AIR CONDITIONING McNARY HYDROELECTRIC POWERHOUSE WITH HEAT PUMPS

by

ROLAND ELWOOD OTT

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APPROVED:

Redacted for Privacy

Professor of Mechanical Engineering Department

In Charge of Major

Redacted for Privacy

Head of Mechanical Engineering Department

Redacted for Privacy

Chairman of School Graduate Committee

Redacted for Privacy

Dean of Graduate School

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CHAPTER I

INTRODUCTION

AIR CONDITIONING McNARY HYDROELECTRIC POWERHOUSE WITH HEAT PUMPS

The air conditioning system which was designed for the McNary powerhouse embodies several unique features which shall be discussed in this paper.

Heat pumps with generator cooling water as the heat source have been used to heat as well as to cool the generator room and other areas of a hydroelectric powerhouse. This installation is probably the first of its kind.

Air conditioning is not entirely new; however, wide use of the advantages of conditioning air has been dependent upon the availability of electric power; therefore, it has followed the development of hydroelectric power which has occurred, in the United States, during the last seventy years.

"The first electric lighting plant with water power as the prime mover was built at Appleton, Wisconsin, in 1882, with a capacity of 250 cp." (10, p.612)

There has been an ever-increased demand for electrical power. Of the potential of 69 millions of water horse power available in the North American continent, approximately 22 millions of water horse power has been developed. (24, p.2)

To partially ease the increasing pressure for more electric power, the McNary Dam and Powerhouse was started in 1947, and it will be completed in 1957. The 14 generators will have a total capacity of 980,000 kilowatts. This project which cost $280,000,000 will have an air conditioned powerhouse. (35, p.22)

The spillway and powerhouse stretch across the Columbia river at the former Umatilla rapids from the Washington shore, south to the Oregon bank of the river.

The powerhouse which adjoins the Oregon shore, is 1422 feet long. Windowless construction was utilized in the design of the main generator room with its beneficial effect for summer cooling and winter heating. Windows were used, however, in the offices which created a special problem for air conditioning this area.

The roof, which averages 77 feet above the generator floor, contains a one-inch insulation barrier. Massive concrete walls, which enclose the generators, create a large thermal lag that reduces the maximum cooling load.

The gigantic generators, 51'-8" in diameter, lose approximately 3.3 per cent of their 70,000 kilowatt capacity in the form of heat which is caused by friction, windage and electrical losses. This heat is removed by circulating water through coils which cool large volumes of air which is recirculated within the generator casing.

This warm water, which would endlessly discharge heat into the mighty Columbia river, was selected as a topic for a study to determine whether a portion of this energy might be reclaimed.

As a result of this study, the heat pump was selected as the prime mover of this energy, and in addition to heating, cooling capacity was also provided for the powerhouse.

Since it is desirable not only to describe but also to compare the factors of a problem with known data, a research for information was made so that a comparison could be made between accepted standards and the various parts of the McNary air conditioning system. This study logically began with the early beginnings of air conditioning.

A plan to air condition a building in a tropical country was published, in 1852, by Dr. Piazza Smith. He proposed to take the excess moisture out of the air by compressing it and extracting heat from it. The compressed air was to be allowed to expand which would cool it, and the chilled air was to be circulated. This system included the essential features for summer air conditioning. (3, p.117)

The elements of air conditioning date back almost two centuries before the plans made by Dr. Smith. On June 30, 1666, Robert Boyle commenced the first regular psychrometric observations at Oxford. (13, p.65)

In the year of 1777, M. Bonnemain is credited with the first use of hot water as a heating medium. (12, p.5)

Saussure discovered, in 1786, the necessity of whirling a wet bulb thermometer to get a true reading. (13, p.65)

"John Dalton, in the third of four important papers contributed to the Manchester Society in 1801, first announced that water vapor exists mixed with air and that evaporation is proportional to the temperature, whether in air or in vacua." (13, p.65)

It was Apjohn of Dublin who in 1835 disclosed the fact that the wet-bulb temperature was an indication of total heat.

The first comprehensive psychrometric tables were published by James Glaisher in London in 1847. (13, p.65)

The foundation of air conditioning as it is known today is credited to William H. Carrier who, on December 8, 1911, presented a paper before the American Society of Mechanical Engineers on his "... Rational Psychrometric Formulae". This occasion is regarded as the first recognition of air conditioning by the engineering profession, and 1936 "... marks its first general acceptance as an important modern convenience by the public". (4, p.42)

The complete air conditioning of a building is listed by G. A. Van Brunt as "... the simultaneous control of temperature, humidity, motion, distribution, dust, bacteria, odors, and toxic gases". (45, p.114)

These factors will be discussed in this paper; however, bacteria control will only be referred to as it relates to dust. The control of toxic gases will be limited to the ventilation of the battery room, oil storage room, and the gate repair pits.

CHAPTER II

WHY AIR CONDITION POWERHOUSES?

A modern hydroelectric powerhouse hums with the whirling of giant generators and auxiliary electrical equipment which is constantly controlled by delicate instruments. If they are to function properly, the atmosphere must be kept at relatively even temperatures and humidity, with sufficient motion to insure proper air distribution and be free from dust.

The health of the personnel should also be considered as a factor in the operation of a large installation. Human comfort which can be provided by the use of air conditioning can partially offset the disadvantages of working in a windowless building.

It was the development of fluorescent lighting that was instrumental in the successful design of buildings without windows. The use of sufficient incandescent lamps to light a windowless building was costly, and a large quantity of heat had to be removed by the air conditioning system. (46, p.130)

The McNary powerhouse was designed as a windowless building with the exception of the offices. A building of this type is slightly less expensive to erect than conventional ones, and they are also tight against air leakage. Without large heat gains as well as large heat losses, usually associated with generator-room windows, a windowless building can be air conditioned at a lower cost. (15, p.99)

Temperature control throughout the year is one of the most important factors of an air condition system. The temperature of the air surrounding small circuit breakers, rated up to 100 amps. at 600 volts, should be kept as constant as possible. This is to insure that they will function as accurately as they have been calibrated since the tripping mechanism responds to heat generated by the flow of an electric current. The testing of these breakers is performed in an air conditioned room which is held within $\pm$ 1°F at 77°D.B. (40, p.526)

"The essential application of air conditioning in automatic telephone exchanges is now quite an unchallenged principle. The open relays necessary for continuous adjustment of close contacts demand effective dust and temperature control." (7, p.280)

The relative humidity effects the operation of totalizing recorders. It was found at the Beauharnois powerhouse, which is located in Quebec, Canada, that during the winter season the relative humidity was as low as 15%. The galvanometer needle of the totalizing recorders retained a static charge; however, this was overcome by raising the relative humidity of the air in a localized area of the instrument. (Appendix A, 9)

Excess water vapor in the air which occurs during the summer months may cause condensation in the lower galleries.

"The ever present humidity in the powerhouse shows itself by condensation on cold water piping to the air-cooling units and to the bearing pots on the generator." (Appendix A, 9)

"Hydro-electric plants are often located in river valleys or gorges where humidity is normally higher than in other areas. The nights are usually cooler and days somewhat warmer than are found in the surrounding, more open country. Consequently an atmospheric condition arises which is conducive to the condensation of moisture on electrical equipment." (20, p.1054)

The presence of precipitated moisture accelerates the growth of microscopic organisms. Paint deteriorates; exposed metal surfaces are corroded; and electrical insulation breaks down. (17, p.321)

Ventilating air is used to carry this excess moisture out of the galleries; however, the dew point temperature of the circulated air must be maintained appreciably below the dry bulb temperature within the galleries.

Condensation on the walls of the control rooms, cable vaults and wherever switchboards and instruments are located may cause stray currents. (22, p.383)

"Oil filled cables, transformers, and disconnecting switches are endangered if their insulating oils absorb moisture." (22, p.383)

The air within a powerhouse should be relatively free from dust. Foreign bodies in the air enter a building with ventilating air, and they are also generated by the deterioration of materials within the building.

Atmospheric dust in the vicinity of the average powerhouse is a mixture of various organic and inorganic substances. Approximately one hundred-millionth of the air volume is made up of dust which contains "...coal, cinders, ashes, sand, loam, vegetable and animal fibers, animal tissue ..." and bacteria. (21, p.320)

Actual impurities which exist in any particular sample of air will vary with nearness to industrial sections of cities and sand storm areas.

"Accumulation of dust within electrical equipment has been the cause of many failures". This dust conducts electrical currents which cause the grounding of windings, one of the most frequent sources of trouble. Sparking which may be caused by the accumulation of dust can cause excessive maintenance of collector rings and commutators. (41, p.1)

All electrical equipment must be cleaned periodically, and this cost can be reduced if the atmosphere, surrounding the machines, is kept clean.

Gritty dust which enters the bearings is the principal cause for injury to machinery. (21, p.321)

Dust carried into an air compressor is a special problem; however, as an indication of the wear to mechanical parts caused by dust, reference is made to a test which was made with filtered and unfiltered air.

"Reductions as high as 80 per cent in maintenance costs have been shown by experience to be possible through installation of air filters on compressors and internal combustion engines, as compared to operation with unfiltered air." (8, p.454)

Dust also settles upon white walls and lighting fixtures which materially reduces the illumination. The maintenance cost of continually cleaning is considerable, and the frequency of painting which is required to keep the walls and ceilings bright can be reduced by removing the dust from the air before it enters the building as well as precipitating the dust which is generated from within. (21, p.321)

Safeguarding the health of employees increases the total productive effort of any group of workers. Air conditioning can help reduce part of the tremendous cost of the common cold; it has been estimated that in American factories, alone, a loss of 100 million working days occurs each year and costs up to two billion dollars. (14, p.384)

"The concensus of opinion among the medical profession at the present time is that probably more than 90 per cent of the cases of pulmonary tuberculosis are conveyed by the dust or infected air that we breathe or by the food or drink taken into the body." (27, p.574)

The human body handles approximately five times the weight of air than it does of food or drink which should indicate that the air quality is not an overrated commodity. (7, p.279)

Dr. Joseph H. Kler, M.D., of New Brunswick, New Jersey, reported in an article appearing in the Archives of Otonlaryngology, published by the American Medical Association, that there are fewer colds in air conditioned plants in contrast with non-air conditioned plants. This study was made over a two-year period. He said that the common cold was responsible for more than one-third of the total days lost in American factories. (14, p.384)

Dr. A. G. Young, who is the medical director of Corey Hill hospital in Brookline, Massachusetts, declared that "infective agents do not thrive in the respiratory tracts of people under ideal air conditions". Dr. Young has been in charge of experiments which have been conducted on "...the effects of air conditioning on health." (1, p.101)

Empirical tests have been made to relate time lost by personnel to the type of atmosphere in which they work, that is, whether the building was air conditioned or not.

