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THE DESIGN OF A DIFFUSER AND HEAT EXCHANGER FOR A FOUR MEGAWATT ELECTROGASDYNAMIC TEST FACILITY

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FOREWORD

This report was prepared by James Van Kuren of the Electrogasdynamics Test Branch, Flight Mechanics Division, Air Force Flight Dynamics Laboratory (FDL), Wright-Patterson Air Force Base, Ohio. The design of a diffuser and heat exchanger for the FDL Four Megawatt Electrogasdynamics Facility is described.

This report represents part of the "in-house" design effort under Project 1426, "Experimental Simulation of Flight Mechanics." The design and analysis was a group effort of the Mechanical Engineering Section. Aerodynamic requirements and special consultation were furnished by the Thermo-Kinetics Branch under the supervision of Mr. F. J. A. Huber.

ABSTRACT

The design of a diffuser and heat exchanger for a hypersonic aerodynamic test facility is presented in detail.

This diffuser and heat exchanger are an integral part of the Flight Dynamics Laboratory Four Megawatt Electrogasdynamic Test Facility. Convective heat flux from 9000°F air to the diffuser wall and heat exchanger tubes is analyzed. Thermal and mechanical stresses are predicted as part of a complete design. Auxiliary equipment required to give a complete cooling system is described.

This technical documentary report has been reviewed and is approved.

P. P. Antonatos

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SYMBOLS

A	Area, square feet
D	Diameter, feet
E	Modulus of elasticity, pounds per square inch
G	Mass rate of flow at the minimum cross section area, $Lb_m/FT^2 HR$
I	Cross section moment of inertia of tube, Ft^4 or In^4
L	Length of tube, feet
LMTD	Log mean temperature difference (Equation 12), ^oR
N	Number of rows of tubes
P	Pressure, pounds/square inch
Pr	Prandtl number
Re	Reynolds number
S	Stress, pounds/square inch
T	Temperature, ^{o}Rankine
U	Overall heat transfer coefficient for a wall, $BTU/Hr \cdot Ft^2 \cdot {}^oR$
V	Velocity, feet/hour
W	Force per unit tube length
f	Friction factor
g	Acceleration of gravity
h	Convective heat transfer coefficient, $BTU/Hr \cdot Ft^2 \cdot {}^oR$
k	Thermal conductivity, $BTU/Hr \cdot Ft \cdot {}^oR$
r	Radius, feet
t	Thickness, feet
x	Distance, feet
ΔP	Pressure loss through heat exchanger
ΔT	Temperature difference between bulk air and water
μ	Dynamic viscosity, $Lb_m/Ft\text{-}Hr$
ρ	Density, Lb_m/Ft^3
ω	Natural frequency of tube vibration, cycles/sec

SUBSCRIPTS

a	Water channel
air	Bulk air property
c	Cylinder
e	Elastic failure
eq	Equivalent
i	Inside
i_n	Inlet
M	Mean
n	Natural
o	Outside
out	Outlet
W	Wall
Water	Bulk water property
∞	Free stream

INTRODUCTION

The Research and Technology Division Electro-Gasdynamics Facility is an electric arc heated hypersonic aerothermodynamic test facility. During test operation, air at a pressure up to 1000 psia passes through a direct current high voltage arc at a rate of up to 1 pound per second. Currently 4 Megawatts of D.C. power are available, a maximum of 66 percent of which is transferred to the air flow. Future plans call for extension to 8 Megawatts, therefore, all wind tunnel components were designed for this power level. Gas conditions leaving the heater were assumed to be 9000°F total temperature at 1 pound per second and supersonic velocity. The air is expanded to hypersonic velocities into an open test section where models are tested under simulated stagnation point conditions. A low vacuum down to 20 μ Hg absolute, is required in the test section to establish this flow. A hypersonic to subsonic diffuser is placed in the system to recompress the air and reduce the vacuum volume pumping requirement. To protect the pumps, the air is cooled in an air to water heat exchanger.

The project reported herein consisted of the design and installation of the diffuser and heat exchanger system for this wind tunnel. This work was started in 1960 and completed in its entirety by the Facility Engineering Branch.

CRITERIA

Available Utilities

The diffuser and heat exchanger system ordered by the operating engineers, had certain specifications and had to be designed within certain constraints. Additional criteria were added by the Facilities Branch due to the location of the system in a building where some unusual utilities were available but floor space was limited.

The usual supplies of water and electricity were available. Compressed air at 100 psi pressure and 150,000 gallons of freon 12 stored at -40°F were existing in the building. A high pressure air system for 1 pound per second at 1200 psi, an electrical power supply for 4 Megawatts of D.C. power, and a vacuum system for 30,000 cubic feet per minute at 20 to 500 microns of mercury absolute pressure, were installed in the building for this wind tunnel under previous contract. The vacuum pumps are Roots-type, rotary, positive displacement pumps limited to 150°F inlet temperature.

Gas Properties

The arc air heater and nozzle system performance were predicted by other personnel of the Aerodynamic Division to produce 1 pound per second of air at 9000°F total bulk temperature and hypersonic velocities. Velocity profiles across the diffuser entrance would vary for different test article sizes and shapes.

General Requirements

To keep vacuum volume pumping requirements within existing limits, it was required that maximum pressure drop for the heat exchanger should not exceed a fraction of the pressure after the normal shock wave for the Mach number of the test conditions. No melting or structural instability was allowed and all joints and seals were to be designed for the vacuum condition.

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DIFFUSER DESIGN

General Description

A perfect diffuser causes an isentropic compression of the high velocity low pressure gas to high pressure and subsonic speeds. In an ideal, one dimensional, supersonic flow a decrease in cross sectional area in the flow direction causes a decrease in Mach number and velocity. The opposite is true for subsonic flow. Deceleration would take place if the wind tunnel duct area was decreased to that at which sonic velocity exists and then increased to further reduce the velocity. In actual application any attempt to compress the flow produces shock waves with an inherent loss of pressure. The best hypersonic wind tunnel diffusers produce about 1.25 times the normal shock pressure ratio for the entering flow Mach number.

The internal geometry of this diffuser was given by the aeronautical engineers. The purpose of this part of the project was to analyze the heat transfer to the inside surface and then to design an adequate structure to contain the flow.

Cooling Analysis

Because of the lack of information, it was assumed that the heat transfer at the inside surface was that of a fully developed turbulent pipe flow. Tests have shown that this assumption was a usefully accurate prediction. Actually, peak values of heat transfer could be higher locally. The empirical equation for the inside surface coefficient is given in Reference 20 as,

$$h = 0.023 \frac{k}{D} Re^{0.8} Pr^{0.4} \qquad \frac{BTU}{HR\text{-}FT^2\text{-}^\circ R} \tag{1}$$

where k is the thermal conductivity, D is the diameter, Re is the Reynolds number and Pr is the Prandtl number. All fluid properties are evaluated at the film temperature which is given as the average of the free stream and the wall temperatures.

