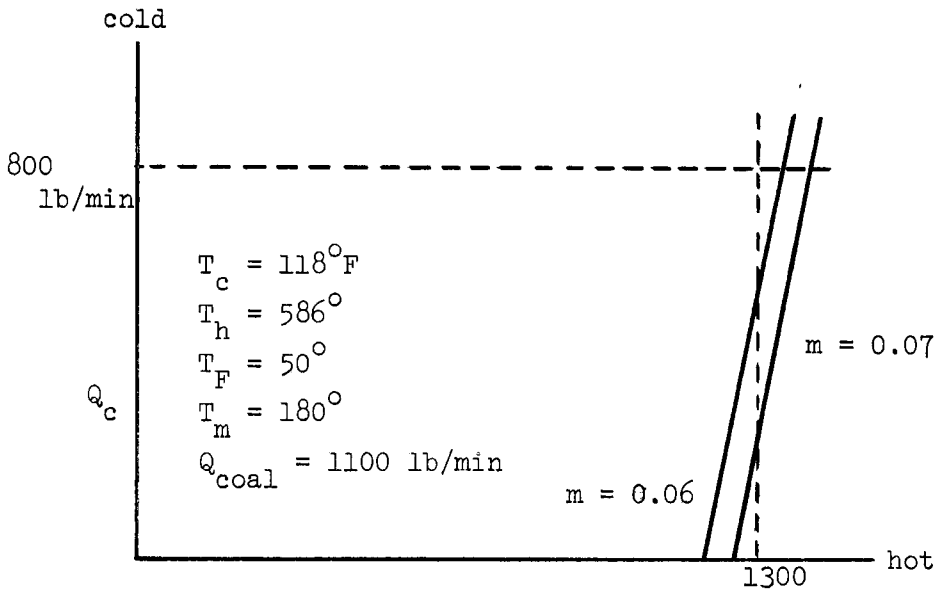


Figure 4-4

Change in Moisture Content of Coal

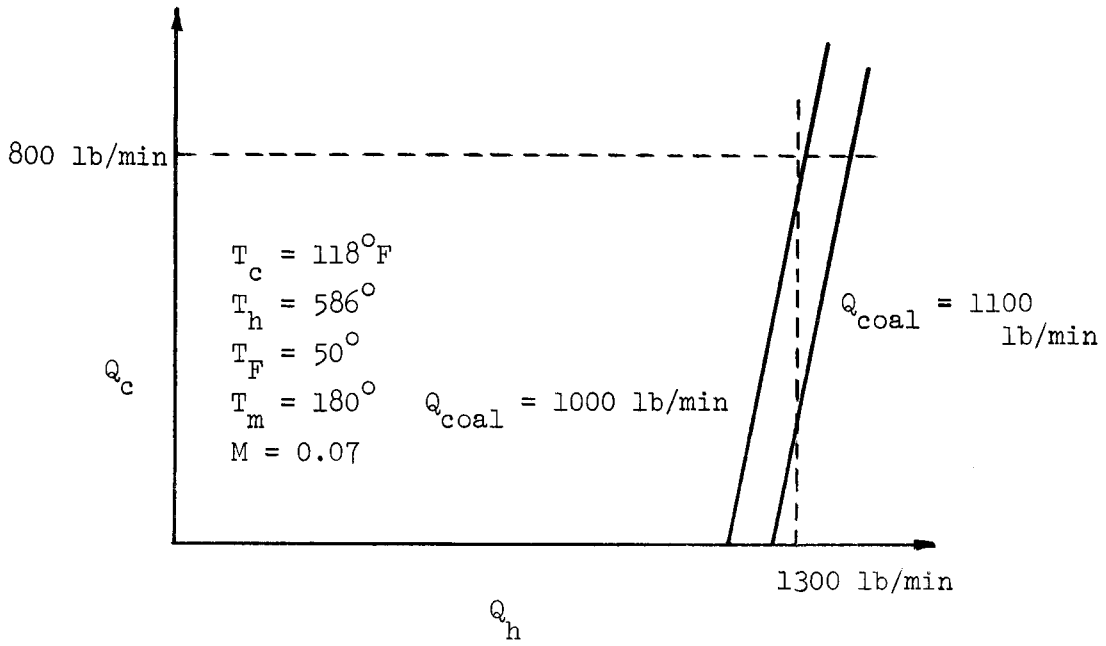


Figure 4-5

Change in Rate of Coal Delivery

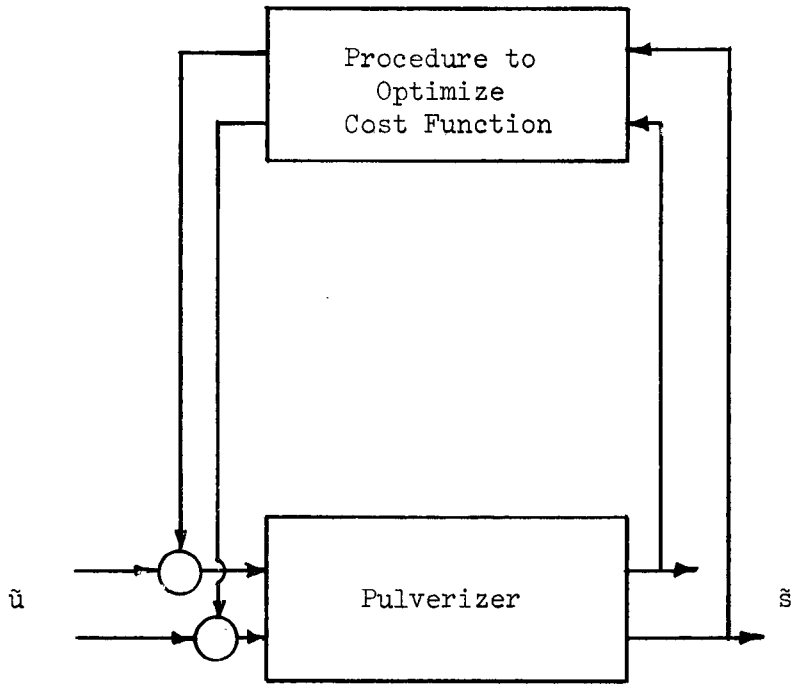


Figure 4-6

The Optimization Problem

temperature must be maintained as given in the heat balance equation.

The function to be optimized is given by

$$f(\bar{x}) = Q_c + Q_h$$

where Q_c and Q_h are the mass flow rates of cold and hot primary air. The volume flow rates are shown in Equations (3-11) and (3-12). The constraint equation is given by Equation 6 which may be restated as

$$g(\bar{x}) - T_m = 0 \tag{4-3}$$

The method of Lagrangian multipliers is used to examine the model for optimum operation. In this method, an objective function

$$F(\theta_c, \theta_h, \Delta p, n, \lambda) = Q_c(\theta_c, \Delta p, n) + Q_h(\theta_h, \Delta p, n) + \lambda(g(\bar{x}) - T_m) = 0 \tag{4-4}$$

is formed and a system of equations

$$\frac{\partial F}{\partial \bar{x}} = 0 \tag{4-5}$$

is derived. The roots of this set of equations are possible optimum operating points if they meet conditions set by the operating constraints which are

$$0\% \leq \theta_c \leq 100\% \tag{4-6a}$$

$$0\% \leq \theta_h \leq 100\% \tag{4-6b}$$

$$4 \leq \Delta p \leq 8 \tag{4-6c}$$

$$0 \leq n \leq 1173. \tag{4-6d}$$

Equation 4-5 becomes

$$\partial F/\partial \theta_c = \rho_c \partial Q_c/\partial \theta_c + \lambda \partial g(\bar{x})/\partial \theta_c = 0 \qquad (4\text{-}7a)$$

$$\partial F/\partial \theta_h = \rho_h \partial Q_h/\partial \theta_h + \lambda \partial g(\bar{x})/\partial \theta_h = 0 \qquad (4\text{-}7b)$$

