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FFTF DUMP HEAT EXCHANGER DESIGN & DEVELOPMENT

APPLIED TECHNOLOGY

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FIRST JOINT US/JAPAN SEMINAR ON BRP

PLANT COMPONENTS

US PAPER 7

Abstract

FFTF DUMP HEAT EXCHANGER DESIGN AND DEVELOPMENT

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This report is a brief summary of the history of procurement of the Dump Heat Exchangers for FFTF; it outlines the various organizations and respective roles. A description of the design basis and features of the large main heat transport system dump heat exchangers is included. Design evolution is discussed and testing performed as part of the design process, including finned tube qualification, header hydraulic tests, air-pressure drop tests, insulated panel tests and friction/wear testing, is summarized. Manufacturing experience, including development of an integral finned tube, development of welding procedures for joining the finned tubes, and header fabrication is also presented.

Testing results obtained with the prototype unit are described, as are additional feature model tests that were performed to define possible methods of improving heat transfer performance for the plant units.

The paper concludes with a summary of construction experience, status of plant modifications and a description of expected plant operation, including decay heat removal.

FFTF DUMP HEAT EXCHANGER DESIGN AND DEVELOPMENT

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1.0 INTRODUCTION

The Fast Flux Test Facility (FFTF), located on the Hanford Reservation near Richland, Washington, is a major center for developing liquid metal fast breeder reactor technology in the United States. Aside from its primary function of testing fuels and materials for use in future breeder reactors, the FFTF will also be a proving ground for components to be used in liquid metal systems.

The FFTF reactor core is cooled by the sodium in the Primary Heat Transport System (radioactive). The heat is transferred to a non-radioactive secondary system via a sodium-to-sodium Intermediate Heat Exchanger (IHX). Finally, the heat is dissipated to the atmosphere through the sodium-to-air Dump Heat Exchanger (DHX) system. The DHX system is composed of 12 DHX modules and will dissipate the full reactor heat load of 400 MWt. The reactor has 3 main heat transfer loops, each of which contains 4 DHX modules. The DHX modules within a given loop operate in parallel; each module dissipates 1/12 of the total reactor heat load. Isolation valves are provided for each module so that a given loop may operate on the three remaining DHX modules should a malfunction arise in any one of the four.

The sodium-to-air heat exchanger module shown in Figure 1 is comprised of several major subcomponents. These include the following:

1. Sodium boundary-tube bundle and header assembly.
2. Electric motor drive forced draft fan.
3. Air plenum and ducting system, including tube bundle housing, exhaust stack, fan inlet and fan outlet ducts.
4. Oil fired heating system.
5. Structural support system.
6. Airflow control devices, including modulating and shut-off equipment with actuators.
7. Inlet and outlet piping from the headers to the interface point with the Heat Transfer System piping.

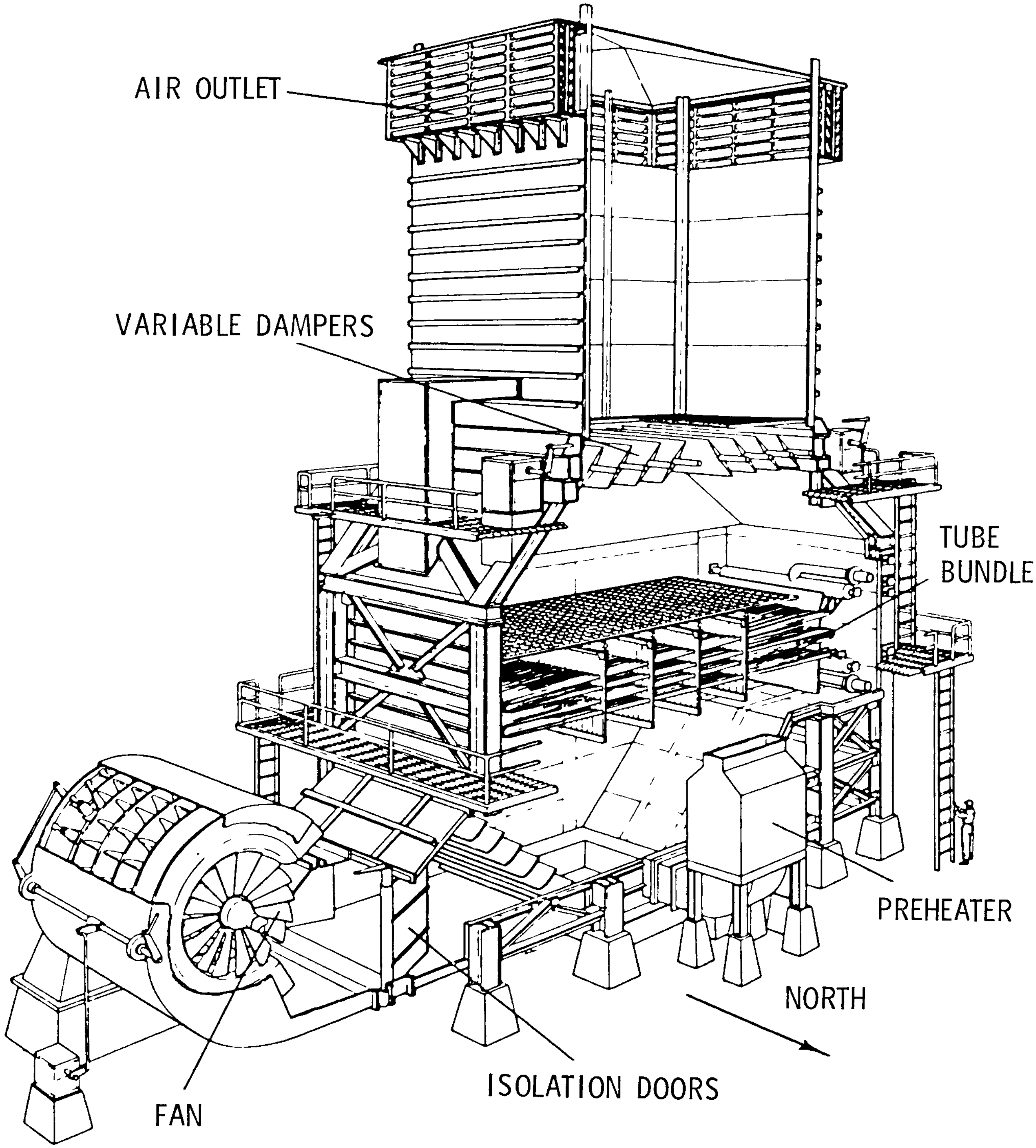

FIGURE 1 HTS/DHX, 33 MWT Dump Heat Exchanger - Cutaway View

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The stainless steel tube bundle and header assembly forms the heart of the heat exchanger and is comprised of 66 tubes with integral fins, each making four passes across the air stream between the inlet and outlet header assemblies. The tubes are supported by four tube support assemblies (TSAs) and are fixed at the header ends.

The DHX system in FFTF performs numerous functions other than its primary function of removing reactor heat. The DHX system is designed to control the temperature of the reactor and heat transport systems and to minimize any structural damage resulting from normal and off-normal temperature transients. The control system is required to operate under a variety of circumstances from full power, steady-state conditions to decay heat removal under total loss of electrical power. The DHX system must also be capable of a controlled preheat in preparation for sodium fill, must limit plant heat losses under extended loss of power conditions, and must be capable of accommodating numerous failure mode conditions.

2.0 SPECIFICATION DEVELOPMENT

The design of the sodium-to-air heat exchanger is governed by the design specification HWS-1530. The development of this specification was based on the RDT Standard E4-7T for Sodium-to-Air Heat Exchangers which was prepared as part of the LMFBR Standards Program for the U. S. Atomic Energy Commission (now the U. S. Department of Energy - DOE). HWS-1530 invokes numerous national codes and standards. The governing design criteria are contained in Section III of the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code with appropriate high temperature code cases. Appropriate standards for architectural, civil, mechanical, and electrical disciplines together with Hanford Site Standards and State and City Codes and Regulations are also included.

In addition, the DHX system must withstand catastrophic natural phenomenon such as earthquakes and tornadoes and maintain its capability to safely shut down the reactor and provide adequate cooling thereafter. The DHX must maintain integrity of the sodium boundary and provide controlled removal of reactor decay heat following the occurrence of a Design Basis Earthquake. The DHX itself is not inherently tornado hardened. Decay heat removal following the design basis tornado is provided by one heat transport loop which contains four DHX modules. In this loop, the DHX's are tornado hardened by adding a supplemental 1.27 cm (1/2") steel plate enclosure around the four modules. The control system for this loop is also tornado hardened and is provided with emergency electrical and pneumatic supplies.

The specified service life of the FFTF and the DHX is 20 years (175,200 hours) with a 75% plant factor. For analysis purposes, this time was distributed as follows:

Full Power Operation - 131,400 hours
For purposes of creep calculation, the operating condition with the largest creep damage factor was assumed for the full 131,400 hours.

Operation During Transients - 13,650 hours
The transients used for design are defined in Table 1 and 2.
Isothermal Testing - 700 hours at 427°C (800°F).
Refueling - 29,450 hours with DHX at standby with sodium temperature below 375°C (707°F).

The DHX is designed for the following steady state operating conditions:

Tube Side	Initial	Rated	Advanced
Sodium Inlet Temp C,(F)	412 (773)	518(965)	518(965)
Sodium Outlet Temp C,(F)	268 (515)	375(707)	324(615)
Sodium Flow $(x10^6)$Kg/hr, (lbs/hr)	.65 (1.43)	.66(1.45)	.48(1.06)
Max. Sodium Pressure Drop m,Na	40.5		
Heat Load (MWt/module)	33.3	33.3	33.3
Air Side			
Ambient Air Temperature C, (F)	32(90)		
Wind Velocity kph,(mph)	19.3 (12)		
Barometric Pressure (MmHg)	760		
Humidity (%)	15		

The operating requirements for transient conditions are defined in HWS-1530 and are based on a failure mode analysis of the reactor and heat transport systems. Both likely and unlikely events were considered. The DHX control system is designed to maintain the temperature transients within the envelope defined by these specific thermal transients.

TABLE 1

NORMAL CONDITION DESIGN THERMAL TRANSIENTS

EVENTS	SYSTEM LOCATION	TEMPERATURE °C, (F) INITIAL	FINAL	NUMBER OF OCCURRENCES
1. Heatup of Dry System	All locations	Ambient	204,(400)	10
2. Cooldown of Dry System	All locations	204,(400)	Ambient	10
3. Normal Startup	Secondary Hot Leg	204,(400)	538,(1000)	843
	Secondary Cold Leg	204,(400)	394,(741)	843
4. Normal Shutdown	Secondary Hot Leg	538,(1000)	204,(400)	118
	Secondary Cold Leg	364,(471)	204,(400)	118

TABLE 2

UPSET AND EMERGENCY THERMAL TRANSIENTS
AT THE DHX INLET NOZZLE

EVENT NUMBER AND DESCRIPTION		DESIGN EVENTS	NOTES
U1	Reactor Scram	658	1 2 3
U2	Safety or Control Rod Drop	20	2 3
U3X	Loss of Electrical Power to One Primary Pump	7	2 3
U4X	Loss of Electrical Power to One Secondary Pump	7	2 3
U5X	Loss of Airflow Through One DHX Module	7	2 3
U6X	Loss of Airflow Through One DHX Unit (All Four Modules in One Loop)	7	2 3
U7	Closure of Isolation Valve on One DHX Module	7	2 3
U9X	Scram With Excess Airflow in One DHX Unit (All Four Modules in one Loop)		

NOTES:

1) This event is assumed to begin with DHX inlet/outlet temperatures at inital design conditions, Page 4.

2) This event is assumed to begin with DHX inlet/outlet temperatures at rated design conditions, Page 4.

3) This event is assumed to begin with DHX inlet/outlet temperatures at advanced design conditions, Page 4.