Before the heat transfer analysis could be continued it was necessary to assume the wall thickness. Water cooling was selected as the simplest and most practical method. For this case the diffuser wall was subject to external pressure of 150 psi and the inside pressure was in the order of one psi. The length to diameter ratio of the straight section of the diffuser was four and the buckling pressure was found by the use of the equation below (Reference 23).

$$P_e = \frac{2.60\,E\,(t/D)^{5/2}}{L/D - 0.45\,(t/D)^{1/2}} \tag{2}$$

The material selected for the inside wall was U.S. Steel Corp. T-1 steel. This steel was suitable for this application because of its high strength at elevated temperature. It has better thermal conductivity than stainless steel, is 4 times as corrosion resistant as mild steel and has a lower coefficient of thermal expansion. The material sizes chosen were based on past experience and availability.

Once the wall material and thickness were established, surface temperatures and cooling requirements were computed by the methods given in most heat transfer texts. The overall heat transfer coefficient is,

$$UA_m = \frac{1}{\frac{1}{h_i A_i} + \frac{\ln r_i/r_o}{2\pi kL} + \frac{1}{h_o A_o}} \qquad (3)$$

and the inside wall temperature is,

$$T_i = T_{air} - (T_{air} - T_{water}) \frac{UA_M}{h_i A_i} \qquad (4)$$

To evaluate h_o it was necessary to define the equivalent diameter used in Equation 1. The cross section is an annulus, therefore,

$$D_{eq} = \frac{4\ \text{Area}}{\text{Perimeter}} = \frac{4\pi D t_a}{2\pi D} = 2 t_a \qquad (5)$$

A simplification of Equation 1 is given in Reference 26 as,

$$h_o = 0.00134 \frac{V^{0.8}}{D^{0.2}} (T_{water} + 100) \qquad (6)$$

where T is in degrees fahrenheit.

The water velocity required to keep the inside surface below melting temperature and the waterside surface below boiling was calculated.

Thermal Expansion

Longitudinal thermal stress of the diffuser was controlled in two ways. The center of the heat exchanger was chosen as the fixed point of the system. The upstream end of the diffuser was allowed to expand freely. Differential expansion of the inner shell of the diffuser with respect to the outer shell was allowed in two ways, by a bellows expansion joint in the outer shell in the supersonic section, and by the differential expansion of dissimilar metals in the subsonic section. The presence of a large quantity of water keeps the shells at nearly the same temperature. The outside shell is 304 stainless steel, a material with a larger coefficient of expansion than that of the inside shell which is T-1 steel. Resulting stresses are at an acceptable level of about 13,000 psi.

Flange Design

Future plans for the facility included different Mach number and Reynolds number flow conditions. For these future modifications it would be necessary to have a different geometry of the diffuser. Therefore, the diffuser was fabricated in two parts. The downstream part consisting of the divergent cone section, can be retained for the modified diffuser.

The flange joints of the diffuser presented special cooling and sealing problems. Internal pressures of the order of 20μ Hg absolute and a marginal vacuum pumping capacity necessitate positive seals. High heating rates over the inside surface made it necessary to cool all parts in contact with the flow. Water was piped into a stilling chamber on the upstream flange. The channel was designed so that the water accelerated past the flange which had no internal corners to cause stagnation points that could be potential hot spots. (See Detail B, Figure 2). At the parting flanges a double seal was designed to seal water from the room and from the hot gas. The flanges were slotted to allow the water to pass directly through, thereby producing the required cooling effect (View D-D, Figure 2). Water flow was parallel with the gas flow so that the coldest water could extract heat from the hottest gas. By this method, wall temperatures were kept to a minimum.

Stress Analysis

Stresses due to pressure loads were calculated in the various parts. Hoop stress in the outer shell is given by:

$$S = \frac{Pr}{t} \tag{7}$$

The elastic stability of the divergent section under external water pressure of 150 psig was also checked. This section expands to 4 feet internal diameter at the downstream end of the diffuser. Charts for computing the critical pressure of truncated conical shells are found in Reference 7. In calculations for simply supported ends, the diffuser had a marginal safety factor for the originally chosen thickness. Shell thickness was increased from $\frac{1}{4}$ to $\frac{3}{8}$ inch to increase the safety factor. Welded flanges provided some additional stiffness. Circumferential stiffeners were impractical because of the coolant flow. Longitudinal stiffeners added considerable strength because of the double wall construction, and have an additional advantage of providing flow guides to keep the coolant evenly distributed around the shell.

HEAT EXCHANGER DESIGN

General Requirements

As previously stated, the air flow was 1 pound per second at an assumed 9000°F. This extreme temperature caused primary concern for the prevention of melting or burn-out of any parts in contact with the flow. A secondary criterion was the condition of gas leaving the cooler and entering the vacuum pumps; the pumps were limited to 150° inlet temperature. The air pressure drop caused by the cooler, later proved to be an important factor, because it was determined that the diffuser pressure recovery was lower than the predictions.

Other than these special requirements, sound engineering practice could be applied to make a reasonably reliable system. Water and air systems had to be positively sealed because of the required vacuum condition, water velocities had to be sufficiently high to provide adequate cooling but low enough to preclude erosion, air pockets were to be prevented in the water system, and thermal expansion allowance was to be provided. Inspection and maintenance procedures were considered in the design, and the safety of the operating personnel is a continuing consideration.

Gas Heat Transfer

This particular topic is the most unique part of this project. The gas conditions of temperature and weight flow presented by this facility had never occurred. No data was available on heat transfer and pressure drop at these conditions and very little information was available on air properties. For these reasons, every attempt was made to be conservative in making these assumptions. In general, two problems were considered, first a provision had to be made for adequate cooling of peak local heating and second the entire heat load had to be carried away. The difficulty in the first problem was the prediction of stagnation line heat transfer on the first row of tubes. Four different methods were compared. None of the methods were directly applicable: two methods were for stagnation point heating of hypersonic re-entry bodies, one was for subsonic incompressible flow at relatively higher Reynolds numbers and one was for low supersonic flow at much lower temperatures. The methods compared within 20 percent. The method giving the highest value was used for wall temperature calculations.

The formula developed in Reference 13 is

$$h_c = 0.763\, k\, Pr^{0.4} (dV/dx \,.\, \rho/\mu)^{1/2} \tag{8}$$

and for a circular cylinder

$$dV/dX = 4 V_{\infty} / D_c \tag{9}$$

In the stagnation region the tube temperature can be calculated approximately by writing a heat balance for transfer through a cylindrical wall. The overall heat transfer coefficient is given by Equation 3 where h_o is given by Equation 8 and h_i is given by Equation 1. The wall temperature outside is

$$T_w = -(T_{air} - T_{water}) \frac{UA_M}{h_c A_c} + T_{air} \tag{10}$$

The tube wall temperature on the air side was calculated to be approximately 500°F. This calculation did not account for the three dimensional relief due to lower heating at points away from the stagnation line, therefore, actual operating temperatures would be lower.

