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IDAHO NATIONAL ENGINEERING LABORATORY

FEASIBILITY/CONCEPTUAL DESIGN STUDY FOR BOISE GEOTHERMAL SPACE HEATING DEMONSTRATION PROJECT BUILDING MODIFICATIONS

PREPARED FOR THE

ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION

IDAHO OPERATIONS OFFICE UNDER CONTRACT E (10-1)-1375

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ABSTRACT

Based on the premise that a geothermal fluid source of quality and having thermal properties similar to the existing Old State Penitentiary well water can be confirmed in the same general location, a feasibility/conceptual design effort for the Boise Space Heating Demonstration Project was started to determine practical ways to use this energy. The design activity included studies to establish methods by which certain public, State of Idaho buildings in Boise could economically be converted to utilize geothermal energy for space heat in lieu of presently installed more conventional heating systems. The first portion of that conceptual design effort is the required building modifications and is the basis for this report.

SUMMARY

The objective of this feasibility and conceptual design effort is concentrated on the potential for converting to geothermal heating, the State Capitol building, other buildings that make up the Capitol Mall, the State Veteran's Home and some of the more conversion compatible buildings that comprise the Boise State University Campus. This report** is directed toward a description of building modifications required to accomplish such a conversion.

1. From an individual building standpoint, and with the exception of the Len B. Jordan Building*, the physical conversion to geothermal heating appears to be a relatively straight forward modification. All components used in these estimates are state-of-the-art and are used in conventional hot water heating systems.

2. The proposed building modifications discussed in this report eliminate most, but not all of the present requirements for steam. The small dependency on steam that will remain was done purposely for several reasons, some of which are as follows:

   a. In many of the buildings, small steam unit space heaters are installed in equipment rooms or other areas open only to maintenance personnel. Changing these to hot water units could not be economically justified at this time. Once the geothermal systems are installed and operationally proven then these small devices can be changed out with maintenance or operating funds over a convenient and extended period of time.

   b. Most of the buildings use steam for heating potable water. As mentioned in the report, conversion of water heaters to geothermal heat was not addressed at this time primarily due to the lack of geothermal source information. As the resource information becomes available, conversion designs can be prepared.

* Difficult to access the HVAC coil areas, possibly achievable only by the temporary removal of a load bearing wall.

** Definitions for abreviations and units used throughout this report can be found in the Glossary section of the appendix.

c. For the State Veteran's Home, 180°F water is required in the kitchen area. With present geothermal source assumptions, this temperature cannot be attained but as proposed, geothermal heat will be used for preheating to 150°F.

d. In the State Capitol Building, steam is supplied to radiators just inside the entrance ways. These radiators, like the unit space heaters in other buildings, can be changed out later after operational characteristics for the overall modification are known.

3. A summary of the estimated conversion costs by building is included as Table 1. As a means of comparing or attempting to qualify the individual estimates, an estimated cost per million BTU/hr was computed for the purchase and installation of a typical gas fired boiler. This figure was multiplied by the individual heat ratings of the geothermal systems. It is apparent that relatively good correlation between the two estimates does exist. In comparing the mechanics of both items, the geothermal conversion figure represents the hardware and installation costs necessary to extract the required amount of heat from the geothermal water while the boiler figure represents the hardware and installation costs for a gas fired unit that will produce the same required amount of heat by conventional methods.

Comparing this latest estimate with the order of magnitude estimate for building system conversions published earlier (Ref. 1), the total $268,700 figure compares with $165,000 plus 25% for Design and Construction Engineering or a final figure of $206,250. The estimate published in this report is thus 30% higher than the order of magnitude one. It is felt that this amount of deviation is not out of line especially since the earlier estimate was made without benefit of individual heating system characteristics but only as a function of rated heat output.

4. Using an average heating load factor for the Boise area and published Degree Day data, the amount of heat per building for an average heating season was calculated. This amount of heat was then converted to a natural gas cost by applying a 75 percent overall heating plant efficiency factor and the current 1975 Intermountain Gas Company commercial service rates. Comparing the conversion cost estimate with the average gas consumption cost, both in 1975 dollars, one year's operational fuel savings will almost cancel

out the cost to convert each building.

5. No design or construction obstacle has been revealed to date that would significantly change the project scope. Looking at each building independently, the estimated cost of conversion is in line with the estimated cost to install conventional gas fired devices. In total, a very attractive yearly heating fuel savings can be realized. If the distribution and discharge system conceptual design estimates are still within the order of magnitude range, the operational savings will result in a short overall capital investment amortization period (ten years or less).

TABLE 1

ESTIMATED CONVERSION COSTS & COMPARISON

	Building	Estimated Conversion Cost	Heat Rating Of Conversion In 10^6 Btu/hr	Estimated Comparison Cost For Gas Fired Boilers Ac $5560/10^6 BTU/hr	Geothermal Water Required In GPM	Average Fuel Cost Per Season Based On Natural Gas At 1975 Commercial Rates
1.	Idaho State Capitol	$ 13,800	2.25	$ 12,510	227	$ 11,500
2.	State Veteran's Home	29,000	4.34	24,13[illegible]	438	21,400
3.	Len B. Jordan Office	26,900	6.37	35,41[illegible]	[illegible]55	30,900
4.	Idaho Supreme Court	23,200	3.72	20,683	49	18,300
5.	Idaho State Library	15,700	5.10	28,356	170	25,000
6.	New State Office	15,800	1.80	10,008	120	9,400
7.	School of Business - BSU	22,500	7.46	41,478	298	36,000
8.	Library - BSU	38,800	8.66	48,150	347	41,000
9.	Liberal Arts - BSU	28,700	3.54	19,682	403	17,700
10.	Student Union - BSU	54,300	6.08	33,805	244	29,400
	Totals	$ 268,700	49.32	$ 274,219	2651	$ 240,600
	Two Years Escalation @ 10%/year	56,427		57,586		
		$ 325,127		$ 331,805		
	25% Contingency	81,283		82,950		
		$ 406,410		$ 414,755		

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1.0 BACKGROUND INFORMATION

Geothermal waters of moderate temperature (<200°F) have been utilized for space heating in isolated cases throughout the world. Perhaps the most significant utilization is the capital city of Rekyjavik, Iceland where 85,000 people have virtually all of their space heat provided from hot water geothermal wells. In general, unless the hot water was already available, geothermal heating systems in this country have seldom been viewed as an economic alternative to fossil fuels. Since the 1973 oil embargo and with the spiraling costs of such fuels, it is believed that direct, deliberate efforts to employ geothermal energy for space heating will lead to its eventual widespread use and a significant savings to the nation's fossil fuel budget.

Geothermal space heating has been attempted on a modest scale at only two United States localities, the oldest of which is the goethermal heating system in Boise, Idaho, which has served the Warm Springs residential area since 1890. This system, with water at 170°F pumped from two 400-ft deep wells, at one time served 400 homes and business establishments but presently serves only about 120 homes. These two wells are known as the Old Penitentiary wells and are thought to be drilled intersecting the foothills fault geologic plane.

The downtown area and the adjacent capitol complex are situated within one mile of the Foothills Fault and as can be seen in Figure 1, many of the State buildings are located within 1/2 mile of this geologic zone. (A well on the Capitol grounds produces warm water.) At a slightly greater distance is the campus of Boise State University.

2.0 INTRODUCTION

Through previous preliminary investigations of existing heating systems within various public, State and Federal buildings (Ref. 2), ten of the systems, all State owned buildings, were determined to be convertible to geothermal heating with a relative minimum amount of individual building modifications. An important criteria for modification of these buildings was that existing heating systems would not be eliminated but would be retained on standby for backup or to supplement the geothermal system once actual change overs have been made.

The ten buildings included in the detailed feasibility and conceptual design studies were:

Capitol Mall Area:
Idaho State Capitol Building
L. B. Jordan State Office Building
Idaho Supreme Court Building
New State Office Building (under construction)
Idaho State Library Building

Boise State University:
School of Business Building
Library Building
Liberal Arts Building
Student Union Complex

State Veteran's Home

Assumptions concerning the geothermal fluid for this project were based on data obtained from the Old Penitentiary wells. The validity and apparent reliability of these assumptions and data as they are extrapolated for this project become of paramount importance to the overall success of the design. The 170°F water temperature is thought to be a realistic to conservative estimate and noncorrosive, almost potable water quality characteristics are expected to be duplicated. (Reference both the Penitentiary wells and the Capitol grounds well.)

Without an actual well location to use as the starting point for distribution considerations, a central accumulation site (for pumps, flow controls, etc. installations) located approximately 2050 feet northeast of the Veteran's Home within the Foothills Fault area, was assumed. Service conditions throughout were considered to be suitable for standard schedule 40 black iron pipe.

With these basic assumptions, the consequences or magnitude of design changes that would be involved if the assumptions are later proven to be inaccurate was considered. For all practical purposes, if a source of hot water is not found with temperatures of at least 150°F then the project scope and definition will require a major change. On the other hand, if an adequate quantity of water is confirmed with temperatures above 170°F then piping, pumps and heat exchanger equipment can be resized accordingly with subsequent favorable cost advantages. If the chemical quality of the water is corrosive, toxic or in general has an abnormally high dissolved solids or gas content then in all probability it would also exhibit higher enthalpy or temperature characteristics. Design concepts could thus be altered, to incorporate a "closed loop" water circulation system for distribution purposes and a relatively short coupled "supply well - heat exchanger - reinjection well" geothermal fluid loop at the well site. Otherwise, materials throughout the various systems would have to be selected on the basis of corrosive resistance, and the discharge system concept would have to account for the corrosive, toxic or dissolved contaminants with severe ultimate cost and operational disadvantages.

The conceptual design for the Boise Geothermal Space Heating Demonstration Project was divided into three separate areas for resolution:

Necessary Building Modification Requirements

Distribution System Requirements From The Foothills Fault Area to The Various Buildings

Environmentally Sound Discharge System Requirements

This report addresses itself to the Building Modifications. Supplemental reports will be issued to cover the other areas of activity.

3.0 BUILDING MODIFICATIONS

Each building was examined for ease of adding heat exchanger equipment without eliminating or making existing equipment inoperable. The extent to which equipment would be added was limited to practical considerations. The objective being to effectively utilize geothermal energy but not to redesign and replace entire building air conditioning systems.

Original heating system designs were based on good engineering practice utilizing the ARI winter design temperature of -10°F and 5890 Degree Days per average heating season in the Boise area. For all conversions the present installed heating capacity was used for establishing the size of geothermal heating equipment.

Hot water fin tube coils for each building were selected based on McQuay [1]. Coil computations were made using Trane [2] information but McQuay was chosen because of their wide range of fin and number of coil arrangement options. For the heat loads inherent to the existing structures and the established air flows in these buildings, increased design flexibility was an important evaluation criteria. These coils are manufactured with copper tubes, aluminum fins and though not "off-the shelf", are standard cataloged items. If later material deposition and compatibility studies prove these materials unsatisfactory, then other selections can be made.

Materials selected for all tube and shell heat exchangers were cast iron heads, carbon steel tubes and a carbon steel outer shell. If later these materials are determined unsatisfactory, other selections are available without excessive cost penalties.

(1) McQuay Division, McQuay - Perfex, Inc., Minneapolis, Minnesota.

(2) Trane Company, LaCrosse, Wisconsi, 54601.

Existing energy requirements for potable hot water were also compiled for the ten buildings under consideration. This information was obtained either directly from construction drawings or estimated. It was felt, however, that to properly justify conversion to geothermal heating for this application should wait until more information about the actual geothermal water source is known. For example:

1) The quality may be such that direct use is acceptable.
2) Temperatures could be higher or lower than the assumed 170°F temperature thus greatly changing heat exchanger calculations.

For all hot water coil installations discussed in this report, some method of equalizing air flow resistance between the hot section and cold section of each air handling system configuration must be made. This can be accomplished in a number of ways, for example, a "dummy" coil section could be added to the cooling coil section, a damper section could be added with an initial setting made by measurement of air flow after installation and then the dampers locked in position, or an expanded metal "screen" could be added in the cooling coil section that would equal the resistance of the new hot water coils. The latter technique was employed for the modifications in this report. Modification drawings show the addition of new coils, piping arrangements, and a circulation pump for each building, however, the addition of resistance equalizing screens are not shown. Estimated modification costs do reflect these screens, however, and to have shown them on the modification drawings would have served no real constructive purpose.

Each fin tube hot water coil or tube and shell heat exchanger system will require thermostatically controlled geothermal water flow modulating devices. These flow control devices, which are motor actuated throttling valves, will be interlocked electrically with the circulation pump or pumps for the building. The control scheme for all the buildings will be as follows:

1) Existing building temperature sensing devices will originate electrical or pneumatic signals calling for heat. These signals, if from more than one device per system, will require a master, temperature averaging, thermostatic control device from which signals open and close the flow control valve and start or stop a circulation pump. If one thermostat per system is used then this device will perform

the same functions described above. It is suggested that existing controls be utilized wherever possible and a manual switch over device be installed to enable selection of steam or hot water heat control.

2) For multizone units and dual air supply systems, multiple thermostats already exist but are connected to modulating damper systems. These control devices will remain as installed.

3) Control functions between multiple air handling/coil systems have to be arranged such that if any one system requires hot water then the circulation pump or one pump, in a multiple arrangement, will be activated.

Individual building conversion cost estimates were derived by obtaining verbal quotes from local distributors on all major items. Equipment quotes used by manufacturer were: McQuay - fin tube hot water coils; Bell & Gossett[3] - tube and shell water/water heat exchangers and circulation pumps; Fisher[4] - pressure control valves; Robertshaw[5] - thermostatically controlled valves. Average material and installation cost figures were obtained for pipe, wiring, conduit and all miscellaneous items required for the modifications. (Ref. 3) After computing basic conversion costs, percentage allowances were added to cover such items as general contractor overhead and profit, engineering design and project contingencies.

For future geothermal water requirement considerations, data was compiled on several other existing public buildings that under preliminary investigation appear to be good candidates for conversion. This information was not used in this report but is required for distribution and discharge considerations.

(3) Bell & Gossett - ITT, Morton Grove, Illinois, 60053

(4) Fisher Controls Company, Marshalltown, Iowa

(5) Robertshaw Controls Company - Milford Division, 155 Hill Street, Milford, Conn., 06460.

BOISE, IDAHO, GEOTHERMAL SPACE HEATING PROJECT

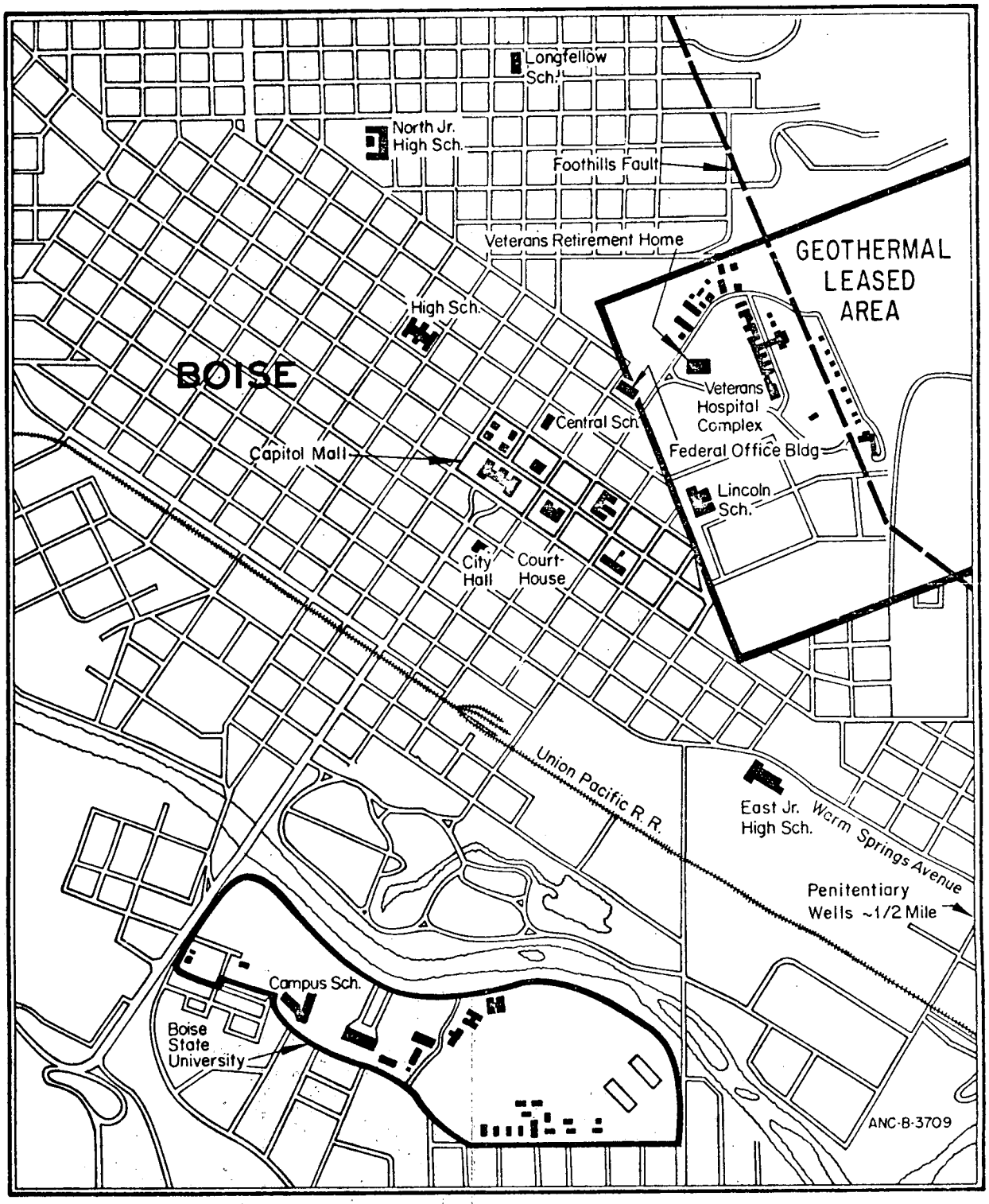


Figure 1. Boise, Idaho, Geothermal Space Heating Project

Figure 3.1-0 State Capitol

3.1 Idaho State Capitol Building

The existing system uses a single steam/water heat exchanger rated at 2,250,000 BTU/hr with provisions to add a second exchanger for future added heat load.

Conversion to geothermal will involve the installation of a conventional tube and shell water/water heat exchanger along with the present steam/water exchanger and valving to isolate either from the building circulation system. A circulation pump for the geothermal water and automatic control valves will complete the building modifications.

Using 170°F inlet water temperature and a 20°F drop in temperature across the exchanger, the dimensions of an exchanger to duplicate the present heating capacity is approximately 16 in. dia. by 9 ft in length with 6 in. dia. pipe connections on the geothermal side and 4 in. dia. pipe connections on the circulation, heating side. Geothermal water flow at maximum rated heating capacity will be approximately 227 gpm. Supply and discharge pipe size to and from the building will be 3-1/2 in. dia. The estimated conversion cost for this building in 1975 dollars without contingency or escalation is $13,800.

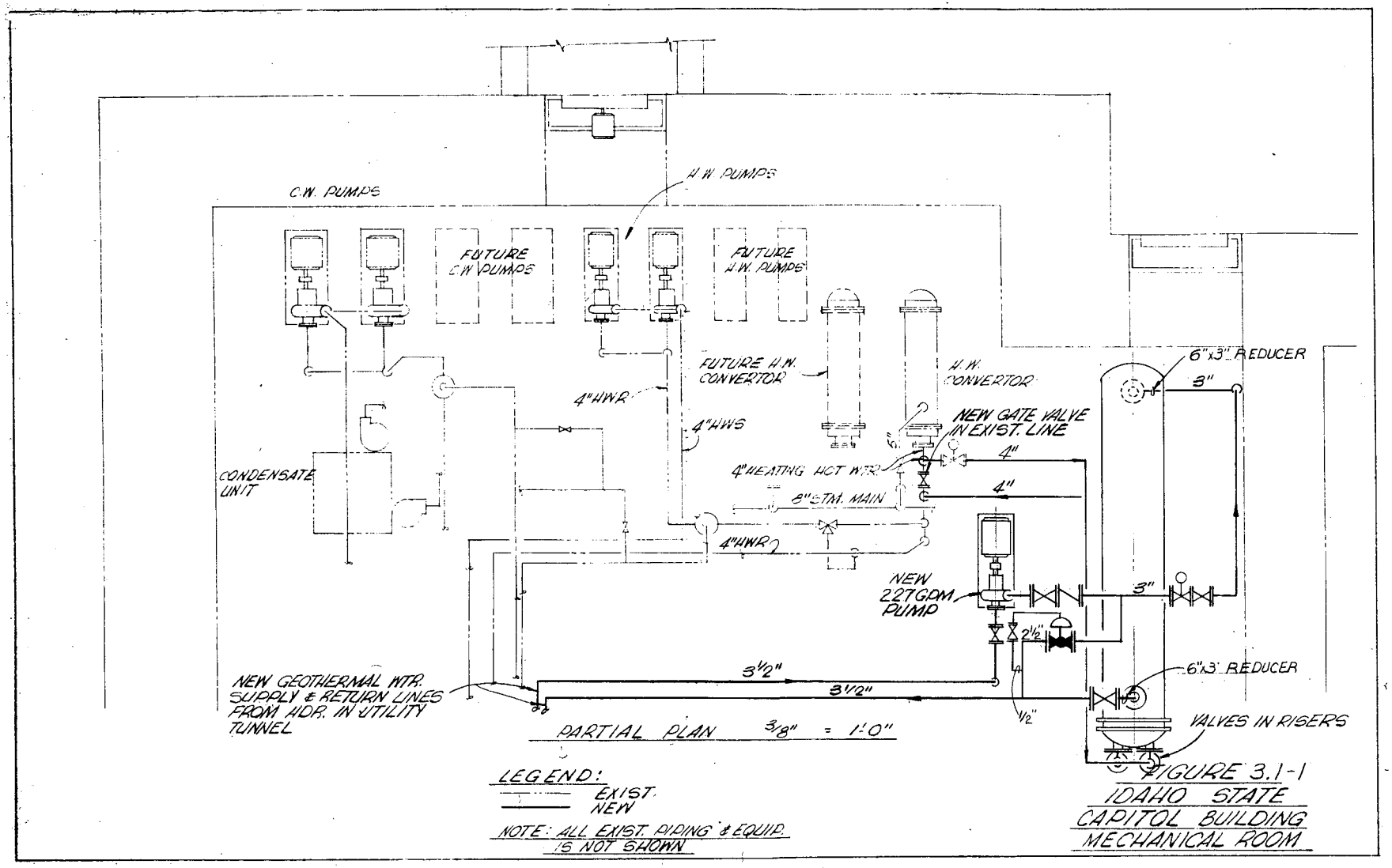


Figure 3.1-1 Idaho State Capitol Building Mechanical Room

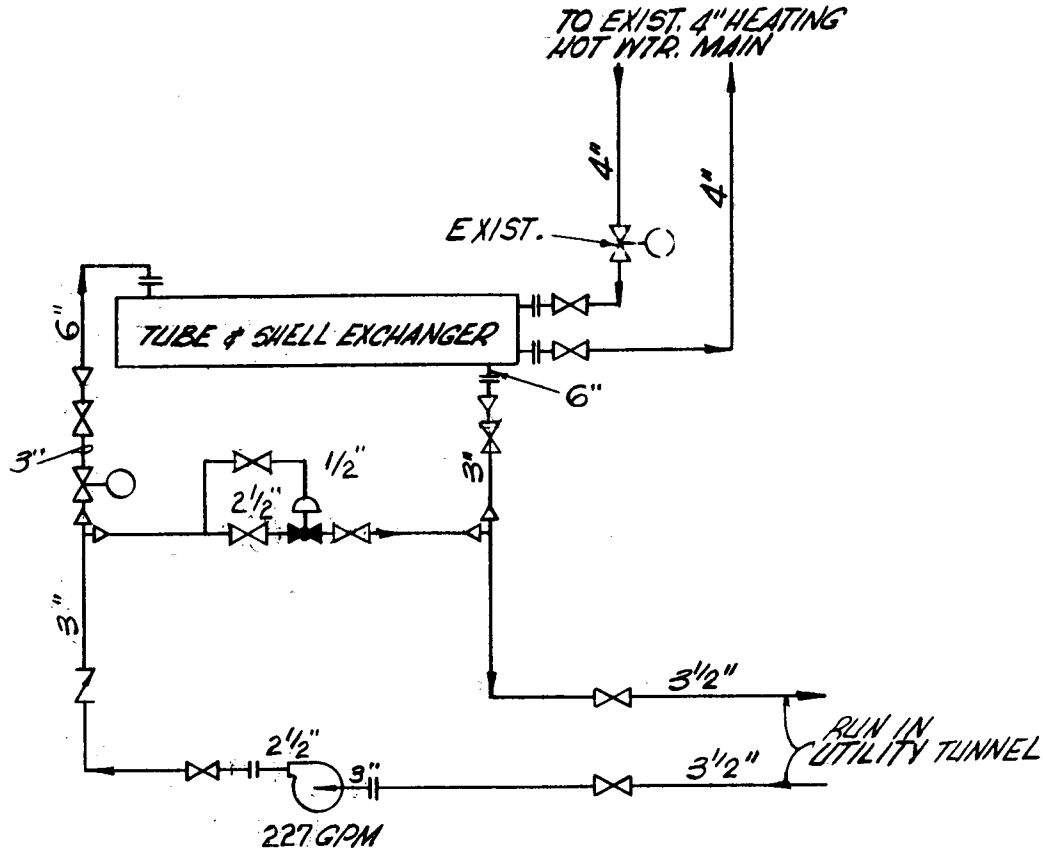


FIGURE 3.1-2
IDAHO STATE CAPITOL BLDG.

Figure 3.1-2 Idaho State Capitol Building

Figure 3.2-0 State Veteran's Home

3.2 State Veteran's Home

The existing heating system is partially integrated with the domestic hot water supply for the building. Present steam/water exchanger rated capacities include 2,855,000 BTU/hr for comfort heating, 1,120,000 BTU/hr for laundry/kitchen hot water and 560,000 BTU/hr for other domestic hot water requirements. These heating capacities correspond to 285 gpm of water at 140°F for comfort heating, 16 gpm of water at 180°F for the laundry/kitchen use and 11 gpm of water at 140°F for other domestic uses. The basic geothermal source assumption limits the design from obtaining a 180°F water temperature but the flow rates of 16 gpm plus 11 gpm can be obtained at an intermediate temperature, such as 150°F.

The proposed method of utilizing geothermal energy for this building is to install a water/water tube and shell heat exchanger in parallel with the existing comfort heating exchanger with the new exchanger sized to duplicate present capacities and to install another exchanger for the domestic hot water to supply 150°F water for use at that temperature and to serve as a preheater for the laundry/kitchen water. The higher temperature water can be heated from 150°F with electric units. Approximate physical dimensions of equipment to accomplish this are:

Comfort Heating - 16 in. dia. by 10 ft long exchanger
- Geothermal Connections - 4 in. dia. pipe
- Circulated Hot Water Connections - 4 in. dia. pipe

Potable Hot Water - 16 in. dia. by 10 ft long exchanger
- Geothermal Connections - 4 in. dia. pipe.
- Circulated Hot Water Connections - 1-1/2 in. dia. pipe

Geothermal water circulation pumps and automatic control valves will be required. Total maximum geothermal water flow will approach 460 gpm with supply and discharge lines to the building sized at 5 in. dia. The estimated building conversion cost in 1975 dollars without escalation or contingency is $29,000.

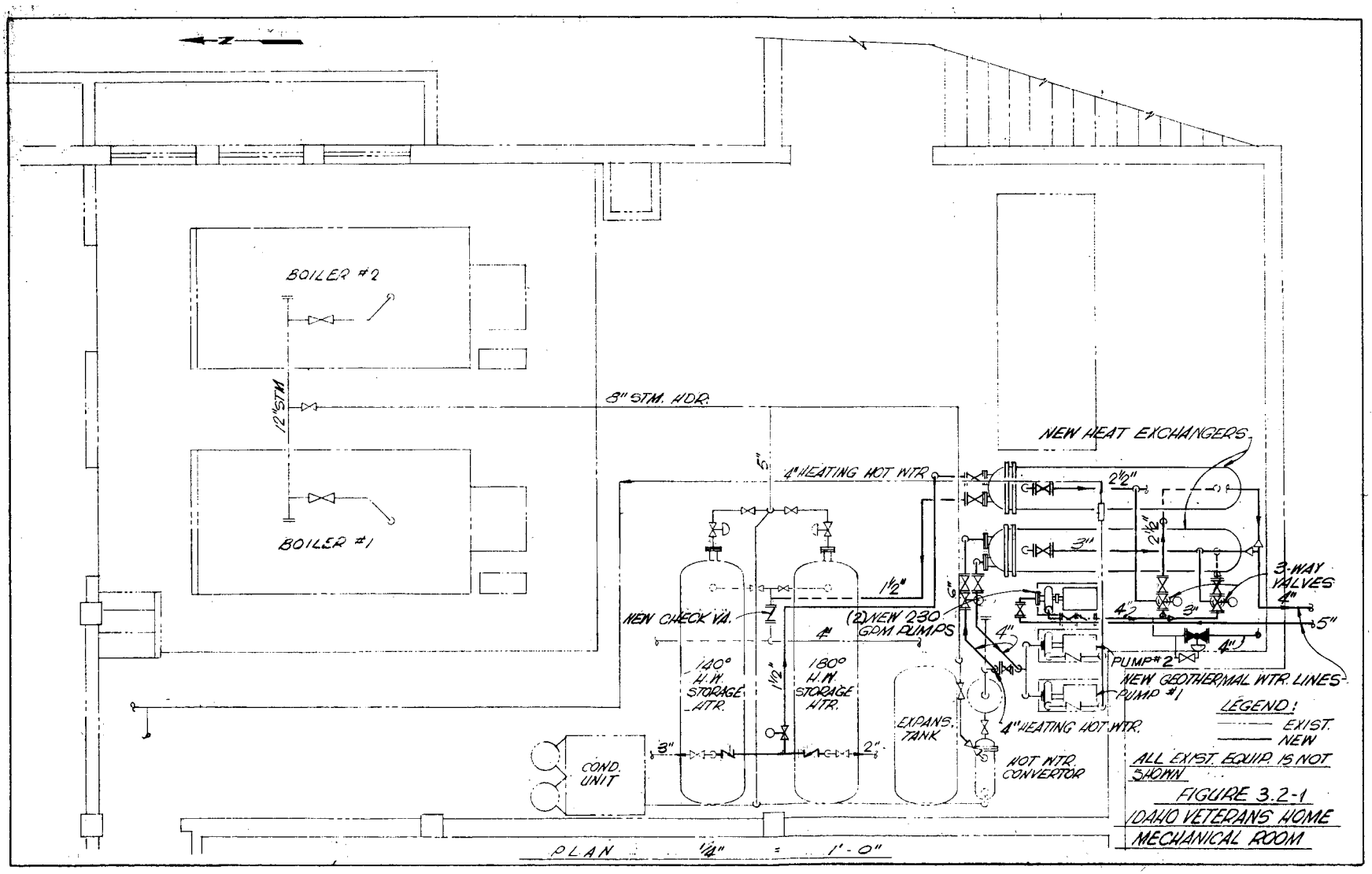
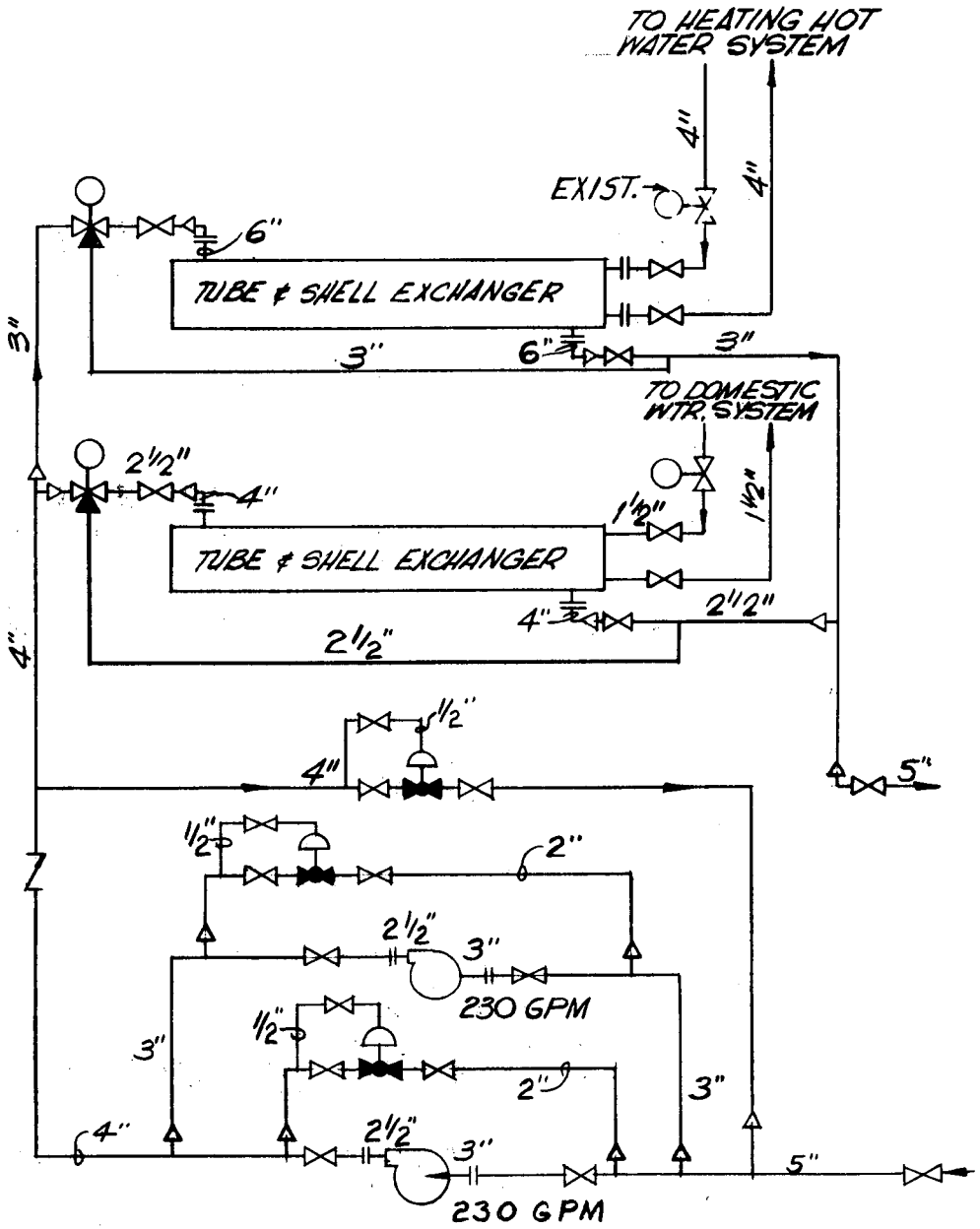


Figure 3.2-1 Idaho Veterans Home Mechanical Room



FLOW RATES SHOWN ARE CALCULATED VALUES PLUS 5%

FIGURE 3.2-2
IDAHO VETERAN'S HOME

Figure 3.2-2 Idaho Veteran's Home

Figure 3.3-0 Len B. Jordan Office

3.3 Len B. Jordan Office

Two central plant type forced air HVAC units are utilized for the major portion of the building and one small package HVAC unit for the Emergency Operation Center (E.O.C.) area. Both large units have 100 HP fan motors and each system is rated at 51,700 SCFM. The E.O.C. unit has a 25 HP fan motor and the system is rated at 12,000 SCFM. Steam coils are used in the hot deck sections and chilled water coils in the cold decks. All three systems have room for additional coils in the heating sections. See Table 3.3-1 for total coil changes.