TABLE 2 (Con't)

EVENT NUMBER AND DESCRIPTION	** DESIGN EVENTS	NOTES
E1 Primary Pump Mechanical Failure		3
E2A Secondary Pump Mechanical Failure		2 3

** The DHX is designed to accommodate five events of the most severe Emergency Event under the temperature conditions corresponding to either note 2 or 3. If the consecutive occurrance of any two emergency events produces a more severe effect than two cycles fo the most severe isolated Emergency Event, the DHX must also accommodate this more severe sequence. Only one consecutive event combination need be considered.

The DHX is also designed for off-normal operating conditions, such as:

- Decay heat removal during postscram conditions in a natural convection mode on the air side.
- Decay heat under total loss of electrical power.
- Minimized system heat losses during periods of extended loss of power conditions.
- Dry preheat of the DHX tube bundles to 204°C (400°F) for sodium fill.

These modes of operation do not significantly affect the high temperature design considerations but do have a major impact on the control system design and DHX system configuration.

Lesser design considerations for the sodium-to-air heat exchangers include the following:

- Elevation of the tube bundle thermal center to facilitate natural circulation in the sodium system.
- Natural air circulation.
- Sodium remelt.
- Ability to drain.
- Transportation considerations.
- Maintainence considerations involving tube replacement.
- Sodium fire suppression.
- Tube vibration.
- Ease of construction and weld inspection.
- High wind loads and rain protection.

3.0 PROCUREMENT

The DHX system for FFTF was developed and procured through the efforts of several organizations. The original approach called for design, fabrication and testing of a lead or prototype unit before manufacturing the plant units. Much of the raw material was purchased in plant quantities for reasons of economy. When a critical need for the components developed at the construction site, it was infeasible to conduct a complete prototype development and testing program prior to fabricating the plant units. The design was evaluated and full production of the plant units was immediately initiated. The first completed unit was shipped to the Liquid Metal Engineering Center, near Los Angeles, California where it was tested at power with sodium in the Sodium Component Test Installation. The remaining 11 units were fabricated and shipped directly to the site for installation.

The major participants in developing, fabricating and installing the DHX units at FFTF are as follows:

- USERDA (now USDOE) is the government agency responsible for the FFTF project. ERDA participated in major design reviews and approved top level criteria, funding and schedules.

- Westinghouse Hanford Company, the prime contractor to ERDA for design, construction, and operation of the FFTF operates the Hanford Engineering Development Laboratory (HEDL). This organization was responsible for development of the Sodium-to-Air heat exchanger specification and management of the Struthers Wells contract for design and fabrication of the DHX modules.
- Struthers Wells Corporation was contracted by HEDL to provide design and fabrication of 12 DHX Modules to be installed in FFTF. Struthers was responsible for the structural, hydraulic and thermal design of the DHX. They performed the ASME Section III Class I pressure boundary analysis and other design design efforts including architectural, civil/structural, thermal, electrical and mechanical. As the main contractor for procurement of the DHX, Struthers procured hardware from several subcontractors, including:

    1) Air moving system including motor, variable speed coupling, fan and associated inlet housing, and speed control systems.
    2) Oil-fired preheater and associated control system.
    3) Structural steel air plenum insulating panels, exhaust stack and rain hood.

- Liquid Metal Engineering Center (LMEC) was responsible for installation and testing of the prototype DHX and its control systems.

Figure 2 is an organization chart showing contractor interfaces.

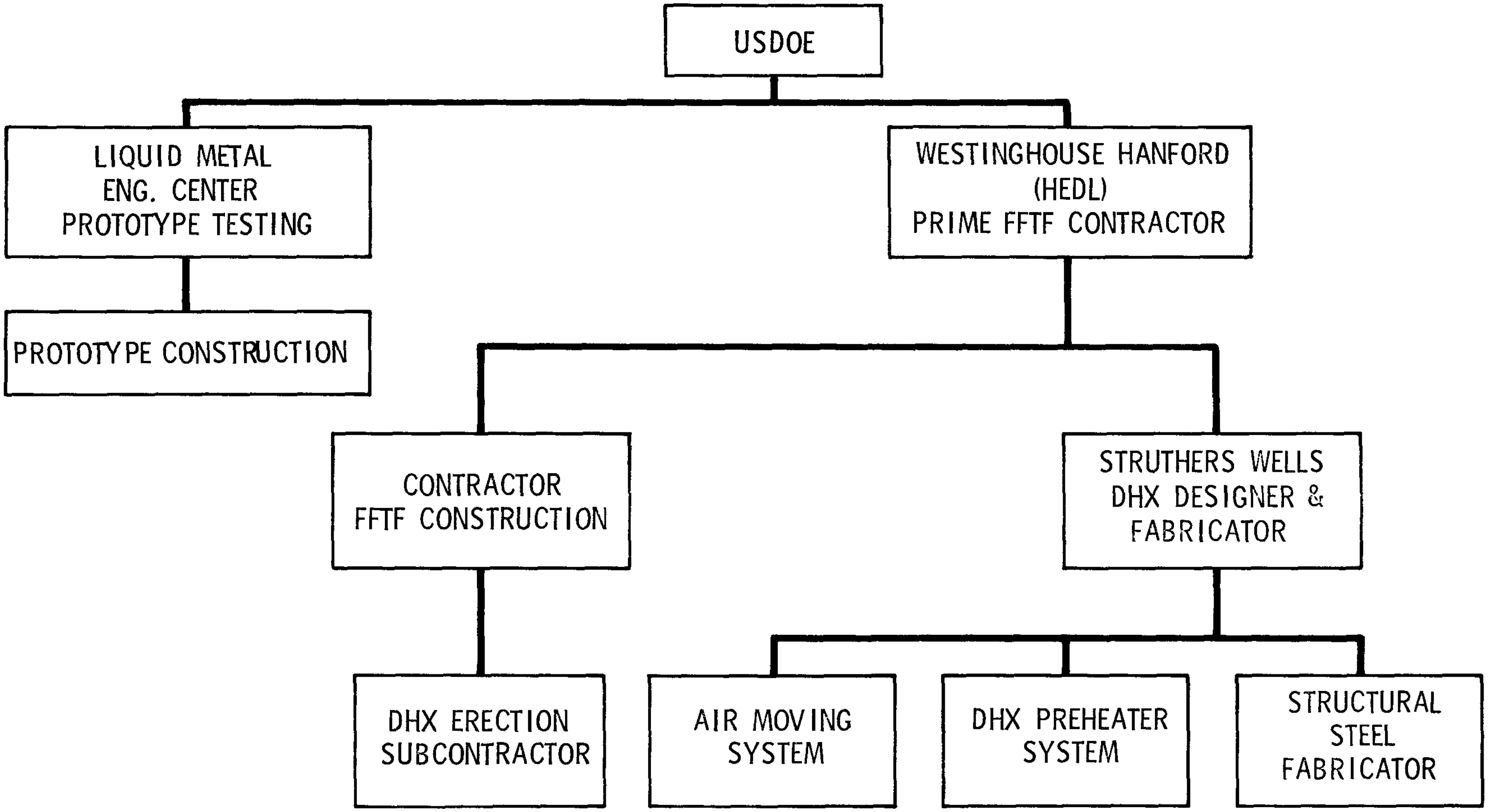

FIGURE 2 Organization for Dump Heat Exchanger Procurement

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4.0 DESIGN, MANUFACTURING & INITIAL FEATURE TESTING

Three requirements significantly impacted the design of the DHX. These requirements are as follows:

1. The application of ASME Section III, Class 1 stress analysis requirements to a serpentine coil heat exchanger.

2. The application of severe seismic design criteria.

3. The multitude of operational transients above 427°C (800°F) sodium inlet temperature, which required strain racheting and creep fatigue evaluations.

Other requirements contributing to a lesser degree are described in Section 2.0.

Availability of materials and manufacturing capability were also considered. During the design of the DHX, it was necessary to develop autogenous tube to tube welding equipment and techniques, an extended fin surface acceptable to ASME Section III, Class 1, and a unique approach to inlet/outlet header design. Ancillary equipment suppliers were developed for control equipment, oil fire preheaters, fan and drive train assemblies, and air flow dampers that were capable of meeting exacting seismic design and system controllability requirements.

Early design efforts centered primarily on the tube bundle arrangement to satisfy steady-state performance requirements and ASME pressure calculations. Each component was sized with appropriate stress margins that were later used for the combination of thermal and seismic stresses, since much of the actual design criteria, including transient and seismic conditions, was developed as the design progressed.

Final design effort centered on meeting thermal and seismic design of the ASME Section III, Class 1 Code, High Temperature design (FRA-152-3) and seismic criteria. (JABE-WADCO-02).

4.1 Tube Bundle

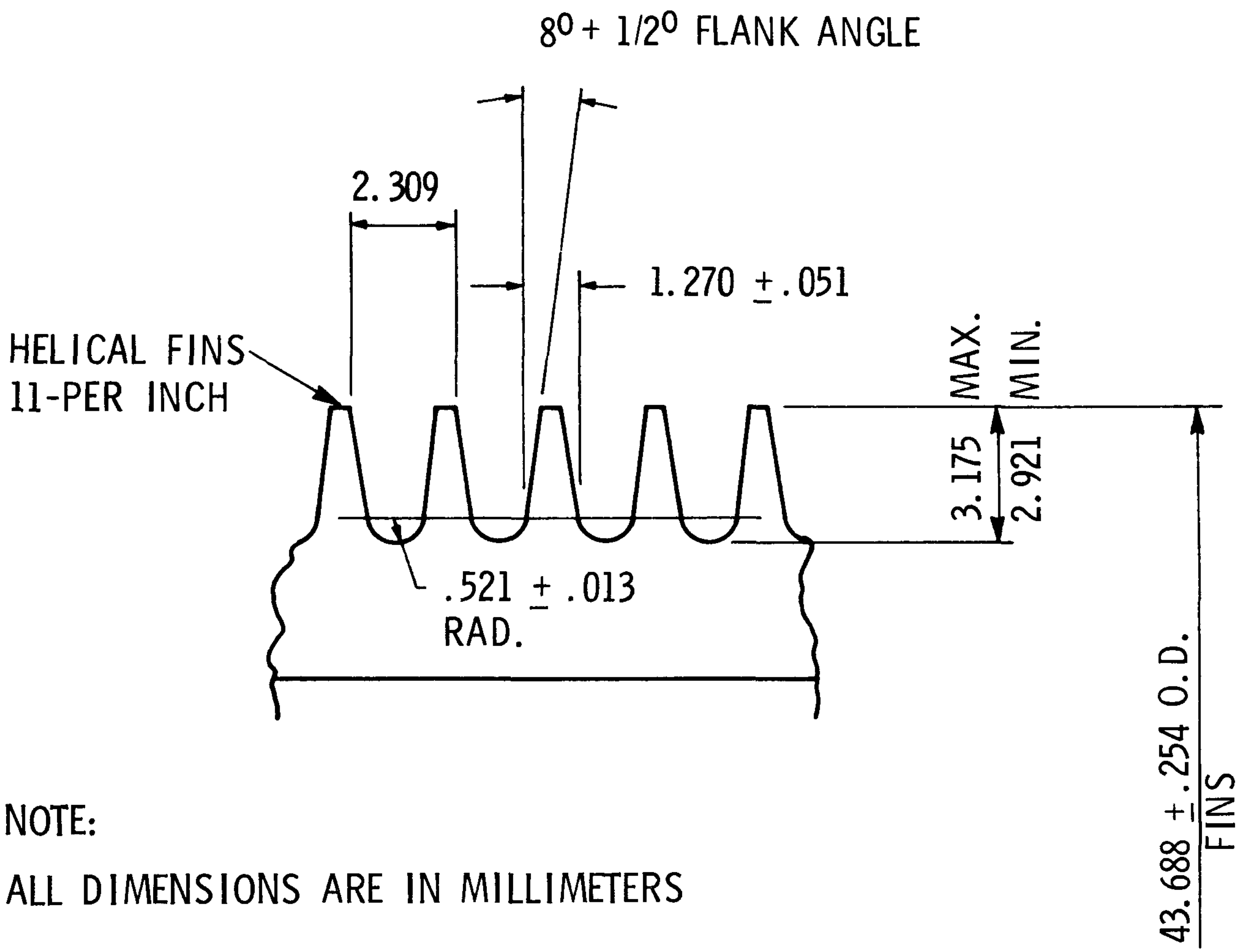
The initial tube bundle design effort involved selecting a tube bundle configuration that would provide the necessary thermal and hydraulic performances for steady-state conditions. The use of the bare tubing was rejected because of the size of the resulting tube bundle. Shipping limitations would have made it necessary to ship each bundle in two halves and to complete assembly and testing in the field. It was also felt that the ratio of the bundle's width to length, which approached 1.0, would make air flow distribution a more difficult problem. Quality assurance requirements, and the requirements of ASME Section III, Class 1 design precluded the use of commercially available extended surface welded fin tubing.