High Temperature Coils

A staggered tube arrangement was chosen because it gives the most effective heat transfer. With this configuration the second row of tubes would be subject to the same stagnation heating as the first row. A compromise was made between high water velocities in the tubes and high surface temperatures. The constraint in this case was the tube temperature calculated from steady state heat balance considerations. It was desirable to cool the air in a minimum length to reduce the overall pressure drop. To accomplish this the use of extended surface (fins) was investigated using the method in Reference 15. It was determined that the extended surface would reach melting temperature if placed in the first 12 rows of tubes. For this reason the first twelve rows had to be void of uncooled flanges, brackets or supports. A vertical tube arrangement was chosen for the first rows so that the tubes would be self-supporting. (See Figure 5). Four 4-inch pipes acted as manifolds and as feet for the plain coils. Water flows into the first manifold at the hot end of the heat exchanger and is supplied to 20 vertical 1-inch copper tubes in parallel. Six tubes are in series and the water again enters a header. A second duplicate coil is located in series for a total of 12 rows of tubes with 20 tubes per row. When the flow is in parallel channels the normal practice is to place orifices at the inlet of each tube to balance the flow in the circuits. However, in the present application, the flow resistance in the tubes is high enough to substitute for orifice effects.

With the vertical tube arrangement and tube bends in the high points of the system there exists a possibility for air pockets to form. A calculation of the forces on an air bubble shows that it would be quickly swept downstream at the 20 ft/sec water velocities.

Since the tubes are not restrained at the top there is an increased bending moment induced by the aerodynamic load. There was some concern over the possible vibration of the tubes excited by fluctuations in the flow or by aeolian vibrations on the tubes. The aerodynamic load was calculated by the simple formula for drag on a cylinder in cross flow,

$$F_D = C_D \tfrac{1}{2} \rho V^2 A$$

where C_D is the drag coefficient ρ is the air density. V the velocity and A the frontal area of the tube. The load was of the order of 0.10 pound, which was negligible for the 1-inch tubes. The natural frequency of the tubes was given by Reference 29 as,

$$\omega = 3.55 \sqrt{\frac{g E I}{W l^4}} \tag{11}$$

where E is the modulus of elasticity, I the moment of inertia, W the weight per foot of tube and l the length of tube. The natural frequency is about 13 cycles per second compared to 1000 to 2000 quoted in Reference 27 for the aeolian vibrations.

Finned Coils

Analysis of the fin temperature mentioned previously gave temperatures of 550 to 590°F if copper fins which are an integral part of the tube are placed after 12 rows of plain coils. At the low density conditions predicted for this heat exchanger, boundary layer growth is very rapid, approaching $\frac{1}{4}$ inch in the length of one fin diameter. For this reason the largest fin spacing commercially available, five fins per inch, was chosen, on the basis of log mean temperature difference. (Which applies only to cases of constant U but yields reasonable results for this case)

$$LMTD = \frac{\Delta T\ in - \Delta T\ out}{\log_e \Delta t\ in / \Delta t\ out} \tag{12}$$

where Δt is the temperature difference between the coolant and the gas. It was determined that for a commercially available coil with 36 tubes across the face, 20 rows of coils were required to cool the air temperature down to 150°F. Standard fin coils are usually manifolded on the side and the tubes are orientated horizontally. This appeared to meet the requirements of this project because there would be no support problems and maintenance would be relatively easy. Three coils were chosen, two with 6 rows of tubes and one with 8 rows. Feeder lines for the water were designed so that the water can enter the bottom of the headers and be discharged from the top. Air which goes to the high point of the system is then forced out. Each header was furnished with a bleed valve to facilitate removal of the air during initial operation. Pressure drop of the air passing through the heat exchanger was estimated by the method given in References 17 and 18 for low Reynolds numbers. Data in these reports was presented in graphs of $f = \Delta P \rho\, g/2G^2 N$ versus Re for a range of tube spacings. Pressure drop was 0.1 psi for the design point. For conditions of lower Reynolds number, one or two of the finned coils could be removed and the opening in the box could be blanked off. Thus a variety of operating conditions could be accommodated at minimum pressure losses. This is an extremely important factor for blow down wind tunnels because it directly affects the time available for testing. The finned coil installation is shown in Figure 7.

HEAT EXCHANGER SHELL DESIGN

The shell required for the gas-to-water heat exchanger was similar in size to the structure of an existing supersonic wind tunnel. The design could be directly duplicated if the wall temperatures were kept below the design temperature of that structure. A quick calculation of the flat plate heat transfer showed that an uncooled wall would melt within a few seconds. The internal vacuum made it necessary for any integral cooling passages to be very elaborate. A relatively simple solution was applied however by using plate coil, which was laid against the inside walls in the high temperature section of the box.

A manhole was designed into the shell in the section between the plain coils and the finned coils. This would facilitate inspection and maintenance of the coils. The 2-foot-square hatch was calculated from a method given in Reference 21 for a uniformly loaded plate supported at the edges. Cross stiffeners were required to keep deflections low enough to maintain the vacuum seal. A flat rubber gasket was used for sealing. Tapped holes eliminated the problems of bolt seals. The inlet end of the box was provided with a large square flange for connection to the diffuser. The outlet end had a short section of 24-inch pipe with a bellows expansion joint for connection to the vacuum system. The shell structure was made longer than necessary to facilitate future addition of coils for a planned power increase to 8 megawatts.

EXPANSION JOINT

The layout of the system was such that the deflection of the vacuum pipe was perpendicular to the axis of the expansion joint. Manufacturer's data claimed that lateral deflection was determined by angular displacement of the corrugations. Space was limited so that a maximum of 3 corrugations could be used. A simple geometric calculation showed that lateral displacement could be $\frac{3}{8}$ inch. Measured deflection of the vacuum pipe was $\frac{3}{4}$ inch, therefore an anchor brace was designed to restrain the vacuum pipe at the heat exchanger connection.

WATER SYSTEM

The water system to be used for the heat exchanger presented some problems. Wright Field water with a high mineral content was found to be very unsatisfactory for cooling at high heat transfer rates. Cooling tower water was available in the building but it was also impure. Therefore, a separate demineralized water system that was cooled by tower water in a tube-in-shell heat exchanger was used. Later it was learned that distilled water was available on the base and could be supplied by tank truck. However, this water was slightly acidic, and would absorb oxygen and corrode the system. A spray tank from a de-aerator was obtained from the Air Force surplus and converted to a vacuum type de-aerator. The tank was placed high in the building (above 34 feet) and a vacuum source was attached to the tank. A leveling device, and a circulating pump were used to maintain water in the tank.

The water-to-water heat exchanger was designed for 1200 GPM of water at 125 psi to be cooled from 160° to 110°F on the shell side. On the tube side, 1000 GPM of 15 psig tower water would rise from 90° to 150°F to create the worst condition of maximum power input from the tunnel on a hot summer day. A favorable factor was the heat capacity of the system which represents 20 percent of the heat input for the maximum run time of 30 minutes. The exchanger was designed with a removable head on the tube side to allow tube cleaning.