Based on the calculations shown in Tables 3.3-3 through 3.3-6, coils with identical face areas as the existing steam coils can be installed that will duplicate the heating capacity ratings. For the 100 HP systems the increased resistance to airflow as a result of added coils would increase the fan static pressure requirements by 1.1 inches of water. Adjusted fan speed to overcome this resistance would add less than 9% to the fan horsepower requirement. Likewise, a coil added to the 25 HP system would increase the static pressure requirement by 1.2 inches of water and the fan horsepower requirement by less than 10%. From the data available, it appears that sufficient reserve capacity exists in all system motors. Fan speed adjustments for the 100 HP systems will amount to an increase from 855 RPM to 900 RPM, only a 5.4% change. From actual fan speed measurements, these systems apparently are operating above the design value at a speed of 1050 RPM. This indicates that actual air flow rates are above the 51,700 SCFM value and the coil face velocities above the design value. It is recommended that the coil face velocities be kept below 1000 FPM or closer to the 940 FPM design value to minimize air turbulence and fan static pressure. Again, horsepower, air flow resistance increase and heat exchange rates were based on the design value of 51,700 SCFM.

The 25 HP system would require a fan speed adjustment from the 1650 RPM value to 1765 RPM or a 7% increase.

Total geothermal water flow required to the building would be 265 gpm and main building supply and discharge lines sized at 5 in. dia. Reference Figures 3.3-1 through 3.3-4 and calculation sheets, Tables 3.3-1 through 3.3-6, for total proposed modifications.

Even though the Len B. Jordan building is a good candidate for conversion from an individual equipment standpoint, field inspections will verify that access to the coil sections of the individual air handlers is almost impossible with the size coils required. It would appear that partial demolition of a load bearing wall is required to either remove existing coils or install additional ones. The estimated cost to convert this building, if the load bearing wall interference can be resolved without removal, is $26,900. This figure is in 1975 dollars without escalation or contingency.

TABLE 3.3-1

SUMMARY

BUILDING Len B. Jordan State Office

TOTAL MAXIMUM HOT WATER (GEOTHERMAL) REQUIRED FOR THIS BUILDING - 254.7 GALLONS PER MINUTE

TEMPERATURE OF WATER TO THE COILS ASSUMED TO BE 170°F AND THE TEMPERATURE OF WATER FROM THE COILS ASSUMED TO BE 120°F UNLESS NOTED OTHERWISE.

HOT WATER COIL CALCULATIONS, MODEL NUMBERS AND NOMENCLATURE USED ARE BASED ON Mc QUAY.

	PER AIR HANDLING SYSTEM								PER FIN TUBE HOT WATER COIL								
NO[1]	AIR[2] FLOW IN (SCFM)	AIR[3] DESIGN IN(°F) IN	OUT	AIR[4] FLOW IN (SCFM)	AIR[5] FLOW (ΔP)	FAN[6] ΔHP	HEATING[7] CAPACITY IN (BTU/HR)	WATER FLOW IN (GPM)	FIN TUBE COIL MODEL NUMBER TYPE	FIN SERIES	TUBE ROWS "C"	SIZE (W X L X D) IN (INCHES)	PIPE[8] SIZE	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	WATER[10] FLOW (ΔP) (PSI)	COILS PER SYSTEM
2	51700	51	97	51700	1.1	8.93	2600000	104	5WL	08	03	33x120x6	1-1/2	27.5	940	4.92	2
1	12000	5	95	12000	1.23	2.32	1166400	46.7	5WS	08	04	21x96x7.5	2	14	860	1.29	1

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
4 DESIGN AIR FLOW THROUGH THE HEATING COIL SECTION UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
5 INCREASE IN AIR PRESSURE REQUIRED TO OVERCOME THE ADDITIONAL RESISTANCE OF THE HOT WATER COILS WHILE MAINTAINING THE DESIGN AIR FLOW UNITS - INCHES OF WATER
6 INCREASE IN FAN MOTOR HORSEPOWER REQUIRED TO PRODUCE THE ADDITIONAL AIR PRESSURE
7 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR
8 PIPE DIAMETER OF THE INLET AND OUTLET CONNECTIONS FOR THE INDIVIDUAL COILS UNITS - INCHES
9 AVERAGE VELOCITY OF THE AIR AT THE HOT WATER COIL UNITS - FEET PER MINUTE
10 PRESSURE LOSS OF THE HOT WATER AS IT FLOWS THROUGH EACH COIL UNITS - POUNDS PER SQUARE INCH

TABLE 3.3-2

BUILDING Len B. Jordan State Office

HEATING SYSTEM INFORMATION DERIVED FROM CONSTRUCTION DRAWINGS, BUILDING SPECIFICATIONS AND FIELD DATA

	PER AIR HANDLING SYSTEM								PER FIN TUBE STEAM COIL							
NO[1]	AIR[2] FLOW IN (SCFM)	FANS[3]	FAN RPM[4] DESIGN / MEAS.	FAN HP[5] DESIGN / MEAS.	MOTOR[6] VOLTS PHASE	FAN[7] SP @ AIR FLOW	FAN[8] AIR VEL. FPM	MEASURED AIR FLOW IN (SCFM)	COILS PER SYSTEM	SIZE (W X L) IN INCHES	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	AIR[10] PER COIL (SCFM)	AIR[11] DESIGN IN(°F)	AIR[11] DESIGN OUT(°F)	HEATING[12] CAPACITY IN (BTU/HR)
2	51700	1	855 / 1050	100 / 80	460/3	6.5	2100		2	33x120	27.5	940	25850	51	97	1300000
1	12000	1	1650 / 1650	25 / 21	460/3	5.5	2350		2	21x96	14	860	12000	5	95	583000

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 NUMBER OF IDENTICAL FANS PER SYSTEM
4 FAN SPEED BOTH DESIGN AND THAT MEASURED AT INSTALLATION UNITS - REVOLUTIONS PER MINUTE
5 FAN DRIVE MOTOR HORSEPOWER BOTH DESIGN AND THAT MEASURED AT INSTALLATION
6 DRIVE MOTOR VOLTAGE AND PHASE; ALL VOLTAGE FREQUENCIES ARE 60 HERTZ
7 RATED STATIC AIR PRESSURE PRODUCED BY THE FAN AT THE DESIGN AIR FLOW UNITS - INCHES OF WATER
8 AVERAGE VELOCITY OF THE AIR AT THE FAN DISCHARGE UNITS - FEET PER MINUTE
9 AVERAGE VELOCITY OF THE AIR AT THE STEAM COIL UNITS - FEET PER MINUTE
10 DESIGN AIR FLOW PER STEAM COIL UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
11 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
12 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR

TABLE 3.3-3

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Len B. Jordan State Office

RATING OF SYSTEM 51700 (SCFM) 1300000 (BTU/HR) NUMBER OF SYSTEMS (2)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 33 X 120/144 = 27.5 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 51700 /2(27.5)= 940 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 46 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 1300000/(500)X(50) = 52 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WL TYPE COIL WITH 3,4 ROWS = 16

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 52 / 16 = 3.25

7) $\Delta T_A/\Delta T_{inlet}$ = 46 / 170 (°F) - 51 = 46 / 119 = .387

$\Delta T_W/\Delta T_A$ = 50 / 46 = 1.09

FROM FIGURE 8 : M_t = .710

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 3 /(.71 X 9.4) =.450

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}}$ / 2 = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .101

$R_{f2} = R_{ft} - R_{f1}$ = .450 - .101 = .349

FROM FIGURE 6 : FIN SERIES = 08 R_{f2} = .300

COIL MODEL NUMBER BECOMES: 5WL-0803C-33x120 NUMBER OF COILS REQ'D 2

9) FROM FIGURE 9 : ΔP_A = 1.1 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(51700)X(1.1) = 8.93

10) FROM FIGURE 12 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(3.5 + 7.0) =

11.34 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 4.92 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.3-4

NEW FAN SPEED ESTIMATE SHEET

BUILDING Len B. Jordan State Office

RATING OF SYSTEM 51700 (SCFM) 2600000 (BTU/HR) NUMBER OF SYSTEMS (2)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$FAN\ BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(\ 51700\ )X(\ 6.5\ ) =$ 52.76

$FAN\ BHP_2 = (FAN\ BHP_1)+(FAN\ BHP_+) = (\ 52.76\ )+(\ 8.93\ ) =$ 61.69

$RPM_2 = (RPM_1)X(FAN\ BHP_2/FAN\ BHP_1)^{1/3} = (\ 855\ )X(61.69\ /\ 52.76)^{1/3} =$ 900

CHANGE IN FAN SPEED = $(RPM_2 - RPM_1)/RPM_1 =$ .054 OR X 100 = 5.4 % INCREASE

DEFINITIONS:

$FAN\ BHP_1$ = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

$FAN\ BHP_+$ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9,FROM THE COIL SELECTION SHEET).

$FAN\ BHP_2$ = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

TABLE 3.3-5

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Len B. Jordan State Office

RATING OF SYSTEM 12000 (SCFM) 1166400 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 21 X 96 /144 = 14 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 12000 / 14 = 860 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 90 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 1166400 /(500)X(50) = 46.7 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WS TYPE COIL WITH 2, 3, 4 ROWS = 14

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 46.7 / 14 = 3.34

7) $\Delta T_A/\Delta T_{inlet}$ = 90 / 170 (°F) - 5 = 90 / 165 = .545

$\Delta T_W/\Delta T_A$ = 50 / 90 = .556

FROM FIGURE 8 : M_t = 1.04

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 4 /(1.04 X 8.6) = .447

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}} / 2$ = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .10

$R_{f2} = R_{ft} - R_{f1}$ = .447 - .10 = .347

FROM FIGURE 6 : FIN SERIES = 08 R_{f2} = .315

COIL MODEL NUMBER BECOMES: 5WS-0804C-21x96 NUMBER OF COILS REQ'D 1

9) FROM FIGURE 9 : ΔP_A = 1.23 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(12000)X(1.23) = 2.32

10) FROM FIGURE 12 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(.85 + 1.9) =

2.97 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 1.29 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.3-6

NEW FAN SPEED ESTIMATE SHEET

BUILDING Len B. Jordan State Office

RATING OF SYSTEM 12000 (SCFM) 1166400 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$FAN\ BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(\underline{12000})X(\underline{5.5}) = \underline{10.36}$

$FAN\ BHP_2 = (FAN\ BHP_1)+(FAN\ BHP_+) = (\underline{10.36})+(\underline{2.32}) = \underline{12.68}$

$RPM_2 = (RPM_1)X(FAN\ BHP_2/FAN\ BHP_1)^{1/3} = (\underline{1650})X(\underline{12.68}/\underline{10.36})^{1/3} = \underline{1765}$

CHANGE IN FAN SPEED = $(RPM_2 - RPM_1)/RPM_1$ = .07 OR X 100 = 7 % INCREASE

DEFINITIONS:

$FAN\ BHP_1$ = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

$FAN\ BHP_+$ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9,FROM THE COIL SELECTION SHEET).

$FAN\ BHP_2$ = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

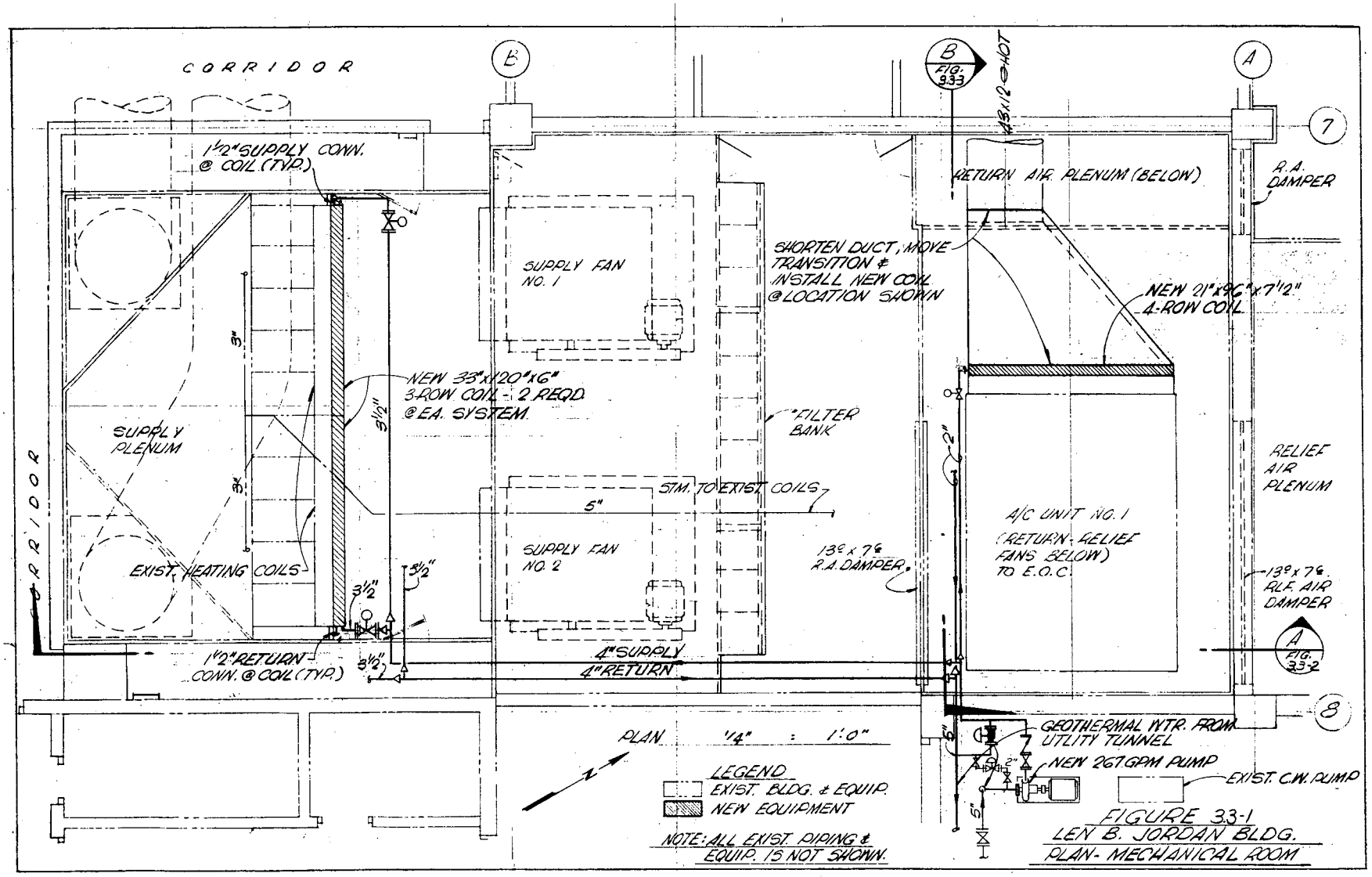


Figure 3.3-1 Len B. Jordan Building Mechanical Room

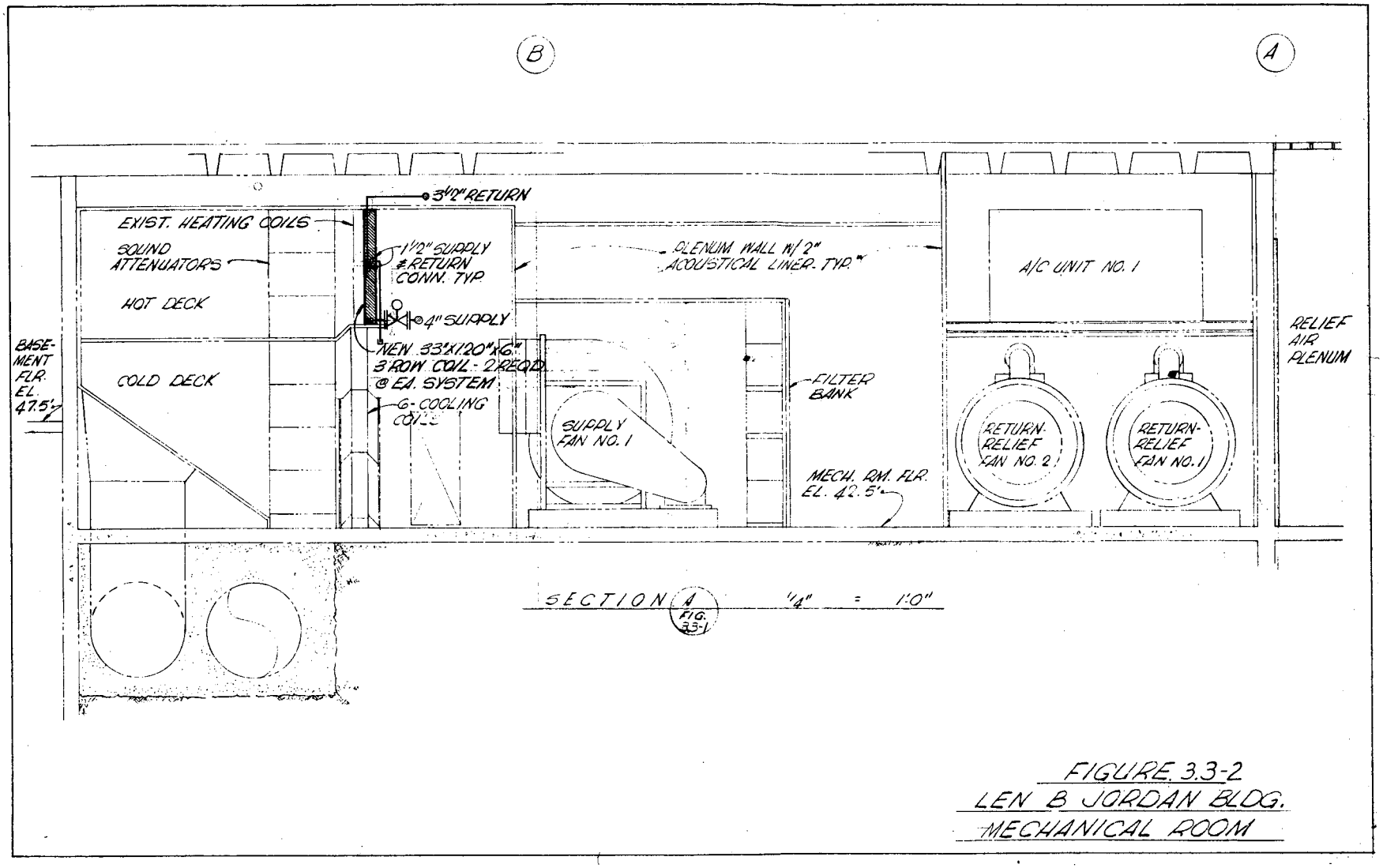


Figure 3.3-2 Len B. Jordan Building Mechanical Room

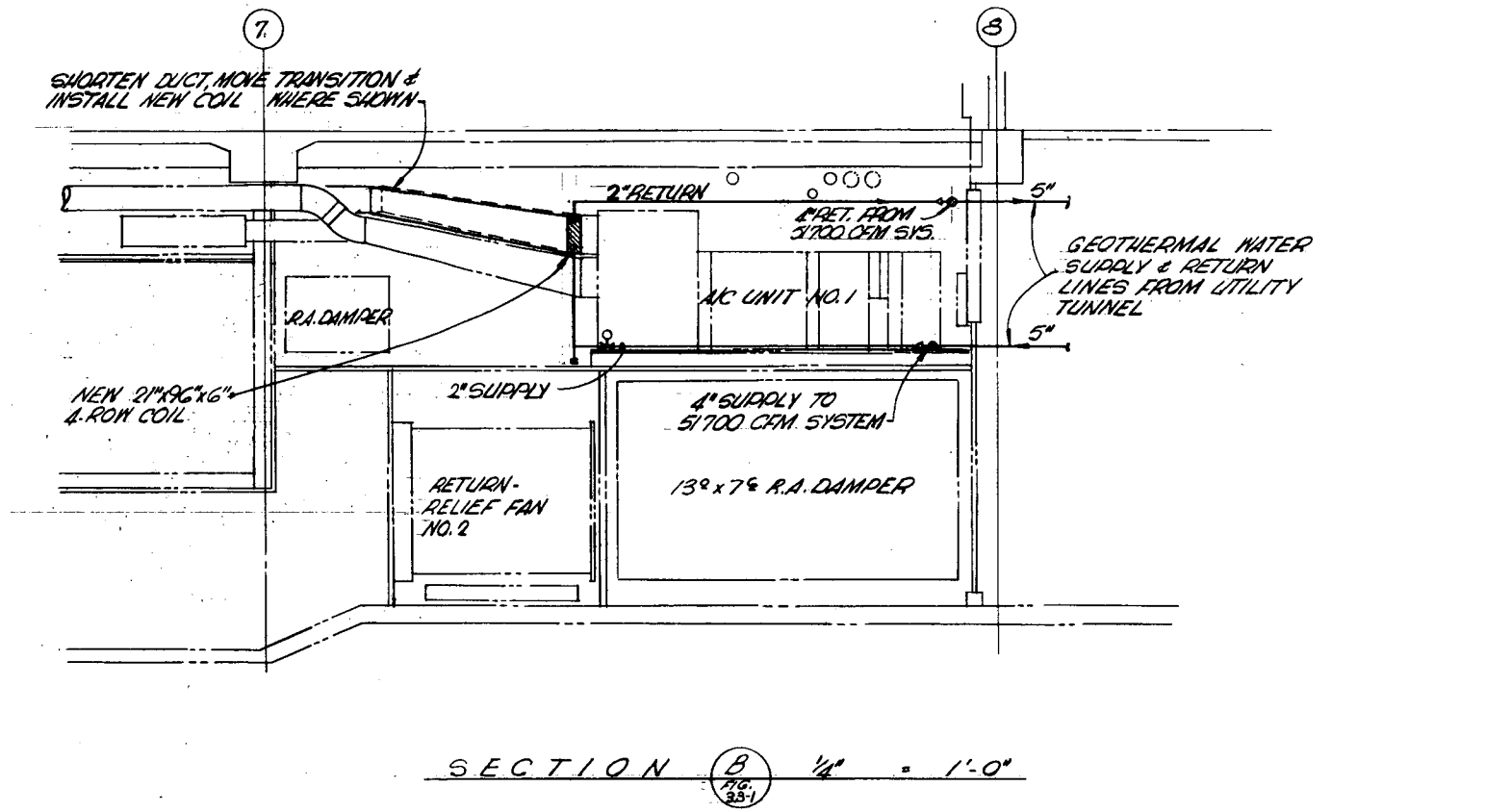
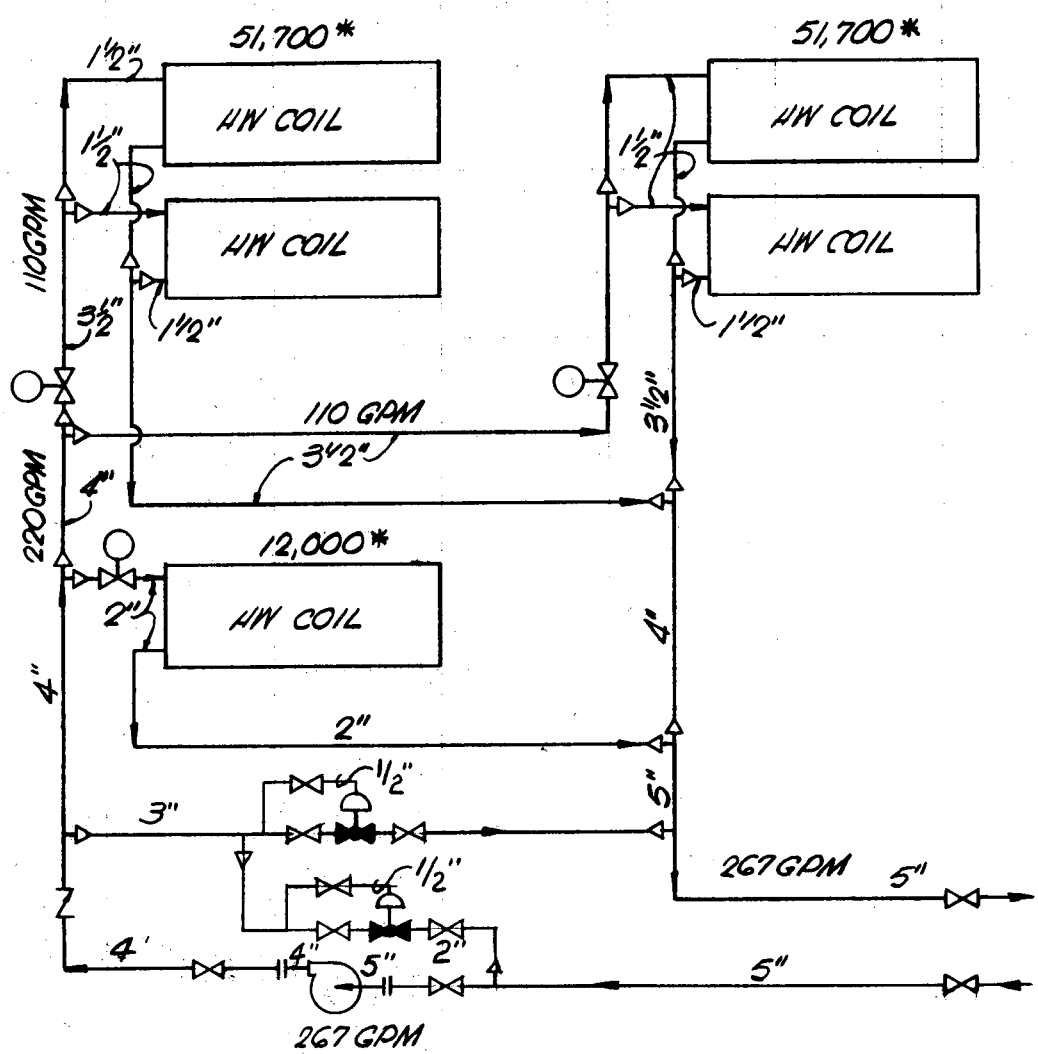


FIGURE 3.3-3
LEN B. JORDAN BLDG.
MECHANICAL ROOM

Figure 3.3-3 Len B. Jordan Building Mechanical Room



FLOWS SHOWN ARE CALCULATED VALUES PLUS 5%
* RATED AIR FLOW OF SYSTEM IN SCFM

FIGURE 3.3-4
LEN B. JORDAN BLDG.

Figure 3.3-4 Len B. Jordan Building

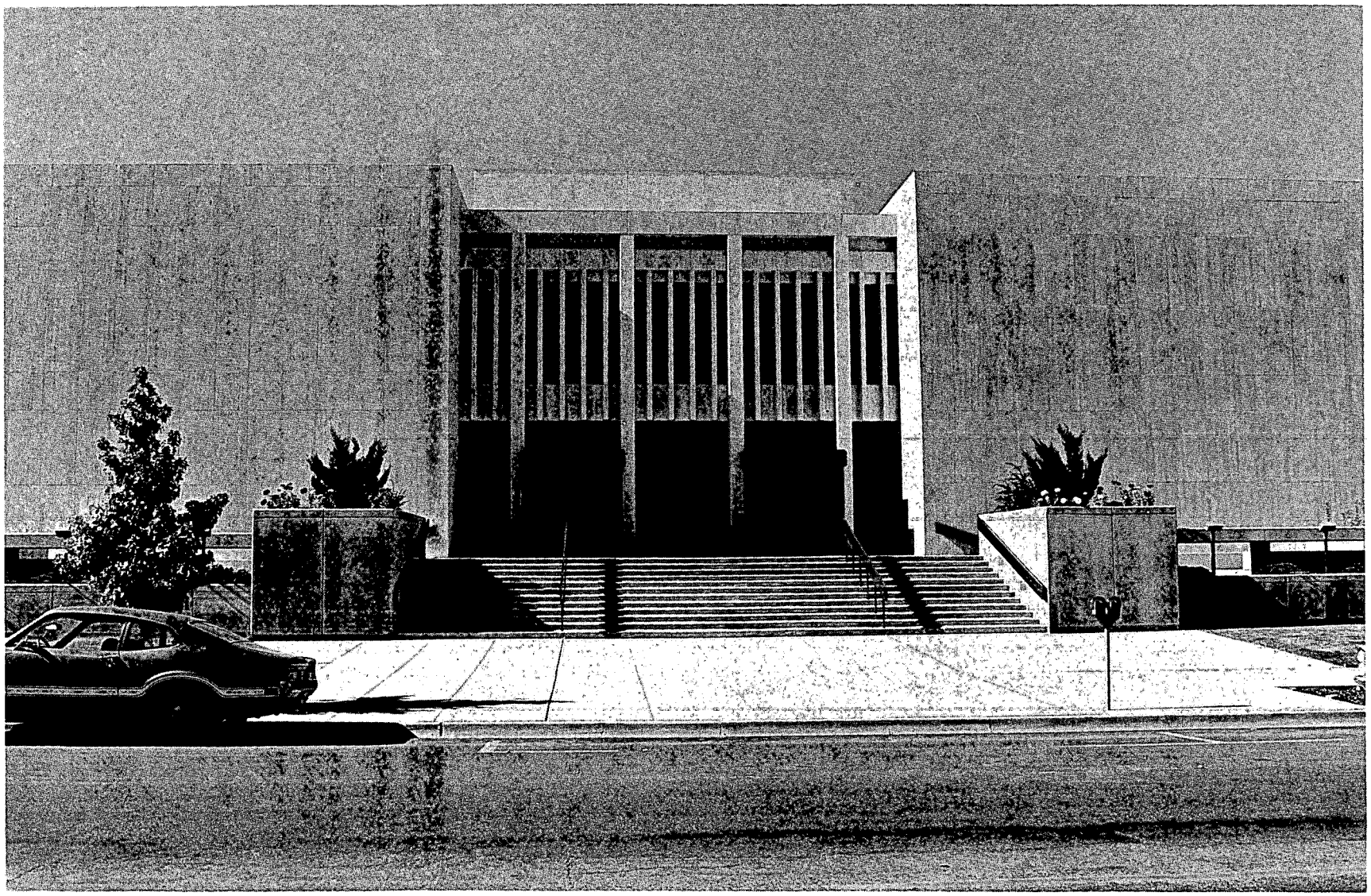


Figure 3.4-0 Idaho Supreme Court

3.4 Idaho Supreme Court

One large central plant type forced air HVAC unit located in the basement currently takes care of the bulk of the building. A small package multizone HVAC unit in the penthouse satisfies the Courtrooms and Judges Chambers. The large unit is powered with a 125 HP motor and is rated at 63,600 SCFM. The small unit uses a 15 HP motor and is rated at 15,000 SCFM. Both units lend themselves to conversion, however, the penthouse unit will require quite extensive sheet metal modification to the heating coil section. Steam coils are used presently in the heating section of both units and chilled water coils in the cooling sections.

Based on the calculation sheets, Tables 3.4-1 through 3.4-6, hot water coils with identical face areas as the existing steam coils can be installed that will duplicate the heating capacity ratings. The increased resistance to airflow as a result of added coils will increase the fan static pressure requirements for the 125 HP system by .26 inches of water. Adjustments to the fan speed to overcome additional resistance will result in only a 2% increase in horsepower requirements. A coil added to the 15 HP system will increase the static pressure required by .38 inches of water resulting in an additional 6% in horsepower.

From the information available, it appears that enough reserve capacity will be available in the existing drive motors. For the 125 HP unit a very minor speed adjustment is indicated with the new fan speed being 760 RPM, a 1.4% increase above original design. The 15 HP unit calls for a 15.1% increase in fan speed to 1000 RPM.

Total geothermal water flow to the building will be 150 gpm and could be supplied through 4 in. dia. supply and discharge mains. Reference Figures 3.4-1 through 3.4-5, Tables 3.4-1 through 3.4-6 for proposed modifications. The estimated building conversion cost in 1975 dollars without escalation or contingency is $23,200.

TABLE 3.4-1

SUMMARY

BUILDING Idaho Supreme Court

TOTAL MAXIMUM HOT WATER (GEOTHERMAL) REQUIRED FOR THIS BUILDING - 149.0 GALLONS PER MINUTE

TEMPERATURE OF WATER TO THE COILS ASSUMED TO BE 170°F AND THE TEMPERATURE OF WATER FROM THE COILS ASSUMED TO BE 120°F UNLESS NOTED OTHERWISE.

HOT WATER COIL CALCULATIONS, MODEL NUMBERS AND NOMENCLATURE USED ARE BASED ON Mc QUAY.

	PER AIR HANDLING SYSTEM								PER FIN TUBE HOT WATER COIL								
NO[1]	AIR[2] FLOW IN (SCFM)	AIR[3] DESIGN IN(°F) IN	AIR[3] DESIGN IN(°F) OUT	AIR[4] FLOW IN (SCFM)	AIR[5] FLOW (ΔP)	FAN[6] ΔHP	HEATING[7] CAPACITY IN (BTU/HR)	WATER FLOW IN (GPM)	FIN TUBE COIL MODEL NUMBER TYPE	FIN SERIES	TUBE ROWS "C"	SIZE (W X L X D) IN (INCHES)	PIPE[8] SIZE	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	WATER[10] FLOW (ΔP) (PSI)	COILS PER SYSTEM
1	63600	51	94	63600	.255	2.55	2976000	119.2	5WH	08	02	42x107.5x5	1-1/2	31.4	507	1.03	4
1	15000	51	97	15000	.380	.90	744000	29.8	5WH	08	03	21x107.5x6	1-1/2	15.7	955	3.70	1

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
4 DESIGN AIR FLOW THROUGH THE HEATING COIL SECTION UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
5 INCREASE IN AIR PRESSURE REQUIRED TO OVERCOME THE ADDITIONAL RESISTANCE OF THE HOT WATER COILS WHILE MAINTAINING THE DESIGN AIR FLOW UNITS - INCHES OF WATER
6 INCREASE IN FAN MOTOR HORSEPOWER REQUIRED TO PRODUCE THE ADDITIONAL AIR PRESSURE
7 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR
8 PIPE DIAMETER OF THE INLET AND OUTLET CONNECTIONS FOR THE INDIVIDUAL COILS UNITS - INCHES
9 AVERAGE VELOCITY OF THE AIR AT THE HOT WATER COIL UNITS - FEET PER MINUTE
10 PRESSURE LOSS OF THE HOT WATER AS IT FLOWS THROUGH EACH COIL UNITS - POUNDS PER SQUARE INCH

TABLE 3.4-2

BUILDING Idaho Supreme Court

HEATING SYSTEM INFORMATION DERIVED FROM CONSTRUCTION DRAWINGS, BUILDING SPECIFICATIONS AND FIELD DATA

	PER AIR HANDLING SYSTEM								PER FIN TUBE STEAM COIL							
NO[1]	AIR[2] FLOW IN (SCFM)	FANS[3]	FAN RPM[4] DESIGN / MEAS.	FAN HP[5] DESIGN / MEAS.	MOTOR[6] VOLTS PHASE	FAN[7] SP @ AIR FLOW	FAN[8] AIR VEL. FPM	MEASURED AIR FLOW IN (SCFM)	COILS PER SYSTEM	SIZE (W X L) IN INCHES	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	AIR[10] PER COIL (SCFM)	AIR[11] DESIGN IN(°F) IN	AIR[11] DESIGN IN(°F) OUT	HEATING[12] CAPACITY IN (BTU/HR)
1	63600	1	750 /	125 /	208/3	6	2100		4	42x107.5	31.4	507	15900	51	94	744000
1	15000	1	875 /	15 /	208/3	2.5	1500		1	21x107.5	15.7	955	15000	51	97	744000

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 NUMBER OF IDENTICAL FANS PER SYSTEM
4 FAN SPEED BOTH DESIGN AND THAT MEASURED AT INSTALLATION UNITS - REVOLUTIONS PER MINUTE
5 FAN DRIVE MOTOR HORSEPOWER BOTH DESIGN AND THAT MEASURED AT INSTALLATION
6 DRIVE MOTOR VOLTAGE AND PHASE; ALL VOLTAGE FREQUENCIES ARE 60 HERTZ
7 RATED STATIC AIR PRESSURE PRODUCED BY THE FAN AT THE DESIGN AIR FLOW UNITS - INCHES OF WATER
8 AVERAGE VELOCITY OF THE AIR AT THE FAN DISCHARGE UNITS - FEET PER MINUTE
9 AVERAGE VELOCITY OF THE AIR AT THE STEAM COIL UNITS - FEET PER MINUTE
10 DESIGN AIR FLOW PER STEAM COIL UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
11 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
12 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR

TABLE 3.4-3

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Idaho Supreme Court

RATING OF SYSTEM 63600 (SCFM) 2976000 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 42 X107.5/144 = 31.35 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 13400 / 31.35 = 507 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 43 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 744000 /(500)X(50) = 29.8 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WH TYPE COIL WITH 1,2,3,4 ROWS = 14

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 29.8 / 14 = 2.13

7) $\Delta T_A/\Delta T_{inlet}$ = 43 / 170 (°F) - 51 = 43 / 119 = .361

$\Delta T_W/\Delta T_A$ = 50 / 43 = 1.16

FROM FIGURE 7 : M_t = .675

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 2 /(.675X5.07) = .584

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}}$ / 2 = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .130

$R_{f2} = R_{ft} - R_{f1}$ = .584 - .130 = .454

FROM FIGURE 5 : FIN SERIES = 08 R_{f2} = .420

COIL MODEL NUMBER BECOMES: 5WH-0802C-42x107.5 NUMBER OF COILS REQ'D 4

9) FROM FIGURE 9 : ΔP_A = .255 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(63600)X(.255) = 2.55

10) FROM FIGURE 11 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(1.1 + 1.1) =

2.38 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 1.03 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.4-4

NEW FAN SPEED ESTIMATE SHEET

BUILDING Idaho Supreme Court

RATING OF SYSTEM 63600 (SCFM) 2976000 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$FAN\ BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(63600)X(6) = 59.91$

$FAN\ BHP_2 = (FAN\ BHP_1)+(FAN\ BHP_+) = (59.91)+(2.55) = 62.46$

$RPM_2 = (RPM_1)X(FAN\ BHP_2/FAN\ BHP_1)^{1/3} = (750)X(62.46/59.91)^{1/3} = 760$

CHANGE IN FAN SPEED = $(RPM_2 - RPM_1)/RPM_1$ = .014 OR X 100 = 1.4 % INCREASE

DEFINITIONS:

$FAN\ BHP_1$ = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

$FAN\ BHP_+$ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9,FROM THE COIL SELECTION SHEET).