$$\partial F/\partial \Delta p = \rho_c \partial Q_c/\partial \Delta p + \rho_h \partial Q_h/\partial \Delta p + \lambda \partial g(\bar{x})/\partial \Delta p = 0 \qquad (4\text{-}7c)$$

$$\partial F/\partial n = \rho_c \partial Q_c/\partial n + \rho_h \partial Q_h/\partial n + \lambda \partial g(\bar{x})/\partial n = 0 \qquad (4\text{-}7d)$$

$$\partial F/\partial \lambda = g(x) - T_m = 0. \qquad (4\text{-}7e)$$

This system of equations is then solved using the numerical approach shown in Figure 4-7. Results of this approach show that optimization of the cost function necessitates operating at the upper limits of the control variables shown in Equation 4-6. The partial derivatives in Equation 4-7 were computed and used in the computational routine illustrated in Figure 4-7. This computational precedure presented some problems in finding reasonable solutions. First, the routine often converged toward the minimum value of the cost function instead of the maximum. Second, the functions for Q_c and Q_h have discontinuities at θ equal to 100% causing the computer routine to exit with an error message.

In order to compare the present control system on the pulverizer with a control system using the model to compute control setting, the cost function was computed for a set of typical operating conditions. The control schemes for each are illustrated in Figures 4-8 and 4-9 with the operating conditions shown. The computed value of the cost function, $Q_c + Q_h$, for each system is shown.

In summary, it has been shown how the pulverizer system was analyzed using some modern system analysis techniques. A model for

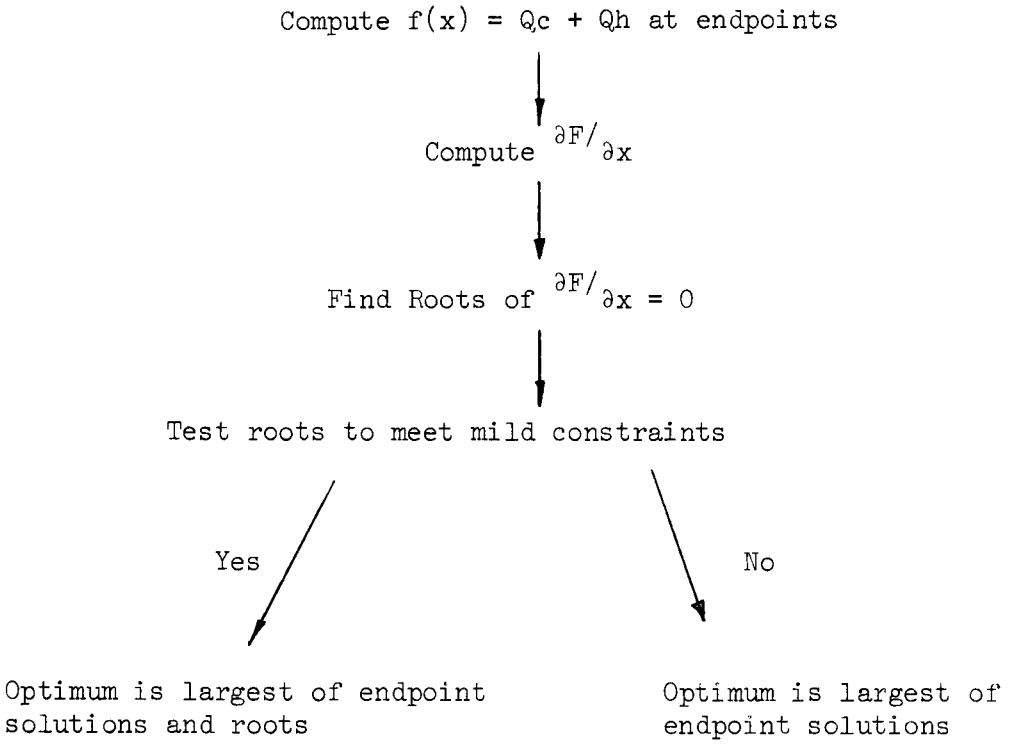
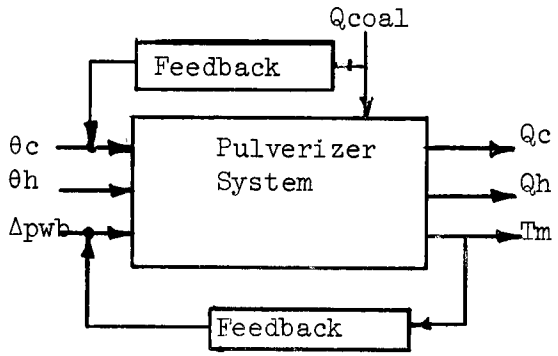