Because of the lack of floor space in the building, the water-to-water heat exchanger was mounted on top of the 6-foot diameter vacuum pipe. This required a special saddle design and an analysis of the column supports of the vacuum pipe. As previously mentioned the vacuum pipe moved ⅜ inch, therefore the heat exchanger feet were furnished with elongated bolt holes and coated with graphite to allow the vacuum pipe to slide back and forth. This technique was successful on the vacuum pipe supports.

Pump specification was a routine problem. The water pressure losses of the circuit were calculated for the required flow and then used to determine pump horsepower. Another pump was furnished to supply 500 psi water at 300 GPM to the arc heater.

ELECTRICAL REQUIREMENTS

Three phase 440 volt power for the pumps was available in the building. Each pump required a 150 HP motor, which was furnished with combination starter-circuit breakers with lock-out push buttons located at the pumps.

INSTRUMENTATION

Instrumentation was made as simple as possible principally for budgetary reasons. Three flow meters were provided, one for tower water and one each for the flows to the pumps. These were orifice meters with local indicators and pneumatic transmitters for future remote indication. Bourdon pressure gages were used to give pump pressures and head losses for each separate cooling loop. Dial type thermometers measured temperature rises for each separate cooling loop. Visual observation during the initial runs permitted correction of any anomolies in the system. Sufficient valving of the separate loops facilitated the corrections.

TEST RESULTS

The FDL Four Megawatt Electrogasdynamic Facility was first operated on 9 December 1963 at a power level of 1.78 megawatts to the heater and 0.37 pounds per second mass flow. Because of the lower power and mass flow rate, the heat exchanger was operated with two banks of fin coils removed. The remaining bank contained 6 rows of tubes. Outlet air temperatures were 62°F, which was well below the required 150°F. Water temperature rise was 7°F for 2 MW compared to design temperature rise of 50°F for 8 MW input. The only problem that occurred was the faulty operation of some of the thermometers. Subsequent operation at up to 4.7 megawatts and 0.49 pounds per second proved the design was more than adequate.

REFERENCES

REPORTS

1. R. E. Thomas and J. D. Lee, "The Ohio State University 12-Inch Hypersonic Wind Tunnel System," The Ohio State University Research Foundation Report Nr 559-2, August 1959.

2. Ames Research Staff, "Equations, Tables, and Charts for Compressible Flow," NACA Report Nr 1135, 1953.

3. R. F. Thomas, "Some Real Gas Effects in the Hypersonic Flow of Air," M. S. Thesis, Department of Aeronautical Engineering, The Ohio State University, 1956.

4. F. C. Hansen, "Approximations for the Thermodynamic and Transport Properties of High Temperature Air," NASA, TN 4150, March 1958.

5. S. Feldman, "Hypersonic Gas Dynamic Charts for Equilibrium Air," AVCO RAD, January 1957.

6. K. K. Yoshikawa and E. D. Katzen, "Charts for Air-Flow Properties in Equilibrium and Frozen Flows in Hypervelocity Nozzles," NASA, TN-D-693, April 1961.

7. P. P. Radkowski, "Buckling of Thin Truncated Conical Shells Subjected to Uniform External Pressure," WAL TR 893.3/1, May 1958.

8. D. O. Horning, "Vacuum Vessel Construction Techniques and Sealing Methods," University of California, Institute of Engineering Research Report Nr HE-150-84.

9. Ying-Nien Yu, "A Summary of Design Techniques for Axisymmetric Hypersonic Wind Tunnels," NATO AGARDograph 35, November 1958.

CATALOGS

10. Fabrication of U.S.S. Stainless Steels, ADUCO-15595 [illegible] Second Edition, 1956.

11. Ladish General Catalog Nr 55, Ladish Company, 1954.

12. Anaconda Copper and Copper Alloys, Anaconda Publication, B-32, 1962.

ARTICLES

13. M. Sibulkin, "Heat Transfer to an Incompressible Turbulent Boundary Layer and Estimation of Heat Transfer Coefficients at Supersonic Nozzle Throats," Journal of the Aeronautical Sciences, February 1956.

14. D. R. Bartz, "A Simple Equation for Rapid Estimation of Rocket Nozzle Convective Heat Transfer Coefficients," Jet Propulsion Magazine, January 1957, p49.

15. J. G. Bartas, "Gas-Side Wall Temperatures in Rib-Backed Liquid-Cooled Combustion Chambers," Jet Propulsion Magazine, July 1957.

REFERENCES, cont'd

16. W. J. Graff, "Thermal Conductance of Metal Joints," Machine Design, September 15, 1960.

17. S. L. Jameson, "Tube Spacing in Finned-Tube Banks," Transactions of the ASME, November 1945.

18. A. Y. Gunter and W. A. Shaw, "A General Correlation of Friction Factors for Various Types of Surfaces in Crossflow," Transactions of the ASME, November 1945.

19. K. A. Gardner, "Efficiency of Extended Surface," Transactions of the ASME, November 1945.

BOOKS

20. W. H. McAdams, Heat Transmission, McGraw-Hill, Third Edition, 1954.

21. R. J. Roark, Formulas for Stress and Strain, McGraw-Hill, Third Edition, 1954.

22. Tables of Thermal Properties of Gases, National Bureau of Standards Circular 564, 1 November 1955.

23. V. M. Faires, "Design of Machine Elements," Third Edition, Macmillan Co., 1955.

24. Dr. H. Schlichting, "Boundary Layer Theory," Pergamon Press, 1955.

25. M. Jakob and G. A. Hawkins, "Elements of Heat Transfer," Third Edition, Wiley, 1957.

26. A. I. Brown and S. M. Marco, "Introduction to Heat Transfer," Second Edition, McGraw-Hill, 1951.

27. R. C. Binder, "Fluid Mechanics," Third Edition, Prentice-Hall, Inc., 1955.

28. J. K. Vennard, "Elementary Fluid Mechanics," Third Edition, Wiley, 1954.

29. W. T. Thomson, "Mechanical Vibrations," Second Edition, Prentice-Hall, 1953.

APPENDIX

DATA AND DESIGN

TABLE 1

HEAT EXCHANGER CALCULATION

ROW	T (AIR) °R	T (WATER) °R	T (AVG) = Ta+Tw/2 °R	μf @T (AVG) LB/FT-SEC	DG/μf	$h_m D_o / K_f$
1	9460	620	5040	5.4×10^{-5}	173.6	6.2
2	8218	617	4418	4.9×10^{-5}	191.4	6.75
3	7262	615	3943	4.7×10^{-5}	199.5	7.00
4	6479	613	3553	4.3×10^{-5}	218.0	7.20
5	5803	612.5	3215	4.0×10^{-5}	234.4	7.30
6	5229	611.25	2920	3.9×10^{-5}	240.5	7.60
7	4788	610.3	2699	3.6×10^{-5}	260.5	7.80
8	4407	609.5	2508	3.4×10^{-5}	276.0	8.20
9	4029	608.7	2319	3.3×10^{-5}	284.0	8.10
10	3702	608.0	2155	3.2×10^{-5}	293.0	8.50
11	3415	607.4	2011	3.1×10^{-5}	303.0	8.60
12	3171	606.9	1889	2.8×10^{-5}	335	9.00
				LB/FT-HR		
13	2947	606.4	1776	.102	331	8.90
14	2720	605.9	1663	.094	360	9.10
15	2525	605.5	1565	.090	375	9.40
16	2350	605.1	1478	.088	385	9.50
17	2194	604.8	1399	.081	413	10.0
18	2046	604.5	1325	.080	422	10.1
19	1918	604.2	1261	.078	434	10.5
20	1801	604.0	1203	.076	444	11.0
21	1693	603.8	1148	.075	450	11.1
22	1595	603.6	1099	.072	468	11.5
23	1506	603.43	1054	.068	504	12.0
24	1423	603.3	1013	.067	504.2	12.2
25	1349	603.16	976	.065	520.0	12.3