$FAN\ BHP_2$ = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

TABLE 3.4-5

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Idaho Supreme Court

RATING OF SYSTEM 15000 (SCFM) 744000 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 21 X 107/144 = 15.7 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 15000 / 15.7 = 955 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 46 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 744000 /(500)X(50) = 29.8 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WH TYPE COIL WITH 1,2,3,4 ROWS = 7

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 29.8 / 7 = 4.26

7) $\Delta T_A/\Delta T_{inlet}$ = 46 / 170 (°F) - 51 = 46 / 119 = .387

$\Delta T_W/\Delta T_A$ = 50 / 46 = 1.09

FROM FIGURE 8 : M_t = .710

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 3 /(.71 X 9.55) = .442

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}}$ / 2 = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .090

$R_{f2} = R_{ft} - R_{f1}$ = .442 - .090 = .352

FROM FIGURE 6 : FIN SERIES = 08 R_{f2} = .300

COIL MODEL NUMBER BECOMES: 5WH-0803 NUMBER OF COILS REQ'D 1

9) FROM FIGURE 9 : ΔP_A = .380 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(15000)X(.380) = .895

10) FROM FIGURE 12 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(1.4 + 6.5) =

8.53 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 3.70 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.4-6

NEW FAN SPEED ESTIMATE SHEET

BUILDING Idaho Supreme Court

RATING OF SYSTEM 15000 (SCFM) 744000 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$FAN\ BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(15000)X(2.5) = 5.89$

$FAN\ BHP_2 = (FAN\ BHP_1)+(FAN\ BHP_+) = (5.89)+(.90) = 6.79$

$RPM_2 = (RPM_1)X(FAN\ BHP_2/FAN\ BHP_1)^{1/3} = (875)X(6.79/5.89)^{1/3} = 1007$

CHANGE IN FAN SPEED = $(RPM_2 - RPM_1)/RPM_1$ = .151 OR X 100 = 15.1 % INCREASE

DEFINITIONS:

$FAN\ BHP_1$ = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

$FAN\ BHP_+$ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9,FROM THE COIL SELECTION SHEET).

$FAN\ BHP_2$ = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

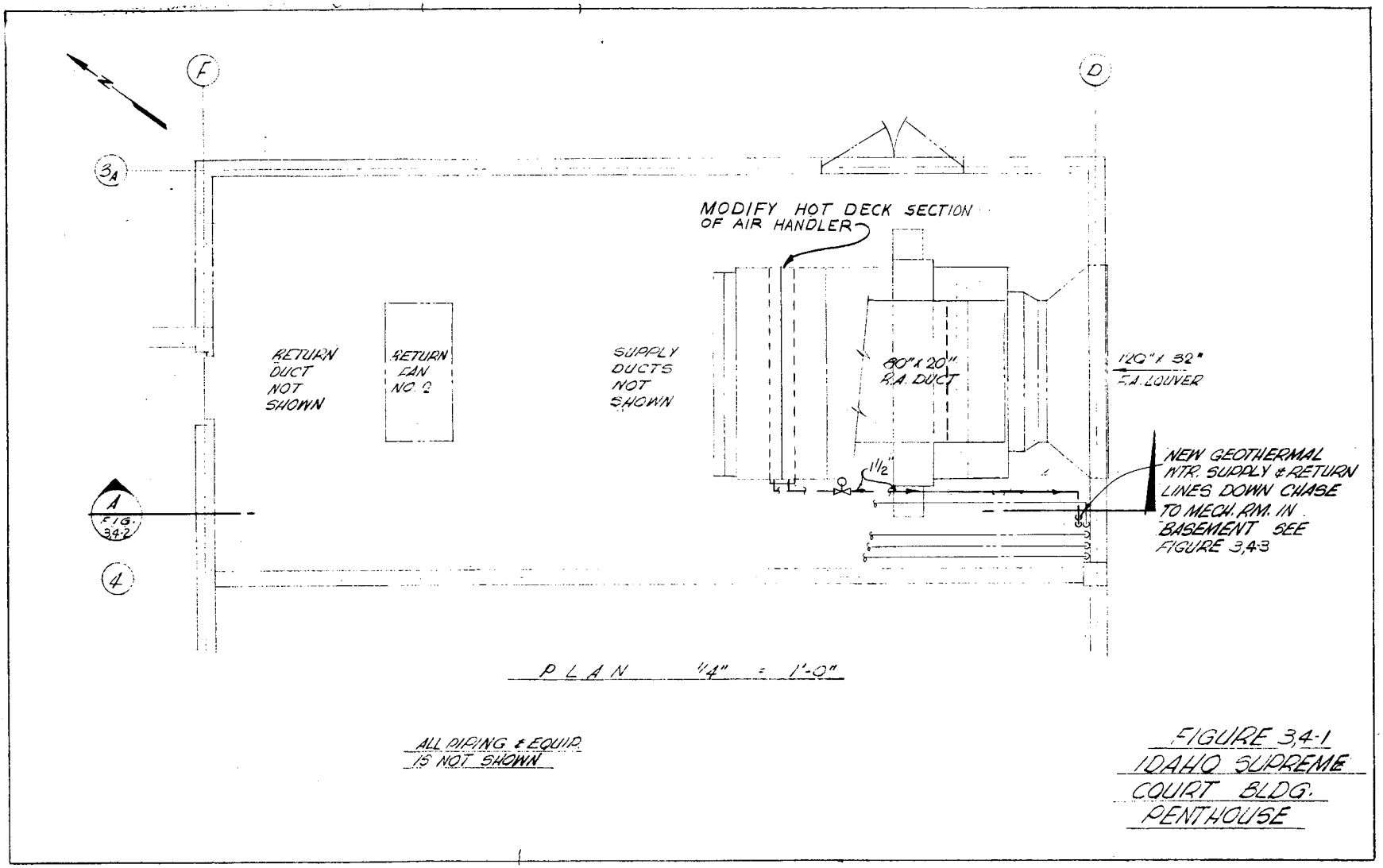


Figure 3.4-1 Idaho Supreme Court Building Penthouse

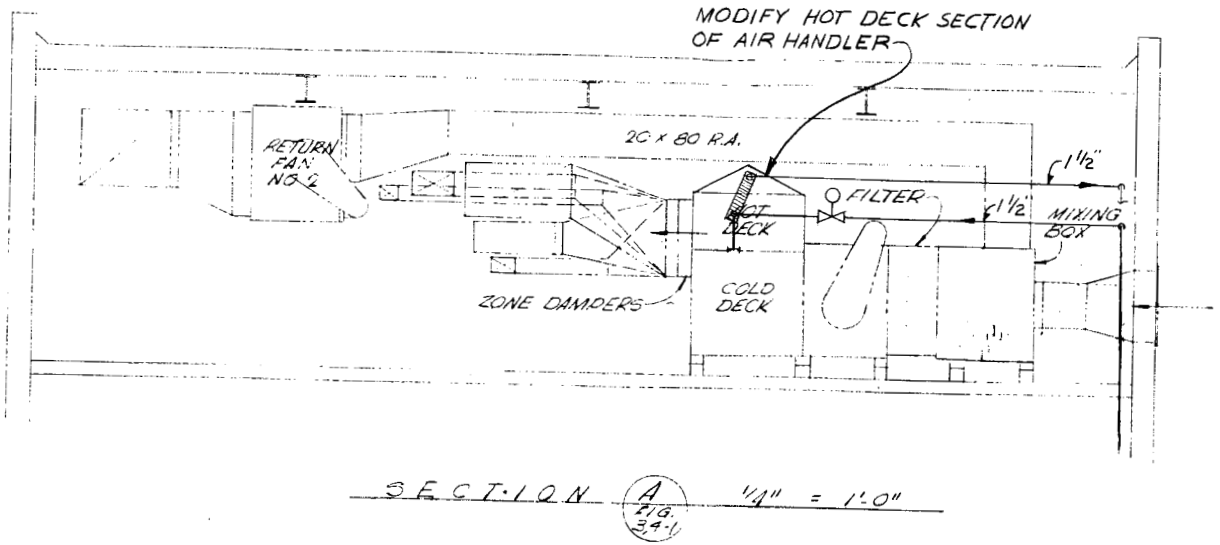


FIGURE 3.4-2
IDAHO SUPREME
COURT BLDG.
PENTHOUSE

Figure 3.4-2 Idaho Supreme Court Building Penthouse

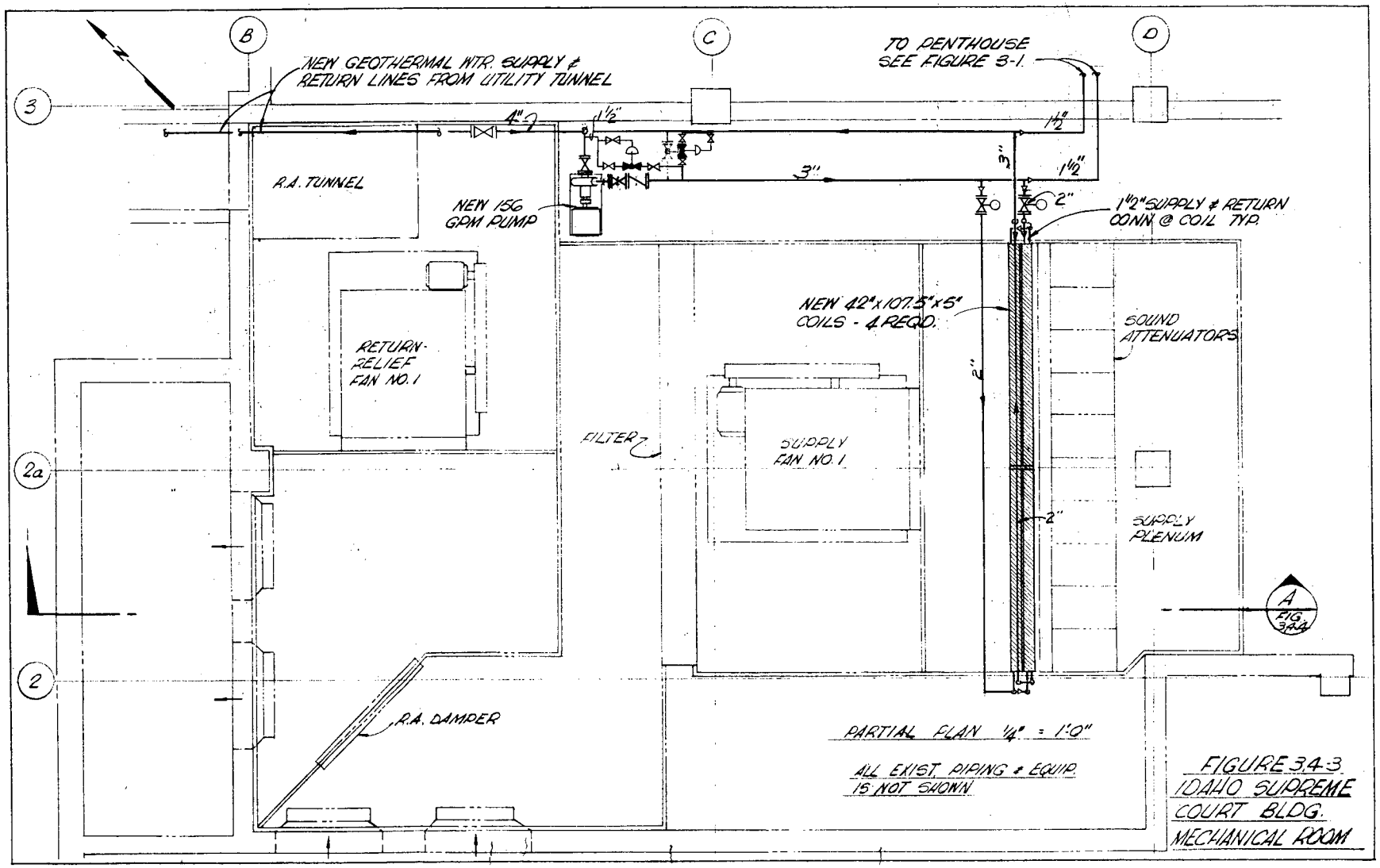


Figure 3.4-3 Idaho Supreme Court Building Mechanical Room

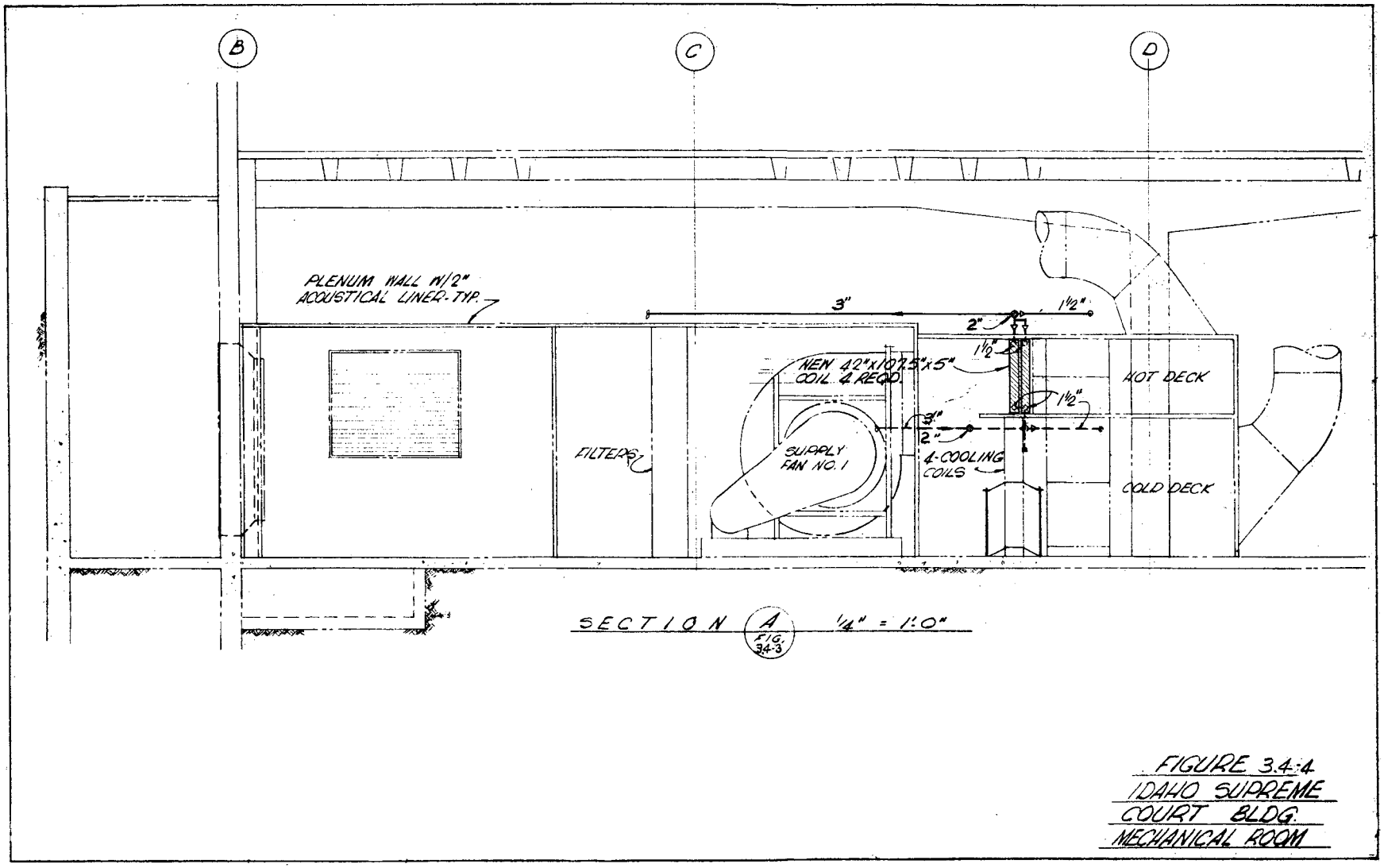
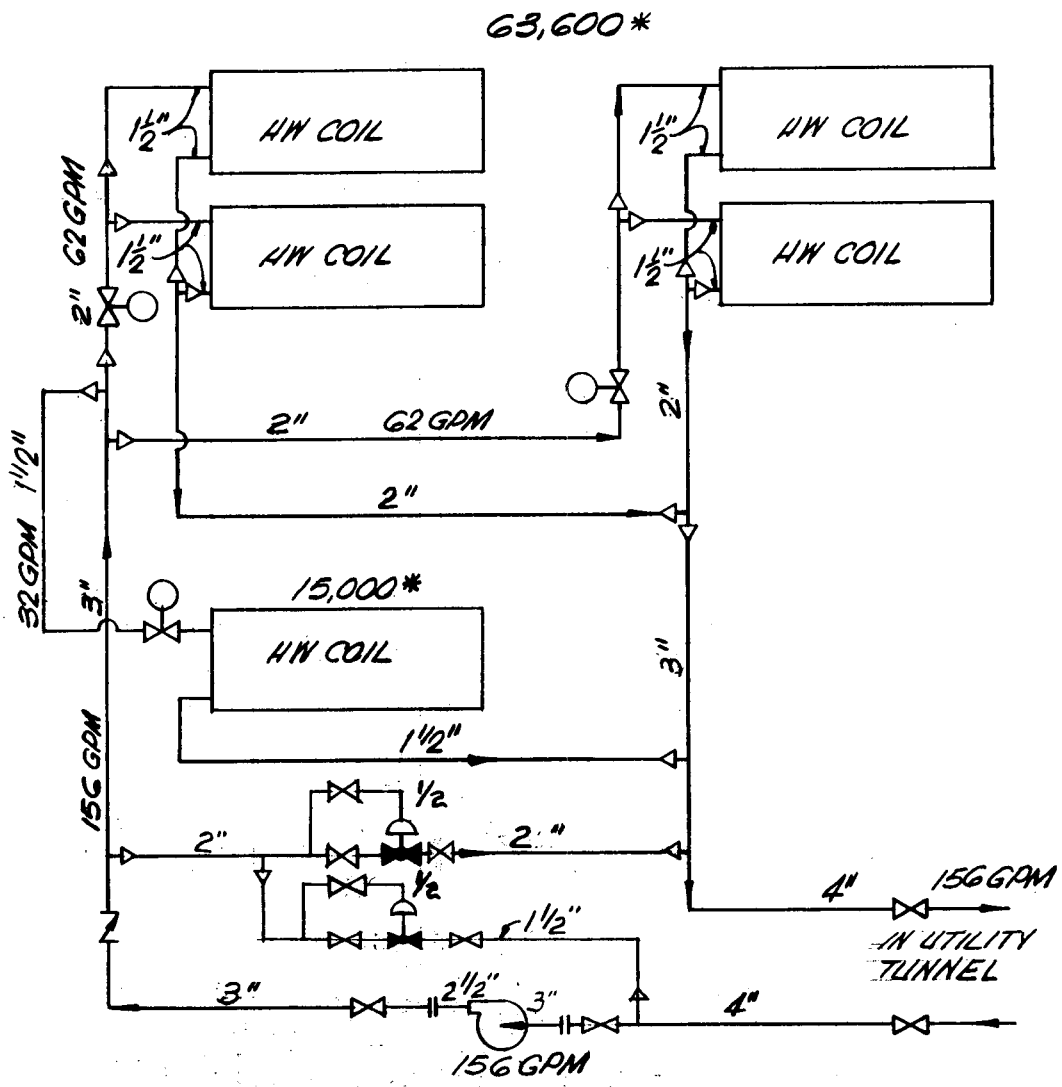


Figure 3.4-4 Idaho Supreme Court Building Mechanical Room



FLOWS SHOWN ARE CALCULATED VALUES PLUS 5%
*RATED AIR FLOW OF SYSTEM IN SCFM

FIGURE 3.4-5
IDAHO SUPREME COURT BLDG.

Figure 3.4-5 Idaho Supreme Court Building

Figure 3.5-0 Idaho State Library

3.5 Idaho State Library

One built-up central plant HVAC unit serves the entire building. This unit has a 30 HP fan motor and is rated at 30,600 SCFM. The air distribution system is of the dual, hot-cold, type resulting in a maximum air flow across the heating coils of only 27,000 SCFM. Steam and chilled water coils are used in the respective hot and cold sections of the system. Space is available to add the extra coils needed.

Based on Tables 3.5-1 through 3.5-4, identical face area hot water coils can be installed that will duplicate the present heating capacity. Resistance to airflow as a result of the added coils will increase the fan static pressure requirement by 1.2 inches of water. Fan speed adjustments to overcome this resistance will add a significant 17% to the motor horsepower requirement.

With a 17% increase in fan brake horsepower it appears that a motor change will be required. Fan speed will have to be increased by 10.3% to 953 RPM.

Maximum geothermal water flow requirements will approach 180 gpm and the building supply and discharge lines sized at 4 in. dia. Reference Figures 3.5-1 through 3.5-3 for total proposed modifications. The estimated conversion cost for this building in 1975 dollars without escalation or contingency is $15,700.

TABLE 3.5-1

SUMMARY

BUILDING Idaho State Library

TOTAL MAXIMUM HOT WATER (GEOTHERMAL) REQUIRED FOR THIS BUILDING - 170 GALLONS PER MINUTE

TEMPERATURE OF WATER TO THE COILS ASSUMED TO BE 170°F AND THE TEMPERATURE OF WATER FROM THE COILS ASSUMED TO BE 120°F UNLESS NOTED OTHERWISE.

HOT WATER COIL CALCULATIONS, MODEL NUMBERS AND NOMENCLATURE USED ARE BASED ON Mc QUAY.

	PER AIR HANDLING SYSTEM								PER FIN TUBE HOT WATER COIL								
NO[1]	AIR[2] FLOW IN (SCFM)	AIR[3] DESIGN IN(°F) IN	OUT	AIR[4] FLOW IN (SCFM)	AIR[5] FLOW (ΔP)	FAN[6] ΔHP	HEATING[7] CAPACITY IN (BTU/HR)	WATER FLOW IN (GPM)	FIN TUBE COIL MODEL NUMBER TYPE	FIN SERIES	TUBE ROWS "C"	SIZE (W X L X D) IN (INCHES)	PIPE[8] SIZE	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	WATER[10] FLOW (ΔP) (PSI)	COILS PER SYSTEM
1	32600	50	128	27000	1.20	5.1	2550000	170	5WM	12	04	24x114x8-3/8	2-1/2	19	710	1.92	2

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
4 DESIGN AIR FLOW THROUGH THE HEATING COIL SECTION UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
5 INCREASE IN AIR PRESSURE REQUIRED TO OVERCOME THE ADDITIONAL RESISTANCE OF THE HOT WATER COILS WHILE MAINTAINING THE DESIGN AIR FLOW UNITS - INCHES OF WATER
6 INCREASE IN FAN MOTOR HORSEPOWER REQUIRED TO PRODUCE THE ADDITIONAL AIR PRESSURE
7 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR
8 PIPE DIAMETER OF THE INLET AND OUTLET CONNECTIONS FOR THE INDIVIDUAL COILS UNITS - INCHES
9 AVERAGE VELOCITY OF THE AIR AT THE HOT WATER COIL UNITS - FEET PER MINUTE
10 PRESSURE LOSS OF THE HOT WATER AS IT FLOWS THROUGH EACH COIL UNITS - POUNDS PER SQUARE INCH

TABLE 3.5-2

BUILDING Idaho State Library

HEATING SYSTEM INFORMATION DERIVED FROM CONSTRUCTION DRAWINGS, BUILDING SPECIFICATIONS AND FIELD DATA

	PER AIR HANDLING SYSTEM								PER FIN TUBE STEAM COIL							
NO[1]	AIR[2] FLOW IN (SCFM)	FANS[3]	FAN RPM[4] DESIGN / MEAS.	FAN HP[5] DESIGN / MEAS.	MOTOR[6] VOLTS PHASE	FAN[7] SP @ AIR FLOW	FAN[8] AIR VEL. FPM	MEASURED AIR FLOW IN (SCFM)	COILS PER SYSTEM	SIZE (W X L) IN INCHES	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	AIR[10] PER COIL (SCFM)	AIR[11] DESIGN IN(°F) IN	OUT	HEATING[12] CAPACITY IN (BTU/HR)
1	32600	1	865	30	460/3	3.5	2000		4	24x114	19	710	13500	50	128	2550000

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 NUMBER OF IDENTICAL FANS PER SYSTEM
4 FAN SPEED BOTH DESIGN AND THAT MEASURED AT INSTALLATION UNITS - REVOLUTIONS PER MINUTE
5 FAN DRIVE MOTOR HORSEPOWER BOTH DESIGN AND THAT MEASURED AT INSTALLATION
6 DRIVE MOTOR VOLTAGE AND PHASE; ALL VOLTAGE FREQUENCIES ARE 60 HERTZ
7 RATED STATIC AIR PRESSURE PRODUCED BY THE FAN AT THE DESIGN AIR FLOW UNITS - INCHES OF WATER
8 AVERAGE VELOCITY OF THE AIR AT THE FAN DISCHARGE UNITS - FEET PER MINUTE
9 AVERAGE VELOCITY OF THE AIR AT THE STEAM COIL UNITS - FEET PER MINUTE
10 DESIGN AIR FLOW PER STEAM COIL UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
11 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
12 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR

TABLE 3.5-3

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Idaho State Library

RATING OF SYSTEM 32600 (SCFM) 2550000 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 24 X 114 /144 = 19 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 13500 / 19 = 710 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 78 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 1275000 /(500)X(50) = 85 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WM TYPE COIL WITH 4 ROWS = 24

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 85 / 24 = 3.54

7) $\Delta T_A/\Delta T_{inlet}$ = 78 / 170 (°F) - 50 = 78 / 120 = .650

$\Delta T_W/\Delta T_A$ = 50 / 78 = .641

FROM FIGURE 8 : M_t = 1.56

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 4 /(1.56 X 7.1) = .361

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}}$ / 2 = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .095

$R_{f2} = R_{ft} - R_{f1}$ = .361 - .095 = .266

FROM FIGURE 6 : FIN SERIES = 12 R_{f2} = .230

COIL MODEL NUMBER BECOMES: 5WM-1204C-24x114 NUMBER OF COILS REQ'D 2

9) FROM FIGURE 9 : ΔP_A = 1.2 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(27000)X(1.2) = 5.09

10) FROM FIGURE 12: CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(1.9 + 2.2) =

4.43 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 1.92 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.5-4

NEW FAN SPEED ESTIMATE SHEET

BUILDING Idaho State Library

RATING OF SYSTEM 32600 (SCFM) 2550000 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$FAN\ BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(\ 27000\ )X(\ 3.5\ ) =$ 14.84

$FAN\ BHP_2 = (FAN\ BHP_1)+(FAN\ BHP_+) = (\ 14.84\ )+(\ 5.09\ ) =$ 19.93

$RPM_2 = (RPM_1)X(FAN\ BHP_2/FAN\ BHP_1)^{1/3} = (\ 865\ )X(\ 19.93/\ 14.84\ )^{1/3} =$ 954

CHANGE IN FAN SPEED = $(RPM_2 - RPM_1)/RPM_1 =$ .103 OR X 100 = 10.3 % INCREASE

DEFINITIONS:

$FAN\ BHP_1$ = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

$FAN\ BHP_+$ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9, FROM THE COIL SELECTION SHEET).

$FAN\ BHP_2$ = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

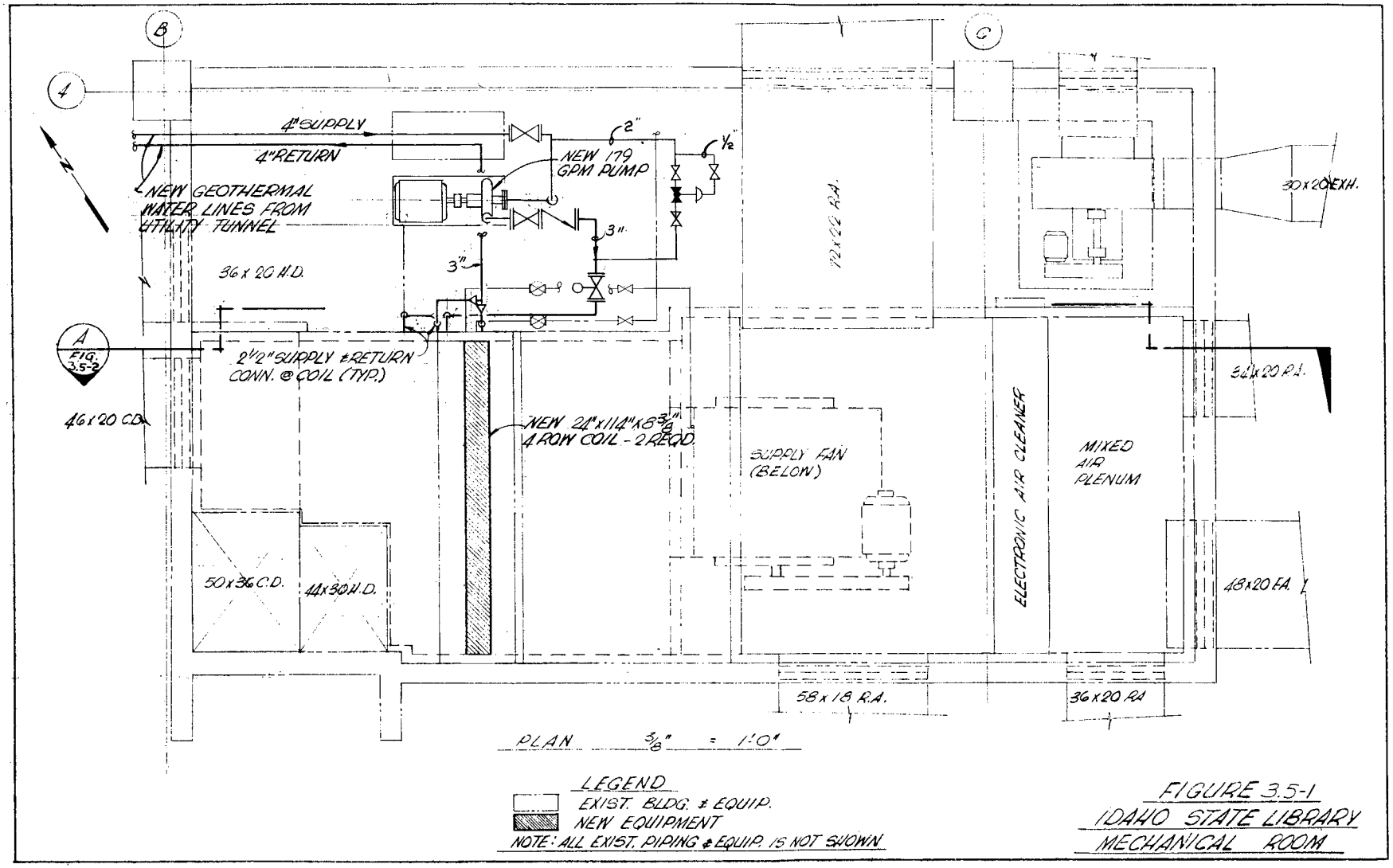


Figure 3.5-1 Idaho State Library Mechanical Room

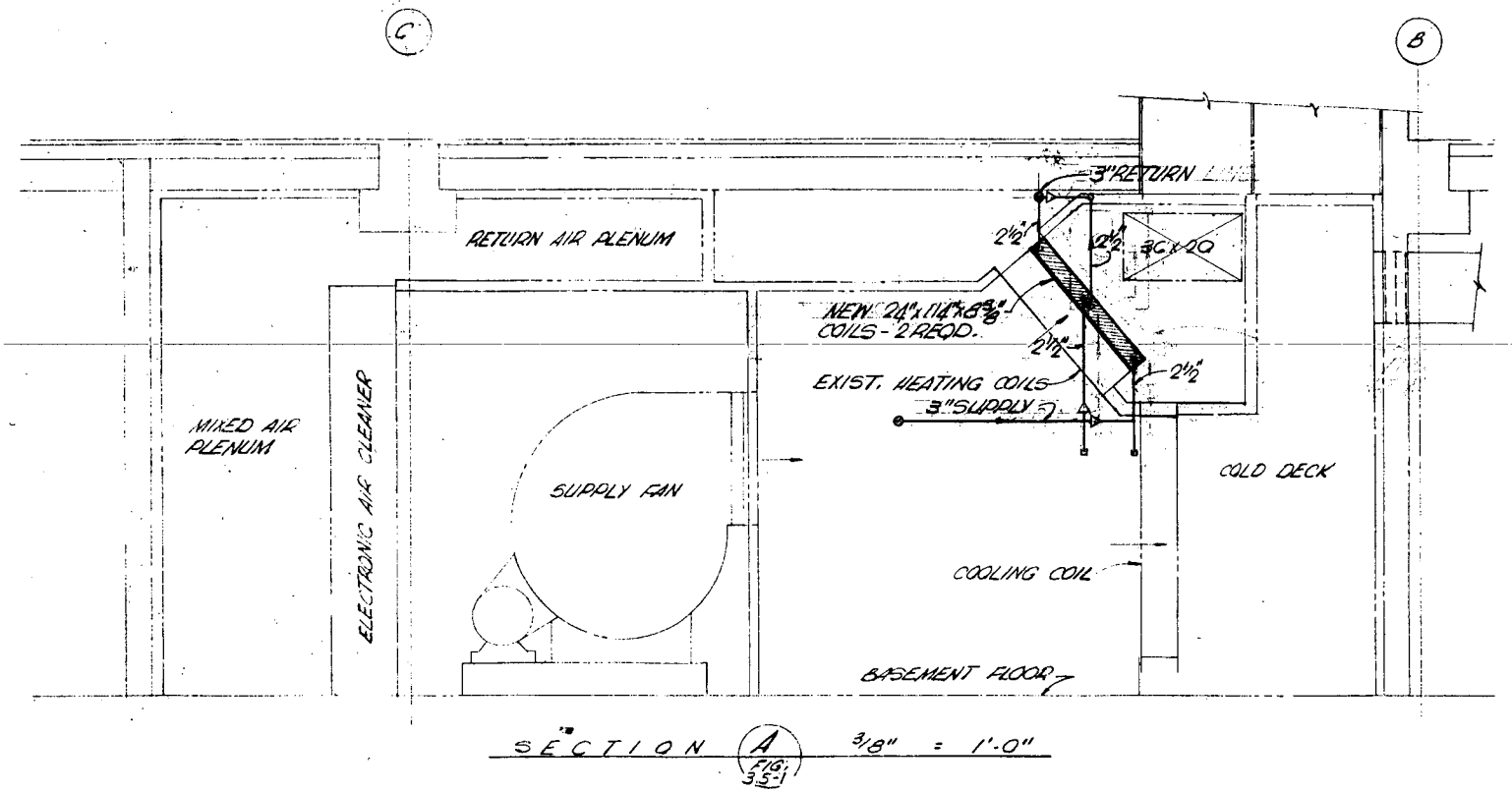
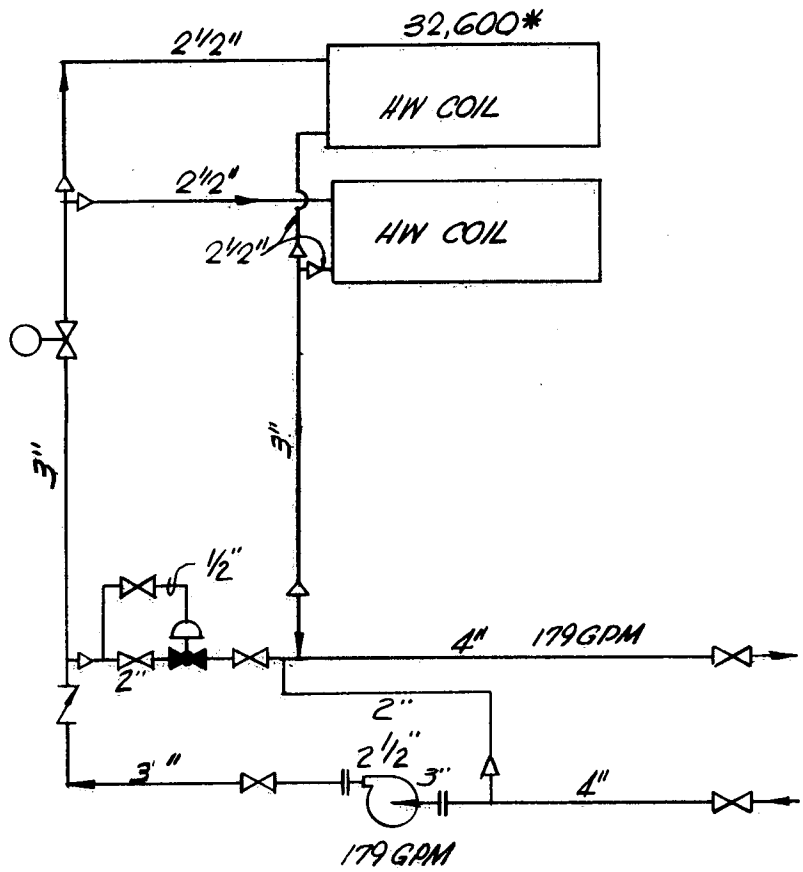