Initial experimentation with integrally machined fins indicated that production would be a lengthy and expensive undertaking. Based on samples of commercially available extruded finned tubing and preliminary development results, it was believed possible to develop an extruded fin tube of reliable quality, within schedule requirements. Accordingly, tube bundle design proceeded in parallel with finned tube development and other aspects of DHX design outlined earlier.

During later extruded fin development work, it was determined that the required fin surface would not be produced with a single pass extrusion operation. Two pass extrusion would have caused concern about quality. Because the double processing required annealing the tube between extrusion steps, both the costs and the uncertainties of extruded fin tube production were considered unacceptable.

After an intensive effort involving numerous potential machining subcontractors and machine tool manufacturers, a machined fin (Fig.3) meeting all necessary design requirements was developed at Struthers Wells. During the course of DHX fabrication, over 32 km (20 miles) of machined fin tubing were produced. The machined fin tube was qualified for use through a series of simulated sodium transient tests. The transients selected for the tests were calculated to produce material fatigue at least equivalent to that calculated for the DHX service life. Post-test examinations of the machined fin specimens did not reveal any cracks or other damage. Also, there was no significant difference in tensile properties or results of a 180 degree bend test before and after qualification testing.

A second manufacturing consideration, which had a significant impact on tube bundle design, was the development of stainless steel, tube to tube, automatic homogenous (autogenous) welding capabilities. Because of the number of tube to tube, tube to tube bend, and tube bend to header welds involved, it was believed necessary to use an automatic non-filler metal welding technique, in order to minimize both cost and possible quality concerns associated with more conventional approaches. The availability of autogenous welding equipment capable of welding within the close geometry of the tube bundle, limited weld configurations to relatively thin wall, tube to tube sections. The physical size of the welding equipment also imposed requirements on tube bundle design in order to provide adequate clearance for the welding equipment in the weld areas. Developments in tube to tube autogenous welding equipment during the DHX project allowed the critical weld details to be increased in thickness; this was a crucial factor in the eventual success of the DHX mechanical design effort.



NOTE:

ALL DIMENSIONS ARE IN MILLIMETERS

FIGURE 3 DHX Machined Finned Tube

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The initial tube bundle design featured 66 serpentine coils (Figure 4). Each coil was fabricated from four straight lengths of extended finned tube with the ends stripped and prepared for welding to 180-deg. bends (actually 179-deg. 30 min. to provide for the slope), and to the 45- and 90-deg. bends which are used to attach the finned tube to the headers. Two barrel type headers were used to distribute sodium inlet and outlet flow.

The inside diameter of the serpentine coil tubes was established at 31.5 mm (1.237-in. nominal), based on hydraulic design considerations, to maintain tube bundle sodium pressure drop within the desired limits. Factors influencing the choice of the internal diameter included the necessity for a high enough sodium side pressure drop to assure good sodium flow distribution across the bank of tubes, within the limits of the FFTF's secondary sodium pump design criteria. Minimum I.D. was specified at 25.4 mm (1 in.) in order to minimize sodium blockage. Tube size was, of course, a major factor in thermal design of the tube bundle. Finned tube configuration was originally based on the requirements of the extrusion process. It was necessary to maintain appropriate relationships between desired extruded fin height of 3.0 mm (1/8 in.) and the resulting tube wall thickness.

A basic tube wall thickness (base of fin to inside surface of tube) of 2.41 mm (0.095 in.) was chosen, which limited the combination of pressure and static bending stresses to less than 50 per cent of high temperature primary stress levels. Seismic stress calculations were based on equivalent static loads of 1.5 times the peak ground response accelerations for the site pending the completion of dynamic analysis.

Various header designs made of weldments were considered and rejected. The present headers were machined from single extrusions. The cross section of the extrusion is a heavy wall cylinder which is machined as a single piece, including integral stub nozzles for serpentine attachments. In order to maintain equal sodium flow through all 66 serpentines over a wide range of operating conditions, an internal flow distribution baffle was designed to be welded to the inside of the headers. A full scale model of the header and baffle was made and used to test flow distribution at various flow rates. Test results confirmed that the resulting baffle design was adequate.

Other components of the header assemblies include a 20.3 cm (8 in) inlet nozzle, one 20.3 cm (8 in) outlet nozzle, and two elliptical heads with integral trunnions for header support, one on either end of the header barrel.

The serpentine coils are supported by tube support assemblies (TSA's) placed at intervals along their length. In order to permit serpentine assemblies to be fabricated as sub-assemblies and applied to the headers at final assembly, it was determined to design the TSA's in vertical segments. It was then possible to fabricate every other serpentine coil with the TSA segment in place around the serpentine tubes. Thus, at final assembly, the TSA's are assembled, as the serpentines are sequentially applied to the headers. Tube support assemblies are then completed by applying top and bottom cross bars, side bars and segment spacer dowel pins. The dowel pins have tolerances to ensure that in the event of TSA shifting or differential segment thermal growth, the dowel pin will keep the segments from binding the serpentine coil tube. The TSA's are approximately 3.66 m. (12 ft.) wide, 1.52 m. (5 ft.) high and 25.4 mm (1 in.) thick. They are fixed (bolted to the structure) on one side and allowed to expand to the other side.

TSA design was involved because of thermal gradients and differential thermal expansion between the top and the bottom of the TSA assembly.

The initial design called for TSA's on 1.07m (42 in) spacing in order to support the serpentine and minimize seismic stresses, and in order to establish tube natural vibration frequencies at acceptable levels. This close spacing created design problems because of the differential expansion of the hot headers and relatively cool first TSA. The latter is only 1.07 m (42 in) from the header which resulted in high bending stesses in the finned tube. The close TSA spacing also created a problem at the return bend end of the tube bundle by restraining the serpentine's tendency to bend down because of differential expansion of the top or hotter tube on each hairpin versus the lower cooler tube.

In order to reduce the foregoing problems, it was felt necessary to eliminate a number of the TSA's. This, in turn, meant that the specified tube natural vibration frequency limitations had to be relaxed, which aroused much concern about air flow induced vibration during operation. Preliminary air flow vibration testing was authorized and carried out. The test results showed that the air stream did not have enough energy to produce tube vibrations for the 1.52m (60 in) spacing between supports. Based on these tests, prototypical air flow vibration tests using five full length U-tubes and actual air flow conditions were run with on four TSA's at 2.11 m (83 in) spacing. These tests were successfully completed in ASME Paper No. 75-PWR-B. It was concluded that a less than 150 per cent of the design air velocity, insufficient energy is available to cause tube vibration problems.

The thermal expansion of the serpentine coils is so severely restrained by the supports that high normal forces exist between the supports and fin tubes. The friction factors used for the design between the tube and its supports are critical, and the validity of the factor used had

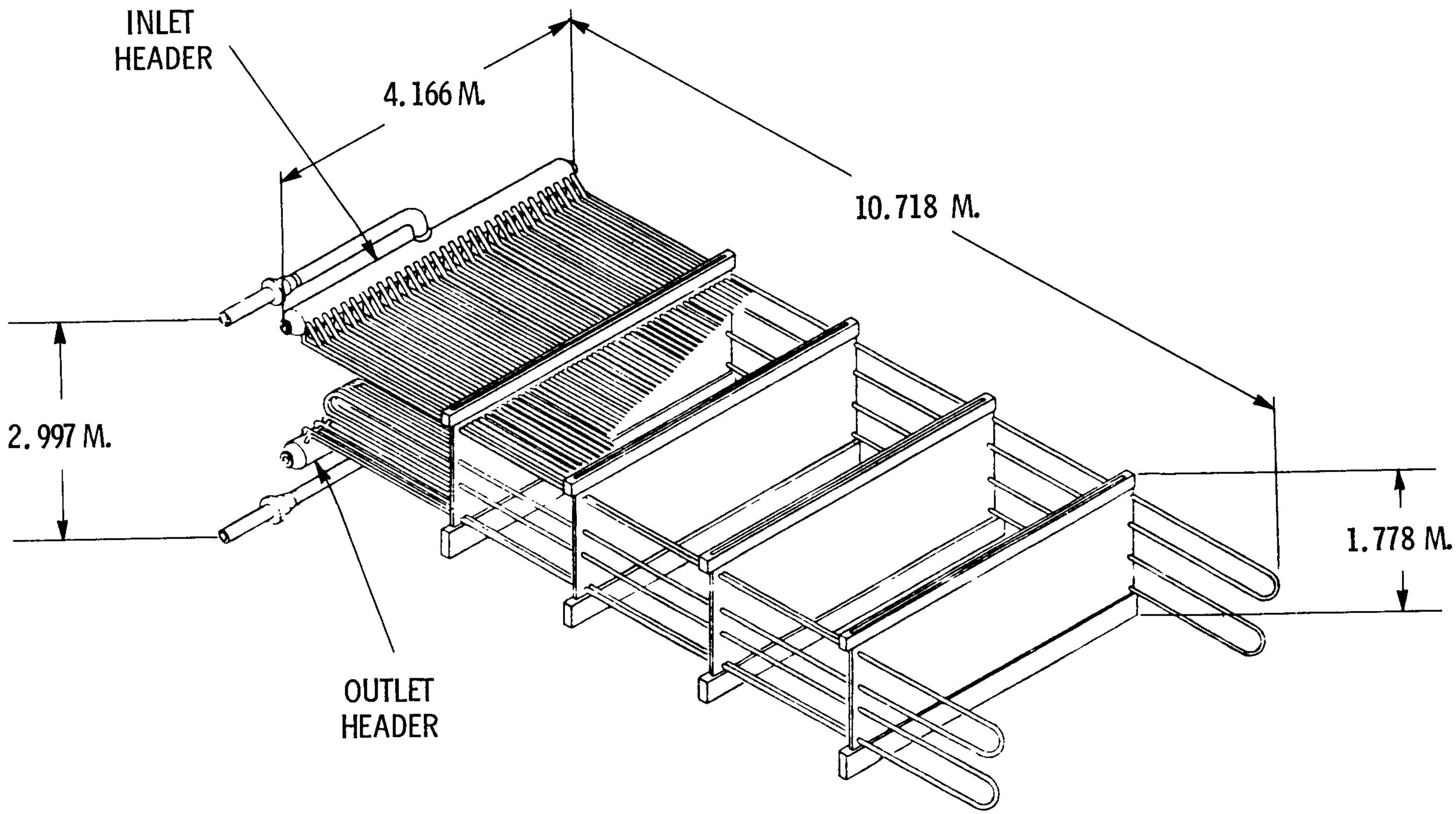

FIGURE 4 Initial Design of Tube Bundle

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of the factor used had to be established. Both static and dynamic testing programs were initiated to establish the required break-in friction factors and to determine "wear-in" characteristics and fatigue damage for the finned tubes placed in prototypic tube support plates under cyclic motion while loaded with the maximum expected normal loads. Westinghouse/ARD performed these prototypic tests with results establishing friction coefficients of .8 and adequate fatigue life.