Figure 4-7

Flow Diagram of Numerical Procedure for Optimization

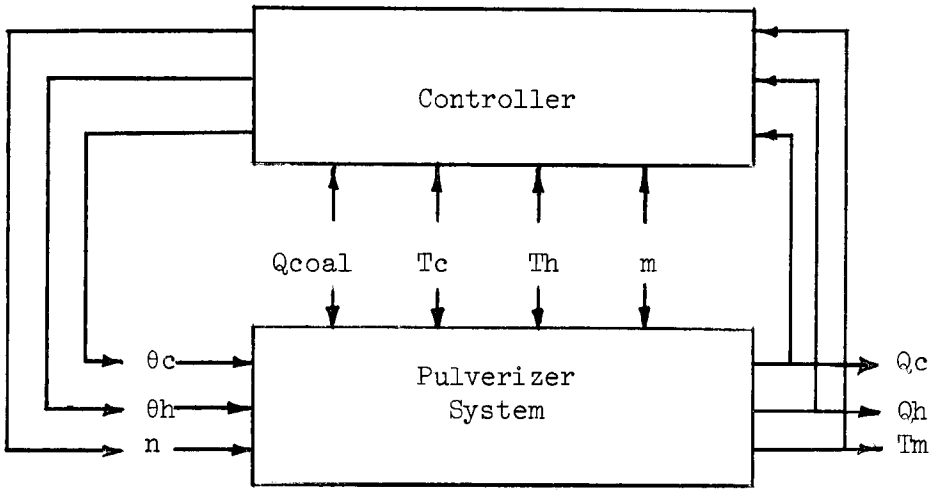
Operating Conditions:
m = 0.06
Qcoal = 1100 lb./min.
Tc = 118°F
Th = 586°F
T_F = 50°F
T_m = 180°F



Cost Function = Qc + Qh = 1600 lbs./min.

Figure 4-8

Present System Control



Cost Function = Qc + Qh = 1800 lb./min.

Figure 4-9

Control Using Model

the system was developed in terms of control, disturbance, and state variables. Possible optimum operating strategies were discussed and quantitative effects of the variables were shown. The operation of this system can be improved in terms of the cost function chosen. However, this strategy must be modified in reality since it is important to maintain controllability in the event of disturbances. Quantitative effects on the cost function of changing the operating point can be seen from these graphs. It may be feasible to improve operation of the pulverizer by operating closer to the operation limits. However, it will be necessary to explore the dynamics of the system before this conclusion may definitely be drawn.

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APPENDIXES

APPENDIX I

PULVERIZER HEAT BALANCE

This appendix uses a simplified pulverizer heat balance to develop Equation 3-13 which is one of the pulverizer state equations. A similar development for a different purpose can be found in Reference 7.

The basis for a pulverizer heat balance is straight forward. The heat which is furnished to the process is balanced against the disposition of this heat in the process. It is assumed that this balance is continuously maintained in the pulverizer and that the output of the pulverizer consisting of pulverized coal and primary air is in thermal equilibrium.

The heat which is added to the process comes from two sources:

1. The heat furnished by the primary air.
2. The heat from the crushing action of the motor.

The disposition of this heat goes to four places:

1. Most of the moisture in the coal is evaporated to a vapor at a temperature of T_m.
2. The coal is heated to the temperature of the coal-air mixture (T_m).
3. Residual moisture in the coal is heated to the pulverizer temperature (T_m)
4. Some heat is lost by radiation.