FIGURE 3.5-2
IDAHO STATE LIBRARY
MECHANICAL ROOM

Figure 3.5-2 Idaho State Library Mechanical Room



FLOWS SHOWN ARE CALCULATED VALUES PLUS 5%
* RATED AIR FLOW OF SYSTEM IN SCFM

FIGURE 3.5-3
IDAHO STATE LIBRARY

Figure 3.5-3 Idaho State Library

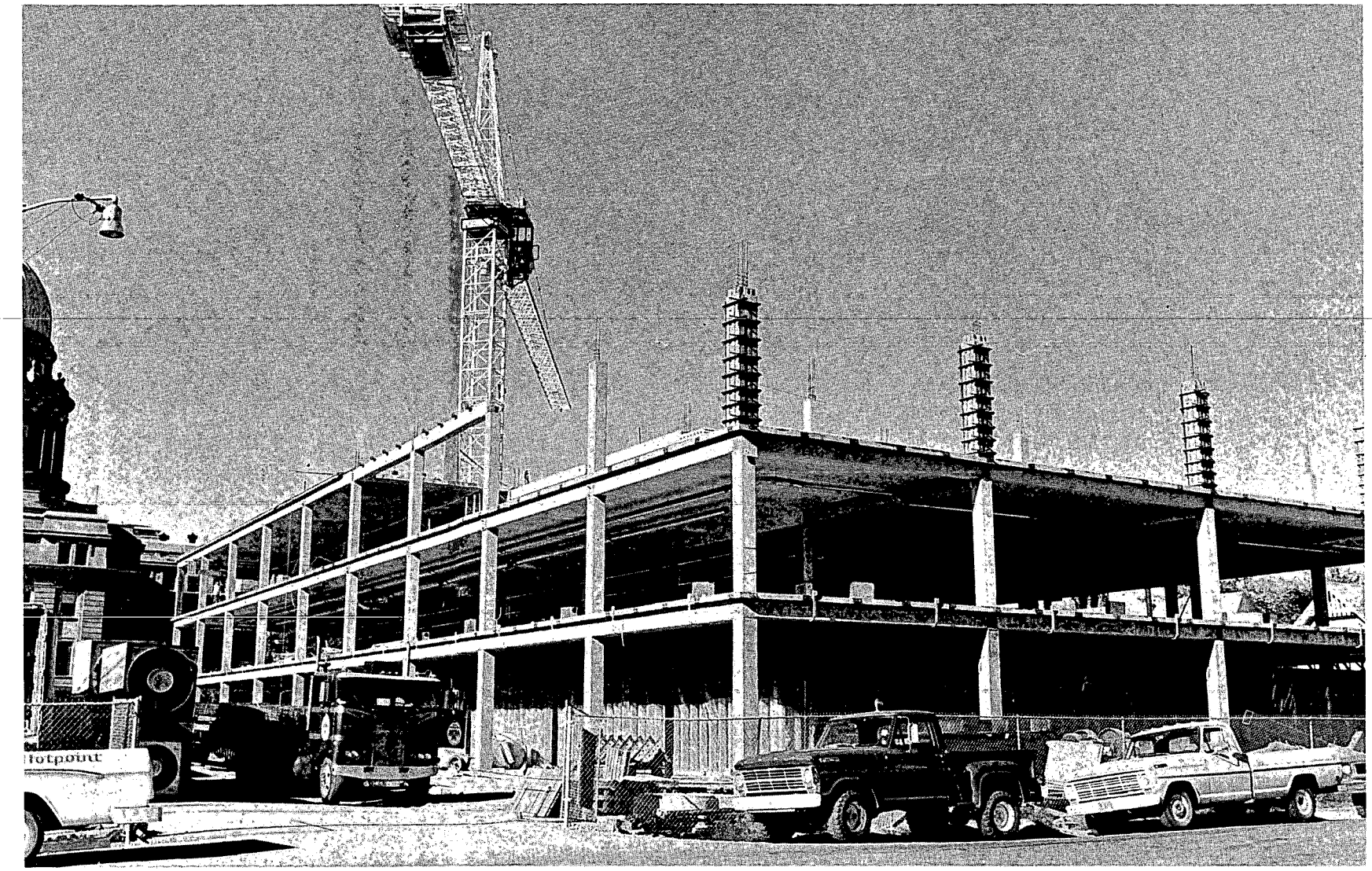


Figure 3.6-O New State Office Building

3.6 New State Office Building

This building, which is currently under construction, will utilize one forced air multizone HVAC unit for perimeter heating and cooling with the interior portions of the building relying on electrical energy for heat. Steam and chilled water coils are planned for the multizone unit. A 40 HP motor will drive the fan.

As shown in Tables 3.6-1 through 3.6-4, hot water coils with the same face areas can be installed along with the steam coils. Both coil sets will have identical heat capacity ratings. These additional coils will add resistance equal to .53 inches of water to the static pressure requirements for the fan. A fan horsepower increase of 5% is required which doesn't appear to warrant a change in motor. A fan speed increase of 2.9% to 970 RPM can probably be accomplished within the adjustable range of the motor drive pulley.

Reference Figures 3.6-1 through 3.6-3 along with the Tables for total proposed modifications. The geothermal flow requirement will be approximately 120 gpm for maximum heating conditions. This building will be served, then, with 3-1/2 in. dia. supply and discharge piping. The estimated building conversion cost in 1975 dollars without escalation or contingency is $15,800.

TABLE 3.6-1

SUMMARY

BUILDING New State Office

TOTAL MAXIMUM HOT WATER (GEOTHERMAL) REQUIRED FOR THIS BUILDING - 120* GALLONS PER MINUTE

TEMPERATURE OF WATER TO THE COILS ASSUMED TO BE 170°F AND THE TEMPERATURE OF WATER FROM THE COILS ASSUMED TO BE 120°F UNLESS NOTED OTHERWISE.

HOT WATER COIL CALCULATIONS, MODEL NUMBERS AND NOMENCLATURE USED ARE BASED ON Mc QUAY.

	PER AIR HANDLING SYSTEM								PER FIN TUBE HOT WATER COIL								
NO[1]	AIR[2] FLOW IN (SCFM)	AIR[3] DESIGN IN(°F) IN	OUT	AIR[4] FLOW IN (SCFM)	AIR[5] FLOW (ΔP)	FAN[6] ΔHP	HEATING[7] CAPACITY IN (BTU/HR)	WATER FLOW IN (GPM)	FIN TUBE COIL MODEL NUMBER TYPE	FIN SERIES	TUBE ROWS "C"	SIZE (W X L X D) IN (INCHES)	PIPE[8] SIZE	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	WATER[10] FLOW (ΔP) (PSI)	COILS PER SYSTEM
1	25220	70	139	24155	.53	2.01	1800000	120*	5WH	14	03	36x66x6	1-1/2	16.5	488	2.32	3

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
4 DESIGN AIR FLOW THROUGH THE HEATING COIL SECTION UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
5 INCREASE IN AIR PRESSURE REQUIRED TO OVERCOME THE ADDITIONAL RESISTANCE OF THE HOT WATER COILS WHILE MAINTAINING THE DESIGN AIR FLOW UNITS - INCHES OF WATER
6 INCREASE IN FAN MOTOR HORSEPOWER REQUIRED TO PRODUCE THE ADDITIONAL AIR PRESSURE
7 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR
8 PIPE DIAMETER OF THE INLET AND OUTLET CONNECTIONS FOR THE INDIVIDUAL COILS UNITS - INCHES
9 AVERAGE VELOCITY OF THE AIR AT THE HOT WATER COIL UNITS - FEET PER MINUTE
10 PRESSURE LOSS OF THE HOT WATER AS IT FLOWS THROUGH EACH COIL UNITS - POUNDS PER SQUARE INCH

* Flow Based on 170°F Inlet and 140°F Outlet.

TABLE 3.6-2

BUILDING New State Office

HEATING SYSTEM INFORMATION DERIVED FROM CONSTRUCTION DRAWINGS, BUILDING SPECIFICATIONS AND FIELD DATA

	PER AIR HANDLING SYSTEM								PER FIN TUBE STEAM COIL							
NO[1]	AIR[2] FLOW IN (SCFM)	FANS[3]	FAN RPM[4] DESIGN / MEAS.	FAN HP[5] DESIGN / MEAS.	MOTOR[6] VOLTS PHASE	FAN[7] SP @ AIR FLOW	FAN[8] AIR VEL. FPM	MEASURED AIR FLOW IN (SCFM)	COILS PER SYSTEM	SIZE (W X L) IN INCHES	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	AIR[10] PER COIL (SCFM)	AIR[11] DESIGN IN(°F)	AIR[11] DESIGN OUT(°F)	HEATING[12] CAPACITY IN (BTU/HR)
1	25220	1	940	40	480/3	6			3	36x66	16.5	488	8051	70	139	600000

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 NUMBER OF IDENTICAL FANS PER SYSTEM
4 FAN SPEED BOTH DESIGN AND THAT MEASURED AT INSTALLATION UNITS - REVOLUTIONS PER MINUTE
5 FAN DRIVE MOTOR HORSEPOWER BOTH DESIGN AND THAT MEASURED AT INSTALLATION
6 DRIVE MOTOR VOLTAGE AND PHASE; ALL VOLTAGE FREQUENCIES ARE 60 HERTZ
7 RATED STATIC AIR PRESSURE PRODUCED BY THE FAN AT THE DESIGN AIR FLOW UNITS - INCHES OF WATER
8 AVERAGE VELOCITY OF THE AIR AT THE FAN DISCHARGE UNITS - FEET PER MINUTE
9 AVERAGE VELOCITY OF THE AIR AT THE STEAM COIL UNITS - FEET PER MINUTE
10 DESIGN AIR FLOW PER STEAM COIL UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
11 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
12 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR

TABLE 3.6-3

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING New State Office

RATING OF SYSTEM 25220 (SCFM) 1800000 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 36 X 66/144 = 16.5 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 8051 / 16.5 = 488 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 69 (°F) 4) ASSUME ΔT_W = 30 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 600000 /(500)X(30) = 40 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WH TYPE COIL WITH 1, 2,3,4ROWS = 12

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 40 / 12 = 3.33

7) $\Delta T_A/\Delta T_{inlet}$ = 69 / 170 (°F) - 70 = 69 / 100 = .690

$\Delta T_W/\Delta T_A$ = 30 / 69 = .435

FROM FIGURE 8 : M_t = 1.58

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 3 /(1.58 X 4.88) = .389

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}}$ / 2 = 170 (°F) + 140 / 2 = 155 (°F)

FROM FIGURE 3 : R_{f1} = .11

$R_{f2} = R_{ft} - R_{f1}$ = .389 - .11 = .279

FROM FIGURE 6 : FIN SERIES = 14 R_{f2} = .240

COIL MODEL NUMBER BECOMES: 5WH-1403C-36x66 NUMBER OF COILS REQ'D 3

9) FROM FIGURE 9 : ΔP_A = .53 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(24155)X(.53) = 2.01

10) FROM FIGURE 12 : CORRECTION FACTOR = 1.05

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.05)X(2.1 + 3.0) =

5.36 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 2.32 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.6-4

NEW FAN SPEED ESTIMATE SHEET

BUILDING New State Office

RATING OF SYSTEM 25220 (SCFM) 1800000 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$FAN\ BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(24155)X(6.0) = 22.75$

$FAN\ BHP_2 = (FAN\ BHP_1)+(FAN\ BHP_+) = (22.75)+(2.01) = 24.76$

$RPM_2 = (RPM_1)X(FAN\ BHP_2/FAN\ BHP_1)^{1/3} = (940)X(24.76/22.75)^{1/3} = 967$

CHANGE IN FAN SPEED = $(RPM_2 - RPM_1)/RPM_1$ = .029 OR X 100 = 2.9 % INCREASE

DEFINITIONS:

$FAN\ BHP_1$ = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

$FAN\ BHP_+$ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9,FROM THE COIL SELECTION SHEET).

$FAN\ BHP_2$ = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

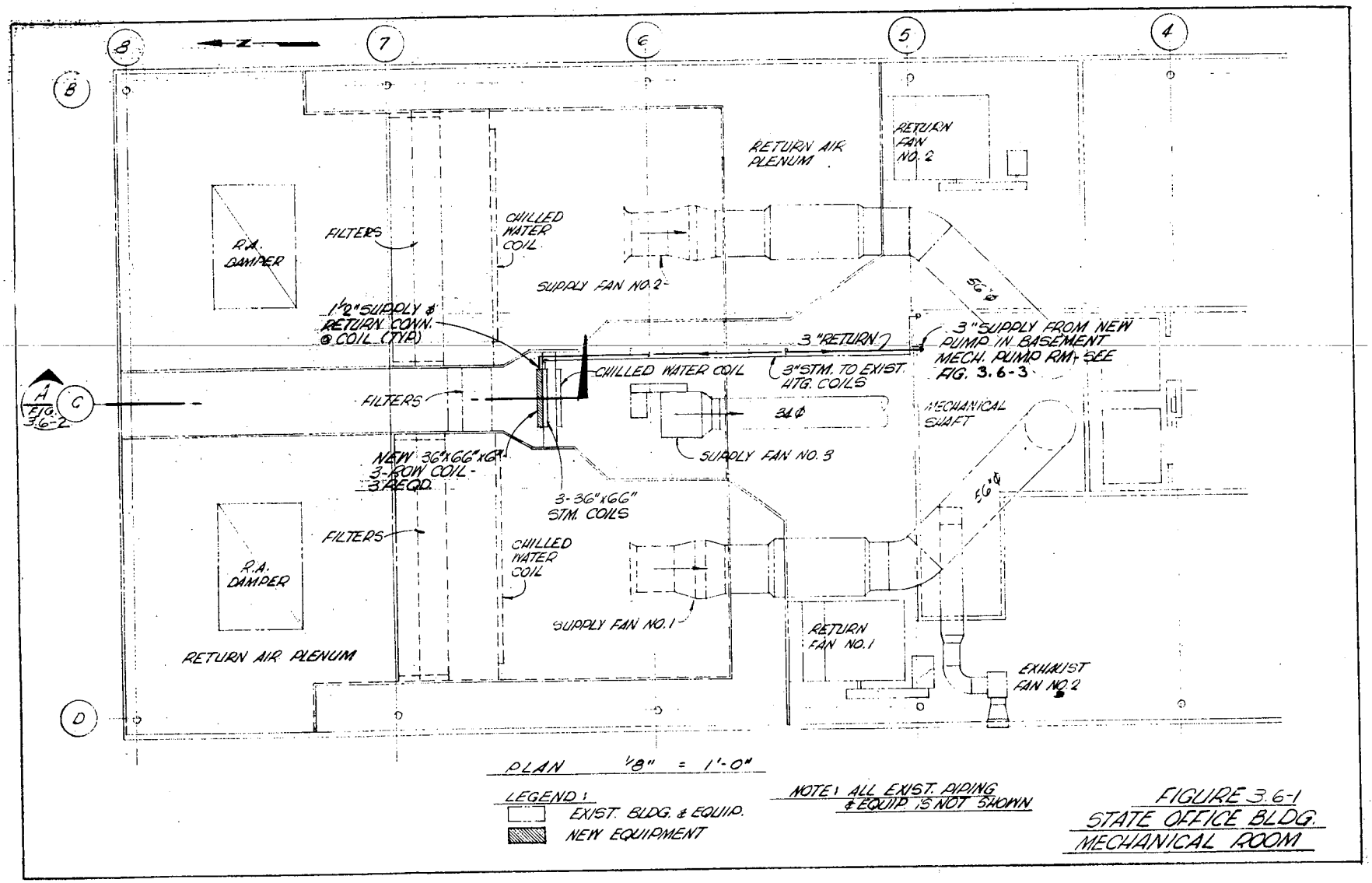


Figure 3.6-1 State Office Building Mechanical Room

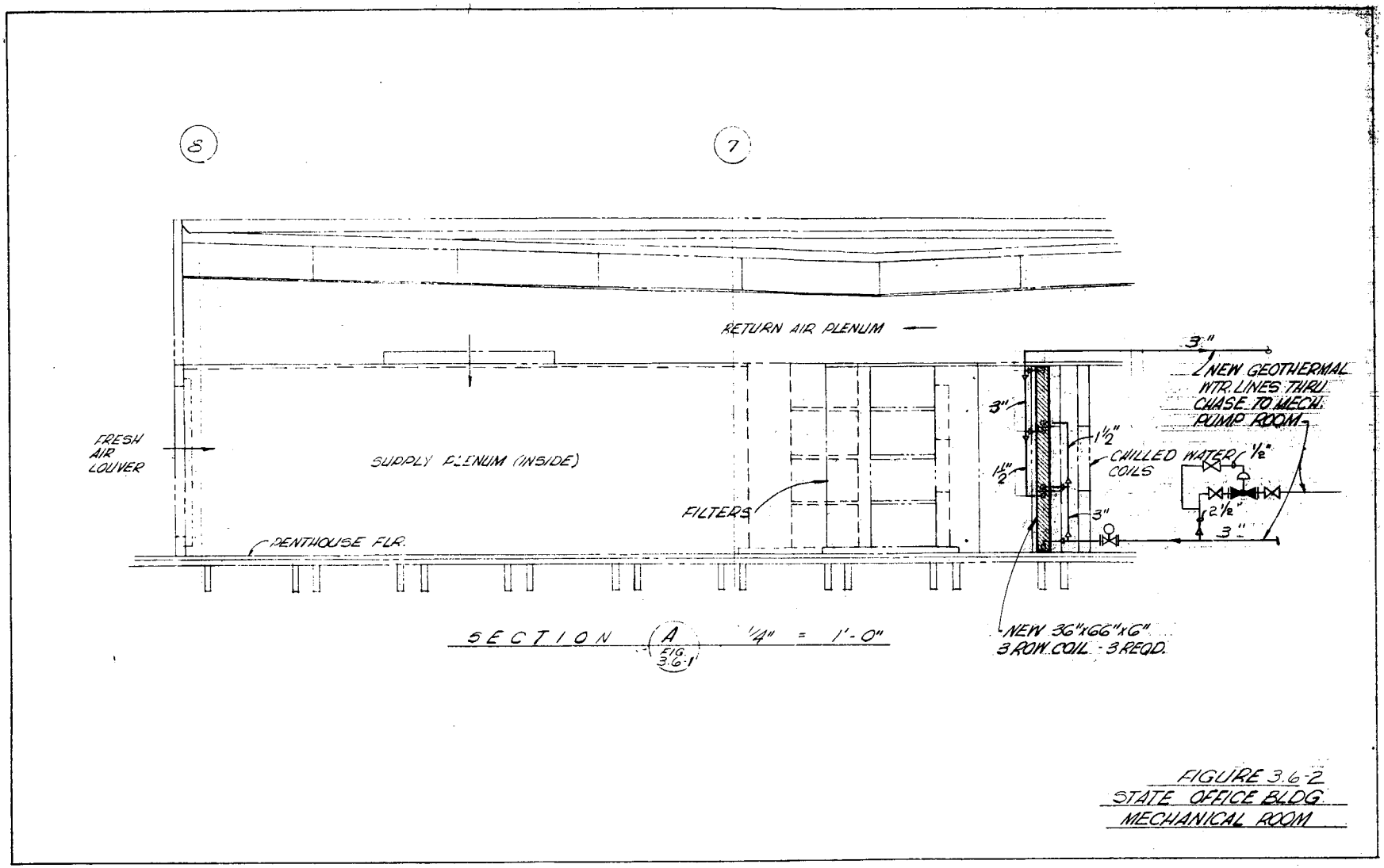
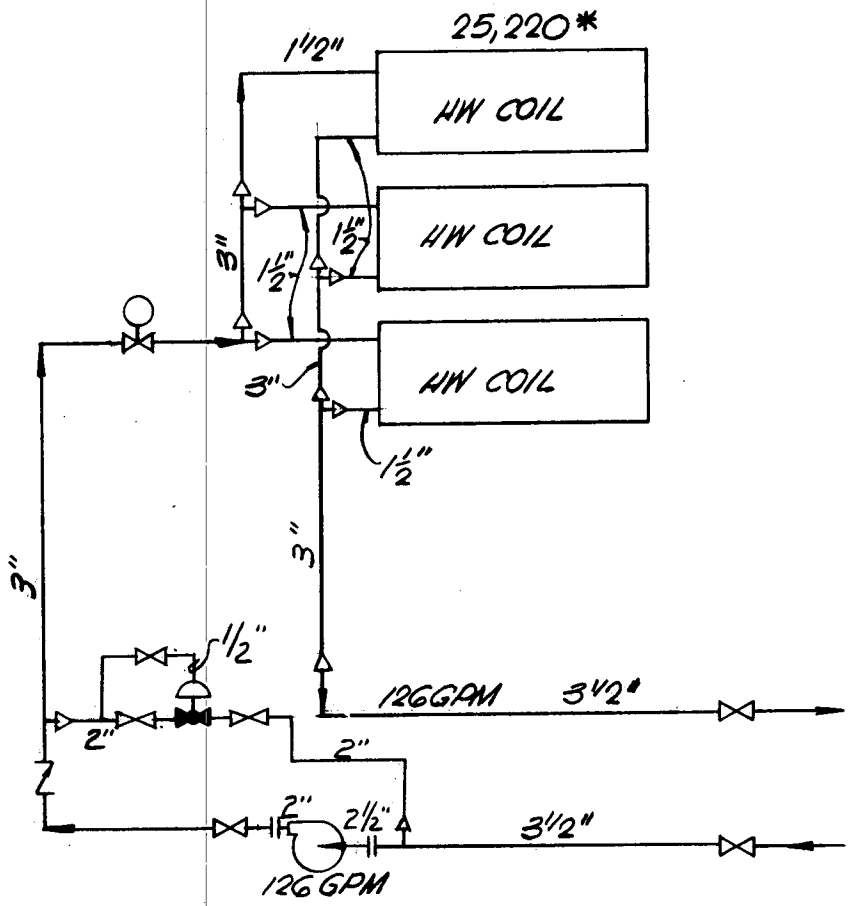


Figure 3.6-2 State Office Building Mechanical Room



FLOWS SHOWN ARE CALCULATED VALUES PLUS 5%
*RATED AIR FLOW OF SYSTEM IN SCFM

FIGURE 3.6-3
STATE OFFICE BLDG.

Figure 3.6-3 State Office Building

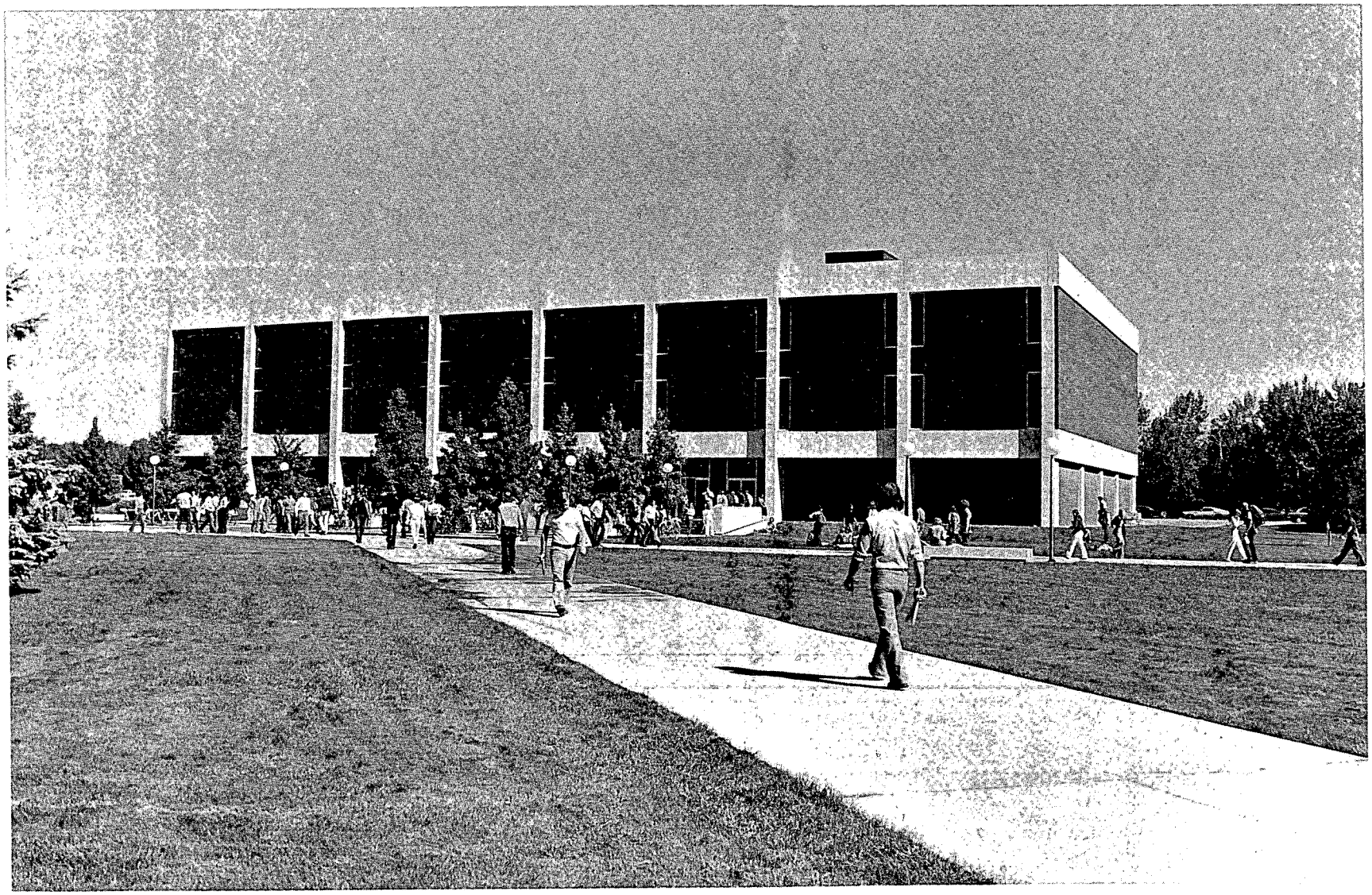


Figure 3.7-0 School of Business - BSU

3.7 School of Business - BSU

Two central plant HVAC units serve the building. Both systems use 75 HP fan motors and are rated at 37,550 SCFM each. Steam and chilled water coils provide respectively, heating and cooling. From Tables 3.7-1 through 3.7-4, the addition of hot water coils with the same face area as the steam can be installed that will duplicate the heating capacity already present. The additional coils will add .83 inches of water pressure to the fan static pressure requirement, and 6.5% to the horse power required.

From the data obtained it appears that fan speed adjustments to 1045 RPM, a 4.4% increase, will not require a change in drive motors. Also, from the balance reports, these systems are now operating above 1045 RPM which indicates a coil face velocity higher than 1000 FPM. It is recommended that this coil face velocity not exceed 1000 FPM to keep air turbulence and fan static pressure within more favorable limits.

Reference Figures 3.7-1 through 3.7-3 for total modifications proposed. Geothermal water flow to the building will approach 300 gpm under maximum heating conditions. Geothermal supply and discharge mains serving the building will be 5 in. dia. pipe. The estimated building conversion cost in 1975 dollars unescalated and without contingency is $22,500.

TABLE 3.7-1

SUMMARY

BUILDING School of Business - BSU

TOTAL MAXIMUM HOT WATER (GEOTHERMAL) REQUIRED FOR THIS BUILDING - 298.4 GALLONS PER MINUTE

TEMPERATURE OF WATER TO THE COILS ASSUMED TO BE 170°F AND THE TEMPERATURE OF WATER FROM THE COILS ASSUMED TO BE 120°F UNLESS NOTED OTHERWISE.

HOT WATER COIL CALCULATIONS, MODEL NUMBERS AND NOMENCLATURE USED ARE BASED ON Mc QUAY.

	PER AIR HANDLING SYSTEM								PER FIN TUBE HOT WATER COIL								
NO[1]	AIR[2] FLOW IN (SCFM)	AIR[3] DESIGN IN(°F) IN	AIR[3] DESIGN IN(°F) OUT	AIR[4] FLOW IN (SCFM)	AIR[5] FLOW (ΔP)	FAN[6] ΔHP	HEATING[7] CAPACITY IN (BTU/HR)	WATER FLOW IN (GPM)	FIN TUBE COIL MODEL NUMBER TYPE	FIN SERIES	TUBE ROWS "C"	SIZE (W X L X D) IN (INCHES)	PIPE[8] SIZE	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	WATER[10] FLOW (ΔP) (PSI)	COILS PER SYSTEM
2	37550	55	101	37550	.83	4.89	3730968	149.2	5WS	12	03	24x120x6	1-1/2	20	939	2.43	2

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
4 DESIGN AIR FLOW THROUGH THE HEATING COIL SECTION UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
5 INCREASE IN AIR PRESSURE REQUIRED TO OVERCOME THE ADDITIONAL RESISTANCE OF THE HOT WATER COILS WHILE MAINTAINING THE DESIGN AIR FLOW UNITS - INCHES OF WATER
6 INCREASE IN FAN MOTOR HORSEPOWER REQUIRED TO PRODUCE THE ADDITIONAL AIR PRESSURE
7 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR
8 PIPE DIAMETER OF THE INLET AND OUTLET CONNECTIONS FOR THE INDIVIDUAL COILS UNITS - INCHES
9 AVERAGE VELOCITY OF THE AIR AT THE HOT WATER COIL UNITS - FEET PER MINUTE
10 PRESSURE LOSS OF THE HOT WATER AS IT FLOWS THROUGH EACH COIL UNITS - POUNDS PER SQUARE INCH

TABLE 3.7-2

BUILDING School of Business - BSU

HEATING SYSTEM INFORMATION DERIVED FROM CONSTRUCTION DRAWINGS, BUILDING SPECIFICATIONS AND FIELD DATA

	PER AIR HANDLING SYSTEM								PER FIN TUBE STEAM COIL							
NO[1]	AIR[2] FLOW IN (SCFM)	FANS[3]	FAN RPM[4] DESIGN / MEAS.	FAN HP[5] DESIGN / MEAS.	MOTOR[6] VOLTS PHASE	FAN[7] SP @ AIR FLOW	FAN[8] AIR VEL. FPM	MEASURED AIR FLOW IN (SCFM)	COILS PER SYSTEM	SIZE (W X L) IN INCHES	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	AIR[10] PER COIL (SCFM)	AIR[11] DESIGN IN(°F) IN	AIR[11] DESIGN IN(°F) OUT	HEATING[12] CAPACITY IN (BTU/HR)
2	37550	1	1000 1175	75 50	480/3	6	2250	42338	2	24x120	20	939	18775	55	101	1865484

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 NUMBER OF IDENTICAL FANS PER SYSTEM
4 FAN SPEED BOTH DESIGN AND THAT MEASURED AT INSTALLATION UNITS - REVOLUTIONS PER MINUTE
5 FAN DRIVE MOTOR HORSEPOWER BOTH DESIGN AND THAT MEASURED AT INSTALLATION
6 DRIVE MOTOR VOLTAGE AND PHASE; ALL VOLTAGE FREQUENCIES ARE 60 HERTZ
7 RATED STATIC AIR PRESSURE PRODUCED BY THE FAN AT THE DESIGN AIR FLOW UNITS - INCHES OF WATER
8 AVERAGE VELOCITY OF THE AIR AT THE FAN DISCHARGE UNITS - FEET PER MINUTE
9 AVERAGE VELOCITY OF THE AIR AT THE STEAM COIL UNITS - FEET PER MINUTE
10 DESIGN AIR FLOW PER STEAM COIL UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
11 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
12 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR

TABLE 3.7-3

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING School of Business - BSU

RATING OF SYSTEM 37550 (SCFM) 3730968 (BTU/HR) NUMBER OF SYSTEMS (2)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 24 X 120/144 = 20 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 18775 / 20 = 939 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 46 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 1865484 /(500)X(50) = 74.6 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WS TYPE COIL WITH 2,3,4 ROWS = 16

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 74.6/ 16 = 4.66

7) $\Delta T_A/\Delta T_{inlet}$ = 46 / 170 (°F) - 55 = 46 / 115 = .40

$\Delta T_W/\Delta T_A$ = 50 / 46 = 1.09

FROM FIGURE 8 : M_t = .75

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 3 /(.75 X 9.39) = .426

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}} / 2$ = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .085

$R_{f2} = R_{ft} - R_{f1}$ = .426 - .085 = .341

FROM FIGURE 6 : FIN SERIES = 08 R_{f2} = .325

COIL MODEL NUMBER BECOMES: 5WS-1208C-24x120 NUMBER OF COILS REQ'D 2

9) FROM FIGURE 9 : ΔP_A = .83 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(37550)X(.83) = 4.89

10) FROM FIGURE 12 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(2.4 + 2.8) =

5.62 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 2.43 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.7-4

NEW FAN SPEED ESTIMATE SHEET

BUILDING School of Business - BSU

RATING OF SYSTEM 37550 (SCFM) 3730968 (BTU/HR) NUMBER OF SYSTEMS (2)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$FAN\ BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(\ 37550\ )X(\ 6\ ) = 35.37$

$FAN\ BHP_2 = (FAN\ BHP_1)+(FAN\ BHP_+) = (\ 35.37\ )+(\ 4.89\ ) = 40.26$

$RPM_2 = (RPM_1)X(FAN\ BHP_2/FAN\ BHP_1)^{1/3} = (\ 1000\ )X(40.26\ /\ 35.37)^{1/3} = 1044$

CHANGE IN FAN SPEED = $(RPM_2 - RPM_1)/RPM_1$ = .044 OR X 100 = 4.4 % INCREASE

DEFINITIONS:

$FAN\ BHP_1$ = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

$FAN\ BHP_+$ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9, FROM THE COIL SELECTION SHEET).