4.2 Support Structure and Air Plenum

The tube bundle is installed about 7.62 m (25 ft) above ground level, within an insulated plenum and supported by structural steel frame, which is anchored to the foundation. The steel frame consists of a bolted connection design of rolled sections including columns, struts, braces and beams. The structure encloses and supports the heat exchanger equipment, including plenum, primary sodium piping, a tube bundle and sodium headers. The structure is designed to withstand both dead loads and live loads including component weights, thermal expansion, seismic forces, tornado winds and sodium piping reactions. The thermal design for the structure was based on a maximum plenum air temperature of 260°C (500°F) below the tube bundle and 427°C (800°F) steady-state operating temperature at or above the tube bundle with a maximum transient air temperature of 538°C (1000°F).

The air plenum provides the necessary air passage from the forced draft fan to the sodium-to-air heat exchanger and continues upward to the stack and rainhood. In order to provide for the necessary thermal insulation and sodium fire containment the insulation panels were designed with an inner liner of 10 gauge carbon steel (A243 for high temperature service and corrosion resistance), three inches of block insulation (A.P. Green Insblok No. 19) and an outer liner of .48 cm (3/16") carbon steel. This construction was tested under normal operating thermal conditions to determine adequacy of the design. The results showed that the panel corners and flanged weld seams should not fail under normal conditions. No long term life type tests were conducted, but computer analysis of the panels showed fatigue life beyond the required service conditions. Since the design requires that both the sodium and sodium fires be contained within the casing, additional testing for sodium jet impingement was conducted by L.M.E.C. on a typical insulation panel to determine that no penetration or significant damage occurred to the panel. In order to maintain complete isolation of the plenum in case of a sodium fire or during stand-by operation, a pneumatic actuator controlled isolation gate and stack damper are provided at each.

An important part of the plenum design included determining the air flow requirements and distribution of air equally over the heat exchanger surfaces. Since air side pressure drop will directly influence the required fan horsepower for a given air flow, an arrangement was selected for the plenum which minimized the required number of expansions, contractions and turns of the air flow path. The total air side pressure was then calculated for the initial core design air flow of 1,030,000 kg/hour (2,265,000 /hr). This required the use of a 900 HP Forced Draft Fan/Motor assembly. Since the size of the fan was so important to the success of the design and existing data did not allow calculation of tube bundle pressure drop with a high degree of certainty, experimental tests were undertaken using actual fin tubes in a prototypic design layout. Using the test results and detailed calculations of pressure loss in the plenum, total pressure drop was calculated at 27 cm (10.93") of H_2O and a 1250 HP fan was selected with an available static pressure rise of 30 cm (11.8") of H_2O at design conditions.

A more detailed account of DHX design experience is contained in ASME Paper No. 76-JPGC-NE-7 entitled "Considerations in Designing High Temperature Dump Heat Exchanges".

5.0 PROTOTYPE TESTING

From September 1974 to June 1976, construction, testing and disassembly of the prototype unit took place at the Liquid Metal Engineering Center near Los Angeles, California. The test loop to which the DHX was attached was the Sodium Components Test Installation (SCTI) currently undergoing expansion and modification to test the first generation of modular steam generators for the fast breeder program. The loop contained a 35 MWt heater capable of producing enough heat to evaluate design and off normal DHX heat removal capability. The test program shown in Figure 5 was divided into the following phases:

- Erection
- Preoperational Inspection and Checkout
- Steady State Operation
- Transient Operation
- Modification Testing
- Post Service Inspection and Disassembly.

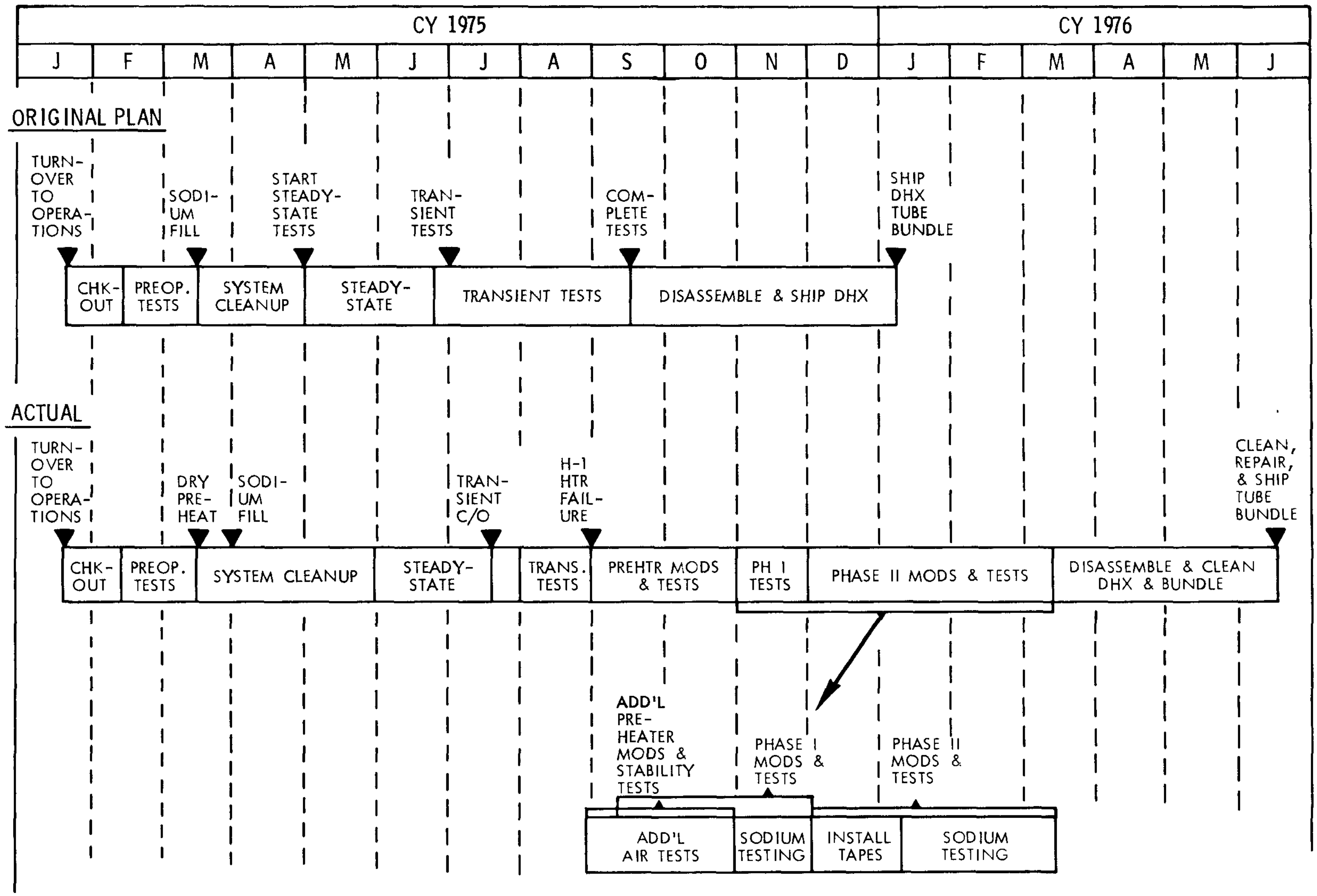
5.1 Erection

Erection procedures for the HTS/DHX were prepared by Struthers Wells Corporation (SWC) for use at LMEC and were subsequently modified for use at FFTF based on LMEC experience. A summary report dealing with problems encountered during DHX erection at LMEC was prepared by the erection contractor.

Table 3 itemizes the problems that were identified as well as probable causes, how they were solved at LMEC, and recommended action for FFTF. In general, the problems were caused by minor design inconsistencies and drawing discrepancies and were readily corrected during the erection process. Valuable experience on installation sequencing and reassembly was gained and applied to FFTF, which resulted in a reduction in assembly time. Erection at LMEC was completed in $\sim$ 4 months. Figure 6 shows a composite of various DHX assembly phases.

5.2 Inspections

The preoperational tests were performed as an overcheck of the DHX installation and to make functional checks of controls and equipment. Specific objectives of these tests were to: 1) determine DHX actuator failure modes; 2) determine the freedom of movement of the preheater air duct dampers, variable inlet vanes, exhaust stack dampers, and isolation gates; 3) perform a functional checkout of the DHX controller by manually operating over full range while measuring fan speed control signal, fan speed, and actual positions of the fine and coarse dampers; 4) verify interlock functions for proper operation.



* PHASE I INCLUDES AIR FLOW MEASUREMENT SYSTEM MODIFICATIONS.
PHASE II INCLUDES INSTALLATION OF ASME ORIFICE TO VALIDATE SODIUM FLOW.

FIGURE 5 HTS DHX Test Program Schedule Showing Original Plan and Actual Activities

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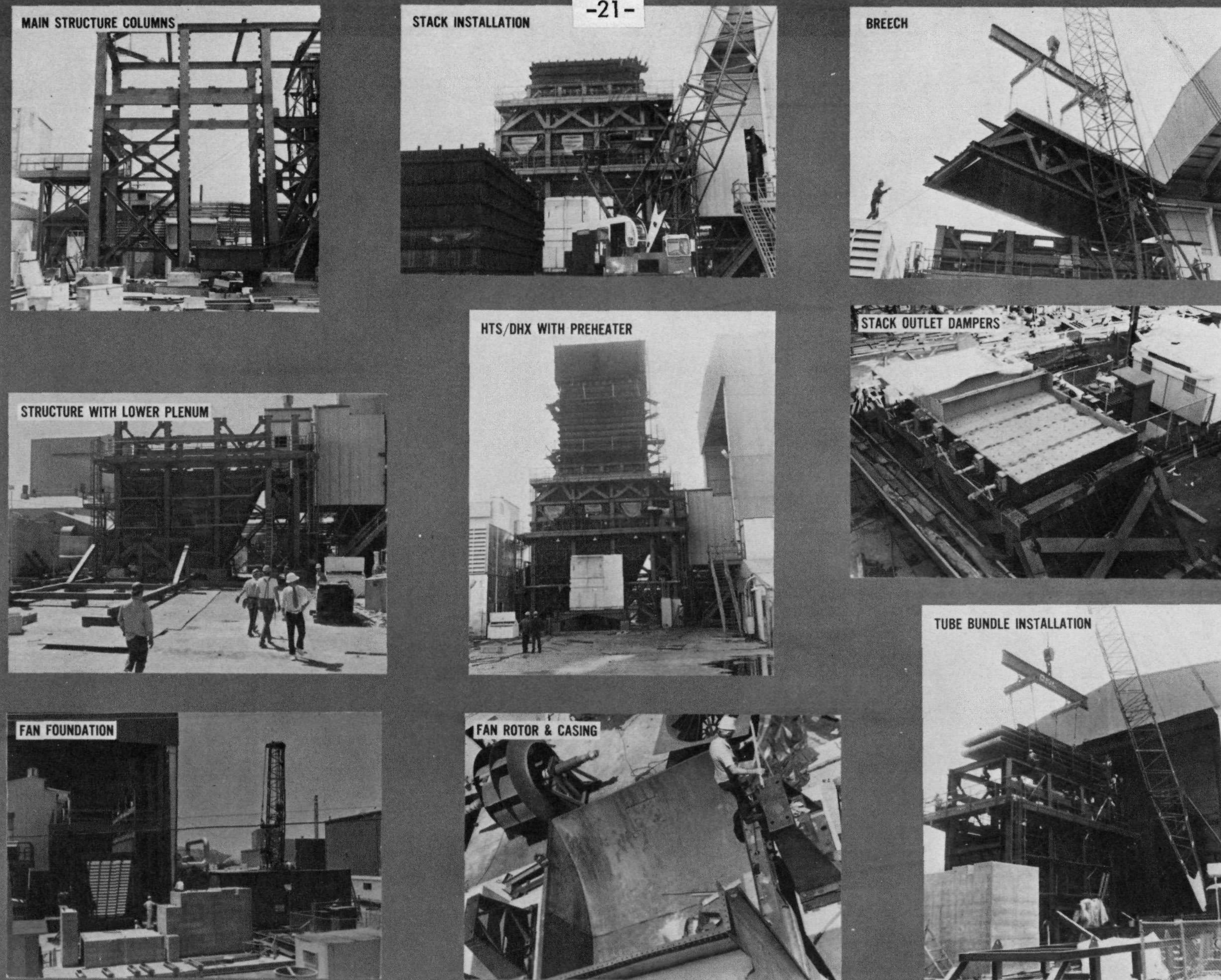