The following symbols will be used in the development of the heat balance equation:

Q_h = hot air flow, cubic feet per minute

ρ_c = cold air density, pounds per cubic foot

ρ_h = hot air density, pounds per cubic foot

T_c = cold air temperature, degrees R

T_h = hot air temperature, degrees R

C_p = 0.245 = specifies heat of air

C_c = 0.325 = specifies heat of coal

T_m = mill temperature, degrees R

K_m = crushing constant for motor, BTU per pound

K_R = Radiation loss constant

T_F = coal temperature entering mill, degrees F

T_P = Primary air temperature, degrees F

m = moisture content of coal, percent

r = residual moisture

h_{f_1} = enthalpy of water vapor at T_m

h_{f_2} = enthalpy of water at T_F

Q_{coal} = coal flow, pounds per minute

Q_c = cold air flow, cubic feet per minute

The heat supplied by the primary air in BTU/minute is

$$q_1 = (Q_c\rho_c + Q_h\rho_h)(T_p - T_m)(C_p) \qquad \text{(A1-1)}$$

where

$$\rho_c = 0.075 \times (528/T_c) \qquad \text{(A1-2)}$$

and

$$\rho_h = 0.075 \times \left(\frac{528}{T_h}\right) \quad . \tag{A1-3}$$

The primary air temperature may be replaced by an equivalent quantity which is

$$T_p = \frac{Q_c\rho_c T_c + Q_h\rho_h T_h}{Q_c\rho_c + Q_h\rho_h} \quad . \tag{A1-4}$$

Equation A1-1 becomes

$$q_1 = (Q_c\rho_c + Q_h\rho_h)\left(\frac{Q_c\rho_c T_c + Q_h\rho_h T_h}{Q_c\rho_c + Q_h\rho_h} - T_m\right) C_p \tag{A1-5}$$

The heat supplied by the crushing action of the pulverizer motor is approximated by

$$q_2 = K_m Q_{coal} \quad \text{BTU/min.} \quad . \tag{A1-6}$$

where K_m is evaluated to be 13 BTU/pound of coal crushed from the experimental data shown in Appendix 2.

The heat required to evaporate the moisture in the coal at the temperature of T_m is

$$q_3 = Q_{coal}\left(\frac{m - r}{1 - r}\right)(h_{f_1} - h_{f_2}) \quad \text{BTU/min.} \tag{A1-7}$$

For T_m ranging from 160°F to 200°F, h_{f_1} ranges from 1130 to 1150 BTU. The enthalpy of water at the temperature T_F may be approximated by 18 BTU for a range of temperatures from 30°F to 60°F. So, $h_{f_1} - h_{f_2}$ may be approximated by

$$h_{f_1} - h_{f_2} \doteq 1120 \tag{A1-8}$$

with a reasonable degree of accuracy. Then Equation (A1-7) becomes

$$q_3 = Q_{coal}\left(\frac{m - r}{1 - r}\right) \quad (1120) \qquad \text{(A1-9)}$$

The heat used to raise the temperature of the dry coal to the mill temperature, T_m, is

$$q_4 = Q_c(1 - m)(C_c)(T_m - T_F) \quad \text{BTU/min.} \qquad \text{(A1-10)}$$

The heat required to raise the residual moisture to the pulverizer temperature is

$$q_5 = (Q_{coal})(1 - m)\ r\ (T_m - T_F) \quad \text{BTU/min.} \qquad \text{(A1-11)}$$

The residual moisture is less than 1% and q_5 is less than 1% of the total heat used in the process. In order to simplify the resulting expression, q_5 will be dropped from the heat balance with only a small effect on the accuracy of the results.

The heat lost by radiation from the pulverizer depends the surface area of the pulverizer, the ambient temperature, and the temperature of the pulverizer. As a useful approximation, this is calculated by the expression

$$q_6 = K_R\ T_m \quad . \qquad \text{(A1-12)}$$

Equating the heat lost with the heat gained gives

$$q_1 + q_2 = q_3 + q_4 + q_6 \quad . \qquad \text{(A1-13)}$$

Solving this expression for the pulverizer temperature, T_m, is

straight forward and gives the result that

$$T_m = \{(.245)(T_c Q_c \rho_c + T_h Q_h \rho_h) + 13Q_{coal} + T_f(1-m)(.325)(Q_{coal}) + (Q_{coal})\left[\frac{r-m}{1-r}\right](1120)\} / \{(.245)(Q_c \rho_c + Q_h \rho_h) + (Q_{coal})(1-m)(.325)+K_R\} \quad (A1\text{-}14)$$

which is used as Equation 3-13.

