MODELING AND SIMULATION OF A CAMPUS CENTRAL CHILLED WATER SYSTEM

By

EUY-JOON LEE

Bachelor of Engineering
Yonsei University
Seoul, Korea
1981

Master of Science
Oklahoma State University
Stillwater, Oklahoma
1983

Submitted to the Faculty of the Graduate College
of the Oklahoma State University
in partial fulfillment of the requirements
for the Degree of
DOCTOR OF PHILOSOPHY
July, 1987

OKLAHOMA STATE UNIVERSITY LIBRARY

MODELING AND SIMULATION OF A CAMPUS

CENTRAL CHILLED WATER SYSTEM

Thesis Approved:

Jerald D. Parker
Thesis Adviser

R. M. Moretti

A. J. Ghajar

R. Ramakumar

Norman N. Durham
Dean of the Graduate College

1291697

ACKNOWLEDGMENT

I CAN DO ALL THINGS THROUGH HIM WHO STRENGTHENS ME. (Philippians 4:13). First of all, with my whole heart, I bring glory to God who led me to attain this small result.

I would like to express my deepest appreciation to my thesis adviser, Dr. Jerald Parker, for his warm guidance and patient commitment to me. I also greatly appreciate the time and interest offered by my doctoral committee members, Drs. Peter Moretti, Afshin Ghajar, and Ramachandra Ramakumar.

I am also thankful to Don Shepherd and his people in the OSU Physical Plant, who were primary sources of financial and technical support.

My wife, Sook-Jong, and sons, Jae-Hyun and Jae-Woo, have given me the desire to undertake this valuable work. I also would like to thank my parents for their supports and sacrifices which have been a source of motivations.

TABLE OF CONTENTS

Chapter Page

LIST OF TABLES

LIST OF FIGURES

Figure		Page

NOMENCLATURE

ASHRAE	- American Society of Heating, Refrigeration, and Air-Conditioning Engineers
ASME	- American Society of Mechanical Engineers
ADAPTIVE	- Self-controlled with external disturbances
BUILDING LOOP	- A hydronic circuit inside a building
BCV	- Building control valve
BSIM	- Building simulation program
Btuh	- Btu per hour
CCW	- Central chilled water system
CCV	- Coil control valve
C.F.	- Cooling factor (cooling supplied/cooling used)
CHILLER LOOP	- A hydronic circuit to circulate a constant flow rate through chillers
COMMON PIPE	- A pipe interfacing between the loops
CFM	- Cubic feet per minute
Cv	- Valve coefficient, gpm/(psi$^{0.5}$)
DDC	- Direct digital control
DELP	- A constant set pressure through a variable speed pump.
DH	- Enthalpy difference, Btu/lbm
DISTRIBUTION LOOP	- A hydronic circuit between building loops and chiller loops to distribute the chilled water
DPa	- Actual pressure drop across coil, psi

DPd	- Design pressure drop across coil, psi
DPk	- Pressure drop across k'th element
DPvj	- Pressure drop across j'th CCV
DPt	- Pressure drop of total piping system
DPv	- Pressure drop across the valve
DTp	- Design primary temperature difference, F
DTs	- Design terminal temperature difference, F
ED	- Equivalent diameter
EL	- Equivalent length
ff	- Friction factor
FRa	- Actual flow rate of a coil, gpm
FRd	- Design flow rate of a coil, gpm
F	- Fahrenheit
gna	- Gain of actuator
gns	- Gain of sensor
gnv	- Gain of valve
gpm	- Flow rate of gallon per minute
HVAC	- Heating, Ventilation, Air-Conditioning
HYDRONICS	- Single-phase energy transport system
Kp,i,d	- Gains of PID controller
LOOP	- A hydronic circuit
N.C.	- Normally closed valve
N.O.	- Normally open valve
OSU	- Oklahoma State University
P	- Atmospheric pressure
PID	- Proportional, Integral, Derivative gains
Pj	- Pump coefficients

PR	- Pressure rise through pump
psi	- Pounds per square inches
Pws	- Saturation pressure, psia
R_0	- Water density
SCS	- Staefa control system
Tdb	- Dry-bulb temperature, F
ton	- Cooling capacity of 12000 Btu/hr
toua	- Time constant of actuator
toup	- Time constant of process
tous	- Time constant of sensor
touv	- Time constant of valve
TI	- Texas Instrument
TS	- Sampling time (sec)
Tsr	- Temperature of the secondary return
Tss	- Temperature of the secondary supply
Twb	- Wet-bulb temperature, F
u	- Water viscosity
V	- Water velocity
W	- Humidity ratio
Wcp	- By-pass flow rate, gpm
Wps	- Primary supply flow rate, gpm
Wss (W_s)	- Secondary supply flow rate, gpm
X(n)	- State-variable array
=%	- Equal percentage type valve
17DA	- A trane electric chiller model (centrifugal)
17FA	- A trane steam chiller model (centrifugal)

Subscripts

c	- Corrected value
in	- Inlet
n	- New value
o	- Old value
out	- Outlet
t	- Trial value

Superscripts

'	- derivative

CHAPTER I

INTRODUCTION

1.1 Central Chilled Water System

A Central Chilled Water (CCW) system is an air-conditioning system using water as a medium to cool many thermal loads at various locations from a central plant. The multiple air-conditioning loads interconnected with a CCW system can create the following important economic advantages and energy conservation opportunities. First, the CCW can reduce total installed cooling capacity by taking advantage of non-simultaneous occurrence of individual peak cooling loads. Second, the CCW system can operate one or more chillers to their capacity at high efficiency rather than operate each separate refrigeration system at part loads most of the time.

These aspects of the CCW system led the Oklahoma State University (OSU) administrators to convert the original de-centralized system into a CCW system in the mid-seventies. As shown in Figure 1, about thirty percent of the energy supplied to OSU goes for cooling purposes and the cooling cost in 1986 was estimated to be over one million dollars. Figure 1 also shows the rate of increase in cooling costs, which is quite steady at about 0.25 million dollars per year during the past five years.

As E. E. Davidson, the Vice President of Oklahoma State University,

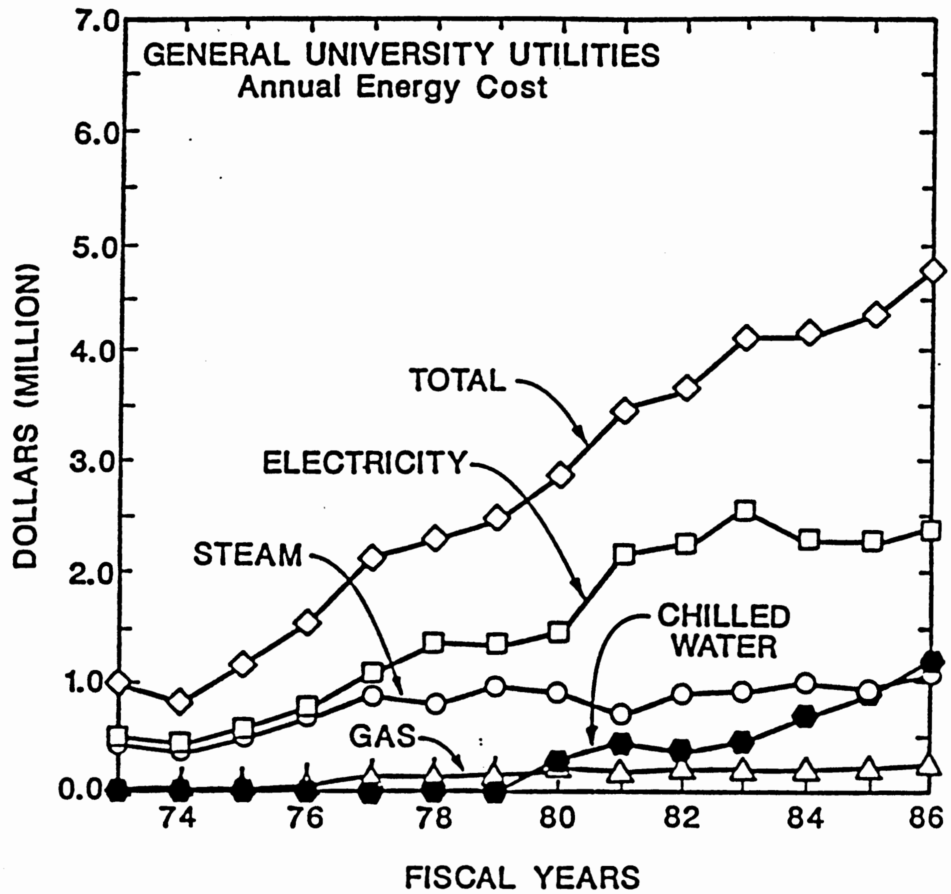


Figure 1. Increasing Cooling Costs

said concerning the CCW system in the Daily O'Collegian newspaper on October 15, 1985, a fifty percent savings have been seen by moving toward a CCW system. The OSU Board of Regents approved more than 1.7 million dollars in expenditures for Phase 3 of the OSU CCW system development plan which includes the installation of another 4,200 ton electric chiller. The completion of Phase 3 CCW project will make the total capacity 12,000 tons with a flow capacity of 18,600 gpm. According to the previous research (7) on the OSU cooling expansion capacity study, Phase 3 will cover the campus cooling load without further expansions for the next several decades. The next assignment for the CCW system is its energy-efficient operation to take full advantage of the centralized system. In this case, the steadily increasing cooling costs shown in Figure 1 could be reduced significantly.

Any large central chilled water system piping loop can be grouped as building, chiller, and distribution loops. This study was initiated to examine these loops, using both hydraulic and thermal analysis. The models developed for this purpose provide optimum solutions for the problems faced by OSU CCW systems. The building loop with cooling coils must be a variable flow system (constant temperature system), which varies the flow rates to the coils as the thermal loads change, by regulating the control valves on the coils and the line to the main distribution line. Thus, a problem of optimum control valve selection based on both steady-state and dynamic analysis must be solved to provide a constant return temperature to the main return line.

According to Haines' classification (53), there are three kinds of distribution piping systems: 1. direct return, 2. reverse return, and

3. circular link type systems. The circular link type distribution system should be the best among the three systems in terms of controllability. However, the OSU CCW distribution system belongs to the first type which is the most difficult to control the flow rates needed for each building loop. Figure 2 shows a simplified layout of the OSU CCW distribution system, which is a typical direct return system. Each building branch loop is expressed as a dotted line in Figure 2. Pressure and flow characteristics of the branch are defined with a branch coefficient which is analytically similar to a valve coefficient. It is desirable for any distribution system to be a variable flow system, as the building demand varies. The variable flow system produces a constant temperature differential by changing its flow rate.

The chiller loops in the central plant require constant flow rates through each chiller. The pipe connected between the chiller loops and the distribution loops, called a "common" pipe, functions as an interface between the loops. This is very important and critical for the successful operation of a large CCW system such as the one at OSU. Ideally, no flow in the common pipe in a central plant is desired but it is not always possible to match the flows of the chillers and distribution pumps. The present location of this common pipe creates a blending problem upstream of the chillers, which impairs efficient chiller operation and control. The blending often supplies warm temperature to the buildings by mixing the warm return temperature and the cool supply temperature from the chillers.

Reduction in chiller efficiency occurs when the distribution return temperature is low. Most of the building loops on the OSU campus do not effectively regulate the return temperature because of poor design

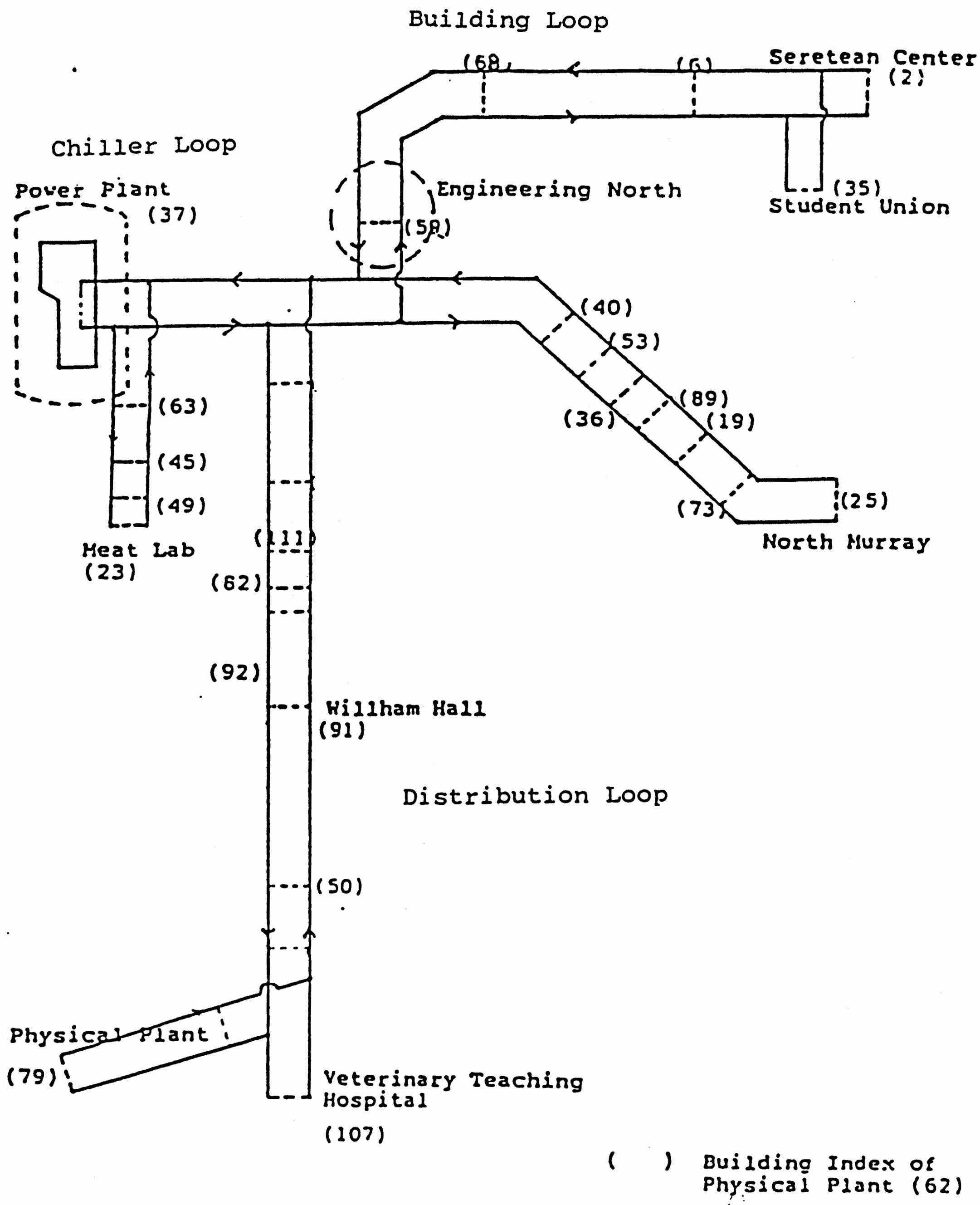


Figure 2. OSU CCW System Model

and mechanical failures in the building control valves. As a result water is usually returned to the plant at too low a temperature.

1.2 Problem Statement

Figure 3 shows a photograph of the Engineering North building interface which includes the main supply, the common pipe, the secondary supply, the secondary return, and the main return line. Early in the study on the OSU CCW system (7), it was found that the primary source of most problems which occurred in the CCW system was due to the improper operation of the building control valves. These valves were installed to control the return water temperature to the main return line. A previous report on the campus cooling capacity expansion (61) made the following observations on existing designs and operation. First, the cooling equipment in most building was quite overdesigned (7). This over-estimation led to oversizing of other components such as coils, valves, and pumps. Quite often the HVAC system worked quite well at the design condition but would not operate well in partial load condition.

Because of the inoperation of building control valves the building loops are now operated as a constant flow system. The secondary loop was originally designed for a de-centralized system where some building loops had their own chiller. The modifications of the original chilled water system resulted in the removal of the local chillers and connection to the CCW distribution lines. The change caused large reductions of flow resistance in the secondary loop so that the existing secondary pump would draw much more of the primary water than required. At reduced loads on the buildings, less flow is required through the coils. Under these conditions the cool main supply water was by-passing

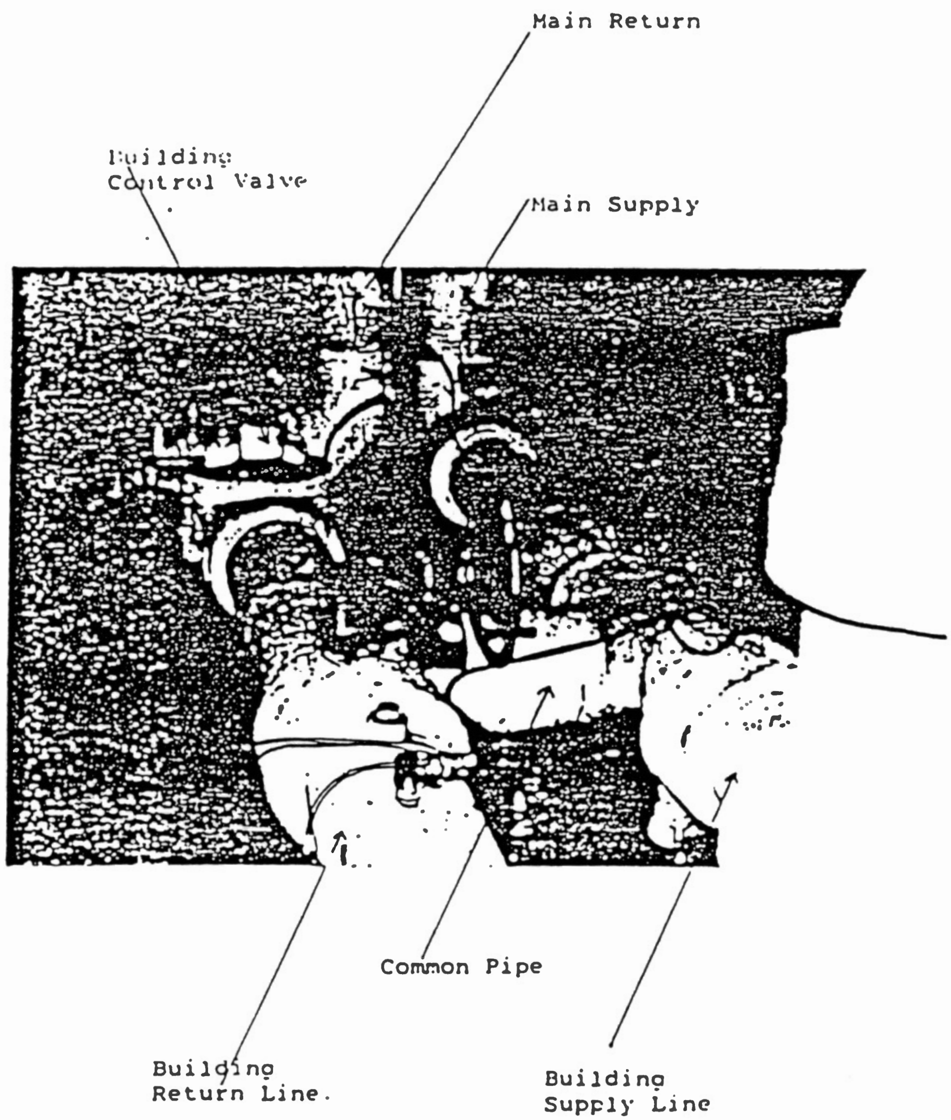


Figure 3. Actual Building Loop Interface

through the common pipe without picking up heat at the terminals.

1.3 Literature Review

A large central chilled water system can be divided into hydraulic, thermal, and control systems for study. Recently, the thermal environmental group in the American Society of Mechanical Engineers (ASME) has emphasized the importance of modeling and simulation of buildings and campus hydronic systems for energy efficient system operation (1). The American Society of Heating, Refrigeration, and Air-Conditioning Engineers publishes four very useful handbooks to keep up with new technology, including hydronics. These books deal with Fundamentals, Equipment, Systems, and Applications (2).

Hodge (8) clearly explained the fundamentals of fluids and thermal aspects to analyze and design an energy efficient system. Hodge also stressed system simulation, especially for the energy system development and effective operation. Wood (21) developed a systematic way of simulating the piping systems utilizing the Hardy-Cross method as presented in his paper and later prepared a manual for his program (22). Stoecker et al. (16), (17), and (18) have, since the mid-seventies, contributed to the modeling and simulation of HVAC equipment and CCW system operation. Stoecker utilized the multivariable Newton-Raphson technique as the piping network simulation method. His books on thermal system design (20) and refrigeration and air-conditioning (19) uniquely dealt with the curve fitting technique to model energy related systems, such as pumps, coils, and chillers, etc. Useful ideas of analysis and design of a HVAC system may be obtained form McQuiston and Parker's book (5) while the chapter on fluid flow provides the basic concept of a CCW system and the chapter also briefly explains a configuration of a variable flow system.

Hydronics, according to Coad (13), has become the most commonly used energy transfer system for central air-conditioning purposes. Coad explained the reason as "easy" controllability of the system compared to other energy transfer systems, such as all-air or two-phase systems. Hansen (4) established the basic concepts of nearly all aspects of hydronic systems, especially the areas of distribution systems and automatic controls in the CCW system loops. He also classified, as Coad (14) did, the distribution systems as constant flow systems and variable flow systems. Coad (15) also explored configurations of the chiller loops, which he showed gave better performance than conventional chiller loops.

Haines (53) was mainly concerned with the control aspects of HVAC systems but he also published on the control of CCW system (50). Haines summarized the method of control valve selection at the interfaces between the building loop and the main distribution lines (51). The computer based monitoring and control system described by Haines (52) has now become important areas of research and a focus of attention of industries. The actual cases concerning problems in campus-type system were documented and optimum solutions for the particular cases were presented at Purdue University's CCW conference (3).

As pointed out by the previous researchers, like Williamson (3) and Bahnfleth (3), most of the problems in campus CCW systems occurred at the interface of the chiller-distribution loop and the interface of the distribution-building loop. Hawk (3) in 1976 used a computer to simulate campus buildings and recommended the primary-secondary loop system for several campus buildings. Mannion devised a flow control system for the troubled common pipe line to increase the effectiveness of the secondary chilled water system performance (56), (57). So far, all efforts have been directed toward increasing the performance which requires higher cooling effects with less chilled water being pumped from the central location. Gladstone (30) explained the procedures of balancing and adjusting the water flow at the pump or at the coils of the CCW system.

Carlson (12) pointed out key problems and developed figures to determine the amount of primary flow rate, depending upon the various conditions of the building loop system operating parameters: thermal loads on coils, pump flow rate, valve characteristics, and other design requirements.

Information on specific problems on the OSU CCW system were obtained through the inquiries with Mr. McKinzie of the OSU Physical Plant and Mr. Cummings of the OSU Power Plant and their internal memoranda (63). An expansion study, conducted by Parker and Moretti (61), indicated a chilled water storage system as a feasible option, theoretically, but the storage option was not found to be economical under the electric rate structure OSU had at that time. This expansion study strongly recommended modifications of the building loops into a constant temperature systems.

The main task of this study is focused primarily on the successful modification of existing systems into more satisfactory systems by simulating the central chilled water systems.

1.4 Objectives

The purpose of this study is to develop methodology for predicting the behavior of the campus the CCW systems. A general method to examine any type of large CCW system has to be developed first to recommend designs that will result in optimum performance. The CCW system is simulated by a building loop model, a chiller loop model, and a distribution loop model. The generalized method should be suitable for simulating systems such as the one at Oklahoma State University to detect unforeseen problems and to recommend suitable modifications.

Steady-state building loop models using two different methods (direct-substitution method and iterative method) are used to predict pressures, flow rates, and temperatures at specified locations for various combinations of the following operating parameters:

1. coil thermal load variations
2. coil control valve characteristics
3. pump characteristics
4. building control valve characteristics
5. pipe size and arrangements.

A macro distribution model is used to detect possible reasons for the unnecessary pressure losses in distribution lines to prevent overpumping. This model simulates the modification of pipe diameters as well as different modes of the distribution pumps. The control mode of the distribution pumps and their characteristics need to be known from the local system loop studies. The goal of this DSIM (Distribution Loop Simulation) is to provide a cost-effective way of pumping for any future expansion which might be added to the present distribution system.

Emphasis is placed on developing a large CCW system model which has three distinctive loops: building, chiller, and distribution loops.

CHAPTER II

OPTIMUM SELECTION OF THE SYSTEM COMPONENT

This chapter discusses the basic guide-lines to select several important components of a successful large central chilled water system. The major components considered are chillers and their pipings, coils and their connections with other system components, distribution pipings, and valves. The methods presented in this thesis have been drawn from the work of well-known people in these areas and will be used for further study. The criterion for the selection was overall cooling system efficiency, which is defined as the cooling used (ton) over the cooling produced (ton) from the central plant.

2.1 Chillers and Pipings

Chillers are used to cool the warm return water from a load. The compressors usually consume substantial energy compared to other equipment. Operation at design load conditions is usually desired for the compressive chillers but the ideal situation is seldom met because of variable loads. Generally, the leaving temperature from a chiller is controlled to be constant and the load to each chiller is determined by the return temperature.

Coad (13) proposed a concept of multiple chiller operation to meet the continuously varying load by discrete operation of unit chillers.

This proposed system is shown in Figure 4.a. Under partial load conditions, the system shown in Figure 4.a (post-common pipe system) has advantage over Figure 4.b (pre-common pipe system) during sequential operation. In Figure 4.a, the chillers will unload in sequence starting with the Unit 1, whereas in Figure 4.b, all chillers will unload equally permitting a blending when their pumps are operating.

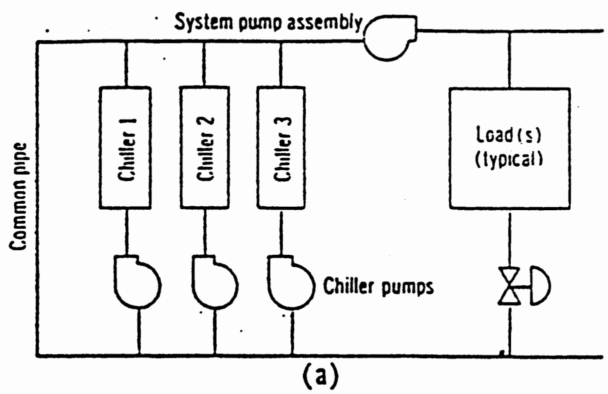
Trane Company (43) provides data with engineers newsletters to recommend optimum chiller connections for improving the overall cooling efficiency. One of the methods suggested by them is to monitor the direction and flow rate on common pipe. Then, the information from the monitoring device is used to control the chillers and the distribution pumps optimally.

A chiller capacity is selected based on the expected cooling load (ton) and CCW diversity factor. The diversity factor is defined as the selected chilling capacity over the estimated cooling load for the campus. The cooling capacity of a campus is normally designed with the diversity factor between 0.6 and 0.7.

The OSU Physical Plant has expanded its capacity Phase by Phase as the cooling load increases. Figure 5 shows the past and predicted future OSU campus cooling loads and the steps taken to meet these loads. With the present gas and electric rates, electrically operated chillers are preferred since they operate more economically than steam driven or absorption types.

2.2 Coils and Pipings

Coils used in the air-water heat exchange in the buildings are an important part of the chilled water system. Ideally, variable air flow



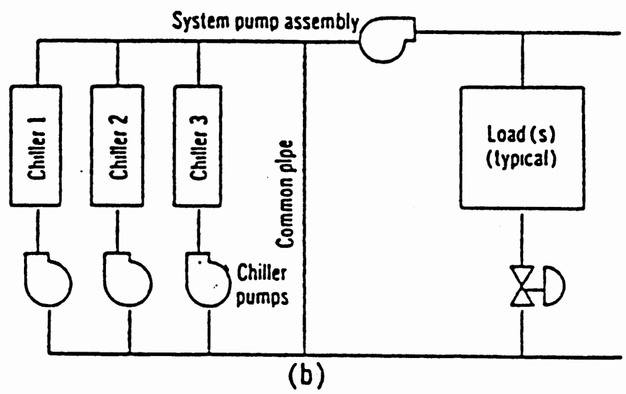


Figure 4. Two Chiller Loop Pipings (15)

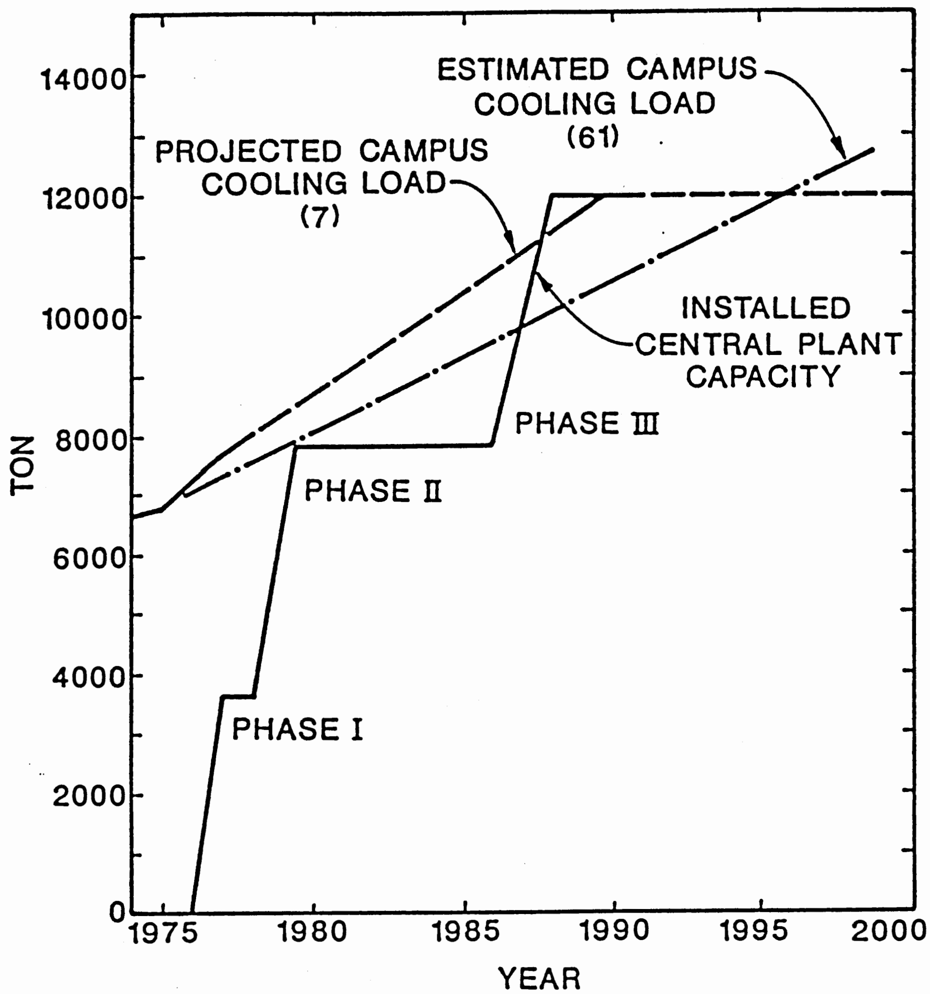
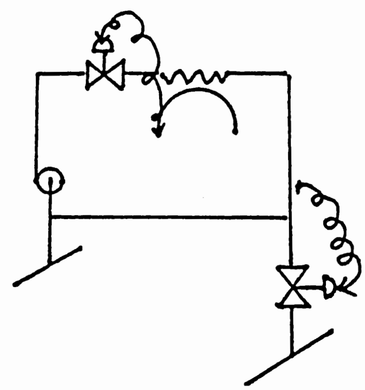


Figure 5. Projected Campus Cooling Load

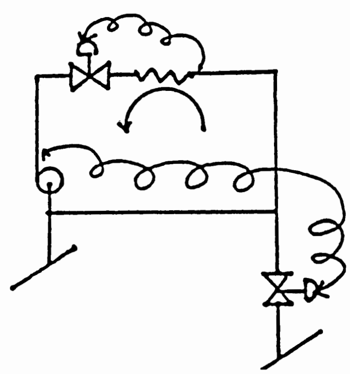
rates and variable water flow rates are desired as thermal loads change. However, in many coils, the flow rates of air and water are constant so that the outlet temperatures in both streams vary as the load changes. The coils in a building transfer heat to the water inside the tubes. The water flow rate through the coil should change with variations of inlet air dry/wet bulb temperature because the outlet air temperature is usually controlled to be constant. Water inlet/outlet temperatures should be controlled to be constant to assure dehumidification and the meeting of maximum loads.

When a central plant serves several buildings, those buildings may vary greatly in size, load and internal pressure losses. If the system distribution pumps are required to provide sufficient pressure to overcome the building losses, some severe energy penalties will result. It is typical, therefore, to provide secondary pumps, at least in the "larger" buildings, to avoid this penalty. The important criterion when connecting secondary (building) pumps is to provide hydraulic separation. That is, the operation of the secondary pump should not influence the pressure differential in the primary distribution system. Additionally, it is desirable to control the return water temperature from the building to maximize the water temperature differential in the primary distribution.

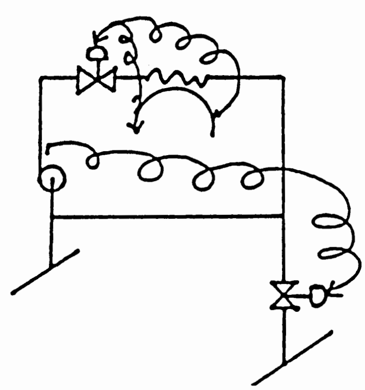
There are possibly three types of such constant return water temperature system available as shown in Figure 6. a, b, and c for this study. With the system of Figure 6.a, the temperature sensor modulates the control valve to maintain a constant return water temperature as the load varies. The control valve sees only the pressure differential between the primary supply and the primary return because the by-pass



(A) RETURN WATER CONTROL METHOD



(B) SUPPLY WATER CONTROL METHOD



(C) WET-BULB TEMPERATURE CONTROL METHOD

Figure 6. Three Air Handling Unit (Coil) Pipings

(common pipe) line has no restrictions. The by-pass line provides hydraulic isolation between primary and secondary system as the common pipe between the chiller loop and the distribution loop. However, the building supply water temperature will increase as building load decreases with this arrangement of Figure 6.a. This system may not be satisfactory where a low dew-point is required for humidity control. The return water temperature can be reset as a function of building humidity to satisfy this requirement. The humidity problems may be solved by proper coil selection and use of a suitable low inlet water temperature.

Another system shown in Figure 6.b is sensing two critical locations in water-side: the secondary (building supply and coil water outlet. The inlet water temperature is held constant by controlling the building control valve. The coil control valve must maintain a constant outlet water temperature for the variable load from the air-side (inlet air status in terms of Tdb/Twb).

The system of Figure 6.c uses wet-bulb temperature difference method to estimate the cooling load by sensing the dry/wet bulb temperatures across the cooling coil. The enthalpy difference with selected terminal temperature difference in water-side estimates the needed water flow rate through the coil. The conditions for the air and water can be met with this type of system but this system needs extra sensing devises such as the wet bulb temperature sensors.

With one of these three systems, the temperature of water returning to the plant can be controlled so that chiller performance and capacity can be maintained.

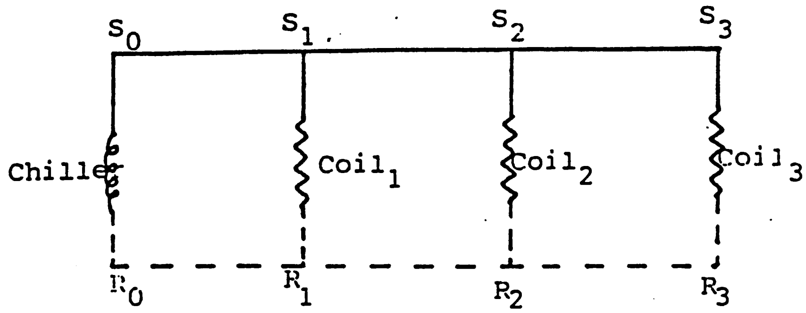
2.3 Distribution Pipings

Distribution lines play an important role in large central chilled water systems because they serve as a link or bridge between the central chilled water plant and the buildings. The three kinds of distribution loop systems have been described previously.

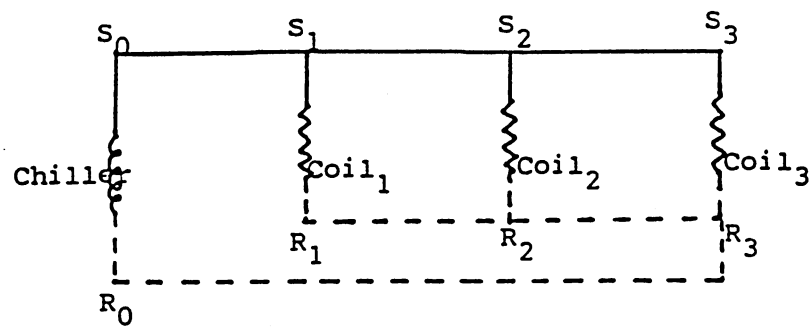
The direct return system is a two pipe system. One pipe comes from the central plant and goes to the farthest branch and the other is the return to the main plant. The branch pressure drop between the supply and the return lines near the plant is, therefore, very large compared to the one at near the farthest end. The major disadvantage of this system is the difficulty of flow control in each branch because of the variations in their pressure drops with locations and with load. The flow rate at the end point is very crucial for the satisfactory operation of the distribution system. At OSU this is the point at which a pressure is maintained to control the main distribution pumps.

A reverse return system allows the same pressure drop for each branch regardless of the location of the branches. It is easier to control flow rates in each branch than with the direct return system. The cost is greater than the direct return system since more piping is required. It is difficult to design a large and complex central distribution system as a reverse system when more than one branch are involved.

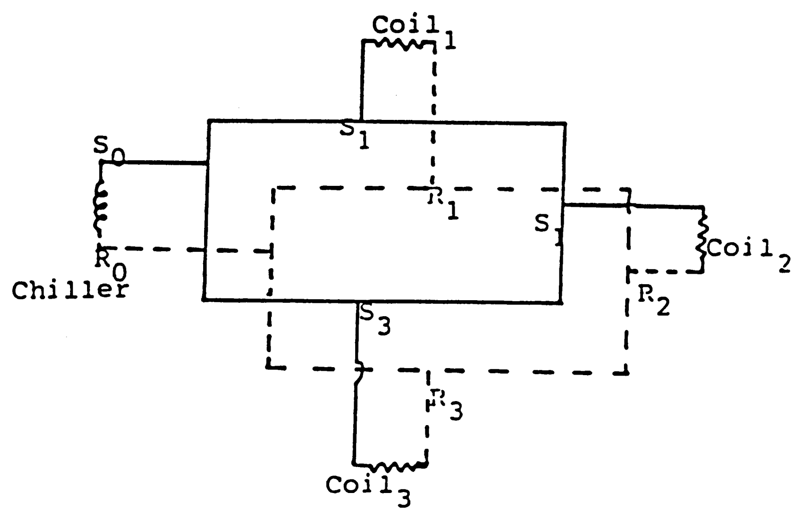
Ring type systems permit the flexibility for future expansion. Haines (50) and Hansen (6) recommended this type of system over the others because of its controllability and the reliability. The major advantage of this system is the possibility of loop isolation. Any loop of more importance may be served separately by using an isolated loop.



(A) DIRECT RETURN SYSTEM



(B) REVERSE RETURN SYSTEM



(C) RING TYPE SYSTEM

Figure 7. Three Distribution Piping Systems

It is operated very stably whenever or wherever an accident occurs in a local loop. This loop system has the highest central cooling efficiency, and therefore, cools more with less chilled water being pumped from the central location.