HEAT EXCHANGER (Continued)

ROW	K_f @T(AVG) BTU/FT-SEC-^oR	h_m BTU/FT^2-SEC-^oR	Q BTU/SEC	T(AIR) ^oR	T(WATER) ^oR	C_p @T(AIR) BTU/LB ^oR
1	2.310^{-5}	$171(10^{-5})$	370	1242	2.68	.298
2	2.110^{-5}	$170(10^{-5})$	284	956	2.12	.2975
3	1.910^{-5}	$159(10^{-5})$	232.5	783	1.685	.297
4	1.810^{-5}	$155.5(10^{-5})$	201.0	676	1.450	.297
5	$(1.7)10^{-5}$	$148.9(10^{-5})$	170.0	574	1.230	.296
6	$(1.4)10^{-5}$	$127.6(10^{-5})$	129.6	441	0.94	.294
7	$(1.3)10^{-5}$	$122.0(10^{-5})$	112.4	381	0.81	.293
8	$(1.35)10^{-5}$	$132.(10^{-5})$	111.0	378	0.800	.292
9	$(1.30)10^{-5}$	$126.4(10^{-5})$	95.0	327.0	0.69	.290
10	$(1.20)10^{-5}$	$122.0(10^{-5})$	83.0	287.0	0.60	.289
11	$(1.10)10^{-5}$	$113.0(10^{-5})$	70.0	244.0	0.51	.287
12	1.05×10^{-5}	$113.0(10^{-5})$	64.0	224.0	0.46	.286
	BTU/FT-HR ^oR					
13	.042	$(124.6)10^{-5}$	64.0	227.0	0.47	.284
14	.039	$(118)10^{-5}$	55.0	195.0	0.40	.282
15	.037	$(115)10^{-5}$	48.8	175.0	0.35	.280
16	.036	$(114)10^{-5}$	43.5	156.0	0.31	.278
17	.035	$(117)10^{-5}$	41.0	148.0	0.296	.276
18	.033	$(111)10^{-5}$	35.2	128.0	0.254	.274
19	.031	$(109)10^{-5}$	31.4	117.0	0.230	.270
20	.030	$(110)10^{-5}$	29.0	108.0	0.210	.266
21	.029	$(107)10^{-5}$	25.7	97.5	0.186	.263
22	.028	$(107)10^{-5}$	23.35	89.2	0.169	.262
23	.027	$(108)10^{-5}$	21.47	82.9	0.155	.259
24	.026	$(106)10^{-5}$	19.12	74.1	0.1385	.258
25	.025	$(102)10^{-5}$	16.75	67.5	0.1214	.248

TABLE 2

TYPICAL DATA

POWER TO HEATER MW	AIR WEIGHT FLOW LBS/SEC	AIR ENTHALPY (AVG) BTU/LB	DISTILLED WATER ΔT °F	PLAIN COILS ΔT °F	PLAIN COILS ΔP PSI	PLATE COILS ΔT °F	PLATE COILS ΔP PSI	DIFFUSER ΔT °F	DIFFUSER ΔP PSI	FIN COILS ΔT °F	FIN COILS ΔP PSI	AIR EXIT TEMP °F
1.8	0.32	2560	7	3	22	10	7	2	11	*	*	60
1.8	0.34	2460	7	4	26	7	12	5	14	*	*	62
3.1	0.39	3630	11	6	23	12	11	3	12	*	10	65
2.1	0.33	2750	9	11	16	11	11	5	11	7	11	*
4.7	0.49	3600	NO DATA TAKEN, ALL SYSTEMS GO.									