$FAN\ BHP_2$ = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

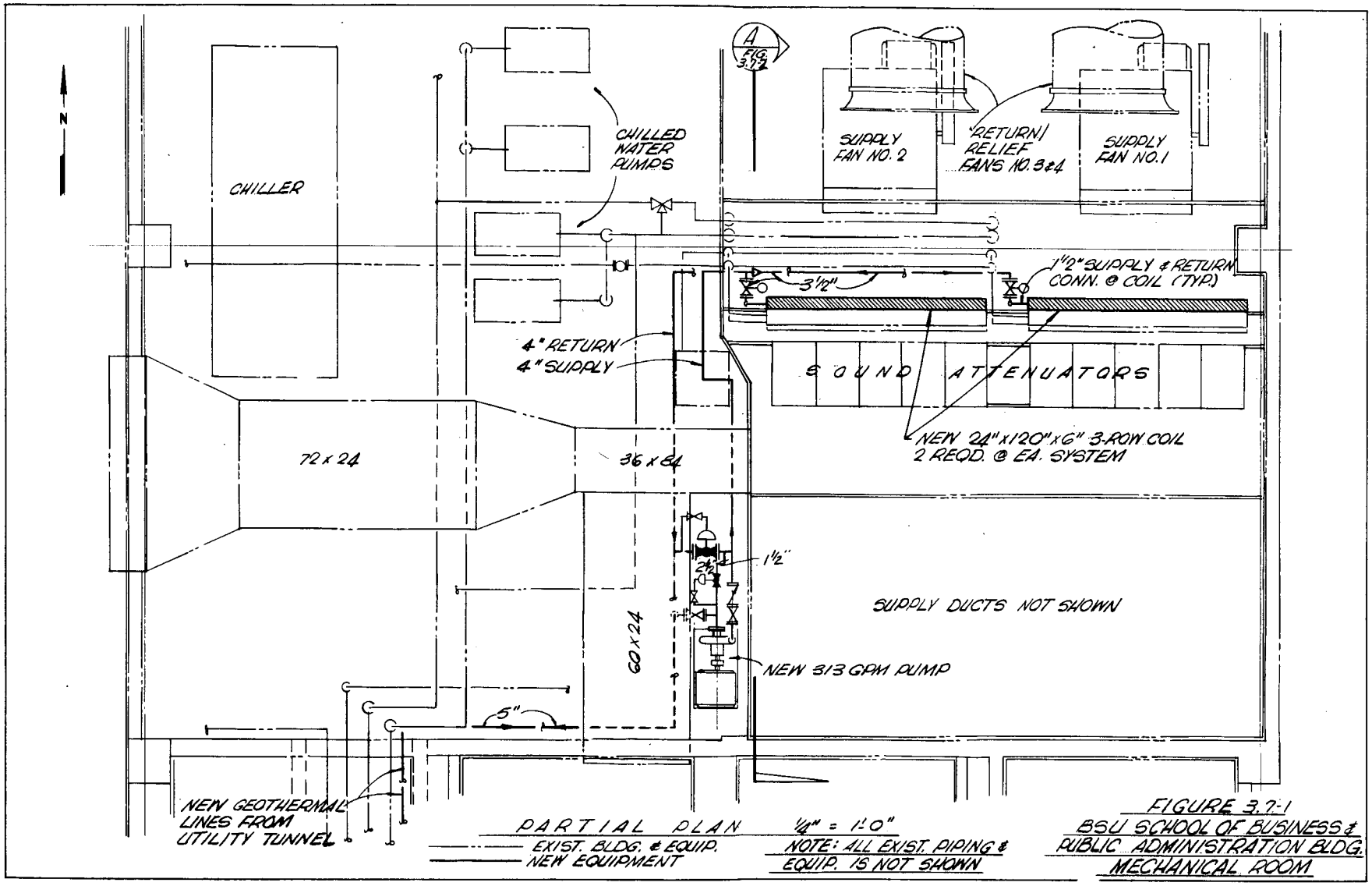


Figure 3.7-1 BSU School of Business & Public Administration Building Mechanical Room

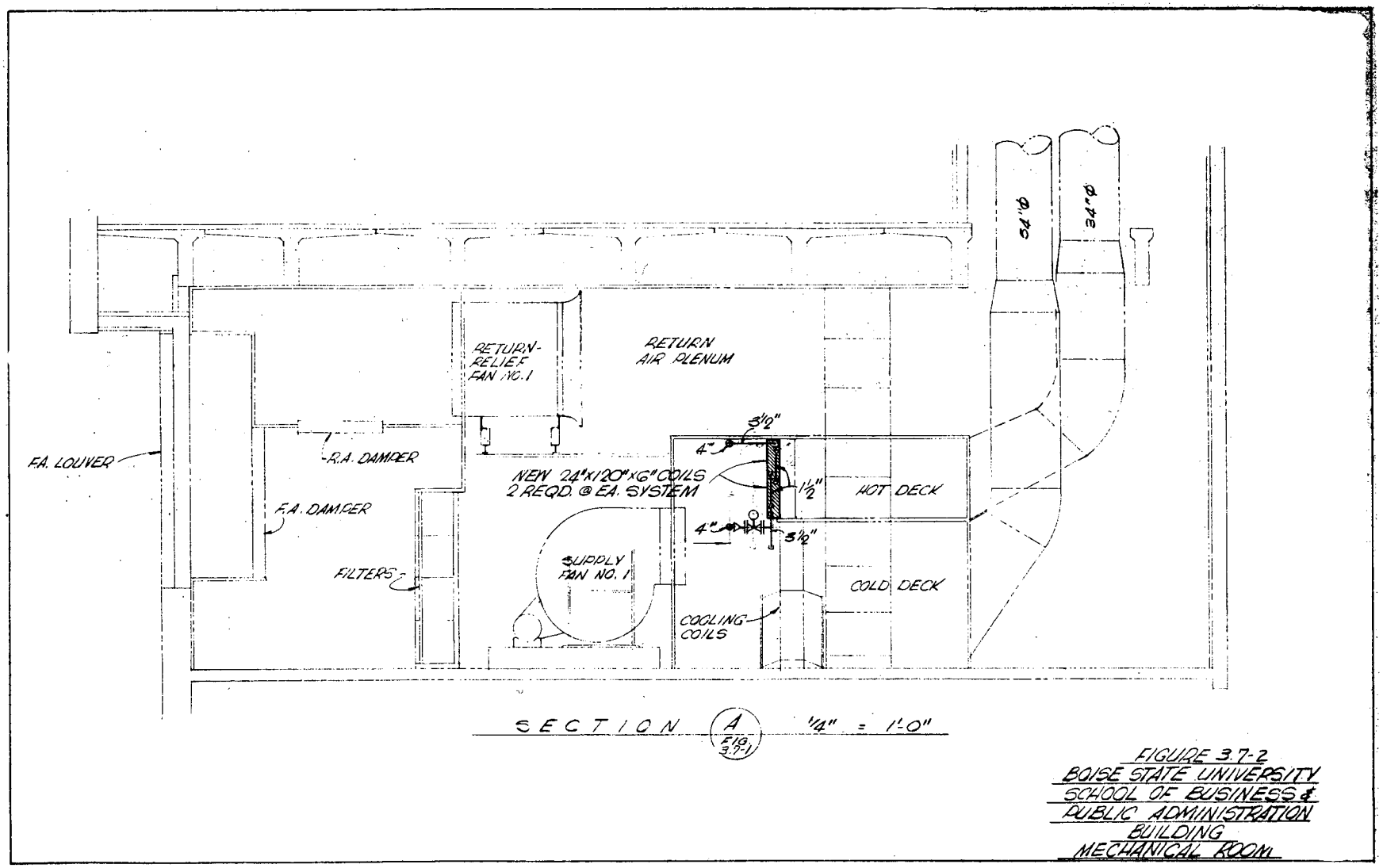
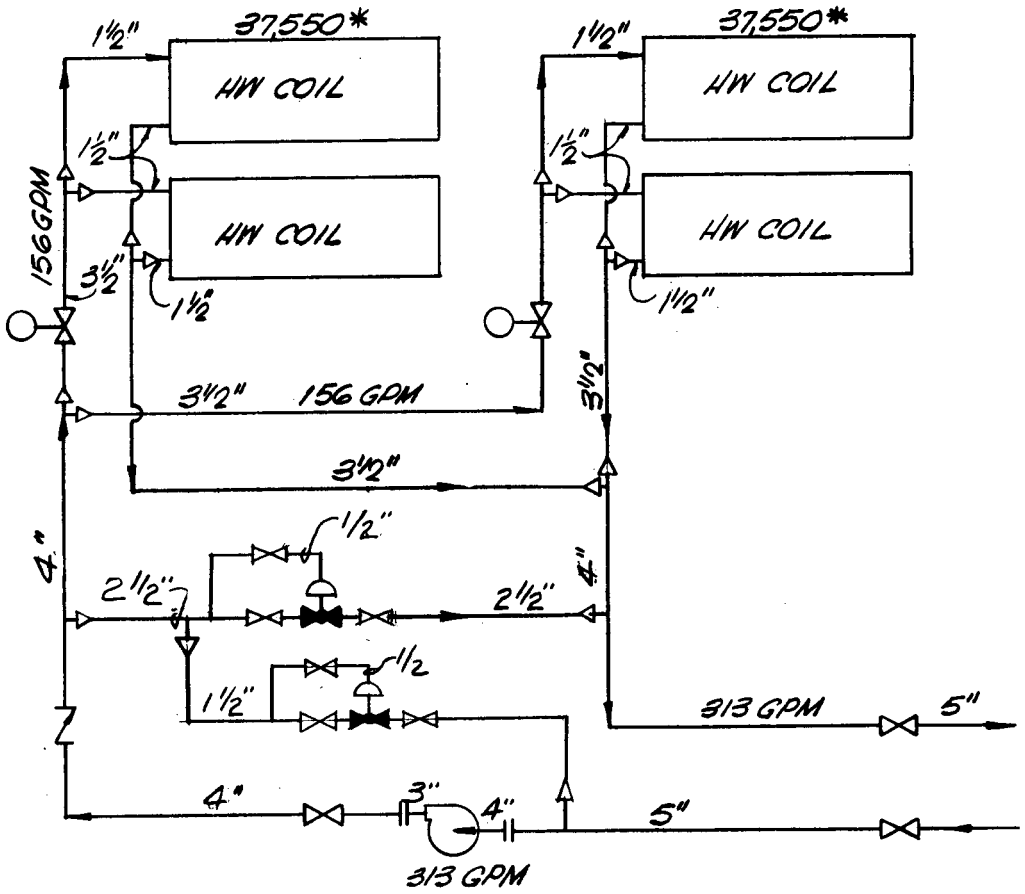


Figure 3.7-2 BSU School of Business & Public Administration Building Mechanical Room



FLOWS SHOWN ARE CALCULATED VALUES PLUS 5%
*RATED AIR FLOW OF SYSTEM IN SCFM

FIGURE 3.7-3
BSU SCHOOL OF BUSINESS &
PUBLIC ADMINISTRATION BLDG.

Figure 3.7-3 BSU School of Business & Public Administration Building

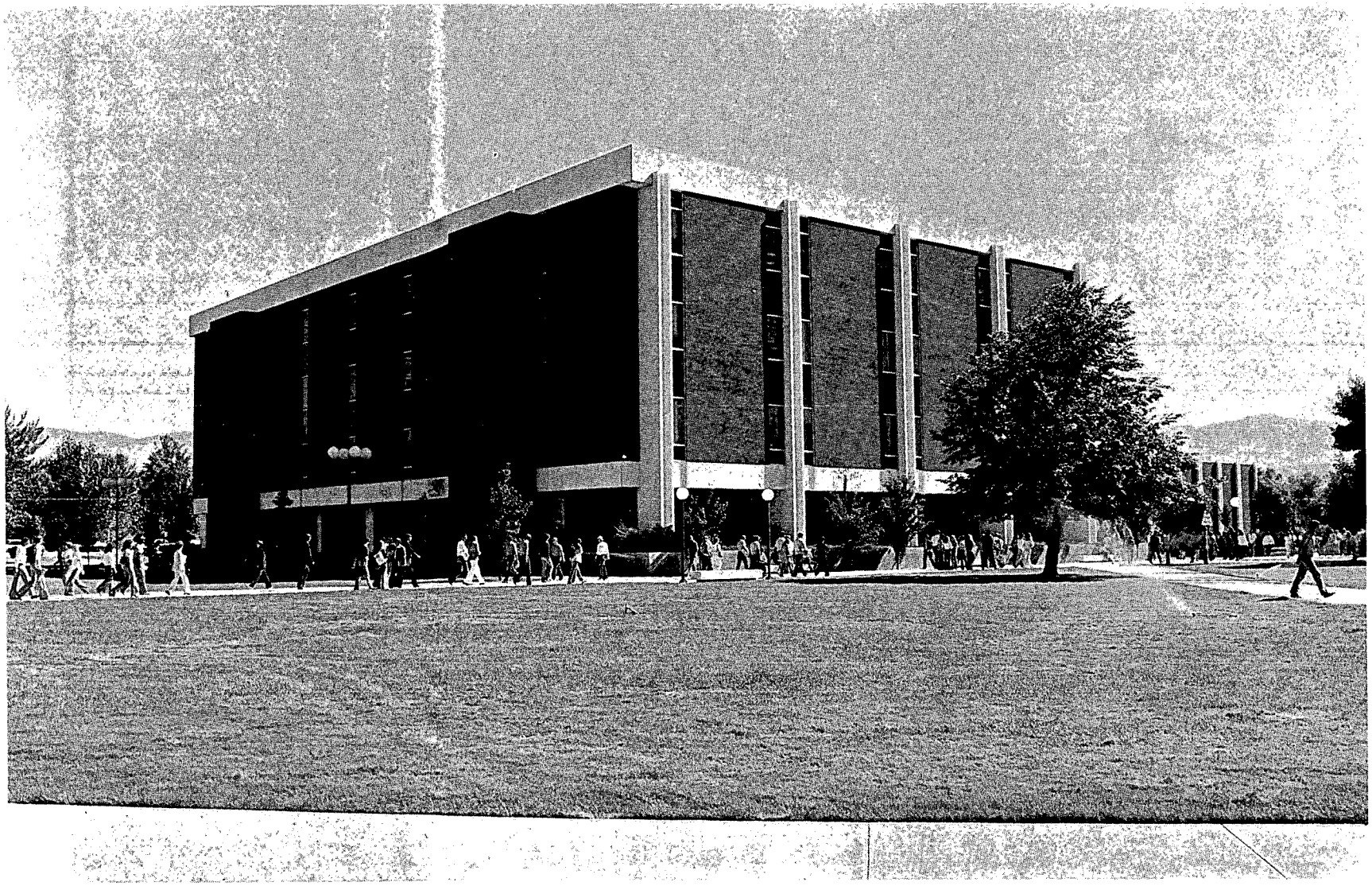


Figure 3.8-0 Library - BSU

3.8 Library - BSU

The original portion of this building uses two forced air packaged multizone HVAC units and a hot water perimeter heat system. Both multizones use 15 HP fan motors with one unit rated at 22,800 SCFM and the other rated at 25,500 SCFM. Subsequently, this building was enlarged and the new areas are served by one large central plant HVAC unit with two 100 HP fan motors. Each fan section is rated at 55,000 SCFM. All the forced air units use steam and chilled water coils. Reference Tables 3.8-1 through 3.8-8 for more detail.

Hot water coils installed to duplicate existing steam coil capacities will produce the following effects:

1) The multizone unit rated at 22,800 SCFM will require a 25% increase in fan horsepower to overcome the 1.04 inches of water, fan static pressure increase.

2) The multizone unit rated at 25,500 SCFM will require a 30% increase in fan horsepower to overcome the 1.40 inches of water, fan static pressure increase.

3) The central plant unit will require a nominal 12% increase in fan horsepower to overcome the 1.42 inches of water, fan static pressure increase.

With the increased power requirements, the two 15 HP systems will definitely require motor changes. New fan speeds will both be around 460 RPM, an approximate 20% to 25% increase. The two 100 HP systems will not require motor changes. Fan speed changes to approximately 900 RPM or a 7.4% increase from the original design speed will be required. However, these two units were noted to be operating at slightly above this speed already with a subsequent 4.2% higher than design air flow.

Maximum geothermal water requirements will approach 365 gpm. Building supply and discharge lines will thus be 6 in. dia. pipe. Reference Figures 3.8-1 through 3.8-5 in addition to the calculations for total modification requirements. The estimated building conversion cost in 1975 dollars without escalation or contingency is $38,800.

TABLE 3.8-1

SUMMARY

BUILDING Library - BSU

TOTAL MAXIMUM HOT WATER (GEOTHERMAL) REQUIRED FOR THIS BUILDING - 346.6 GALLONS PER MINUTE

TEMPERATURE OF WATER TO THE COILS ASSUMED TO BE 170°F AND THE TEMPERATURE OF WATER FROM THE COILS ASSUMED TO BE 120°F UNLESS NOTED OTHERWISE.

HOT WATER COIL CALCULATIONS, MODEL NUMBERS AND NOMENCLATURE USED ARE BASED ON Mc QUAY.

	PER AIR HANDLING SYSTEM								PER FIN TUBE HOT WATER COIL								
NO[1]	AIR[2] FLOW IN (SCFM)	AIR[3] DESIGN IN(°F) IN	AIR[3] DESIGN IN(°F) OUT	AIR[4] FLOW IN (SCFM)	AIR[5] FLOW (ΔP)	FAN[6] ΔHP	HEATING[7] CAPACITY IN (BTU/HR)	WATER FLOW IN (GPM)	FIN TUBE COIL MODEL NUMBER TYPE	FIN SERIES	TUBE ROWS "C"	SIZE (W X L X D) IN (INCHES)	PIPE[8] SIZE	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	WATER[10] FLOW (ΔP) (PSI)	COILS PER SYSTEM
1	25500	50	114	20350	1.40	4.47	1406592	56.3	5WL	12	03	33x109x6	1-1/2	25	815	1.87	1
1	22800	50	95	22800	1.04	3.72	1108080	44.3	5WL	08	03	33x109x6	1-1/2	25	915	1.17	1
2	55000	51	103	55000	1.42	12.3	3073410	123	5WL	10	03	33x120x6	1-1/2	27.5	1000	2.43	2

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
4 DESIGN AIR FLOW THROUGH THE HEATING COIL SECTION UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
5 INCREASE IN AIR PRESSURE REQUIRED TO OVERCOME THE ADDITIONAL RESISTANCE OF THE HOT WATER COILS WHILE MAINTAINING THE DESIGN AIR FLOW UNITS - INCHES OF WATER
6 INCREASE IN FAN MOTOR HORSEPOWER REQUIRED TO PRODUCE THE ADDITIONAL AIR PRESSURE
7 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR
8 PIPE DIAMETER OF THE INLET AND OUTLET CONNECTIONS FOR THE INDIVIDUAL COILS UNITS - INCHES
9 AVERAGE VELOCITY OF THE AIR AT THE HOT WATER COIL UNITS - FEET PER MINUTE
10 PRESSURE LOSS OF THE HOT WATER AS IT FLOWS THROUGH EACH COIL UNITS - POUNDS PER SQUARE INCH

TABLE 3.8-2

BUILDING Library - BSU

HEATING SYSTEM INFORMATION DERIVED FROM CONSTRUCTION DRAWINGS, BUILDING SPECIFICATIONS AND FIELD DATA

	PER AIR HANDLING SYSTEM								PER FIN TUBE STEAM COIL							
NO[1]	AIR[2] FLOW IN (SCFM)	FANS[3]	FAN RPM[4] DESIGN / MEAS.	FAN HP[5] DESIGN / MEAS.	MOTOR[6] VOLTS PHASE	FAN[7] SP @ AIR FLOW	FAN[8] AIR VEL. FPM	MEASURED AIR FLOW IN (SCFM)	COILS PER SYSTEM	SIZE (W X L) IN INCHES	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	AIR[10] PER COIL (SCFM)	AIR[11] DESIGN IN(°F) IN	AIR[11] DESIGN IN(°F) OUT	HEATING[12] CAPACITY IN (BTU/HR)
1	25500	1	370 /	15 /	440/3	1.375	1520		1	33x109	25	815	25500	50	114	1410000
1	22800	1	393 /	15 /	440/3	1.625	1360		1	33x109	25	915	22800	50	95	1110000
2	55000	1	840 / 990	100 / 84.3	480/3	6	2150	57300	2	33x120	27.5	1000	27500	51	103	1540000

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 NUMBER OF IDENTICAL FANS PER SYSTEM
4 FAN SPEED BOTH DESIGN AND THAT MEASURED AT INSTALLATION UNITS - REVOLUTIONS PER MINUTE
5 FAN DRIVE MOTOR HORSEPOWER BOTH DESIGN AND THAT MEASURED AT INSTALLATION
6 DRIVE MOTOR VOLTAGE AND PHASE; ALL VOLTAGE FREQUENCIES ARE 60 HERTZ
7 RATED STATIC AIR PRESSURE PRODUCED BY THE FAN AT THE DESIGN AIR FLOW UNITS - INCHES OF WATER
8 AVERAGE VELOCITY OF THE AIR AT THE FAN DISCHARGE UNITS - FEET PER MINUTE
9 AVERAGE VELOCITY OF THE AIR AT THE STEAM COIL UNITS - FEET PER MINUTE
10 DESIGN AIR FLOW PER STEAM COIL UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
11 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
12 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR

TABLE 3.8-3

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Library - BSU

RATING OF SYSTEM 25500 (SCFM) 1406592 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 33 X 109/144 = 25 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 20350 / 25 = 815 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 64 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 1406592 /(500)X(50) = 56.3 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WL TYPE COIL WITH 3, 4 ROWS = 16

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 56.3 / 16 = 3.52

7) $\Delta T_A/\Delta T_{inlet}$ = 64 / 170 (°F) - 50 = 64 / 120 = .533

$\Delta T_W/\Delta T_A$ = 50 / 64 = 781

FROM FIGURE 8 : M_t = 1.1

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 3 /(1.1 X 8.15) = .355

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}} / 2$ = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .10

$R_{f2} = R_{ft} - R_{f1}$ = .335 - .10 = .235

FROM FIGURE 6 : FIN SERIES = 12 R_{f2} = .220

COIL MODEL NUMBER BECOMES: 5WL-1203C-33x109 NUMBER OF COILS REQ'D 1

9) FROM FIGURE 9 : ΔP_A = 1.4 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(20350)X(1.40) = 4.47

10) FROM FIGURE 12 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(1.1 + 2.9) =

4.32 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 1.87 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.8-4

NEW FAN SPEED ESTIMATE SHEET

BUILDING Library - BSU

RATING OF SYSTEM 25500 (SCFM) 1406592 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

FAN BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(20350)X(1.375) = 4.39

FAN BHP_2 = (FAN BHP_1)+(FAN BHP_+) = (4.39)+(4.47) = 8.86

RPM_2 = (RPM_1)X(FAN BHP_2/FAN BHP_1)$^{1/3}$ = (370)X(8.86 / 4.39)$^{1/3}$ = 468

CHANGE IN FAN SPEED = (RPM_2 - RPM_1)/RPM_1 = .265 OR X 100 = 26.5% INCREASE

DEFINITIONS:

FAN BHP_1 = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

FAN BHP_+ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9,FROM THE COIL SELECTION SHEET).

FAN BHP_2 = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

TABLE 3.8-5

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Library - BSU

RATING OF SYSTEM 22800 (SCFM) 1108080 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 33 X 109/144 = 25 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 22800 / 25 = 915 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 45 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 1108080 /(500)X(50) = 44.3 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WL TYPE COIL WITH 3, 4 ROWS = 16

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 44.3 / 16 = 2.77

7) $\Delta T_A/\Delta T_{inlet}$ = 45 / 170 (°F) - 50 = 45 / 120 = .375

$\Delta T_W/\Delta T_A$ = 50 / 45 = 1.11

FROM FIGURE 8 : M_t = .68

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 3 /(.68 X 9.15) = .482

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}} / 2$ = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .11

$R_{f2} = R_{ft} - R_{f1}$ = .482 - .11 = .372

FROM FIGURE 6 : FIN SERIES = 08 R_{f2} = .306

COIL MODEL NUMBER BECOMES: 5WL-0803C-33x109 NUMBER OF COILS REQ'D 1

9) FROM FIGURE 9 : ΔP_A = 1.04 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(22800)X(1.04) = 3.72

10) FROM FIGURE 12 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(.7 + 1.8) =

2.7 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 1.17 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.8-6

NEW FAN SPEED ESTIMATE SHEET

BUILDING Library - BSU

RATING OF SYSTEM 22800 (SCFM) 1108080 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$FAN\ BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(22800)X(1.625) = 5.82$

$FAN\ BHP_2 = (FAN\ BHP_1)+(FAN\ BHP_+) = (5.82)+(3.72) = 9.54$

$RPM_2 = (RPM_1)X(FAN\ BHP_2/FAN\ BHP_1)^{1/3} = (393)X(9.54/5.82)^{1/3} = 463$

CHANGE IN FAN SPEED = $(RPM_2 - RPM_1)/RPM_1$ = .179 OR X 100 = 17.9% INCREASE

DEFINITIONS:

$FAN\ BHP_1$ = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

$FAN\ BHP_+$ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9,FROM THE COIL SELECTION SHEET).

$FAN\ BHP_2$ = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

TABLE 3.8-7

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Library - BSU

RATING OF SYSTEM 55000 (SCFM) 3073410 (BTU/HR) NUMBER OF SYSTEMS (2)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 33 X 120/144 = 27.5 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 27500 / 27.5 = 1000 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 52 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 1536705 /(500)X(50) = 61.5 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WL TYPE COIL WITH 3, 4 ROWS = 16

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 61.5 / 16 = 3.84

7) $\Delta T_A/\Delta T_{inlet}$ = 52 / 170 (°F) - 51 = 52 / 119 = .437

$\Delta T_W/\Delta T_A$ = 50 / 52 = .962

FROM FIGURE 8 : M_t = .83

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 3 /(.83X 10) = .361

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}}$ / 2 = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .095

$R_{f2} = R_{ft} - R_{f1}$ = .361 - .095 = .266

FROM FIGURE 6 : FIN SERIES = 10 R_{f2} = .230

COIL MODEL NUMBER BECOMES: 5WL-1003C-33x120 NUMBER OF COILS REQ'D 2

9) FROM FIGURE 9 : ΔP_A = 1.42 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(55000)X(1.42) = 12.26

10) FROM FIGURE 12 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(1.4 + 3.8) =

5.62 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 2.43 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.8-8

NEW FAN SPEED ESTIMATE SHEET

BUILDING Library - BSU

RATING OF SYSTEM 55000 (SCFM) 3073410 (BTU/HR) NUMBER OF SYSTEMS (2)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$FAN\ BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(55000)X(6) = 51.81$

$FAN\ BHP_2 = (FAN\ BHP_1)+(FAN\ BHP_+) = (51.81)+(12.3) = 64.11$

$RPM_2 = (RPM_1)X(FAN\ BHP_2/FAN\ BHP_1)^{1/3} = (840)X(64.11/51.81)^{1/3} = 902$

CHANGE IN FAN SPEED = $(RPM_2 - RPM_1)/RPM_1$ = .074 OR X 100 = 7.4 % INCREASE

DEFINITIONS:

$FAN\ BHP_1$ = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

$FAN\ BHP_+$ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9,FROM THE COIL SELECTION SHEET).

$FAN\ BHP_2$ = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

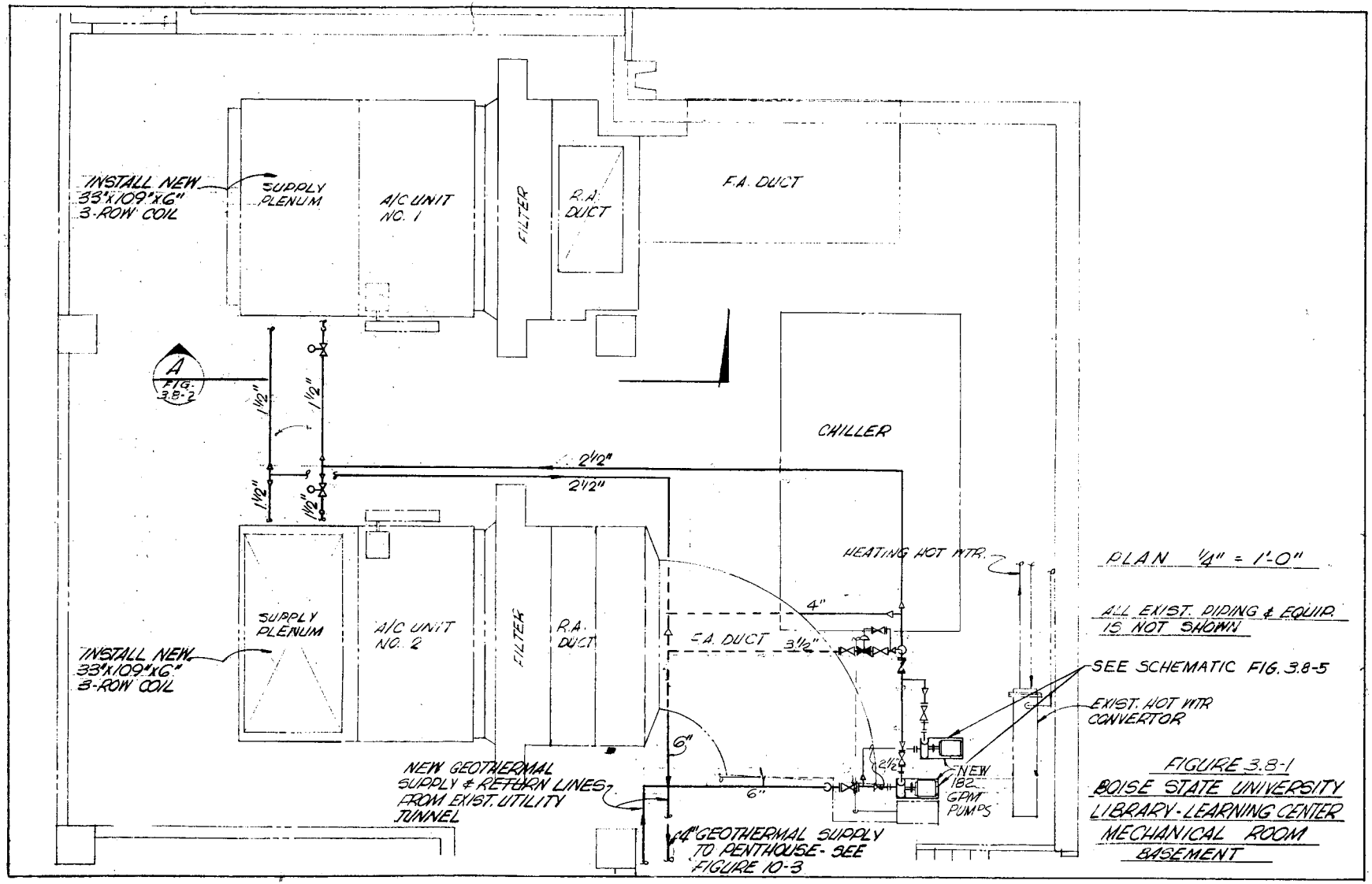


Figure 3.8-1 BSU Library Learning Center Mechanical Room Basement

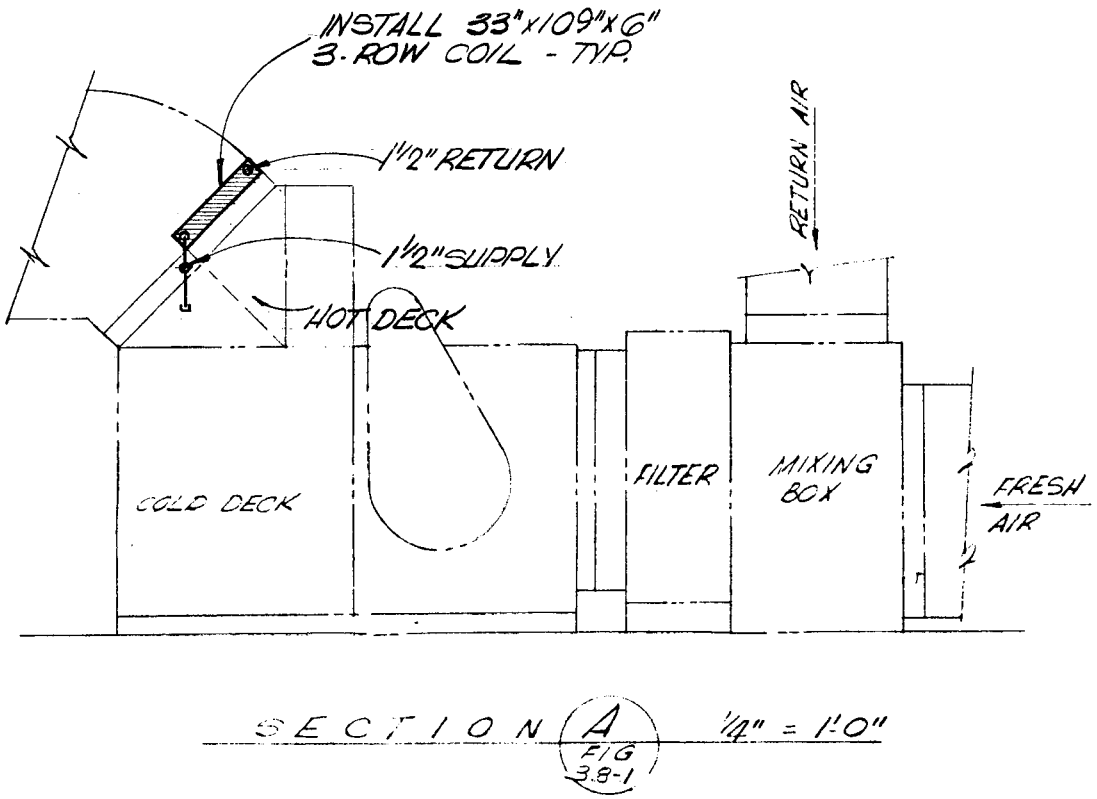


FIGURE 3.8-2
BOISE STATE UNIVERSITY
LIBRARY - LEARNING CENTER
MECHANICAL ROOM

Figure 3.8-2 BSU Library - Learning Center Mechanical Room

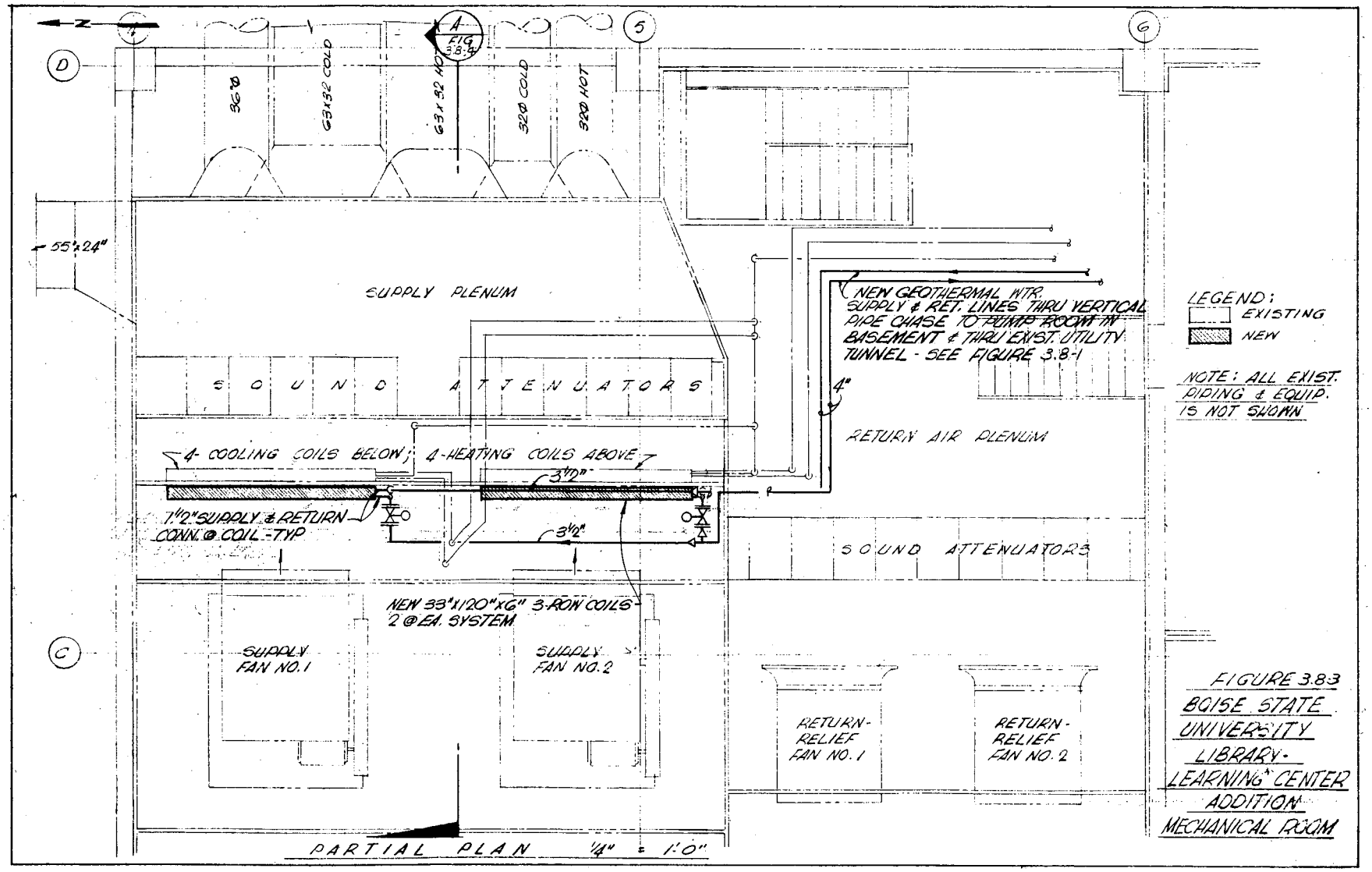


Figure 3.8-3 Boise State University Library-Learning Center Addition Mechanical Room