FIGURE 6 HTS/DHX Erection at LMEC

TABLE 3

ERECTION PROBLEMS

	Problem	Probable Cause	Fix At LMEC	Recommended Fix at FFTF
1.	Location of airflow preheater temperature monitors interferes with lower and upper preheater dampers.	Design Error. Came from switch to one-blade from two blade damper.	Relocated to Preheater side of duct at LMEC.	Revise drawing to relocate T/C penetrations.
2.	Stack damper linkage interferes with breech structure. Six-blade damper only opens 80°.	Design error.	Bent linkage arms out about 1" away from structure to remove interference.	Bend linkage arms out about 1" away from structure to remove interference.
3.	Preheater dampers do not close fully.	Improper installation of linkage.	Modified linkage to revise design at LMEC.	Issue clarification to Bechtel to modify support plate for linkage to ensure proper location.
4.	Stack damper actuator arm installed on wrong side. Located 180° out of position. Dampers open when should close.	Drawings not clear.	Reverse polarity, accept as is.	Reverse arm, install actuator in correct position.
5.	Preheater alarm circuit board wiring incorrect.	Fabrication error.	Rewired Board.	Rewire board prior to shipment.
6.	Preheater insulation section left out on floor.	Improper installation.	Subcontractor called back to complete job.	Does not apply.
7.	Water drips on dampers and drops on tube bundle during rain.	Inherent consequence of DHX design.	Operate preheater in low-fire mode to keep bundle warm and evaporate moisture.	Same.

TABLE 3 (cont.)

	Problem	Probable Cause	Fix At LMEC	Recommended Fix at FFTF
8.	Stack damper seals bent.	Unknown.	Problem did not appear to affect damper leakage. Accept as is.	Accept on a one-for-one basis.
9.	Light and site ports on breech leak water. Manways also.	Ports designed to be removable - requires gasket.	Same as No. 7 above.	Same as No. 7 above.
10.	Insulation of bypass duct not specified on drawing.	Design omission.	Insulated up to 90° bend per SW instruction.	SW to incorporate instruction on FFTF drawing.
11.	Seal welding on breech not complete.	Inaccessible during assembly.	Packed areas with insulation and cover with welded plate.	Different assembly procedure for breech used at FFTF, involves pre-welding panel seams to eliminate problem.
12.	Panel holes smaller than insulation for headers and I/O pipes.	Interface information on insulation design not fed back.	Insulation thickness reduced at panel holes.	Same.
13.	Preheater insulation panel seal welds leak.	Improper seal welding.	Reweld.	Follow weld procedures.
14.	Water pools on preheater panel and leaks through damper.	Design problem.	Holes drilled to provide leakage path away from damper.	Revise drawings to provide drain path.
15.	Flow sensing switch hook-up to pneumatic lines not identified for preheater airflow (FS-21881).	Drawing requirement not identified.	Hook-up completed based on SW rep. instructions.	Revise drawing to include information.
16.	Regutron coupling control O & M Manual discrepancy. (Fan speed indicator off when fan off so no indication of fan speed during fan coastdown.)	Engineering error	SW requested to modify O & M Manual. Discovered wrong manual supplied.	O & M Manual discovered to be wrong one. Correct manual supplied. Simple rewire to be done in field.

TABLE 3 (cont.)

	Problem	Probable Cause	Fix at LMEC	Recommended Fix at FFTF
17.	Masking tape left on headers, I/O pipes.	Improper installation.	Removed tape and cleaned surfaces.	Add to check list and receipt inspection requirement at FFTF.
18.	Preheater structure gussets.	Design modification.	Fixed in field.	To be fixed in shop.
19.	Smoke detector/ladder interference.	Interface design error.	Chop off ladder four feet.	Does not apply.
20.	Spacer plate required between motor and coupling to limit motor float.	Drawing omission.	Mycarda plate added at LMEC.	Installation instructions describe requirements.
21.	Flanges at ends of beam cover panels interfered with welds.	Oversize weld or overrun on length of casing plate.	Remove end of panel flanges.	Field fix.
22.	Some horizontal panel joints could not be welded per drawings.	Interface error.	Removed 6 inches of the vertical flanges, welded the joints, then replaced flanges.	Weld the panels on the ground and put in place assembled.
23.	Some floor panels could not be seal welded.	Design and fabrication error.	Welded from outside.	Same.
24.	Joints behind columns and tube bundle supports could not be seal welded.	Design error and not following SWC recommendations during erection.	Cut flange seal welded from side.	Modify erection sequence.
25.	TSA brackets were warped causing fit-up problems with panels.	Design error - not allowing large enough slots in panels to allow for warpage from welding.	Enlarged slots in field.	Enlarge slots in shop.
26.	Some upper bracket panels did not fit.	Design error - accumulation of fab. tolerances.	Hammered and/or jacked into place.	Same as at LMEC.

TABLE 3 (cont.)

	Problem	Probable Cause	Fix at LMEC	Recommended Fix at FFTF
27.	Turning fixture trunnion did not fit on shipping fixture.	Design error.	Bolt holes enlarged, replaced bolts because of damage.	Same as at LMEC.
28.	Fit-up of tube bundle support clips and brackets to lower TSA support beams.	Design error - weld distortion and tolerance accumulation.	Cut level on both ends of lower cross bars.	Eliminate weld distortion by proper fixturing.
29.	TSA support clips could be installed backwards.	Fabrication error - clips not marked per drawing.	Same as above.	Ensure that clips are per drawing which shows "X" for inside.
30.	Interference between lifting beams and header.	Drawing error.	Spacers added to lifting beams.	Add spacers to lifting beam.
31.	Casing erection interfered with tube bundle installation.	Erection procedure oversight.	TSA bracing removed on the ground.	Remove TSA bracing on the ground.
32.	Delays resulting from interface parts not being available.	Problem encountered with insertion of bushings into swing plates at SW.	Workaround scheme implemented with temporary header supports.	Ensure all interface parts are on hand before bundle installation and that all lower brackets and the lower section of TGA are installed and aligned prior to opening the shipping fixture.
33.	Drilling of dowel pin holes at the fixed header end was very time consuming.	Installation of inlet/outlet pipes prior to availability of pins.	Holes drilled after installation of inlet/outlet pipes. Access limited. Had to use two pins instead of three.	Holes have been partially drilled in the shop. Do not install loops until all work on the header support system is concluded.

TABLE 3(cont.)

	Problem	Probable Cause	Fix at LMEC	Recommended Fix at FFTF
34.	Difficulty in drilling holes in swing plates and pillow blocks in the field.	Design choice error.	Drill holes in the swing plate and pillow blocks in the shop. Plug weld the holes in the column and burn and ream matching holes in columns to suit.	Same as LMEC.
35.	The .070" to .080" clearance between hinge plates and pipe support plate could not be held.	Warped support beam caused by shop welding.	Champfered hinge plate and mounting plate.	Mating surface milled at shop. Tolerance may be relaxed on a one-for-one basis.
36.	Welding of panel joints, during breech and breech casing erection.	Erection procedure too difficult to accomplish.	Used alternate erection procedure.	Use alternate erection procedure (O & M Manual Ch. 9, Step 28 Option 2).
37.	Erector considered weld inadequate for lifting damper per instructions.	Design/experience conflict between designer and erector.	Additional weld was added.	Lift damper in three parts.
38.	Damper did not fit to breech structure.	Interface error.	Removed cap plate extension and replaced after breech and damper were installed in place.	Cap plate extensions shipped loose, but lifting of damper in three parts minimized the problem.
39.	Fan base should be shimmed, not grouted as shown on SW drawing.	Design error	Installed shims.	Change drawing, have shims available at FFTF.
40.	Insulation inside preheater may not take high temperatures.	Design error, design specified a fiberglas PVC mastic coat.	Insulation secured with metal lath.	O & M Manual revised.
41.	Final insulated closeout panels installed in header support area did not fit.	Design/fabrication error.	Field fit.	Field fit.

TABLE 3 (cont.)

	Problem	Probable Cause	Fix at LMEC	Recommended Fix at FFTF
42.	Low controller signal alarm wired in fail safe mode which causes meaningless alarm.	Design error.	Rewire.	Wire plant units correctly.
43.	Fan speed indicator out of calibration.	Wrong meter.	Correct shunt resistor.	Not applicable.
44.	Annunciator on module field panel would not accept both N.O. and N.C. type contacts.	Not known.	Rewired by service representative.	Field fix.
45.	Preheater T/C's type J instead of K.	Design oversight.	Accept as is.	Correct units supplied.

The preoperational tests were generally successful after some expected minor controller and mechanical adjustments. However, during a test which was to demonstrate that the isolation gates would not close prematurely against a fan pressure of 2.54cm (1 in) H_2O, the actuator rods were bent. The rods were straightened and reinstalled with no recommendation made for redesign of the plant units, because the incident occurred under test conditions which will not be duplicated during normal operation at FFTF.

In-service inspections were performed throughout the DHX test program. These inspections provided data to permit evaluation of changes which took place in the tube bundle and overall component geometry during performance testing. Inspections to determine the "before", "during", and "posttesting" conditions were conducted under LMEC test procedures. The initial inspections provided the basis for subsequent inspection results.

An extensive series of tests was performed to characterize air flow distribution and vibration in the DHX, particularly in the area of the tube bundle. The experimental program was augmented by an analytical effort performed by Argonne National Laboratory. No damaging vibrations were detected. Results from airflow distribution tests were inconclusive because of turbulence encountered within the DHX.

5.3 Steady-State Operation

Steady-State Operation consisted of preheat, sodium fill, isothermal sodium purification runs and ultimately full power operation at a series of flow rates and temperatures.

Initial DHX dry preheat, sodium fill, and drain tests were performed during March and April of 1975. Data taken during the dry preheat portion of the tests indicate that a fairly uniform temperature was maintained across the tube bundle during heat-up. The tubes nearest the plenum walls lagged only slightly behind the remainder of the tube bundle during the heat-up period. The electric trace heaters on the DHX headers were discovered to be inadequate when the header temperatures failed to attain the heat-up rate of the tube bundle during dry preheat. Although the electric heaters are not a part of the oil-fired preheat system that was being tested, the discovery of their inadequacy was an important result of the dry preheat tests. The difficulty was overcome by rewiring to provide 480 volts to the heaters instead of the original 277 volts. The electrical heater design was subsequently changed and additional heaters were placed on the headers during installation of the DHX units at FFTF.

High sodium impurity levels and cold edge rows during this initial test series resulted in the plugging of an outer row tube (number 66). The plugged tube condition was discovered during April after a series of cold trapping operations were performed to try to clean up the system. While increasing the system temperature in preparation for a hot drain, it was noted that tube number 66 failed to respond with the rest of the tube bundle, indicating that a plugged condition existed. Attempts to unplug the tube were unsuccessful and the condition prevailed through the remainder of the test program.

DHX sodium preheat tests were performed during June of 1975. Problems with the preheater control system were encountered and it became apparent that using the preheater inlet to outlet temperature difference (ΔT) depends upon the heat losses from the DHX, which may vary with ambient conditions. Setting the ΔT to a particular value may then result in preheater inlet temperature that vary with the ambient conditions. Based on the experience of the sodium preheat tests, the control system was modified to use the desired preheater inlet temperature for the primary controller.

Subsequent stability tests for the modified system were successfully performed during September 1975. Another important result of the stability test concerns the control system's inability to limit tube bundle heating or cooling rates. Based on the test experience, it was concluded that the preheater control system cannot be relied upon to increase or decrease tube bundle temperature within set rate limits without operator attendance. If totally unattended preheater operation is desired, additional modifications to the control system are required. Operator requirements for FFTF are being evaluated to assess the need for such modifications.