It was reported by the Procter and Gamble Company, of Cincinnati, Ohio, that the loss of employees' services from colds and other respiratory illnesses, was reduced from an average of 2.2 days to 1.6 days during one year after the offices were air conditioned. (2, p.194)

A survey was made in several large buildings in various sections of the country, and it was concluded by the use of statistical records that a 28 per cent reduction in the illnesses which keep an employee from work could be attributed to the air conditioning of these buildings. (18, p.551)

A very comprehensive health survey was conducted at the Detroit Edison Co. which involved 1500 employees. The study was carried on for 140 weeks. Air conditioned as well as non-air conditioned quarters housed the personnel. The employees submitted 200,000 separate weekly reports.

The survey included several factors for comparative analysis as follows:

Number of colds or infections, severity and time lost from work because of the infections, seasonal differences and epidemics, differences in the occurrence of infections between men and women, and the influence of age on infections. Reliable mathematical procedures were used to test the results and to determine that differences were real.

"The most significant finding of the study on the basis of the survey was that each 100 employees in air conditioned quarters lost fewer days from work over the period of the study because of respiratory infections than did employees located in non-air conditioned quarters. However, this significant difference between the two groups was too small to warrant an unqualified statement in favor of air conditioning quarters. It was considered as evidence but not absolute proof." (47, p.677)

The reasons why a hydroelectric plant should be air conditioned have been discussed so that a better appreciation could be made of the air conditioning system which has been designed for the McNary powerhouse.

CHAPTER III

HEAT PUMPS & HEAT SOURCE

With the need for air conditioning of a hydroelectric powerhouse, found to be desirable, it is necessary to investigate the cost of a system which controls the temperature, humidity, air movement, and removal of dust from the air.

An investigation of the problem, "When is Complete Air conditioning of the Modern Factory Advisable," was conducted by H. A. Mosher. A graph is shown on page 306, in the June 1945 issue of "Heating Piping & Air Conditioning". The graph is used to compare complete air conditioning with a system in which only ventilation is considered. Both of the capital costs per thousand square feet, in dollars, and the annual operating costs per thousand square feet, in dollars, are plotted against the internal heat load in B.T.U. per hour per square foot which is designated as H. (33, p.p. 306-7)

From the graph it is noted that an air conditioning system can be justified for a building which has an internal gain greater than H = 5 over a ventilating system on economic grounds alone.

The reasons for this economic justification are as follows:

1. Heat from lights require larger volume of air for ventilation than for air conditioning. The first cost of

the two systems is almost proportional to the volumes of air circulated.

2. New air diffusers permit lower temperature air to be introduced without causing objectional drafts.

The exceptions to cost comparison are as follows:

1. Buildings with good natural ventilation have large volumes of naturally circulated air without added capital costs. The "stack effect" is used to advantage.

2. Areas with low internal heat gains (less than H = 5) require only a minimum of ventilation. (33, p.p. 306-7)

The internal heat load of the generator room at McNary is 1,654,300 B.T.U. per hour; the area of the generator room is 116,000 square feet. H, for this area at McNary, is as follows:

$$H = \frac{1,654,300}{116,000} = 14.26 \text{ B.T.U. per hour per square foot.}$$

A cost analysis of an air conditioning system made for a temporary building which housed the Abbott Laboratories is as follows: (28, p.72)

Original cost	\$20,000
Salvage value after 2 years	9,000
Net capital cost	\$11,000

The cost analysis of an air conditioning system for Abbott Laboratories is continued as follows:

Amortization, $11,000 for 2 yrs.	$5,500
" 9,000 " 15 yrs.	600
Interest, insurance & taxes 3% x $20,000	600
Maintenance, 40 tons @ $2.00/ton	80
Power & water, 1000 hr. @ $0.68	680
Total annual cost	$7,460

From the tabulation it is shown that the $20,000 investment was paid off in two years. Based upon a five-month cooling period, the cost would be $1,500 per month. With a monthly payroll of $40,000 and an average increase in employee efficiency of less than four per cent, this investment would break even. In an eight-hour day, an increase of four per cent would be only twenty minutes. (28, p.72)

Another installation is listed for which the actual cost of the operation of approximately 12,000 tons of refrigeration is included as follows:

	Electric Current	Operating Engr. Labor	Supplies & Materials	Total
Yearly unit cost per gross sq. ft. of build. area	$00.0258	$00.0172	$0.035	$00.0465
Yearly cost per rated ton	14.50	9.65	1.95	26.00
Yearly cost per occupant	8.50	5.65	1.20	15.35

The system, for which the operating costs are given, serves several buildings of the National Park Service and Department of Interior, in Washington, D. C. This air conditioning system was considered to be the world's largest in 1939. (38, p.p.375-78)

A study was conducted by the Corps of Engineers, North Pacific Division, Hydroelectric Design Branch, regarding the relative merits of several types of heating systems. It was necessary to compare the cost of each scheme for an operational period of 50 years in order to determine the most economical installation.

Electric resistance type heating is in operation at the Bonneville powerhouse which permitted inspection. The first cost of this installation was nominal, but a large block of firm power is required to operate the system each year.

The cost estimates which were made for the McNary powerhouse are included in this paper. The three methods which were considered seriously include resistance heating, heating directly with warm air from the generators, and two heat pump schemes.

A fourth method of heating, which proposed to use supplementary coils in the generators, is described for information only since the difficulties were too great to consider this system for the McNary powerhouse. With this heating arrangement, additional coils are placed ahead of the regular coolers which require extra space within the generator casing. A larger overall diameter of the generator housing is required. The increased width of the generator room would add to the cost of the powerhouse.

With the supplementary cooler arrangement, the generator cooling water would need to be modulated at full load to raise the air temperature of the air re-entering the generator windings to 40°C which is the maximum permissible.

It should be noted that the estimates include firm power at $14.50 per kilowatt-year, but this rate has been raised to $17.50. In analyzing the cost estimates, the new price of firm power should be considered since it would increase the cost of the resistance type of heating by $10,425 annually. The same increased cost of electricity would add only $2,370 to the annual cost of heat pump Scheme 1, and $2,250 to the heat pump Scheme 2.

The system of heating in which hot air is discharged directly from the generators involves a difficult filter problem. Even with an electrostatic precipitator for each generator, which is not more than 90 per cent efficient, a small amount of dust would collect in the machine which would need to be removed and would require a shutdown for maintenance.

Since the systems which use either hot air directly from the generators or employ supplementary coolers to supply hot water were thought to be undesirable from the standpoint of the generator construction and maintenance, it was decided to choose between electric resistance heating and the heat pump.

From the estimated cost analyses, the heat pump was the best choice. In addition to the reduced cost of heating, a source of chilled water was made available for cooling the various areas of the powerhouse in which personnel are located as well as to provide cooling capacity for lowering the temperature of the generator room.

The cost estimate of the heat pumps for Scheme 2 proposed the use of four heat pump compressors; however, three centrifugal compressors were actually used. This substitution did not change the cost of the system enough to warrant a complete recalculation of the estimate.

The following cost estimates included in the design data which was computed for the McNary powerhouse are shown on pages 20, 21, 22 and 23.

COST ESTIMATE -- RESISTANCE HEATING

Electric blower type heaters		$31,200
S. S. transformer capacity @ 4.00/K.V.A.		13,900
4160 V. switchgear - 1 position		4,400
4160 V. wire & conduit		13,800
4160/480 V. transformers & control centers		46,500
480 V. wiring & conduit		10,000
Labor included in above items		
*Ventilation same for all systems		______
	Total	119,800
	Contingencies 15%	17,970
		137,770
	Engr. & Overhead 15%	20,665
	Total Cost	$158,435

Annual Cost

Estimated average life of equipment 25 yrs.

Replacement of plant	$\frac{158,435}{50}$	$ 3,169
Maintenance	158,435 x 0.02	3,169
Firm power requirements	3475 KW @ $14.50/yr.	50,400
Simple interest @ 3%	158,435 x 0.03	4,753
Amortize in 50 yrs.	158,435 x 0.0089	1,410
	Total Annual Cost	$62,901

*Cost of ventilating the powerhouse has not been added as it will be equal for all schemes.

COST ESTIMATE -- HOT AIR FROM GENERATORS

Cost of electrostatic precipitator, duct work, dampers & controls (14 units)	$111,965
Increased cost of generators	420,000
Labor included in above items	
Ventilation same for all systems	
Total	531,965
Contingencies 15%	79,794
	611,759
Engr. & Overhead 15%	91,764
Total Cost	$703,523

<u>Annual Cost</u>

Estimated average life of equipment 25 yrs.

Replacement of plant	111,965 x 1.3/50	$ 2,912
Maintenance	111,965 x 1.3 x 0.02	2,912
Simple interest @ 3%	703,523 x 0.03	21,105
Amortize in 50 yrs.	703,523 x 0.0089	6,260
Total Annual Cost		$33,189

COST EXTIMATE --- HEAT PUMPS SCHEME 1

Centrifugal compressors (8)		$110,500
Heat exchangers & controls		75,990
Ductwork & installation		6,890
Piping		24,389
Pumps		4,176
S. S. transformer capacity @ $4.00/K.V.A.		3,600
4160 V. switchgear - 1 position		4,400
4160 V. wire & conduit		2,400
4160/480 V. transformers & control centers		20,000
480 V. wire & conduit		2,100
Electrical labor (Mech. included above items)		17,500
Ventilation same for all systems		______
	Total	$271,945
	Contingencies 15%	40,791
		312,736
	Engr. & Overhead 15%	46,911
	Total Cost	$359,647

Annual Cost

Estimated average life of equipment 25 yrs.