APPENDIX II

EXPERIMENTAL RESULTS

This appendix contains a description of the equipment used to collect data for establishing parameters in the model. Also, examples of the data collected is shown.

In order to evaluate constants for the state equations for hot air flow to the pulverizer, cold air flow to the pulverizer, and pulverizer temperature, the following system variables must be measured. The following variables were available from the control panel in the control room: Q_c, θ_c, θ_h, Δpwb, T_c, T_n, and T_m. Fan speed, n, was measured by an installed tachometer. Pulverizer base pressure, P_B, was measured by an available static pressure tap on the pulverizer. Coal moisture, m, was measured by a weight difference determination in the coal analysis laboratory. Residual moisture, R, was estimated from the expected performance of the pulverizer to be about 1%. The coal temperature, T_f, was readily measured from a sample from the coal feeder. An accurate determination of the hot and cold air flows was more difficult and required the installation of special test equipment. Averaging pitot tubes were installed in the base of pulverizer A-1 to measure the total primary air flow illustrated in Figure A2-1. Calibration of these pitot tubes was made from the data shown in Figure A2-2. The velocity, v, in feet per minute could be calculated as

$$v = 4004\ C\sqrt{h_v} \quad \text{ft./min.} \qquad \text{(A2-1)}$$

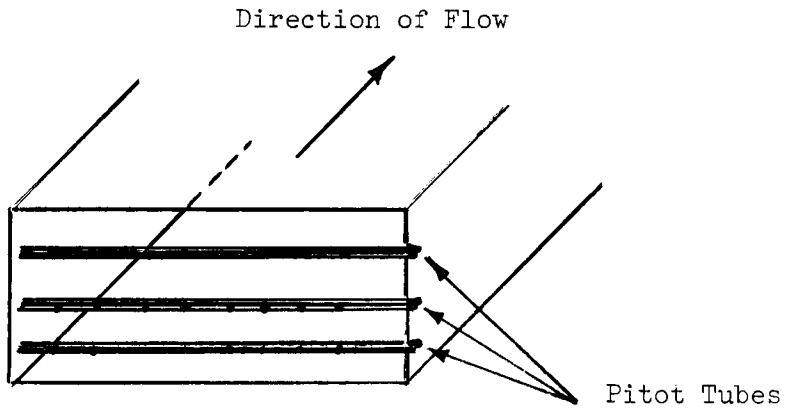


Figure A2-1

Averaging Pitot Tubes

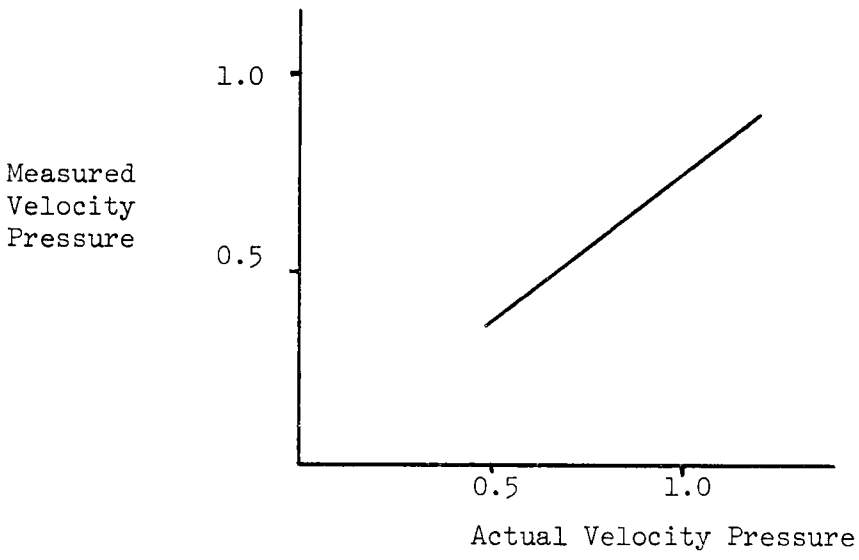


Figure A2-2

Calibration Curve

where C is the correction for a non-standard pitot tube and h_v is the velocity pressure. The velocity pressure is measured as the difference between the total pressure and the static pressure as illustrated in Figure A2-3. For the averaging pitot tubes installed, C was evaluated to be 1.2.