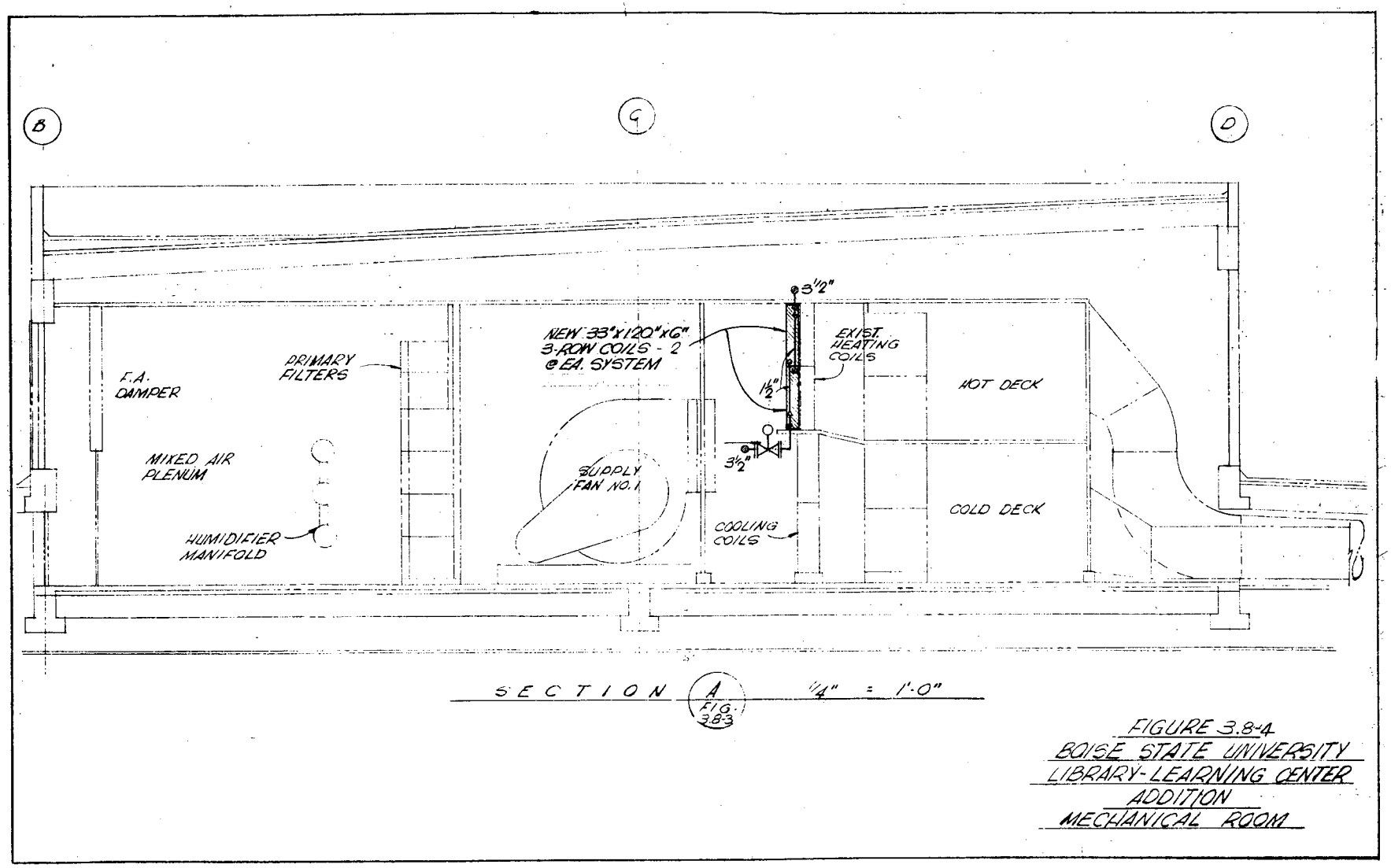


Figure 3.8-4 Boise State University Library-Learning Center Addition Mechanical Room

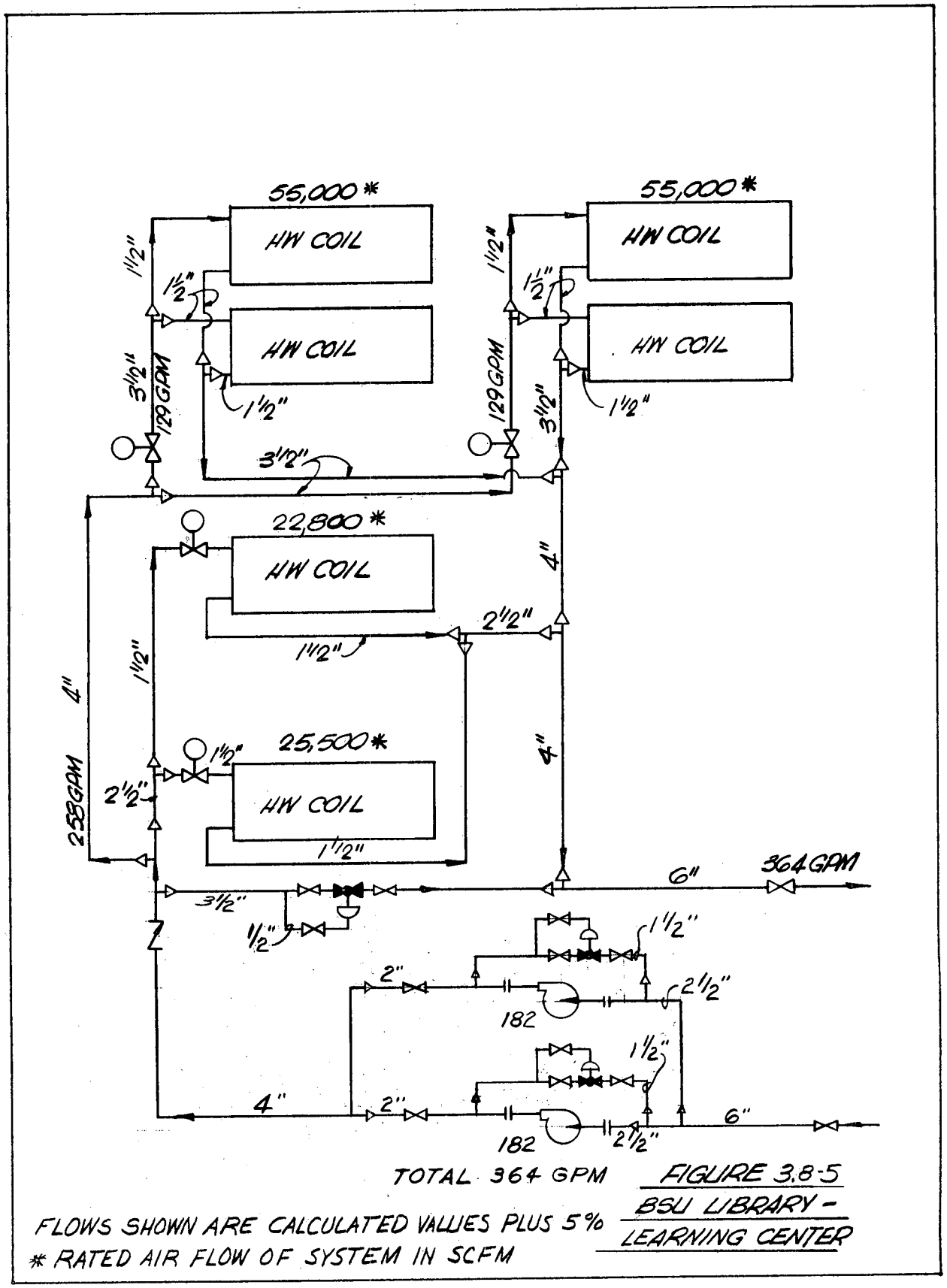


Figure 3.8-5 Boise State University Library-Learning Center

Figure 3.9-0 Liberal Arts Builsing - BSU

3.9 Liberal Arts Building - BSU

This building uses a forced air, packaged, multizone HVAC unit with hot water perimeter heating. The multizone unit uses a 15 HP fan motor and is rated at 25,400 SCFM. Steam and chilled water coils are utilized in the coil sections. Perimeter heating hot water is provided by a steam/water heat exchanger rated at 2,800,000 Btu/hr, a significant part of the total building heating capacity.

Conversion to geothermal will involve the installation of hot water coils in the multizone unit identical in face area and rated heating capacity as the existing steam coils. The installation of a water/water heat exchanger in parallel with the existing steam/water convertor will also be required.

The addition of coils in the multizone unit require a minor 6% increase in fan horsepower to overcome the .33 inches of water, fan static pressure increase. The new fan speed for this added resistance will be 460 RPM or an 8% change. From the balance reports, this particular fan motor is apparently operating close to the rated 15 HP and a possible change in motor is indicated.

Approximate physical dimensions of a tube and shell, water/water heat exchanger that will satisfy the heating capacity will be 18 in. dia. by 10 ft in length. Geothermal water pipe connections will be 6 in. dia. and circulating hot water connections 4 in. dia.

Maximum geothermal water flows will approach 425 gpm and will require building supply and discharge lines of 5 in. dia. Reference Figures 3.9-1 through 3.9-3 and Tables 3.9-1 through 3.9-4 for total proposed modifications. The estimated conversion cost without escalation or contingency is $28,700.

TABLE 3.9-1

SUMMARY

BUILDING Liberal Arts - BSU

TOTAL MAXIMUM HOT WATER (GEOTHERMAL) REQUIRED FOR THIS BUILDING - 404.4* GALLONS PER MINUTE

*Includes (1) shell and tube water/water heat exchanger 375 GPM; 170°F inlet; 150°F outlet.
TEMPERATURE OF WATER TO THE COILS ASSUMED TO BE 170°F AND THE TEMPERATURE OF WATER FROM THE COILS ASSUMED TO BE 120°F UNLESS NOTED OTHERWISE.

HOT WATER COIL CALCULATIONS, MODEL NUMBERS AND NOMENCLATURE USED ARE BASED ON Mc QUAY.

	PER AIR HANDLING SYSTEM								PER FIN TUBE HOT WATER COIL								
NO[1]	AIR[2] FLOW IN (SCFM)	AIR[3] DESIGN IN(°F) IN	AIR[3] DESIGN IN(°F) OUT	AIR[4] FLOW IN (SCFM)	AIR[5] FLOW (ΔP)	FAN[6] ΔHP	HEATING[7] CAPACITY IN (BTU/HR)	WATER FLOW IN (GPM)	FIN TUBE COIL MODEL NUMBER TYPE	FIN SERIES	TUBE ROWS "C"	SIZE (W X L X D) IN (INCHES)	PIPE[8] SIZE	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	WATER[10] FLOW (ΔP) (PSI)	COILS PER SYSTEM
1	25400	59	99	17021	.325	.87	735307	29.4	5WH	10	02	42x107.5x5	1-1/2	31.4	543	1.03	1

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
4 DESIGN AIR FLOW THROUGH THE HEATING COIL SECTION UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
5 INCREASE IN AIR PRESSURE REQUIRED TO OVERCOME THE ADDITIONAL RESISTANCE OF THE HOT WATER COILS WHILE MAINTAINING THE DESIGN AIR FLOW UNITS - INCHES OF WATER
6 INCREASE IN FAN MOTOR HORSEPOWER REQUIRED TO PRODUCE THE ADDITIONAL AIR PRESSURE
7 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR
8 PIPE DIAMETER OF THE INLET AND OUTLET CONNECTIONS FOR THE INDIVIDUAL COILS UNITS - INCHES
9 AVERAGE VELOCITY OF THE AIR AT THE HOT WATER COIL UNITS - FEET PER MINUTE
10 PRESSURE LOSS OF THE HOT WATER AS IT FLOWS THROUGH EACH COIL UNITS - POUNDS PER SQUARE INCH

TABLE 3.9-2

BUILDING Liberal Arts - BSU

HEATING SYSTEM INFORMATION DERIVED FROM CONSTRUCTION DRAWINGS, BUILDING SPECIFICATIONS AND FIELD DATA

	PER AIR HANDLING SYSTEM								PER FIN TUBE STEAM COIL							
NO[1]	AIR[2] FLOW IN (SCFM)	FANS[3]	FAN RPM[4] DESIGN	FAN RPM[4] MEAS.	FAN HP[5] DESIGN	FAN HP[5] MEAS.	MOTOR[6] VOLTS PHASE	FAN[7] SP @ AIR FLOW	FAN[8] AIR VEL. FPM	MEASURED AIR FLOW IN (SCFM)	COILS PER SYSTEM	SIZE (W X L) IN INCHES	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	AIR[10] PER COIL (SCFM)	AIR[11] DESIGN IN(°F) IN	AIR[11] DESIGN IN(°F) OUT	HEATING[12] CAPACITY IN (BTU/HR)
1	25400	1	425	425	15	15	460/3	1.25		17021	1	42x107.5	31.4	543	17021	59	99	735000

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 NUMBER OF IDENTICAL FANS PER SYSTEM
4 FAN SPEED BOTH DESIGN AND THAT MEASURED AT INSTALLATION UNITS - REVOLUTIONS PER MINUTE
5 FAN DRIVE MOTOR HORSEPOWER BOTH DESIGN AND THAT MEASURED AT INSTALLATION
6 DRIVE MOTOR VOLTAGE AND PHASE; ALL VOLTAGE FREQUENCIES ARE 60 HERTZ
7 RATED STATIC AIR PRESSURE PRODUCED BY THE FAN AT THE DESIGN AIR FLOW UNITS - INCHES OF WATER
8 AVERAGE VELOCITY OF THE AIR AT THE FAN DISCHARGE UNITS - FEET PER MINUTE
9 AVERAGE VELOCITY OF THE AIR AT THE STEAM COIL UNITS - FEET PER MINUTE
10 DESIGN AIR FLOW PER STEAM COIL UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
11 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
12 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR

TABLE 3.9-3

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Liberal Arts - BSU

RATING OF SYSTEM 25400 (SCFM) 735307 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 42 X 107/5/144 = 31.35 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 17021 / 31.35 = 543 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 40 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 735307 /(500)X(50) = 29.4 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WH TYPE COIL WITH 1,2,3,4 ROWS = 14

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 29.4 / 14 = 2.1

7) $\Delta T_A/\Delta T_{inlet}$ = 40 / 170 (°F) - 59 = 40 / 111 = .360

$\Delta T_W/\Delta T_A$ = 50 / 40 = 1.25

FROM FIGURE 7 : M_t = .71

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 2 /(.71 X 5.43) = .519

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}} / 2$ = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .13

$R_{f2} = R_{ft} - R_{f1}$ = .519 - .13 = .389

FROM FIGURE 5 : FIN SERIES = 10 R_{f2} = .315

COIL MODEL NUMBER BECOMES: 5WH-1002C-42x107.5 NUMBER OF COILS REQ'D 1

9) FROM FIGURE 9 : ΔP_A = .325 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(17021)X(.325) = .868

10) FROM FIGURE 11 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(1.1 + 1.1) =

2.38 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 1.03 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.9-4

NEW FAN SPEED ESTIMATE SHEET

BUILDING Liberal Arts - BSU

RATING OF SYSTEM 25400 (SCFM) 735307 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

FAN BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(17021)X(1.25) = 3.34

FAN BHP_2 = (FAN BHP_1)+(FAN BHP_+) = (3.34)+(.868) = 4.21

RPM_2 = (RPM_1)X(FAN BHP_2/FAN BHP_1)$^{1/3}$ = (425)X(4.21 / 3.34)$^{1/3}$ = 459

CHANGE IN FAN SPEED = (RPM_2 - RPM_1)/RPM_1 = .080 OR X 100 = 8 % INCREASE

DEFINITIONS:

FAN BHP_1 = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

FAN BHP_+ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9,FROM THE COIL SELECTION SHEET).

FAN BHP_2 = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

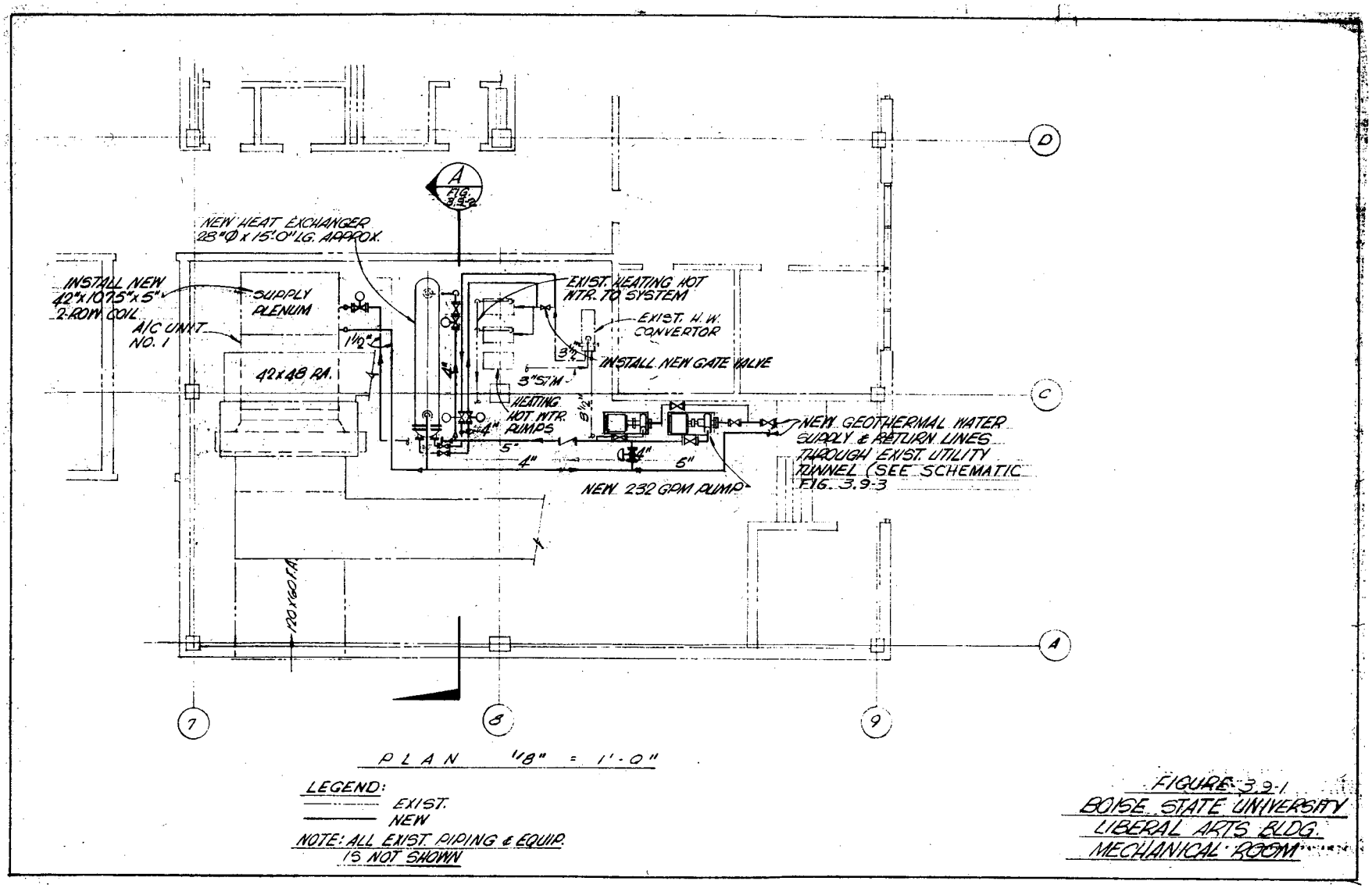


Figure 3.9-1 Boise State University Liberal Arts Building Mechanical Room

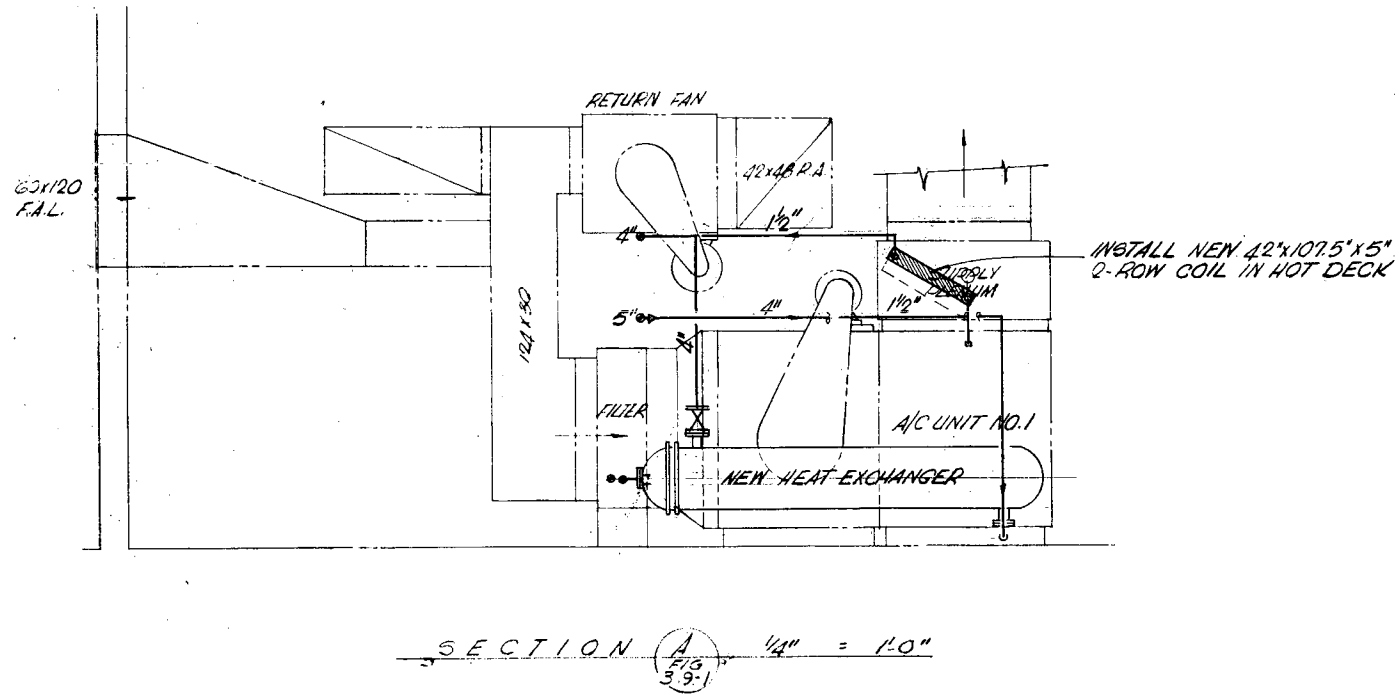
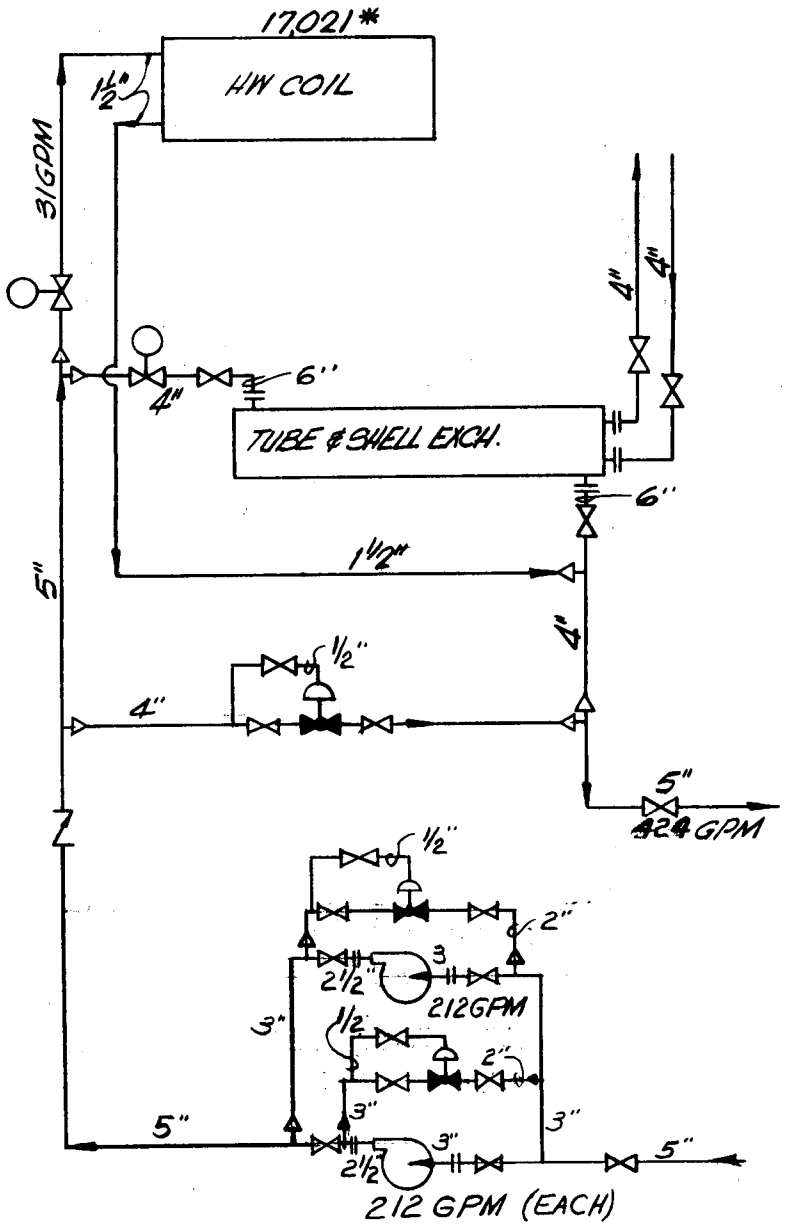


Figure 3.9-2 Boise State University Liberal Arts Building Mechanical Room



* RATED AIR FLOW OF SYSTEM IN SCFM

FLOWS SHOWN ARE CALCULATED VALUES PLUS 5%

FIGURE 3.9-3
BSU LIBERAL ARTS BLDG.

Figure 3.9-3 Boise State University Liberal Arts Building

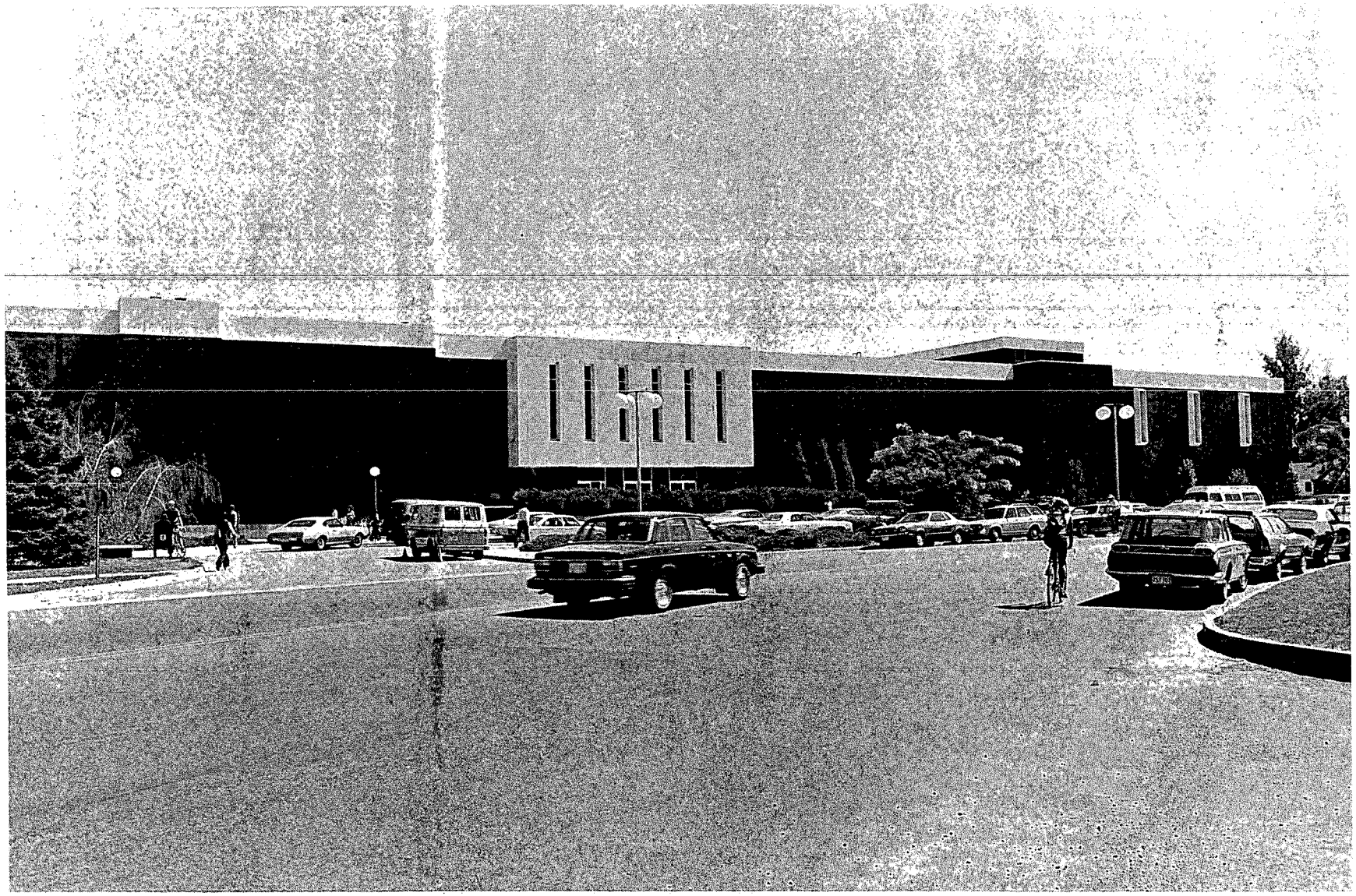


Figure 3.10-9 Student Union - BSU

3.10 Student Union - BSU

The original portion of this building is served by three forced air, packaged,multizone HVAC units. This building was more than doubled in size after original construction and five additional forced air, packaged, multizone HVAC units installed to take care of the added area. Reference Tables 3.10-1 through 3.10-12 for more detail.

Information for the original three units is incomplete but data from the five more recent units indicate that increases in fan motor horsepower required as a result of added hot water coils will be on the order of 30% for the 27,750 SCFM and 22,400 SCFM rated systems and 45% for the 16,800 SCFM rated system. Of the five recent multizone units, increases in fan horsepower for added hot water coils will range from 8% to 46%. As a result of design features particular to these installations, required fan static pressure increases are high.

Hot water coils installed in the five more recent multizone units will have the following effects:

1) The system rated at 19,285 SCFM powered with a 15 HP motor will require 46% more horsepower to overcome the fan static pressure increase of 2.3 inches of water.

2) The system rated at 23,125 SCFM powered with a 15 HP motor will require 31% more horsepower to overcome the fan static pressure requirement of 1.3 inches of water.

3) The system rated at 20,725 SCFM powered with a 15 HP motor will require 43% more horsepower to overcome the fan static pressure requirement of 2 inches of water.

4) The system rated at 11,025 SCFM powered with a 7.5 HP motor will require 40% more horsepower to overcome the fan static pressure requirement of 1.7 inches of water.

5) The system rated at 5,700 SCFM powered with a 5 HP motor will require 8% more horsepower to overcome the fan static pressure additional requirement of .42 inches of water.