Replacement of plant	$\frac{359,647}{50}$	$ 7,193
Maintenance	359,647 x 0.02	7,193
Firm power requirements	790 KW @ $14.50/yr.	11,450
Simple interest @ 3%	359,647 x 0.03	10,788
Amortize in 50 yrs.	359,647 x 0.0089	3,198
	Total Annual Cost	$39,822

COST ESTIMATE -- HEAT PUMPS SCHEME 2

Centrifugal compressors (4)	$116,000
Heat exchangers & controls	75,990
Ductwork & installation	6,890
Piping	34,784
Pumps	2,728
S. S. transformer capacity @ $4.00/K.V.A.	3,200
4160 V. switchgear - 1 position	4,400
4160 V. wire & conduit	2,400
4160/480 V. transformers & control centers	16,000
480 V. wire & conduit	4,000
Electrical labor (Mech. included above items)	22,000
Ventilation same for all systems	
Total	$288,392
Contingencies 15%	43,258
	331,650
Engr. & Overhead 15%	49,747
Total Cost	$381,397

Annual Cost

Estimated average life of equipment 25 yrs.

Replacement of plant	$\frac{381,397}{50}$	$7,628
Maintenance	381,397 x 0.02	7,628
Firm power requirements	750 KW @ $14.50/yr.	10,875
Simple interest @ 3%	381,397 x 0.03	11,442
Amortize in 50 yrs.	381,397 x 0.0089	3,395
Total Annual Cost		$40,968

A source of heat is one of the paramount problems in the design of a heat pump system. Air, ground coils, well water and waste water are used for this purpose.

To operate efficiently a heat pump should have a constant supply of heat at a temperature of $50^{\circ}F$ or higher. A heat pump will function with source heat at a lower temperature, but the coefficient of performance is reduced materially.

The generator cooling water provides an ideal supply of this heat.

It was necessary, however, to determine the maximum generator cooling water temperature which should be used since the life of an electric generator is directly affected by the temperatures at which the various parts are maintained. A study was made into the various aspects of the problem of generator cooling.

The study began with an investigation into the sources of heat generated by the electrical currents and friction. The maximum temperature at which the generator can operate was studied and a test was made at Bonneville Dam to determine the limited temperature of the air from the coolers which fixes the highest cooling water temperature which can be maintained without injuring the insulation of the generator.

Electrical losses in the laminations, losses caused by the load current, frictional losses in bearings, windage and fan losses produce heat within the generator. Hysteresis and eddy current losses within the laminations caused by exciting current only are considered approximately constant regardless of load. Losses caused by the load current in the windings (I^2R), eddy current losses in the copper and losses in the laminations and iron surrounding them, vary as the square of the load current. Frictional, windage and fan losses are independent of the load unless the air circulated is varied with the load. (25, p.282)

The heat generated within a machine must be removed continually. If an adequate means is not available to remove the heat even for a short time, the load on the generator must be reduced.

Air circulated from outside the generator room was the first method used for cooling generators. With this arrangement, dust collected in the machines which necessitated considerable maintenance.

In recent years water coils have been used to cool the air which is recirculated within the enclosed generator casings. Either fans or the whirling of blades on the periphery of the rotor cause the circulation of air.

Damage to the insulation as well as its electrical resistance value is affected by high temperatures. Changes in the temperature periodically affect the life of the insulation.

The following tabulation indicates the maximum temperatures for different classes of insulation: (25, p.288)

Class	Description	Max. Temp.
A	Cotton, silk, paper and similar materials not treated, impregnated or immersed in oil	95°C
	Similar materials as above but treated, impregnated or in oil	105°C
B	Mica, asbestos	125°C
C	Fireproof and refractory	Limits not determined

At different temperatures the insulating quality changes.

"The insulation resistance in megohms varies inversely with the temperature at a rate depending upon the type of insulation, the degree of moisture present, and the condition of the surface." (32, p.157)

It is reported that if a generator is held at a constant temperature, the life of insulation is increased. Mr. Edgar Knowlton, of the General Electric Company, reported that "... the life of insulation is considerably increased if the machine is not subject to extremes of temperature. Sudden changes of temperature are especially serious..." (26, p.541)

J. Elmer Housley, of the Aluminum Company of America, indicated that the temperature of the generator should be kept as constant as is possible. "Any important temperature difference between the shut-down period and the operating period should be avoided because the contraction and expansion of the metal in the coils puts a strain on the relatively brittle insulating material." "Repeated expansion and contraction over a long period of time will undoubtedly shorten the life of the insulation. It appears desirable, therefore, to limit the temperature differential." (20, p.1055)

It was necessary to determine the highest temperature of generator cooling water which would not be detrimental to the insulation of the windings in the generators.

During the design of the McNary powerhouse heat pump system, the relationship of the cooling water temperature to the temperature of the air leaving the coils was investigated. Heat balance tests were made at Bonneville Dam which is on the Columbia river, 40 miles east of Portland.

The information obtained from a test of Unit No. 6, in June, 1948, is as follows:

	Load Mva	Cooling Water Temp. °C In	Cooling Water Temp. °C Out	Temp. of Air Leaving Coolers °C
Actual	68.0	5.1	12.0	22.5
Calc.	68.0	0.0	6.9	17.4
Calc.	68.0	0.0	13.33 (56°F)	23.83
Calc.	68.0	0.0	23.33 (74°F)	33.83

The first entry of the tabulation was from the actual test. The second entry is based upon the fact that a reduction of 5.1°C in the temperature of the incoming water temperature would lower the outlet water temperature by the same temperature increment with the load and water flow constant. For the third and fourth entries, the outlet water temperatures were assumed to be 13.33°C (56°F) and 23.33°C (74°F) respectively, which would be obtained by modulating the water flow. The temperature difference between the water leaving the coils and the air off the coils was estimated to be 10.5°C since the cooling load was assumed to be similar to the test.

The coolers used in the generators in the Bonneville powerhouse are mounted radially; whereas the coolers to be used in the McNary powerhouse are mounted perpendicular to the radius of the generators. The hot air circulates along the casing of the generators at Bonneville before it enters as well as after it leaves the cooling coils. With this arrangement, warm and cool areas occur alternately along the periphery of the casing. The generators for the McNary powerhouse are designed for air flow through the coolers before striking the casing.

Despite the difference in the location of the coolers, the data derived from the test made at Bonneville was used to advantage in computing the values of the foregoing tabulation.

Other information was found concerning the relationship between the temperature of the cooling water and the temperature of the air leaving the coolers.

From the General Electric Review, October 1929, data on the temperatures of cooling water from generators indicated that for an inlet water temperature of 25°C, approximately 10°C may be allowed for the water temperature rise. For 30°C inlet water, the water temperature rise is limited to 5°C so that the water from the cooler will not exceed 35°C and thus hold the leaving air temperature below a maximum of 40°C. (26, p.540)

The design temperature of the cooling water, which was selected for the heat source to the evaporators of the heat pumps to be used at McNary, was determined from the maximum allowable summer operating temperature and the relative value of the heat which could be reclaimed from the pre-heat coils and increased coefficient of performance of the heat pumps.

Data obtained from the tests made at Bonneville indicated that the water temperature rise through the generators should be approximately 6.9°C or 12.4°F. This water temperature rise through the generators added to an assumed maximum summer water temperature of 65°F would give a maximum cooling water temperature of 77.4°F. The value of 74°F was selected as the maximum cooling water temperature to use.

The selection of 74°F is substantiated by cost data which indicates that the higher the design temperature of generator cooling water the lower will be the initial installation cost as well as the annual cost. At lower temperatures of cooling water, increased heat pump capacity will be required to temper the incoming ventilating air, and the coefficient of performance of the heat pumps will be reduced which decreases their efficiency.

The temperature of 74°F is used only for design conditions, and it will be gradually lowered as the outside temperature rises.

The system which is used at McNary to "pump" the low quality energy to a higher useable level is basically similar to the "warming machine" which was described by

Lord Kelvin in 1852. His heat pump used fresh air as the refrigerant. The air was to be expanded and cooled in an air engine, reheated in a coil placed in outside air, and further heated and raised to atmospheric pressure by an air compressor. The heated air was then to be circulated within the building. (31, p.145)

Even though the heat pump was proposed almost 100 years ago, it had remained as working models in laboratories until comparatively recent times. The first residential installation was probably made in Scotland in 1927. (5, p.811)

Up to 1947 between 300 to 400 heat pumps were installed throughout the United States. The capacities ranged from 1/2 h.p. to 300 h.p. (42, p.161)

Recently it has been found practical to heat and cool hydroelectric powerhouses and administration buildings with heat pumps. (11, p.84)

"The reservoir water from Austin Dam in Texas, for example, is used in a heat pump application that furnishes year-around air conditioning to the Lower Colorado River Authority's three story office building at the dam site." (11, p.84)

In another area of the United States, at Pickstown, South Dakota, a heat pump is being used to heat and cool the areas of Fort Randall powerhouse generally occupied by personnel. (Appendix A, 1)

There is a definite lack of engineering data regarding the design of heat pumps. The 1952 edition of the "Heating Ventilating Air Conditioning Guide" does not contain heat pump design information; however, it is suggested that present installations should be studied to aid in the engineering of new heat pump installations. (5, p.812)

Heat pumps utilize equipment which is similar to that used for refrigeration systems, and since data is available regarding refrigeration it will not be included in this paper. It must be noted, however, that information is needed which applies the data used for refrigeration to the temperatures encountered in the design of heat pumps.

The capacity, water quantities, coefficient of performance and temperatures used in the design of the McNary heat pumps will be included in this description since these factors are a departure from the conventional refrigeration system.

The internal control of the heat pumps will be included in this section, but the control of the capacity of the entire system by the selection and operation of the three machines will be described in chapter five in the section on controls.

The high efficiency of the heat pump, as it is compared to direct resistance heating, is the factor on which the low operating cost of this type of system is based.