2.4 Valves

Another important component of the CCW system is the control valve. The proper selection and sizing of valves for the control of chilled water flow requires an understanding of the characteristics of both the valve and the system in which it is to be used. An important control valve characteristic is the valve coefficient, Cv, discussed in this section. The shape of the valve characteristic curve is very important to the successful operation of the system. A pressure drop ratio, r, is used to investigate the system effect and to help in valve selection. The factor r is defined as the pressure drop across a valve over the pressure drop across a system except the valve. This section will primarily discuss valve types and sizing method. The selection procedure provided in this section is based on both steady-state analysis and dynamic analysis. Information on selecting the gains of the valve and controllers can be obtained in Shinskey's book (65) using the concept of dynamic analysis. The effect of the valve gains on the dynamic performance of the system is discussed later in this section.

In terms of lift-flow characteristic curves, there are basically two types of valves: linear and equal-percentage. Figure 8 shows the lift-flow characteristic curves for such valves. Linear valve has a higher valve gain than the equal-percentage valve at the low valve openings. A precise valve with higher value of rangeability (maximum

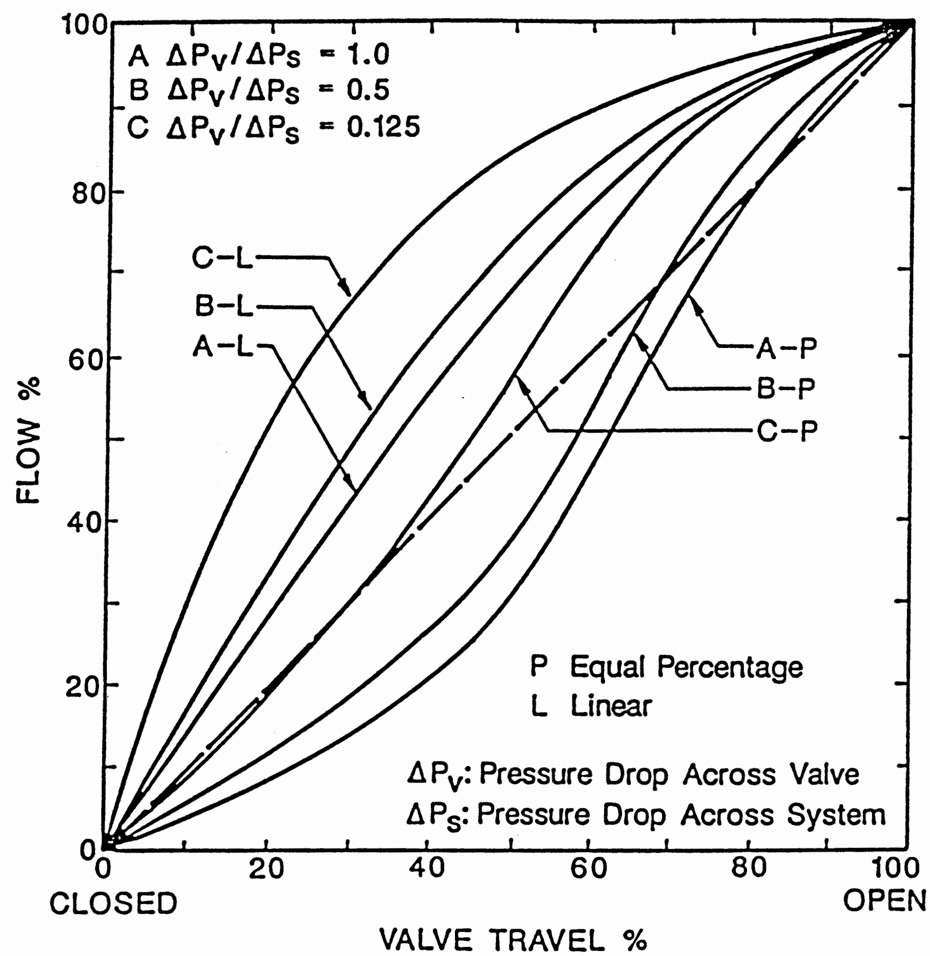


Figure 8. Valve Characteristic Curves

over minimum controllable flow) is usually recommended based on a steady-state analysis. However, a dynamic analysis indicates that an equal-percentage valve is strongly recommended to improve the dynamic system performance because of the lower gains at the lower load.

A linear type valve is usually not recommended in CCW systems where low loads can occur because of the higher flow over little valve movement at low load as compared to the equal-percentage type valve. A high valve gain in conjunction with a set of high gain of controller can result in unstable operations. This can result in the problems like chattering valves, which is one of the major causes of valve failures.

The observations made by Haines (51) and Hansen (4) are:

- The linear type valve does not control as precisely as the equal-percentage valve especially at partial load.
- The equal-percentage valve with a design ratio of pressure drop, r, of 0.5 or higher has a nearly linear effect on flow at high flow rates.
- A low design ratio of pressure drop, r, causes the equal-percentage valve to approach the shape of the linear valve curve.
- Optimum value of r for controllability is regarded as 0.5 as stated in references (4), (29), and (51).

For reasons given above, an equal-percentage valve is generally preferred for a fine control purpose. The size of the building pump can be selected with the largest design pressure drop across the coil control valve and the proposed ratio, r. For heating purposes, valves are normally open (N.O.) while the ones used for cooling purposes are normally closed (N.C.) to avoid the excessive use of chilled water.

But, the coil control valve type installed in the Engineering North building was N.O. to ensure flow to some rooms (such as computer rooms) in case of failure of air pressure system.

The procedure for sizing a valve is described as follows. The coil control valve is designed to control the water flow rate of a coil to meet the changing thermal loads. The problem of selecting and sizing a coil control valve or building control valve involves the following:

- Flow and Pressure Information

  maximum and minimum flow rate at minimum and maximum pressure

- Valve Characteristics

  port style

  flow vs lift curves

  rangeability (maximum/minimum controllable flow)

  maximum capacity, Cv

- System Effect

  installation effect

  pressure ratio, r

- Sizing Calculation

  manufacturer's table (size)

  computer, nomogram, and formula

The procedure to size a valve can be summarized as follows:

1. estimate a total system terminal pressure drop, DPt
2. select a minimum pressure drop across a valve, DPv

$$DPv = \left(\frac{r}{r + 1}\right) * DPt \qquad (2.1)$$

where r is the pressure drop ratio

3. obtain a maximum flow rate, FRv

$$FRv = DF * SF * FRd \tag{2.2}$$

where

DF .. Diversity factor

1.0 for CCV and 0.65 for BCV

SF .. safety factor, 1.25

FRd .. design flow rate

coil flow for CCV

sum of coil flows for BCV

4. select a valve size from a manufacturer's catalogue (select the smaller size when the design point falls on the point between two valve sizes).

As an example, Figure 9 shows valve curves of Staefa Control System (SCS) reproduced in English system of units. Table I contains the data for the SCS valve sizes. Table II summarizes the valve selection for Agriculture (AG) Hall building based on the SCS valve data. The design pressure drop, DPt, across the main distribution line was assumed as 25 psi and 0.5 of r was used to select the control valves. But, this valve selection procedure did not take into account the dynamic effects of the system components.

A dynamic system with the following components: actuator, valve, sensor, and controller can be modeled with a simulation package, PARASOL-II. For simplicity, each component may be modeled as a first order dynamic system whose time constant can be obtained from the manufacturer's catalogue. PARASOL-II was used to simulate the dynamic performance of the system after all variables were defined and initialized. The PARASOL simulation program has two simulation blocks, one is for digital controller and the other for the system of actuator,

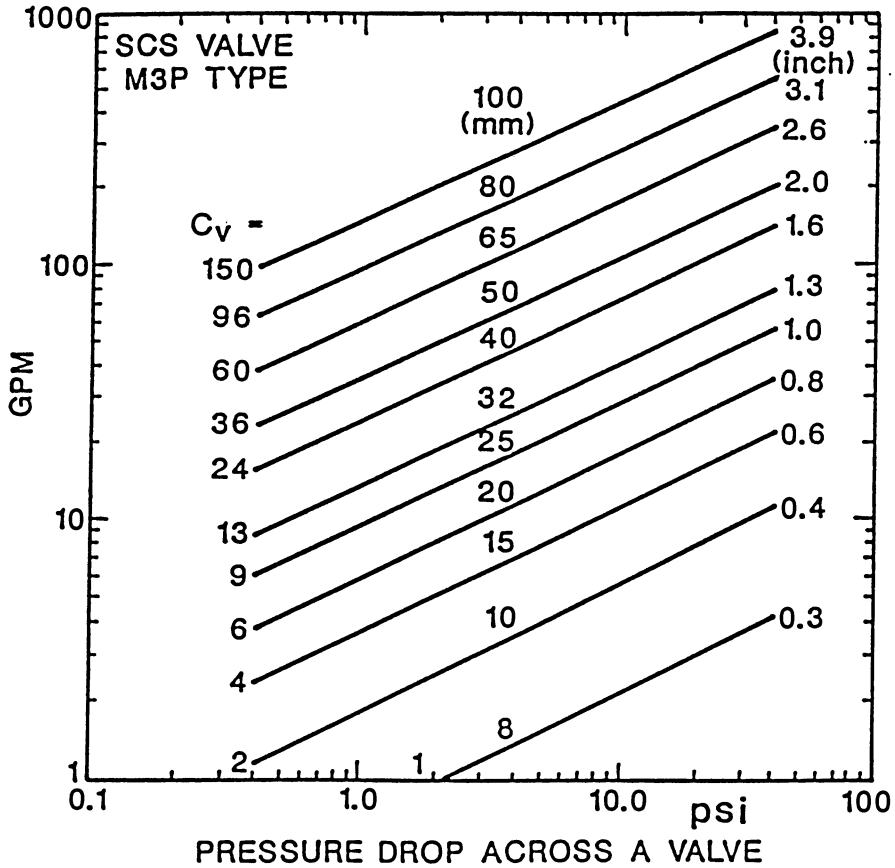


Figure 9. SCS Valve Curve

TABLE I

STAEFA CONTROL SYSTEM (SCS) VALVE DATA

Type	Size (mm)	Cv ($GPM/PSI^{0.5}$)
M3P08	08/15	1
M3P10	10/15	2
M3P15	15	4
M3P20	20	6
M3P25	25	9
M3P32	32	13
M3P40	40	24
M3P50	50	36
M3P65F	65	60
M3P80F	80	96
M3P100F	100	150

TABLE II

AGRICULTURE HALL VALVE SELECTION

Coil I.D.	Design gpm	Design ft water	Control Valve, Cv	Scs Valve Size, mm(in)
1 (N)	36.3	8.6	16	32 (1.26)
2	60.0	19.6	27	40 (1.57)
3	20.4	2.4	9	25 (0.98)
4	30.8	3.2	14	32 (1.26)
5	20.4	2.4	9	25 (0.98)
6	30.8	3.2	14	32 (1.26)
7	20.4	2.4	9	25 (0.98)
8	30.8	3.2	14	32 (1.26)
9	27.1	2.0	12	32 (1.26)
10	47.4	6.4	21	40 (1.57)
11	55	4	25	40 (1.57)
12	55	5	25	40 (1.57)
13	45	3	20	40 (1.57)
14	57	7	25	40 (1.57)
15	74	8	33	50 (1.97)
16 (E)	88	10	39	50 (1.97)
17	180	15	81	80 (3.15)
18	175	21 *	78	80 (3.15)
19	165	20	74	80 (3.15)
20	140	9	63	65 (2.56)
21	110	5	49	65 (2.56)
22 (H)	95	12	42	65 (2.56)
23	30	12	13	32 (1.26)
	1,593.4	21 (9.09 psi)		100 (3.94)

valve, process, and sensor.

Sampling time was given as 0.1 seconds based on a principle called Nyquist's Sampling Theorem:

> "If an analog signal is uniformly sampled as a rate at least twice its highest frequency content, then the original signal can be reconstructed from the samples."

Other operating parameters for the dynamic simulation are listed as follows:

actuator gain (inch/mA), gna,	...	0.2
actuator time constant (sec), toua,	...	1.0
valve gain (gpm/inch), gnv,	...	10.0
valve time constant (sec), touv,	...	1.0
process gain (F/gpm), gnv,	...	0.0267
process time constant (sec), toup,	...	1.5
sensor gain (mA/F), gns,	...	0.25
sensor time constant (sec), tous,	...	0.5

Initial gains for the PI controller simulated were determined based on Ziegler-Nichols' open loop step response method (66) experimentally. For most HVAC control applications, the two-mode PI control gives good results. The initial gains for PI control may be calculated from the following formulas (43):

$$Kp = (M*F*0.4)/(R*L) \tag{2.3}$$

$$Ki = Kp/(0.4*L) \tag{2.4}$$

where

M = Actuator step change (0.0 to 1.0)

R = Rate in measured variable units per second

F = Full range measured variable units (i.e., span)

L = Process lag time in seconds

From the sample step response test of the building control valve in the Engineering North building, the following values were chosen:

R = 3 F / 3 sec = 1 (F/sec)

L = 1 sec

F = 47-44 = 3 F

M = 0.1 (10 percent change assumed)

Substituting these into the gain factor equations (2.3) and (2.4) gives:

Kp = (0.1*3*0.4)/(1*1) = 0.12

Ki = (0.12)/(0.4*1) = 0.3

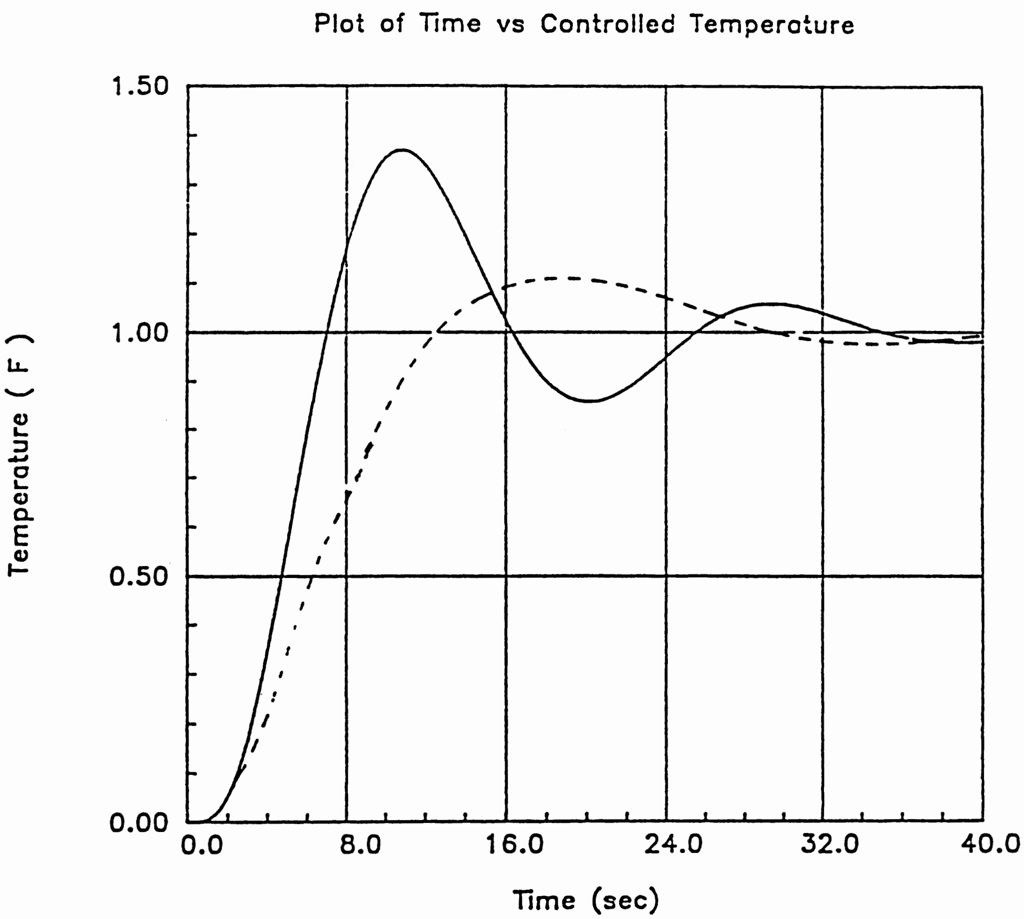
A PARASOL program is listed in Appendix F with the set of state variable equations for each component and controller equations. The effect of valve gains, gnv's, on dynamic system performance was tested by changing the values from 10 to 20. Unit step input was introduced to see the time response of the controlled variable, the temperature at the secondary supply. Figure 10 shows the result of the simulation to study the effect of the valve gains. The valve with a lower gain such as equal-percentage valve runs more stably with better dynamic performances as shown in Figure 10. A manual adjustment of the gain factors is required for new sample interval. These gains are easily calculated using the following formulas (43):

$$Kp_n = Kp_o \tag{2.5}$$

$$Ki_n = Ki_o * (TS_n/TS_o) \tag{2.6}$$

The suffix "n" denotes the new term and "o" the old term. TS stands for the sampling time in seconds.

To run the example system in Texas Instrument (TI) personal



Valve gain, gnv

______ 20

\- - - 10

Figure 10. The Effect of Valve Gain on Dynamic System Performance

computers, the following steps are required. First, a file called "TEST.P" must be edited by suing an editor such as Turbo. A sample TEST.P is listed in the last page of Appendix F. Then, insert the PARASOL program disk into a TI computer. A set of important commands is given by:

```
A>par2 run.p (to run PARASOL simulation program)
.
.
.
    P2>$ifil
        filename: B:TEST.P
    P2>$ 0.1 40 run (run with 0.1 sec time step till 40 sec)
    -   -
    -   - (results)
    -   -
```

To change the operating parameters for the same system, the following commands are mostly used:

```
P2> 10 @ gnv (to change the valve gain into 10)
P2>  .2 @ Ki (to change the integral gain value into 0.2)
P2>  .1 20 run (to run the modified system)
-   -
-   - (results)
-   -
P2>$exit (get out of the program mode)
```

CHAPTER III

MODELING OF CENTRAL CHILLED WATER SYSTEM

3.1 Central Chilled Water System Components

Modeling refers to a mathematical representation of a system with all the necessary parameters and variables. Some of the variables can be input (assumed or known) and some are outputs (unknown). The behavior of a system may usually be represented by a set of equations relating the variables for a given set of parameters. Mathematical modeling of each of the components and loops constituting a large central chilled water system will discussed in this chapter.

A central chilled water system is composed of three principal mechanical elements: piping systems, heat exchangers, and prime movers. Another important element is the control valve. Modeling technique for the piping systems, heat exchangers, and prime movers will be described in the following sections. Two different approaches (direct-substitution method and iterative method) will be outlined with a discussion of the general laws that are applied to simulate a large system loop.

3.1.1 Piping Systems

Energy system using a fluid as the working medium usually utilize some form of conduit or pipe. Piping systems used in conjunction with energy systems may range in complexity from simple to complex. Regardless of the complexity, relatively simple principles may be used to describe the piping system mathematically.

The pressure drop due to friction or elevation in pipes, fittings, and valves may be derived and obtained from references such as Tranes' (23). There are three primary elements which contribute to the pressure drop in a piping system; they are:

- Straight Pipes and Fittings
- Heat Exchangers (Chillers and Coils)
- Control Valves

The pipes are characterized by means of equivalent length and diameter. All fittings in the system are transformed into equivalent length using Tables from references (23) and (18). See, for example, Table III. The equivalent lengths of fittings and valves are added to the pipe length to give the total pipe equivalent length (EL) in feet for a specified section. These equivalent lengths must be calculated based on the same diameter. The pressure drop of water flowing through a straight pipe or pipe with fittings may be expressed by the equation

$$Pi - Pj = C * (W^2) \tag{3.1}$$

where

Pi pressure at pipe inlet, psi

Pj pressure at pipe outlet, psi

W water flow rate, gpm

C pressure drop coefficient, psi/(gpm^2)

The pressure drop coefficient C may be evaluated by the equation

TABLE III

FITTING LOSSES IN EQUIVALENT FEET OF PIPE (18)

NOMINAL PIPE OR TUBE SIZE (In.)	SMOOTH BEND ELBOWS: 90° Std*	90° Long Rad.	90° Street*	45° Std*	45° Street*	180° Std*	SMOOTH BEND TEES: Flow-Thru Branch	Straight-Thru Flow: No Reduction	Reduced ¼	Reduced ½
3/8	1.4	0.9	2.3	0.7	1.1	2.3	2.7	0.9	1.2	1.4
½	1.6	1.0	2.5	0.8	1.3	2.5	3.0	1.0	1.4	1.6
¾	2.0	1.4	3.2	0.9	1.6	3.2	4.0	1.4	1.9	2.0
1	2.6	1.7	4.1	1.3	2.1	4.1	5.0	1.7	2.3	2.6
1¼	3.3	2.3	5.6	1.7	3.0	5.6	7.0	2.3	3.1	3.3
1½	4.0	2.6	6.3	2.1	3.4	6.3	8.0	2.6	3.7	4.0
2	5.0	3.3	8.2	2.6	4.5	8.2	10	3.3	4.7	5.0
2½	6.0	4.1	10	3.2	5.2	10	12	4.1	5.6	6.0
3	7.5	5.0	12	4.0	6.4	12	15	5.0	7.0	7.5
3½	9.0	5.9	15	4.7	7.3	15	18	5.9	8.0	9.0
4	10	6.7	17	5.2	8.5	17	21	6.7	9.0	10
5	13	8.2	21	6.5	11	21	25	8.2	12	13
6	16	10	25	7.9	13	25	30	10	14	16
8	20	13	—	10	—	33	40	13	18	20
10	25	16	—	13	—	42	50	16	23	25
12	30	19	—	16	—	50	60	19	26	30
14	34	23	—	18	—	55	68	23	30	34
16	38	26	—	20	—	62	78	26	35	38
18	42	29	—	23	—	70	85	29	40	42
20	50	33	—	26	—	81	100	33	44	50
24	60	40	—	30	—	94	115	40	50	60

$$C = \frac{(0.0311) * ff * EL}{ED^5 * 2.31} \tag{3.2}$$

where

ff friction factor

EL equivalent length

ED equivalent diameter

The conversion factor 0.0311 in Equation (3.2) is derived from the empirical Darcy-Weisbach equation. The other conversion factor 2.31 is to change the units of pressure from feet of water into psi.

The value of ff must be determined first in order to compute the pressure drop coefficient in Equation (3.2). The equation that determines the value of ff depends on the type of flow in the section (laminar or turbulent). The program from this study uses Reynolds number (Re) as the criterion to determine whether the flow is laminar or turbulent. Reynolds number is defined as:

$$Re = \frac{Ro * V * D}{u} \tag{3.3}$$

where

Ro water density, slugs/ft^3

V water velocity, ft/sec

D pipe diameter, ft

u water viscosity, lbf-sec/ft^2

If we assume the overall water temperature throughout the system to be 47.5 F, water density 62.4 lbm/ft^3, and water viscosity to be $28.6*10^{-6}$ lbs*sec/ft^2, Equation (3.3) becomes

$$Re = 0.066 * W * 10^5 / 2.86 / ED \tag{3.4}$$

where

W water flow rate, gpm

ED equivalent pipe diameter, in

Whether the flow is laminar or turbulent depends on the value of Re. If Re < 2,000, then the flow is laminar so that the friction factor is

$$ff = 64 / Re \tag{3.5}$$

If Re > 2,000, then the flow is considered as turbulent. The ff, in this case, can not be expressed explicitly. The ff may be calculated iteratively using Colebrook equation (8) with assumed pipe roughness, e. The Colebrook equation is

$$\frac{1}{\sqrt{ff}} = \log \left(\frac{e}{3.7 * ED} + \frac{2.51}{Re\sqrt{ff}}\right)^{-2} \tag{3.6}$$

where

ED section diameter, in

e pipe roughness

Another equation for the ff is implicit which uses Churchill's equation (10). This is also very useful because it covers the entire range of Re numbers of practical interest. The equation is

$$A = \left[2.457 * \ln \frac{1}{\left(\frac{7}{Re}\right)^{0.9} + 0.27 * \left(\frac{e}{ED}\right)}\right]^{16}$$

$$B = \left(\frac{37530}{Re}\right)^{16}$$

$$ff = 8 * \left[\left(\frac{8}{Re}\right)^{12} + (A + B)^{-1.5}\right]^{\frac{1}{12}} \tag{3.7}$$

The relative roughness, e depends on a pipe material. At the CCW system of the OSU, the pipes between 30 to 10 inch are mostly made of concrete with epoxy inside (about 0.001 inch roughness) and the pipes below 8 inch are plastic tube (62). During the development of the computer program, the calculation of the friction factor naturally lends itself to being one module of the model. A general friction factor subroutine may be written and checked out quite independently with figures acquired from several references (5), (8), and (23).

3.1.2 Heat Exchangers

The specification of chillers usually includes the flow and pressure data so that equivalent length may be evaluated for the selected flow rate of a chiller (See Appendix D). Generally, the pressure drop through the chillers is required to be constant. The pressure drop across a coil will be assumed to vary according to the well-known law:

$$GPMa = GPMd * \left(\frac{DPa}{DPd}\right)^{0.5}$$

$$DPa = DPd * \left(\frac{GPMa}{GPMd}\right)^{2} \qquad (3.8)$$

where

GPMa actual gallons per minute

GPMd design gallons per minute

DPa actual pressure drop

DPd design pressure drop

For both coils and chillers the thermal load in terms of cooling ton to chiller and coil is expressed as:

$$\text{Tons} = \frac{(\text{Ti} - \text{To}) * \text{GPM} * (12000)}{500} \tag{3.9}$$

where

Tons cooling load

Ti water inlet temperature, F

To water outlet temperature, F

Figure 11 shows the heat transfer characteristics of a typical coil such as a variable water flow system. At partial load, there is a big output change for small input valve movement whereas little change is shown near the design condition. This is why most coils show instability at low partial loads.

The wet-bulb temperature method (30) can estimate the thermal load applied to the coils. This procedure requires the assumption that air flow is constant near design cfm. For a specified cfm, the cooling load in terms of Btuh can be found from the equation:

$$\text{Btuh} = 4.5 * \text{cfm} * \text{DH} \tag{3.10}$$

where

4.5 constant

cfm design air flow rate

DH enthalpy difference

The enthalpy difference in the air-side is found by plotting the entering and leaving wet-bulb and dry-bulb temperatures on the psychrometric chart. Once the total load in Btuh is known, the required water flow rate is calculated from an equation:

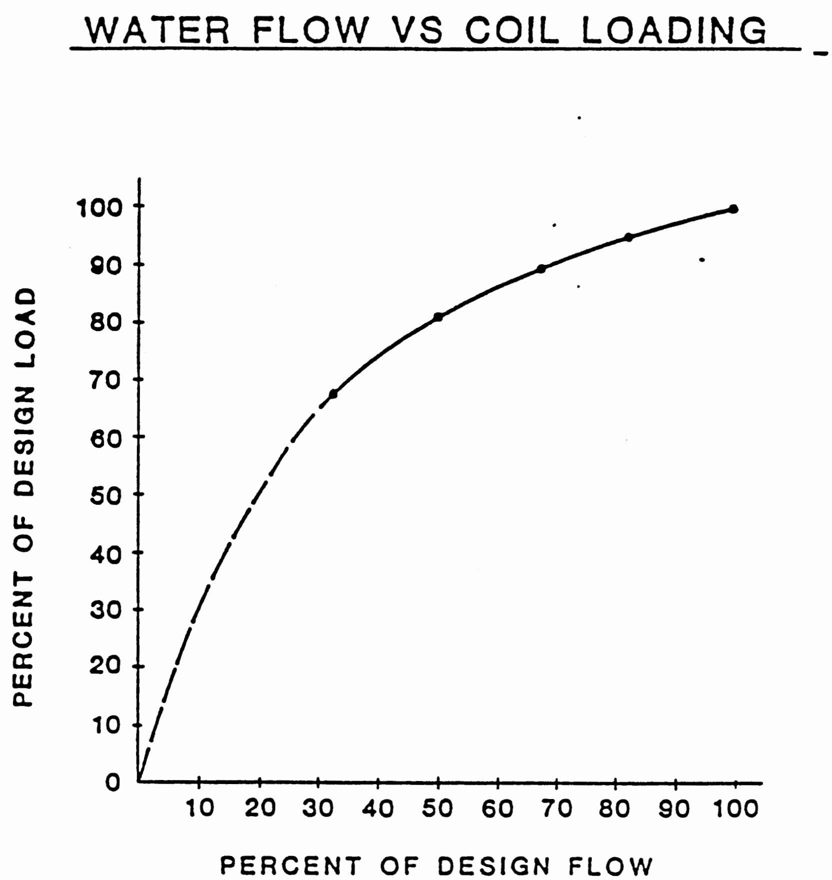


Figure 11. Coil Characteristics

$$gpm = \frac{Btuh}{(500) * DT} \tag{3.11}$$

where

Btuh cooling load

DT temperature difference between inlet and outlet, F

Equation (3.11) shows that the constant flow system would produce a varying outlet temperature as the load varies. If the outlet temperature of a coil is to be held constant, then, obviously the flow rate of water must change as the load changes.

Equipment performance data should supply at least the following information:

Chiller - design flow rate (gpm)
design pressure drop (psi)
design cooling load (ton)

Coil - cooling load (ton)
design pressure drop (psi)
design flow rate (gpm)

Data for several coils were obtained from the architect room in the Physical Plant and the Power Plant of the OSU.

For instance, a coil existing in the Engineering North building is selected to illustrate the load estimation method with Equations (3.10) and (3.11). The coil is designed with the following data:

design air flow rate = 10,500 cfm

air inlet condition = 83.0/68.6 F (dry/wet bulb temperature)

air outlet condition = 63.1/60.0 F

The enthalpy difference may be calculated by the following two ways:

one is to use the psychrometric chart as shown in Figure 12 and the other way is to use the equations available in reference (2).

Using the chart, the enthalpy difference DH can be obtained as 9 Btu/lbma because 34 (inlet) minus 25 (outlet) Btu/lbma gives 9 Btu/lbma. At off-design condition such as 78/65 F of inlet air, the enthalpy difference DH can be 5 Btu/lbma. Therefore, the loads and the needed flow rates for each case can be estimated as follows:

$$\text{Btuh} = 4.5 * \text{cfm} * \text{DH}$$

$$= 4.5 * 10{,}500 * 9 = 425{,}250 \text{ Btu/h} = 35.4 \text{ ton}$$

$$\text{gpm} = \frac{\text{Btuh}}{(500)\ \text{DT}}$$

$$= (425{,}250)/500/12 = 70.9 \text{ gpm}$$

(on-design)

$$\text{Btuh} = 4.5 * 10{,}500 * 5 = 236{,}250 \text{ Btu/h} = 19.7 \text{ ton}$$

$$\text{gpm} = (236{,}250)/500/12 = 39.4 \text{ gpm}$$

(off-design)

Using the equations without the chart, the following steps should be employed to calculate the loads.

1. Estimate the saturation pressure (Pws) with the wet-bulb temperature (Twb)
2. Calculate the humidity ratio, W
3. Calculate the enthalpies for the inlet/outlet conditions
4. Calculate the enthalpy difference, DH

The saturation pressure over liquid water for the temperature range of 32 to 392 F is given by:

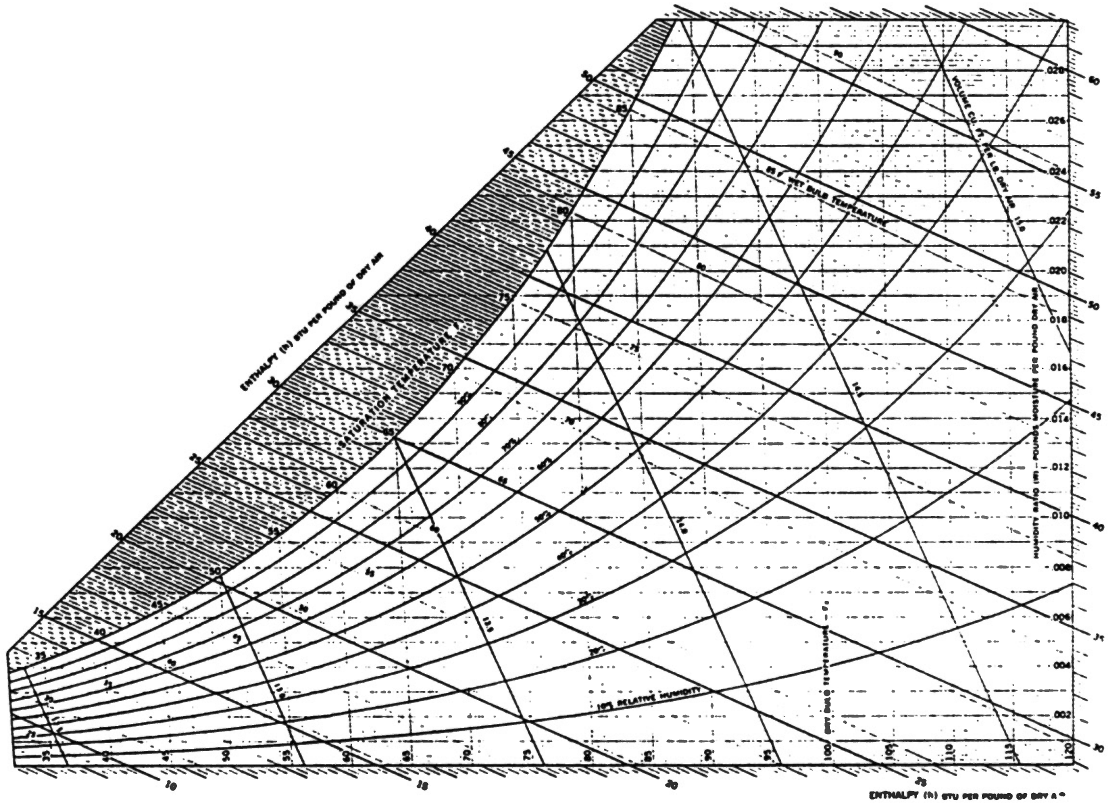


Figure 12. Psychrometric Chart (5)

$$\ln(Pws) = C1/T + C2 + C3*T + C4*T^2 + C5*T^3 + C6*\ln(T) \qquad (3.12)$$

where

C1 = -10,440.4

C2 = -11.2946669

C3 = -0.02700133

C4 = $0.12897060*10^{-4}$

C5 = $-0.24780680*10^{-8}$

C6 = 6.5459673

ln = log e

Pws = saturation pressure, psia

T = absolute temperature, R (F+459.67)

The humidity ratio W is given by:

$$W = 0.62198*Pws/(P-Pws) \qquad (3.13)$$

where

Pws = saturation pressure, psia

P = atmospheric pressure, psia

The moist air enthalpy then becomes:

$$h = 0.240*t + W*(1061+0.444*t) \text{ (Btu/lbm)} \qquad (3.14)$$

where

t = temperature, F

W = humidity ratio

Using the above equations, the following enthalpy differences were obtained:

DH_o = ho = 33.7 - 25.0 = 8.7 (Btu/lbm)

DH_n = hn = 29.5 - 25.0 = 4.8 (Btu/lbm)

where

DH_o enthalpy difference at the design condition

DH_n enthalpy difference at another condition

The advantage of using the equation is that the enthalpy may be obtained with the different operating pressure other than the atmospheric pressure with Equation (3.13).

3.1.3 Valves

To model a control valve, the information valve type and capacity is needed. Most manufacturers publish valve capacity tables in terms of a flow coefficient Cv. This is defined as "flow rate in gallons of water that will pass through the valve in one minute at one psi pressure drop." The flow rate at a pressure drop, DP is assumed to be predicted by the formula:

$$W = Cv * (DP^{0.5}) \tag{3.15}$$

where

DP desired pressure drop across the valve

Cv valve coefficient, gpm/(psi^{05})

Valve capacity tables from a manufacturer's catalogue should show Cv and flow rate at various pressure drops. The operating pressure other than the design condition is given by:

$$DPa = DPd * (FRa/FRd)^2 \tag{3.16}$$

where

DPa = actual pressure drop across the valve, psi (off-design)

DPd = design pressure drop across the valve, psi (on-design)

GPMa = actual flow rate, gpm

GPMd = design flow rate, gpm

The characteristic curve of a control valve can be obtained as in Figure 8. As stated before, the gain at instant operating point as in

Figure 8 is very important for a better dynamic operation of the system including the valve. The lower the valve gains are, the better it performs at low partial loads as shown in Figure 10.

3.1.4 Prime Mover

An integral part of many energy systems is the pumps. Pumps come in an amazing variety of sizes and types and present many diverse characteristics. A pump does not operate in isolation, and the complementary characteristics of a piping system into which a pump is placed are of equal concern. It is the combination of pump and system characteristics that determines at what level the pump as well as the system will operate. Figure 13 shows the interaction between the piping system and the pump. The system curve number 1 is representing a minimum flow and the curve number 2 for a possible maximum flow curve, which is depending on the layout and the components of a piping system. The operating point for each case in terms of pressure and flow can be found by a iterative method.

Centrifugal pumps will be considered in this study. Extensive use will be made of manufacturer's performance data (Appendix D) and specifications. Appendix C summarizes how to obtain the equation for a pump using a curve fitting technique. Appendix D includes available central chilled water equipment performance data. There are many sources of pump information in references (24) and (25).

Stoecker (20) explains a curve fitting technique to model the performance of a pump. The pressure rise is a function of the flow rate and the function is this study was third degree polynomial of the flow rate for a typical pump representation. The form of the equation used

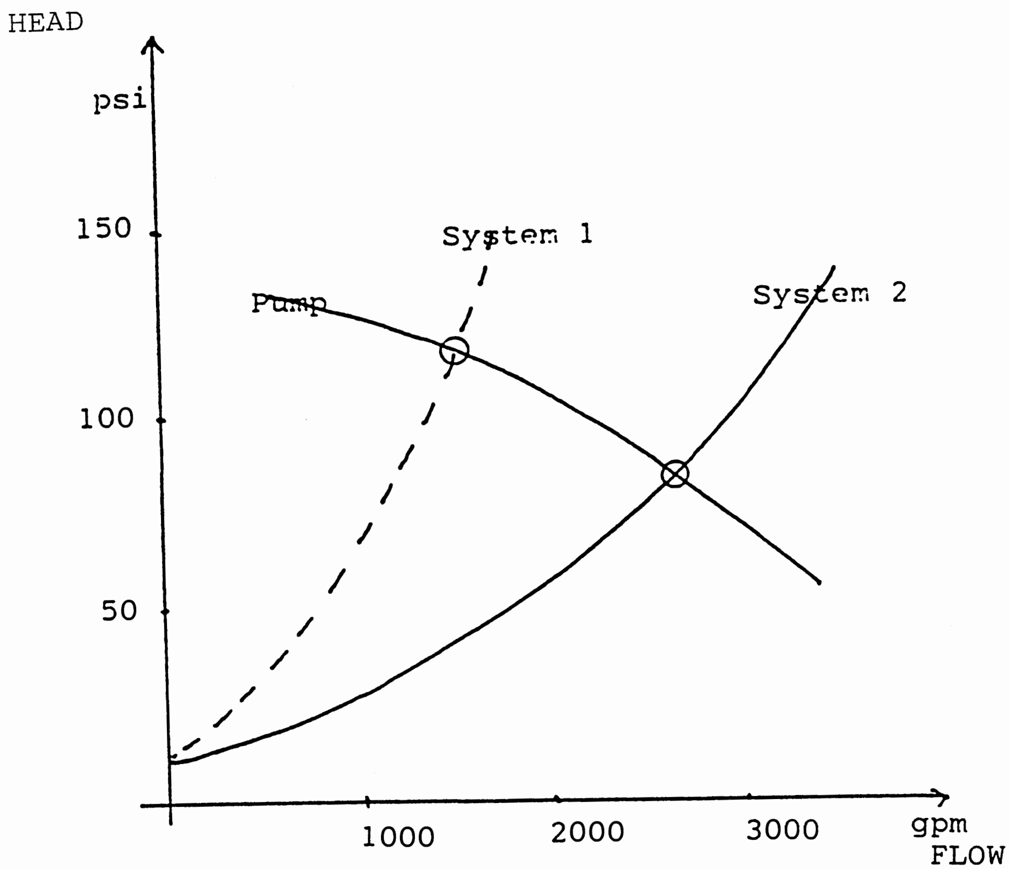


Figure 13. System and Pump Characteristics

to describe the pressure rise across a constant speed pump is:

$$P_2 - P_1 = P_{c1} + P_{c2}*W + P_{c3}* W^2 + P_{c4}*W^3 \qquad (3.17)$$

where

P_2 pressure at pump inlet, psi

P_1 pressure at pump outlet, psi

W water flow rate through pump, gpm

P_{ci}'s pump coefficients

Appendix C illustrates how to determine the pump coefficients, P_{ci}'s, using four data points and the routine GAUSSY.