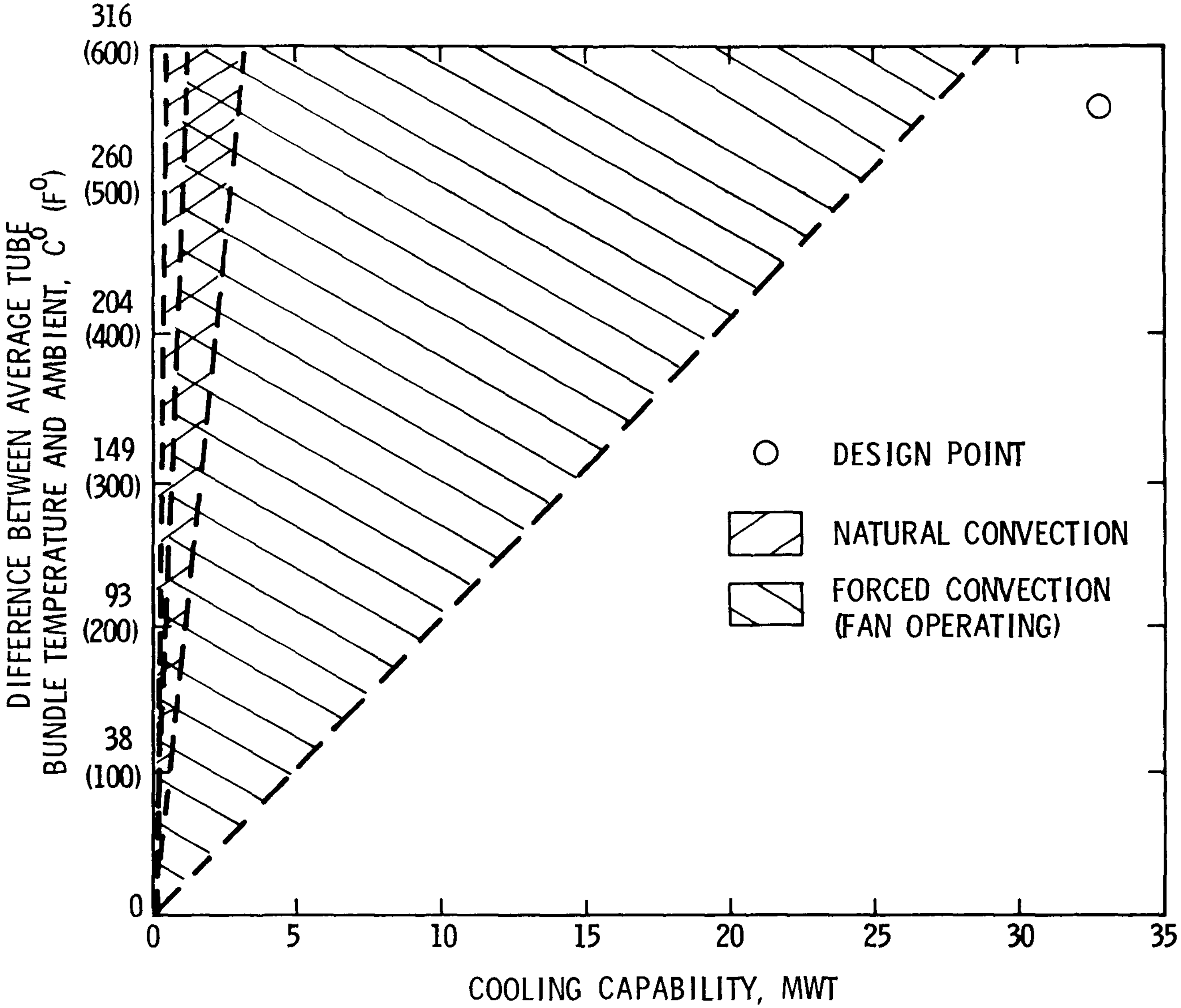

FIGURE 7 HTS/DHX Cooling Capability Comparison of Natural and Forced Convection Capabilities

HEDL 7710-126.5

Based on original design calculations, the preheater was sized to have a capacity of 2.1 x 10^6 Btu/hr (0.62 MW). Questions arose during the test program as to whether or not the preheater was actually performing at its rated capacity when difficulties were observed in bringing the tube bundle to the desired temperature.

We have concluded that the DHX preheater is capable of operating in excess of its 2.1 x 10^6 Btu/hr (0.62 MW) design rating. Difficulties encountered in attaining desired tube bundle temperatures must therefore be attributed to higher than expected heat losses.

Steady state performance tests were conducted to determine the full range of the DHX thermal performance capability. The minimum recorded power level with the fan operating was approximately 0.9 megawatts. This power level is well below the maximum natrual convection capability of the DHX. A comparison of natural and forced convection results is shown in Figure 7. The overlap is sufficient to allow a smooth transition between forced convection and natural convection operating modes.

The maximum cooling capability of the DHX was found to be $\sim$ 80% of the 33.3 megawatt design value. Results were adjusted to account for the plugged tube. The cause of this discrepancy is a lower than anticipated air-side heat transfer coefficient. There are apparently two factors which combined to lead to a non-conservative estimate of the heat transfer coefficient in the original design calculations.

The first factor involves the unusually wide spacing between rows of finned tubes in the DHX tube bundle. Since heat transfer data for such wide spacing was not available at the time the design calculations were performed, a factor to account for heat transfer degradation due to this spacing was estimated by extrapolation of data for bare tubes. The factor thus selected to be applied to the air-side heat transfer coefficient was 0.92. More recent correlations for finned tube heat transfer coefficients indicate a factor of 0.7 should have been applied.

The second factor affecting the heat transfer coefficient involves a correction to account for boundary layer velocity profile distortion caused by variations in physical properties with temperature between the surface of the fins and the air stream. A factor of 0.8 would be used today for this term, which was not considered in the original design. Information concerning this effect was not available in the literature at the time of the original design review.

5.4 Isothermal Heat Losses

During shutdowns and standby conditions, the DHX is shut down and the air-side isolated. Heat losses from the DHX are made up by pump work and core decay heat assisted as necessary by the electrical trace heat system on the heat transport loops. Results from LMEC are shown in Figure 8. Heat losses exceeded specification losses by ~ 50% probably because of higher than anticipated infiltration losses.

5.5 Transient Tests

A series of HTS/DHX transient tests were conducted at LMEC during Ausut and September of 1975. The DHX was subjected to reactor scram and other upset transient conditions to confirm that the limits of the design thermal transients would not be exceeded. Tests were limited to those upset conditions that would not significantly reduce the service life of the module being tested. Sufficient information was to be generated to permit accurate analysis of all predictable transients.

The transient tests consisted of four basic simulated conditions: 1) reactor SCRAM; 2) loss of airflow; 3) isolation valve closure; and 4) 133% over power condition. The tests that were actually performed deviated somewhat from those called for in the test request. These deviations were due primarily to test facility limitations and to the less than expected thermal capability of the DHX itself. Results from the transient tests are being used within the Systems Dynamics and Thermal Analysis group of FFTF Engineering to improve the accuracy of analytical modeling of the DHX. Results from those efforts will be presented soon in a separate report. In general, transients were found to be less severe than predicted. The simulation of the loss of a module in a unit of four with the remaining three picking up the load showed that the DHX control system could not lower the outlet temperature as quickly as predicted. The impact on plant operation is still being evaluated.

5.6 Modification Testing

Realization of the lower than expected heat transfer during the early stages of steady-state testing resulted in the development of an extensive program of additional testing and analytical work to improve DHX performance. A number of proposed methods for improving the overall heat transfer capability were tested at LMEC. These methods included the following: 1) installation of bypass baffles to reduce air flow between the outer tube rows and DHX wall; 2) installation of a straightener; 3) installation of a false floor in the lower plenum to reduce reciculation and improve air flow characteristics; and 4) attachments of stainless steel tape strips to the existing tubes to delay boundary layer separation. None of the above methods, which were tested in various combinations, provided enough improvements to attain the 33.3 megawatt design goal.

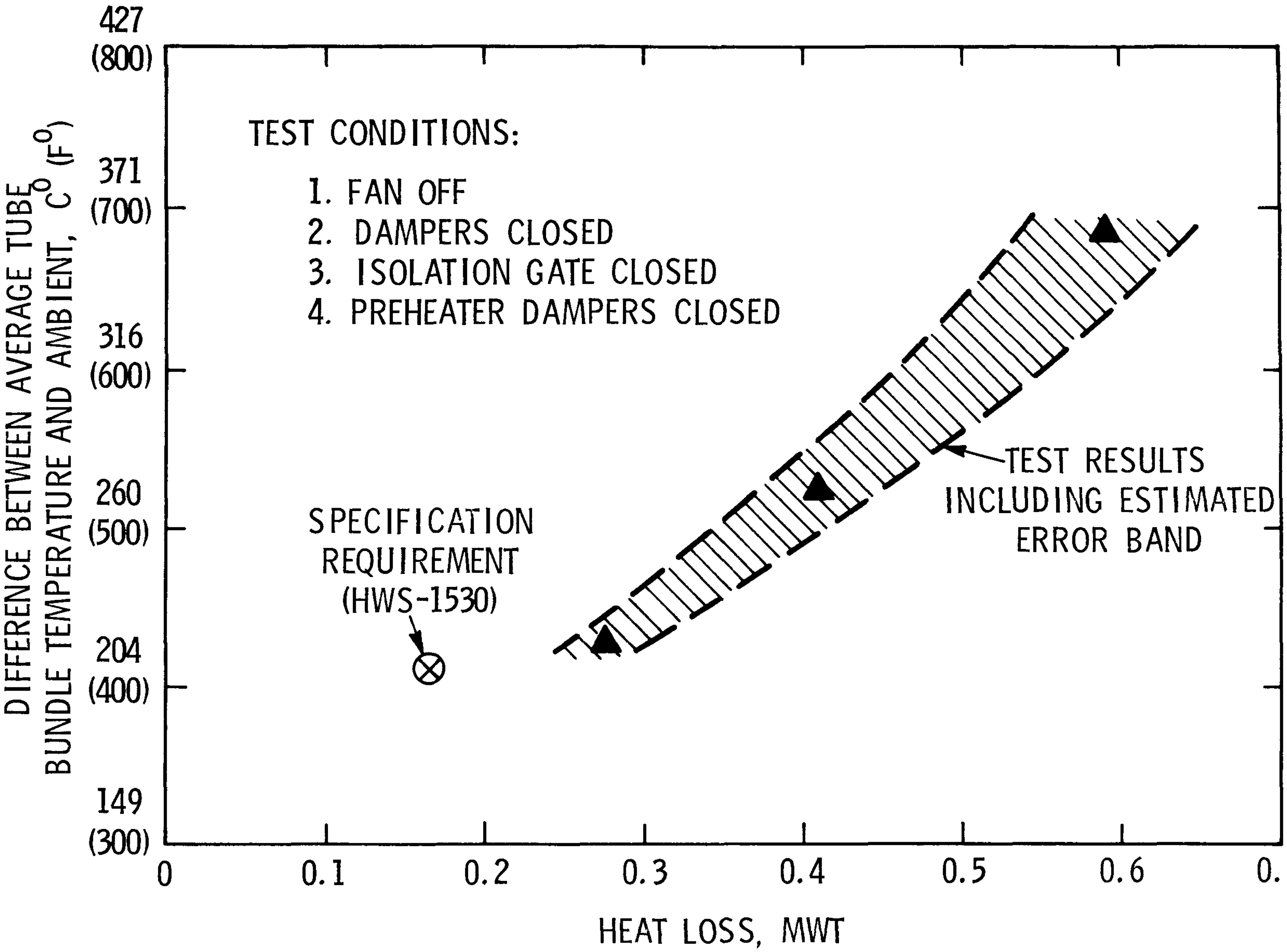

FIGURE 8 Isothermal Heat Loss from the HTS/DHX

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Evaluation of the results of the LMEC steady state tests indicated that the installation of bypass baffles provided a significant improvement to DHX performance. Although this improvement alone was not enough to increase performance to the 33.3 megawatt level, it was considered significant enough to warrant its incorporation into the plant DHX units. Table 4 includes predicted performance for the modified plant units with bypass baffles as well as the unmodified DHX tested at LMEC.