Coefficient of performance, cp, is a term which describes this efficiency is shown as follows: (42, p.3)

$$cp = \frac{\text{Heat delivered, } Q_h}{\text{Work } (Q_h - Q_c)} \qquad \text{(Heating cycle)}$$

Q_h = Heat rejected by the system

Q_c = Heat absorbed by the system (Refrigerating effect)

$Q_h - Q_c$ = Energy ideally added by compressor

Based upon the Carnot cycle, the theoretical coefficient of performance is designated as follows:

$$cp = \frac{T_h}{T_h - T_c} \qquad \text{(Heating cycle)}$$

T_h = Condensing temperature, absolute

T_c = Evaporation temperature, absolute

The theoretical coefficient of performance indicates the limits for a practical heat pump system. The overall efficiency of the compressor and the component parts of the heat pump system convert the theoretical value of cp into an actual coefficient of performance which may be expected in actual practice.

The required input horsepower is affected by two opposing factors in an actual installation. At higher evaporating temperature, the pressure and density of the refrigerant gas are increased which requires a greater horsepower to compress the gas. A compressor set to operate at a given speed will move a greater weight of refrigerant, and it will also "pump" more heat. (42, p.65)

At a higher evaporator temperature with a constant condenser temperature and a constant quantity of heat delivered, the coefficient of performance is increased which reduces the horsepower required per unit of heat delivered by the condenser. Also for a system with a lower condensing temperature and a constant evaporator temperature and a constant quantity of heat delivered, the coefficient of performance is raised.

If a machine is to be used for both heating and cooling, a balance must be found in the selection of chilled water temperature for cooling and a condenser temperature for heating which will permit the machine to operate at a constant r.p.m. without overloading the motor on heating.

Output of a heat pump on both the heating and cooling cycles is affected by factors which are independent of the evaporating and condensing temperatures.

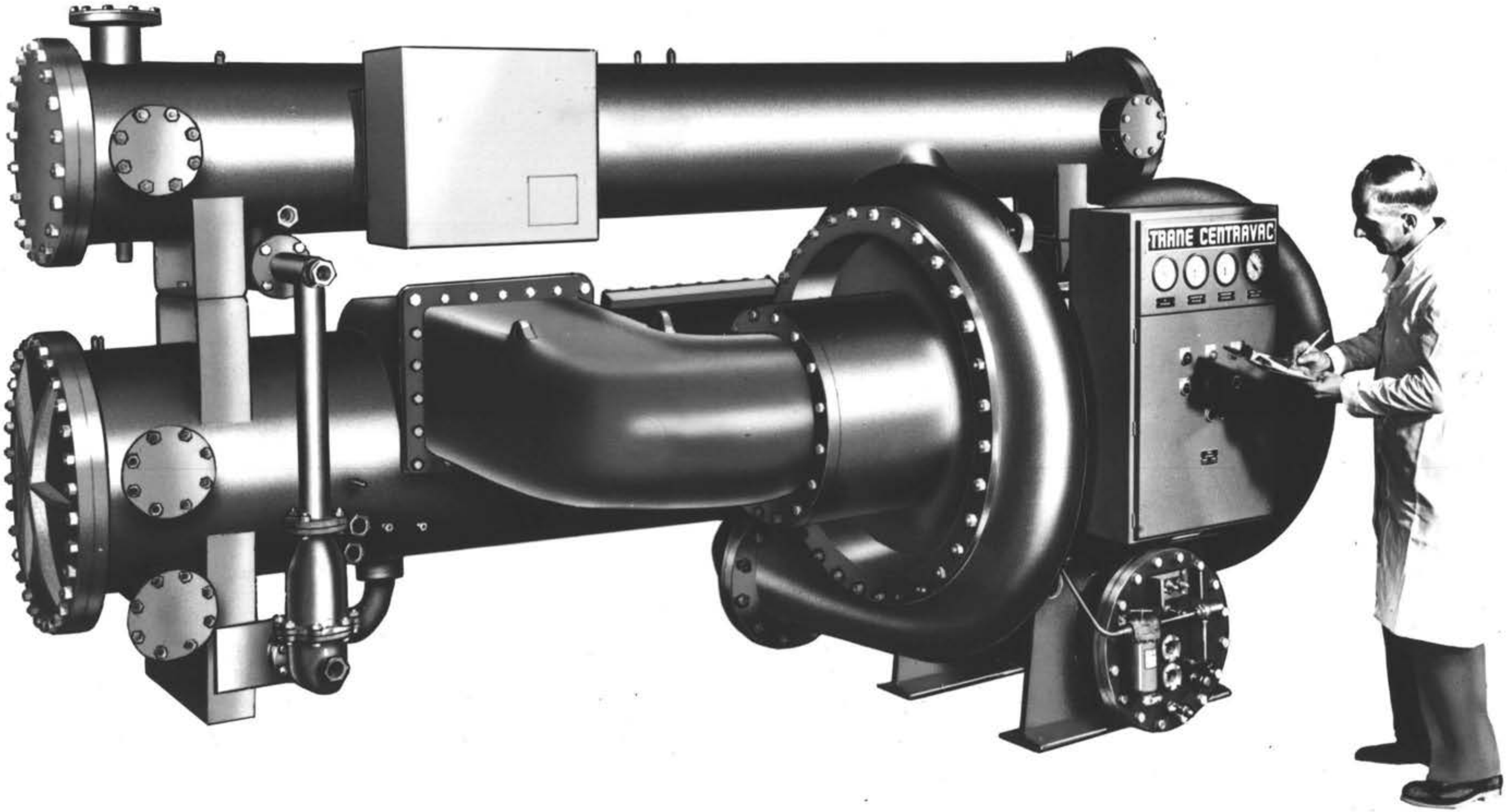
The use of stages from the lowest temperature at the heat source to the highest temperature of the condenser, is normally embodied in a centrifugal compressor system. The reduced compression ratio of each stage increases the overall efficiency of the equipment.

Since a centrifugal compressor operates in a negative pressure range on both the high side (condenser) and the low side (evaporator), air and other noncondensible gases tend to leak into the refrigeration system. A purge pump is needed to first exhaust the system when it is put into operation as well as to periodically remove these gases. This horsepower must be added to that of the compressor.

Impellers of a centrifugal compressor must revolve at a high rate of speed to provide the required difference in pressure between the evaporator and condenser. This speed is a serious mechanical design problem. These difficulties have been surmounted, however, in the design of centrifugal compressors which have been installed and which have proved very satisfactory by years of service.

The particular type of machines which are to be installed in the McNary powerhouse are hermetically sealed centrifugal compressors. A picture of one of these units is included to give the relative size and general arrangement of the equipment.

The heat pump consists of two impellers in stages which are mounted back to back on the shaft extensions of an electric motor which is hermetically sealed within the casing. No shaft seal is required to hold out atmospheric air pressure. (43, p.23)


TRANE CENTRAVAC

Vanes are located at the inlets of both the first stage and second stage impellers to modulate the refrigerant gas; this regulates the load carried by the machine. These vanes are connected, through a ring and gear mechanism, to an automatic air diaphragm operator which is controlled, in turn, by external air pressure. The variation of the air pressure which is caused by changing load, will be described in chapter five in the section on controls. (43, p.8)

Copper tubes, which are assembled in bundles, carry the water through the condenser and the evaporator. The tubes are removeable for cleaning without removing the connecting water piping which is connected to marine-type water boxes. (43, p.13)

A float valve regulates the flow of liquid refrigerant to the evaporator.

A purge compressor is employed to remove the noncondensible gases. Oil is separated from the refrigerant with heat from an electric resistance element. (43, p.p.18-19)

The two main bearings, which are the only moving parts in the main compressor, are lubricated with a force feed system which must be in operation before the compressor will start. (43, p.17)

The sequence of starting the machines begins with the chilled water and condenser water pumps. With these pumps in operation, a time delay relay is energized. After a brief interval of time, the lubrication oil pump is started automatically. With sufficient oil pressure available, an electric current is permitted to flow to the low temperature cutout. If the refrigerant temperature is above the minimum to prevent freeze-up of the tubes, the starting circuit is completed to the lockout relay which interrupts the current caused either by a rise in motor temperature or an excessive refrigerant pressure. It can be reset only with a manual push button which requires inspection of the equipment before a new start is attempted. (43, p.p. 36-37)

With all the foregoing safety controls operating in their normal running positions, the magnetic starter automatically energizes the compressor motor.

If an overload which is caused by rapidly rising evaporation temperature should be placed upon the motor, the load is automatically reduced. A control transformer, which measures the motor load, takes over the control of the inlet vanes to the compressor, and the load is limited even though the return water temperature may be "calling" for more capacity. (43, p.p. 36-37)

Three heat pumps are to be installed in the McNary powerhouse with a heating capacity of 2,362,500 B.T.U. per hour each. The rating is based upon heating 550 GPM of water in the condenser with an entering temperature of $97^{\circ}F$ and passing 650 GPM through the evaporator with an entering temperature of $74^{\circ}F$. Each unit has a capacity of 165 tons as a refrigerating machine which chills 550 GPM with a leaving temperature of $46^{\circ}F$ and with 626 GPM passing through the condenser. Twenty-four GPM of water is used to cool the auxiliaries.