A variable speed pump can vary the pressure rise as the speed of the pump changes. In this study a pump controller has been chosen that will cause the pump to maintain a constant pressure rise at the pump regardless of the flow rate. The equation used to describe this pressure rise is

$$P_2 - P_1 = DELP \qquad (3.18)$$

where

P_2 pressure at pump outlet, psi

P_1 pressure at pump inlet, psi

DELP a constant pressure, psi

By varying the flow rate to keep the pressure constant, the variable speed pump has almost flat flow-pressure characteristic. Mostly this type of pump is recommended as the building pump for a better variable system because the pressure rise is relatively constant even at the low load.

3.2 Central Chilled Water System Loops

To model a hydronic loop, a set of many equations is required to represent all of the piping loop elements mathematically. Each loop can be divided into small sections by nodes. The number of sections and nodes will influence the number of the variables. Constant values of the parameters are inputted to run the simulation under given conditions. A numerical method is used to find a set of variables which satisfy all the equations in sections.

The major variables are pressure and flow in specified sections. Temperatures may then be calculated in any of the sections by utilizing energy balance principles.

The first chiller loop model requires the chiller and distribution pump operation status. The pressure drop and total demand flow of a campus load need to be assumed to study the central chiller loop mainly. The output variables are the pressure drops, flow rates, and temperatures to meet all the equations satisfactorily.

The operational parameters for the building loop are: 1. coil thermal load, 2. coil control valve type and characteristics, 3. piping arrangement, and 4. pump type and characteristics. The hydraulic performance of the building loop is mainly concerned due to the parametric changes.

The thermal load variational effect can be examined with a diversity factor, which was defined by Carlson (12) as the actual load over the design load to the secondary loop. The secondary overpumping effect can be studied with a balance factor which is the actual flow rate over the design flow rate for the secondary building loop. In this thesis, a cooling factor (CF) was introduced to consider the both effects at the same time, which is defined as the supplied cooling over the utilized cooling to a building loop in terms of cooling ton.

Ideally, a cooling factor close to one is desirable but the value tends to be greater because of the high degree of overpumping to the coils or short circuiting through the common pipe. The combined effect may also be visualized and checked with the simulated results and the results of other investigators to improve the building loop performance.

The major objective of this thesis is to develop a method to simulate a total campus distribution system and to predict the overall behavior of the large central chilled water system. It seems logical to first study the building loop and the chiller loop and use the results in the simulations for the distribution system simulation and prediction of overall system behavior.

A general procedure is presented for building loops, chiller loops, and distribution loops as follows:

For building loop simulation:

1. assume thermal loads in each coil and characteristic of coil and entire piping system
2. assume the temperature difference in building and distribution loop, DTs and DTp
3. calculate coil flow rates, gpm
4. calculate coil heads, psi
5. calculate total secondary flow
6. determine the needed primary flow

For the central chiller loop, the possible combinations of the central distribution pump operation may be obtained. A series of pump curves can be generated depending on the status of the pumps turned on or off.

For distribution loop simulation:

1. input the branch flow rates for loads assumed above
2. calculate total campus demand, gpm
3. input the assumed status of distribution pumps
4. calculate the pressure difference supply and return at the plant, psi
5. calculate the section head losses, psi
6. predict the pressure drop required in building

Other information may be found by running the building loop simulation program. The head loss for the coil control valve may be estimated by the above method to meet a given thermal load condition. The operational parameters for the distribution model are distribution pump operation status and the required amount of primary water to each branch, which should come from building loop models. Then, the distribution loop model simulates the behavior of the hydronic system to predict pressure drops and flow rates in specified sections of the distribution lines for the given condition from the other loops.

3.3 Natural Laws

This section reviews physical laws applicable to the problem under study, piping network. The major laws are continuity law and Kirchhoff's law.

3.3.1 Continuity Law

Each specified point in the system is considered to be a node (either a two-way or three way junction). Under steady state conditions, the total mass flow rate into a node equals to the total mass flow rate out from the node:

$$(m)_{in} = (m)_{out} \tag{3.19}$$

since m = Ro * W

$$(Ro)*(W)_{in} = (Ro)*(W)_{out} \tag{3.20}$$

if the density of the water Ro is assumed to remain constant, Equation (3.16) is reduced to

$$(W)_{in} = (W)_{out} \tag{3.21}$$

Equation (3.21) leads to the three equations which are available to describe the volume flow rate balance at each point to the type of node:

$$(W)_{in} = (W)_{out} \quad \text{(two-way node)} \tag{3.22}$$

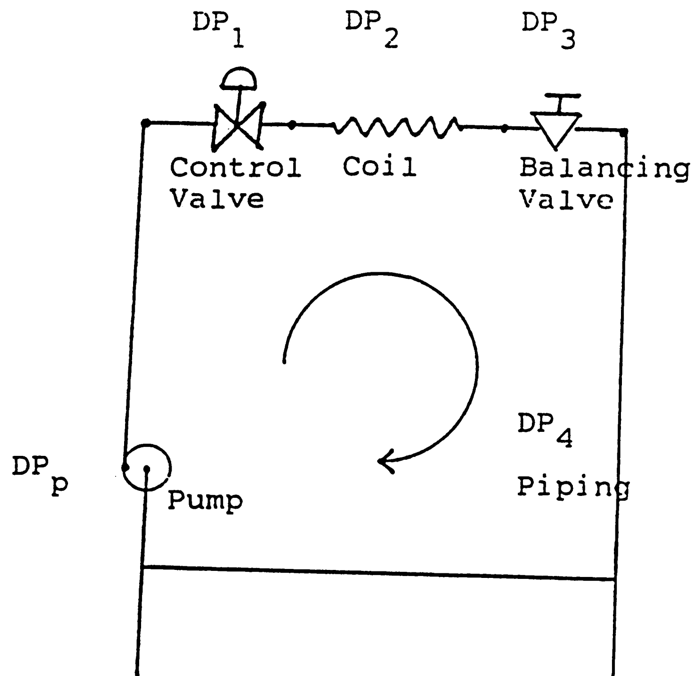
$$(W)_{in} = (W_1 + W_2)_{out} \quad \text{(three-way diverting node)} \tag{3.23}$$

$$(W_1 + W_2)_{in} = (W)_{out} \quad \text{(three-way mixing node)} \tag{3.24}$$

The user must provide the continuity equations and the following loop equations to match properly the number of equations to the number of unknowns.

3.3.2 Kirchhoff's Law

Kirchhoff's voltage law states that the algebraic sum of all branch voltages around any closed loop of a network is zero at all instants of time. This law can be applied to the hydraulic circuit because voltage and current are analogous to pressure drop and flow. Figure 14 shows a simple resistive network with pressure reference directions assigned for



$$DP_p = DP_1 + DP_2 + DP_3 \div DP_4$$

Figure 14. Kirchhoff's Law for Hydronics

the elements and clockwise loop direction selected for the application of the Kirchhoff's law. Starting at a node, we assign a positive sign to the pressure drop if the polarity marks occur in the order + to -, a negative sign for the opposite order. Thus, we write the equation:

$$-DPp + DP_1 + DP_2 + DP_3 + DP_4 = 0 \tag{3.25}$$

The pressure drops due to the pipings, fittings, and valves can be substituted for the DP_i's in Equation (3.21). The DPp stands for the pressure rise due to the pump. Interestingly, the equation for the DP_i's are of nonlinear form so that it is very difficult to solve the system equation by hand.

3.3.3 Direct-substitution Method

Two methods are available to describe a hydronic loop system. One is a direct-substitution method and the other is an iterative method. The direct-substitution method is very useful and effective when one of the system component equations is not known and must be found. The general steps were previewed in the section 3.2. The method calculates the needed information step by step to estimate the desired values.

The iterative method, which will be thoroughly explained in the next section, is suitable to a system whose component equations are all known. However, it is sometimes difficult to obtain all the component equations to be simulated. The characteristic curve of the control valve is the typical case of the situations. The direct method to find the pressure drop to meet the other conditions is very useful for the case.

In this study, the direct method was used to simulate the total campus system and the iterative method was used to verify the results from the developed models and simulations. The concept of the two methods is well applied to building loop as well as distribution loop system.

3.3.4 Iterative Method

Multivariable Newton-Raphson Method is one of the widely used iterative methods. Two methods for solving nonlinear systems are generally available: (1) successive substitution and (2) Newton-Raphson iteration. Successive substitution is the common iterative approach. In this method, values of unknown variables are assumed and an iterative process delineated that allows corrections to the assumed values until convergence is achieved. Generally, for a specific simulation problem, many successive substitution iterative paths can be developed. Some of the paths will converge rapidly, some slowly, and some will diverge. The Newton-Raphson iterative approach solves the system simultaneously and, if well behaved (all initial values well assumed), will generally avoid convergence problems. The Newton-Raphson approach will be pursued as the major technique for solving systems of nonlinear equation derived from a physical piping system.

The multivariable Newton-Raphson method is closely tied with the Newton-Raphson method for one variable (or unknown). Generally, we wish to find the correct value of x, x_c, such that

$$f(x_c) = 0 \tag{3.26}$$

since in general

$$f(x) = y(x) \doteq 0 \tag{3.27}$$

A truncated Taylor series about x_c can be written as

$$y(x) = y(x_c) + y'(x_c)(x - x_c) \tag{3.28}$$

If a trial guess x_t is made for x_c, then

$$y(x_t) = y(x_c) + y'(x_c)(x - x_c) \tag{3.29}$$

But since $y(x_c) = 0$ and since x_c is not known, Equation (3.29) can be approximated as

$$y(x_t) = 0 + y'(x_t)(x_t - x_c) \tag{3.30}$$

Equation (3.30) may be used to solve for an estimate of x_c:

$$x_c = x_t - y(x_t)/y'(x_t) \tag{3.31}$$

The above equation is the classical Newton-Raphson equation for finding the root of an equation in a single variable. Care should be exercised in using Equation (3.31), for expressions with multiple roots as convergence may be erratic or may not occur, especially if the initial trial guess is far from the correct result.

Once a set of many non-linear equations has been derived for a specific problem, the next task is to determine how to solve the set of equations by developing a computer program. The multivariable Newton-Raphson method is developed and adapted for this study, as is the single-variable Newton-Raphson, by considering a Taylor's series expansion. In this case, more than a single independent variable is present, and the more complicated series is

$$y(x_1,x_2, \ldots, x_n) = y(a_1,a_2, \ldots, a_n) + \sum_{j=1}^{n} (\frac{\partial y}{\partial x_j})_a (x_j - a_j)$$

$$+ \frac{1}{2} \sum_{i=1}^{n} \sum_{j=1}^{n} (\frac{\partial^2 y}{\partial x_i \partial x_j})_a (x_i - a_i)(x_j - a_j) + \tag{3.32}$$

Consider a system of three equations in the independent variables x_1, x_2, and x_3:

$$f_1(x_1,x_2,x_3) = 0$$

$$f_2(x_1,x_2,x_3) = 0$$

$$f_3(x_1,x_2,x_3) = 0 \tag{3.33}$$

The correct solution point is (x_{1c},x_{2c},x_{3c}), and the initially assumed solution point is (x_{1t},x_{2t},x_{3t}). The Taylor series for $f_1(x_{1t},x_{2t},x_{3t})$ can be written as

$$\begin{aligned} f_1(x_{1t},x_{2t},x_{3t}) = {} & f_1(x_{1c},x_{2c},x_{3c}) \\ & + (\frac{\partial f_1}{\partial x_1})_c (x_{1t} - x_{1c}) \\ & + (\frac{\partial f_1}{\partial x_2})_c (x_{2t} - x_{2c}) \\ & + (\frac{\partial f_1}{\partial x_3})_c (x_{3t} - x_{3c}) + \ldots \end{aligned} \tag{3.34}$$

and in similar fashion for $f_2(x_{1t},x_{2t},x_{3t})$ and $f_3(x_{1t},x_{2t},x_{3t})$. Making the approximation $(\frac{\partial f_1}{\partial x_j})_c - (\frac{\partial f_1}{\partial x_j})_t$ and realizing that $f_1(x_{1c},x_{2c},x_{3c})$ is zero, Equation (3.34) becomes

$$f_1(x_{1t},x_{2t},x_{3t}) = (\frac{\partial f_1}{\partial x_1})_t\ (x_{1t} - x_{1c})$$
$$+ (\frac{\partial f_1}{\partial x_2})_t\ (x_{2t} - x_{2c})$$
$$+ (\frac{\partial f_1}{\partial x_3})_t\ (x_{3t} - x_{3c}) + \ldots \tag{3.35}$$

and f_2 and f_3 appear as

$$f_2(x_{1t},x_{2t},x_{3t}) = (\frac{\partial f_2}{\partial x_1})_t\ (x_{1t} - x_{1c})$$
$$+ (\frac{\partial f_2}{\partial x_2})_t\ (x_{2t} - x_{2c})$$
$$+ (\frac{\partial f_2}{\partial x_3})_t\ (x_{3t} - x_{3c}) + \ldots$$

$$f_3(x_{1t},x_{2t},x_{3t}) = (\frac{\partial f_3}{\partial x_1})_t\ (x_{1t} - x_{1c})$$
$$+ (\frac{\partial f_3}{\partial x_2})_t\ (x_{2t} - x_{2c})$$
$$+ (\frac{\partial f_3}{\partial x_3})_t\ (x_{3t} - x_{3c}) + \ldots$$

Equation (3.35) constitute a linear system equations for $(x_{it} - x_{ic})$, i = 1, 2, 3, which can be written as a generalized matrix form:

$$\begin{bmatrix} \\ \\ DR_i \\ --- \\ DX_j \\ \\ \\ \end{bmatrix} \cdot \begin{bmatrix} (x_t - x_c)_1 \\ (x_t - x_c)_2 \\ \\ \\ \vdots \\ \\ (x_t - x_c)_n \end{bmatrix} = \begin{bmatrix} R_1 \\ R_2 \\ \\ \\ \vdots \\ \\ R_n \end{bmatrix} \tag{3.36}$$

$i = 1, 2, 3, \ldots n$

$j = 1, 2, 3, \ldots n$

where

x_t trial value

x_c corrected value

R_i residual equation

DR_i/DX_j partial derivatives

So that a set of the improved values for the variables can be calculated with the inversed equation of Equation (3.36).

$$\begin{bmatrix} (x_t - x_c)_1 \\ (x_t - x_c)_2 \\ \cdot \\ \\ \cdot \\ \cdot \\ (x_t - x_c)_n \end{bmatrix} \cdot \begin{bmatrix} \\ \dfrac{DR_i}{DX_j} \\ \\ \end{bmatrix}^{-1} = \begin{bmatrix} R_1 \\ R_2 \\ \cdot \\ \\ \cdot \\ \cdot \\ R_n \end{bmatrix} \qquad (3.37)$$

and the new corrected values for the next iterative pass become

$$x_{c1\ new} = x_{t1\ old} - (x_t - x_c)_1$$

$$x_{c2\ new} = x_{t2\ old} - (x_t - x_c)_2$$

$$\cdot \qquad \cdot \qquad \cdot$$

$$\cdot \qquad \cdot \qquad \cdot$$

$$x_{cn\ new} = x_{tn\ old} - (x_t - x_c)_n$$

When $(x_{it} - x_{ic})$, $i = 1, 2, \ldots n$, are sufficiently close to zero the procedure has converged and x_{1c}, x_{2c}, ..., and x_{nc} are the required solution to the nonlinear system.

Stoecker gives a seven-step procedure for implementing this technique:

1. Rewrite all the equations in the form R_i $(x_1, x_2, x_3, \ldots x_n) = 0$, $i = 1, 2, \ldots n$.
2. Assume values of x_{1t}, x_{2t}, ..., and x_{nt}.
3. Calculate R_i $(x_{1t}, x_{2t}, \ldots, x_{nt})$ for $i = 1, 2, \ldots, n$.
4. Compute DR_i/DX_j at $(x_{1t}, x_{2t}, \ldots, x_{nt})$ for $1 = 1, 2, \ldots, n$ and $j = 1, 2, \ldots, n$.
5. Solve the set of linear equations [Equation (3.35)].
6. Compute the new values from Equation (3.38).
7. Using the values from step 6 in step 2, test for convergence, and if not converged, repeat steps 2 to 7.

A computer program was developed based on the above algorithm as visualized in Figure 15 of the flow chart. This routine, which requires all the component equations for a solution set of the non-linear equations. Figure 15 shows the flow-diagram for a building loop analysis. The program attempts to obtain a converged set of the solutions for hydraulic equations satisfying the given thermal conditions.

This procedure was used to analyze any piping layout for this study. Based on this simulation technique, several results have been obtained with relatively faster run time than when using other routines.

3.4 System Model and Data Collection

To simulate a large piping system without simplification would require enormous time and computer capabilities due to the huge amount of unmanageable data. It, therefore, was concluded that some form of

* FLOW CHART OF COMPUTER PROGRAM *

BSIM.CMN
* Create common variables

DATA.FOR
* Store data needed

INIT.FOR
* Set initial values

A

RESD.FOR
* Calculate
Residual values

PDRV.FOR
* Evaluate
Partial derivatives

GAUSSY.FOR
* Estimate
Corrected values

CONVERGE ? Yes Go to B

LAST ITERATION ? Go to A
Yes
B

TEMP.FOR
* Calculate
Temperatures

TSET.FOR
* Compare
The Temperatures

CNTL.FOR
* Control
Valve positions

LAST COUNTER ? Yes Go to C

Go to A

C

RESULT.FOR
* Summarize the results

Figure 15. Iterative Method Logic Diagram

simplification was in order. The macro distribution model may be viewed as a two-input system for evaluating the pressure drops at each branch. One input came from the status of the central distribution pump operations. Flow rates at each branch are considered as the other input. The flow rates were obtained primarily from the overall building loads. One of the building loop simulation's capability is to estimate the needed primary flow rate to meet the given thermal loads. The central plant itself can be examined very carefully in a different analysis called chiller loop study.

Flow rates from the building loop study are coupled with the distribution model. The distribution pump status from the central chiller loop study is also connected to the distribution loop analysis. Thus, the hydraulic and thermal behavior of the central distribution line can be predicted. In the simulation utilizing the iterative method, a building loop coefficient was introduced to express pressure-flow characteristics of a building, using the building loop simulation program. Then, a distribution loop model predicts the behavior of the distribution loop.

To predict the behavior of the OSU CCW system, the layout and hardware of each building loop were first obtained from the shop drawings located in the Physical Plant blueprint files. The drawings indicated the location and type of cooling coils, the length and diameter of the distribution lines and the number of pipe fitting in the buildings. Records in the power plant showed the important data on the distribution pumps, as given in Appendix D. However, several of the drawings for the building loop were incomplete and outdated which resulted in major limitations on this study.

buildings. Records in the power plant showed the important data on the distribution pumps, as given in Appendix D. However, several of the drawings for the building loop were incomplete and outdated which resulted in major limitations on this study.

CHAPTER IV

SIMULATION OF CENTRAL CHILLED WATER SYSTEM

4.1 Simulation Approach

Computer simulation generally refers to the solutions of the equations of mathematical model representing behavior of a real physical system through the use of computer program. The major advantage of computer simulation is its flexibility and cost effectiveness to test and modify a physical system through parametric changes. Any large central chilled water system may be studied and modeled locally for building loops and central chiller loops. The CCW system also may be simulated in a macroscopic way for the large distribution system study. A focus of simulation is, therefore, determined depending upon the interest of the system analyst. Building or chiller loops can be examined if the major problems were detected in those loops.

The distribution loop simulation model will be used to understand the total system from a macroscopic view. A key factor to couple these models is the branch coefficient, defined as branch flow rate over square root of branch pressure drop, which is similar to the valve coefficient from an analytical stand-point. The simulation approach of the large system can be explained by the following five steps:

1. Draw distribution system lines with multiple branches based on the longest run.

2. Input the status of distribution pumps.

3. Input the required amount of branch flow rate

4. Estimate the pressure rise through the distribution pumps.

5. Calculate the pressure drops at the specified sections of interest.

A required amount of flow rate can be found for a building or branch under given thermal load conditions in terms of cooling 'ton' which is equivalent to 12,000 Btu/hr of thermal load. The total flow rate may be obtained by summing up all the branch flow rates. The equation for the distribution pump is selected based on the operational mode of the distribution pumps at a specific time. The pressure rise across the section of the main distribution pumps may be calculated based on the selected pump operation mode. Finally, the pressure across any branch may be obtained with the aid of Kirchhoff's law for the hydronic loops. The pressure-drops will be used to select an optimum building control valves for the secondary loops.

The distribution loop analysis requires the results or information from building loop studies. The building model also can test several other configurations. As stated in Chapter III, there were two approaches to model and simulate a hydronic loop depending on the availability of all the component equations, especially for the control valve. The model utilizing the direct method tried to find the pressure drop across the control valve to meet the other conditions. But, the model utilizing the iterative method may examine whether the main return temperature is constant or not in addition to the amount of the primary flow drawn from the main distribution lines. The results with the iterative approach are listed in Appendix A for different cooling load conditions. The results calculated from both approaches were compared and they are fairly close. The difference in the results with different approaches is primarily due to the error bands given to the controlled temperatures for the control valve with the iterative approach. A dynamic control characteristic cannot be predicted with these two modeling approaches because they are utilizing the concept of steady-state analysis. The dynamic response of system components can be analyzed by using the simulation package PARASOL as shown in Chapter II to select an optimum valve gain and other components. The flow rates were calculated based on both simulation approaches of the building loops. These flow rates were mainly used for the simulation of the large distribution loop.

From the central chiller loop study, the status of the distribution pumps can be obtained. The status (or mode) can be decided by a series of pump curves depending on the on-off status of each pump in the central plant. The distribution pump data given in Appendix D were obtained from the OSU Physical Plant and were used to draw a series of curves depending upon the status of the pumps, which are all connected in parallel. The status of the distribution pumps (turned-on or -off) determines the coefficients of the distribution pumps, which were obtained by the curve fitting technique as in Appendix C. The pump equation is, then, replaced at the driving section of the distribution model. Thus, the pressure rise for a total campus demand flow rate can be determined. The pressure drops at the branches may be calculated with the values of the pressure rise through the pump at the plant by using the hydronic loop law. This main distribution model, therefore, tested the following parametric effects:

1. distribution pump operating status
2. piping arrangement and modifications
3. branch (building) demand variations

Actually, the distribution pumps at the OSU central plant are controlled by a preset pressure drop at the farthest locations. The control scheme has not been working well because most of the present building loops are not well controlled systems. The purpose of the first parametric study with the pump status is to find the most effective way of pumping to meet the condition for a variable flow system. This can be accomplished by finding an optimum combination of the distribution pumps to meet the thermal loads at branches for the variable flow system.

Second, the effect of a line modification was examined because some of the sections are observed with high-head losses. The examination was focused on the possible reasons for the high head losses: smaller diameter or excessive flow demand at near branches. The real cause for this will be detected by the second simulation study.

Third, the distribution model tested the effect of the future addition to the present distribution line on the hydraulic performance of the distribution line. The test was for the addition of Gallagher Hall and Cordell Hall, which should be connected to the branch near the central plant. Another test was for a feasible addition of Vet Med Hospital building, which is connected to a branch far away from the plant.

4.2 Building Loop Simulation

The first example model represents the hydraulic loops inside a building, which are also called secondary loops, shown in Figure 16.

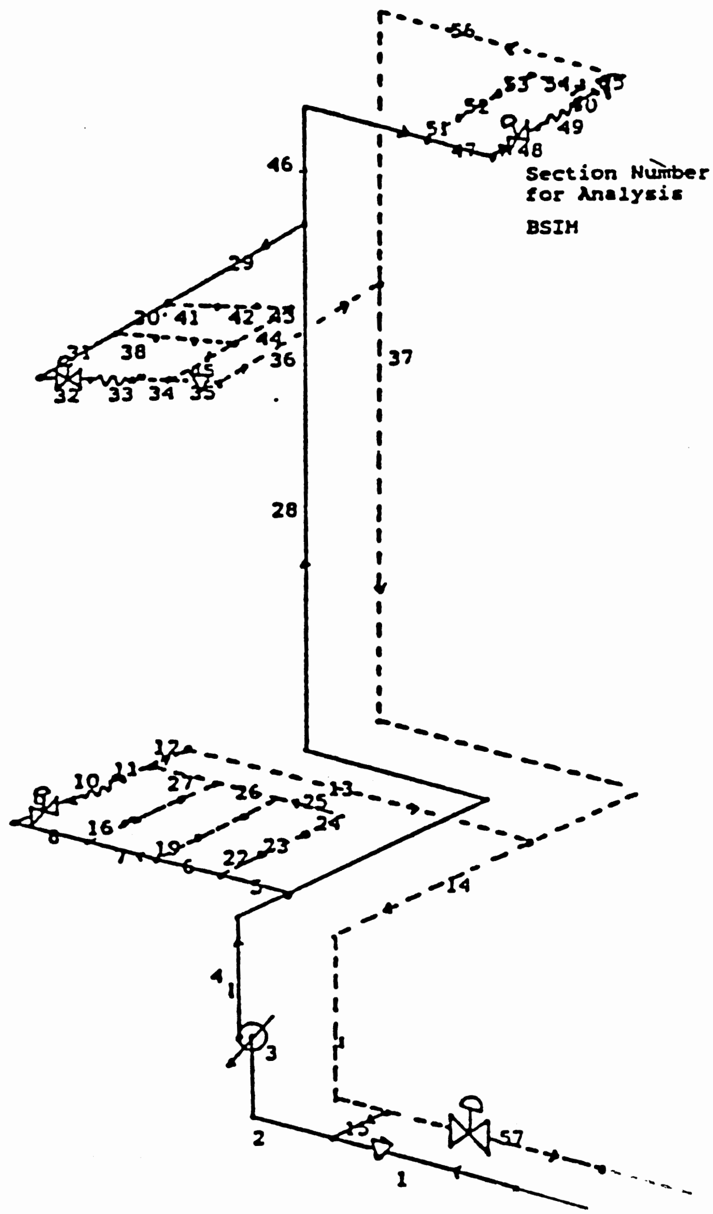


Figure 16. Engineering North Building Loop Model

The example represents the Engineering North building of Oklahoma State University. There are nine cooling coils and one secondary supply pump, which is a variable speed pump. The variable speed pump should be controlled by a set-pressure at the farthest point in the building loop just the same as the variable speed pumps at the central plant are controlled.

One objective is to maintain the main return temperature with proper selection of control valve and controller gains. Another goal with this simulation is to match the variable building demand with the variable building supply by making the cooling factor approach a value of 1.0. The variable building loads may be described with the variable flow rate, Wss, and the terminal temperature difference, DTs. The variable flow rate can be the sum of the flows needed for the coils. The coil flow rate depends on the load to the coil and the terminal temperature difference. In the same way, a variable building supply can be expressed with the variable flow rate, Wps, and the temperature difference, DTp, across the main distribution line.

Practically, it is difficult and requires a fine tuning of controller gains and an optimum type of control valve to maintain the temperatures of each stream. References (52) (53) show that a control method for an entire building loop must be proposed to make both the people inside a building and the people at the central plant happy simultaneously. Three types of such constant temperature systems were previewed with Figure 6. a, b, and c. of section 2.2 Those methods are: 1. return water control method, 2. supply water control method-I, 3. supply water control method-II. Method 2 differs from method 3 in terms of the coil outlet control method. The second method senses the coil outlet water temperature to control the valve on the coil to maintain a constant return temperature to the main distribution line. The third method uses the wet-bulb temperature method to directly estimate the needed water flow through a coil to produce a predetermined temperature difference, DTs. In both methods, it is extremely important to supply water to the coil at a preset temperature. This is the advantage of the second and third method over the first method, which is the return water temperature control method. The humidity control problem with the first method at a low cooling load can be overcome with the second or the third method because the supply temperature can be adjusted anytime.

This study focused on the second system because the third system requires extra wet-bulb temperature sensors and more complicated controllers than the second method. Therefore, the building loop simulations are based on the second control method, which is the proposed control method for the building loop. The piping information on equivalent length and diameter of the model is shown in Appendix E, which is based on a blue-print obtained from the OSU Physical Plant. The major task is to determine flows, pressures, and temperatures at specified locations, and the required amount of the primary flow drawn into a building to meet the changing thermal loads.

The inputs are the cooling loads (ton) to each coil. The pressure drop and the flow rate of each coil at design condition must also be specified as well as the characteristics of the pumps. The design temperature difference of the coils and the design temperature difference of the distribution lines must also be specified. The key parameters were 1. thermal load variation (partial thermal load effect), 2. coil valve type and characteristics, 3. piping arrangements, and 4. pump type and characteristics. The effects of these parameters on the hydraulic performance of the building loop were considered.

Specifically, the second parametric study on the effect of the valve type and characteristics on the performance can only be possible with the iterative method because the valve information must be specified in terms of a characteristic curve. More details are reviewed again later in this chapter dealing with program running procedure.

The general procedure for the analysis using the direct substitution method involves the following steps:

1. Flow rate through each coil for the given thermal load is determined by

$$\text{Flow} = 12000*\text{ton}/500/\text{DTs} \tag{4.1}$$

where

ton ... cooling load

DTs ... design terminal temperature difference, F

Flow ... desired coil flow rate, gpm

2. Pressure drop through each coil is determined based

$$\text{DPa} = \text{DPd}*(\text{FRa}/\text{FRd})^2 \tag{4.2}$$

where

DPa ... actual pressure drop across coil, psi

DPd ... design pressure drop across coil, psi

FRd ... design flow rate of coil, gpm

FRa ... actual flow rate of coil, gpm

3. The needed flow rate for a secondary loop is calculated by summing up all the coil flow rates

$$Ws = \sum_{i=1}^{n} (Flow)_i \tag{4.3}$$

where

Ws ... needed secondary flow, gpm

$(Flow)_i$ ... flow rate at coil i, gpm

n ... number of coil

4. Pressure rise through the secondary pump is calculated for a selected pump in terms of the secondary flow rate.

$$PR = P_0 + P_1 * Ws + P_2 * Ws^2 + P_3 * Ws^3 \tag{4.4}$$

where

PR ... pressure rise through the pump

P_j ... pump coefficients

Ws ... secondary flow rate

5. The pressure drop across the coil control valve (CCV) is, then, calculated according to the loop law

$$DPv = PR - \Sigma\, (DP)\ k \tag{4.5}$$

where

DPv ... pressure drop across the CCV, k

PR ... pressure rise through the secondary pump

$\Sigma\, (DP)\ k$ ... sum of pressure drops except the valve, k

For example, the pressure drops at nine coil control valves and their valve coefficients were obtained using this method. First, a very low, 5 ton cooling load to each coil was assumed. Figure 17 shows the

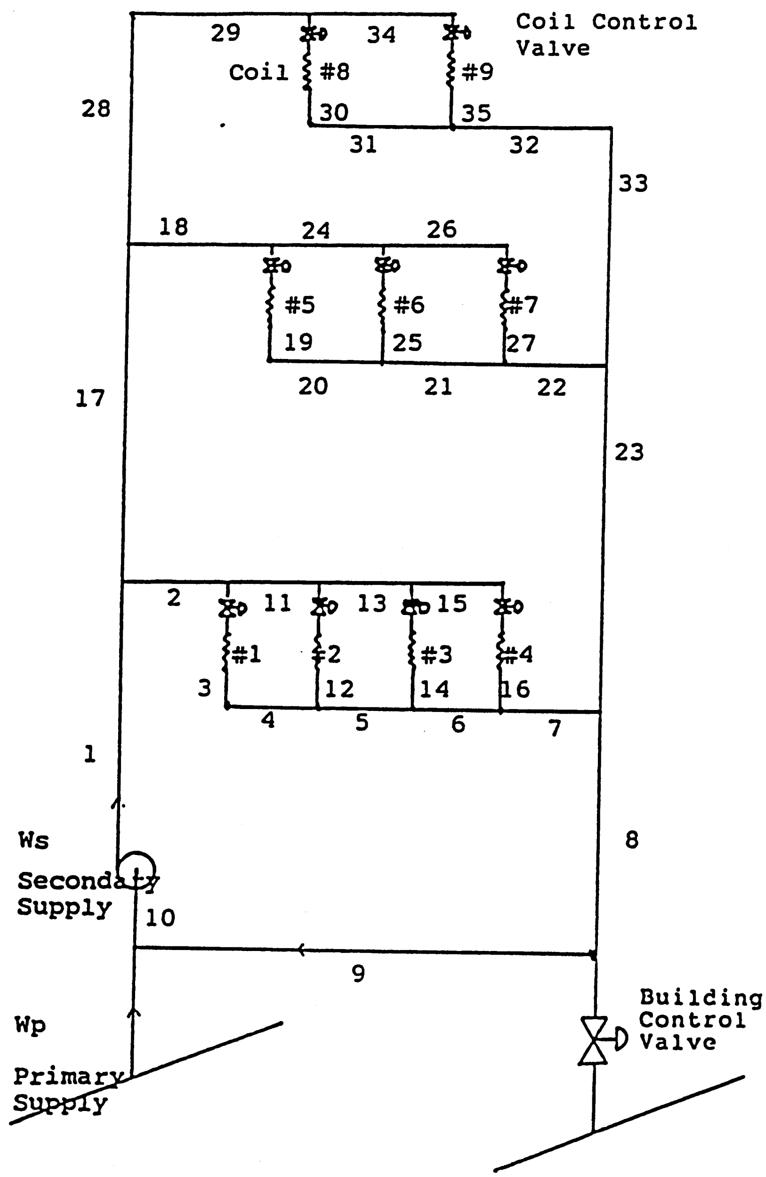


Figure 17. Simplified Engineering North Building Loop

simplified diagram of the Engineering North building loop. The pump was considerable overdesigned originally. The pump equation is:

$$PR = 27.68 - 5.405E\text{-}3*Ws + 2.002E\text{-}5*Ws^2 - 7.003E\text{-}8*Ws^3 \qquad (4.6)$$

The coefficients were obtained by the curve fitting technique as in Appendix C. The design conditions of each coil are listed in Table IV, which were obtained from the OSU Physical Plant. The design temperature difference of the distribution lines was 16 F, with the main supply 40 F and the return 56 F.

1. coil flow rate is

Flow = 12000*ton/500/DTs

= 12000*(5)/500/(12) = 10 (gpm) per each coil

2. actual pressure drop through coil #1 is

$$DPa = DPd*(FRa/FRd)^2$$

$$= (5.97)*(10/92)^2 = 0.07 \text{ (psi)}$$

3. secondary flow rate is

$$Ws = \sum_{i=1}^{n} (Flow)_i \; k = 9*(10) = 90 \text{ (gpm)}$$

4. pressure rise through the pump is

$$PR = 27.68-5.405E\text{-}3*(Ws) + 2.202E\text{-}5*(Ws^2)-7.003E\text{-}8*(Ws^3)$$

$$= 27.68-5.405E\text{-}3*(90) + 2.202E\text{-}5*(90^2)-7.003E\text{-}8*(90^3)$$

$$= 27.32 \text{ (psi)}$$

5. The pressure drop across the valve is, then, calculated since the coil and the other section head losses were calculated as in Chapter III.

$$DPv1 = PR - \Sigma \text{ (DP) } k$$

TABLE IV

COOLING COIL DATA OF THE ENGINEERING NORTH

Coil I.D.	Design Flow gpm	Design Head psi
1	92	5.97
2	108	3.46
3	55	2.38
4	59	2.64
5	56	4.03
6	102	2.38
7	105	4.98
8	61	4.98
9	81	5.93

= 27.32 - 0.47 - 26.85 (psi)

This high value of pressure drop is due to the oversized pump and the low thermal load. Also, this high pressure for the building pump could lead to unstable operation of the control valve. Therefore, a smaller pump with a flat pressure-flow characteristic curve is recommended with a variable speed motor. More runs were executed to visualize the effect of thermal loads and the type of pump, which will be shown in the following section. The effect of these changes on the pressure drops of the coil control valves will be examined.

4.3 Building Loop Simulation Results

Several runs were completed to determine the effects of different cooling loads on pressure drops across the control valves. Also, the simulations allow for the estimate of the range of the pressure drops and the flow rates for given cooling loads so that the control valve can be sized according to the results. The primary flow can also be estimated with the demand flow and the two design temperature differences, DTs and DTp, across the building (secondary) and distribution (primary) lines. Figure 18 shows one of the building simulation results at low, medium, and high cooling loads to each coil with the oversized building pump. The blue-print showed that the building was designed with 299 cooling tons, based on water-side.

Engineering North building, with the configuration of Figure 6.b, was simulated to see the effect of the load variations on pressure drops for the coil control valve with other parameters fixed as follows:

1. Secondary supply pump
2. Secondary design temperature difference, 12 F (44-56 F)

*** BUILDING SIMULATION RESULTS ***

- COIL THERMAL LOADS -

COIL I.D.	COOLING LOAD, TON
1	5.000000
2	5.000000
3	5.000000
4	5.000000
5	5.000000
6	5.000000
7	5.000000
8	5.000000
9	5.000000

- FLOW INFORMATION -

PRIMARY,	BY-PASS,	SECONDARY FLOW,GPM
67.50000	22.50000	90.00000

- VALVE INFORMATION -

VALVE I.D.	FLOW,GPM	HEAD,PSI	Cv
1	10.00000	26.84878	1.929913
2	10.00000	26.76955	1.932774
3	10.00000	26.66963	1.936384
4	10.00000	26.47131	1.943624
5	10.00000	26.54176	1.941043
6	10.00000	26.66699	1.936480
7	10.00000	26.66498	1.936553
8	10.00000	26.48735	1.943035
9	10.00000	26.53080	1.941444

*** BUILDING SIMULATION RESULTS ***

- COIL THERMAL LOADS -

COIL I.D.	COOLING LOAD, TON
1	15.00000
2	15.00000
3	15.00000
4	15.00000
5	15.00000
6	15.00000
7	15.00000
8	15.00000
9	15.00000

- FLOW INFORMATION -

PRIMARY,	BY-PASS,	SECONDARY FLOW,GPM
202.5000	67.50000	270.0000

- VALVE INFORMATION -

VALVE I.D.	FLOW,GPM	HEAD,PSI	Cv
1	30.00000	22.19874	6.367325
2	30.00000	21.48394	6.472384
3	30.00000	20.58644	6.611966
4	30.00000	18.80155	6.918699
5	30.00000	19.43556	6.804915
6	30.00000	21.14897	6.523440
7	30.00000	21.61973	6.452025
8	30.00000	18.94391	6.892290
9	30.00000	19.33498	6.822238

*** BUILDING SIMULATION RESULTS ***

- COIL THERMAL LOADS -

COIL I.D.	COOLING LOAD, TON
1	25.00000
2	25.00000
3	25.00000
4	25.00000
5	25.00000
6	25.00000
7	25.00000
8	25.00000
9	25.00000

- FLOW INFORMATION -

PRIMARY,	BY-PASS,	SECONDARY FLOW,GPM
337.5000	112.5000	450.0000

- VALVE INFORMATION -

VALVE I.D.	FLOW,GPM	HEAD,PSI	Cv
1	50.00000	11.52320	14.72933
2	50.00000	9.537627	16.19011
3	50.00000	7.044509	18.83832
4	50.00000	2.086342	34.61437
5	50.00000	3.847702	25.48997
6	50.00000	8.607134	17.04277
7	50.00000	9.914833	15.87915
8	50.00000	2.487331	31.70181
9	50.00000	3.573440	26.44831

Figure 18. Simulation Results for Different Cooling Loads

3. Primary design temperature difference, 16 F(40-56 F)

A simple energy balance principle was applied to the loops to verify the results and to justify the tendencies of the variable flow system. Engineering North building was originally designed with a 299 ton cooling load with 10 F difference. The design secondary supply flow is:

$$Wss = ton * 12{,}000 / 500 / DTs$$

$$= (299)*(12{,}000)/500/10 = 718 \text{ gpm with 10 F difference}$$

and

$$Wss = (299)*(12{,}000)/500/12 = 598 \text{ gpm with 12 F difference}$$

Thus, the design primary flow should be

$$Wps = Wss * (DPs/DTp)$$

$$= (598)*(12/16) = 448.5 \text{ gpm}$$

Three data points were obtained from the building loop simulation to correlate the cooling used and supplied with the DTp kept constant. The relationship appears as linear as expected with the results shown in Figure 18.