TABLE 4. FFTF Performance Predictions, 412°C (773°F) Inlet 268°C (515°F) Outlet, 32°C (90°F) Ambient and 456°C (853°F) Inlet, 313°C (595°F) Outlet

	412°C (773°F Inlet	456°C (853"F) Inlet
Unmodified	27.4 MW	31.5 MW
With Baffles	28.9 MW	33.0 MW

5.7 Post Testing Inspection and Disassembly

A comparison of post-test results with pre-test data revealed the following:

1. A comparison of the DHX dimensional measurements taken before and after testing showed differences which were greater than could be accounted for by the different ambient temperatures at which they were taken. In addition, differences were sometimes positive and sometimes negative. The randomness of these dimensional differences is somewhat indicative of the difficulty encountered in taking measurements of this type. It is also conceivable that the dimensional changes were a result of the thermal excursions to which the DHX was subjected during testing.
2. The inspections made revealed no visible wear or galling at the contacting surfaces of the tubes and tube support assemblies nor at the isolation gate foundation bearing plate. There did appear to be some erosion in the portion of the TSA/Bottom Bracket Lubrite Plate which was exposed to the airstream during testing. However, this occurrence does not affect the lubricating qualities of that portion of the plate which actually contacts the support bracket and, therefore, is of no consequence.
3. A comparison of structural measurements taken before and after testing indicated no significant differences in the dimensions taken.
4. After a portion of the thermal performance testing had been completed, parts of the isolation gate seals were found to have

suffered fatigue failure. The remaining portions of the seals were removed and subsequent tests were conducted without the seals in place. Isolation gate seals on all plant DHX units will be replaced with seals of a new design to prevent recurrence of this problem at FFTF.

Disassembly was conducted in a routine fashion. Since the prototype was the 12th unit for construction at FFTF, the module was disassembled in subassemblies to expedite site erection.

Prior to shipment the tube bundle was drained and cleaned by LMEC using a heated alcohol method. The cleaning was complicated by residual sodium collected in the U-bends of the tube bundle. Local heating and repeated flushing with alcohol were used as well as tipping the tube bundle. The plugged tube was removed and a new tube installed by LMEC personnel prior to cleaning.

6.0 HEDL FEATURE MODEL TESTING

Shortly after the indication that the Dump Heat Exchanger was operating at reduced power level, HEDL undertook a series of scale model tests to characterize possible reasons for the malperformance and to evaluate possible modifications. Testing was conducted in a step wise fashion, with initial water testing using dye and bubble injection to evaluate flow distribution. After water screening tests were complete, a four-row model employing steam inside the tubes was used to evaluate heat transfer coefficients. Because of limitations on steam temperature this testing was followed by testing with an electrically heated test section capable of matching operating temperature. After an unsuccessful test of one concept from HEDL feature model testing at LMEC, an improved feature model was constructed and used to test various extended fin geometries. A brief description of the testing performed follows.

6.1 Water Flow Testing

The hydraulic model was an enclosed structure made of 1.91 cm (3/4 - inch) plexi-glass supported by 25.4 cm (10-inch) channel iron. It was approximately 2.74 m (nine feet) long with a cross-section surface of 15.24 cm (six inches) wide by .432 m (17 inches) tall. Water was supplied to the test section by a 10.16 cm (four-inch) piping system which was connected to a pump capable of 11.4 m^3/min (3000 GPM) at 14.1 kg/cm^2 (200 psi). Flow control in the range of 1.89 m^3/min (100 to 500 GPM. 38. was accomplished by means of a penumatically actuated, diaphram-operated valve and was based on duplicating Reynolds number at full DHX airflow.

The model was designed to accommodate up to four tube rows in the test section. Each tube row consisted of seven tubes and two half-tubes. They were aligned such that the distance from centerline to centerline duplicated plant dimensions. The half-tubes were flush with the top and bottom plates of the plexi-glass channel. Figure 9 shows the hydraulic model with one tube row installed.

When a test was being conducted that required more than one tube row, the distance between tube rows was .36m (14 inches). The initial tube row was situated approximately .75m (2-1/2 feet) downstream from a flow straightener that was located at the test section inlet.

Modifications were made to the channel such that screens, dummy tubes, baffles, or other modifications could be positioned at a number of locations relative to the tube rows.

A five-prong dye rake was inserted in the water model about four inches upstream of the first tube row. A camera was set up in a manner such that photographs of the flow patterns could be taken after the introduction of dye into the water passing across the tubes. Evidence of the flow profile and determination of the pressure drops of dummy tube rows, baffling downstream of the tube rows, supplement fin geometries, screens upstream of the tubes and steel strips applied directly to the tubes. These improvements were designed to increase the airside heat transfer coefficient without a significant penalty on pressure drop. The water flow tests helped to select the geometries with the most promise for further testing.

6.2 Steam Heated Testing

As shown in the schematic in Figure 10, room air entered a 142 m^3/min (5000 CFM) centrifugal blower through a manually adjusted butterfly damper. The blower was driven by a 15-hp motor which operates at 1750 rpm. Air emerged from the blower into a .4m x .4m (16" x 16") steel duct, through two 90-degree bends, and then expanded into a .3m x .61m (12" x 24") duct. Next, the air was straightened through an air monitor and the total and static air pressures were measured via a pitot tube network. The pressures were subtracted to give velocity head and fed back to a pressure meter indicating cubic feet per minute (CFM). Located behind the air monitor was the upstream inlet air temperature and pitot rake used in determining the inlet air velocity profile. A similar pair of Resistance Temperature Detectors (RTDs) and pitot rakes followed the test section. The matched pair of RTDs was used to measure the air temperature across the test section. The air was then passed through a flow straightener and duct outside of the building.