The heat supplied by the condenser is computed upon the ASRE standard for rating and testing water-cooled refrigerant condensers with a fouling factor of 0.001 on the water side of the tubes. The fouling factor = $1/h_f$; h_f = fouling allowance on the water side in B.T.U. per hour per square foot of internal cooling surface of the tubes per degree Fahrenheit mean temperature difference. (6, p.432)

The coefficient of performance of the heat pumps to be installed at McNary is based upon the maximum heating capacity of 2,362,500 B.T.U. per hour with a compressor input of 150 BHP. In addition to the compressor motor, the horsepower of the auxiliaries and water pumps has been included to give the cp as follows:

$$\text{Input} = \left(\frac{150 + 20 + 15}{0.80^{*}} + \frac{0.33 + 0.25 + 0.33}{0.70^{*}}\right)2545$$

$$= 590{,}200 \text{ B.T.U./hr.}$$

$$\text{cp} = \frac{2{,}362{,}500}{\text{Input}} = \frac{2{,}362{,}500}{590{,}200} = \underline{4} \quad \text{(Heating)}$$

*Estimated motor efficiencies

With a cooling capacity of 165 tons, 145 BHP is required, and the cp is as follows:

$$\left(\frac{145 + 20 + 15}{0.80^{*}} + \frac{0.33 + 0.25 + 0.33}{0.70^{*}}\right) 2545$$

$$= 577{,}000 \text{ B.T.U./hr.}$$

$$\text{cp} = \frac{165 \times 12{,}000}{\text{Input}} = \frac{165 \times 12{,}000}{577{,}000}$$

$$= \underline{3.43} \qquad \text{(Cooling)}$$

As a comparison with the coefficient of performance ratings for other large heat pump installations, reference is made to "Table 9 Brown Boveri Heat-Pump Installations" listed on page 645, of the August, 1947 issue, volume 69, of Mechanical Engineering. The coefficient of performance values ranged from 2.62 to 4.4, and the heat output varied from 460,000 B.T.U. per hour to 6,000,000 B.T.U. per hour. (36, p.645)

The heat pumps to be used to heat the McNary powerhouse have a design coefficient of performance which compares favorably with other large installations that are already in operation.

*Estimated motor efficiencies

CHAPTER IV

VENTILATION

The ventilation of enclosed spaces has been a necessity ever since man moved from the outdoors.

The Romans endeavored to control the temperature of buildings by allowing air to escape through openings in the roofs. Sir Christopher Wren attempted to properly ventilate the House of Parliament in 1660. (13, p.62)

The real development of ventilation awaited the introduction and general use of electric motors which could turn various types of centrifugal and axial flow fans.

After years of "rule of thumb" methods of rating ventilating systems, an attempt was made to standardize fresh air requirements. A committee composed of James D. Hoffman, E. Vernon Hill, and Frank T. Chapman, reported in "The American Society of Heating and Ventilating Engineering Transactions", dated 1916, on the "Minimum Ventilation Requirements for Public and Semi-Public Buildings..." (19, p.47)

The minimum air quantity of fresh air per occupant of a factory was listed as 1500 CFH or 25 c.f.m. Public and semi-public locker rooms, coat rooms, dressing rooms and wash rooms were to have 6 air changes per hour except for rooms with windows opening directly outside with an area which equals or exceeds one eighth of the floor area. (19, p.49)

One air change per hour is the quantity of air in cubic feet equal to the cubical content of a room. Thus it is assumed that the entire air volume within a room would be changed in one hour.

Toilet room ventilation was indicated to require 35 c.f.m. for each water closet or urinal with not less than 6 air changes per hour. (19, p.49)

In 1919, another minimum ventilation standard was published in "Domestic Engineering." For windowless rooms with exposed walls above ground, 2100 CFH (35 c.f.m.) was required per person. For interior rooms, 5 air changes per hour plus 1200 CFH (20 c.f.m.) for each occupant was indicated to be the minimum ventilation requirement. Underground rooms required 6 air changes per hour plus 900 CFH (15 c.f.m.) for each occupant. Toilets required 10 air changes per hour plus 1500 CFH (25 c.f.m.) for each water closet and urinal. For locker rooms and coat rooms, 6 air changes per hour plus 600 CFH (10 c.f.m.) for each locker and each person's wearing apparel was indicated. (34, p.49)

Since the design of a ventilating system for a hydroelectric powerhouse requires special information, the Resident Engineers of the major plants in the United States and Canada were contacted for data concerning the actual air volumes used for ventilation.

Table 1 is the compiled data which was received in response to letters which were sent by the author.

The number of air changes used for various areas of the McNary powerhouse were also included in Table 1 so that a comparison could be made between the McNary system and several existing installations.

The following notations indicate the sources of information for Table 1, as well as special remarks concerning the entries in this table:

Source A, Table 1 (Appendix A, 2)

The number of air changes is based upon all outside air, recirculated air, or a mixture of each; however, all contaminated air is exhausted from the building.

Source B, Table 1 (Appendix A, 6)

The number of air changes per hour indicates the quantities of air distributed throughout the powerhouse and returned for recirculation during the winter and the quantities of air which will be exhausted from the generator room during the summer. Air which is contaminated, such as from the toilets and locker rooms, kitchen, battery room, dark room, oil treatment room, storage areas, and water treatment rooms will be exhausted at all times.

Source C, Table 1 (Appendix A, 11)

The number of air changes is based upon either all fresh air or reused air which passes through more than one room.

Source D, Table 1 (Appendix A, 8)

The number of air changes for the generator room is based upon a fifteen-foot-high zone for severe winter conditions and upon a complete volume of the generator room as long as the 256 kilowatts is sufficient to maintain a temperature of 65°F in the generator room.

Source E, Table 1 (Appendix A, 3)

Air for ventilation is reused and is exhausted outside.

Source F, Table 1 (Appendix A, 1)

The number of air changes is based upon the heat load and the outside temperature. The ventilating air is a mixture of fresh air and recirculated air.

Source G, Table 1 (Appendix A, 5)

The number of air changes is based upon all outside air for summer operation except for the control room and kitchen. A mixture of fresh air and recirculated air ventilate the control room, and reused air, which is drawn from another room, ventilates the kitchen. For winter operation, a mixture of outside and recirculated air ventilates all the rooms except the generator room and kitchen. In these areas, reused air is circulated. Also during the winter the number of air changes are varied as follows:

Generator room 15 changes per hour

Control room 8.6 changes per hour

Source H, Table 1 (Appendix A, 10)

The number of air changes is based upon varied conditions for summer and winter seasons. The ventilating air is a mixture of fresh air and recirculated air.

TABLE 1. NUMBER OF AIR CHANGES PER HOUR

Area Ventilated	Sources of Information							
	A	B	C	D	E	F	G	H
Air Compressor Rm.	6	-	1	-	-	4	10	6
Battery Room	6	10	2	-	6	10	14	5
Cable Spreading Rm.	6	4	-	-	-	-	2	6
Communications Rm.	6	-	2	-	-	3	1.3	2
Control Room	6	-	$2\frac{1}{2}$	-	6	10	17.3	6
Electric Shop	6	4	$4\frac{1}{2}$	-	-	6	5.3	6
First Aid Room	12	-	$11\frac{1}{2}$	-	-	7	7.2	6
Gate Repair Pit	-	-	15 to 20	-	-	-	-	2
Generator Room	2	3	$\frac{1}{2}$	3	1	2	3	$2\frac{2}{3}$
Kitchen	12	-	4	-	-	6	10	6
Locker Room	12	-	6	-	12	10	9.4	$8\frac{1}{2}$
Lower Galleries	12	-	$\frac{1}{4}$	1	-	3	1	$15\frac{1}{2}$
Machine Shop	6	6	$4\frac{1}{2}$	-	-	6	4	6
Motor-Gen. Room	6	-	-	-	6	4	5	$6\frac{2}{3}$
Offices	12	-	1	-	6	5	5	$3\frac{1}{3}$
Oil Purification Rm.	6	4	2	-	-	4	2 to 3	$4\frac{2}{3}$
Oil Storage Room	6	-	2	-	-	2	2	$2\frac{1}{2}$
Reception Room	12	-	7	-	-	6	5.2	6
Storage	2	2	$\frac{1}{2}$ to 2	-	-	2	-	2
Toilets	12	10	10 to 14	-	12	10	9	6 to 18
Turbine Room	12	3	$\frac{1}{2}$	-	1	2	2	2
Upper Galleries	6	2	1	1	-	-	1	$15\frac{1}{2}$

Condensation is a problem which must be considered in the design of a powerhouse. (39, p.130)

The reason why condensation should be avoided have already been discussed in chapter two; therefore, only the control of condensation will be considered in this section.

Condensation can be prevented by circulating ventilating air which may or may not be dehumidified or heated depending upon the amount of moisture to be removed and the condition of the air as it enters the galleries.

Information was received from the Office of the Bureau of Reclamation, Denver, Colorado, which included a tabulation of the wet and dry bulb temperatures of the various parts of the Left powerhouse of the Grand Coulee power plant. The unwatering gallery was selected as a typical example which is listed as follows: (Appendix A, 2)

August 1, 1946	Time	D.B.	W.B.	R.H.
Unwatering Gal. L-1	9:30 AM	70.0°F	68.0°F	90.0%
Fan Plenum	9:45 AM	70.0°F	59.5°F	54.0%
January 15, 1947				
Unwatering Gal. L-1	--	58.0°F	55.0°F	83.0%
Fan Inlet	--	67.0°F	50.0°F	27.0%

An analysis of the change in specific humidity of the inlet air to the condition of the air in the galleries indicates that considerable moisture is carried from the galleries by the exhaust air.

For the test made August 1, 1946, the humidity ratio (w = pounds of water vapor per pound of dry air) for the incoming air was found to be 0.0084. The humidity ratio, w, for the condition of the air in the unwatering gallery, L-1, was found to be 0.0143 on the A.S.H.V.E. Psychrometric Chart, 1951. The difference of the specific humidity ratios indicates that water vapor was removed by the ventilating air.

The test of the wet and dry bulb temperatures which was made on January 15, 1947, indicated a difference of 0.0047 in the humidity ratios between the incoming air and the condition of the air in the unwatering gallery, L-1; therefore, moisture was carried out of the gallery with the exhaust air. (Appendix A, 2)

To make a comparison with the design conditions in the vicinity of the Grand Coulee Power Plant, the wet and dry temperatures were selected from the "Heating Ventilating Air Conditioning Guide" for Spokane, Washington. With a summer design condition of 93°F dry bulb and 65°F wet bulb temperatures, the specific humidity ratio would be approximately 0.0068 which is lower than the humidity ratio for the outside air which was used in the test for August 1, 1946, which indicates that the data was taken at conditions which were more severe than usually exists in that area since the ventilating air at a higher humidity ratio can carry away less moisture.

The humidity ratio of the winter test was unseasonably high since both dry bulb and wet bulb temperatures were much above the average.

If it were found that condensation would take place, the incoming air could be dehumidified or heated to avoid reaching the dew point.