* NO DATA DUE TO FAULTY SENSORS

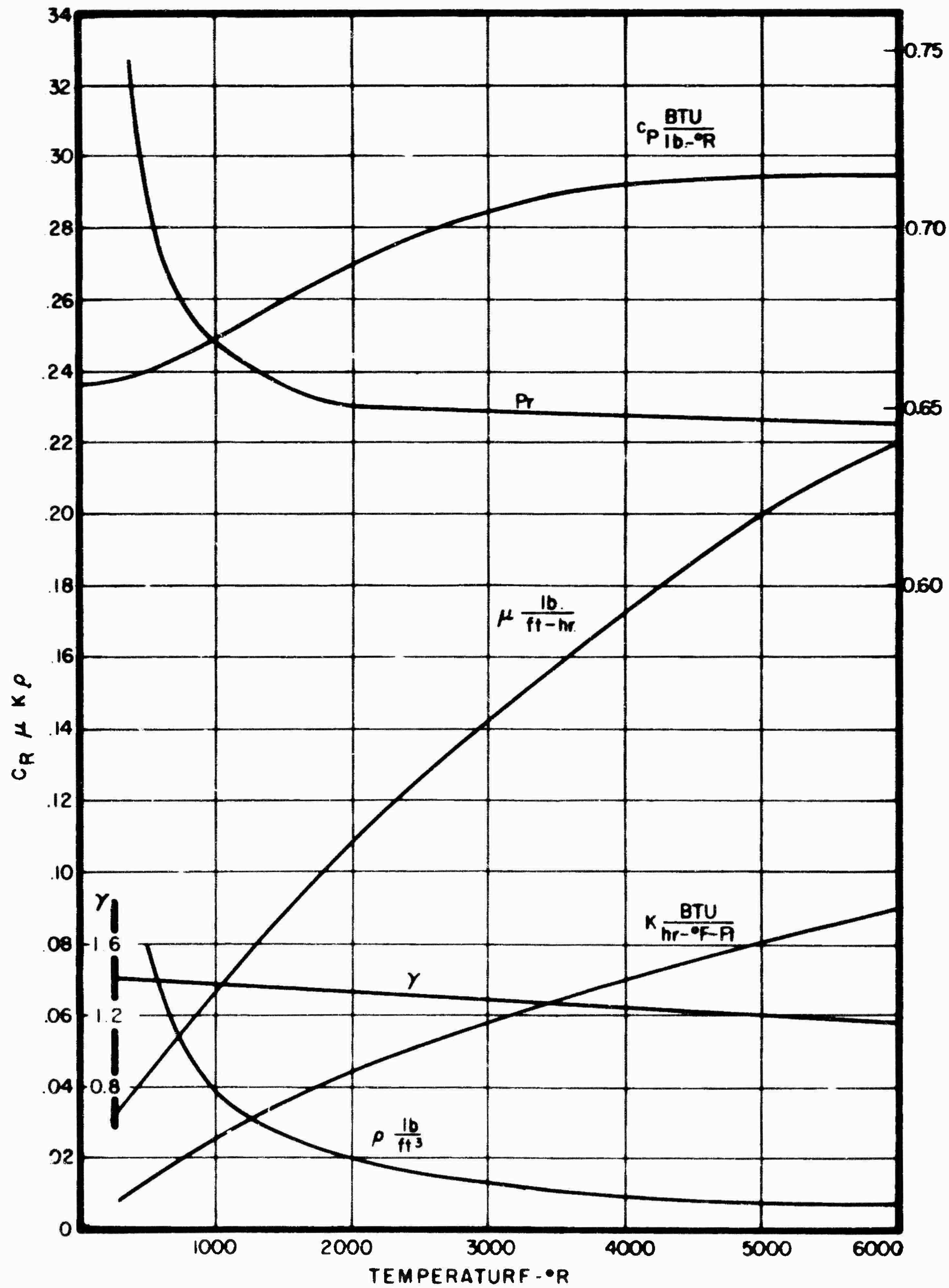

Figure 1. Properties of Air at 1 Atmosphere

Figure 2. Diffuser Section Drawing

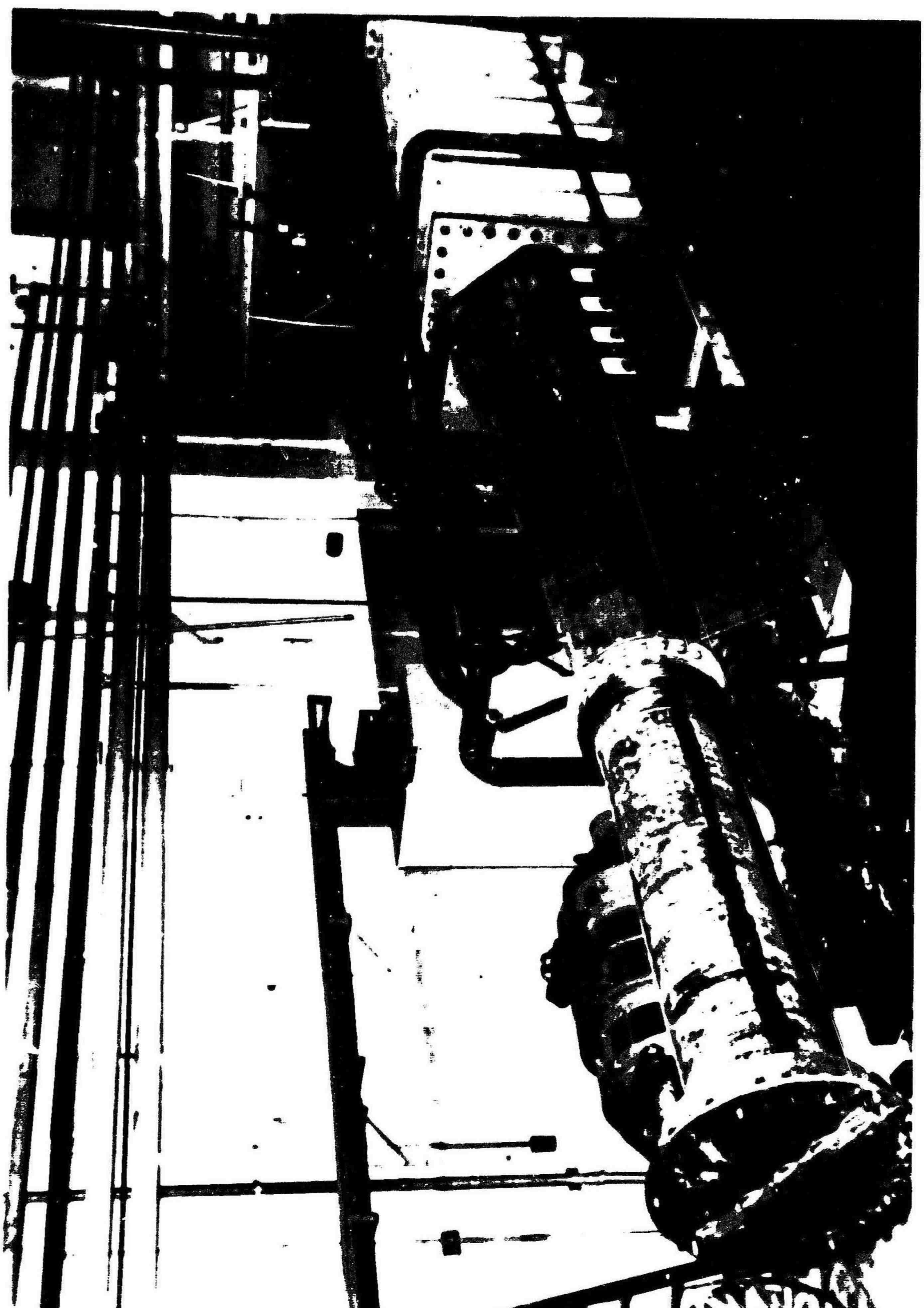