Figure 18 shows also the pressure drop variations due to the load changes. The control valve had to furnish high pressure drops at low loads. In this case, the type of control valves must be selected based on the procedure in Chapter II. An equal percent type of control valve seems best because of the lower gain at the lower load. To improve the reliability of the valve the high pressure should be avoided by reducing the size of the secondary pump or by introducing a small bypass across the valve. The effect of the secondary pump will be introduced later in this section. The effect of the bypass diameters on hydraulic performance was illustrated in reference (29). The range of coil control valve, Cv, may be estimated with a set of cooling loads and the simulation program.

The simulation results were obtained assuming an oversized secondary pump (26 psi at the low load). A simulation at design loads to all coils indicated 43 psi would be enough to meet the load requirement. The optimum size of the building pump can be decided assuming a maximum load, and a suitable safety factor. Several simulations were conducted to provide information on the feasible size of the building pump.

As one of the simulations, the Engineering Building loop was tested with a smaller pump having the characteristics:

$$PR = 16.43 + 2.472E\text{-}3*(Ws) - 22.29E\text{-}5*(Ws^2) + 1.603E\text{-}8*(Ws^3) \qquad (4.7)$$

As shown in Figure 19, the pressure drop across the coil control valve was drastically reduced, which leads to better controllability. The third one in Figure 19 showed the results with the design conditions for each cooling coil. The design Cv should have been based on these results if the load estimation was accurate. But, the Cv's should be reduced if the load is overestimated.

Engineering North building data monitored at the OSU Physical Plant included the main return temperature, the building return temperature, and the controlled building supply temperature. The results will be mentioned in the following section.

4.4 Experimental Verification

The results of measurements on the modified system operation of Engineering North building are shown in Figure 20. Shown are the main

*** BUILDING SIMULATION RESULTS ***

- COIL THERMAL LOADS -

COIL I.D.	COOLING LOAD, TON
1	5.000000
2	5.000000
3	5.000000
4	5.000000
5	5.000000
6	5.000000
7	5.000000
8	5.000000
9	5.000000

- FLOW INFORMATION -

PRIMARY,	BY-PASS,	SECONDARY FLOW,GPM
67.50000	22.50000	90.00000

- VALVE INFORMATION -

VALVE I.D.	FLOW,GPM	HEAD,PSI	Cv
1	10.00000	16.01228	2.499042
2	10.00000	15.93265	2.505263
3	10.00000	15.83313	2.513139
4	10.00000	15.63481	2.529028
5	10.00000	15.70526	2.523350
6	10.00000	15.83049	2.513349
7	10.00000	15.82848	2.513508
8	10.00000	15.65085	2.527732
9	10.00000	15.69430	2.524230

*** BUILDING SIMULATION RESULTS ***

- COIL THERMAL LOADS -

COIL I.D.	COOLING LOAD, TON
1	15.00000
2	15.00000
3	15.00000
4	15.00000
5	15.00000
6	15.00000
7	15.00000
8	15.00000
9	15.00000

- FLOW INFORMATION -

PRIMARY,	BY-PASS,	SECONDARY FLOW,GPM
202.5000	67.50000	270.0000

- VALVE INFORMATION -

VALVE I.D.	FLOW,GPM	HEAD,PSI	Cv
1	30.00000	11.35662	8.902186
2	30.00000	10.64182	9.196305
3	30.00000	9.744321	9.610489
4	30.00000	7.859425	10.63360
5	30.00000	8.593441	10.23382
6	30.00000	9.720567	9.622224
7	30.00000	9.702487	9.631185
8	30.00000	8.103787	10.53846
9	30.00000	8.494859	10.29303

*** BUILDING SIMULATION RESULTS ***

- COIL THERMAL LOADS -

COIL I.D.	COOLING LOAD, TON
1	38.00000
2	45.00000
3	23.00000
4	24.00000
5	23.00000
6	43.00000
7	44.00000
8	25.00000
9	34.00000

- FLOW INFORMATION -

PRIMARY,	BY-PASS,	SECONDARY FLOW,GPM
448.5000	149.5000	598.0000

- VALVE INFORMATION -

VALVE I.D.	FLOW,GPM	HEAD,PSI	Cv
1	76.00000	18.07599	17.87568
2	90.00000	15.72057	22.69909
3	46.00000	16.31279	11.38921
4	48.00000	12.86124	13.38443
5	46.00000	9.821222	14.67828
6	86.00000	6.709618	33.20088
7	88.00000	3.851303	44.84137
8	50.00000	9.770243	15.89622
9	68.00000	7.178901	25.37755

Figure 19. Simulation Results with a Smaller Pump

supply temperature variations and the main return temperature variations over a specified time period. It verifies the expected status of return water temperature to the main distribution line and the building supply temperature. The measured data show some variations of the return water temperature due to the lack of perfect coil valve control.

Figure 20.a shows the temperature variations of primary return, secondary supply, and primary supply during 30 hours starting at 7:53 a.m. on February 6, 1987. Although the temperature of the primary supply varied the return water temperature remained fairly constant at 56 F with 1 F variation. The primary supply temperature was fluctuating due to the free cooling at the central plant on that day. The control point at the secondary supply was preset at 50 F to provide a desirable return water temperature. But, it was reported that the wrong sensing locations were read. On March 24, 1987, another measurement was made, and the results are shown in Figure 20.b. The building supply water temperature was very unstable during the time period and it was difficult to maintain the set temperature of 44 F. A swing of the return temperature was detected, probably because four of the nine cooling coils in the Engineering North building were not properly regulated. A reverse flow (short circuiting) without passing through the coils was observed because the main return temperature was less than the secondary return temperature. In Figure 20.c, on the next day, a P-I controller was introduced and reduced the instability problem, but the building return temperature still varied as the coil load varied due to the uncontrolled coils.

The proposed system of Figure 6.b would need another temperature sensor at the location of the primary supply if the primary supply

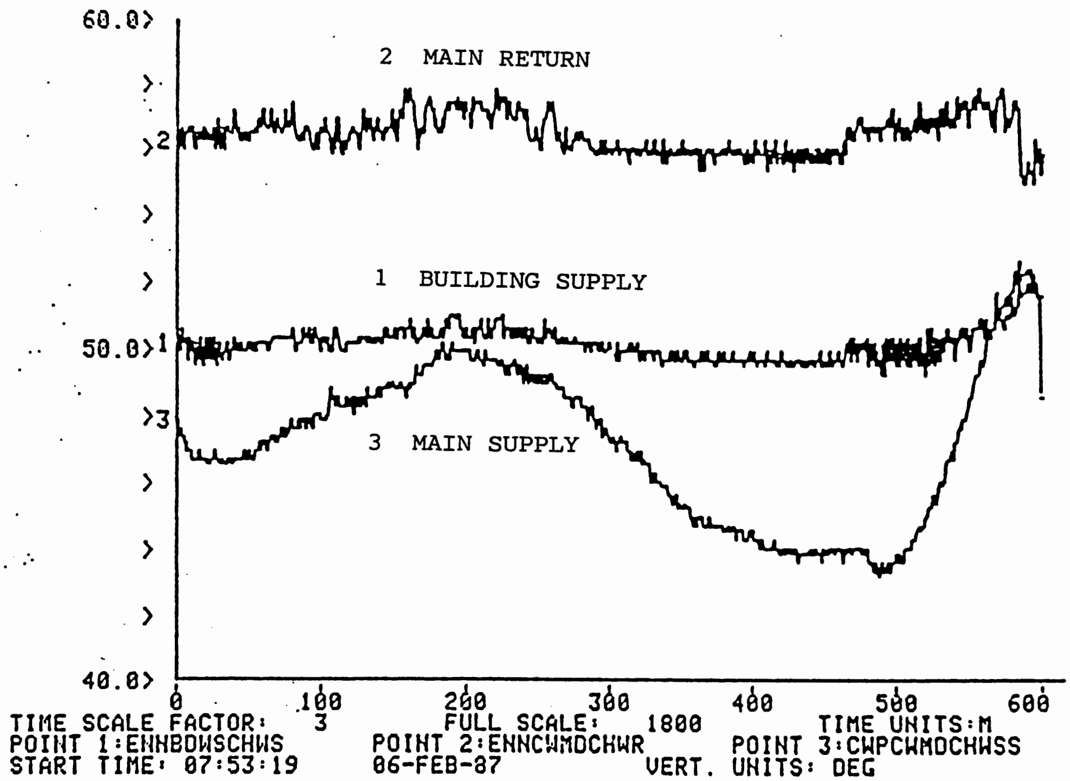


Figure 20.a Experimental Data of a Building Loop (06-Feb-87)

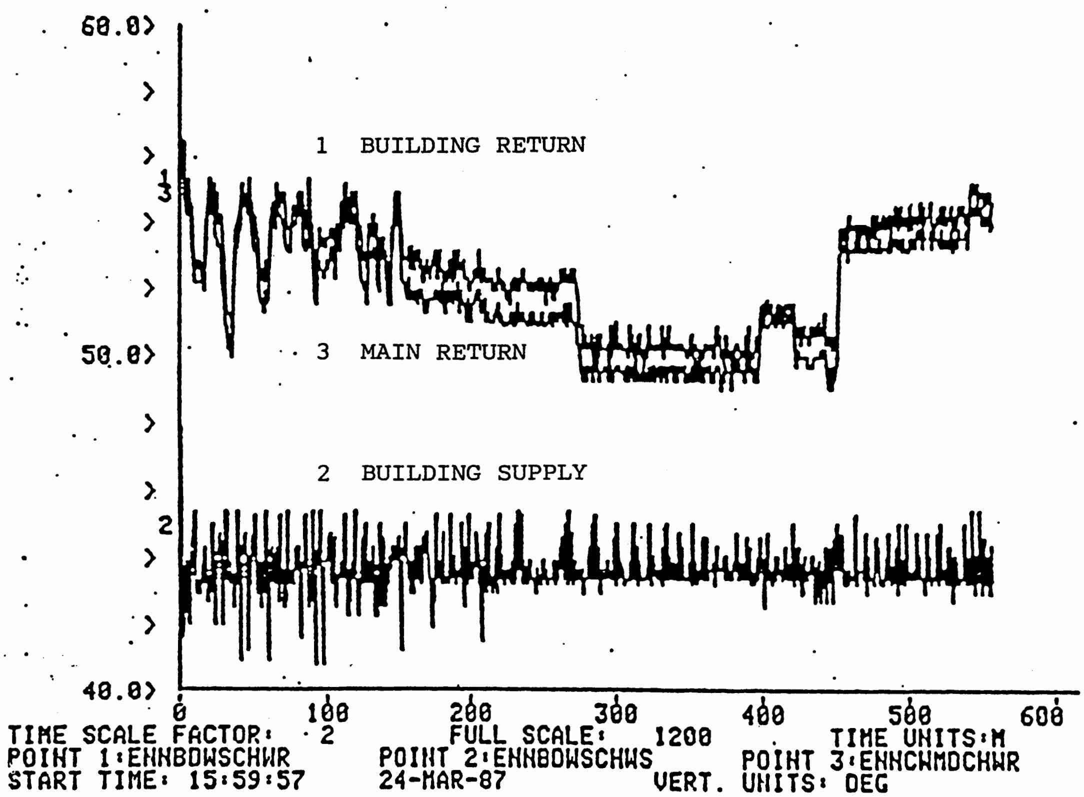


Figure 20.b Experimental Data of a Building Loop
(24-Mar-87)

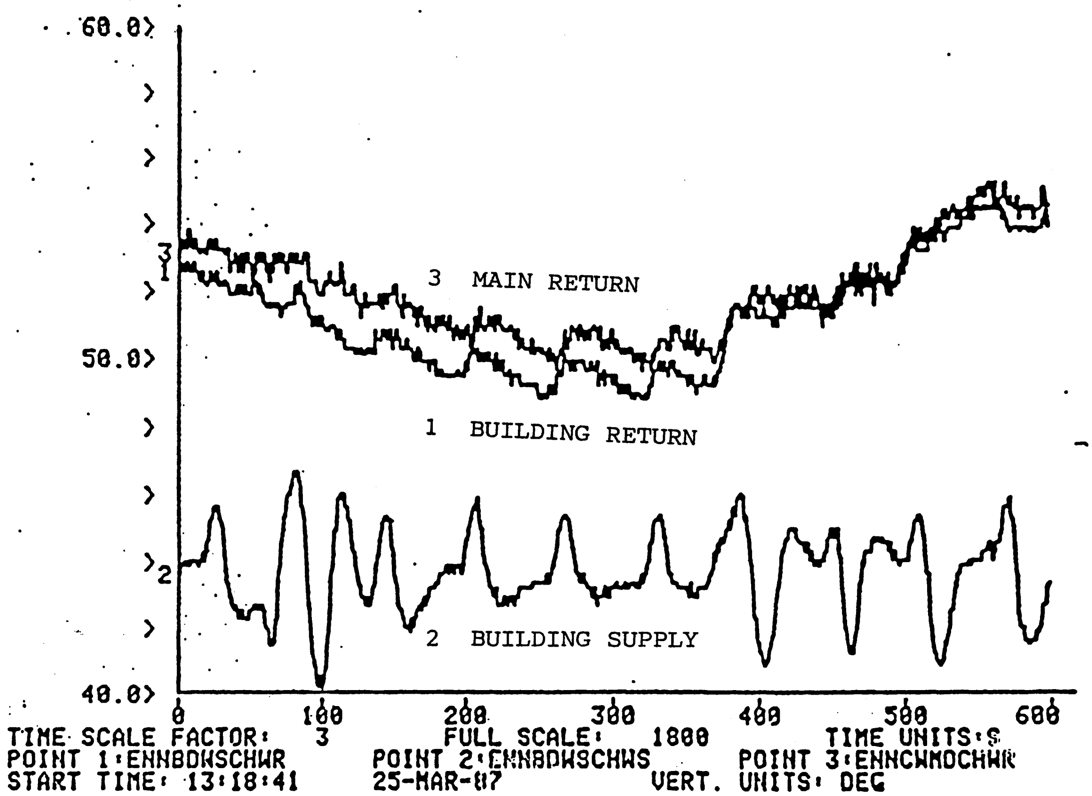


Figure 20.c Experimental Data of a Building Loop (25-Mar-87)

temperature were higher than the secondary supply. In this situation, the temperature difference between the primary supply and the secondary supply could be used to control the automatic building control valve.

However, instability of the control point (supply temperature) was observed at low load, probably due to the proportional gain, and the system responded too quickly. Problems in valve dynamics should be solved to make sure that the constant temperatures could be maintained. To make the proposed control method work, all the coils in a building should have the right valve and the controller to maintain the desired outlet water temperature. To overcome bypassing in the common pipe, a check valve (57) may be required to allow one directional flow in the common pipe. Another but more expensive approach is to use a water-to-water heat exchanger, as in the Student Union, to separate the building loop from the main line.

Distribution loop simulation results are based on the results of the studies on the building loops and chiller loops.

4.5 Theoretical Verification

From the results of the building loop investigations, a control method for the secondary building loop was proposed (Figure 21). This variable flow building system consists of a coil control valve, building control valve, and a variable speed pump. There are four strategic locations in this system called the primary supply, the secondary supply, the secondary return and the primary return. The best system demands the least amount of primary water and maintains the primary return at the desired temperature.

The load removed from the secondary loop by primary flow must be

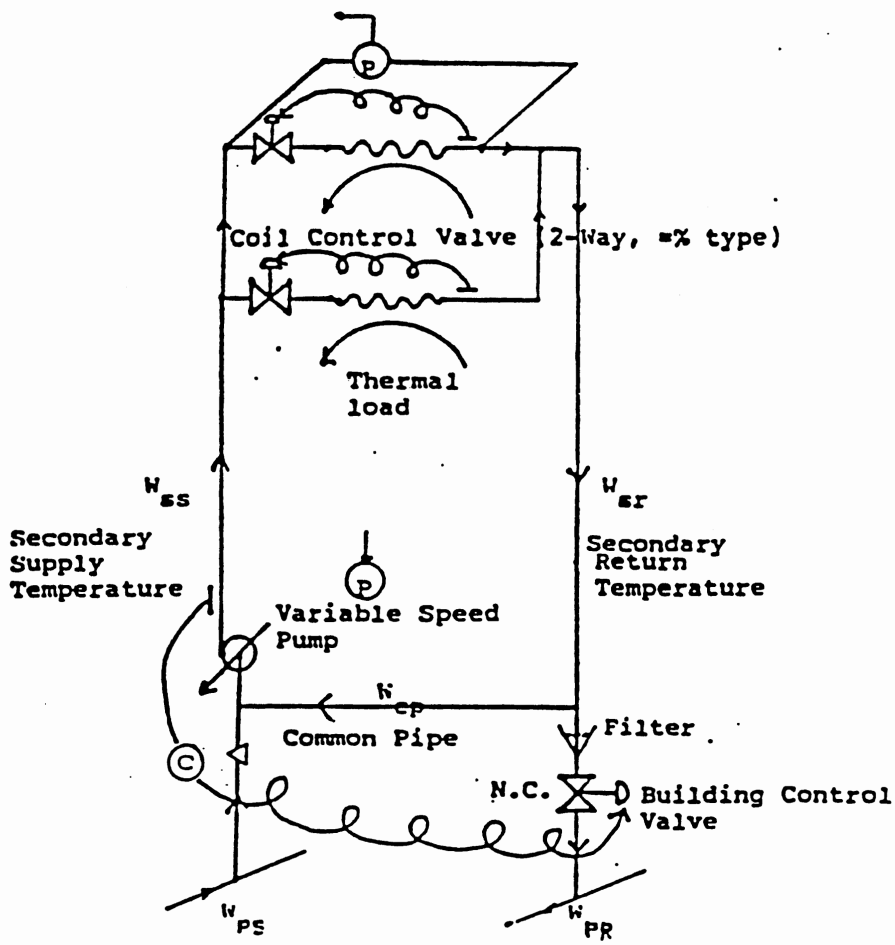


Figure 21. A Proposed Control Method for Buildings

equal to load removed from the secondary loop terminals by the secondary flow. Referring to Figure 21, the load, flow, and temperature difference relationships should be written as:

$$500 * Wp * DTp = \text{primary loop load} \quad (4.8)$$

$$500 * Ws * DTs = \text{secondary loop load} \quad (4.9)$$

Because the primary load is equal to the secondary load,

$$500 * Wp * DTp = 500 * Ws * DTs \quad (4.10)$$

where

DTp primary temperature difference, Tpr-Tps

DTs secondary temperature difference, Tsr-Tss

Canceling the terms leads to the following equation:

$$Wp = Ws * DTs / DTp \quad (4.11)$$

Thus, the flow rate of a building and the flow rate into the building should be varied properly to keep the temperatures constant. For the first example run of Engineering North in Figure 16, the results were obtained as:

Ws = 90.0 gpm

DTs = Tsr-Tss = 56-44 = 12 F

DTp = Tpr-Tps = 56-40 = 16 F

Thus

Wp = (90)*(12/16) = 67.5 gpm

And, the simulated result was 67.5 gpm as shown in Figure 18. The percent primary flow is

% primary flow = Wpa/Wpd

$$= (67.5)/(448.5) = 15.05\%$$

This indicates a linear relationship between the cooling used with Wss and supplied with Wsp.

Figure 22 compares the controlled secondary loop and an uncontrolled system to visualize the reduction in primary flow draw. The secondary loop draws too much primary flow if the flow is not controlled properly with a coil control valve and building control valve. The degree of unbalance can be explained with the cooling factor in the diagram.

The uncontrolled building loop draws twice as much primary flow as the controlled building loop over the range of thermal loads in a building loop with a 2.0 cooling factor. This trend was verified with the Engineering North building simulation program results and measurements. According to the measurement data on March 25, 1987, the flow rate of the building supply was about 150 gpm. Two possible cases can be presumed depending upon the direction and the flow rate (50 gpm assumed) through the building common pipe. For the desirable case the cooling factor and percent primary flow can be calculated as follows:

$$CF = \frac{Wps * DTp}{Wss * DTs} = \frac{(100)(51-40)}{(150)(51-44)} = 1.0$$

$$\%\ \text{primary draw} = \frac{Wps_a}{Wps_d} = \frac{100}{(0.2)(450)} = \frac{100}{90} = 1.1$$

For the case of a short circuiting, the CF and the percent draw can be calculated as:

$$CF = \frac{(200)(51-40)}{(150)(52-40)} = 1.2$$

$$\text{COOLING FACTOR (CF)} = \frac{\text{COOLING SUPPLIED}}{\text{COOLING USED}} = \frac{Wps*DTp}{Wss*DTs},$$

where

Wss	secondary supply flow rate, gpm
Wps	primary supply flow rate, gpm
DTs	terminal temperature difference, F
DTp	primary temperature difference, F
CF	degree of over-supply to a building

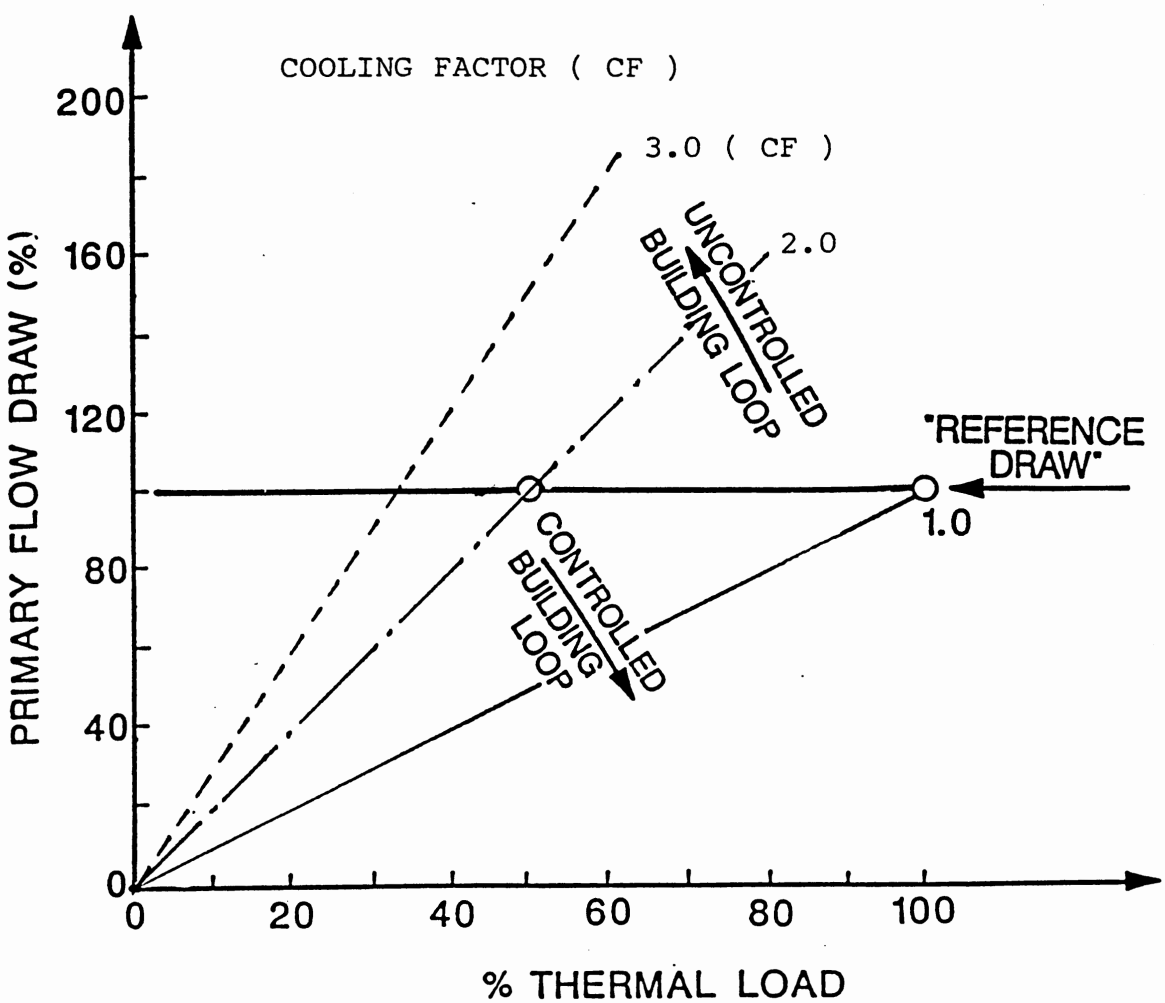


Figure 22. Simulation Results and Model Verifications

$$\% \text{ primary draw} = \frac{200}{90} = 2.2$$

This means twice as much water was supplied to the building loop as was needed because of the short circuiting.

A thirty percent flow reduction was assumed to calculate the annual savings from the modification of the secondary building loop to a constant temperature system. In addition to the savings at the chillers, a predicted pumping cost savings from this modification for 53 buildings under the following assumptions was calculated:

Assumptions: Electric rate, 0.04 $/kwh (Rate)

Flow reduction ratio, 0.3 (Ratio)

A conversion factor, 0.746 kw/hp

Variable speed pump operation

The equations used are:

Present Annual Operating Cost, HP1

HP1 = Rate * BHP(k) * 0.746 * OHRS(k) (4.12)

Future Annual Operating Cost, HP2

HP2 = Rate * (Ratio**3)*BHP(k) * 0.746 * OHRS(k) (4.13)

where

BHP(k) design horse power for each pump, hp

OHRS(k) annual operational hours, hr/yr

Annual savings from the modification was calculated next by summing up the savings in the buildings and the plant. The possible saving estimated was about

112,000 dollars per year

Within 5 years, the investment ($10,000 per building assumed) for the modification into the constant temperature system would be paid off.

Some buildings which are low-rise and near the central plant may not need the building pumps because they have high enough pressure differential to force the water inside the pipings of such buildings.

4.6 Chiller Loop Study

The second model represents the central chilled water plant of OSU as shown in Figure 23. The model has three chiller loops called Phase 1 (2,400 ton and 1,200 ton chillers in parallel), Phase 2 (4,200 tons), and Phase 3 (4,200 tons) as of 1987. Each chiller has its own chiller pump(s) of constant flow rate and the sizes are shown in Appendix D. Appendix E summarizes the equivalent lengths and diameters of the pipings in the central plant. The design flow rate and pressure drop of the chillers are shown in Appendix D. The detailed piping and location of the chillers and pumps were obtained from the drawings in the OSU Physical Plant.

As shown in Figure 23, an idealized distribution loop in this model is connected to the chiller loops, and carries the variable campus load. A common pipe is required to isolate the behavior of these constant flow chiller loops from the variable distribution demand flows. This unmatching of supply and demand at the central plant has led to blending due to the excess or deficit flow in the common pipe. The configuration of the present chiller loops is functionally the same as the pre-common pipe system shown in Figure 4.b, which brings cool mixed inlet water temperature to the chillers when the chillers overrun as the load decreases.

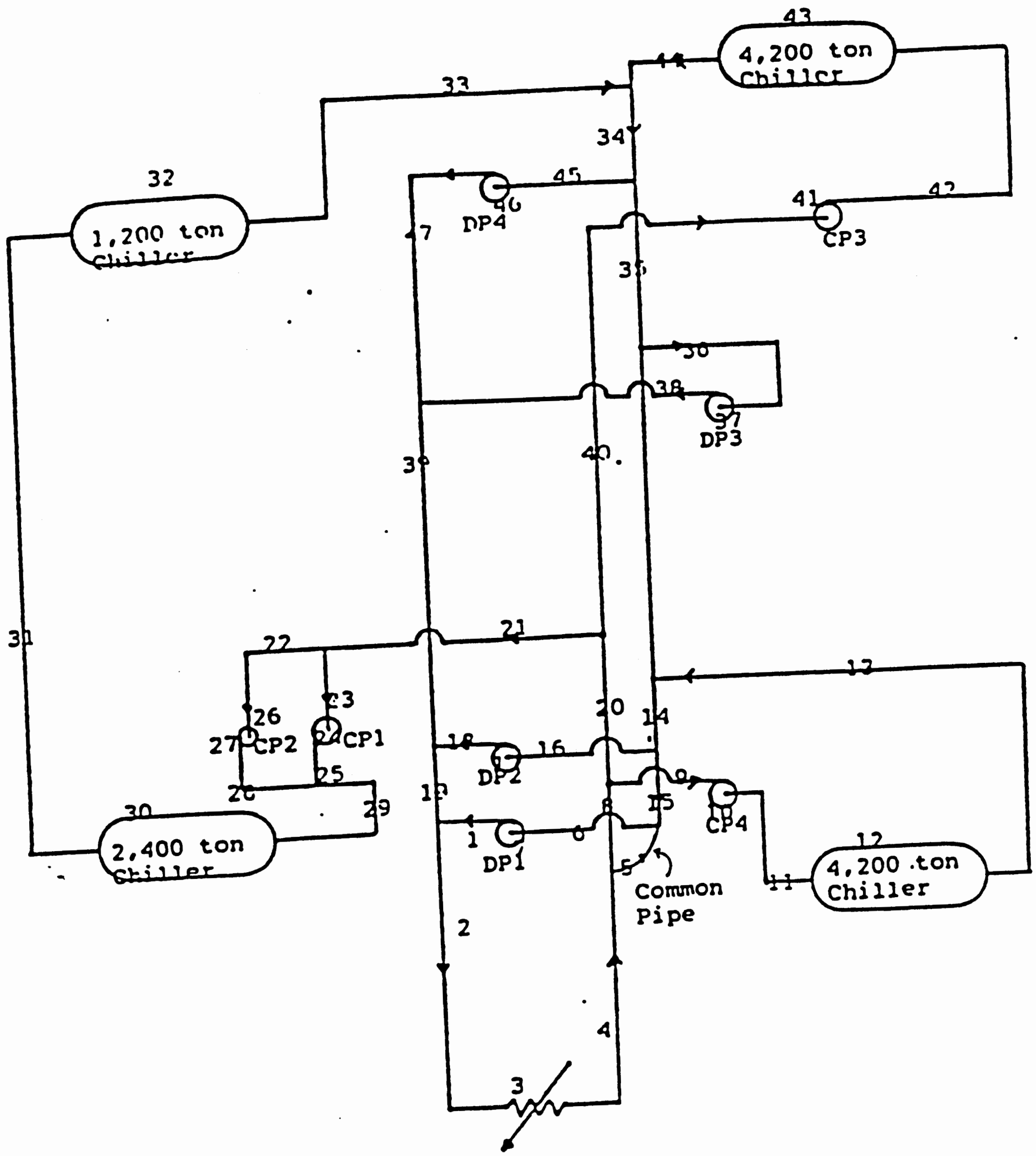


Figure 23. A Central Chiller Loop Model

The proposed system as shown in Figure 4.a (post-common pipe system) can unload the chiller sequentially starting from the unit 1, as the total campus load decreases.

The chiller loop model has 47 sections plus the common pipe. Thus, the 48 variables are enough to describe the behavior because the other flow rates are known with the chiller and pump operation plan. To study the chiller loop, the pressure drop and demand flow of the campus must be known or estimated and input properly. This information can be obtained from the distribution loop model, described in the next section.

Depending upon campus load, the distribution model gives the pressure drop and flow rate fixing the distribution pump operating status. The flow direction and the flow rate through the common pipe can be estimated for a given operating conditions of the multiple chillers. An optimum operation plan of multiple dissimilar chillers was introduced in reference (36).

The modification of the building loops into constant temperature systems is, therefore, very important because this brings a constant temperature to the chillers. A relocation of the common pipe (section 5 in Figure 23) has been proposed and studied. In central plant study, the present and the modified system have been examined with two different systems to prevent the blendings leading to inefficient chiller operations. The study compared basically two systems of Figure 4.a (post-common pipe system) and figure 4.b (pre-common pipe system) which are analogous to the future modified OSU central chilled water plant of Figure 24 and the present OSU central system of Figure 23. The system should be modified into one similar to Figure 24 because it has

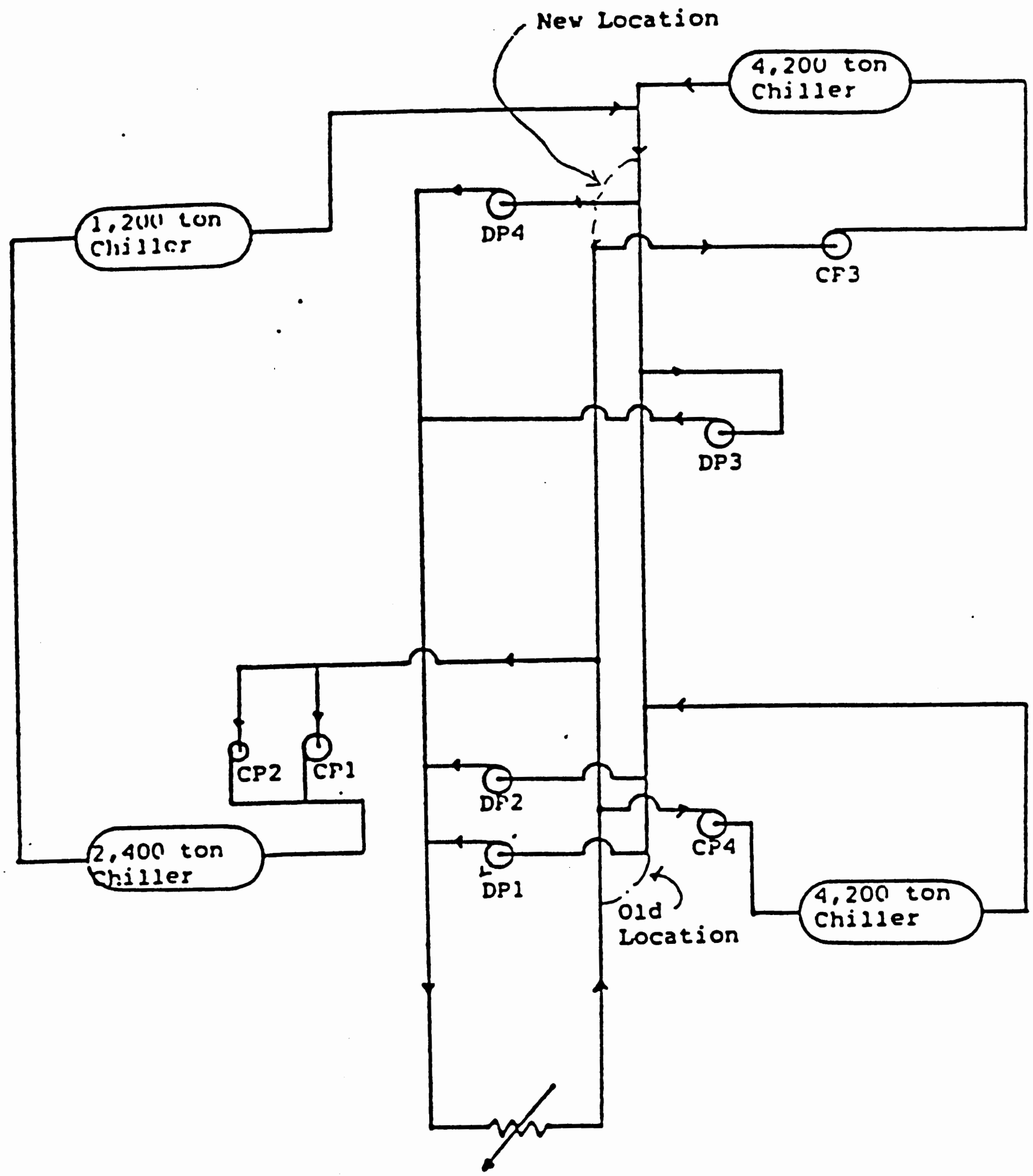


Figure 24. A Recommended Chiller Loop

two major advantages as Coad (13) explained:

1. to prevent warm blending when the demand is greater than the supply under design load condition
2. to prevent cool blending when the demand is less than the supply

The distribution pumps were identified with numbers for the different size of the pumps. For instance, the smaller two pumps (3,000 gpm) were numbered as #1 and #2 and the larger two pumps (6,300 gpm) were numbered as #3 and #4 respectively. Several options were derived to meet the variable flow demand from the campus. The combinational mode of the multiple pumps were classified with the options as follows (See Figure 26):

Options	Pump-on
I	1,2,3,4
II	3,4
III	1,2,3
IV	1,3
V	3 (or 4)
VI	1,2
VII	1 (or 2)

Based on the above analysis, the option I means that all the pumps are turned on. The option IV implies one smaller pump and one larger pump operating and the option VI stands for one of the two small pumps turned on.

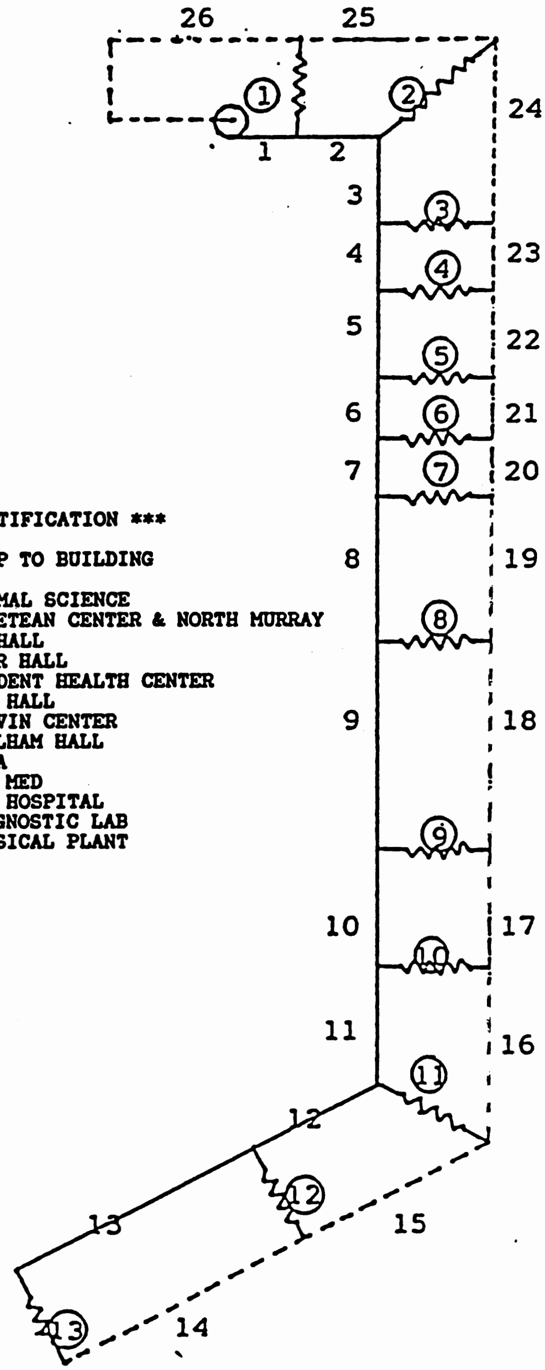
4.7 A Central Chilled Water System Simulation

The third model describes the distribution lines of the OSU central chilled water system in addition to the secondary branches. Figure 25 shows the OSU distribution loop model which was simulated. The objective of this model is to estimate the pressure and flow rates at various locations around the campus for a variety of situations so that a more effective distribution pumping method can be found to improve the overall cooling system efficiency.