To supply the air needed to ventilate the galleries as well as the other areas of the McNary powerhouse, a fresh air conditioner is provided to clean, heat or cool 68,000 c.f.m. of air.

Since dust storms occur periodically in the vicinity of the McNary powerhouse, it is necessary to remove the dust, which is abrasive, from the air with the aid of an electrostatic precipitator.

This type of air filtering device is the result of years of engineering effort in the field of dust filtration.

In 1906, Dr. F. G. Cottrell was the first in this country to utilize the theories of electrostatic precipitation in the removal of gases and dusts commercially. (37, p.2)

Data concerning the electrostatic precipitation process may be found in the "Heating Ventilating Air Conditioning Guide" for 1952, on page 727; however, since this type of air-cleaning device is unique, a brief description is included.

The dust particles pass near a highly charged wire, and after passing through this ionized area between the wire and a grounded electrode, the positively charged dust particles drift away from the positively charged plates towards the negatively charged collector plates.

The efficiency of the unit "...depends upon the inertia of the dust particle, the electrical force tending to pull it toward the charged plates and the length of the plates." (41, p.3)

With a velocity which is too high, some of the particles will pass through without striking the plates, which reduces the efficiency.

"Air velocities of about 500 feet per minute give an efficiency of approximately 85% and 400 feet per minute gives approximately 90%." (41, p.3)

Air borne particles as fine as tobacco smoke are effectively removed.

The efficiencies previously stated as 85% and 90% are based upon the United States Bureau of Standards discoloration method of testing air filters.

Another standard of testing filters is the ASHVE method in which a known weight of very fine dust is added to the air stream; the original quantity of dust is compared with the weight of dust removed by the filter. Both of these standards are used by the air filter industry, but there is as much as 64 to 68 per cent difference in the efficiency test ratings of an identical filter as tested by the two methods. (41, p.4)

The electrostatic precipitator for the McNary powerhouse is rated at 90 per cent efficiency for 68,000 c.f.m. of air. This efficiency is based upon the Bureau of Standards discoloration method.

Manual washing of the plates of the precipitator was determined to be most advantageous since the installation is less expensive and a shut-down period of approximately four hours was not considered objectionable. The adhesive, which holds the precipitated dust to the collector plates after the charge is lost, is applied manually. It is estimated that one washing will be required each month.

A centrifugal pump with a rating of 8 GPM at 110 feet of head, circulates hot water under pressure to remove the dirt and adhesive from the plates of the precipitator.

The incoming fresh air is pre-heated to 63°F with generator cooling water which is supplied at 74°F. A two-row booster coil raises the air temperature to 80°F. For summer operation the booster coil is used to cool and dehumidify the incoming air; however, the pre-heat coils will not be used for cooling.

A sixty-inch, 68,000 c.f.m. fan, rated at 5/8 inches of water, discharges the conditioned air into the circuit breaker gallery at elevation 287.00. The air is distributed into the generator room through adjustable low wall registers. Gratings in the floor of the circuit breaker gallery supply air to the turbine room.

Ventilating air supplied to the generator room, which is in excess of the air quantities needed by adjacent areas, is permitted to escape through vents located in the roof of each bay of the powerhouse. The roof outlets are of non-aspirating type so that changes of wind velocity will not appreciably affect the air flow through the ventilators. Adjustable pressure release dampers, located at the inlet of the roof vents, keep the pressure in the building slightly above atmospheric. With a positive pressure within the powerhouse, the influx of dust will be reduced. (9, p.1696)

At the north end of the upstream wall, grilles allow air to flow from the generator room into the galleries, at elevations 320.00, 333.00 and 346.00. This air not only ventilates the galleries but is also forced into the intake gate repair pit at the south end of the powerhouse. The large quantity of air is supplied to the gate repair pit since paints and explosive mixtures of thinners are used in this area.

A central exhaust fan located on the top of the air shaft draws air from the sump pump room, El. 176.00; lower pipe tunnel, El. 221.00; station service bay, El. 254.00; battery room, El. 254.00; oil storage room, El. 246.50; air compressor room, locker room, toilet and shower, and rigging storage, El. 267.00; machine shop and electric shop, El. 287.00; and various storage rooms. The air is supplied from the generator room and turbine room.

Air also enters the air shaft from the gallery at El. 346.00 to ventilate the galleries whenever the intake gate repair pit fan is not drawing air from them.

There are several features of the ventilating system which will be described in more detail since they involved special problems.

The exhaust air from the locker room is taken through louvers in the lockers; therefore the moisture and odors are effectively removed from the clothing.

Ventilation of the battery room involves the prevention of explosions by the accumulation of hydrogen gas. "The volume of hydrogen gas given off for each fully charged cell is 0.01474 cu. ft. per ampere-hour of charge, irrespective of the rate of charging." An explosive mixture is formed by combining 8 parts of hydrogen with 92 parts of air. (48, p.724)

The McNary battery room contains 180 FME-17 Exide batteries which are rated 1.75 volt at 640 ampere-hours on the eight-hour rate. With a safety factor of two, the cubic feet of air per minute required to ventilate the room is as follows:

$$\text{c.f.m.} = \frac{2 \times 0.01474 \times 180 \times 640 \times 92}{60 \times 8}$$

$$= 650$$

The actual quantity of air exhausted from the battery room at McNary is 667 c.f.m. This volume of air was calculated on the basis of two air changes per hour.

Air from the toilets located in the generator room is exhausted through embedded pipes which extend above the roof.

A separate fan in the special unit, C-14, located in the generator room, supplies ventilating air to the stop log gate repair pit. Air is forced into the room near the floor level, and it is allowed to escape through vents. Twenty changes of air per hour purge the paint and volatile thinner vapors.

Air enters the south end of the fishway control gallery and is exhausted by a fan located at the north end of the generator room.

The air which ventilates the reception room is drawn through the louvered doors in the public toilets. Exhaust air from the control room is drawn through louvers in the doors leading to the toilet, kitchen, and record storage rooms. A common exhaust fan connects through duct work to these two systems.

Heat from the lights above the plastic ceiling in the control room is removed with air which is drawn from the generator room and exhausted.

Air from the outside and also from the generator room is mixed with recirculated air and is filtered, heated, and distributed to the reception room through ceiling outlets. All outside air is mixed with recirculated air during the summer months.

Outside air, mixed with recirculated air, is filtered, heated or cooled, carried through duct work, and diffused through ceiling outlets in the control room, lunch room, and the first aid room.

The offices are served by a unit conditioner which filters, heats, or cools a mixture of outside air and recirculated air. The conditioned air is distributed through ceiling diffusers.

Air is exhausted through vents in the roof of the toilets in the vicinity of the offices. A pressure release dampet in the janitor's closet holds a slight positive pressure in the office building.

Ventilation of the fishway pumphouse is provided by three roof exhaust fans. Air is drawn through three large louvers to cool the fishway pump motors. A maximum of two motors will be in operation at one time. The motors are of the open type which depend upon the air circulation for proper cooling.

The ventilation system is provided with heating and cooling coils and accompanying controls which will be described in chapter five.

CHAPTER V

HEATING COOLING & CONTROLS

Heating and cooling a powerhouse requires properly selected temperatures. Table 2 indicates the design conditions which have been used for several powerhouses located in the United States and Canada. The temperatures of only the generator room and control room were used in this table since these areas are representative of a powerhouse.

The temperature of 60°F, selected for the McNary generator room, may seem to be too low for the area occupied by the operators which will be stationed in this room. It is reasonable, however, since the heating equipment is designed upon an outside temperature of 0°F which will prevail for relatively short periods; therefore the temperature can be maintained at 70°F most of the winter season. Also there is heat dissipated within the building from the electrical equipment which will augment that supplied by the heat pumps.

There is considerable variation in the temperatures selected for the generators rooms as shown in Table 2.

The following notations indicate the sources of information for Table 2:

Source I, Table 2 (Appendix A, 11)

Source II, Table 2 (Appendix A, 6)

Source III, Table 2 (Appendix A, 12)

Source IV, Table 2 (Appendix A, 1)

Source V, Table 2 (Appendix A, 7)

Source VI, Table 2 (Appendix A, 4)

Source VII, Table 2 (Appendix A, 8)

TABLE 2. DESIGN CONDITIONS

Area Conditioned	Sources of Information			
	I	II	III	IV
Generator Room				
Source of Heat	Heat Pump	Elect. Res.	Gen. Water	Elect. Res.
Source of Cooling	Heat Pump	Fresh Air	Pool Water	**
Winter D.B. Temp.	60°F	55°F	45°F	55°F
Summer D.B. Temp.	85°F	---	***	85°F
Control Room				
Source of Heat	Heat Pump	Elect. Res.	Heat Pump	Heat Pump
Source of Cooling	Heat Pump	Refrig.	Heat Pump	**
Winter D.B. Temp.	70°F	70°F	72°F	70°F
Winter R.H.	50%	---	30%	45%
Summer D.B. Temp.	70°F	80°F	72°F*	80°F
Summer R.H.	50%	50%	50%	50%
Outside Conditions				
Winter D.B. Temp.	0°F	5°F	---	-25°F
Summer D.B. Temp.	95°F	95°F	---	95°F
Summer W.B. Temp.	70°F	78°F	---	78°F

* Maximum temperature difference between inside and outside temperature is 20°F.

** Mixture of pool water and heat pump chilled water.

*** The summer D.B. temperature is limited by supply water temperature.

TABLE 2. DESIGN CONDITIONS (CONT.)

Area Conditioned	Sources of Information		
	V	VI	VII
Generator Room			
Source of Heat	Air From Gen.	Elect. Equip.	Elect. Res.
Source of Cooling	Outside Air	Pool Water	Outside Air
Winter D.B. Temp.	40°F	70°F	65°F
Summer D.B. Temp.	---	---	---
Control Room			
Source of Heat	Elect. Res.	Elect. Res.	Elect. Res.
Source of Cooling	Refrig.	Refrig.	Refrig.
Winter D.B. Temp.	72°F	70°F	70°F
Winter R.H.	30%	35% to 45%	---
Summer D.B. Temp.	78°F	80°F	80°F+
Summer R.H.	55%	45% to 55%	53%
Outside Conditions			
Winter D.B. Temp.	-30°F	20°F	10°F
Summer D.B. Temp.	95°F	95°F	95°F
Summer W.B. Temp.	76°F	78°F	78°F

+ Above 80°F, the inside temperature is maintained 15°F below the outside temperature.