Figure 3. Diffuser and Heat Exchanger Overall View

Figure 4. Diffuser and Flow Meter Panel

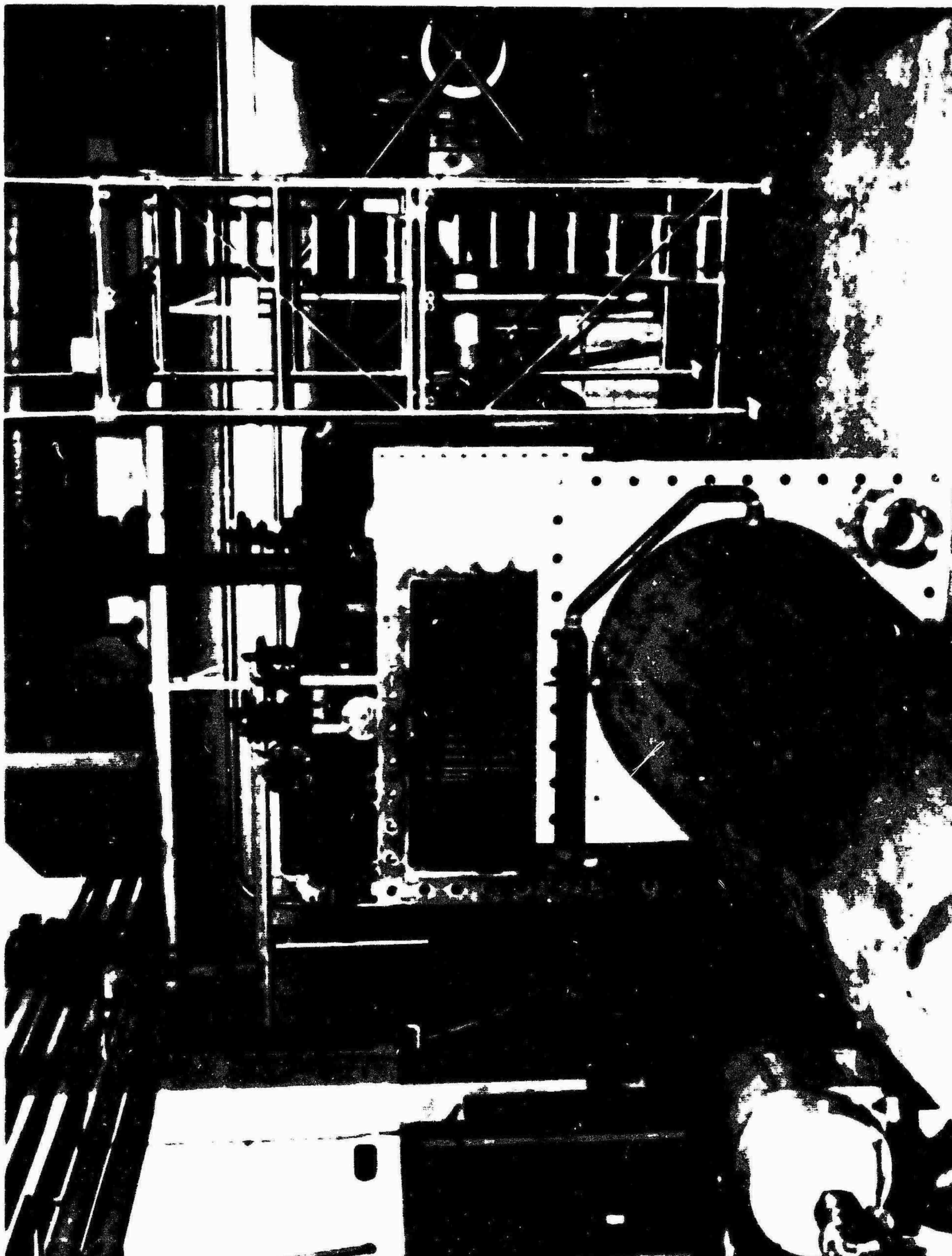

Figure 5. Heat Exchanger High Temperature Coils

Figure 6. High Temperature Coil Assembly Drawing