Shown in Figure 25 is the schematic model of the primary distribution loop with the "longest" run to the Physical Plant building. The main distribution model has 13 sections of supply line and 13 sections of return, for a total of 26 sections in distribution lines. The flow rates to each branch should be known by the previous building simulation program, BSIM or load calculation before using this DSIM for distribution loop analysis.

In this model, Branch #1 describes the loops to the Animal Science building, branch #2 indicates an equivalent section which includes the loop to the Seretean Center and the loop to the North Murray building. Branch #3 is to the AG Hall building, Branch #4 for Kerr-Drummond Hall, Branch #5 for Student Health Center, Branch #6 for IBA Hall, Branch #7 for the loops to the Colvin Center, Branch #8 for Willham Hall, Branch #9 for the USDA Building, Branch #10 for the Vet Med building, Branch #11 for the Vet Med Hospital buildings, Branch #12 for the Diagnostic Lab building, and Branch #13 for the OSU Physical Plant, which is the farthest point from the central plant.

This program requires the control modes (options) of the distribution pumps because OSU CCW system is a variable flow system. Figure 26 shows the curves of the multiple distribution pumps which can produce a series of pump coefficients corresponding to a specific



*** BRANCH IDENTIFICATION ***

I.D.	LOOP TO BUILDING
1	ANIMAL SCIENCE
2	SERETEAN CENTER & NORTH MURRAY
3	AG HALL
4	KERR HALL
5	STUDENT HEALTH CENTER
6	IBA HALL
7	COLVIN CENTER
8	WILLHAM HALL
9	USDA
10	VET MED
11	VET HOSPITAL
12	DIAGNOSTIC LAB
13	PHYSICAL PLANT

Figure 25. A Combined Distribution Loop Model

operation mode. The flow rates of the secondary branches must be input when the branch includes more than one building. The status of the distribution pumps should be input to run the program. The lengths and the diameters of the sections between the secondary branches were obtained from the blue-prints in the OSU Physical Plant and shown in Appendix E.

The procedure to find the pressure drops across the branches is basically the same as the one for the building loop.

1. flow rate through each branch is estimated
2. total campus demand is calculated
3. pressure drop at the central plant is determined depending on the status of the pumps as in Figure 26
4. section pressure drops are calculated
5. if pressure drop at control point is not satisfactory a new pump control mode option is selected. The above steps are repeated until a satisfactory pressure at the selected point(s) is obtained
6. pressure drop for the building control valve is evaluated

Some of the distribution loop simulations are presented. The first run was to see the effect of the distribution pump operation. A 50% thermal load was assumed to each branch. The total simulated campus demand was 11,700 gpm. The mode of the distribution pumps was 4, which means pump #1 and #3 are turned-on (See Figure 26). The estimated pressure drop at the farthest point (the physical plant) for this condition was 10.8 psi which is over-pressurized since the set value is about 5 psi. Therefore, the option was changed to 5 from 4, which means one of the larger capacity pumps (6,300 gpm) was turned on. The

Distribution Pump Options

Options	Pump-on*
I	1,2,3,4
II	3,4
III	1,2,3
IV	1,3
V	3(4)
VI	1,2
VII	1(2)

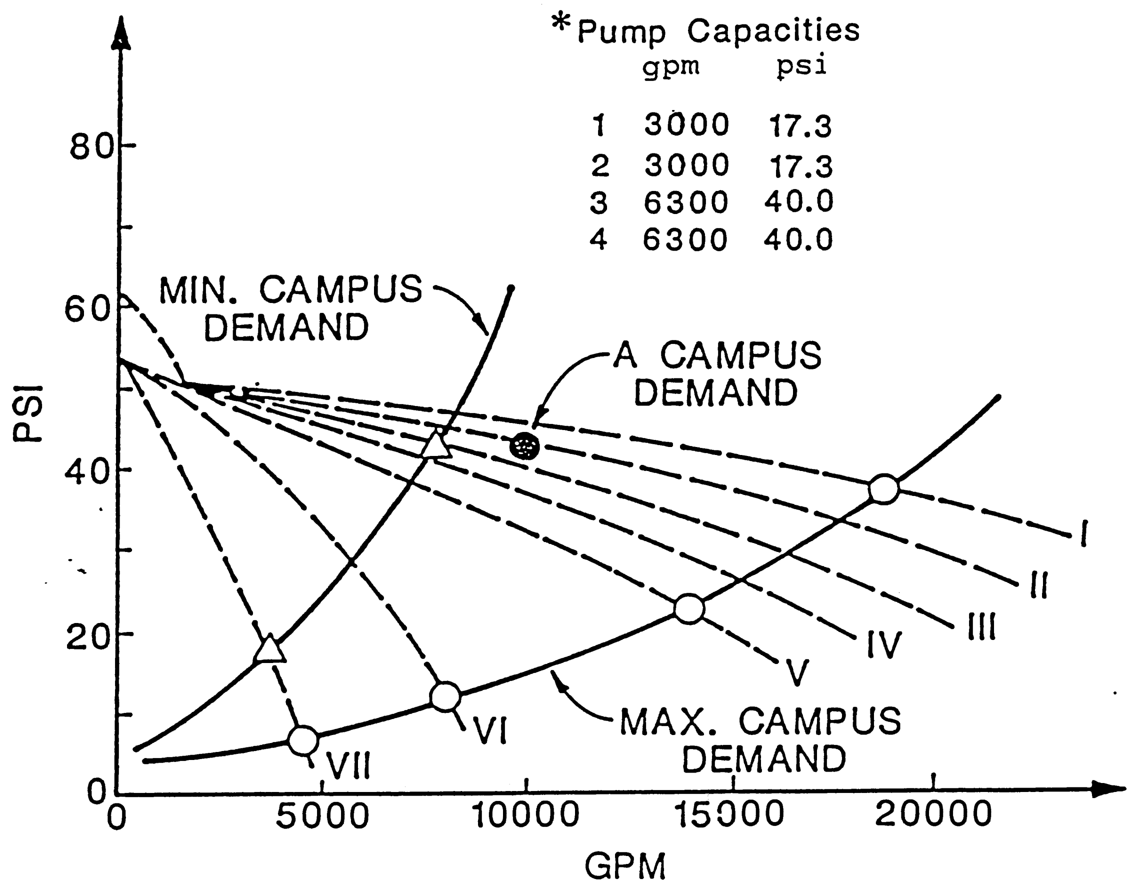


Figure 26. Optimum Operation of Distribution Pumps

pressure drop at the physical plant was 5.0 psi which is very close to the desired value as shown in Figure 27.

The major parameters tested by this model were:

1. distribution pump operating status (Figure 27)
2. piping arrangement and modifications (Figure 28. a & b)
3. future building expansion (Figure 29)

The effects of the three parameters on the effective pumping in the CCW system were examined.

4.8 Results of a CCW System Model

The OSU distribution model simulated the pressure drop and the flow rates in specified sections around the campus. The major parameters that varied were the distribution pump mode, the piping diameter modifications, and campus demand.

Figure 27 shows one of the simulation results to visualize the effect of distribution pump-status on head loss in a specific section. Figure 27 shows the difference of the pressures at several locations including branches depending on the mode of the distribution pumps. An optimum plan for the distribution pumping can be found by the simulations to match variable campus demand.

The model can be used to detect any strange sections which consume excessive pressure drops unnecessarily. Section #9 and section #12 were of concern because they created higher head losses compared to other sections based on simulation results as shown in Figure 28. The higher pressure drop between Willham Hall and the USDA building was due to the pipe length (1,450 feet away). The other higher head loss in section #12 was primarily due to the smaller diameter and the slant piping to

*** CAMPUS SIMULATION RESULTS ***		*** CAMPUS SIMULATION RESULTS ***	
- BRANCH DEMANDS, GPM -		- BRANCH DEMANDS, GPM -	
BRANCH I.D.	DEMANDS, GPM	BRANCH I.D.	DEMANDS, GPM
1	550.0000	1	550.0000
2	6000.000	2	6000.000
3	800.0000	3	800.0000
4	830.0000	4	830.0000
5	110.0000	5	110.0000
6	125.0000	6	125.0000
7	600.0000	7	600.0000
8	850.0000	8	850.0000
9	200.0000	9	200.0000
10	1000.000	10	1000.000
11	440.0000	11	440.0000
12	175.0000	12	175.0000
13	20.00000	13	20.00000
TOTAL DEMAND	11700.00 GPM	TOTAL DEMAND	11700.00 GPM
TOTAL LOAD	7800.000 TON	TOTAL LOAD	7800.000 TON
- DISTRIBUTION PUMP MODE -		- DISTRIBUTION PUMP MODE -	
MODE IS	4	MODE IS	5
- BRANCH HEAD, PSI	& COEFFICIENT, -	- BRANCH HEAD, PSI	& COEFFICIENT, -
33.43598	95.11646	27.65487	104.5868
31.78286	1064.277	26.00175	1176.657
28.03341	151.0957	22.25229	169.5909
26.09195	162.4893	20.31084	184.1680
24.18291	22.36858	18.40180	25.64263
21.85425	26.73881	16.07314	31.17882
19.69318	135.2052	13.91207	160.8627
18.97131	195.1508	13.19020	234.0417
14.08244	53.29556	8.301332	69.41544
13.41326	273.0441	7.632151	361.9733
13.35270	120.4116	7.571589	159.9039
11.51588	51.56909	5.734772	73.07687
10.81114	6.082669	5.030031	8.917532

Figure 27. The Effect of Pump Operation on Hydraulic Performance of the Distribution Line

the Diagnostic Lab from the Vet Med Hospital building. As in Figure 28.a, the original size of pipe in section #12 was 6 inches and the size was increased to 8 inches to see the effect of the modification on the hydraulic performance of the CCW system. The head loss with the 6 inch pipe was calculated at 0.9 psi when the result of Figure 28.a was obtained. The estimated head loss with the larger diameter pipe was about 0.2 psi with all other parameters remaining the same. This modification will allow operation of the present OSU CCW system with pumping head reduced from 28 psi to 26 psi.

The effect of future expansion may be predicted also with this model so that the pressure drops in every section for different operating conditions were estimated. The flow rates at branch #1 was increased by 1000 gpm to consider the possible additions of Cordell and Gallagher Hall to the main lines as shown in Figure 29. The possible expansion of the Noble Research Center was also considered by adding 500 gpm to Branch #12. An interesting result was obtained when more flows were added to the branches at the end of the distribution line such as expansion to the Vet Med Hospital building. A higher pumping head was required to provide the needed demand flow because the head losses developed due to the pipe length.

To summarize, several choices of distribution pump options were examined for the given amount of campus branch flow rates. Also, the effect of campus demand variations was studied with other parameters fixed. Some of the results were shown in Figures 27, 28, and 29. An optimum plan for the distribution pumps can be developed to match the varying demand to the multiple distribution pump operations.

*** CAMPUS SIMULATION RESULTS ***

- BRANCH DEMANDS, GPM -

BRANCH I.D.	DEMANDS, GPM
1	550.0000
2	6000.000
3	800.0000
4	830.0000
5	110.0000
6	125.0000
7	600.0000
8	850.0000
9	200.0000
10	1000.000
11	440.0000
12	175.0000
13	20.00000

TOTAL DEMAND 11700.00 GPM

TOTAL LOAD 7800.000 TON

- DISTRIBUTION PUMP MODE -

MODE IS 4

- BRANCH HEAD, PSI & COEFFICIENT, -

27.53619	104.8119
25.88307	1179.352
22.13362	170.0450
20.19217	184.7084
18.28312	25.72572
15.95446	31.29457
13.79339	161.5533
13.07152	235.1017
8.182655	69.91701
7.513474	364.8208
7.452911	161.1720
5.616095	73.84496
5.000000	8.944272

- SECTION DIAMETER, INCH -

SECTION I.D.	DIAMETER
1	30.00000
2	30.00000
3	19.00000
4	18.00000
5	16.00000
6	14.00000
7	14.00000
8	14.00000
9	14.00000
10	14.00000
11	14.00000
12	6.000000
13	3.000000
14	3.000000
15	6.000000
16	14.00000
17	14.00000
18	14.00000
19	14.00000
20	14.00000
21	14.00000
22	16.00000
23	18.00000
24	19.00000
25	30.00000
26	30.00000

- SECTION HEAD LOSS, PSI -

SECTION I.D.	HEAD LOSS, PSI
1	7.5842708E-02
2	0.8265576
3	1.874728
4	0.9707265
5	0.9545200
6	1.164332
7	1.080535
8	0.3609332
9	2.444434
10	0.3345906
11	3.0281466E-02
12	0.9282832
13	0.2659398
14	0.3501541
15	0.9085324
16	3.0281466E-02
17	0.3345906
18	2.444434
19	0.3609332
20	1.080535
21	1.164332
22	0.9545200
23	0.9707265
24	1.874728
25	0.8265576
26	0.1213484

Figure 28.a The Effect of a Line Size Modification (Before)

*** CAMPUS SIMULATION RESULTS ***

- BRANCH DEMANDS, GPM -

BRANCH I.D.	DEMANDS, GPM
1	550.0000
2	6000.000
3	800.0000
4	830.0000
5	110.0000
6	125.0000
7	600.0000
8	850.0000
9	200.0000
10	1000.000
11	440.0000
12	175.0000
13	20.00000

TOTAL DEMAND 11700.00 GPM

TOTAL LOAD 7800.000 TON

- DISTRIBUTION PUMP MODE -

MODE IS 4

- BRANCH HEAD, PSI & COEFFICIENT, -

25.75677	108.3720
24.10366	1222.109
20.35420	177.3221
18.41275	193.4277
16.50371	27.07709
14.17505	33.20074
12.01398	173.1043
11.29211	252.9481
6.403238	79.03695
5.734056	417.6082
5.673494	184.7257
5.616095	73.84496
5.000000	8.944272

- SECTION DIAMETER, INCH -

SECTION I.D.	DIAMETER
1	30.00000
2	30.00000
3	19.00000
4	18.00000
5	16.00000
6	14.00000
7	14.00000
8	14.00000
9	14.00000
10	14.00000
11	14.00000
12	8.000000
13	3.000000
14	3.000000
15	8.000000
16	14.00000
17	14.00000
18	14.00000
19	14.00000
20	14.00000
21	14.00000
22	16.00000
23	18.00000
24	19.00000
25	30.00000
26	30.00000

- SECTION HEAD LOSS, PSI -

SECTION I.D.	HEAD LOSS, PSI
1	7.5842708E-02
2	0.8265576
3	1.874728
4	0.9707265
5	0.9545200
6	1.164332
7	1.080535
8	0.3609332
9	2.444434
10	0.3345906
11	3.0281466E-02
12	0.2202860
13	0.3545864
14	0.3501541
15	0.2780915
16	3.0281466E-02
17	0.3345906
18	2.444434
19	0.3609332
20	1.080535
21	1.164332
22	0.9545200
23	0.9707265
24	1.874728
25	0.8265576
26	0.1213484

Figure 28.b The Effect of a Line Size Modification (After)

*** CAMPUS SIMULATION RESULTS ***

- BRANCH DEMANDS, GPM -

BRANCH I.D.	DEMANDS, GPM
1	1550.000
2	6500.000
3	800.0000
4	830.0000
5	110.0000
6	125.0000
7	600.0000
8	850.0000
9	200.0000
10	1000.000
11	440.0000
12	175.0000
13	20.00000

TOTAL DEMAND 13200.00 GPM

TOTAL LOAD 8800.000 TON

- DISTRIBUTION PUMP MODE -

MODE IS 4

- BRANCH HEAD, PSI & COEFFICIENT, -

27.68777	294.5694
25.88307	1277.631
22.13362	170.0450
20.19217	184.7084
18.28312	25.72572
15.95446	31.29457
13.79339	161.5533
13.07152	235.1017
8.182655	69.91701
7.513474	364.8208
7.452911	161.1720
5.616095	73.84496
5.000000	8.944272

*** CAMPUS SIMULATION RESULTS ***

- BRANCH DEMANDS, GPM -

BRANCH I.D.	DEMANDS, GPM
1	1550.000
2	6500.000
3	800.0000
4	830.0000
5	110.0000
6	125.0000
7	600.0000
8	850.0000
9	200.0000
10	1000.000
11	940.0000
12	175.0000
13	20.00000

TOTAL DEMAND 13700.00 GPM

TOTAL LOAD 9133.333 TON

- DISTRIBUTION PUMP MODE -

MODE IS 3

- BRANCH HEAD, PSI & COEFFICIENT, -

35.02899	261.8894
33.06606	1130.374
28.55321	149.7140
26.13979	162.3405
23.64989	22.61925
20.58827	27.54864
17.71927	142.5372
16.70352	207.9768
8.787445	67.46813
7.646397	361.6360
7.452911	344.3221
5.616095	73.84496
5.000000	8.944272

Figure 29. The Effect of Future Expansions

4.9 Contribution of This Thesis

According to the literature survey as of 1986, no method was available to analyze a large central chilled water system in a macroscopic way. Even Stoecker's routine was not able to deal with such a large system including the three loops simultaneously because too many variables are involved. Mostly it fails to converge or it took too much computing power, i.e. costs. This thesis developed a loop-by-loop analysis approach by applying direct or iterative method to each loop and by introducing a branch coefficient to couple those loops. As compared with the previous works in this area, the solution procedure is quite system oriented. This study covered the key points on the CCW system selection and operation by attacking the problems uniquely loop by loop. This loop-model approach to the CCW system simulation contributed to the following areas:

- development of a loop-model concept
- an optimum design of a secondary building loop
- parametric study on building loop
- parametric study on distribution loop

This study has developed a method to analyze a large campus type CCW system by an approach illustrated in Figure 30. The figure shows that the user decides the focus of simulation and, then, selected the model of concern. A macro distribution loop model is viewed as a large system model, which has two major inputs: one from the central plant and the other from the branches. The pressure rise to meet a campus demand was evaluated from the chiller loop study and the flow demands at each branch were estimated from the building loop study for the given load and design temperature difference. Building loops may be investigated

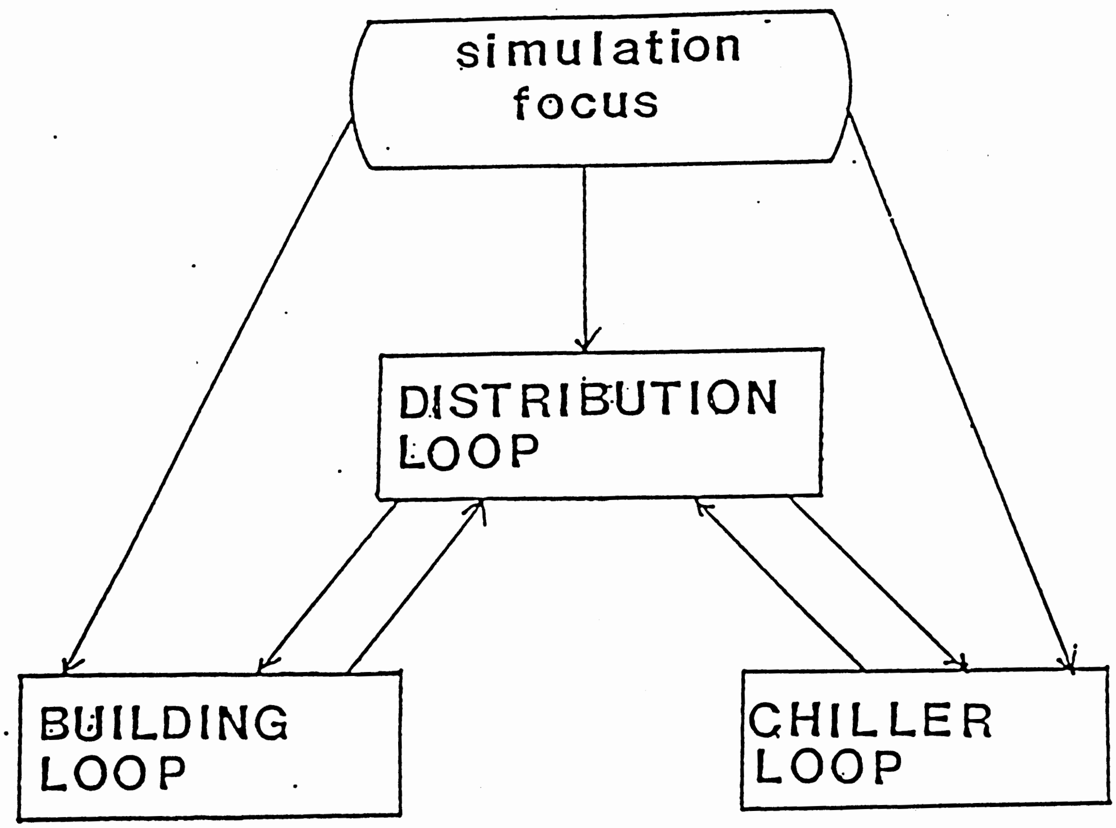


Figure 30. A Loop-Model Approach Concept

to obtain the following two major data: 1. amount of primary supply flow and 2. the temperature of the primary return.

The key concern of the building loop system was to provide a constant return temperature to the main distribution line. The proposed building loop was controlled by monitoring the temperature difference of the primary supply and the secondary supply. The building control valve was selected based on the procedure in Chapter 2.4.

It was possible to investigate a building loop by varying one of the parameters each time for the parametric studies. The system selected for this purpose was the Engineering North building. The building was modeled, simulated and data recorded to specifically check the return water temperature to the main distribution line.

Building valves with proportional control only were found to be unable to handle the control point of the secondary supply temperature and, therefore, integral controller was recommended and installed. A dynamic analysis with a set of initial optimum value of the P-I controller gains and the valve was conducted as in Chapter 2.4 to show the effect of the gains on the dynamic system performance.

Finally, a large distribution system including the data of buildings and central chiller loops was modeled and simulated based on the concept developed in this thesis. The model was useful in conducting parametric studies by changing operating parameters. An optimum plan for the distribution pumps was devised to pump the water more effectively to meet the building loads. One of the results indicated that the present distribution pumping capacity cannot carry a situation of unplanned expansion near the end of the distribution loop with present thermal loads near the central plant. Therefore, future expansions or deletions should be simulated before changes are actually carried out to avoid such a problem.

4.10 Procedure to Run the Programs

This section reviews the procedure to run the computer program for different operating conditions of two models: building loop simulation program and distribution loop simulation program. The general system-loop approach will be summarized again to apply this method to other CCW systems. Finally, several commands and operating systems to run the computer program in VAX system will be reviewed. The simulation procedure differs slightly depending upon the situations for the control valves. As stated in Chapter III, a direct substitution method was preferred to know the valve information while an iterative method was found better to predict variables when all the information of the component including the valve was given.

4.10.1 For Different Operating Conditions

In the case of the direct substitution model, data files such as EN.IN or OSU.IN should be modified to run the system with different operating parameters. The major parameters for the model were

PO,1,2,3 ... pump coefficients
DTS ... secondary design temperature difference
DTp ... primary design temperature difference
CDF(i) ... design flow for coil i, gpm
CDP(i) ... design head for coil i, psi
CCL(j) ... cooling load for coil j, ton
EL(k) ... equivalent length in section k, ft

ED(k) ... equivalent diameter in section k, in

The major parameters for the distribution model includes the following variables:

PC(i,j) ... pump coefficient j for option i

j = 1-4 & i = 1-5

IPUMP ... pump option i

BFR(k) ... flow rate at branch k

In the case of the iterative model, data files such as DATAB.FOR or DATAD.FOR should be modified to run the system with different operating parameters. The major parameters for building simulation program are listed below:

ITYPE - 1 for constant and 2 for variable speed pump

PC(i,j) - pump coefficients 1-j for i pump

VC(i,j) - valve coefficients 1-j for i coil control valve

BCV(i,j) - building control valve coefficient

VPUMP - pressure set across a variable speed pump

ITERMX - maximum allowable iteration number

IVARMX - maximum number of variable

ISECTN - number of section

NCOIL - number of coil

EPSL - a small number to stop the program

EL(n) - equivalent length of section n

ED(n) - equivalent diameter of section n

CVMX(i) - valve capacity, Cv of coil control valve, CCV

SP(x) - stem position of CCV

DVMX(1) - valve capacity, Cv of building control valve, BCV

DVSP(1) - stem position of building control valve

CCV2(i) - characteristic curve of CCV

DCV2(i) - characteristic curve of BCV

HLOAD(i) - Thermal load to each coil, ton

FDC(i) - design flow rate of coil, gpm

HDC(i) - design head loss of coil, psi

DELPSR - pressure differential across the distribution lines

TEMPSP - temperature of primary supply, F

RTSET - coil outlet set temperature, F

STSET - secondary supply set temperature, F

The major parameters involved in the distribution loop model are listed below:

ITERMX - maximum iteration number

IVARMX - maximum number of variables

NSEC - number of sections

NSUB - number of subbranches

EPSL - a small value to terminate the program

EL(n) - equivalent length of section n

ED(n) - equivalent diameter of section n

FR(k) - required flow rate of branch k

IPUMP - distribution pump mode

PC(1,j) - pump coefficient

The user should change the values of the above variables depending upon the system model of interest and the operating conditions.

4.10.2 For a Different System

With the direct approach, five steps may be applied again to a new system. The procedure is:

1. know flow rate in coil or branch
2. determine head losses in coil and piping sections
3. find the flow rate through the pump
4. calculate the pressure rise through the pump (or pumps)
5. determine the head across the control valve.

With the iterative approach, simulation of another CCW system requires the derivation of the system equation and their derivatives, which means to follow the same steps done for the OSU CCW system analysis. These necessary steps are:

1. Determine the focus of simulation
2. Draw the configuration of the system
3. Specify the system with nodes and sections
4. Identify the variable numbers
5. Set up a set of residual equations
6. Set up a set of partial derivative equations
7. Determine the values of the operating parameters
8. Give the initial values to the variables
9. Set the criteria to stop the program
10. Run the program

4.10.3 Program Execution in VAX System

The building simulation program may be run by typing the following command:

$ @ EN

for direct approach

$ @ BSIM

for iterative approach

Then, check the EN.OT or the RESULTB.OT for the summarized input and output results such as Figure 18.

The distribution simulation program may be run by typing the following command:

$ @ OSU

for direct approach

$ @ DSIM

for iterative approach

Then, check the OSU.OT or the RESULTD.OT for the summarized input and output results such as Figure 27. The following VAX command is required to get a hard copy at the room 216 in Engineering North.

$ PRINTOUT OSU.OT/Q=EN1/NOFLAG

Each command file includes the submodules as shown in Appendix E with the complete program listing.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

A loop approach was developed and applied to simulate the behavior of the OSU CCW system. The loop approach requires the study of every building of the campus. A building was selected and modified to one of the constant temperature systems considered in this thesis. The building loop should be changed into a variable flow system to control the temperature and rate of flow water to the coils. Coil valves are needed to maintain control of all the outlet water temperatures from the coil. The Engineering North building was chosen to confirm the control scheme analytically and experimentally. Computer programs were developed to calculate the pressure drops and the valve coefficient of the control valves of the coils and the building for the different cooling loads. The range of the operating valve coefficients was estimated to select optimum value of Cv for the coils and the building with a steady-state analysis. However, a dynamic analysis was necessary to detect the instability problems with the valve and the controller selected. The dynamic analysis with a simulation package PARASOL indicated a lower valve gain was preferred with a set of optimum controller gains, of proportional and integral functions. An equal percentage valve should provide a fine control action with a P-I controller setting.

The chiller loop study revealed some blending problems and control problems, one of the reasons was due to the improper location of the common pipe. The study compared two systems: pre-common pipe system and post-common pipe system. The post-common pipe system was proposed because the post-common pipe system could prevent the part load operation of all of the chillers and the mixing between the warm campus return and the cool by-passing flow when the campus load was reduced. It is easier to control the chillers with the post-common pipe system.

An optimum plan for a campus demand was developed to pump the water more effectively. A tight control of the building flow at near the end of the distribution line was recommended to prevent the excessive pumping head loss in the central plant. A future addition of Gallagher Hall and Cordell Hall was simulated and an optimum distribution pumping technique was developed for the feasible expansion. However, some addition at near the end of the distribution lines should be carefully conducted because the flow increase at the location such as the Vet Med Hospital building contributes much to the head losses due to the long pipe. Based on the simulation results and the actual data, several modifications have been recommended for the building loops, the chiller loops, and the distribution loops. This chapter presents the details of the proposed system configurations and some recommendations based on the simulation study.

5.1 For Building Loop

One of the goals in this study was to design and test a constant return water temperature system, requiring a variable flow system in the secondary building loop. A control method of Figure 20 was proposed to achieve the objectives in the buildings. The control scheme was tested for several possible situations with different cooling loads. The Engineering North building was modeled and simulated to test and validate the major parameters such as return water temperature and the primary flow to the main. The modified Engineering North building loop was also instrumented to confirm the working variable flow system design.

The building loop was modeled to calculate the pressure drop across the coil control valves and the required amount of primary flow for the given thermal loads. Based on some experimental results, the actual building temperature was still fluctuating as the coil load varied due to the uncontrolled coils. All the coils in the building must control the outlet water temperatures by installing a suitable valve and controller. A special check valve on the common pipe would be needed to allow only one-directional flow. A water-to-water heat exchanger, which is in the Student Union at OSU, would separate the building loop and the distribution loop hydraulically, although this is an expensive approach.

Most of the terminal cooling coils were overdesigned and the coils were operating at partial loads. However, it was difficult to check whether the controller and valve were controlling the water outlet temperatures properly. The constant temperatures from the coils are very important because the proposed method (supply water control method) can not control the main return water temperature directly. Therefore, the optimum selection of the coil control valve and controller gains would be another important problem to be answered. The building pumps were also generally oversized. Several simulations were conducted to find an optimum range, which is quite dependent on the diversity factor of a building load.

Most of the campus buildings at OSU need the building pump because they are located far away from the central plant and mostly are high rise buildings. Some buildings do not require the building pump, but they still have to be equipped with the coil control valves and a check valve on the common pipe. This type of system is suitable for the low-rise buildings near the central plant. Two groups can be made for further modifications in OSU campus buildings:

Group I - low-rise buildings near the central plant

Example: Printing and Publication building
Poultry building
Meat Lab
Student Health Center

Group II - high-rise buildings

Example: Dormitory buildings
Most Buildings other than Group I buildings

The proposed system requires the other major components such as temperature sensor, controllers, and actuators and valves. Haines (46) stressed the electronic direct digital controller (DDC) more than the pneumatic controller. DDC has more flexibility in use because it can be programmed for the different conditions rather than wired every time with an analogue controller. An electro-magnetic control valve would be appropriate if a DDC was selected for easy programmability. The valves are selected and sized based on the procedure in Chapter 2.4. The type of control valve should be an equal percentage. The size of the valve can be decided in terms of Cv in a steady-state sense. But, an optimum set of controller gains and valve gains were necessary for a better dynamic system performance. The time response of the selected system can be simulated with PARASOL and the selected system has two system blocks: digital controller and analogue system in case a DDC was used. An example run to decide a set of optimum values for the gains was in Chapter 2.4.

5.2 For Chiller Loop

The present piping layout of OSU central chilled water plant is shown in Figure 23. One of the interesting problems was the location of the common pipe and its effect on overall system performance. At present, the common pipe is located before any chiller connections (pre-common pipe system). The common pipe was about five feet long and 20 inches in diameter to connect the chiller loops and the distribution loop as shown in Figure 23, which is similar to one of Figure 4.b (pre-common pipe system).

The central chiller plant study of Chapter IV recommended the relocation of the common pipe to the other side (post-common pipe system) as shown in Figure 24. As in Figure 24, the common pipe of the present pre-common pipe system, which is located prior to any chiller connections should be removed or blocked and a new common pipe at the designated location should be placed to prevent blending under design load conditions and to control the multiple chillers optimally. This blending, due to the location of the common pipe, was one of the major sources of the inefficient operation of the multiple chillers. This mixing resulted in warm primary supply temperature and cool inlet temperature to the chillers depending on the direction of the flow in the common pipe at a partial load.

A monitoring system with a flow meter and switch on the common pipe is recommended. The poor location of the present common pipe in conjunction with the mismatched distribution flow caused the blendings at the bypass section.

All of the distribution pumps were connected in parallel. A set of pump curves with possible system curves was obtained as in Figure 26. An equivalent pump can be selected based on the status of the distribution pumps (on or off). This pump equation depending on the option was used to estimate the value of the pressure rise through the pump. This curve provides one of two important inputs for the following overall distribution loop study.

5.3 For Overall OSU CCW System

The OSU CCW system was modeled and simulated with the aid of the developed loop-approach. The loop-approach is to analyze the macro distribution loops having the two major inputs: one from the central plant in terms of the pump operation and the other from the branches in terms of flow rate. The loop-approach was very useful for investigating large CCW system loops which have three distinctive loops: building loops, chiller loops and distribution loops. The simulation requires the flow rates at each branch to find an optimum pumping method. Thus, the effect of the pump status at the central plant (Figure 27), piping size modifications (Figure 28) and future expansions (Figure 29) are investigated to propose a better pumping method and to operate the chillers more effectively. The map of distribution pumps was developed to match the demand and supply of chilled water as in Figure 26.

The distribution loop simulation model (DSIM) indicated several excessive head loss sections for given flow demand conditions and pump option. These sections include the section between Willham Hall and USDA Building (section #9) and the section between Vet Med Hospital buildings and the Diagnostic Lab (section #12) as shown in Figure 25. It was interpreted that the long pipe from Willham Hall to the USDA building caused the high pressure drop at the section #9. And, the high pressure drop through the section #12 was due to the slant piping and small size of the pipe to the Physical Plant building from the Diagnostic Lab building. The signals from the farthest points control the speed of the variable speed motor in the central plant to maintain a preset pressure at those locations. Therefore, unnecessary head loss in the distribution lines must be avoided to pump the water more effectively. A modification to change the diameters of the pipings is proposed as the campus demand increases for the future (see Figure 28). A reduction in head loss was observed by increasing the pipe diameter from 6 inches to 8 inches. The head loss was reduced from 0.9 psi to 0.2 psi with other conditions kept the same as in Figure 28. The modification on the pipe diameter led to decreased pumping power at the central plant of about 2.0 psi as shown in Figure 28.a and b.

A feasible expansion for the future addition of Cordell Hall and Gallagher Hall was tested by adding more flow to Branch #1 at 1,000 gpm. To test the expansion of the Noble Research Center 500 gpm was added to Branch #2. The needed pumping head at the central plant was about 28 psi as shown in Figure 29 for the given flow demand. A simulation result showed that some expansion at the end of the direct-return type distribution loop must be carefully added because the flow increase at the locations is very sensitive to the required pumping head at the central plant. Finally, a plant for optimum operation distribution pumps was obtained from Figure 26 by matching the demand and supply as closely as possible.

Interestingly, the major source of most problems in CCW systems was originated from the poor selection of the building control valve. Therefore, a method to select the control valves for the coils and building loops has been prepared by using both the steady-state and the dynamic analysis.

The primary task is to reduce the primary pumping power and CCW system operation costs. To achieve the main goals, the modifications for each loop were considered. A control method (supply water control method) for the building loops was proposed and tested by programs and experiments. A steady-state analysis using the iterative method confirmed the control method. Some experimental data on the Engineering North building showed that the return temperatures were varied because four of the nine cooling coils were not controlling the water outlet temperatures properly. Thus, a more tight control of the cooling coils was recommended with an equal percentage type of control valve. The pressure drops over the coil control valve were estimated for the different cooling loads. The high pressure drops imposed on the control valve at the low load were observed and contributed to the valve instability. A smaller pump (16 psi size) was replaced for the oversized building pump for that reason. A method to control the speed of the pump was necessary by sensing a preset pressure at the farthest coil from the building pump as in Figure 21, which is just the same as the distribution pumps were controlled.

5.4 Suggestion for Further Study

Problems such as coil simulation with different control methods should be investigated further to test the performance of each coil. The source of the poor cooling in certain zones of a building might be from the poor performance of the coil.

A real-time load monitoring system should be studied more by utilizing a micro-processor and the concept of the enthalpy method. Inlet/outlet wet-bulb temperature with an air-flow meter can predict actual load on the air-side and this ultimately controls the flow of water in the coils.

Building control valves should be selected and studied more thoroughly by utilizing a dynamic analysis. A method to control the speed of the building pump should be devised to meet the coil flow demand as sensitively as possible.

Any buildings included in a branch can be studied by considering the branch alone as the main distribution lines investigated. For instance, the pressure drop across Engineering North may be calculated with the branch #2, starting from the section #2 to the section #25 because the Engineering North building is included in that branch. The same method as for the distribution study can be applied to get the value for the pressure drop across the Engineering North building.

A complete package of a large CCW system model requires a lot of data and time for every building and branch. This work could be completed if all the necessary data in every building were available. Some more efforts should be given to complete this package, using methods that may be developed in the future.

CHAPTER VI

SUMMARY

A general method was developed to simulate a large campus central chilled water system. The simulation program predicts the pressures, flows, and temperatures for a system loop under given operating conditions, loop by loop. The system loops considered are building loop, chiller loop, and distribution loop. The basic algorithm to describe the loops came from the hydronic loop law, also known as Kirchhoff's law.

This approach was applied to simulate the various loops of the OSU CCW system. The purpose of the simulation was to provide information on an optimum plan for a given campus flow demand. Several results were obtained for each loop and some modifications were recommended based on analysis of these results.

The building loop model simulations examined a constant temperature system among the possible alternatives. A building control valve with secondary supply water temperature sensor (supply water control method) brings a constant temperature to the cooling coils. All the coils should be installed with proper controller and control valve to keep the water outlet temperature constant. An equal percentage is best for the control valve because of the low gain at the low load. The size and the gains should be obtained using both a steady-state and dynamic analysis. This will improve the overall efficiency of the multiple chiller operation in the central plant.

The modification of the existing system into a constant temperature system will solve most of the problems and make it easy to control the total campus demand and the multiple chillers' supply. The proposed building loop control method was tested on Engineering North building. Some experimental data during low loads showed instability problems, because the controller had just a proportional function. The system responded too quickly. An additional integral function is recommended to avoid the instability problems. A dynamic analysis using the simulation package PARASOL may provide an optimum set of the gains of the controllers and the valve. The building pump flow rates must be controlled to be sensitive to the load change. All of the coil outlet water temperatures must also be controlled to make the proposed control method work.

For the central chiller loops, the by-pass line connecting the chiller loops and the distribution loops should be moved into the back-side as in a post-common pipe system to prevent the blending and to improve system control and efficiency. A series of characteristic curves of the distribution pumps was obtained to provide information for the overall distribution loop simulation.

A distribution loop model from the building loop and chiller loop study was developed to visualize the effect of piping modifications, and future expansion. The distribution loop model simulation indicated several excessive head loss sections described in the previous section. The distribution loop simulation showed the effect of increasing the pipe diameter in a particular section and the reduction in pumping power for the same distribution of loads. An expansion without considering the other effects, especially at the end of the distribution line may cause distribution problems. Detailed performance of the distribution pumps was developed and is available for any future expansion study.

BIBLIOGRAPHY

1. Winner, W. O., "Research Needs in Thermal Systems: Report of the Select Panel on Research Goals and Priorities in Thermal Systems," Mechanical Engineering, Aug 1986, pp. 39-46.

2. American Society of Heating Refrigeration and Air Conditioning Engineers: Handbook Series (Four Volume) Fundamentals, Equipments, Systems, and Applications.