It has been reported from another source that the temperature in working areas should be maintained between 60°F to 72°F depending upon the class of work performed. (19, p.58)

A third source of information indicates that 62°F should be the minimum temperature for factories. (34, p.49)

An analysis of this data would indicate that the temperature of 60°F selected for the McNary generator room is an average value.

One factor which should be considered in regard to the McNary heating plant is the fact that air at 95°F from the unit heaters will be used to heat the generator room. With this heated air, the temperature within the generator room can be raised to 70°F whenever the outside temperature is 10°F or above. This relatively high temperature air would not be available for a room which is warmed by lower temperature air which is heated only by water from the generators.

The winter temperature of 70°F and a relative humidity of 50% was used at McNary for the offices and control room. This condition of air lies within the winter zone as shown on the "A.S.H.V.E. Comfort Chart for Still Air" on page 126 of the Guide.

Winter design temperatures for areas of the McNary powerhouse that are not listed in Table 2 are as follows:

Turbine room, $60^\circ F$; reception room, $70^\circ F$; electric shop, $70^\circ F$; machine shop, $70^\circ F$; wash room and toilets, $75^\circ F$; and locker room, $75^\circ F$.

The heat load for the entire powerhouse totals 11,099,130 B.T.U. per hour. The pre-heat coils, which use water directly from the generators, supply 4,614,000 B.T.U. per hour, and the remainder of 6,485,130 B.T.U. per hour is provided by the heat pumps.

Heat losses and heat gains for the various parts of the powerhouse are based upon heat transmission coefficients of building materials found in the "Heating Ventilating Air Condition Guide." (5, p.p. 167-198)

Since the entire powerhouse is under a slight positive air pressure, losses for infiltration were not calculated.

The internal and the conduction heat gains for the turbine room and generator room were calculated. The excess sun load for the roof of the generator room was figured, but the excess sun load on the massive walls was not considered since they have such a great time lag.

The heat gains for the control room, reception room, and offices were calculated in order to determine the greatest load for sizing the coils either for heating or cooling.

The coils for the units which serve the offices and control room were designed upon the cooling requirements, but the coil for the reception room was figured for the heating load.

Since the office building is exposed on all four sides and also contains many windows, it has different heating and cooling requirements than the rest of the powerhouse; therefore this area is served by a separate unit and duct system.

The control room has a special problem. With limited outside wall area and large heat gains from electrical equipment and lights, this room requires cooling while the remainder of the powerhouse requires heating. To provide this "intermediate cooling", water from the intake of generator number one is supplied to the unit conditioner.

The reception room has a separate unit and duct system since the heating and cooling loads vary with the presence of larger or smaller numbers of visitors.

The locker and wash room and toilet room each have a floor type unit which draws air from the generator room.

Units for the machine shop and electric shop are recirculating type which are mounted near the ceiling.

The fishway pump house control room and toilet are each heated with electric resistance unit heaters. This method of heating was used since it was not considered economical to pump water from the powerhouse.

Eight recirculating type units either heat or cool the turbine room as required. Sixteen standard units and one special unit supply heated or cooled air to the generator room. The special unit heats or cools the air of the south end of the generator room as well as the stop log gate repair pit.

All the units within the powerhouse have drip pans and drains to carry away the condensate which forms on the coils during the cooling period.

Permanent, viscous, washable type filters are included in the units which serve the turbine room, generator room, office building, control room, and the reception room.

Supply and return headers, which are insulated, carry either heated or chilled water. This distribution water system is supplied with three centrifugal water pumps, P-41, P-42, and P-43, which circulate 550 GPM each, against a 65 foot head. The pumps are supplied with 15 h.p. motors. The temperature of the water as it leaves the heat pumps is approximately 105°F in the winter and 46°F in the summer.

It should be noted that the pump numbers as well as other numbers of equipment are used to designate the particular location and operation on two drawings which are included in this paper. The drawings Key Plans, MDP-2.2-3-7/13 and Pneumatic Control Diagram, MDP-2.2-3-7/29 are found in Appendix B.

Booster pump, P-44, aids the water flow to the units which are connected to the headers in the pipe shaft.

Water from the discharge of the generator cooling coils, at a maximum design temperature of 74°F, is circulated through the pre-heat coils by pump, P-54, which is rated at 925 GPM at a head of 70 feet and is supplied with a 30 h.p. motor.

Water from the generator coolers is circulated, through the evaporator for heating and through the condenser for cooling, by three pumps, P-51, P-52, and P-53, which are rated 650 GPM at a 70 foot head and are supplied with 20 h.p. motors.

The pneumatic control system is shown on drawing MDP-2.2-3-7/29; however, certain electrical controls, which do not appear on the pneumatic drawing, will also be included in this paper.

Air from the 125 p.s.i. station service supply is filtered and dried; the pressure is reduced to 19 p.s.i. and 15 p.s.i. in a reduction station. The three air pressures of 125 p.s.i., 19 p.s.i., and 15 p.s.i. are used to operate the control system.

The temperature of the generator cooling water is controlled by the butterfly modulating valves, V-5. This type of valve was selected since it has a very low friction loss characteristic which was necessary for the proper flow by gravity through the generator cooling water system. An air operator changes the position of the butterfly from a fixed minimum opening through a modulated range to full open. If the generator is not in operation, the valve, V-5, is closed tightly. This is accomplished by water pressure from the upstream side of the valve which forces a rubber ring onto the edge of the butterfly. Three-way solenoid valve, V-7, either permits the upstream pressure to act upon the rubber ring, or it allows the water pressure to recede to the downstream pressure.

To start a generator it is necessary to control solenoid valves, V-7, V-8, and V-9. These valves are in an electrical circuit which is controlled by a manual switch in the governor cabinet. If the gates are opened without operating this switch, the solenoid valves will be positioned by an auxiliary which is located on the gates.

Valve V-7 releases the water pressure on the rubber ring of valve V-5. Valve V-8 interrupts the 15 p.s.i. air supply to the operator of valve V-5, and switches the control of valve V-5 to the thermostat T-14. Valve V-9 purges the air line to valve V-6, which opens this valve and permits water to flow from the generator coolers to the generator cooling water supply header which serve the heat pumps and pre-heat coils.

To provide a flow of water through the generator as soon as it is put in operation, a minimum position of valve V-5 is maintained even though the water is not up to the operating temperature. Pressure-reducing valve, V-45, lowers the pressure to thermostat T-14; therefore, valve V-5 can not be completely closed.

The control-point temperature of thermostats T-14 is changed remotely by a change in air pressure to its resetting mechanism. Outside master thermostat, T-20, controls this air pressure in response to an increase or decrease of outside temperature.

The temperature of the generator cooling water is maintained at 74°F for design conditions only, and it is gradually reduced with rising outside temperature; therefore, the operating temperature of the generator is lowered.

Three-way manual air valve, V-43, is used to purge the air line to the operator of valve V-5, which permits it to be in the wide open position even though thermostat T-14 is calling for modulation. With this arrangement, any generator can be operated with maximum cooling water which reduces the internal temperature of the generator.

If a generator is operating but is not providing heated water for the heat pump system, valve V-6 is closed by changing the position of three-way manual air valve V-44 so that 15 p.s.i. air pressure fills the line to the diaphragm of valve V-6.

Thermostat T-19 switches the system from heating to cooling at a pre-determined outside temperature. A zero pressure is maintained in this line for winter operation and 15 p.s.i. pressure for summer operation. This switch-over line controls pressure switches and three-way diaphragm operated air valves.

Connected to this 0 to 15 p.s.i. air line, pressure switch PS-1 operates through a relay to bypass the differential pressure switch, PR-1. This permits the fresh air conditioner fan to run in the summer without pump P-54 in operation. Since differential pressure switch PR-1 prevents the fan from operating unless sufficient differential water pressure is provided to keep the water flowing through the pre-heat coils, this eliminates the possibility of a freeze-up of the coils. An electrical freeze-up thermostat is mounted on the coils, PC-32, which also stops the fan and closes the intake dampers if the air temperature approaches the freezing point.

Pressure switch PS-2 is attached to the summer-winter switchover air line. This double-throw single-pole switch energizes either of two parallel circuits of a relay which, in turn, energizes either of two parallel contacts within the electric room thermostats. One of the circuits is used for heating and one for cooling. These electric thermostats control the capacity of the units located in the generator room and turbine room by varying the length of time that the fans are in operation.

The electric thermostats which control the unit fans in the machine shop and electric shop are switched manually from heating to cooling.

The units which serve the offices, reception room, control room, wash room, locker room, and fresh-air-intake booster coil, each condition ventilating air which must flow at all times. The amount of water flowing through the coil is varied to meet the demand of the heating or cooling loads. A three-way air-controlled mixing valve proportions the water flow either through the coil or through the bypass.

Air-type summer-winter thermostats are supplied with 15 p.s.i. air pressure for winter operation and 19 p.s.i. air pressure for summer operation. This air pressure not only sets the thermostat either for heating or cooling but also provides the air for operation.

The 15 or 19 p.s.i. air line is supplied by a three-way diaphragm operated air valve, V-4, at the air reduction station. Either zero pressure or 15 p.s.i. within the summer-winter switchover line positions this valve.

The winter relative humidity of the offices, reception room, and control room is maintained by steam generated with 7-1/2 KW electric elements immersed in pans of water located in each of the units. Air-type humidistats, in these three areas, operate pressure switches that energize or de-energize the electric elements.

As previously stated in chapter four, a ventilating fan draws 2200 c.f.m. across the area above the control room ceiling from the generator room and discharges the heated air to the outside. The heat gain from the lights is used to offset the heat losses of the control room during the winter.

The control room attic-cooling fan is controlled by thermostat T-24, which is located above the control room ceiling. The setting of this thermostat will be determined in the field to prevent overheating of the plastic ceiling.