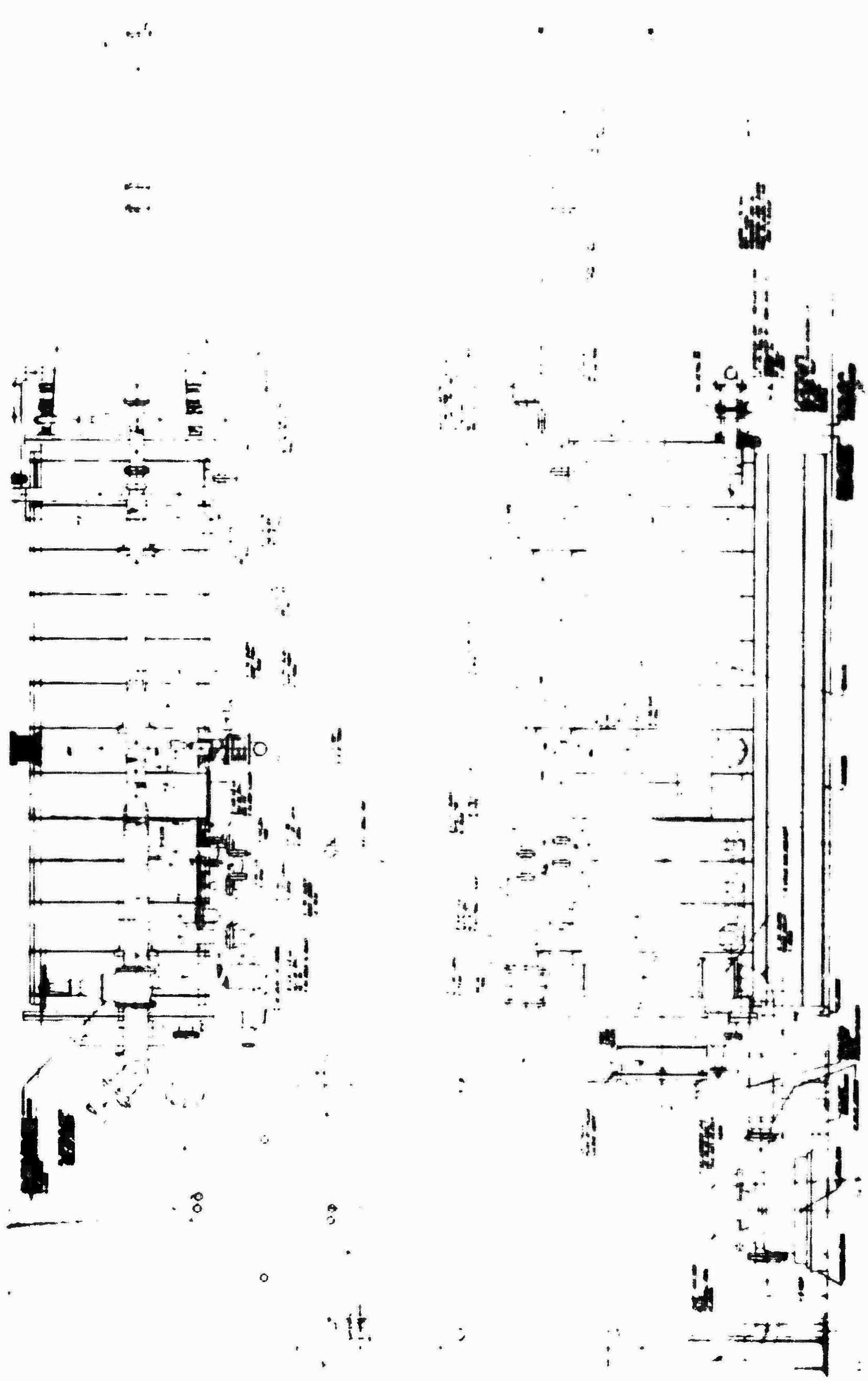

Figure 7. Heat Exchanger Installation Drawing

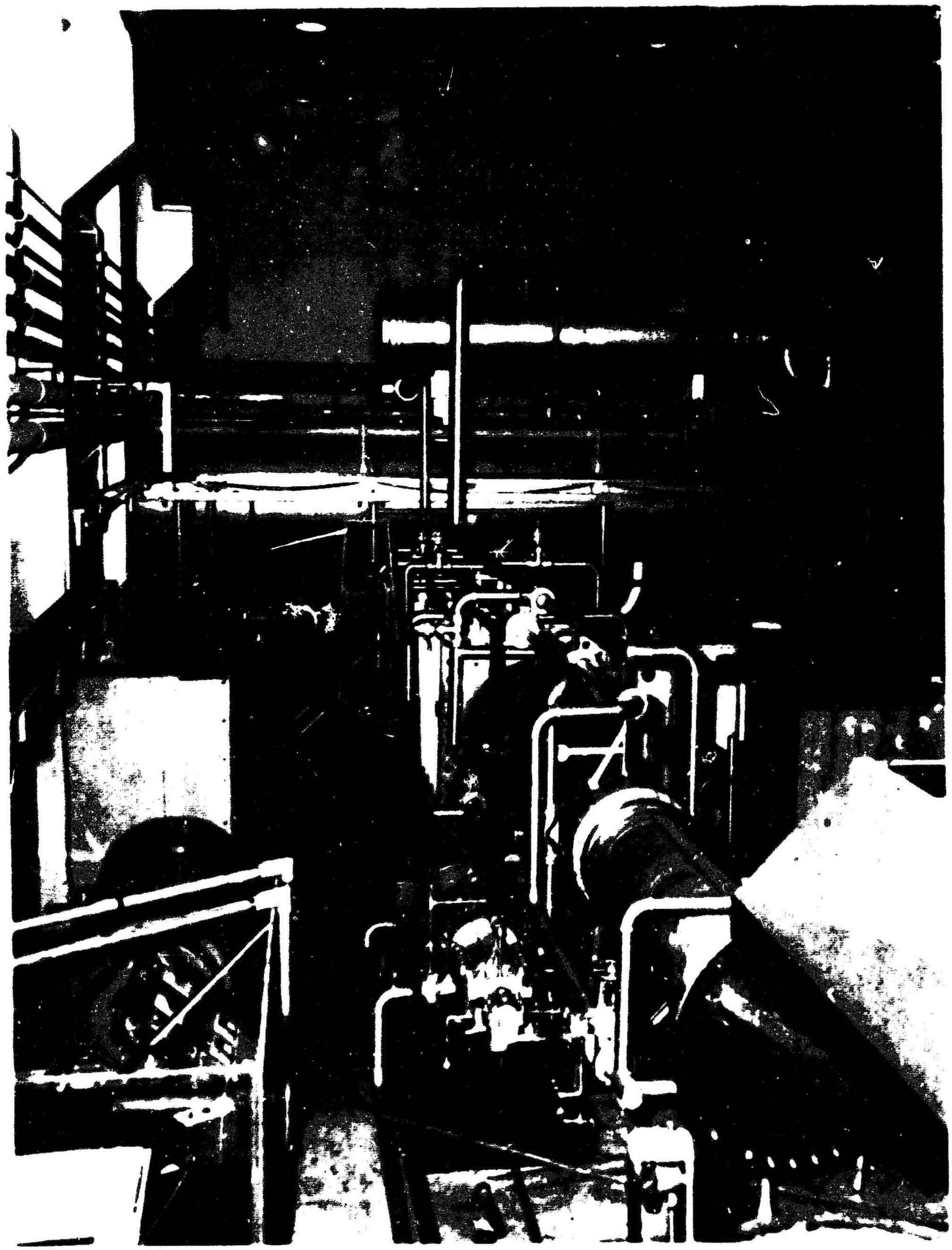

Figure 8. Water System

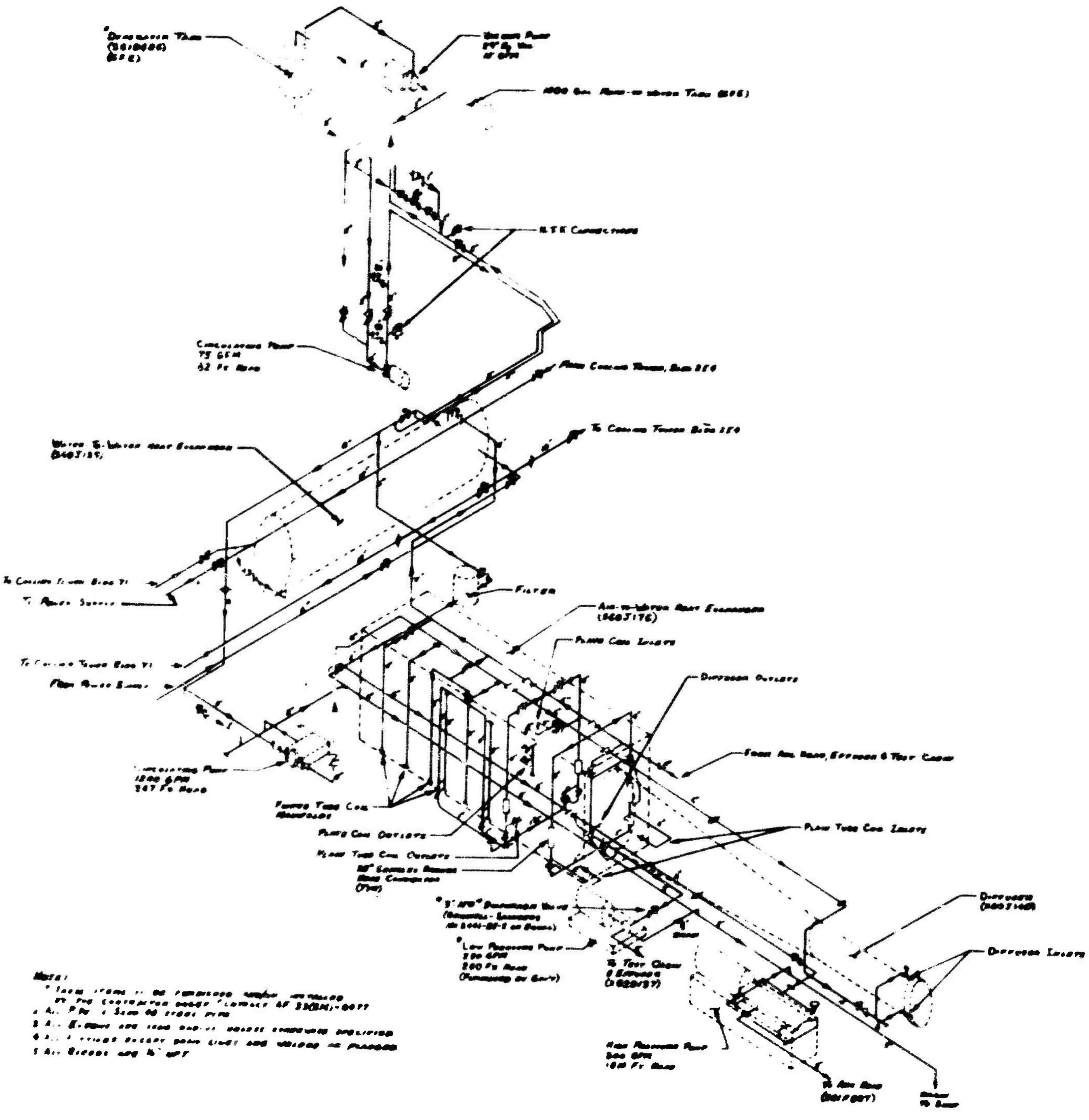

Figure 9. Water System Schematic Drawing