3. Pearson, J. T., Ed., Central Chilled Water Conference Proceedings and Addendum, Purdue University, West Lafayette, Indiana, 1976.

4. Hansen, E. G., Hydronic System Design and Operation, McGraw-Hill Book Co., New York, 1985.

5. McQuiston, F. C. and J. D. Parker, Heating, Ventilation, and Air Conditioning, New York: John Wiley and Sons, 1982.

6. Notes from Professor F. C. McQuiston's Lecture on Advanced HVAC Design and Analysis, March 1984.

7. Parker, J. D., Feasibility Study for Central Chilled Water Plant and Distribution, Dept. of Mechanical and Aerospace Engineering, Oklahoma State University, 1974.

8. Hodge, B. K., Analysis and Design of Energy Systems, Prentice-Hall Inc., New Jersey, 1985.

9. Pita, E. G., Air Conditioning Principles and Systems: An Energy Approach, John Wiley & Sons, Inc., New York, 1981.

10. Ingels, D. M., Computer Modeling and Simulation in Chemical Engineering, John Wiley and Sons, New York, 1972.

11. Franks, R. G., Modeling and Simulation in Chemical Engineering, John Wiley and Sons, New York, 1972.

12. Carlson, G. F., "Central Plant Chilled Water Systems-Pumping & Flow Balance: Part I, II, III," ASHRAE Journal, April 1972, pp. 42-48.

13. Coad, W. J., "Variable Flow in Hydronic Systems for Improved Stability, Simplicity, and Energy Economics," ASHRAE Transaction, Vol. 91, Pt. 1, 1985.

14. Coad, W. J., "Integrated Loop System for Campus Cooling," Heating, Piping, and Air Conditioning, May 1973, pp. 84-91.

15. Coad, W .J., "Water Chillers: Changing Technology," Heating, Piping, and Air Conditioning, July 1981, pp. 35-41.

16. Stoecker, W. F., "A Generalized Program for Steady-State System Simulation," ASHRAE Semiannual Meeting, Philadelphia, Pennsylvania, Jan. 24-28, 1971.

17. Stoecker, W. F., S. R. Richter, G. E. Higges, and A. R. Baker, "A Computer Program to Simulate the Flow Rates, Temperatures, and Pressures of a Central Chilled Water System," ASHRAE Transactions, Vol. 82, Part I, 1976.

18. Al-Quarani, "Simulation of Flow Rates, Pressures, and Temperatures in Central Chilled Water Systems," M.S. Thesis, Dept. of Mechanical and Industrial Engr., Univ. of Illinois at Urbana-Champaign, Urbana, Ill., 1977.

19. Stoecker, W. F., Refrigeration and Air Conditioning, 2nd Ed., McGraw-Hill Book Co., New York, 1985.

20. Stoecker, W. F., Design of Thermal Systems, 2nd Ed., McGraw-Hill Book Co., New York, 1980.

21. Wood, D. J., "Computer-Aided Analysis of Piping Systems," Plant Engineering, Sept. 15, 1979, pp. 183-193.

22. Wood., D. J. User's Manual Computer Analysis of Flow in Pipe Networks Including Extended Period Simulation, Univ. of Kentucky, Lexington, 1980.

23. "Flow of Fluids Through Valves, Fittings, and Pipes," Technical Paper No. 410., The Crane Co., Chicago, Ill., 1976.

24. Karassik, J., Centrifugal Pump Clinic, Marcel Dekker, Inc., New York, 1981.

25. McNaughton, K. J., The Chemical Engineering Guide to Pumps, McGraw-Hill Publishing Co., New York, 1984.

26. Greene, R. W., The Chemical Engineering Guide to Valves, McGraw-Hill Publishing Co., New York, 1984.

27. Computerized Evaluation of Energy Requirement, Equipment Selection, and Economic Comparison for Building Systems User's Manual, Gate Information Center, Southwest Research Institute, San Antonio, Texas, October, 1971.

28. Notes from K. N. Reid's Lecture on Advanced Fluid Power Control, March 1983.

29. Hutchison, J. W., ISA Handbook of Control Valves, 2nd Ed., Instrument Society of America, 1976.

30. Gladstone, J., Air Conditioning Testing/Adjusting/Balancing A Field Practice Manual, 2nd Ed., Van Nostrand Reinhold Company, New York, 1981.

31. Carlson, G. F., "Pump Energy and Flow Balance Analysis." Heating, Piping, and Air Conditioning, October 1982, pp. 71-84.

32. Hibbard, D., "Retrofit-Balancing of Hydronic System," Heating, Piping, and Air Conditioning, March 1984, pp. 105-110.

33. Merrick, R. C., "A Guide to Selecting Manual Valve," Chemical Engineering, Sept. 1 1986, pp. 52-62.

34. Phillips, D., "Energy Conservation with Automatic Flow Control Valves," Heating, Piping, and Air Conditioning, Dec. 1984, pp. 91-92.

35. Spethmann, D. H., "Load Reset" A Basic Optimization Tool," Heating, Piping, and Air Conditioning, Dec 1977, pp. 35-40.

36. Spethmann, D. H., "Optimized Control of Multiple Chillers," ASHRAE Transaction, Vol. 91, Pt. 2.

37. Bailey, S. J., "Software Power is the Strategic Value Added to IBM PCs Used in Control," Control Engineering, June 1986, pp. 81-85.

38. Tao, W., "Coil Design and Selection," Heating, Piping, and Air Conditioning, Dec 1985, pp. 66-73.

39. Coad, W. J., "Ideal Terminal Control and Distribution System: I," Heating, Piping, and Air Conditioning, Oct 1984, pp. 127-128.

40. Goff, K. W., "A Systematic Approach to DDC Design," ISA Journal, Dec 1966, pp. 44-54.

41. Hamilton, D. C., Leonard R. G., and Pearson J. T., "Dynamic Response Characteristics of a Discharge Air Temperature Control System at Near Full Load and Part Heating Load," ASHRAE Trans, Vol. 80, Part I, 1974, pp. 181-194.

42. Nesler, C. G. and Stoecker, W. F., "Selecting the Proportional and Integral Constants in the Direct Digital Control of Discharge Air Temperature," ASHRAE Trans, Vol. 90, Part 2B, pp. 834-844.

43. Major Control Companies' Booklets on the Fundamentals of Control of Thermal Systems in Buildings.

44. Stoecker, W. F., Rosario, L. A., Heidenreich, M. E., and Phelan, T. R., "Stability of an Air-Temperature Control Loop," ASHRAE Trans, Vol. 84, Part I, 1978, pp. 35-53.

45. Symposium on HVAC Control Process Modeling, ASHRAE Trans., Vol. 88, Pt. 1.

46. Haines, R. W., "DDC Operating Descriptions," Heating, Piping, and Air Conditioning, Dec 1985, pp. 121-122.

47. Haines, R. W., "Monitoring and Control Without Computers," Heating, Piping, and Air Conditioning, Oct 1984, pp. 125.

48. Haines, R. W., "Monitoring and Control Systems: IX," Heating, Piping, and Air Conditioning, April 1983, pp. 83-86.

49. Haines, R. W., "Software for DDC-II," Heating, Piping, and Air Conditioning, July 1985, pp. 116-117.

50. Haines, R. W., "Central Chilled Water Plant Control: II," Heating, Piping, and Air Conditioning, Jan 1981, pp. 135-136.

51. Haines, R. W., "Control Valve Selection," Heating, Piping, and Air Conditioning, Sept 1980, pp. 100-102.

52. Notes from Computer Based Monitoring and Control System of R. W. Haines in Oklahoma Engineering Foundation, Oklahoma City, March 1, 1985.

53. Haines, R. W., Control Systems for Heating, Ventilation and Air Conditioning, 3rd Ed., Van Nostrand Reinhold Co., New York, 1983.

54. Hansen, E. G., "Central Chilled Water Distribution Systems Design," ASHRAE Journal, Nov 1974, pp. 47-52.

55. Bahnfleth, D. R., "Pumping Interfaces Between the Chiller Plant Distribution System and the Buildings,"Purdue University Central Chilled Water Conference Proceedings. 1976.

56. Mannion, G. F. and J. R. Mannion, "Diversification of Large Central Plant Pumping: a Solution," Heating, Piping, and Air Conditioning, Dec 1972, pp. 56-60.

57. Mannion, G. F., U. S. Patent No. 3729,051.

58. Rishel, J. B., "System Analysis for a Campus-Type Chilled Water System," ASHRAE Transaction, Vol. 91., Pt. 1, 1985.

59. Burr, G. C. and Pate III, M. E., "Conversion of Campus Central Plant from Constant Flow to Variable Flow at Univ. of West Florida," ASHRAE Transaction, Vol. 90. Pt. 1., 1985.

60. Climatological Data of Stillwater, Oklahoma 1893-1980, Research Report P-821, Feb 1982, Agricultural Experimentation Station, Division of Agriculture, O.S.U.

61. Parker, J. D. and P. M. Moretti, "A Study of Chilled Water Expansion for the OSU Campus," Dept. of Mechanical and Aerospace Engineering, O.S.U., Stillwater, Oklahoma, 1985.

62. McKinzie, V., Personal Communication, Engineering Division in Physical Plant at O.S.U., 1985.

63. Cummings. L., Internal Memorandum, Power Plant, O.S.U., 1984.

64. Hakel, T., Personal Communication, HVAC Technician in Physical Plant at O.S.U., 1986.

65. Shinskey, F. G., Process-Control System, McGraw-Hill Inc., New York, 1979.

APPENDIX A

SAMPLE SIMULATION RESULT OF A BUILDING LOOP

*** CHILLED WATER CIRCUIT SIMULATION INPUT ***

VARIABLE FLOW SYSTEM

COIL SET TEMP(F)	56
PUMP SET TEMP(F)	44
BLDG SET PRESSURE(PSI)	20
MAIN DIFFERENTIAL PRESSURE(PSI)	20
TOTAL COIL	9

COOLING LOAD

COIL I.D. #	LOAD(ton)
1	20
2	20
3	20
4	20
5	20
6	20
7	20
8	20
9	20

HYDAULIC RESULTS:

COIL I.D. #	FLOW(gpm)
1	28.88
2	28.99
3	29.60
4	29.52
5	29.40
6	30.09
7	29.87
8	26.80
9	29.11
BLDG FLOW	264.27
BY-PASS FLOW	38.34
PRIMARY FLOW	225.93

THERMAL RESULTS:

COIL I.D. #	OUTLET WATER TEMP(F)
1	56.6
2	56.6
3	56.2
4	56.3
5	56.3
6	56.0
7	56.1
8	56.7
9	56.4
BLDG SUPPLY	42.4

SIMULATION COMPLETED

APPENDIX B

SAMPLE SIMULATION RESULT OF DISTRIBUTION LOOP

*** DISTRIBUTION WATER CIRCUIT SIMULATION INPUTS ***

DISTRIBUTION PUMP OPTION IS 1

PUMP 1,2,3, AND 4 ON

SECTION LENGTH (FT) & DIAMETER(IN)

50.00000	30.00000
600.0000	30.00000
650.0000	19.00000
360.0000	18.00000
300.0000	16.00000
200.0000	14.00000
200.0000	14.00000
100.0000	14.00000
1450.000	14.00000
250.0000	14.00000
150.0000	13.00000
705.0000	12.00000
800.0000	6.000000
80.00000	30.00000
600.0000	30.00000
650.0000	19.00000
360.0000	18.00000
300.0000	16.00000
200.0000	14.00000
200.0000	14.00000
100.0000	14.00000
1450.000	14.00000
250.0000	14.00000
150.0000	13.00000
690.0000	12.00000
790.0000	6.000000

REQUIRED BRANCH FLOW RATE (GPM)

900.0000	DEMAND (GPM) AT BRANCH	1
4500.000	DEMAND (GPM) AT BRANCH	2
900.0000	DEMAND (GPM) AT BRANCH	3
800.0000	DEMAND (GPM) AT BRANCH	4
120.0000	DEMAND (GPM) AT BRANCH	5
130.0000	DEMAND (GPM) AT BRANCH	6
160.0000	DEMAND (GPM) AT BRANCH	7
120.0000	DEMAND (GPM) AT BRANCH	8
155.0000	DEMAND (GPM) AT BRANCH	9
130.0000	DEMAND (GPM) AT BRANCH	10
220.0000	DEMAND (GPM) AT BRANCH	11
10.00000	DEMAND (GPM) AT BRANCH	12
25.00000	DEMAND (GPM) AT BRANCH	13

*** DISTRIBUTION WATER CIRCUIT SIMULATION OUTPUTS ***

VARIABLE VALUES

3.7082721E-02	HEAD(PSI) OF SECTION	1
0.4449927	HEAD(PSI) OF SECTION	2
0.5438368	HEAD(PSI) OF SECTION	3
3.433585	HEAD(PSI) OF SECTION	4
8.8440798E-02	HEAD(PSI) OF SECTION	5
0.7703944	HEAD(PSI) OF SECTION	6
6.7512050E-02	HEAD(PSI) OF SECTION	7
3.350948	HEAD(PSI) OF SECTION	8
0.2122653	HEAD(PSI) OF SECTION	9
0.1436915	HEAD(PSI) OF SECTION	10
7.0928149E-03	HEAD(PSI) OF SECTION	11
5.869569	HEAD(PSI) OF SECTION	12
1.7361110E-02	HEAD(PSI) OF SECTION	13
5.9768884E-04	HEAD(PSI) OF SECTION	14
0.0000000E+00	HEAD(PSI) OF SECTION	15
4.731003	HEAD(PSI) OF SECTION	16
0.0000000E+00	HEAD(PSI) OF SECTION	17
2.2525406E-02	HEAD(PSI) OF SECTION	18
0.0000000E+00	HEAD(PSI) OF SECTION	19
0.1149532	HEAD(PSI) OF SECTION	20
0.0000000E+00	HEAD(PSI) OF SECTION	21
4.7333863E-02	HEAD(PSI) OF SECTION	22
0.0000000E+00	HEAD(PSI) OF SECTION	23
0.8369467	HEAD(PSI) OF SECTION	24
0.0000000E+00	HEAD(PSI) OF SECTION	25
1.7144097E-02	HEAD(PSI) OF SECTION	26

129.3718	FLOW COEFF OF BRANCH	1
649.8534	FLOW COEFF OF BRANCH	2
137.7691	FLOW COEFF OF BRANCH	3
127.7066	FLOW COEFF OF BRANCH	4
19.18313	FLOW COEFF OF BRANCH	5
20.98937	FLOW COEFF OF BRANCH	6
25.89472	FLOW COEFF OF BRANCH	7
20.33389	FLOW COEFF OF BRANCH	8
26.36305	FLOW COEFF OF BRANCH	9
22.15705	FLOW COEFF OF BRANCH	10
37.96486	FLOW COEFF OF BRANCH	11
1.899668	FLOW COEFF OF BRANCH	12
4.752129	FLOW COEFF OF BRANCH	13

48.39561	HEAD(PSI) OF BRANCH	1
47.95062	HEAD(PSI) OF BRANCH	2
42.67578	HEAD(PSI) OF BRANCH	3
39.24220	HEAD(PSI) OF BRANCH	4
39.13123	HEAD(PSI) OF BRANCH	5
38.36084	HEAD(PSI) OF BRANCH	6
38.17837	HEAD(PSI) OF BRANCH	7
34.82742	HEAD(PSI) OF BRANCH	8
34.56783	HEAD(PSI) OF BRANCH	9
34.42413	HEAD(PSI) OF BRANCH	10
33.58009	HEAD(PSI) OF BRANCH	11
27.71053	HEAD(PSI) OF BRANCH	12
27.67602	HEAD(PSI) OF BRANCH	13

SIMULATION COMPLETED

APPENDIX C

CURVE FITTING FOR PUMPS AND VALVE

Suppose the following data were obtained from a pump curve.

FLOWRATE(gpm)	HEAD(psi)
0	53
50	52
100	49
150	42

It is desired to determine the coefficients P_1, P_2, P_3, and P_4 from the following relationship:

$$HD = P_1 + P_2*W + P_3*W^2 + P_4*W^3$$

Where

P_1, P_2, P_3, and P_4	pump coefficients
W	water flow rate(gpm)
HD	pressure drop(psi)

The third degree curve can be passed through these points and four equations can be obtained for the four data points by inserting the respective flows and heads in the pump characteristic equations:

$$53 = P_1 + P_2*0 + P_3*0^2 + P_4*0^3$$
$$52 = P_1 + P_2*(50) + P_3*(50)^2 + P_4*(50)^3$$
$$47 = P_1 + P_2*(100) + P_3*(100)^2 + P_4*(100)^3$$
$$42 = P_1 + P_2*(150) + P_3*(150)^2 + P_4*(150)^3$$

These are solved simultaneously to give the following coefficients:

P_1 = 53

P_2 = 1.9E-2

P_3 = -9.9E-4

P_4 = 3.9E-6

The above result can be obtained by typing the following command:

```
$ @ PUMP
```

Then, it will print out the coefficients in VAX computer system. The program is located at [U5754AB]PUMP.FOR and the PUMP.COM is a command file which has the following commands:

```
$ FOR PUMP
$ LINK PUMP,LESYQ,GAUSSY
$ RUN PUMP
```

APPENDIX D

EQUIPMENT PERFORMANCE DATA

The 2400 ton Steam Chiller

[illegible] SERIAL NO. [illegible]

MACHINE OPERATING CHARGE [illegible] LB [illegible]

COMPONENT	ITEM	HORSEPOWER	RPM	BRAND, MODEL, REMARKS
COMPRESSOR	100	[illegible]	[illegible]	CARRIER [illegible]
DRIVER	200	[illegible]	[illegible] FULL LOAD	[illegible]
SPEED INCREASER	600		[illegible]	

ITEM	100	[illegible]	200	300		500	703	600
VESSELS — COMPONENT	COMP. OIL COOLER	OIL COOLER	COOLER	REFRIG. CONDENSER		[illegible]	PURGE CONDENSER	STEAM CONDENSER
BRAND	[illegible]		CARRIER	CARRIER		CARRIER	CARRIER	[illegible]
MODEL	[illegible]		17J061	17J061		[illegible]	BF 20	
TUBES — DESIGN SCALE	[illegible]		.0005	[illegible]			.005	
NO. PASSES	2		1	•			2	
LENGTH, FT.	[illegible]		22	15			40.2	
QUANTITY	[illegible]		922	1650			COIL	
GAUGE, BAG & FIN.	[illegible]						19	
O.D., IN.	[illegible]		3/4	3/4			3/4	
SURFACE, [illegible]	[illegible]		[illegible]	[illegible]			8.75	
TUBE SIDE — MATERIAL	[illegible]		FINNED COPPER	FINNED COPPER			COPPER	
FLUID — TYPE	CLEAR WATER		CLEAR WATER	CLEAR WATER			CLEAR WATER	
LOAD, TONS	•		[illegible]	•			•	
FLOW, GPM	10		[illegible]	[illegible]			15	
Δ P, PSI	1.5		[illegible]	[illegible]			11	
ENTG., °F	85		[illegible]	[illegible]			85	
LVG., °F	95		[illegible]	95			95	
WORKING, PSIG	[illegible]		150	150			150	
DESIGN, PSIG	[illegible]		150	150			150	
SHELL SIDE — DESIGN, PSIG	[illegible]		150	185		225	385	
RELIEF, PSIG			150	•		225	385	
REFRIGERANT TEMP., °F			[illegible]					

17M-44

The 1200 ton Steam Chiller

[illegible] SERIAL NO. [illegible]

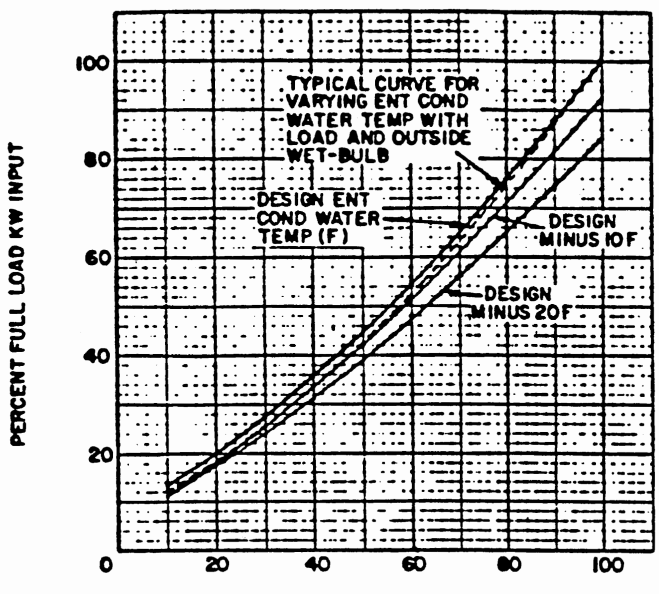
MACHINE OPERATING CHARGE [illegible] LB [illegible]

COMPONENT	ITEM	HORSEPOWER	RPM	BRAND, MODEL, REMARKS
COMPRESSOR	100	[illegible]	[illegible]	CARRIER [illegible]
DRIVER	200	[illegible]	[illegible] FULL LOAD	TURBINE [illegible]
SPEED INCREASER	600		[illegible]	

ITEM	100	[illegible]	200	300			703	600
VESSELS — COMPONENT	COMP. OIL COOLER	OIL COOLER	COOLER	REFRIG. CONDENSER			PURGE CONDENSER	STEAM CONDENSER
BRAND	[illegible]		CARRIER	CARRIER			CARRIER	[illegible]
MODEL	[illegible]		17FA87	17FA89			[illegible]	
TUBES — DESIGN SCALE			.0005	[illegible]				
NO. PASSES	[illegible]		1	2			[illegible]	
LENGTH, FT.	[illegible]		[illegible]	[illegible]				
QUANTITY	[illegible]		[illegible]	[illegible]			1	
GAUGE, BAG & FIN.			.035	.035			20	
O.D., IN.	3/4		3/4	3/4				
SURFACE, [illegible]	[illegible]		[illegible]	[illegible]			[illegible]	
TUBE SIDE — MATERIAL	COPPER		FINNED COPPER	[illegible]			[illegible] COPPER	
FLUID — TYPE	CLEAR WATER		CLEAR WATER	CLEAR WATER			CLEAR WATER	
LOAD, TONS	•		[illegible]	•			•	
FLOW, GPM	[illegible]		[illegible]	[illegible]				
Δ P, PSI	[illegible]		[illegible]	[illegible]				
ENTG., °F	85		[illegible]	[illegible]			85	
LVG., °F	95		[illegible]	[illegible]			95	
WORKING, PSIG	[illegible]		150	150			[illegible]	
DESIGN, PSIG	[illegible]		150	150			[illegible]	
SHELL SIDE — DESIGN, PSIG	[illegible]		[illegible]	225			[illegible]	
RELIEF, PSIG			153	•			[illegible]	
REFRIGERANT TEMP., °F			[illegible]					

17FA

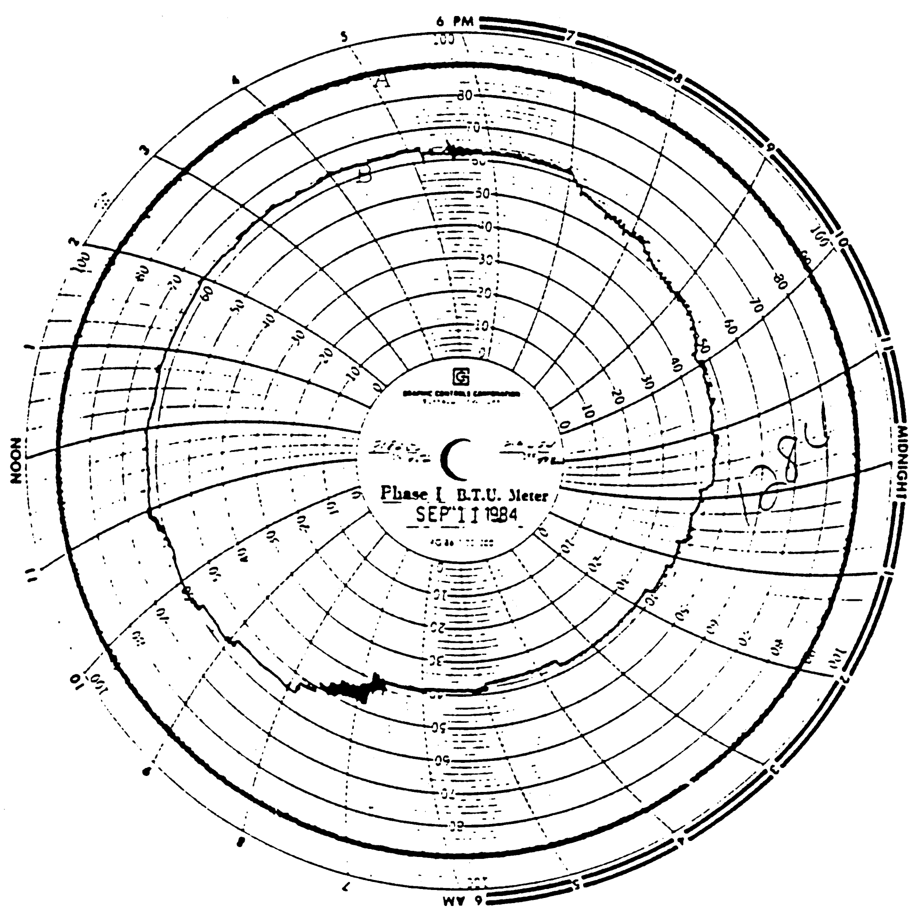
DESIGN DATA SHEET



Typical Performance at Constant Speed with Variable Inlet Guide Vanes (17CB, 17FA, 17MP40, 17MP400)

Btu Meter Monitoring Phase-I

A: Chilled Water Flow Rate Produced
B: Chilled Water Temperature Differential Produced

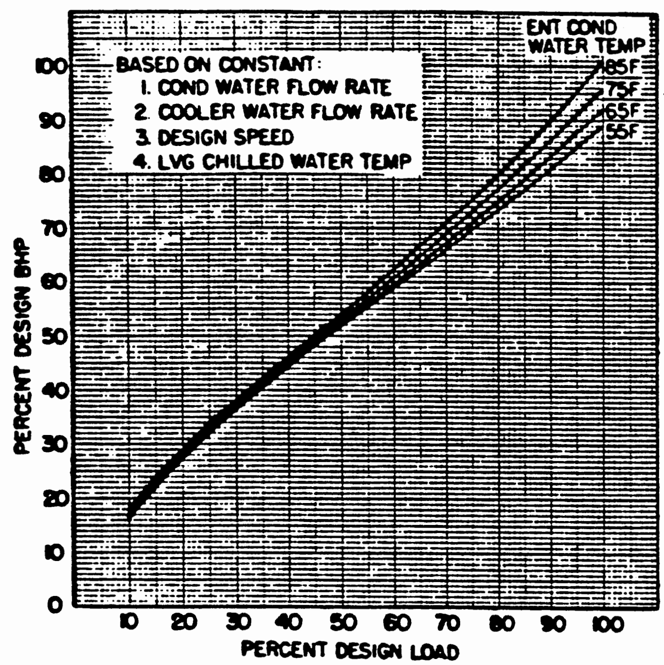


The 4200 ton
Electric
Chiller

17DA

JOB DESIGN DATA FOR REFRIGERATION MACHINE, SERIAL NO. 31470				
MACHINE OPERATING CHARGE [illegible] LB. (APPROX.) OF R-500				
COMPONENT:	ITEM	HORSEPOWER	RPM	BRAND, MODEL, REMARKS
COMPRESSOR	100	3400 BHP	3572	CARRIER 17DA[illegible]
DRIVER	800	4000	1200 FULL LOAD	IDEAL ELECTRIC
SPEED INCREASER	600	4000	3572 1200	LUFKIN [illegible]

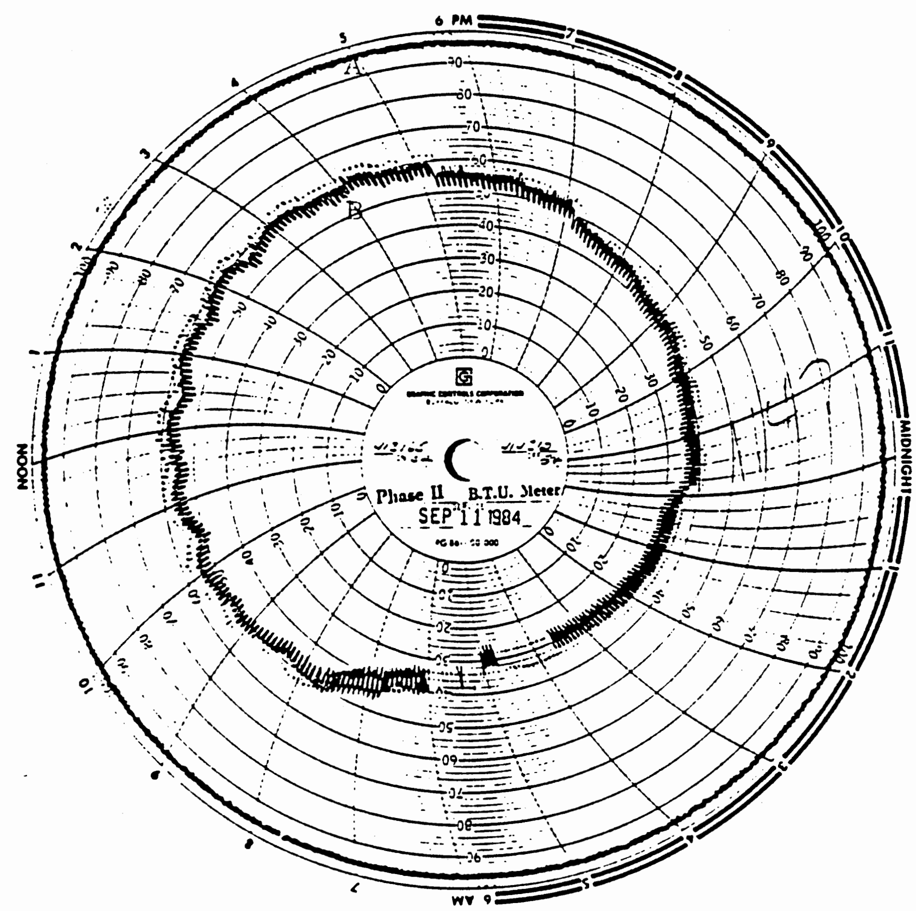
VESSELS		ITEM	104		200	300	400	500	703
		COMPONENT	COMP. OIL COOLER		COOLER	REFRIGERANT CONDENSER	ECONOMIZER	STORAGE TANK	PUMPOUT CONDENSER
		BRAND	BASCO		CARRIER	CARRIER		[illegible]	[illegible]
		MODEL	TYPE 500		17DA37	17DA37		400 CU. FT	[illegible]
TUBE SIDE	TUBES	DESIGN SCALE	.001		.0005	.001			.002
		NO. PASSES	4		2	2			4
		LENGTH, FT.	2		22	22			4.75
		QUANTITY	116		2346	4431			32
		GAUGE, BWG, & FIN.	22		20	20			19
		O.D., IN.	3/8		3/4	3/4			3/4
		SURFACE, INSIDE, SQ.FT.	16		17627	15587			20.5
		MATERIAL	ADMIRALTY		COPPER	COPPER			COPPER
	FLUID	TYPE	CLEAR WATER		CLEAR WATER	CLEAR WATER			CLEAR WATER
		LOAD, TONS	-		4200	-			33
		FLOW, GPM	15		6300	11961			60
		Δ P, PSI	1.5		10.2	6.6			6.5
		ENT., °F	[illegible]		56°	85°			[illegible]
		LVG., °F	95°		40°	95°			95
		OPERATING, PSIG.	150		150	150			150
		DESIGN, PSIG.	150		150	150			150
SHELL SIDE		DESIGN, PSIG.	300		150	185			225
		RELIEF, PSIG.	-		150	185			225
		REFRIGERANT TEMP., °F	-		37.1° SUCT	97.4 COND			-

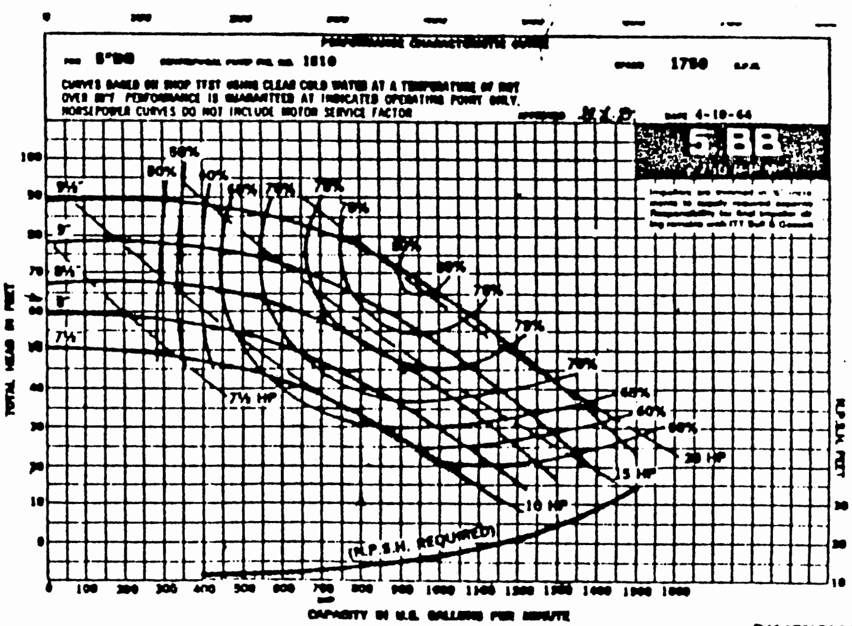


Typical Performance at Constant Speed with Variable Inlet Guide Vanes (17DA, 17EA)

Btu Meter Monitoring Phase-II

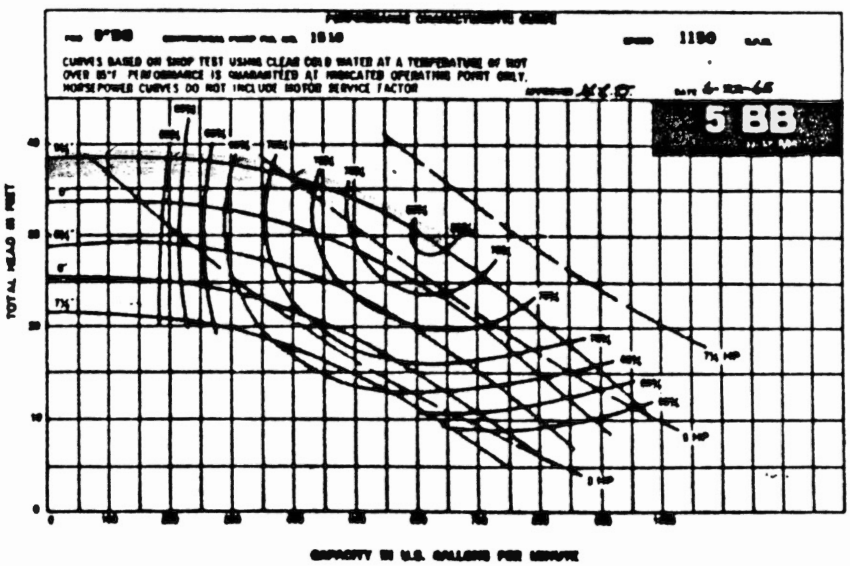
A: Chilled Water Flow Rate Produced
B: Chilled Water Temperature Differential Produced



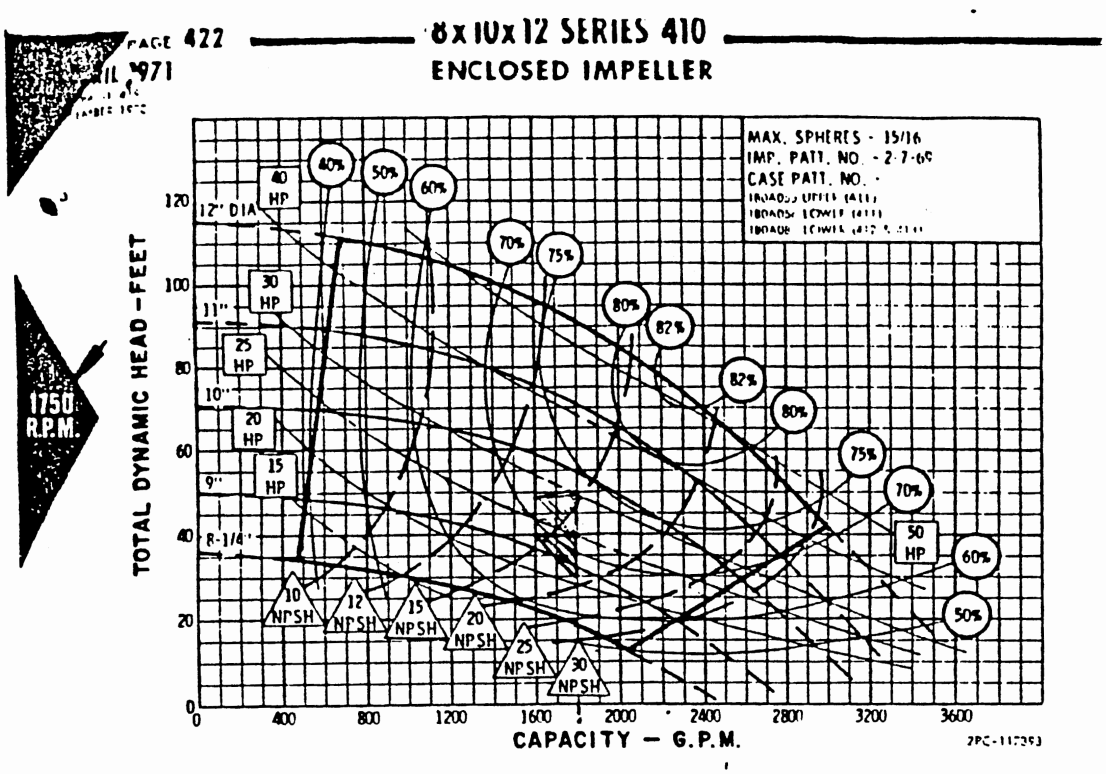


DIMENSIONS ON PAGE 20

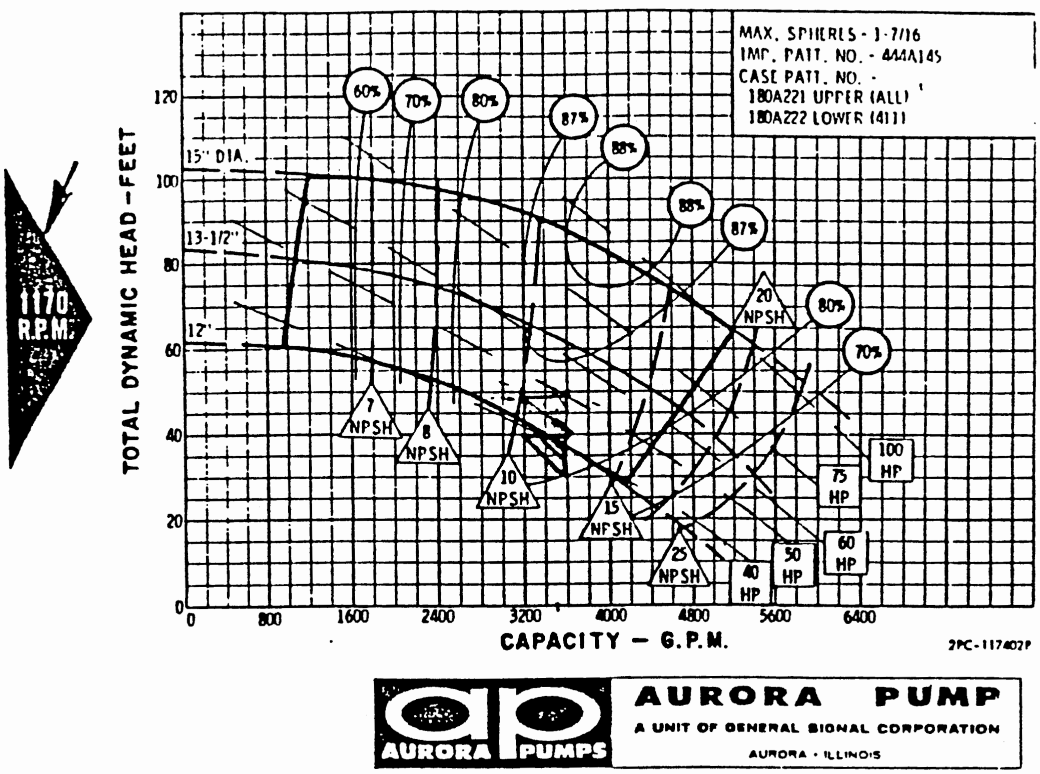
An Old Building Pump (Engineering North)