The flow of primary air, Q_m, could be calculated as

$$Q_m = VA \quad ft^3/min. \tag{A2-2}$$

where A is the area of the duct. A is equal to 11.3 ft^2 at the installation point.

The temperature of the primary air, T_p, was measured by an installed thermocouple in the duct. The density, ρ, of the primary air could then be calculated as

$$\rho_{T_p} = \frac{(0.075)\ (528)}{T_p + 460} \quad . \tag{A2-3}$$

The mass flow rate, Q, of primary air is

$$Q = Q_m \times \rho Tp \quad . \tag{A2-4}$$

There is an equivalence of mass flow rates which is

$$Q = Q_c + Q_h \quad . \tag{A2-5}$$

Solving Equation A2-6 simultaneously with a heat balance on the primary air which is

$$Q(T_p + 460^\circ) = Q_c(T_c + 460^\circ) + Q_h(T_h + 460^\circ) \quad . \tag{A2-6}$$

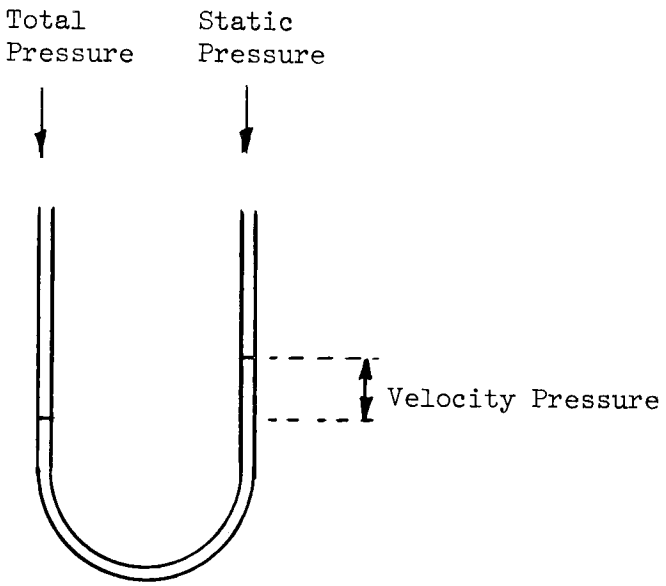


Figure A2-3

Measuring Velocity Pressure

gives a solution for Q_c and Q_h. The volume flow rates can then be calculated as

$$Q_1 = Q_c/\rho_c$$
$$Q_2 = Q_h/\rho_h \qquad \text{(A2-7)}$$

where ρ_c and ρ_h are the densities of the cold air flow and hot air flow respectively.

Typical data is shown in Table III.

Table III

Typical Experimental Data

Test 1		Test 2
45	θ_c	65
55	θ_h	65
1070	n	1070
4	Δpwb	4
14	P_B	14
180	T_m	180
117	T_c	117
586	T_h	586
1100	Q_{coal}	1100
76	T_o	76
467	T_p	425
1100	Q_h	1200
400	Q_c	600

VITA

George Franklin Pierce, the son of Dr. and Mrs. John Franklin Pierce, was born in Pittsburgh, Pennsylvania on February 28, 1948. In 1966, he graduated from West High School in Knoxville, Tennessee, and in 1970, he received the Bachelor of Science degree from the Department of Electrical Engineering at the University of Tennessee in Knoxville. He received the degree of Master of Science in Electrical Engineering in December, 1973 at the University of Tennessee in Knoxville.

He is happily married to the former Kay Monica Harvey.