From air balance data available, all systems appear to be operating close to the rated motor horsepower. It will be necessary to change all eight system motors, however, the present 5 HP system could utilize one of the 10 HP motors, the 7.5 HP system could likewise use one of the 15 HP motors and the two 10 HP systems could use two of the 15 HP motors.

New fan speeds were established for the five systems on which data was sufficiently complete:

1) 19,285 SCFM rated system - The new fan speed will be 1296 RPM or a 24.4% increase over design, however, this system is presently operating at 1125 RPM and the new value will represent only a 15.2% increase above the present speed.

2) 23,125 SCFM rated system - This fan is operating above design speed at 950 RPM and the new speed will be 1067 RPM, a 20.2% increase over design but only a 12.3% increase over the present speed.

3) 20,725 SCFM rated system - The new fan speed will be 1315 RPM, a 28.9% increase over design. Presently this fan is operating at a speed below design and a corresponding air flow 12.6% below the rated capacity.

4) 11,025 SCFM rated system - A 27.4% increase over the design fan speed is required. This represents a value of 1083 RPM and only a 11.1% increase over the present operating speed.

5) 5,700 SCFM rated system - This system is operating, presently, very close to design and the new fan speed will be 917 RPM, a minor 8.3% increase.

Total maximum geothermal water flow for the building could approach 250 gpm and building supply and discharge lines will be 5 in. dia. pipe. Reference 3.10-1 through 3.10-5 in addition to the calculations for total modifications required. The estimated conversion cost for this building in 1975 dollars without escalation or contingency is $54,300.

TABLE 3.10-1

SUMMARY

BUILDING Student Union - BSU

TOTAL MAXIMUM HOT WATER (GEOTHERMAL) REQUIRED FOR THIS BUILDING - 243.6 GALLONS PER MINUTE

TEMPERATURE OF WATER TO THE COILS ASSUMED TO BE 170°F AND THE TEMPERATURE OF WATER FROM THE COILS ASSUMED TO BE 120°F UNLESS NOTED OTHERWISE.

HOT WATER COIL CALCULATIONS, MODEL NUMBERS AND NOMENCLATURE USED ARE BASED ON Mc QUAY.

	PER AIR HANDLING SYSTEM							PER FIN TUBE HOT WATER COIL								
NO[1]	AIR[2] FLOW IN (SCFM)	AIR[3] DESIGN IN(°F) IN	OUT	AIR[4] FLOW IN (SCFM)	AIR[5] FLOW (ΔP)	FAN[6] ΔHP	HEATING[7] CAPACITY IN (BTU/HR)	WATER FLOW IN (GPM)	FIN TUBE COIL MODEL NUMBER TYPE	FIN SERIES	TUBE ROWS "C"	SIZE (W X L X D) IN (INCHES)	PIPE[8] SIZE	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	WATER[10] FLOW (ΔP) (PSI)	COILS PER SYSTEM
1	19285	63	108	19285	2.3	6.96	938880	37.6	5WS	10	03	24x102x6	2	17	1134	.79	1
1	23125	63	91	23125	1.28	4.65	703920	28.2	5WH	10	02	30x99x5	1-1/2	20.6	1121	1.40	1
1	20725	63	103	20725	2.0	6.51	890400	35.6	5WL	10	03	24x102x6	1-1/2	17	1219	1.31	1
1	11025	63	108	11025	1.71	2.96	535440	21.4	5WL	12	03	18x84x6	1-1/2	10.5	1050	.56	1
1	5700	63	120	5700	.42	.38	351360	14.1	5WH	10	03	33x51x6	1-1/2	11.7	488	0	1
1	27750	60	99	27750			1181800	47.3					2	31.8	873		1
1	22400	60	88	22400			673900	27.0					1-1/2	25	896		1
1	16800	45	90	16800			809500	32.4					1-1/2	18.2	923		1

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
4 DESIGN AIR FLOW THROUGH THE HEATING COIL SECTION UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
5 INCREASE IN AIR PRESSURE REQUIRED TO OVERCOME THE ADDITIONAL RESISTANCE OF THE HOT WATER COILS WHILE MAINTAINING THE DESIGN AIR FLOW UNITS - INCHES OF WATER
6 INCREASE IN FAN MOTOR HORSEPOWER REQUIRED TO PRODUCE THE ADDITIONAL AIR PRESSURE
7 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR
8 PIPE DIAMETER OF THE INLET AND OUTLET CONNECTIONS FOR THE INDIVIDUAL COILS UNITS - INCHES
9 AVERAGE VELOCITY OF THE AIR AT THE HOT WATER COIL UNITS - FEET PER MINUTE
10 PRESSURE LOSS OF THE HOT WATER AS IT FLOWS THROUGH EACH COIL UNITS - POUNDS PER SQUARE INCH

TABLE 3.10-2

BUILDING Student Union - BSU

HEATING SYSTEM INFORMATION DERIVED FROM CONSTRUCTION DRAWINGS, BUILDING SPECIFICATIONS AND FIELD DATA

	PER AIR HANDLING SYSTEM								PER FIN TUBE STEAM COIL							
NO[1]	AIR[2] FLOW IN (SCFM)	FANS[3]	FAN RPM[4] DESIGN / MEAS.	FAN HP[5] DESIGN / MEAS.	MOTOR[6] VOLTS PHASE	FAN[7] SP @ AIR FLOW	FAN[8] AIR VEL. FPM	MEASURED AIR FLOW IN (SCFM)	COILS PER SYSTEM	SIZE (W X L) IN INCHES	FACE AREA IN (SF)	AIR[9] FACE VEL. FPM	AIR[10] PER COIL (SCFM)	AIR[11] DESIGN IN(°F) IN	AIR[11] DESIGN IN(°F) OUT	HEATING[12] CAPACITY IN (BTU/HR)
1	19285	1	980 / 1125	15 / 14	208/3	1.75	1400	17680	1	24x102	17	1134	19285	63	108	939000
1	23125	1	888 / 950	15 / 14.2	208/3	1.75	1380	21359	1	30x99	20.6	1121	23125	63	91	704000
1	20725	1	1020 / 975	15 / 10.3	208/3	1.75	1240	18115	1	24x102	17	1219	20725	63	103	890000
1	11025	1	850 / 975	7.5 / 6.7	208/3	1.6	1100	9835	1	18x84	10.5	1050	11025	63	108	535000
1	5700	1	847 / 850	5 / 3.3	208/3	1.6	1230	5502	1	33x51	11.7	488	5700	63	120	351000
1	27750	1		15	208/3	1			1		31.8	873	27750	60	99	1820000
1	22400	1		10	208/3	1			1		25	896	22400	60	88	674000
1	16800	1		10	208/3	1			1		18.2	923	16800	45	90	810000

1 NUMBER OF IDENTICAL SYSTEMS IN THE BUILDING
2 RATED AIR FLOW OF THE SYSTEM UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
3 NUMBER OF IDENTICAL FANS PER SYSTEM
4 FAN SPEED BOTH DESIGN AND THAT MEASURED AT INSTALLATION UNITS - REVOLUTIONS PER MINUTE
5 FAN DRIVE MOTOR HORSEPOWER BOTH DESIGN AND THAT MEASURED AT INSTALLATION
6 DRIVE MOTOR VOLTAGE AND PHASE; ALL VOLTAGE FREQUENCIES ARE 60 HERTZ
7 RATED STATIC AIR PRESSURE PRODUCED BY THE FAN AT THE DESIGN AIR FLOW UNITS - INCHES OF WATER
8 AVERAGE VELOCITY OF THE AIR AT THE FAN DISCHARGE UNITS - FEET PER MINUTE
9 AVERAGE VELOCITY OF THE AIR AT THE STEAM COIL UNITS - FEET PER MINUTE
10 DESIGN AIR FLOW PER STEAM COIL UNITS - STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE
11 DESIGN TEMPERATURES OF THE AIR ENTERING AND LEAVING THE HEATING COIL SECTION UNITS - DEGREES FAHRENHEIT
12 DESIGN HEATING CAPACITY OF THE SYSTEM UNITS - BRITISH THERMAL UNITS PER HOUR

TABLE 3.10-3

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Student Union - BSU

RATING OF SYSTEM 19285 (SCFM) 938880 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 24X 102/144 = 17 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 19285 / 17 = 1134 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 45 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 938880 /(500)X(50) = 37.6 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WS TYPE COIL WITH 2,3,4 ROWS = 16

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 37.6 / 16 = 2.35

7) $\Delta T_A/\Delta T_{inlet}$ = 45 / 170 (°F) - 63 = 45 / 107 = .421

$\Delta T_W/\Delta T_A$ = 50 / 45 = 1.11

FROM FIGURE 8 : M_t = .83

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 3 /(83 X 11.3) = .314

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}} / 2$ = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .123

$R_{f2} = R_{ft} - R_{f1}$ = .319 - .123 = .196

FROM FIGURE 6 : FIN SERIES = 10 R_{f2} = .196

COIL MODEL NUMBER BECOMES: 5WS-1003C-24x102 NUMBER OF COILS REQ'D 1

9) FROM FIGURE 9 : ΔP_A = 2.3 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(19285)X(2.3) = 6.96

10) FROM FIGURE 12 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(.48 + 1.2) =

1.81 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = .79 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.10-4

NEW FAN SPEED ESTIMATE SHEET

BUILDING Student Union - BSU

RATING OF SYSTEM 19285 (SCFM) 938880 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$FAN\ BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(\ 19285\ )X(\ 1.75\ ) =$ 5.30

$FAN\ BHP_2 = (FAN\ BHP_1)+(FAN\ BHP_+) = (\ 5.30\ )+(\ 6.96\ ) =$ 12.26

$RPM_2 = (RPM_1)X(FAN\ BHP_2/FAN\ BHP_1)^{1/3} = (\ 980\ )X(\ 12.26/\ 5.30\ )^{1/3} =$ 1296

CHANGE IN FAN SPEED = $(RPM_2 - RPM_1)/RPM_1 =$ .244 OR X 100 = 24.4 % INCREASE

DEFINITIONS:

$FAN\ BHP_1$ = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

$FAN\ BHP_+$ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9,FROM THE COIL SELECTION SHEET).

$FAN\ BHP_2$ = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

TABLE 3.10-5

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Student Union - BSU

RATING OF SYSTEM 23125 (SCFM) 703920 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 30 X 99/144 = 20.625 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 23125 /20.625 = 1121 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 28 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 702920 /(500)X(50) = 28.2 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WH TYPE COIL WITH 1,2,3,4 ROWS = 10

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 28.2 / 10 = 2.82

7) $\Delta T_A/\Delta T_{inlet}$ = 28 / 170 (°F) - 63 = 28 / 107 = .262

$\Delta T_W/\Delta T_A$ = 50 / 28 = 1.78

FROM FIGURE 7 : M_t = .480

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 2 /(.48 X 11.21) = .372

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}} / 2$ = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .110

$R_{f2} = R_{ft} - R_{f1}$ = .372 - .110 = .262

FROM FIGURE 6 : FIN SERIES = 10 R_{f2} = .230

COIL MODEL NUMBER BECOMES: 5WH-1002C-30x99 NUMBER OF COILS REQ'D 1

9) FROM FIGURE 9 : ΔP_A = 1.28 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(23125)X(1.28) = 4.65

10) FROM FIGURE 11 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(1.3 + 1.7) =

3.24 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 1.40 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.10-6

NEW FAN SPEED ESTIMATE SHEET

BUILDING Student Union - BSU

RATING OF SYSTEM 23125 (SCFM) 703920 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$$\text{FAN BHP}_1 = (.000157)X(\text{SCFM})X(\text{SP}_1) = (.000157)X(\underline{23125})X(\underline{1.75}) = \underline{6.35}$$

$$\text{FAN BHP}_2 = (\text{FAN BHP}_1)+(\text{FAN BHP}_+) = (\underline{6.35})+(\underline{4.65}) = \underline{11.00}$$

$$\text{RPM}_2 = (\text{RPM}_1)X(\text{FAN BHP}_2/\text{FAN BHP}_1)^{1/3} = (\underline{888})X(\underline{11.0}/\underline{6.35})^{1/3} = \underline{1067}$$

CHANGE IN FAN SPEED = $(\text{RPM}_2 - \text{RPM}_1)/\text{RPM}_1$ = .202 OR X 100 = 20.2 % INCREASE

DEFINITIONS:

FAN BHP_1 = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

FAN BHP_+ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9, FROM THE COIL SELECTION SHEET).

FAN BHP_2 = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

TABLE 3.10-7

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Student Union - BSU

RATING OF SYSTEM 20725 (SCFM) 890400 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 24 X 102/144 = 17 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 20725 / 17 = 1219 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 40 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 890400 /(500)X(50) = 35.6 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WL TYPE COIL WITH 3,4 ROWS = 12

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 35.6 / 12 = 2.97

7) $\Delta T_A/\Delta T_{inlet}$ = 40 / 170 (°F) - 63 = 40 / 107 = .374

$\Delta T_W/\Delta T_A$ = 50 / 40 = 1.25

FROM FIGURE 8 : M_t = .71

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 3 /(.71 X 12.19) = .347

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}} / 2$ = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .105

$R_{f2} = R_{ft} - R_{f1}$ = .347 - .105 = .242

FROM FIGURE 6 : FIN SERIES = 10 R_{f2} = .210

COIL MODEL NUMBER BECOMES: 5WL-1003C-24x102 NUMBER OF COILS REQ'D 1

9) FROM FIGURE 9 : ΔP_A = 2 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(20725)X(2) = 6.51

10) FROM FIGURE 12 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(.5 + 2.3) =

3.02 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 1.31 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.10-8

NEW FAN SPEED ESTIMATE SHEET

BUILDING Student Union - BSU

RATING OF SYSTEM 20725 (SCFM) 890400 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$FAN\ BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(20725)X(1.75) = 5.69$

$FAN\ BHP_2 = (FAN\ BHP_1)+(FAN\ BHP_+) = (5.69)+(6.51) = 12.20$

$RPM_2 = (RPM_1)X(FAN\ BHP_2/FAN\ BHP_1)^{1/3} = (1020)X(12.20/5.69)^{1/3} = 1315$

CHANGE IN FAN SPEED = $(RPM_2 - RPM_1)/RPM_1$ = .289 OR X 100 = 28.9 % INCREASE

DEFINITIONS:

$FAN\ BHP_1$ = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

$FAN\ BHP_+$ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9,FROM THE COIL SELECTION SHEET).

$FAN\ BHP_2$ = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

TABLE 3.10-9

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Student Union - BSU

RATING OF SYSTEM 11025 (SCFM) 535440 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 18 X 84 /144 = 10.5 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 11025 / 10.5 = 1050 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 45 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 535440 /(500)X(50) = 21.4 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WL TYPE COIL WITH 3,4 ROWS = 9

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 21.4 / 9 = 2.38

7) $\Delta T_A/\Delta T_{inlet}$ = 45 / 170 (°F) - 63 = 45 / 107 = .421

$\Delta T_W/\Delta T_A$ = 50 / 45 = 1.11

FROM FIGURE 8 : M_t = .83

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 3 /(.83 X 10.5) = .344

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}}$ / 2 = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .125

$R_{f2} = R_{ft} - R_{f1}$ = .344 - .125 = .219

FROM FIGURE 6 : FIN SERIES = 12 R_{f2} = .190

COIL MODEL NUMBER BECOMES: 5WL-1203C-18x84 NUMBER OF COILS REQ'D 1

9) FROM FIGURE 9 : ΔP_A = 1.71 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(11025)X(1.71) = 2.96

10) FROM FIGURE 12 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(0 + 1.2) =

1.30 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = .562 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.10-10

NEW FAN SPEED ESTIMATE SHEET

BUILDING Student Union - BSU

RATING OF SYSTEM 11025 (SCFM) 535440 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$FAN\ BHP_1 = (.000157)X(SCFM)X(SP_1) = (.000157)X(\ 11025\ )X(\ 1.6\ ) =$ 2.77

$FAN\ BHP_2 = (FAN\ BHP_1)+(FAN\ BHP_+) = (\ 2.77\ )+(\ 2.96\ ) =$ 5.73

$RPM_2 = (RPM_1)X(FAN\ BHP_2/FAN\ BHP_1)^{1/3} = (\ 850\ )X(\ 5.73\ /\ 2.77)^{1/3} =$ 1083

CHANGE IN FAN SPEED = $(RPM_2 - RPM_1)/RPM_1 =$ .274 OR X 100 = 27.4% INCREASE

DEFINITIONS:

$FAN\ BHP_1$ = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

$FAN\ BHP_+$ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9,FROM THE COIL SELECTION SHEET).

$FAN\ BHP_2$ = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

TABLE 3.10-11

HOT WATER FIN TUBE COIL SELECTION SHEET

BUILDING Student Union - BSU

RATING OF SYSTEM 5700 (SCFM) 351360 (BTU/HR) NUMBER OF SYSTEMS (1)

USING Mc QUAY COILS:

1) COIL (W X L)/144 = 33 X 51/144 = 11.7 FACE AREA (SF)

2) (SCFM)/FACE AREA (SF) = 5700 / 11.7 = 488 FACE VELOCITY (FPM)

3) $\Delta T_A = T_{A_{out}} - T_{A_{in}}$ = 57 (°F) 4) ASSUME ΔT_W = 50 (°F)

5) RATED (BTU/HR)/(500)X(ΔT_W) = 351360 /(500)X(50) = 14.1 (GPM) OF WATER

6) FROM TABLE 1:

NUMBER OF FEEDS FOR A 5WH TYPE COIL WITH 1,2,3,4 ROWS = 11

GPM/FEED = (GPM) OF WATER/NUMBER OF FEEDS = 14.1 / 11 = 1.28

7) $\Delta T_A/\Delta T_{inlet}$ = 57 / 170 (°F) - 63 = 57 / 107 = .533

$\Delta T_W/\Delta T_A$ = 50 / 57 = .877

FROM FIGURE 8 : M_t = 1.17

8) R_{ft} = NUMBER OF ROWS/(M_t)X(FACE VELOCITY/100) = 3 /(1.17 X 4.88) = .525

$T_{W_{ave}} = T_{W_{in}} + T_{W_{out}}$ / 2 = 170 (°F) + 120 / 2 = 145 (°F)

FROM FIGURE 3 : R_{f1} = .185

$R_{f2} = R_{ft} - R_{f1}$ = .525 - .185 = .340

FROM FIGURE 6 : FIN SERIES = 10 R_{f2} = .340

COIL MODEL NUMBER BECOMES: 5WH-1003C-33x51 NUMBER OF COILS REQ'D 1

9) FROM FIGURE 9 : ΔP_A = .42 (INCHES OF WATER)

ADDITIONAL FAN BRAKE HORSEPOWER = (.000157)X(SCFM)X(ΔP_A) =

(.000157)X(5700)X(.42) = .376

10) FROM FIGURE 12 : CORRECTION FACTOR = 1.08

ΔP_W = (CORRECTION FACTOR)X($\Delta P_h + \Delta P_t$) = (1.08)X(0 + 0) =

0 (FEET OF WATER)

ΔP_W = (.4335)X(FEET OF WATER) = 0 (PSI)

* Reference APPENDIX "A" for definitions of terms and procedures used.

TABLE 3.10-12

NEW FAN SPEED ESTIMATE SHEET

BUILDING Student Union - BSU

RATING OF SYSTEM 5700 (SCFM) 351360 (BTU/HR) NUMBER OF SYSTEMS (1)

Since the order of magnitude of change in speed and power requirements relative to the existing fan operating conditions is all that is required, the following formulas pretaining to fans in general were used. Actual fan operating curves published by the respective fan manufacturers for the installed models can be consulted to determine more exact values.

$$\text{FAN BHP}_1 = (.000157)\text{X}(\text{SCFM})\text{X}(\text{SP}_1) = (.000157)\text{X}(5700)\text{X}(1.6) = 1.43$$

$$\text{FAN BHP}_2 = (\text{FAN BHP}_1)+(\text{FAN BHP}_+) = (1.43)+(.38) = 1.81$$

$$\text{RPM}_2 = (\text{RPM}_1)\text{X}(\text{FAN BHP}_2/\text{FAN BHP}_1)^{1/3} = (847)\text{X}(1.81/1.43)^{1/3} = 917$$

CHANGE IN FAN SPEED = $(\text{RPM}_2 - \text{RPM}_1)/\text{RPM}_1$ = .083 OR X 100 = 8.3 % INCREASE

DEFINITIONS:

FAN BHP_1 = THE ORIGINAL HORSEPOWER REQUIRED TO PRODUCE THE RATED AIR FLOW AT THE RATED STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

FAN BHP_+ = THE ADDITIONAL HORSEPOWER REQUIRED TO OVERCOME THE RESISTANCE INTRODUCED BY THE ADDED FIN TUBE HOT WATER COILS (STEP 9, FROM THE COIL SELECTION SHEET).

FAN BHP_2 = THE NEW HORSEPOWER REQUIREMENT TO PRODUCE THE RATED AIR FLOW AT THE NEW STATIC PRESSURE WITHOUT ALLOWANCES FOR MOTOR EFFICIENCIES OR MECHANICAL LOSSES.

SP_1 = THE RATED AIR STATIC PRESSURE ABOVE AMBIENT PRODUCED BY THE FAN AT THE RATED AIR FLOW UNDER ORIGINAL DESIGN CONDITIONS.

SCFM = RATED AIR FLOW OF THE FAN IN STANDARD AIR CONDITIONS, CUBIC FEET PER MINUTE.

RPM_1 = THE ORIGINAL FAN SPEED IN REVOLUTIONS PER MINUTE.

RPM_2 = THE NEW REQUIRED FAN SPEED IN REVOLUTIONS PER MINUTE.

(.000157)= CONVERSION FACTOR BASED ON STANDARD AIR CONDITION.

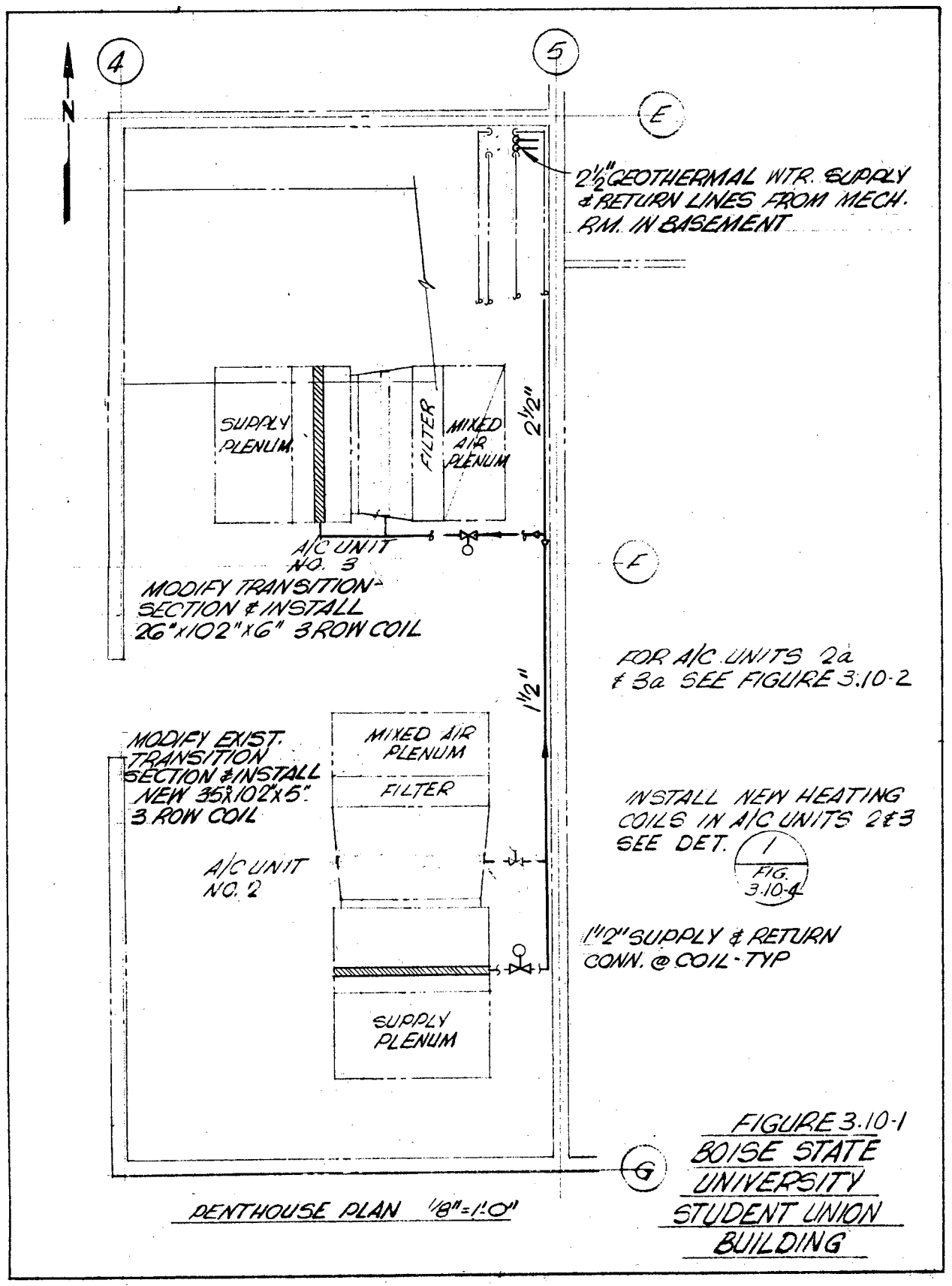


Figure 3.10-1 Boise State University Student Union Building

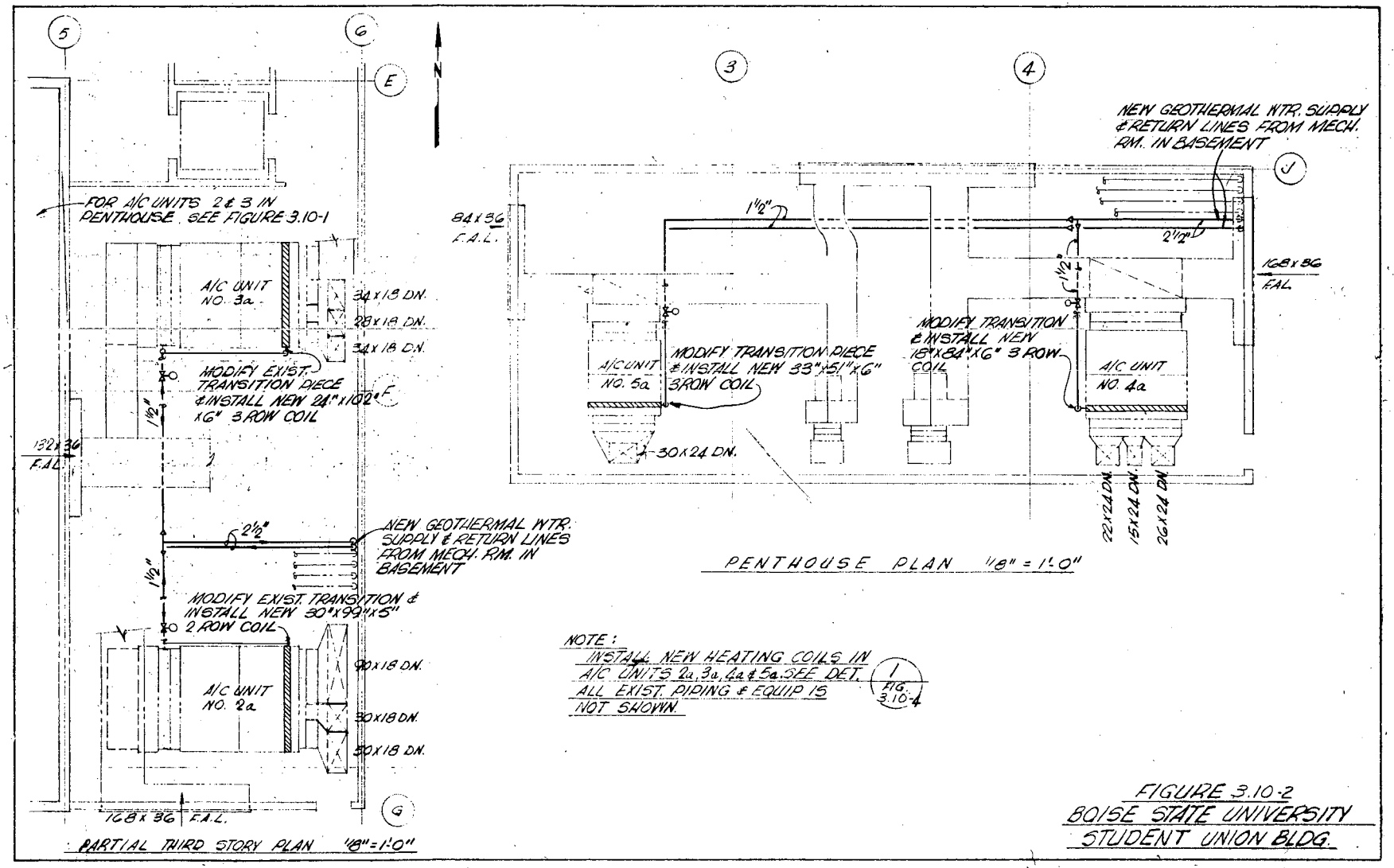


Figure 3.10-2 Boise State University Student Union Building

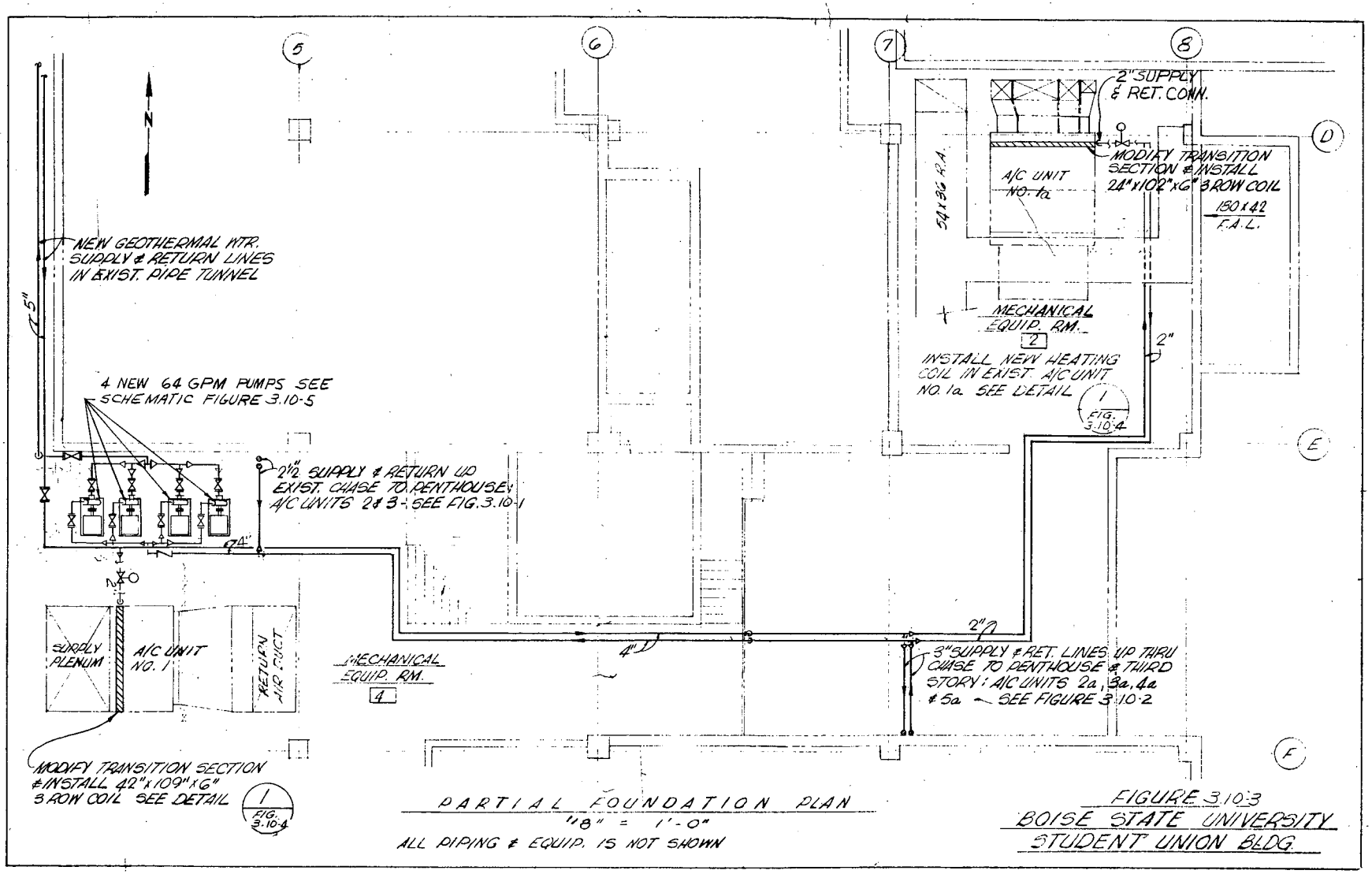


Figure 3.10-3 Boise State University Student Union Building

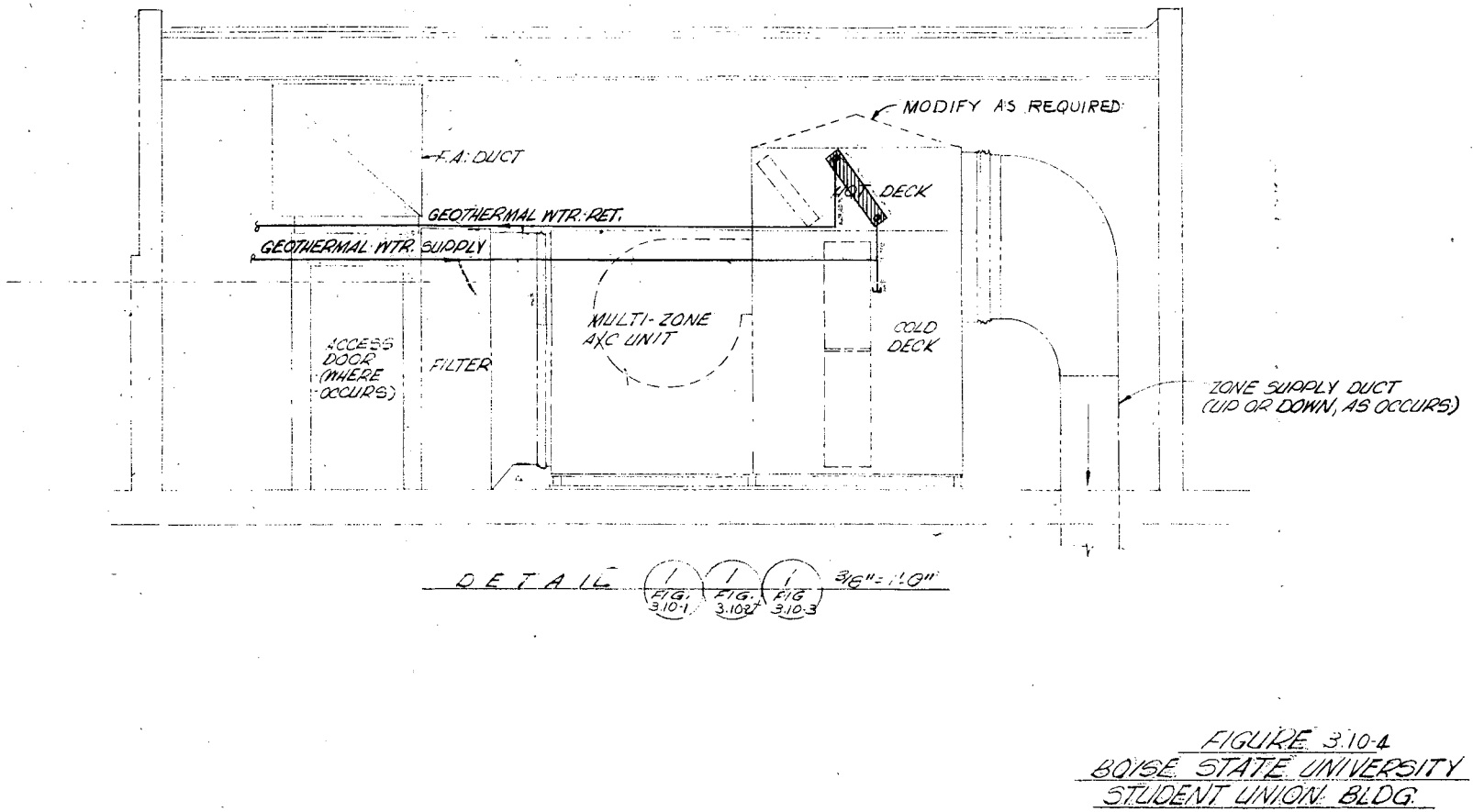


FIGURE 3.10-4
BOISE STATE UNIVERSITY
STUDENT UNION BLDG.