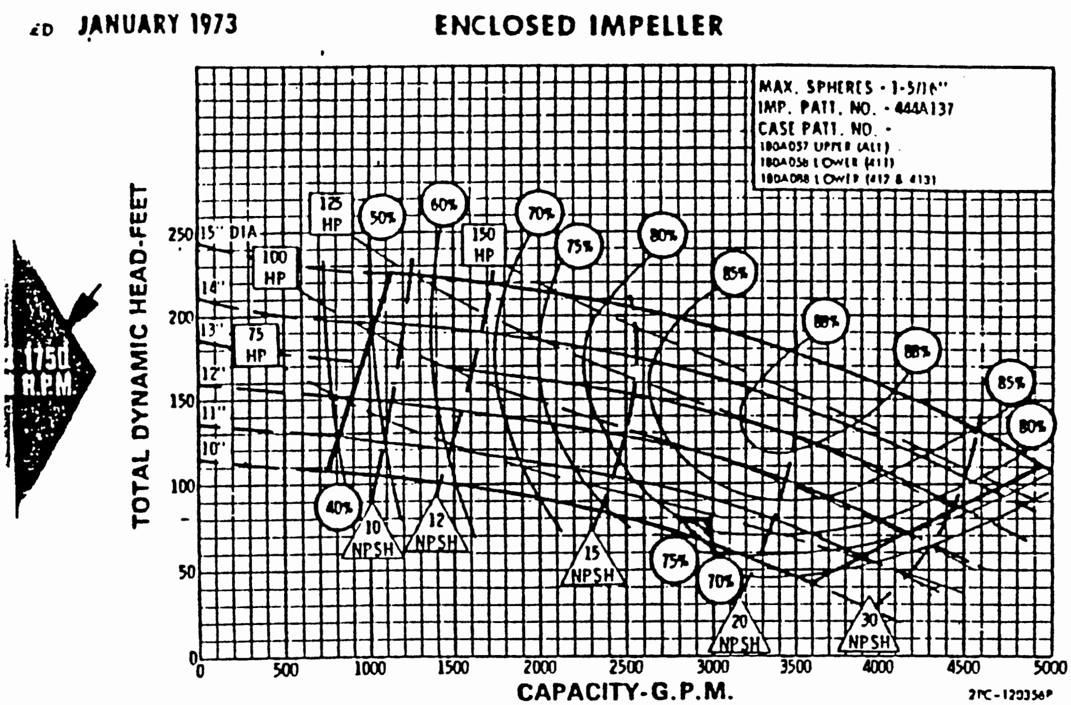
A New Building Pump (Engineering North)



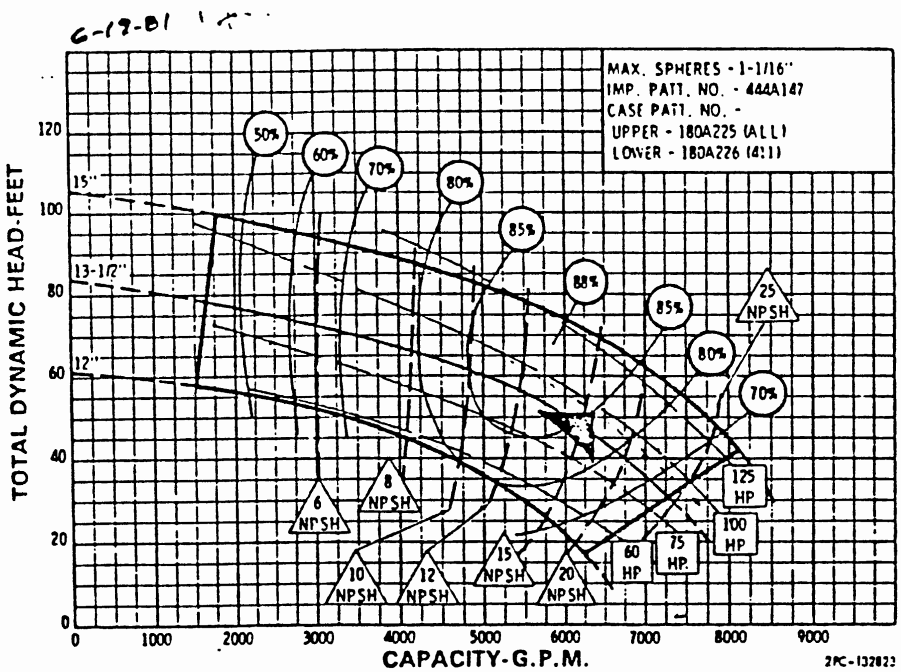
A Chiller Pump



A Chiller Pump

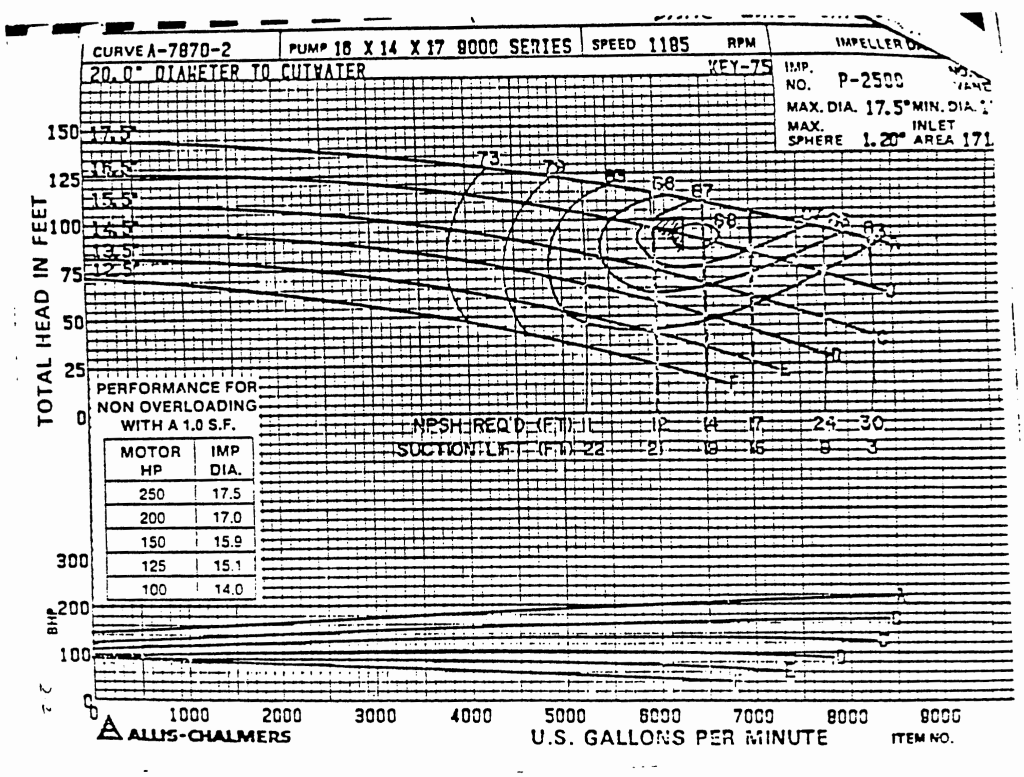


A Distribution Pump

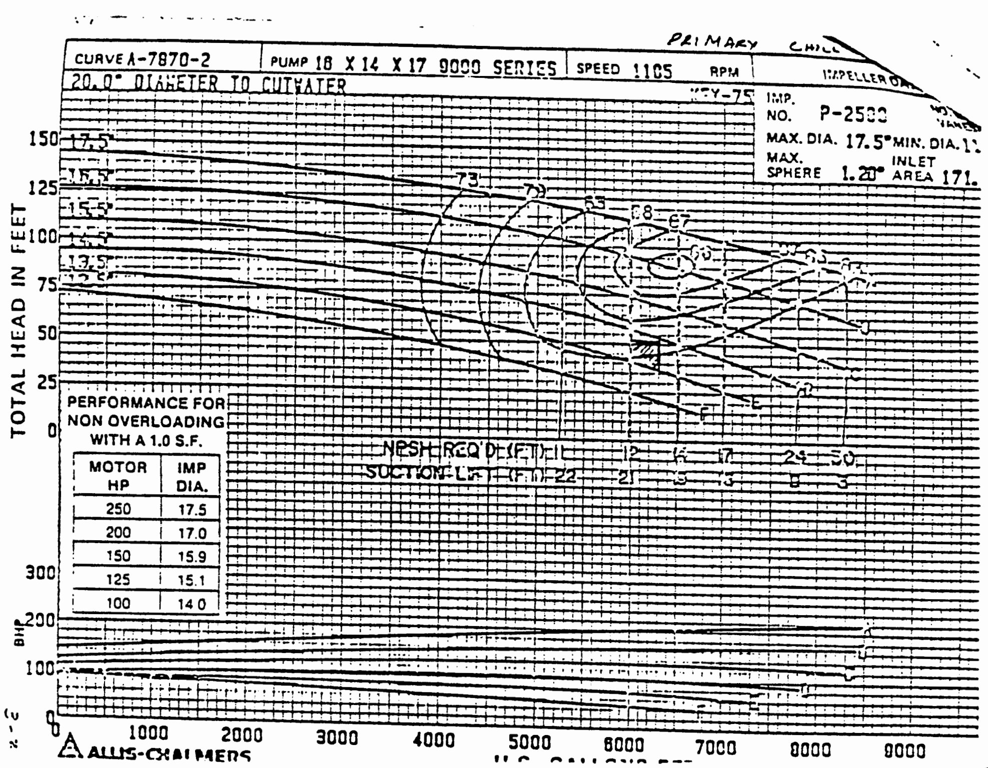


AURORA PUMP
A UNIT OF GENERAL SIGNAL CORPORATION
AURORA · ILLINOIS

A Distribution Pump



A Distribution Pump



MOTOR HP	IMP DIA.
250	17.5
200	17.0
150	15.9
125	15.1
100	14.0

A Chiller Pump

BUILDING		CHILLED WATER PUMP INFORMATION							MOTOR INFORMATION				
NAME	DWG. #	TONS	GPM	HD	MFG	SIZE	SER #	TYPE	RPM	HP	VOLTS	AMPS	MFG
Power Plant Ph 1	D-037-4	1200	3000	80	Aurora	8X10X15B		HSC	VAR	100	460	120	Linc
Power Plant Ph 1	D-037-4	2400	3000	80	Aurora	8X10X15B		HSC	1750	100	460	120	Linc
Power Plant Ph 2	D-037-4	4200	6300	95	Aurora	12X14X18			1750	200	460		US El
Power Plant Ph 3		4200	PROPOSED FOR THE FUTURE										
Library Chiller	D-040-5	1000	3000	80	Worth		A115537	8LR13	1750	75	440	91	Linc
Library Old & New	D-040-5	676	1811	80	Worth		A115538	8LR13	1750	75	440	91	Linc
Ag Hall	D-057-2	635	1750	75	Armstg	8X6	4103423	R	1770	50	208	127.9	US El
Animal Diag. Lab	D-110-2	200	350	86		4X3	EHJ25536		1755	15	200	47	
Animal Husbandry	M-62	55	150	80	Pace	2 1/2X2	2095-1	L	1750	7.5	200	24.1	Relia
Animal Science	D-049-7	98	150	NO PUMP									
Architecture	C-009-1	111	236	60	Pace	2 1/2X2	2095-1	L	1750	7.5	200	24.1	Relia
Bartlett Center	D-003-1	256	390	48	Pace		MUMR70221A		1750	7.5	208		
Bennett Chapel	C-010-1	25	50		Jacuzzi	1.25X1.25		1AM	3450	1.0	208	3.1	Smith
Business Bldg.	D-068-10	207	500	65	Worth	5X4		4LR10	1750	15	208	40	Linc
C.E.R.L.	D-77W-47	25	80	80	Smith	2X1 1/2		P	3450	2	230	5.4	
Civil Eng. Lab	D-026-1	75	140	70	Carver	2 1/2X2	55450BF		3475	5	208-		
Classroom Bldg.	C-044-2	295	800	35	B&G	6X5	927929	1510	1750	10	208	29	Gould
Colvin Center	D-092-5	55	240	50	Aurora	3X2 1/2X9	967-12538	GBPA-BE	1745	5	208	14.6	Marat
Cordell Hall (V.S.)		266	245	70	Aurora	3X2 1/2X9		344A-BF	VAR	7.5	230	18.2	Relia
Eng. North (V.S.)	D-059-4	315	800	45	B&G	6X5	927008-CW	1510	1750	15	208	44	Gould
Eng. South	D-027-3	288	850	60	B&G	6X5	1199250	1510	1750	20	208	56	Gould
Family/Child Science	D-030-4	90	272	70	Aurora	3X2 1/2X9		344	1750	7.5	208		
Gundersen Hall	D-006-9	125	390	66	Carver		67617	3J09	1755	10	220	26.6	Cent

BUILDING		CHILLED WATER PUMP INFORMATION							MOTOR INFORMATION				
NAME	DWG.#	TONS	GPM	HD	MFG	SIZE	SER #	TYPE	RPM	HP	VOLTS	AMPS	MFG
Home Ec. West	C-036-3	402	827	85	Pace	5X4	4011-7	KPS	1750	25	200	72.5	US El
Iba Hall	D-082-1	125	250	20	Carver	2 1/2X2X9Y			1750	10	208		
Journalism/Broad.	D-012-6	113	240	74	Pace	3X2 1/2	2595-1	L	1750	10	200		
Kerr/Drummond	D-066-1	857	1680	90	Worth	10 1/4D	A126858	6LR10	1800	50	208		
Life Science East	D-028-1	372	557	70	Pace		4011	KP	1750	15	208		
Life Science West	D-053-1	320	730	68	Pace	5X4	2AZE37247	4BB-KBS	1750	20	208	56	Marat
Math Science (S.C.)	C-089-1	302	800	90	Pace		2AZE23207	4BD-KPS	1765	[illegible]	208	82	Marat
Meat Lab	D-023-2	114	170										
Morrill Hall	D-004-4	130	320	35	Aurora	4X4X7B		344	1750	5	200	14.6	Auror
North Murray	D-025-2	142	136	75	B&G	3X2 1/2	697946BR	2 1/2	1750	7.5	208	25.4	US El
North Murray	D-025-2	142	126	75			697947BR						
Old Central	D-001-1	40	84	35	Pace	2 1/2X2	2070-5	L	1750	1.5	208	4.8	Cent
Parker Hall	C-055-1	100	300	56	Allis	3X2 1/2		KSH	1800	7.5	208	4.8	Cent
Physical Plant Scvs	C-079-11	25	40			2 1/2X2			3450	1.5	208	4.2	Howel
Physical Sciences	D-051-3	569	1400	40	B&G	8X6X6BB	845501	1510	1750	20	200	58	Linc
Poultry	D-045-3	60	120	51	Dunham	3X2	A9D-1		1725	3	208	8.8	Howel
Pub. Information	C-019-1	69	84	14	B&G	2AB-5-BF	769590	1510	1725	.75	200	2.9	Marat
Publishing/Printing	C-063-4	152	365	60	Pace	4X3	3095-1	L	1750	10	208	30.4	GE
Scott Hall	C-054-1	124	360	56	Allis	4X4		KSH	1800	7.5	208		
Seretean Center	D-002-2	320	763	47	Aurora	4X5X11A	69-10348	OJPA-BF	1735	15	208	44	Marat
SPW Cafeteria	D-056-2	90	240	35	Carver	3X2 1/2		2 1/2YC7	1740	3	208	9.8	Doerp
Student Health	C-111-1	121	213	105	Paco	3X2 1/2	LKN24797A	L	1750	10	230	27	Relia
Student Union	D-035-3	850	1023		Alfa	8X8X8X3	30100-65600	A20-BHAS	HEAT EXCHANGER IN SERVICE 8-84				

BUILDING		CHILLED WATER PUMP INFORMATION							MOTOR INFORMATION				
NAME	DWG.#	TONS	GPM	HD	MFG	SIZE	SER #	TYPE	RPM	HP	VOLTS	AMPS	MFG
Telecommunications	D-073-2	90	135	45		2 1/2X2			1750	3	208		
Telecommunications	D-073-2	90	50	55	B&G				1750		208		
Thatcher Hall	D-013-9	100	316	68	Paco	4X3	3095-1	L	1750	10	200		
Vet Med North (2)	D-039-14	200	552	78	Carver	4X3			1760	20	208	55	Westh
Vet Med South	D-039-14	300	738	75	Carver	5X4			1760	20	208	55	Westh
Vet Med South	D-039-14	300	720		B&G			B	1750	15	208	17	B&G
Vet Med Tch. Hosp.(2)	C-107-3	440	880	75	Paco	6X5	RB34420B	L	1750	25	230	63	Cent
Wentz Hall	C-060-1	245	550	75	Worth	5X4		4LR10	1750	15			
Whitehurst Hall	D-015-8	225	600	40	B&G	6X5	927-928	1510	1750	10	208	29	Gould
Willard Cafeteria	D-77W-30	25	100		Smith	2.5X1.5	224-2538B	1ZBCD	3640	3	220	8	
Willham	D-084-1	970	1680	90	Aurora	8X8X11B	967-9456	OJPA-BF	1750	50	208		

APPENDIX E

COMPUTER PROGRAM LISTING

```
      PROGRAM EN
C
      DIMENSION EL(35),ED(35),DP(35),FR(35)
      DIMENSION CDF(9),CDP(9),CAF(9),CAP(9)
      DIMENSION VAP(9),CVA(9)
      DIMENSION CCL(9)
C
      DATA DTS,DTP/12.,16./
      DATA P0,P1,P2,P3/27.68,-5.405E-3,2.002E-5,-7.003E-8/
C
      OPEN(UNIT=55,NAME='EN.IN',STATUS='UNKNOWN')
      OPEN(UNIT=56,NAME='EN.OT',STATUS='UNKNOWN')
C
C   READ PIPING DATA
C
       DO 10 IS=1,35
      READ(55,100)EL(IS),ED(IS)
 10    CONTINUE
C
C   READ COIL DATA
C
       DO 20 IC=1,9
      READ(55,100)CDF(IC),CDP(IC)
 20    CONTINUE
C
C   READ COOLING LOAD (TON)
C
      DO 30 IL=1,9
      READ(55,100)CCL(IL)
 30    CONTINUE
C
C   CALCULATE THE NEEDED FLOW RATE, GPM
C
      DO 40 IG=1,9
        CAF(IG)=12000.*CCL(IG)/500./DTS
 40     CONTINUE
C
C   ESTIMATE SECTION FLOW RATES, GPM
C
      WS=0.0
      DO 45 ID=1,9
       WS=WS+CAF(ID)
 45     CONTINUE
C
      FR(1)=WS
      FR(2)=CAF(1)+CAF(2)+CAF(3)+CAF(4)
      FR(3)=CAF(1)
      FR(4)=FR(3)
      FR(5)=CAF(1)+CAF(2)
      FR(6)=CAF(1)+CAF(2)+CAF(3)
      FR(7)=FR(2)
      FR(8)=FR(1)
      WP=WS*DTS/DTP
      FR(9)=WS-WP
      FR(10)=FR(1)
      FR(11)=CAF(2)+CAF(3)+CAF(4)
      FR(12)=CAF(2)
      FR(13)=CAF(3)+CAF(4)
      FR(14)=CAF(3)
      FR(15)=CAF(4)
```

```
      FR(16)=FR(15)
      FR(17)=CAF(5)+CAF(6)+CAF(7)+CAF(8)+CAF(9)
      FR(18)=CAF(5)+CAF(6)+CAF(7)
      FR(19)=CAF(5)
      FR(20)=FR(19)
      FR(21)=CAF(5)+CAF(6)
      FR(22)=FR(18)
      FR(23)=FR(17)
      FR(24)=CAF(6)+CAF(7)
      FR(25)=CAF(6)
      FR(26)=CAF(7)
      FR(27)=FR(26)
      FR(28)=CAF(8)+CAF(9)
      FR(29)=FR(28)
      FR(30)=CAF(8)
      FR(31)=FR(30)
      FR(32)=FR(29)
      FR(33)=FR(28)
      FR(34)=CAF(9)
      FR(35)=FR(34)
C
C  ESTIMATE SECTION PRESSURE LOSS, PSI
C
      DO 60 IS=1,35
       DP(IS)=0.0311*0.02*EL(IS)/(ED(IS)**5)*FR(IS)**2/2.31
 60      CONTINUE
C
C  ESTIMATE VARIABLE COIL PRESSURE DROPS
C
      DO 70 IV=1,9
       CAP(IV)=CDP(IV)*(CAF(IV)/CDF(IV))**2
 70      CONTINUE
C
C   PUMP PRESSURE RISE
C
      PDP=P0+P1*WS+P2*WS**2+P3*WS**3
C
C   CALCULATE THE VARIABLE PRESSURE ACROSS VALVES
C
      VAP(1)=PDP-(DP(1)+DP(2)+CAP(1)+DP(3)+DP(4)+
     &      DP(5)+DP(6)+DP(7)+DP(8)+DP(9)+DP(10))
      VAP(2)=PDP-(DP(1)+DP(2)+DP(11)+CAP(2)+DP(12)+
     &      DP(5)+DP(6)+DP(7)+DP(8)+DP(9)+DP(10))
      VAP(3)=PDP-(DP(1)+DP(2)+DP(11)+DP(13)+CAP(3)+DP(14)+
     &      DP(6)+DP(7)+DP(8)+DP(9)+DP(10))
      VAP(4)=PDP-(DP(1)+DP(2)+DP(11)+DP(13)+DP(15)+CAF(4)+
     &            DP(7)+DP(8)+DP(9)+DP(10))
C
      VAP(5)=PDP-(DP(1)+DP(17)+DP(18)+CAP(5)+DP(19)+DP(20)+
     &       DP(21)+DP(22)+DP(23)+DP(8)+DP(9)+DP(10))
      VAP(6)=PDP-(DP(1)+DP(17)+DP(24)+CAP(6)+DP(25)+
     &       DP(21)+DP(22)+DP(23)+DP(8)+DP(9)+DP(10))
      VAP(7)=PDP-(DP(1)+DP(17)+DP(24)+DP(26)+CAP(7)+DP(27)+
     &              DP(22)+DP(23)+DP(8)+DP(9)+DP(10))
C
      VAP(8)=PDP-(DP(1)+DP(17)+DP(28)+DP(29)+CAP(8)+DP(30)+
     &       DP(31)+DP(32)+DP(33)+DP(23)+DP(8)+DP(9)+DP(10))
      VAP(9)=PDP-(DP(1)+DP(17)+DP(28)+DP(29)+DP(34)+CAP(9)+
     &       DP(35)+DP(32)+DP(33)+DP(23)+DP(8)+DP(9)+DP(10))
C
      DO 80 IP=1,9
       PRINT*,IP,CCL(IP)
 80    CONTINUE
```

```
      PRINT*,WS,' SECONDARY GPM'
      PRINT*,WP,' PRIMARY GPM'
      PRINT*,WS-WP,' BY-PASS GPM'
      DO 90 IS=1,35
       PRINT*,IS,DP(IS)
 90    CONTINUE
C
       DO 95 IV=1,9
      PRINT*,IV,VAP(IV),CAF(IV),CAP(IV)
      PRINT*,CAF(IV)/SQRT(VAP(IV)), 'Cv'
 95   CONTINUE
C
C
 100  FORMAT(2F10.3)
C
      WRITE(56,*)' '
      WRITE(56,*)' *** BUILDING SIMULATION RESULTS ***'
      WRITE(56,*)' '
      WRITE(56,*)' '
      WRITE(56,*)' '
      WRITE(56,*)' - COIL THERMAL LOADS - '
      WRITE(56,*)' '
      WRITE(56,*)' COIL I.D.    COOLING LOAD, TON '
      WRITE(56,*)' '
       DO 300 II=1,9
        WRITE(56,*) II, CCL(II)
 300  CONTINUE
C
      WRITE(56,*)' '
      WRITE(56,*)' '
      WRITE(56,*)' - VALVE INFORMATION - '
      WRITE(56,*)' '
      WRITE(56,*)'  VALVE I.D.    FLOW,GPM    HEAD,PSI       Cv     '
      WRITE(56,*)' '
       DO 310 IJ=1,9
      VC=CAF(IJ)/SQRT(VAP(IJ))
      WRITE(56,*)IJ,CAF(IJ),VAP(IJ),VC
 310  CONTINUE
      WRITE(56,*)' '
      WRITE(56,*)' '
      WRITE(56,*)' - FLOW INFORMATION -'
      WRITE(56,*)' '
      WRITE(56,*)' PRIMARY,         BY-PASS,    SECONDARY FLOW,GPM '
      WRITE(56,*)' '
      WRITE(56,*)WP,WS-WP,WS
      STOP
      END
```