As long as the temperature within the control room remains below a pre-determined temperature of approximately 74°F, intermediate-cooling thermostat T-23 is inoperative. The solenoid valve, V-36, remains in the heating position. This valve permits 15 p.s.i. air from the 15 or 19 p.s.i. air line, to set thermostat T-31, for heating operation.

During the heating cycle, thermostat T-23 being inoperative, solenoid valve V-37 is positioned to allow 15 p.s.i. air from the 15 or 19 p.s.i. air line to flow to the diaphragm of the three-way water valves, V-35. Valves V-35 permit hot water to flow through the coil of unit C-31.

In moderately cool weather while the powerhouse requires heating, the internal heat gains from electrical equipment and lights will be great enough to overcome the heat losses from the control room; therefore, intermediate cooling is provided by circulating cool water through coil C-31 from the intake of generator number one.

During intermediate cooling, thermostat T-23 positions three-way valve V-36 to close the branch air line to the 15 or 19 p.s.i. air line and open the lead to the 19 p.s.i. air line. The 19 p.s.i. air pressure re-sets the air type thermostat, T-31, for cooling.

Also during the intermediate cooling cycle, thermostat T-23 positions three-way solenoid valve V-37 to switch the branch air line "A" which connects to the diaphragms of valves V-35 from the 15 or 19 p.s.i. air line to the 0 or 15 p.s.i. air line. With zero pressure on the diaphragms of valves V-35, the three-way water valves are in the proper position to allow cool water to be pumped by P-45 from the intake of generator number one, through coil C-31 to the tail race to waste.

The motor of pump P-45 is started by the closing of the contacts of pressure switch PS-3 with zero air pressure to this switch.

As the outside temperature rises high enough to require cooling for the entire powerhouse, the outside thermostat, T-19, changes the pressure in the summer-winter change-over air line from zero pressure to 15 p.s.i. The actual change-over temperature will need to be determined in the field, but it may be estimated to be 72°F.

For full cooling of the control room, chilled water is circulated through coil C-31.

Thermostat T-31 is already set for cooling with 19 p.s.i. air pressure which is supplied for intermediate cooling.

Also for full cooling, solenoid valve V-37 is in the same position as that for intermediate cooling. The air pressure in the line to the diaphragms is changed to 15 p.s.i. since this line is connected to the summer-winter change-over air line which has been changed to 15 p.s.i. pressure by thermostat T-19.

With 15 p.s.i. air pressure, valves V-35 switch flow positions, and chilled water circulates through coil C-31. Pump P-45 is stopped by pressure switch PS-3 with 15 p.s.i. air pressure.

Two electrical outside anticipating thermostats, T-28 and T-29, provide a "dead spot" for a range of outside temperature during which neither heating or cooling capacity is required. The heat pumps, water pumps, and the units in the generator room and turbine room are electrically disconnected for this range of temperature.

Short-cycling is thus prevented.

The actual settings of thermostats T-28 and T-29 will need to be determined in the field. The high temperature thermostat should be set at approximately the same temperature as the setting of the change-over thermostat, T-19, since intermediate cooling water pump, P-45, will be stopped with 15 p.s.i. air pressure in the summer-winter change-over line.

Overall heat pump capacity for either heating or cooling is controlled by selecting one, two or three machines as required. The return water temperature which indicates the capacity needed affects the bulbs of thermostats T-17 and T-18. One of these thermostats controls the heating and the other the cooling capacity. A selection for heating or cooling is accomplished by the positioning of three-way, air-controlled, air valve V-16.

Under a light heating load, for instance, the return water temperature will be between 103°F and 104°F. Thermostat T-17 modulates the air pressure to the step controller and the first step starts the sequence to put heat pump number two on the line since it is centrally located in the powerhouse.

With a further drop in return water temperature, the step controller is positioned by air pressure to select heat pumps number one and number three. Heat pump number two is cut off the line.

A still lower drop in return water temperature causes the step controller to bring heat pump number two back on the line.

A decrease in heat load demand, which is occasioned by a rise in return water temperature, positions the step controller to stop heat pump number two. With a greater rise in return water temperature, heat pumps number one and number three are removed from the line and heat pump number two takes over the load.

Heat pumps number one and number three are adjusted to modulate internally not less than approximately 45 per cent of their capacity. This is required to allow the return water to rise on receding loads so that heat pump number two can take over the load through the step controller.

The sequence for cooling capacity is similar to that for heating, but rising return water temperature indicates that greater cooling capacity is required. Thermostat T-18 controls the cooling capacity through the step controller.

Selection of exact return water temperatures will be required in the field since they determine the hours of operating time for the heat pumps. The selection of the minimum capacity for heat pumps number one and number three will also be necessary in the field since this factor affects the wear of the individual equipment which should be balanced as closely as possible.

The McNary air conditioning system has been presented with complete descriptions of only the more unique functions. The standard aspects have been left to reference of well-established publications.

CHAPTER VI
CONCLUSIONS

1. In reference to the information in chapter three regarding the advisability of designing a complete air conditioning system, the critical point is considered to be above H = 5, which is 5 B.T.U. per hour per square foot internal heat gain. Buildings with an internal heat gain greater than 5 B.T.U. per hour per square foot are cooled most economically with conditioned air from chilled water coils. The value of H for the McNary generator room is approximately 14 B.T.U./hr. This indicates that complete air conditioning and not merely ventilation was the proper choice.

2. The cost estimates for heating with heat pumps show a yearly saving of approximately $22,000 over an electric resistance type of installation; therefore, the heat pump system was the proper selection on the basis of heating only.

3. The generator cooling water design temperature of 74°F is less than the maximum allowable by the generator manufacturers. The difference of 10.5°C between the generator cooler discharge water temperature and the air temperature off the coils, which was determined during the tests made at Bonneville Dam, added to 23.33°C, equals 33.83°C. This is below 40°C which is the maximum air

temperature which can be recirculated to the hot parts of the generator.

4. The safety controls are adequate to protect the heat pumps against freeze-up, excessive pressure, oil lubrication failure, and excessive motor loads caused by rapidly rising evaporation temperature.

5. The coefficient of performance of 4, for the heating cycle, compares favorably with the Brown Boveri heat pump installations. This value of cp is slightly less than the estimated value of 4.63 which was included in the cost estimate for Scheme 2.

6. The 15 to 20 air changes per hour used for the gate repair pits at McNary is necessary to properly ventilate the explosive mixtures and toxic vapors of paints and thinners.

7. The value of two air changes for the battery room is adequate since it exceeded the quantity of air which was calculated upon the number and rating of the cells installed in the battery room.

8. The turbine room and generator room have been designed for 1/2 of an air change per hour which is much less than for several other sources listed in Table 1, but since these areas are cooled by refrigeration during the summer, the quantity of ventilating air is adequate.

9. Referring to the 1/4 air change per hour of the lower

galleries at the McNary powerhouse, it would seem that insufficient ventilation has been provided, but the fact that the fresh air will be dehumidified with chilled water before it enters the powerhouse, the ventilating air should be able to carry away the excess moisture from the galleries.

10. The electrostatic precipitator, which is 90% efficient based upon the Bureau of Standards methods of testing, provides clean air for the powerhouse. Maintenance of equipment and of the building will be less than for an installation which uses unfiltered air.

11. Positive air pressure is maintained in the powerhouse by the "non-aspirating" type of pressure-release roof vents. Infiltration is eliminated which prevents the entrance of dust into the powerhouse.

12. Water at $74^{o}F$ from the generator coolers is circulated through the pre-heat coils by a 30 h.p., 925 GPM water pump. A total of 4,614,000 B.T.U. per hour of heat is supplied to the heating system; therefore, approximately 45 times as much energy in the form of heat is provided than is used to pump the water.

13. The design temperature of $60^{o}F$ selected for the McNary generator room approximates the average value of $55.7^{o}F$ calculated from Table 2, and also it agrees with other data which was presented regarding the proper temperature

for working areas.

14. Automatic control of the temperature within the control room is maintained throughout the year. During periods that cooling is required, while the remainder of the powerhouse is still on the heating cycle, pool water is circulated through the coil which serves the control room. The changes from heating to intermediate cooling and also to full cooling is accomplished automatically.

15. The temperature of the generator cooling water is gradually reduced by an outside master thermostat as the atmospheric temperature rises; therefore, the generators can operate at a lower temperature which should increase their life. The modulating valve prevents sharp rise or fall of the generator temperature which should also increase the life of the insulation.

16. The air conditioning system is changed from heating to cooling automatically by an outside thermostat. The air pressure in this change-over line is zero p.s.i. for heating and 15 p.s.i. for cooling. The system would "fail safe" to heating if the air pressure should fail for any reason.

17. Short cycling of the equipment is prevented during very mild periods by an "electrical dead spot" which is caused by two electrical outside thermostats. One of these instruments cuts the electric circuit to the heat

pumps, water pumps, and unit fans. Another electrical thermostat closes the circuit at such a temperature that full cooling with chilled water is required for the powerhouse.

18. Wear of the three heat pumps is equalized by adjusting the total running time for each machine. The step-controller, return-water thermostats, and the minimum load setting for heat pumps No. 1 and No. 3, provide the means of distributing the wear of the machines.

19. The McNary air conditioning system will provide proper temperatures, humidity control in necessary areas, air motion and distribution, dust abatement, and removal of explosive and toxic gases where present. Improved operation of equipment and increased efficiency of the personnel who operate the plant is thus provided.

RECOMMENDATIONS

1. More information should be made available regarding the relationship between the generator cooling water temperature, either constant or varying, and its effect upon the life of the electrical insulation of the generator.
2. Data for higher suction temperatures, than now available for refrigeration installations, is needed to aid the design of heat pump systems.
3. A single standard should be available to test and rate all types of air filters.
4. An investigation should be made to standardize the actual quantities of ventilating air which are required for various types of rooms. Experiments should be conducted to determine how much moisture must be removed from areas that are below water levels.

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APPENDIX A

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APPENDIX B

DRAWINGS

Key Plans, MDP-2.2-3-7/13

Pneumatic Control Diagram, MDP-2.2-3-7/29