Figure 3.10-4 Boise State University Student Union Building

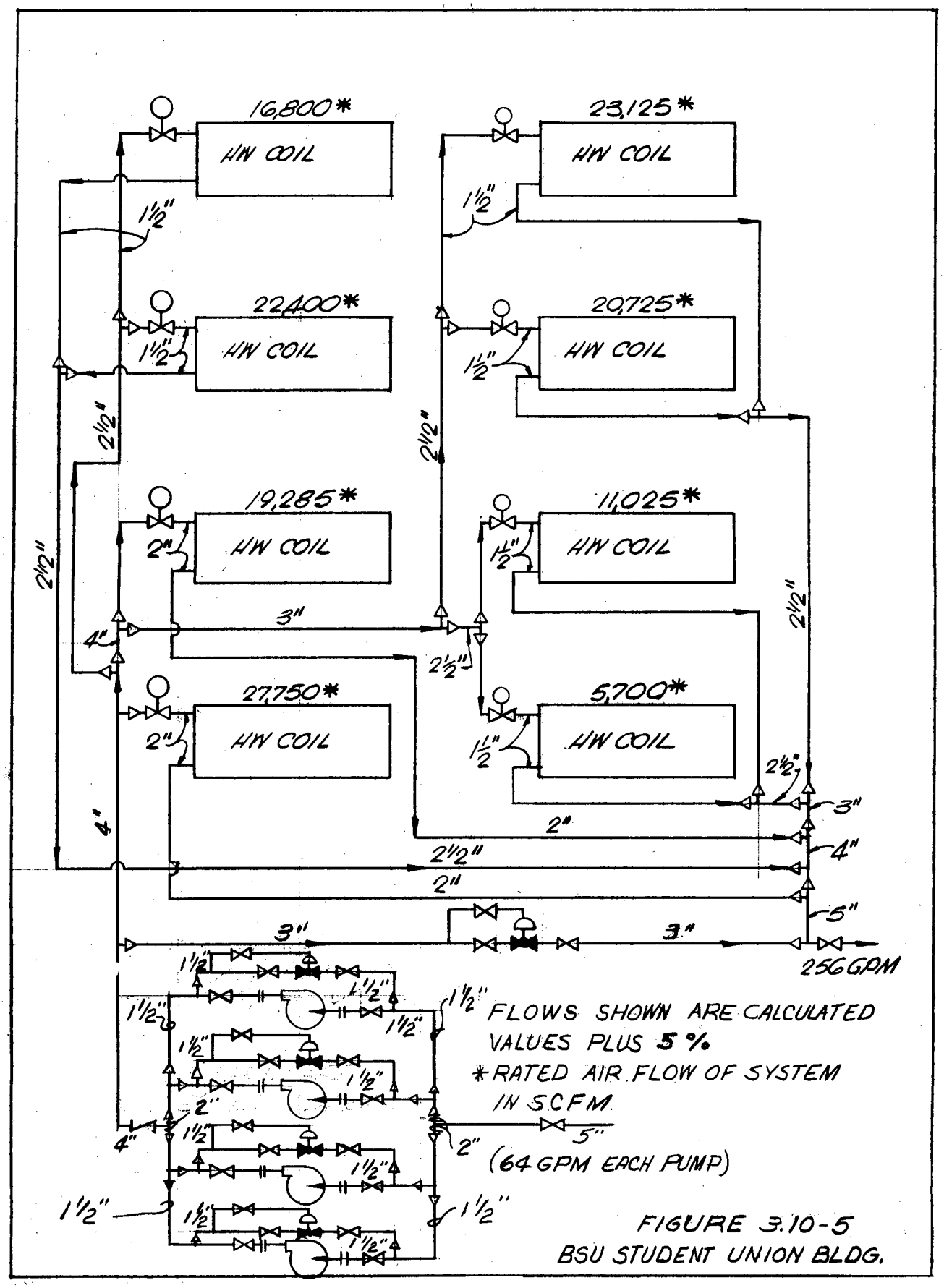


Figure 3.10-5 Boise State University Student Union Building

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APPENDIX A

The calculation steps in sizing hot water coils to equal an existing heating system capacity were divided into ten distinct steps as indicated on the fin tube coil selection sheet for each heating system. Figures and Tables in the following definitions refer to the attachments to this Appendix.

Step Definitions:

1. Using the coil width (W) and length (L) dimensions in inches from the building information sheet, the face area in square feet is determined.

2. For the design air flow in standard air conditions, cubic feet per minute (SCFM) and the face area determined for step 1, the average face velocity in feet per minute is determined.

3. The required change in air temperature (ΔT_A) through the coil is derived by subtracting the entering air temperature ($T_{A\ in}$) from the leaving air temperature ($T_{A\ out}$).

4. The change in geothermal water temperature (ΔT_W) through the coil is an assumed number and is 50°F for these calculations if present conditions will allow. This translates back to the 170°F geothermal water temperature assumed minus 50°F for a 120°F leaving coil water temperature.

5. A geothermal water flow rate in gallons per minute through the coil to produce an equivalent heating capacity is determined by dividing the rated heating capacity in (Btu/hr) by 500 times the assumed change in water temperature (ΔT_W).

6. Using the width dimension (W) in inches from step 1, a type of coil and number of rows selection is made from Table 1. This selection leads to the number of tube feeds per coil and is used to determine the water flow rate per feed by dividing the gallon per minute value determined under step 5 by the number of feeds.

7. The ratio of change in air temperature (ΔT_A) to the inlet temperature differences (ΔT_{inlet} = temperature of the hot water entering minus the temperature of the entering air) is used along with the ratio of change in water temperature (ΔT_w) to the change in air temperature (ΔT_A) to select the heat transfer value (M_t). This M_t value is obtained from either Figure 7 or 8 depending upon the number of tube rows assumed under item 6.

8. After selecting M_t, it is used in the formula R_{ft} = Number of Rows/M_t x (Face Vel./100) to obtain the total heat transfer value (R_{ft}). This formula uses from step 6 the number of rows assumed and from step 2 the calculated face velocity. After determining the average water temperature through the coil ($T_{w\ ave}$), from Figure 3, a heat transfer value (R_{f1}) is selected. This value is used in turn to calculate an R_{f2} value by subtracting R_{f1} from R_{ft}. With this value of R_{f2} a fin series number is obtained from Figures 4, 5 or 6 depending on the type coil assumed under step 6. If a fin series number for an R_{f2} value equal to or less than the calculated value cannot be selected, then another coil type must be assumed and steps 6 through 8 repeated.

9. Using the face velocity, fin series number and the number of rows, from Figure 9 an air pressure drop (ΔP_A) through the coil can be selected. This is in inches of water units. The additional fan horsepower to overcome this pressure is calculated by multiplying the air flow rate (SCFM) by the pressure drop (ΔP_A) in inches of water by .000157.

10. From Figure 10, 11 or 12 a correction factor, header pressure drop (ΔP_h) and tube pressure drop (ΔP_t) are selected. The total water pressure drop (ΔP_w) is determined by multiplying the correction factor by the sum of the pressure drops ($\Delta P_h + \Delta P_t$). This ΔP_w value in feet of water units is converted to pounds per square inch units by multiplying by .4335.

– STANDARD WATER COIL CIRCUITING

FOR TYPE "5W" AND "5B"

NUMBER OF TUBES FED

TYPE	ROWS	"W" DIMENSION (INCHES)												
		6	9	12	15	18	21	24	27	30	33	36	39	42
5BB, 5BS	1, 2	1	1	1	1	1	1	1	–	–	–	–	–	–
5BD	2	2	2	2	2	2	2	2	–	–	–	–	–	–
5WB, 5WQ	1	–	–	2	2½	3	3½	4	4½	5	5½	6	6½	7
5WH	1, 2, 3, 4	–	–	4	5	6	7	8	9	10	11	12	13	14
5WB	2	–	–	4	5	6	7	8	9	10	11	12	13	14
5WL	3, 4	–	–	6	7	9	10	12	13	15	16	18	19	21
5WS	2, 3, 4	–	–	8	10	12	14	16	18	20	22	24	26	28
5WM	4	–	–	12	15	18	21	24	27	30	33	36	39	42
5WD	4	–	–	16	20	24	28	32	36	40	44	48	52	56

NOTES: a. Supply and return connections are on opposite ends of 3 row Type "5WS" Coils. All others have connections on the same end.

b. Type "5BB", "5BD" and "5BS" Coils are not available as standard over 24 inch "W" and 60 inch "FL" dimensions.

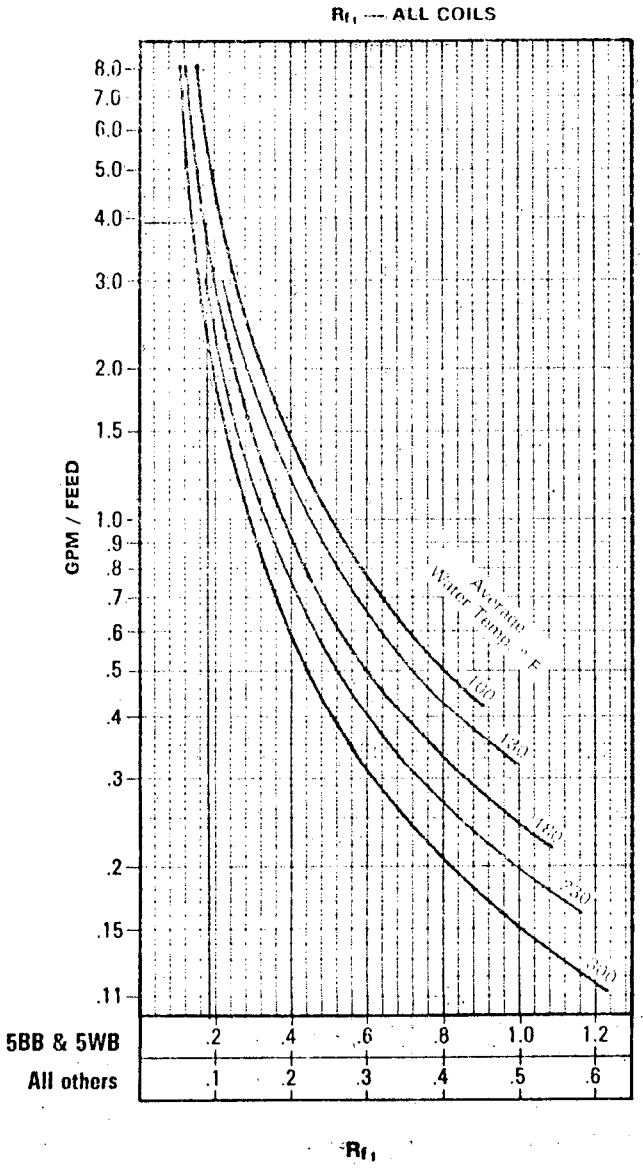
c. Type "5BB" and "5BS" 2 row coils are not drainable when installed for vertical airflow.

d. When ordering Type "5WH", "5WL", "5WS" and "5WM" 3 and 4 row coils, the direction of airflow, horizontal or vertical, must be specified for drainable circuiting.

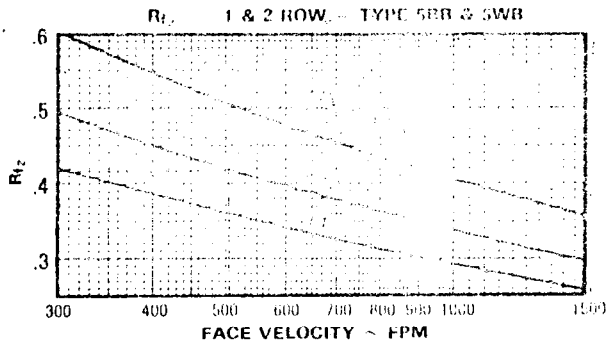
ARI certifies data for horizontal airflow only.

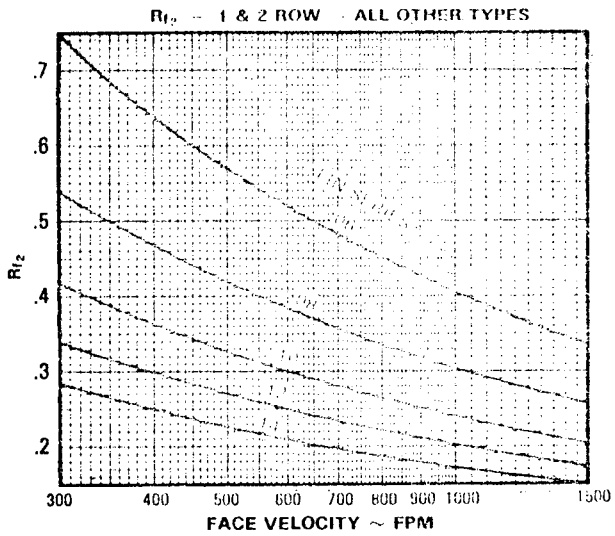
e. When ordering Type "5WM" and "5WD" 4 row coils; the hand of coil, right or left, must be specified.

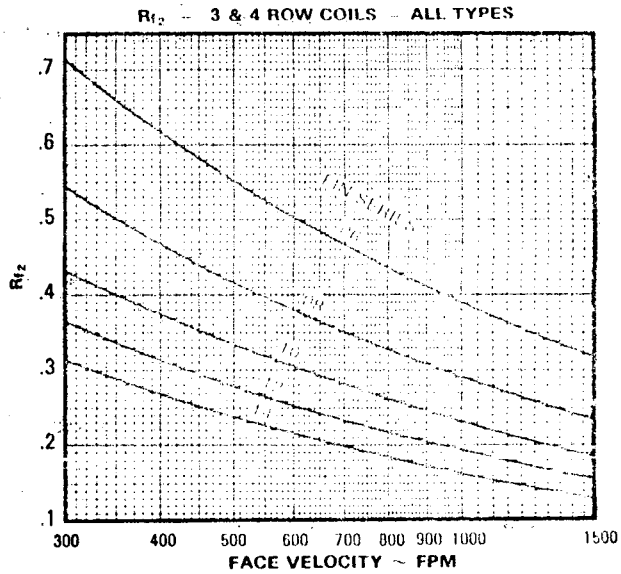
HEAT TRANSFER VALUES — R_f



$R_{f_t} = R_{f_1} + R_{f_2}$

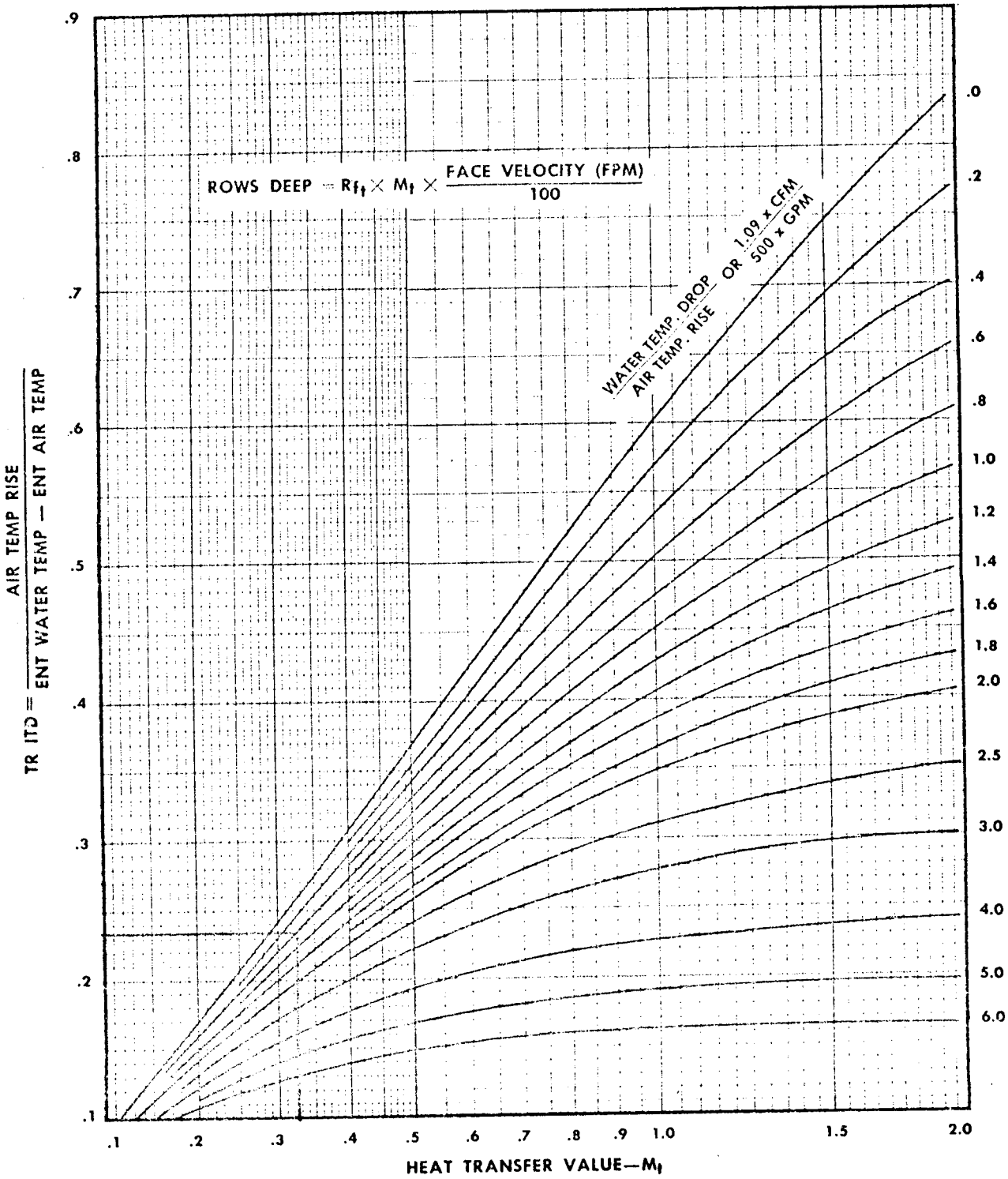






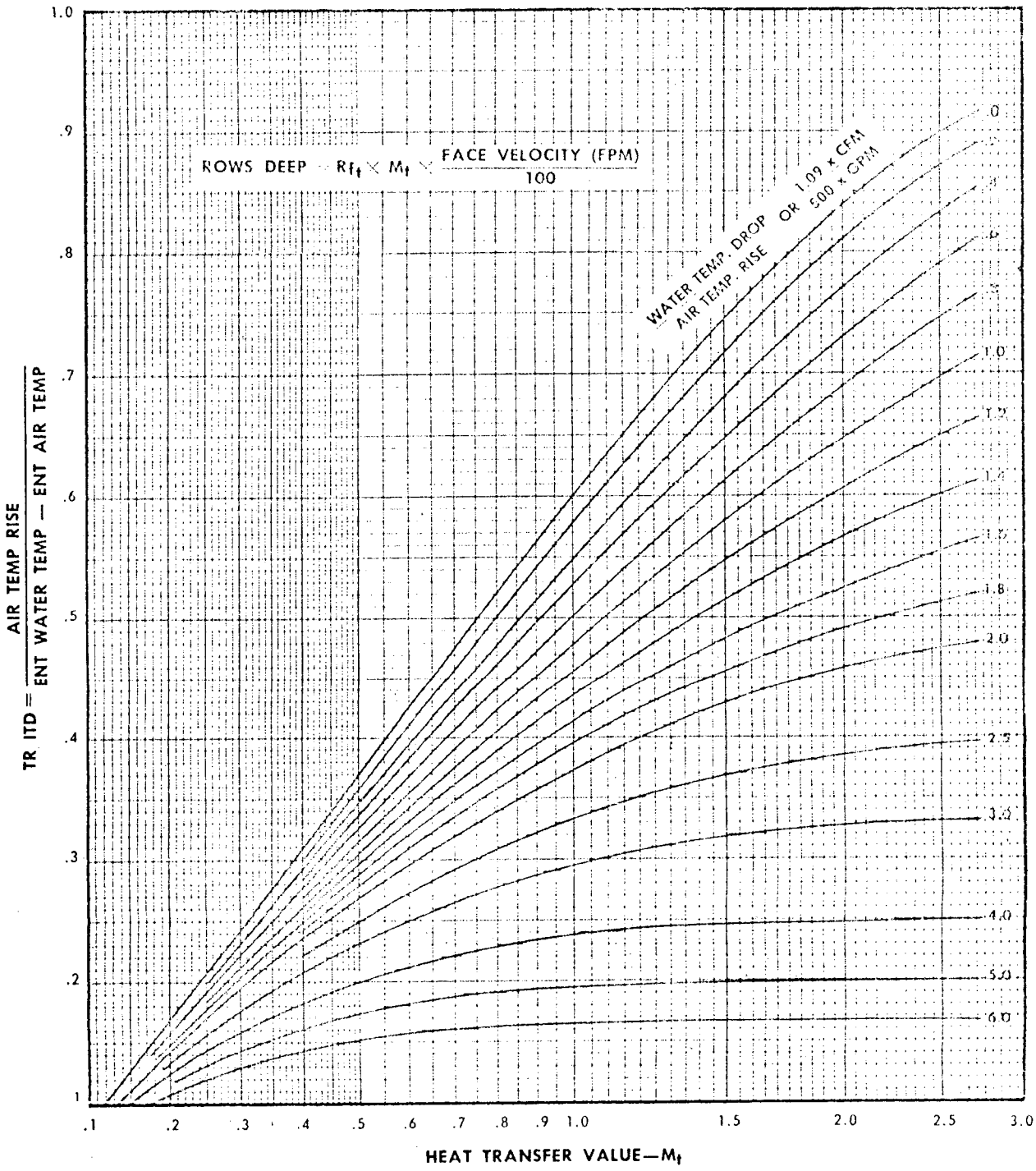
HEAT TRANSFER VALUES—M_t

1 & 2 ROW WATER HEATING COILS

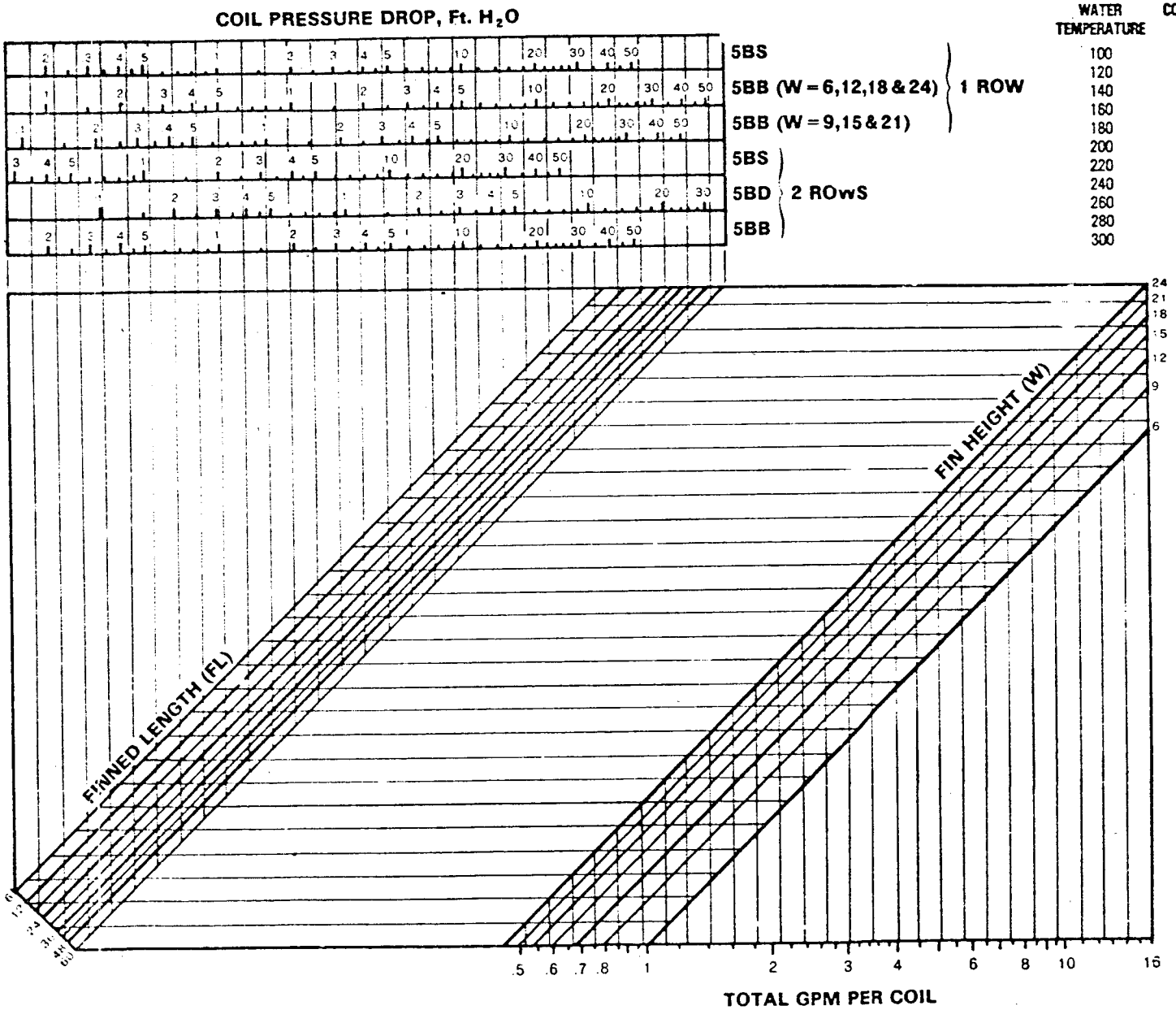


HEAT TRANSFER VALUES—M_t

3 & 4 ROW WATER HEATING COILS

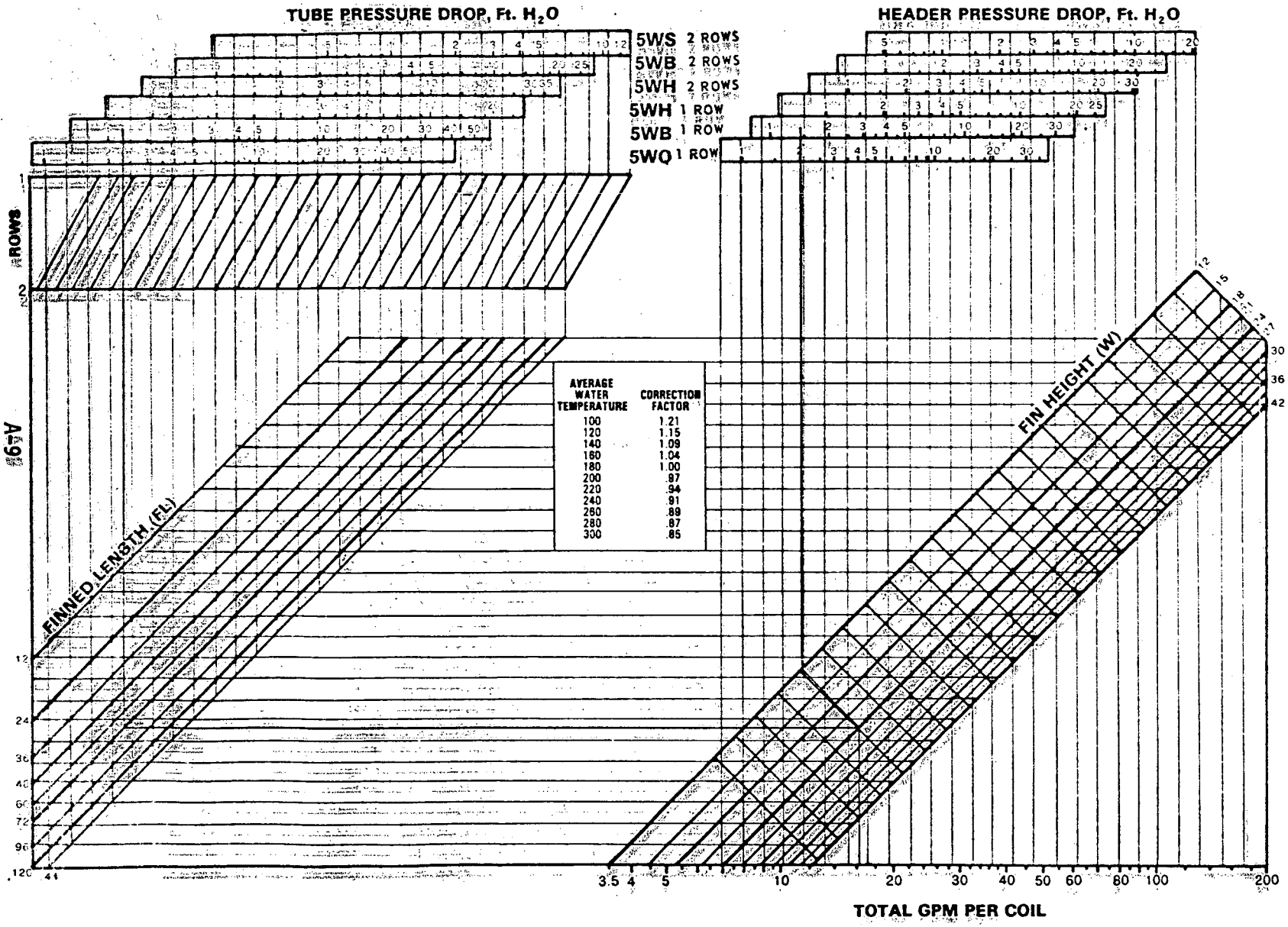


WATER PRESSURE DROP (5B COILS)



AVERAGE WATER TEMPERATURE	CORRECTION FACTOR
100	1.21
120	1.15
140	1.09
160	1.04
180	1.00
200	.97
220	.94
240	.91
260	.89
280	.87
300	.85

WATER PRESSURE DROP (5W — 1 & 2 ROW COILS)



AVERAGE WATER TEMPERATURE	CORRECTION FACTOR
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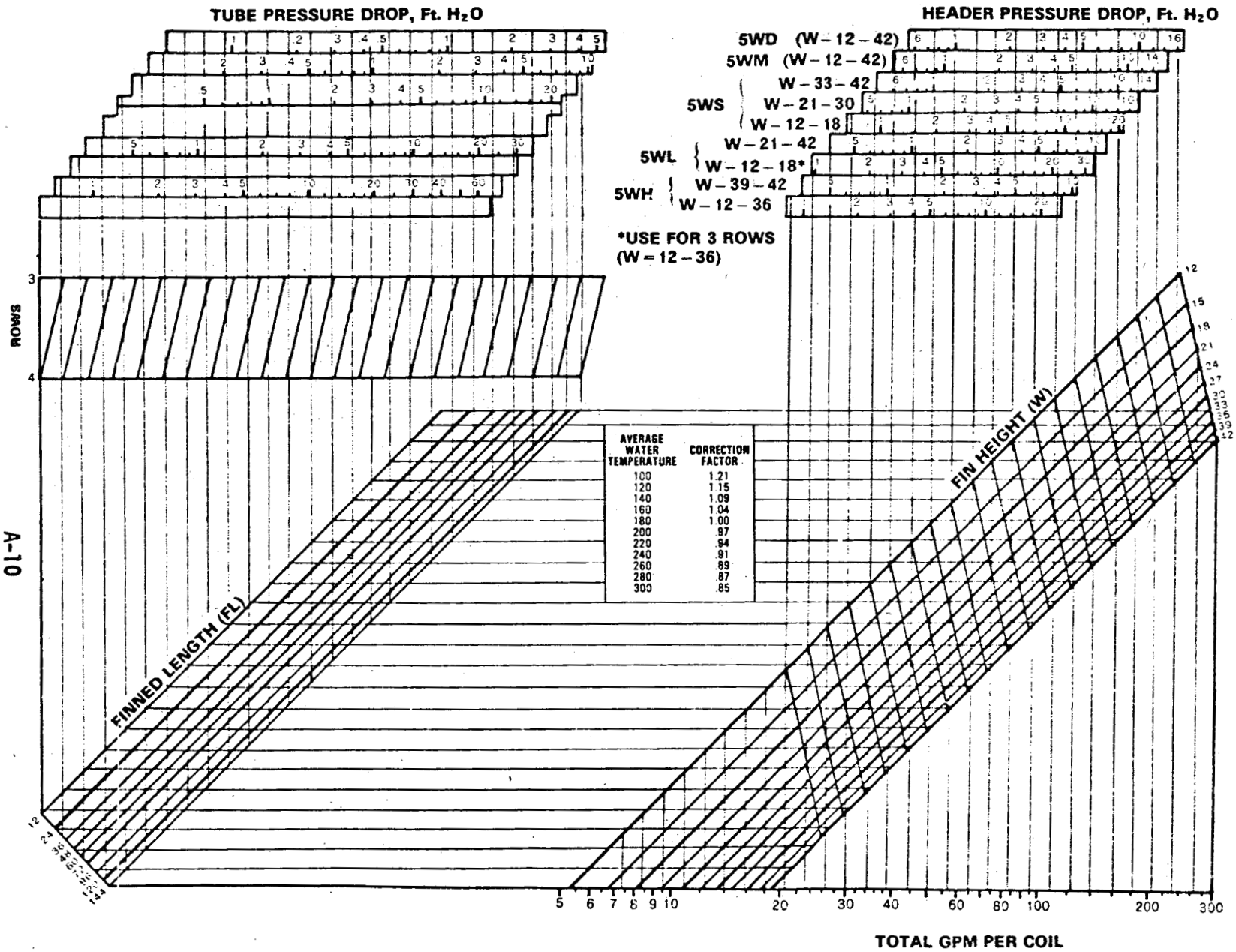
140	1.09
160	1.04
180	1.00
200	.97
220	.94
240	.91
260	.89
280	.87
300	.85

WATER PRESSURE DROP (5W — 3 & 4 ROW COILS)



AVERAGE WATER TEMPERATURE	CORRECTION FACTOR
100	1.21
120	1.15
140	1.09
160	1.04
180	1.00
200	.97
220	.94
240	.91
260	.89
280	.87
300	.85

GLOSSARY

(SCFM) = Standard Air Conditions, Cubic Feet Per Minute Standard Air Conditions Is Defined As Air At 70°F, 29.921 Inches Of Mercury And .07488 Pounds Per Cubic Feet.

(W x L) = Width x Length In Inches.

(W x L x D) = Width x Length x Depth In Inches.

(SF) = Square Feet

(FPM) = Feet Per Minute

(GPM) = Gallons Per Minute

(BTU/hr) = British Thermal Units Per Hour

(°F) = Degrees Fahrenheit

(HP) = Horsepower

(HVAC) = Heating Ventilating Air Conditioning

(RPM) = Revolutions Per Minute

(in) = Inches

(ft) = Feet

(dia) = Diameter

(ARI) = American Refrigeration Institute

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