100.0	5.
150.	4.
150.	3.
150.	3.
150.	4.
200.	4.
200.	4.
100.	5.
1.5	5.
3.	5.
150.	3.
150.	4.
150.	3.
150.	4.
150.	2.
150.	2.
800.	5.
250.	4.
150.	2.5
150.	2.5
150.	3.
200.	4.
300.	5.
150.	4.
150.	2.5
150.	3.
150.	3.
250.	3.
1.	3.
150.	2.5
150.	2.5
150.	3.
100.	3.
150.	2.5
150.	2.5
92.	5.97
108.	3.46
55.	2.38
59.	2.64
56.	4.03
102.	2.38
105.	4.98
61.	4.98
81.	5.93
5.	
5.	
5.	
5.	
5.	
5.	
5.	
5.	
5.	

```
      DIMENSION EL(26),ED(26),DP(26),FR(26)
      DIMENSION BDP(13),BFR(13)
      DIMENSION PC(5,4)
C
      DATA PC(1,1),PC(1,2),PC(1,3),PC(1,4) /54.998,
     & -2.066E-3,1.599E-7,-5.332E-12/
      DATA PC(2,1),PC(2,2),PC(2,3),PC(2,4) /59.999,
     & -1.216E-3,2.997E-8,-1.333E-12/
      DATA PC(3,1),PC(3,2),PC(3,3),PC(3,4) /54.994,
     & -1.831E-3,5.992E-8,-2.662E-12/
      DATA PC(4,1),PC(4,2),PC(4,3),PC(4,4) /49.003,
     & -3.012E-4,-1.099E-7,1.997E-12/
      DATA PC(5,1),PC(5,2),PC(5,3),PC(5,4) /52.865,
     & -1.917E-3,-5.375E-9,-1.154E-12/
C
      OPEN(UNIT=35,NAME='OSU.IN',STATUS='UNKNOWN')
      OPEN(UNIT=36,NAME='OSU.OT',STATUS='UNKNOWN')
C
      IPUMP=3
C
      DO 10 IS=1,26
      READ(35,100)EL(IS),ED(IS)
 10   CONTINUE
C
      DO 20 IB=1,13
      READ(35,100)BFR(IB)
 20   CONTINUE
C
      SUM=0.0
       DO 30 IB=1,13
       SUM=SUM+BFR(IB)
 30   CONTINUE
C
      FR(1)=SUM
      FR(2)=SUM-BFR(1)
      FR(3)=SUM-BFR(1)-BFR(2)
      FR(4)=SUM-BFR(1)-BFR(2)-BFR(3)
      FR(5)=SUM-BFR(1)-BFR(2)-BFR(3)-BFR(4)
      FR(6)=SUM-BFR(1)-BFR(2)-BFR(3)-BFR(4)-BFR(5)
      FR(7)=SUM-BFR(1)-BFR(2)-BFR(3)-BFR(4)-BFR(5)-BFR(6)
      FR(7)=BFR(7)+BFR(8)+BFR(9)+BFR(10)+BFR(11)+BFR(12)+BFR(13)
      FR(8)=BFR(13)+BFR(12)+BFR(11)+BFR(10)+BFR(9)+BFR(8)
      FR(9)=BFR(13)+BFR(12)+BFR(11)+BFR(10)+BFR(9)
      FR(10)=BFR(13)+BFR(12)+BFR(11)+BFR(10)
      FR(11)=BFR(13)+BFR(12)+BFR(11)
      FR(12)=BFR(13)+BFR(12)
      FR(13)=BFR(13)
C
      FR(14)=FR(13)
      FR(15)=FR(12)
      FR(16)=FR(11)
      FR(17)=FR(10)
      FR(18)=FR( 9)
      FR(19)=FR( 8)
      FR(20)=FR( 7)
      FR(21)=FR( 6)
      FR(22)=FR( 5)
      FR(23)=FR( 4)
      FR(24)=FR( 3)
      FR(25)=FR( 2)
      FR(26)=FR( 1)
```

```
-
      DO 40 IS=1,26
      DP(IS)=0.0311*0.02*EL(IS)/(ED(IS)**5)*FR(IS)**2/2.31
 40   CONTINUE
C
      PDP=PC(IPUMP,1)+PC(IPUMP,2)*FR(1)+PC(IPUMP,3)*FR(1)**2+
     &  PC(IPUMP,4)*FR(1)**3
C
      BDP(1)=PDP-(DP(1)+DP(26))
      BDP(2)=PDP-(DP(1)+DP(2)+DP(25)+DP(26))
      BDP(3)=PDP-(DP(1)+DP(2)+DP(3)+DP(24)+DP(25)+DP(26))
      BDP(4)=PDP-(DP(1)+DP(2)+DP(3)+DP(4)+DP(23)+DP(24)+DP(25)+DP(26))
 BDP(5)=P
      BDP(5)=PDP-(DP(1)+DP(2)+DP(3)+DP(4)+DP(5)+
     &         DP(22)+DP(23)+DP(24)+DP(25)+DP(26))
      BDP(6)=PDP-(DP(1)+DP(2)+DP(3)+DP(4)+DP(5)+DP(6)+
     &         DP(21)+DP(22)+DP(23)+DP(24)+DP(25)+DP(26))
      BDP(7)=PDP-(DP(1)+DP(2)+DP(3)+DP(4)+DP(5)+DP(6)+DP(7)+
     &        DP(20)+DP(21)+DP(22)+DP(23)+DP(24)+DP(25)+DP(26))
      BDP(8)=PDP-(DP(1)+DP(2)+DP(3)+DP(4)+DP(5)+DP(6)+DP(7)+DP(8)+
     &   DP(19)+DP(20)+DP(21)+DP(22)+DP(23)+DP(24)+DP(25)+DP(26))
      PRE= DP(1)+DP(2)+DP(3)+DP(4)+DP(5)+DP(6)+DP(7)+DP(8)+DP(9)+
     *  DP(10)+DP(11)+DP(12)+DP(13)
      POS= DP(14)+DP(15)+DP(16)+DP(17)+DP(18)+DP(19)+DP(20)+
     *  DP(21)+DP(22)+DP(23)+DP(24)+DP(25)+DP(26)
      BDP(9)=PDP-(PRE-DP(10)-DP(11)-DP(12)-DP(13)+
     &   POS-DP(14)-DP(15)-DP(16)-DP(17))
      BDP(10)=PDP-(PRE-DP(11)-DP(12)-DP(13)+POS-DP(14)-DP(15)-DP(16))
      BDP(11)=PDP-(PRE-DP(12)-DP(13)+POS-DP(14)-DP(15))
      BDP(12)=PDP-(PRE-DP(13)+POS-DP(14))
      BDP(13)=PDP-(PRE+POS)
C
 100   FORMAT(2F10.3)
C
      PRINT*,' SECTION(12),PSI ',DP(12)
C
      DO 300 IS=1,13
       PRINT*,IS,BFR(IS),'GPM'
 300  CONTINUE
C
      DO 310 IB=1,13
      PRINT*,IB,BDP(IB),'PSI'
 310  CONTINUE
C
      WRITE(36,*)' '
      WRITE(36,*)' *** CAMPUS SIMULATION RESULTS *** '
      WRITE(36,*)' '
      WRITE(36,*)' '
      WRITE(36,*)' '
      WRITE(36,*)' - BRANCH DEMANDS, GPM - '
      WRITE(36,*)' '
      WRITE(36,*)'  BRANCH I.D.   DEMANDS, GPM '
      WRITE(36,*)' '
       DO 500 II=1,13
       WRITE(36,*)II,BFR(II)
 500  CONTINUE
      WRITE(36,*)' '
      WRITE(36,*)' TOTAL DEMAND ',FR(1)
C
      WRITE(36,*)' '
      WRITE(36,*)' '
      WRITE(36,*)' - DISTRIBUTION PUMP MODE - '
      WRITE(36,*)' '
```

```
      WRITE(36,*)' MODE IS ',IPUMP
      WRITE(36,*)' '
      WRITE(36,*)' '
      WRITE(36,*)' - BRANCH HEAD, PSI    &  COEFFICIENT, -'
      WRITE(36,*)' '
      DO 510 IJ=1,13
       WRITE(36,*)BDP(IJ),BFR(IJ)/SQRT(BFR(IJ))
 510  CONTINUE
C
      STOP
      END
```

```
   50.      30.
  600.      30.
  650.      19.
  360.      18.
  300.      16.
  200.      14.
  200.      14.
  100.      14.
 1450.      14.
  250.      14.
  150.      14.
  705.       6.
  800.       3.
  790.       3.
  690.       6.
  150.      14.
  250.      14.
 1450.      14.
  100.      14.
  200.      14.
  200.      14.
  300.      16.
  360.      18.
  650.      19.
  600.      30.
   80.      30.
  450.
 5000.
  700.
  875.
  110.
   30.
  500.
  850.
  850.
  150.
 1445.
  175.
   20.
```

```
$FOR BSIM1
$LINK BSIM1,DATA,INIT,RESD,PDRV,CNTL,GAUSSY,TEMP,TSET,RESULT,OTHR
$RUN BSIM1


        PROGRAM BSIM
C  THIS PROGRAM IS TO CALCULATE FLOW, PRESSURE, AND TEMPERATURE AT
C  VARIOUS LOCATIONS IN A BUILDING LOOP WHICH HAS TEMPERATURE
C  CONTROL VALVES AND VARIABLE SPEED PUMPS.
C
        INCLUDE 'BSIM1.CMN'
C
        OPEN(UNIT=5,NAME='BSIM1.IN',STATUS='UNKNOWN')
        OPEN(UNIT=6,NAME='BSIM1.OUT',STATUS='UNKNOWN')
        OPEN(UNIT=10,NAME='BSIM1.CHK',STATUS='UNKNOWN')
C
C   CALL'DATA' TO OBTAIN  EQUIVALENT LENGTH AND DIAMETER
C
             CALL DATA
C
C   CALL 'INIT' TO INITIALIZE ALL THE VARIABLES
C
             CALL INIT
C
C   CALL 'RESD' TO CALCULATE RESIDUALS OF ALL FUNCTIONS
C
             CALL RESD
C
         ITHMAL  =  1
C
 300     CONTINUE
C
C   ***  START ITERATIONS
C
          DO 100 ITER=1,ITERMX
C
C              WRITE(10,*)ITER,' ITERATION '
C              WRITE(10,*)(V(IVAR),IVAR=1,IVARMX),' VARIABLE VALUES'
C              WRITE(10,*)(R(IVAR),IVAR=I,IVARMX),' RESIDUAL VALUES'
C
C            ASSIGN VARIABLE VALUES
C
            DO 105 IVAR=1,IVARMX
              VOLD(IVAR)=V(IVAR)
 105        CONTINUE
C
C           CALL 'PDRV' TO CALCULATE THE PARTIAL DERIVATIVES.
C
              CALL PDRV
C
C           CALL 'GAUSSY' TO ESTIMATE THE CORRECTION VALUES
C
                   N=IVARMX
C
              CALL GAUSSY(PD,R,X,N)
C
C             WRITE(10,*)(X(IVAR),IVAR=1,IVARMX),' CORRECTED  VALUES'
C
C           CACULATE NEW VALUES FOR THE VARIABLES
```

```
C
                  DO 150 IVAR=1,IVARMX
                   VCORR  = X(IVAR)
                   V(IVAR)=VOLD(IVAR)-VCORR
 150              CONTINUE
C
C           CALL RESIDUAL WITH NEW VARIABLE VALUES
C
                   CALL RESD
C
C  CHECK THE CONVERGENCY BASED ON RESIDUAL VALUES
C
                      DO 200 IVAR=1,IVARMX
 200                  IF(ABS(R(IVAR)).GT. EPSL) GOTO 100
C
C           WAU!!!  CNVG ACHIVED!
C
C              WRITE(10,*)' CONVERGENCY ACHIEVED AT ',ITER,'   ITERATIONS'
C              WRITE(10,*)' LET''S DO THERMAL ANALYSIS '
C              WRITE(10,*)' THERMAL ANALYSIS COUNTER ',ITHMAL
               GOTO 500
C
 100        CONTINUE
C
C     ***   HYDRAULIC ITERATION END
C
C
 500           CONTINUE
C
C       CHECK THE TEMPERATURES AT SENSING POINTS
C
                 CALL TEMP
C
        WRITE(10,*)(V(IV),IV=1,IVARMX),'    FINAL VALUES'
        WRITE(10,*)T2,' T2',T5,' T5',T12,' T12'
C
C       DECIDE VALVE ADJUSTMENTS OR NOT
C
        IF((T5.GE.55.).AND.(T5.LE.57.).AND.
     &     (T12.GE.55.).AND.(T12.LE.57.).AND.
     &     (T2.GE.43.).AND.(T2.LE.45.))THEN
C
           CALL INPUT
C
           CALL OUTPUT
C
           GOTO 999
C
         ELSE
C
               CALL CNTL
C
           ITHMAL=ITHMAL+1
           IF(ITHMAL.LE.40) GOTO 300
C
         END IF
C
C   *** THERMAL ITERATION END
C
 999    CONTINUE
C
        STOP
        END
```

```
      SUBROUTINE DATA
C   THIS ROUTINE IS TO PROVIDE THE DATA FOR 'BSIM.FOR'
C
      INCLUDE 'BSIM1.CMN'
C
      ITERMX = 20
      IVARMX = 18
      ISECTN = 13
      NCOIL  = 2
      EPSL   = 0.1

C
      DO 10 IIS=1,ISECTN
        READ(5,100)EL(IIS),ED(IIS)
 10    CONTINUE
C
       CCV2(1) =200.
       CCV2(2) =150.
C
       DCV2(1) =100.
C
       PERCNT(1)= 0.5
       PERCNT(2)= 0.4
C
       DO 20 IIC=1,NCOIL
         READ(5,100)DLOAD(IIC)
                   HLOAD(IIC)=PERCNT(IIC)*DLOAD(IIC)
 20    CONTINUE
C
       DELPSR=10.0
       TEMPSP=40.0

C
100    FORMAT(F10.2,F10.2)
C
      RETURN
      END


    50.0        4.0
     0.0        0.0
     0.0        0.0
   150.0        2.0
    50.0        3.0
    50.0        3.0
     3.0        2.0
     0.0        0.0
    50.0        3.0
     0.0        0.0
   150.0        2.0
    50.0        3.0
    50.0        3.0
  60.0
  60.0
```

```
      SUBROUTINE INIT
C   THIS ROUTINE IS TO INITIALIZE THE VARIABLES
      INCLUDE 'BSIM1.CMN'
C
      V( 1) = 1.0
      V( 2) = 12.
      V( 3) =  5.0
      V( 4) =  4.5
      V( 5) =  0.5
      V( 6) =  1.0
      V( 7) =  2.0
      V( 8) = 11.
      V( 9) =  1.0
      V(10) =  5.0
      V(11) =  4.5
      V(12) =  0.5
      V(13) =  1.0
C
      V(14) = 200.
      V(15) = 300.
      V(16) = 150.
      V(17) = 150.
      V(18) = 100.
C
C     WRITE(10,*)(V(IVAR),IVAR=1,IVARMX),' INITIAL VARIABLES'
C
      RETURN
      END
```

```
      DIMENSION A(18,18),R(18),V(18),VOLD(18),X(18),PD(18,18)
      DIMENSION EL(13),ED(13)
      DIMENSION CCV2(2)
      DIMENSION DCV2(1)
      DIMENSION HLOAD(2),DLOAD(2),PERCNT(2)
C
      COMMON  /VAR1/ A,R,V,VOLD,X,PD
      COMMON  /VAR2/ EL,ED
      COMMON  /VAR3/ CCV2
      COMMON  /VAR4/ DCV2
      COMMON  /VAR5/ HLOAD,DLOAD,PERCNT
      COMMON  /VAR6/ ITERMX,IVARMX,ISECTN,NCOIL,EPSL
      COMMON  /VAR7/ DELPSR,TEMPSP
      COMMON  /VAR8/ T2,T6,T5,T12
```

```
      SUBROUTINE RESD
C   THIS ROUTINE IS TO CA1CULATE THE RESIDUALS OF FUNCTIONS
      INCLUDE 'BSIM1.CMN'
C
      R(1) = V(1)-0.0311*0.02*EL(1)/(ED(1)**5)*V(14)**2/2.31
      VPUMP=20.0
      R(2) = V(2)-(V(3)+V(4)+V(5)+V(6)+V(7))
      R(3) = V(3)-(V(16)/CCV2(1))**2
      R(4) = V(4)-0.0311*0.02*EL(4)/(ED(4)**5)*V(16)**2/2.31
      R(5) = V(5)-0.0311*0.02*EL(5)/(ED(5)**5)*V(16)**2/2.31
      R(6) = V(6)-0.0311*0.02*EL(6)/(ED(6)**5)*V(15)**2/2.31
      R(7) = V(7)-0.0311*0.02*EL(7)/(ED(7)**5)*V(18)**2/2.31
      R(8) = V(8)-(V(14)/DCV2(1))**2
      R(9) = V(9)-0.0311*0.02*EL(9)/(ED(9)**5)*V(17)**2/2.31
      R(10)= V(10)-(V(17)/CCV2(2))**2
      R(11)= V(11)-0.0311*0.02*EL(11)/(ED(11)**5)*V(17)**2/2.31
      R(12)= V(12)-0.0311*0.02*EL(12)/(ED(12)**5)*V(17)**2/2.31
      R(13)= V(13)-0.0311*0.02*EL(13)/(ED(13)**5)*V(17)**2/2.31
C
      R(14)=V(14)+V(18)-V(15)
      R(15)=V(16)+V(17)-V(15)
C
      R(16)=V(10)+V(11)+V(12)-VPUMP
      R(17)=V(9)+V(10)+V(11)+V(12)+V(13)-V(3)-V(4)-V(5)
      R(18)=V(1)-V(2)+V(3)+V(4)+V(5)+V(6)+V(8)-DELPSR
C
C     WRITE(10,*)(R(IVAR),IVAR=1,IVARMX),' RESIDUAL VALUES '
C
      RETURN
      END


      SUBROUTINE PDRV
C   THIS ROUTINE IS TO EVALUATE THE PARTIAL DERIVATIVE MATRIX
C
      INCLUDE 'BSIM1.CMN'
C
C     PUT ALL PD ZERO ,FIRST
C
      DO 10 II=1,IVARMX
       DO 10 JJ=1,IVARMX
        PD(II,JJ)=0.0
 10   CONTINUE
C
      PD(1,1)  =  1.0
      PD(1,14) = -2.*0.0311*0.02*EL(1)/(ED(1)**5)*V(14)/2.31
C
      PD(2,2)  =  1.0
      PD(2,3)  = -1.0
      PD(2,4)  = -1.0
      PD(2,5)  = -1.0
      PD(2,6)  = -1.0
      PD(2,7)  = -1.0
C
      PD(3,3)  =  1.0
      PD(3,16) = -2.*(V(16)/CCV2(1))*(1./CCV2(1))
C
      PD(4,4)  =  1.0
       CONST   = -2.*0.0311*0.02/2.31
      PD(4,16) =  CONST*EL(4)/(ED(4)**5)*V(16)
```

```
C
        PD(5,5)  =  1.0
        PD(5,16) =  CONST*EL(5)/(ED(5)**5)*V(16)
C
        PD(6,6)  =  1.0
        PD(6,15) =  CONST*EL(6)/(ED(6)**5)*V(15)
C
        PD(7,7)  =  1.0
        PD(7,18) =  CONST*EL(7)/(ED(7)**5)*V(18)
C
        PD(8,8)  =  1.0
        PD(8,14) = -2.*(V(14)/DCV2(1))*(1./DCV2(1))
C
        PD(9,9)  =  1.0
        PD(9,17) =  CONST*EL(9)/(ED(9)**5)*V(17)
C
        PD(10,10)=  1.0
        PD(10,17)= -2.*(V(17)/CCV2(2))*(1./CCV2(2))
C
        PD(11,11)=  1.0
        PD(11,17)=  CONST*EL(11)/(ED(11)**5)*V(17)
C
        PD(12,12)=  1.0
        PD(12,17)=  CONST*EL(12)/(ED(12)**5)*V(17)
C
        PD(13,13)=  1.0
        PD(13,17)=  CONST*EL(13)/(ED(13)**5)*V(17)
C
        PD(14,14)=  1.0
        PD(14,18)=  1.0
        PD(14,15)= -1.0
C
        PD(15,15)= -1.0
        PD(15,16)=  1.0
        PD(15,17)=  1.0
C
        PD(16,10) =  1.0
        PD(16,11) =  1.0
        PD(16,12) =  1.0
C
        PD(17,9) = 1.0
        PD(17,10) = 1.0
        PD(17,11) = 1.0
        PD(17,12) = 1.0
        PD(17,13) = 1.0
        PD(17,3) =-1.0
        PD(17,4) =-1.0
        PD(17,5) =-1.0
C
        PD(18,1) = 1.0
        PD(18,2) =-1.0
        PD(18,3) = 1.0
        PD(18,4) = 1.0
        PD(18,5) = 1.0
        PD(18,6) = 1.0
        PD(18,8) = 1.0
C
C        DO 30 IX=1,IVARMX
C          DO 30 IY =1,IVARMX
C            IF(PD(IX,IY).EQ.0.0) GOTO 30
C            WRITE(10,*)PD(IX,IY),'PARTIALS',IX,' IX',IY,' IY'
C 30       CONTINUE
C
        RETURN
        END
```

```
      SUBROUTINE CNTL
C   THIS ROUTINE IS TO CONTROL THE VALVES AND PUMPS BASED ON
C   TEMPERATURES AT THE STRATEGIC LOCATIONS
C
      INCLUDE 'BSIM1.CMN'
      REAL INC
C
      DATA STSET,RTSET/44.0, 56.0/
      DATA INC,DEC/1.2,0.8/
C
C 55   CONTINUE
C
      IF((T5.GE.55.).AND.(T5.LE.57.))GOTO 61
      IF(T5.LE.RTSET)THEN
        CCV2(1)=DEC*CCV2(1)
      ELSE
        CCV2(1)=INC*CCV2(1)
C       IF(CCV2(1).GE.2000.) CCV2(1)=2000.
      END IF
 61   CONTINUE
C
      IF((T12.GE.55.).AND.(T12.LE.57.))GOTO 62
      IF(T12.LE.RTSET)THEN
        CCV2(2)=DEC*CCV2(2)
      ELSE
        CCV2(2)=INC*CCV2(2)
      END IF
 62   CONTINUE
C
C    EXECESSIVE FLOW MONITORING ON BY-PASS
C
C     IF(V(18).GE.-20.)GOTO 64
C      DCV2(1)=DEC*DCV2(1)
C 64   CONTINUE
C
      IF((T2.LE.45.).AND.(T2.GE.43))GOTO 63
       IF(T2.GE.45.)THEN
       DCV2(1)=INC*DCV2(1)
       ELSE IF(T2.EQ.40)THEN
       DCV2(1)=DEC*DEC*DCV2(1)
       ELSE IF(T2.LE.43)THEN
       DCV2(1)=DEC*DCV2(1)
       END IF
 63    CONTINUE
C
      RETURN
      END
```

```
      SUBROUTINE GAUSSY(A,B,X,N)
C
C  SOLUTION OF SIMULTANEOUS EQN'S BY GAUSS ELIMINATION
C
C    PARAMETERS USED IN GAUSSY
C        A( )  = COEFFICIENT MATRIX
C        B( )  = VARIABLE MATRIX
C        X( )  = CALCULATED VALUES OF THE VARIABLES
C        N( )  = NUMBER OF VARIABLES
C
      DIMENSION A(N,N),X(N),B(N)
C
C   BEGINNING OF ELIMINATION PROCESS
C
      DO 28 K=1,N
C
C   MOVING THE LARGEST COEFF INTO DIAGONAL POSITION
C
      AMAX=0.0
      DO 4 I=K,N
C
C  CHECK COEFF MATRIX
C
      IF(ABS(A(I,K))-ABS(AMAX)) 4,4,2
 2    AMAX=A(I,K)
      IMAX=I
 4    CONTINUE
C
C   TESTING FOR INDEPENDENCE OF EQN'S
C
C
C   CHECK IMAX,K,ABS(AMAX)
C
      WRITE(1,111)IMAX,K,ABS(AMAX)
 111    FORMAT('0',10X,'IMAX=',I5,2X,'K=',I5,2X,'ABS(AMX)=',
     &G15.4)
C
      IF(ABS(AMAX)-0.1E-15) 10,10,14
 10   WRITE(6,12)
 12   FORMAT('0 EQUATIONS ARE NOT INDEPENDENT!')
      RETURN
C
C  EXCHANGE ROW IMAX AND ROW K
C
 14   BTEMP=B(K)
      B(K)=B(IMAX)
      B(IMAX)=BTEMP
      DO 18 J=K,N
      ATEMP=A(K,J)
      A(K,J)=A(IMAX,J)
 18   A(IMAX,J)=ATEMP
C
C  SUBTRACTING A(I,K)/A(K,K) TIMES TERM IN FIRST EQ FROM OTHERS
C
      KPLUS=K+1
      IF(K-N) 22,28,28
 22   DO 24 I=KPLUS,N
      B(I)=B(I)-B(K)*A(I,K)/A(K,K)
      ACON=A(I,K)
      DO 24 J=K,N
 24   A(I,J)=A(I,J)-A(K,J)*ACON/A(K,K)
 28   CONTINUE
```

```
C
C  BACK SUBSTITUTION
C
       L=N
 32    SUM=0.0
       IF(L-N) 34,38,38
 34    LPLUS=L+1
       DO 36 J=LPLUS,N
 36    SUM=SUM+A(L,J)*X(J)
 38    CONTINUE
       X(L)=(B(L)-SUM)/A(L,L)
       IF(L-1) 42,42,40
 40    L=L-1
       GOTO 32
 42    RETURN
       END


       SUBROUTINE TEMP
C      THIS ROUTINE IS TO ESTIMATE TEMPERATURES AT THE VARIOUS
C      STRATEGIC LOCATIONS:
C           T2        SUPPLY INTO BUILDING
C           T5        OUTLET FROM A COIL
C           T12       OUTLET FROM A COIL
C           T6        RETURN TO MAIN
C
       INCLUDE 'BSIM1.CMN'
C
       DATA STSET,RTSET/44.,56./
       DATA T2/40./
C
       T5=HLOAD(1)*12000./500./V(16)+T2
      T12=HLOAD(2)*12000./500./V(17)+T2
C
       T6=(T5*V(16)+T12*V(17))/V(15)
C
       IF(V(18).LE.0.0)THEN
       T2=40.0
       ELSE
       T2=(V(14)*TEMPSP+V(18)*T6)/V(15)
       END IF
C
       RETURN
       END


       SUBROUTINE TSET
C
C   TO DECIDE THE CONTINUATION OF THIS PROGRAM
C   BY COMPARING THE SIMULATED AND PRESET TEMPERATURES
C
       DIMENSION NOK(9)
       INCLUDE 'BSIM1.CMN'
C
       DATA TLO,THI/52.,57./
       DATA TSPLO,TSPHI/43.,45./
C
C   2-WAY COIL CNTROL VALVES
C
       PRINT*,ITHMAL,' ITHMAL '
       PRINT*,(TMP(IC),IC=1,NCOIL),' COIL OUTLET TEMP '
       PRINT*,TSP,' SECONDARY SUPPLY TEMP '
```

```
C
      DO 10 IC=1,NCOIL
       IF((TMP(IC).LE.TLO).OR.(TMP(IC).GE.THI))THEN
        NOK(IC)=1
       ELSE
        NOK(IC)=0
       END IF
 10   CONTINUE
C
      DO 12 ID=1,NCOIL
       IF(NOK(ID).EQ.1) THEN
        IUNC=ID
        PRINT*,ID,' COIL NEED ADJUSTMENT '
        GOTO 100
       END IF
 12   CONTINUE
       PRINT*,' COIL OUTLET TEMP CONTROLLED '
C
C   2-WAY BLDG CONTROL VALVE
C
      IF((TSP.GE.40.0).AND.(TSP.LE.45.0).AND.(V(68).GE.0.0))THEN
       IDO=0
       RETURN
      ELSE
       IDO=1
       RETURN
      END IF
C
 100  CONTINUE
       IDO=1
      RETURN
C
      END


      SUBROUTINE RESULT
C     TO SUMMARIZE THE SIMULATION RESULTS(INPUTS AND OUTPUTS)
C
      INCLUDE 'BSIM1.CMN'
C
      WRITE(6,*) '      '
      WRITE(6,*) '*** CHILLED WATER CIRCUIT SIMULATION INPUTS ***'
      WRITE(6,*) '      '
C
      IF(ITYPE.EQ.1)THEN
       WRITE(6,*)' CONSTANT SPEED PUMP SYSTEM '
      ELSE
       WRITE(6,*)' VARIABLE SPEED PUMP SYSTEM '
       WRITE(6,*)'      '
       WRITE(6,*)RTSET,' COIL SET TEMP(F)'
       WRITE(6,*)STSET,' PUMP SET TEMP(F)'
       WRITE(6,*)VPUMP,' BLDG DIFFERENTIAL PRESSURE SET(PSI)'
      END IF
C
      WRITE(6,*) '    '
      WRITE(6,*)DELPSR,' MAIN DIFFERENTIAL PRESSURE SET(PSI)'
      WRITE(6,*) '    '
C
      WRITE(6,*) NCOIL,'   TOTAL  NUMBER OF COIL'
      WRITE(6,*) '    '
       DO 5 IC=1,NCOIL
        WRITE(6,*)HLOAD(IC),' LOAD(TON) AT COIL #',IC
 5     CONTINUE
C
```

```
C
       WRITE(6,*) '     '
       WRITE(6,*) '     '
       WRITE(6,*) '     '
       WRITE(6,*) '     '
       WRITE(6,*) '*** CHILLED WATER CIRCUIT SIMULATION OUTPUTS ***'
       WRITE(6,*) '     '
       WRITE(6,*)          ' VARIABLE VALUES '
C
        DO 10 IS=1,ISECTN
          WRITE(6,*) V(IS),' HEAD(PSI) OF VARIABLE ',IS
 10     CONTINUE
C
          WRITE(6,*)'   '
        DO 15 IF=ISECTN+1,IVARMX
          WRITE(6,*) V(IF),' FLOW(GPM) OF VARIABLE ',IF
 15     CONTINUE
C
       WRITE(6,*)'    '
       WRITE(6,*)' HYDRAULIC RESULTS:'
       WRITE(6,*)'    '
C
        DO 20 IH=ISECTN+1,ISECTN+NCOIL
       WRITE(6,*)V(IH),'  FLOW(GPM) AT COIL #',IH
 20     CONTINUE
C
       WRITE(6,*)'  '
C
       WRITE(6,*)V(IVARMX-2),'  SECONDARY FLOW(GPM)'
       WRITE(6,*)V(IVARMX-1),'  BY-PASS FLOW(GPM)'
       WRITE(6,*)V(IVARMX),'  PRIMARY FLOW(GPM)'
C
       WRITE(6,*)'    '
C
       WRITE(6,*)' THERMAL RERSULTS:'
       WRITE(6,*)'    '
        DO 25 IT=1,NCOIL
         WRITE(6,*)TMP(IT),' TEMP(F) AT COIL #',IT
 25     CONTINUE
C
       WRITE(6,*)'  '
       WRITE(6,*)TSP,' TEMP(F) OF SECONDARY SUPPLY '
C
       WRITE(6,*)'    '
       WRITE(6,*)'    '
C
       WRITE(6,*)' VALVE INFORMATION:'
       WRITE(6,*)'  '
        DO 30 IV=1,NCOIL
         WRITE(6,*)CCV2(IV),' VALVE COEFF,CV, AT COIL #',IV
 30     CONTINUE
C
       WRITE(6,*)'  '
       WRITE(6,*)DCV2(1),'BLDG CONTROL VALVE COEFF,CV'
C
       WRITE(6,*)'    '
       WRITE(6,*)'  SIMULATION COMPLETED  '
       RETURN
       END
```

```
          SUBROUTINE OTHR
C
        DIMENSION TCHK(50,9)
        INCLUDE 'BSIM1.CMN'
C
        DO 10 ICHK=1,NCOIL
         TCHK(ITHMAL,ICHK)=TMP(ICHK)
 10     CONTINUE
C
        IF(ITHMAL.LE.2)GOTO 100
        DO 12 ICHK=1,NCOIL
        IF(ABS(TCHK(ITHMAL,ICHK)-TCHK(ITHMAL-1,ICHK)).LE.0.01)THEN
        NOT=1
        END IF
 12     CONTINUE
C
 100    CONTINUE
C
        RETURN
        END
```

```
$FOR DSIM
$LINK DSIM,DDATA,DINIT,DRESD,DPDRV,GAUSSY,DRESULT
$RUN DSIM


      PROGRAM DSIM
C
C  THIS PROGRAM IS TO CALCULATE FLOW, PRESSURE, AND TEMPERATURE AT
C  VARIOUS LOCATIONS IN A DISTRIBUTION LOOP.
C
C  THE FLOW RATES TO EACH BRANCH ARE KNOWN
C
      INCLUDE 'DSIM.CMN'
C
      OPEN(UNIT=5,NAME='DSIM.IN',STATUS='UNKNOWN')
      OPEN(UNIT=6,NAME='DSIM.OUT',STATUS='UNKNOWN')
      OPEN(UNIT=7,NAME='DSIM.CHK',STATUS='UNKNOWN')
C
C   CALL'DATA' TO OBTAIN  EQUIVALENT LENGTH AND DIAMETER
C
           CALL DDATA
C
C   CALL 'INIT' TO INITIALIZE ALL THE VARIABLES
C
           CALL DINIT
C
C   CALL 'RESD' TO CALCULATE RESIDUALS OF ALL FUNCTIONS
C
           CALL DRESD
C
C   ***  START ITERATIONS
C
        DO 100 ITER=1,ITERMX
C
      WRITE(7,*)' '
      WRITE(7,*)' '
             WRITE(7,*)ITER,' ITERATION '
             WRITE(7,*)(V(IVAR),IVAR=1,53),' VARIABLE VALUES'
             WRITE(7,*)(R(IVAR),IVAR=I,53),' RESIDUAL VALUES'
C
C         ASSIGN VARIABLE VALUES
C
          DO 105 IVAR=1,IVARMX
            VOLD(IVAR)=V(IVAR)
 105      CONTINUE
C
C         CALL 'PDRV' TO CALCULATE THE PARTIAL DERIVATIVES.
C
            CALL DPDRV
C
C         CALL 'GAUSSY' TO ESTIMATE THE CORRECTION VALUES
C
                N=IVARMX
C
            CALL GAUSSY(PD,R,X,N)
C
            WRITE(7,*)(X(IVAR),IVAR=1,IVARMX),' CORRECTED  VALUES'
C
C         CACULATE NEW VALUES FOR THE VARIABLES
```

```
C
                  DO 150 IVAR=1,IVARMX
                   VCORR  = X(IVAR)
                   V(IVAR)=VOLD(IVAR)-VCORR
 150              CONTINUE
C
C            CALL RESIDUAL WITH NEW VARIABLE VALUES
C
                   CALL DRESD
C
C  CHECK THE CONVERGENCY BASED ON RESIDUAL VALUES
C
                      DO 200 IVAR=1,IVARMX
 200                  IF(ABS(R(IVAR)).GT. EPSL) GOTO 99
C
C            WAU!!!  CNVG ACHIVED!
C
                 WRITE(7,*)' CONVERGENCY ACHIEVED AT ',ITER,'   ITERATIONS'
C                WRITE(7,*)' LET''S DO THERMAL ANALYSIS '
                 GOTO 500
C
 99           CONTINUE
C
 100         CONTINUE
C
C     ***   HYDRAULIC ITERATION END
C
 500         CONTINUE
C
        CALL DRESULT
C
C
        STOP
        END




        DIMENSION V(53),PD(53,53),R(53),VOLD(53),X(53)
        DIMENSION FR(13),BC(13)
        DIMENSION EL(26),ED(26),SF(39),SC(26)
        DIMENSION PC(1,4)
C
        COMMON /V1/ V
        COMMON /V2/ PD
        COMMON /V3/ R
        COMMON /V4/ VOLD
        COMMON /V5/ X
        COMMON /V6/ FR
        COMMON /V7/ BC
        COMMON /V8/ EL
        COMMON /V9/ ED
        COMMON /V10/ SF
        COMMON /V11/ SC
        COMMON /V12/ PC
        COMMON /V13/ NSUB
        COMMON /V14/ NSEC
        COMMON /V15/ IVARMX
        COMMON /V16/ IPUMP
        COMMON /V17/ ITERMX
```

```
      SUBROUTINE DRESD
C
C   THIS ROUTINE IS TO CAlCULATE THE RESIDUALS OF FUNCTIONS
C
      INCLUDE 'DSIM.CMN'
C
      NSUB=13
C
      IVARMX=53
C
      SF(1)=FR(1)+FR(2)+FR(3)+FR(4)+FR(5)+FR(6)+FR(7)+FR(8)+
     &  FR(9)+FR(10)+FR(11)+FR(12)+FR(13)
      SF(2)=FR(2)+FR(3)+FR(4)+FR(5)+FR(6)+FR(7)+FR(8)+
     &  FR(9)+FR(10)+FR(11)+FR(12)+FR(13)
      SF(3)=FR(3)+FR(4)+FR(5)+FR(6)+FR(7)+FR(8)+
     &  FR(9)+FR(10)+FR(11)+FR(12)+FR(13)
      SF(4)=FR(4)+FR(5)+FR(6)+FR(7)+FR(8)+
     &  FR(9)+FR(10)+FR(11)+FR(12)+FR(13)
      SF(5)=FR(5)+FR(6)+FR(7)+FR(8)+
     &  FR(9)+FR(10)+FR(11)+FR(12)+FR(13)
      SF(6)=FR(6)+FR(7)+FR(8)+FR(9)+FR(10)+FR(11)+FR(12)+FR(13)
      SF(7)=FR(7)+FR(8)+FR(9)+FR(10)+FR(11)+FR(12)+FR(13)
      SF(8)=FR(8)+FR(9)+FR(10)+FR(11)+FR(12)+FR(13)
      SF(9)=FR(9)+FR(10)+FR(11)+FR(12)+FR(13)
      SF(10)=FR(10)+FR(11)+FR(12)+FR(13)
      SF(11)=FR(11)+FR(12)+FR(13)
      SF(12)=FR(12)+FR(13)
      SF(13)=FR(13)
C
      DO  10 IR=1,13
         SF(2*IR)=SF(IR)
 10    CONTINUE
C
      DO 14 IP=1,NSUB*2
               SC(IP)=0.0135*0.02*EL(IP)/(ED(IP)**5)
         R(IP)=V(IP)-SC(IP)*SF(IP)**2
 14   CONTINUE
C
      DO 5 IBF=27,39
       IF(V(IBF+14).LE.0.0)THEN
          V(IBF+14)=-V(IBF+14)
          FR(IBF-26)=-FR(IBF-26)
       END IF
       R(IBF)=V(IBF)-FR(IBF-26)/SQRT((V(IBF+14)))
 5    CONTINUE
C
        R(40)=V(40)-(PC(1,1)+PC(1,2)*SF(1)+
     &          PC(1,3)*SF(1)**2+PC(1,4)*SF(1)**3)
C
C  NEED 13 MORE EQN'S
C
      R(41)=V(52)-(V(13)+V(53)+V(26))
      R(42)=V(51)-(V(12)+V(52)+V(25))
      R(43)=V(40)-(V(1)+V(2)+V(3)+V(4)+V(5)+V(6)+V(7)+V(8)+
     &    V(9)+V(10)+V(11)+V(51)+V(24)+V(23)+V(22)+V(21)+V(20)+
     &    V(19)+V(18)+V(17)+V(16)+V(15)+V(14))
      R(44)=V(40)-(V(1)+V(2)+V(3)+V(4)+V(5)+V(6)+V(7)+V(8)+V(9)+
     &    V(10)+V(50)+V(23)+V(22)+V(21)+V(20)+V(19)+V(18)+V(17)+
     &    V(16)+V(15)+V(14))
      R(45)=V(40)-(V(1)+V(2)+V(3)+V(4)+V(5)+V(6)+V(7)+V(8)+V(9)+
     &    V(49)+V(22)+V(21)+V(20)+V(19)+V(18)+V(17)+V(16)+V(15)+
     &    V(14))
```

```
      R(46)=V(40)-(V(1)+V(2)+V(3)+V(4)+V(5)+V(6)+V(7)+V(8)+V(48)+
     &  V(21)+V(20)+V(19)+V(18)+V(17)+V(16)+V(15)+V(14))
      R(47)=V(40)-(V(1)+V(2)+V(3)+V(4)+V(5)+V(6)+V(7)+
     &  V(47)+V(20)+V(19)+V(18)+V(17)+V(16)+V(15)+V(14))
      R(48)=V(40)-(V(1)+V(2)+V(3)+V(4)+V(5)+V(6)+V(46)+
     &  V(19)+V(18)+V(17)+V(16)+V(15)+V(14))
      R(49)=V(40)-(V(1)+V(2)+V(3)+V(4)+V(5)+V(45)+V(18)+V(17)+
     &  V(16)+V(15)+V(14))
      R(50)=V(40)-(V(1)+V(2)+V(3)+V(4)+V(44)+V(17)+V(16)+V(15)+V(14))
      R(51)=V(40)-(V(1)+V(2)+V(3)+V(43)+V(16)+V(15)+V(14))
      R(52)=V(40)-(V(1)+V(2)+V(42)+V(15)+V(14))
      R(53)=V(40)-(V(1)+V(41)+V(14))
C
C
      RETURN
      END



      SUBROUTINE DINIT
C
C     TO INITIALIZE ALL THE VARIABLES
C
      INCLUDE 'DSIM.CMN'
C
      DO 10 IA=1,26
      V(IA)=1.0
 10   CONTINUE
C
      DO 12 IB=41,50
      V(IB)=20.0
 12   CONTINUE
      V(51)=5.
      V(52)=4.
      V(53)=3.
C
      V(40)=40.
C THE VARIABLES OF BLDG COEFF ,BC
      V(27)=224.
      V(28)=1118.
      V(29)=201.
      V(30)=201.
      V(31)=27.
      V(32)=29.
      V(33)=30.
      V(34)=27.
      V(35)=56.
      V(36)=145.
      V(37)=101.
      V(38)=4.
      V(39)=7.
C
      RETURN
      END
```

```
      SUBROUTINE DDATA
C
C   THIS ROUTINE IS TO PROVIDE THE DATA FOR 'BSIM.FOR'
C
      INCLUDE 'DSIM.CMN'
C
      ITERMX = 20
      IVARMX = 53
      NSEC = 26
      NSUB = 13
      EPSL   = 0.1

C
      DO 10 IIS=1,NSEC
        READ(5,100)EL(IIS),ED(IIS)
 10     CONTINUE
C
C   DEMAND FLOW RATE
C
      DO 15 IIT=1,NSUB
        READ(5,100)FR(IIT)
 15     CONTINUE
C
 100    FORMAT(2F10.3)
C
      IPUMP=1
C
      IF(IPUMP.EQ.1)THEN
       PC(1,1)=49.
       PC(1,2)=7.324E-04
       PC(1,3)=-1.199E-07
       PC(1,4)=2.664E-12
      ELSE IF(IPUMP.EQ.2)THEN
       PC(1,1)=0.0
       PC(1,2)=0.0
       PC(1,3)=0.0
       PC(1,4)=0.0
      ELSE IF(IPUMP.EQ.3)THEN
       PC(1,1)=49.
       PC(1,2)=3.632E-04
       PC(1,3)=-1.797E-07
       PC(1,4)=5.325E-12
      ELSE IF(IPUMP.EQ.4)THEN
       PC(1,1)=0.0
       PC(1,2)=0.0
       PC(1,3)=0.0
       PC(1,4)=0.0
      END IF
C
      RETURN
      END
```

```
      SUBROUTINE DPDRV
C
C   THIS MODULE IS TO EVALUATE THE PARTIAL DERIVATIVES
C   THE RESIDUAL EQUATIONS
C
      INCLUDE 'DSIM.CMN'
C
C   PUT ZERO INTO ALL PD
C
       DO 10 II=1,IVARMX
          DO 12 IJ=1,IVARMX
             PD(II,IJ)=0.0
 12    CONTINUE
 10    CONTINUE
C
C  SUPPLY & RETURN SECTION
C
       DO 14 IK=1,NSUB*2
          PD(IK,IK)=1.0
 14    CONTINUE
C
C  BLDG BRANCH
C
       PD(27,27)=1.0
       PD(28,28)=1.0
       PD(29,29)=1.0
       PD(30,30)=1.0
       PD(31,31)=1.0
       PD(32,32)=1.0
       PD(33,33)=1.0
       PD(34,34)=1.0
       PD(35,35)=1.0
       PD(36,36)=1.0
       PD(37,37)=1.0
       PD(38,38)=1.0
       PD(39,39)=1.0
C
       DO 2 IBH=27,39
         IF(V(IBH+14).LE.0.1E-3)THEN
       PD(IBH,IBH+14)=1.*FR(IBH-26)
       ELSE
       PD(IBH,IBH+14)=-(-0.5/(V(IBH+14)**1.5)*FR(IBH-26))
       END IF
 2     CONTINUE
C
       PD(40,40)=1.0
C
C
       PD(41,52)=1.0
       PD(41,13)=-1.0
       PD(41,53)=-1.0
       PD(41,26)=-1.0
C
       PD(42,51)=1.0
       PD(42,12)=-1.0
       PD(42,52)=-1.0
       PD(42,25)=-1.0
C
       PD(43,40)=1.0
       DO 26 IO=1,11
        PD(43,IO)=-1.0
 26    CONTINUE
       DO 28 IP=14,24
```

```
          PD(43,IP)=-1.0
 28     CONTINUE
          PD(43,51)=-1.0
C
        PD(44,40)=1.0
        DO 30 IQ=1,10
          PD(44,IQ)=-1.0
 30     CONTINUE
        DO 32 IR=14,23
          PD(44,IR)=-1.0
 32     CONTINUE
          PD(44,50)=-1.0
C
        PD(45,40)=1.0
        DO 34 IS=1,9
          PD(45,IS)=-1.0
 34     CONTINUE
        DO 36 IT=14,22
          PD(45,IT)=-1.0
 36     CONTINUE
          PD(45,49)=-1.0
C
        PD(46,40)=1.0
        DO 38 IV=1,8
          PD(46,IV)=-1.0
 38     CONTINUE
        DO 40 IW=14,21
          PD(46,IW)=-1.0
 40     CONTINUE
        PD(46,48)=-1.0
C
        PD(47,40)=1.0
        DO 42 IX=1,7
          PD(47,IX)=-1.0
 42     CONTINUE
        DO 44 IY=14,20
        PD(47,IY)=-1.0
 44     CONTINUE
        PD(47,47)=-1.0
C
        PD(48,40)=1.0
        DO 46 IZ=1,6
          PD(48,IZ)=-1.0
 46     CONTINUE
        DO 48 IA=14,19
        PD(48,IA)=-1.0
 48     CONTINUE
        PD(48,46)=-1.0
C
        PD(49,40)=1.0
        DO 50 IB=1,5
        PD(49,IB)=-1.0
 50     CONTINUE
        DO 52 IC=14,18
        PD(49,IC)=-1.0
 52     CONTINUE
        PD(49,45)=-1.0
C
        PD(50,40)=1.0
        DO 54 ID=1,4
        PD(50,ID)=-1.0
 54     CONTINUE
        DO 56 IE=14,17
        PD(50,IE)=-1.0
 56     CONTINUE
        PD(50,44)=-1.0
```

```
C
      PD(51,40)=1.0
      DO 58 IF=1,3
      PD(51,IF)=-1.0
 58   CONTINUE
      DO 60 IG=14,16
      PD(51,IG)=-1.0
 60   CONTINUE
      PD(51,43)=-1.0
C
      PD(52,40)=1.0
      PD(52,1)=-1.0
      PD(52,2)=-1.0
      PD(52,14)=-1.0
      PD(52,15)=-1.0
      PD(52,42)=-1.0
C
      PD(53,40)=1.0
      PD(53,1)=-1.0
      PD(53,14)=-1.0
      PD(53,41)=-1.0
C
      IVARMX=53
C     DO 100 I=1,IVARMX
C       DO 100 J=1,IVARMX
C         IF(PD(I,J).NE.0.0)WRITE(7,*)PD(I,J),'OF EQN ',I,' & VAR',J
C 100   CONTINUE
C
      RETURN
      END


      SUBROUTINE DRESULT
C
C     TO SUMMARIZE THE SIMULATION RESULTS (INPUTS AND OUTPUTS)
C
      INCLUDE 'DSIM.CMN'
C
      WRITE(6,*) '      '
      WRITE(6,*) '*** DISTRIBUTION WATER CIRCUIT SIMULATION INPUTS ***
      WRITE(6,*) '      '
C
C     DISTRIBUTION PUMP OPTIONS (I THRU VII)
C
      WRITE(6,*) '  '
      WRITE(6,*)' DISTRIBUTION PUMP OPTION  IS',IPUMP
C
      WRITE(6,*)' '
      IF(IPUMP.EQ.1)THEN
       WRITE(6,*)'  PUMP 1,2,3, AND 4 ON '
      ELSE IF(IPUMP.EQ.2)THEN
       WRITE(6,*)'  PUMP     3  AND 4 ON '
      ELSE IF(IPUMP.EQ.3)THEN
       WRITE(6,*)'  PUMP  1,2,  AND 3 ON '
      ELSE IF(IPUMP.EQ.4)THEN
       WRITE(6,*)'  PUMP     1  AND 3 ON '
      ELSE IF(IPUMP.EQ.5)THEN
       WRITE(6,*)'  PUMP     3  OR  4 ON '
      ELSE IF(IPUMP.EQ.6)THEN
       WRITE(6,*)'  PUMP     1  AND 2 ON '
      ELSE IF(IPUMP.EQ.7)THEN
       WRITE(6,*)'  PUMP     1  OR  2 ON '
      END IF
```

```
C
       WRITE(6,*)' '
       WRITE(6,*)' '
C
C  PIPING DATA
C
       WRITE(6,*)'  '
       WRITE(6,*)'  SECTION LENGTH (FT) & DIAMETER(IN)   '
       WRITE(6,*)' '
       DO 1 IZ=1,NSEC
       WRITE(6,*)EL(IZ), ED(IZ)
 1     CONTINUE
       WRITE(6,*)' '
       WRITE(6,*)' '
       WRITE(6,*)' REQUIRED BRANCH FLOW RATE (GPM) '
       WRITE(6,*)' '
        DO 3 IA=1,NSUB
         WRITE(6,*) FR(IA),' DEMAND (GPM) AT BRANCH ',IA
 3      CONTINUE
C
       WRITE(6,*) ' '
       WRITE(6,*)' '
       WRITE(6,*)' '
       WRITE(6,*)' '
       WRITE(6,*) '      '
       WRITE(6,*) '      '
       WRITE(6,*) '      '
       WRITE(6,*) '      '
       WRITE(6,*) '*** DISTRIBUTION WATER CIRCUIT SIMULATION OUTPUTS ***'
       WRITE(6,*) '      '
       WRITE(6,*)' '
       WRITE(6,*)        ' VARIABLE VALUES '
C
       WRITE(6,*)' '
        DO 10 IS=1,NSEC
          WRITE(6,*) V(IS),' HEAD(PSI) OF SECTION  ',IS
 10     CONTINUE
C
          WRITE(6,*)'  '
        DO 15 IF=26+1,39
          WRITE(6,*) V(IF),' FLOW COEFF OF BRANCH  ',IF-26
 15     CONTINUE
C
       WRITE(6,*)'  '
        DO 20 IG=41,53
          WRITE(6,*) V(IG),' HEAD(PSI) OF BRANCH   ',IG-40
 20     CONTINUE
C
       WRITE(6,*)'   '
       WRITE(6,*)'  SIMULATION COMPLETED  '
       RETURN
       END
```

APPENDIX F

A PARASOL PROGRAM LISTING

```
$dfsb cntrl
#u1 = e u1 + $$
u = Kp e * Ki u1 * + $$
r = 1 $$
e = r y - $$
endsb
$dfsb sys
#dd = gna u * dd - toua / $$
#wps = gnv dd * wps - touv / $$
#tsin = gnp wps * tsin - toup / $$
#sot = gns tsin * sot - tous / $$
y = tsin $$
endsb
0 &u1 $ic
0 &dd $ic
0 &wps $ic
0 &tsin $ic
0 &sot $ic
1 8 3 $frmt
$dffn print .t r y u $$
.1 @.tsmp
.1 @.dtp
0.2 @gna
1.0 @toua
100 @gnv
1.0 @touv
0.0267 @gnp
1.5 @toup
0.25 @gns
0.5 @tous
0.12 @Kp
0.3 @Ki
```

2

VITA

Euy-Joon Lee

Candidate for the Degree of

Doctor of Philosophy

Thesis: MODELING AND SIMULATION OF A CAMPUS CENTRAL CHILLED WATER SYSTEM

Major Field: Mechanical Engineering

Biographical:

Personal Data: Born in Seoul, Korea, October 29, 1958, the son of Kap and Yang-Soon Lee. Married to Sook-Jong on December 5, 1981 and father to two sons, Jae-Hyun and Jae-Woo.

Education: Graduated from Ehwa High School, Seoul, Korea, in February, 1977; received Bachelor of Science degree in Mechanical Engineering from Yonsei University in February, 1981; received Master of Science degree from Oklahoma State University in July, 1983; completed requirements for the Doctor of Philosophy degree at Oklahoma State University in July, 1987.

Professional Experience: Teaching Assistant, Mechanical Engineering, Yonsei University, March, 1981 to August, 1981; Research Assistant, Fluid Power Research Center, Oklahoma State University, September, 1982, to May, 1983; Teaching Associate, School of Mechanical and Aerospace Engineering, Oklahoma State University, September, 1983, to May, 1987; Research Associate, School of Mechanical and Aerospace Engineering, Oklahoma State University, June, 1984, to June, 1987.