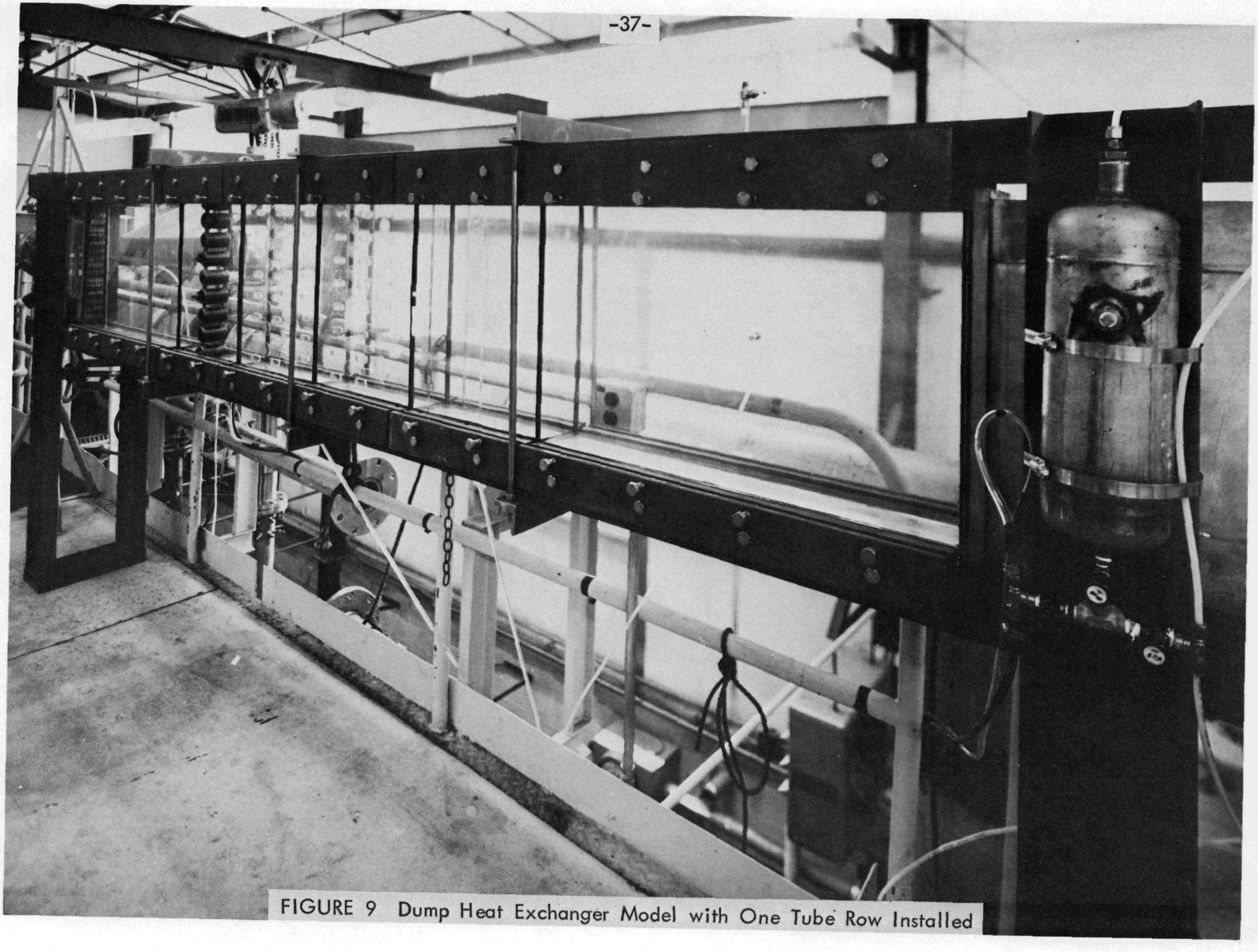

FIGURE 9 Dump Heat Exchanger Model with One Tube Row Installed

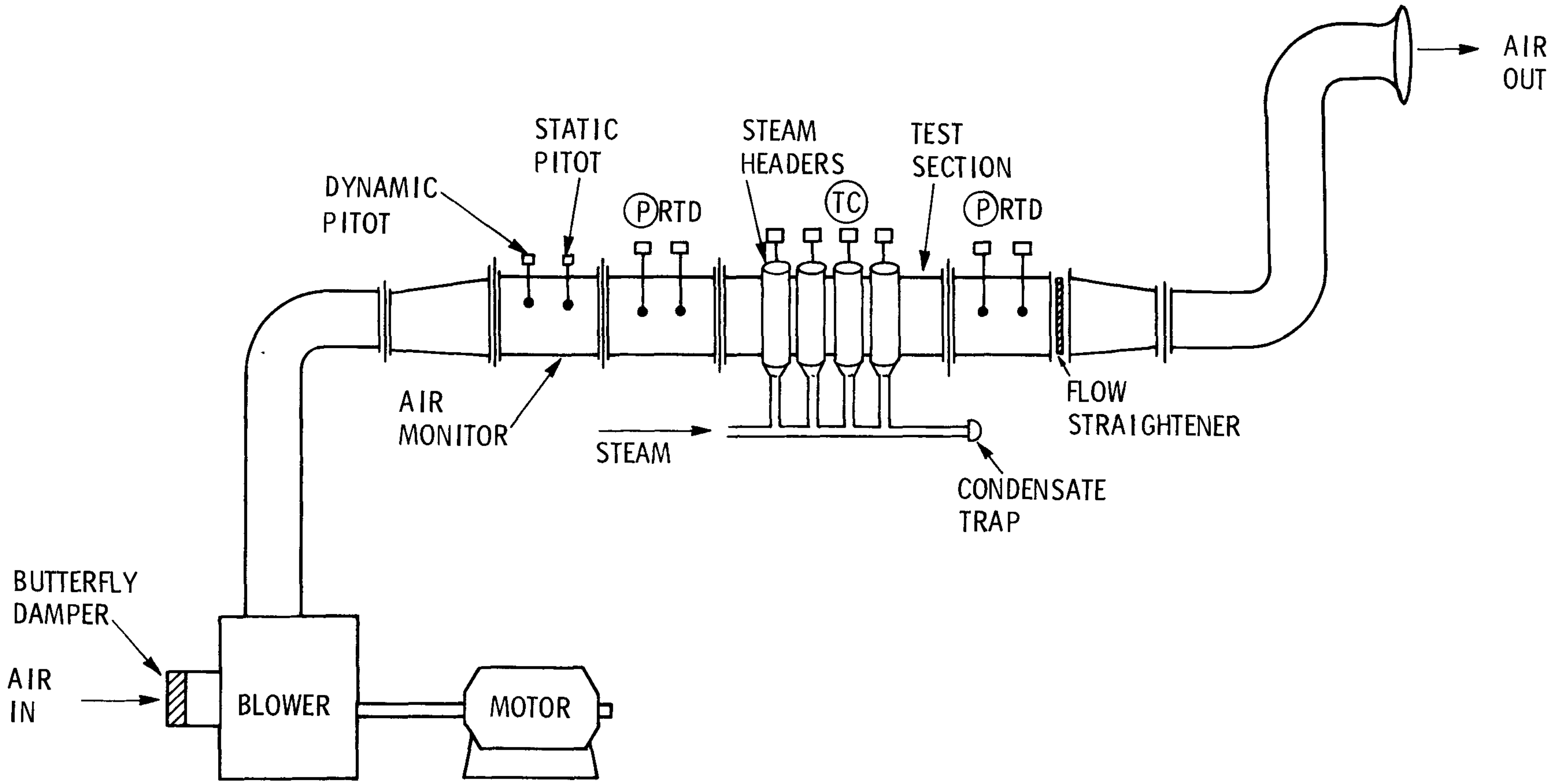

FIGURE 10 Heat Transfer Model - Schematic

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The test section consisted of up to four tube rows with six prototype DHX tubes[3.76 cm (1.48 inches) root diameter] per row. The tubes were supported and contained in steel box-like ducts with inner dimensions of .324 m (12-3/4") high, .61m (24" wide), and .356m (14") long. The four tube row boxes were removable from the test section and could be replaced by empty boxes. Each tube row box had inlet and outlet steam manifolds connected to the tubes and was instrumented with thermocouples. Also, each box had the capability of supporting modifications to the tube row.

Figure 11 shows the test sections being installed. Figure 12 is a closeup of the tube arrangement within a single test section. The completed test model is shown in Figure 13.

Saturated steam reaches the model at about 121°C (250°F), as indicated by the steam manifold thermocouples and at 1.06 kg/cm^2 (15 psig), as indicated by a pressure gage on the inlet steam line. Table 5 summarizes results obtained with the steam. Because of the low steam temperature and the assumptions required to account for the condensing steam film coefficient on the inside of the pipe, the steam results were evaluated in relative terms versus a base line of unmodified tube assemblies. The tape strips and the supplemental fin configurations were selected for further testing at prototypic temperatures in an electrically heated model.

6.3 Electrically Heated Testing

The model for this testing was very similar to that used for steam heating. The steam was replaced by cartridge heaters cast into the tubes with molten aluminum. Additional instrumentation was also added.

The addition of the electrically heated tube bundle permitted modelling of the bundle pressure drop and heat transfer characteristics up to prototopic DHX temperatures of 427°C (800°F). These higher temperatures and higher heat fluxes introduced several areas of potential problems or uncertainties. The higher tube temperatures require the use of shielded thermocouples/RTDs in order to reduce temperature errors arising from radiation effects. The large heat fluxes in the tubes make precise measurement of the inside tube wall temperatures very difficult. These high heat fluxes also develop radial temperature distributions in the tubes, impeding the definition of average inner wall temperatures. Gaps also developed between the aluminum and the tube wall during repeated cycling making precise determination of temperature distributions very difficult.

Initial tests with the strips proved successful and a program of strip optimization was performed to establish strip thickness, orientation and edge shape. A marked effect was found on pressure drop at increased temperature. An optimum strip configuration was selected for testing at LMEC and a 25% increase in heat transfer coefficient was predicted. Actual improvements were found to be on

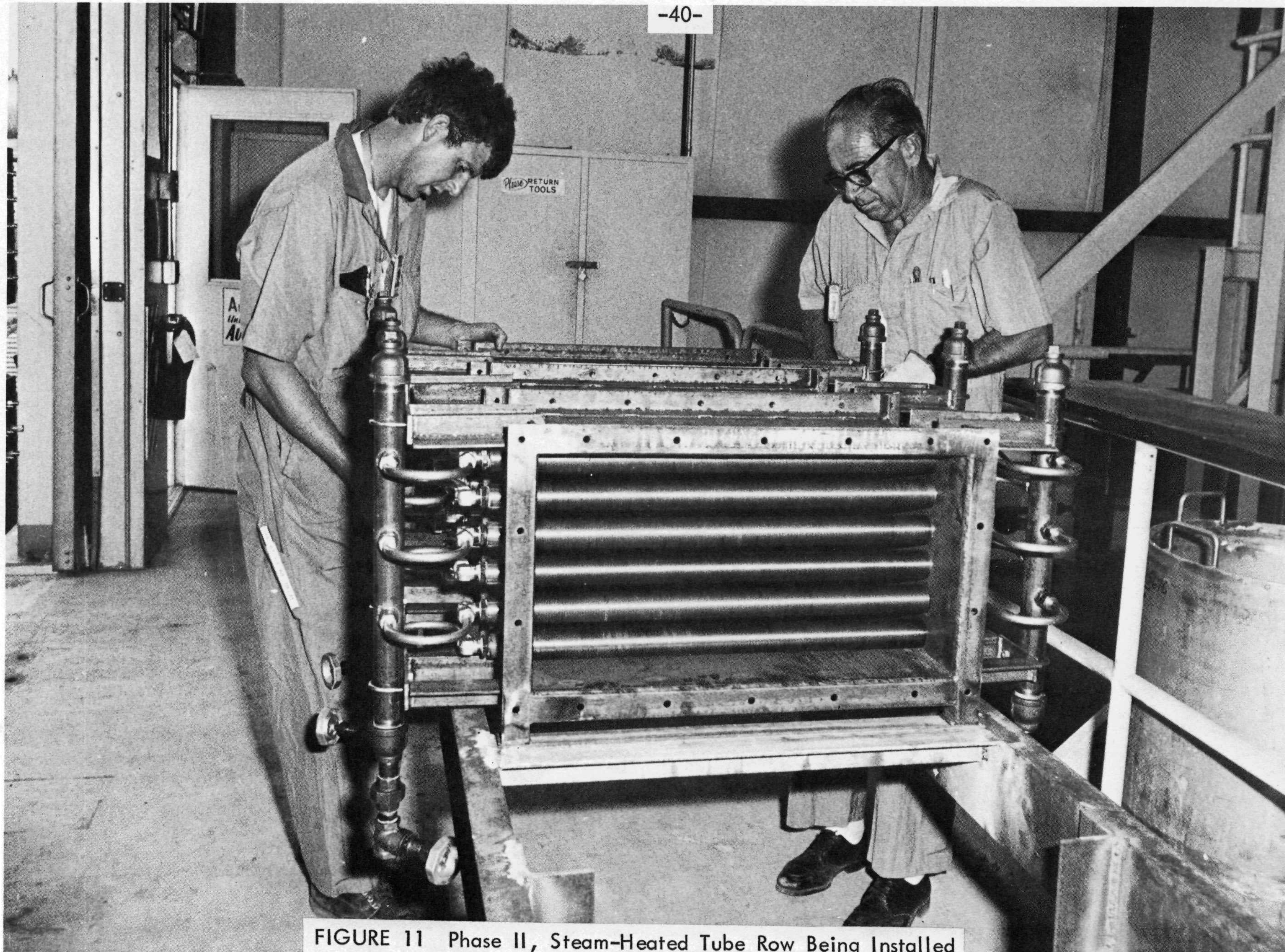

FIGURE 11 Phase II, Steam-Heated Tube Row Being Installed

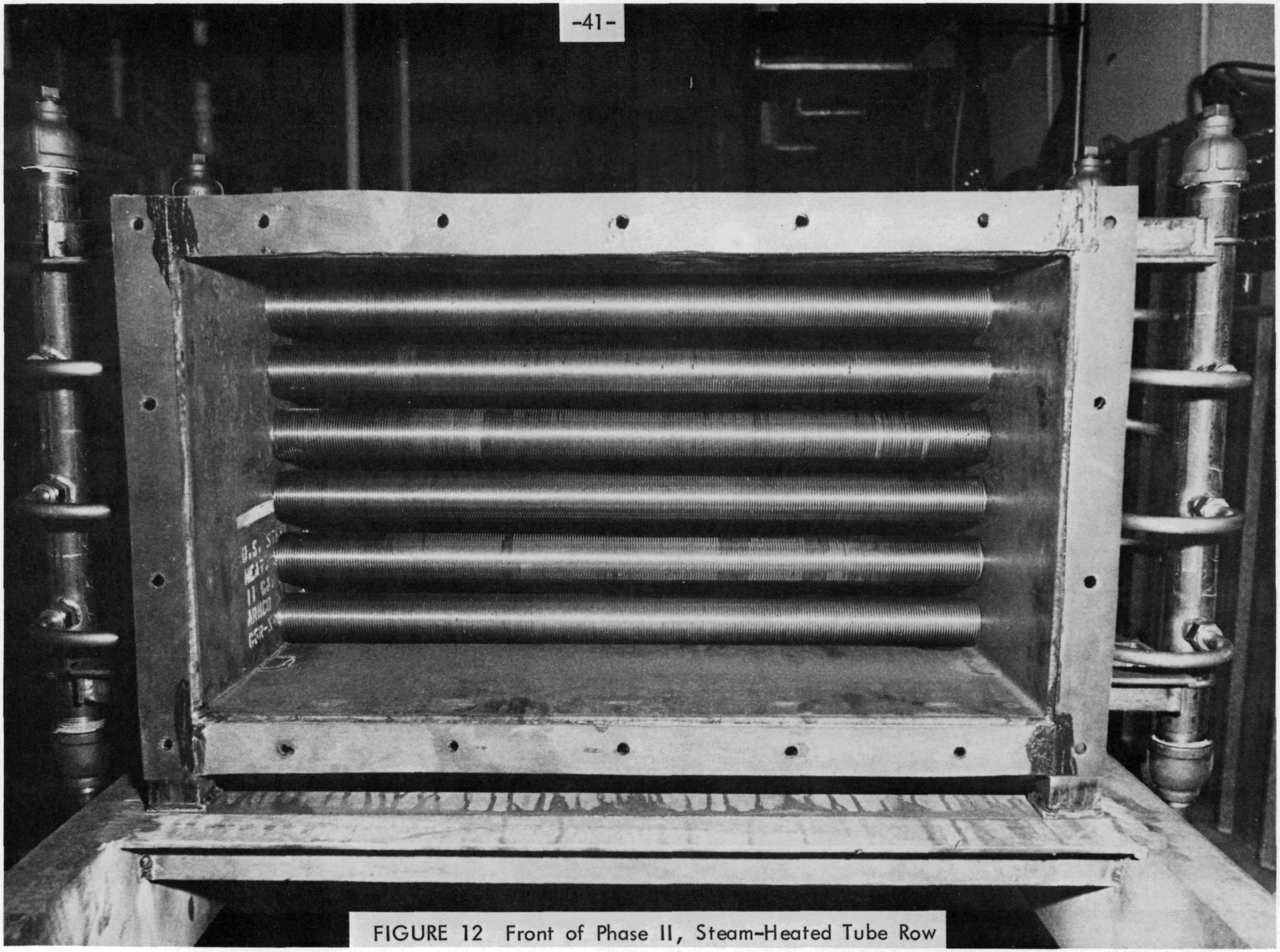

FIGURE 12 Front of Phase II, Steam-Heated Tube Row

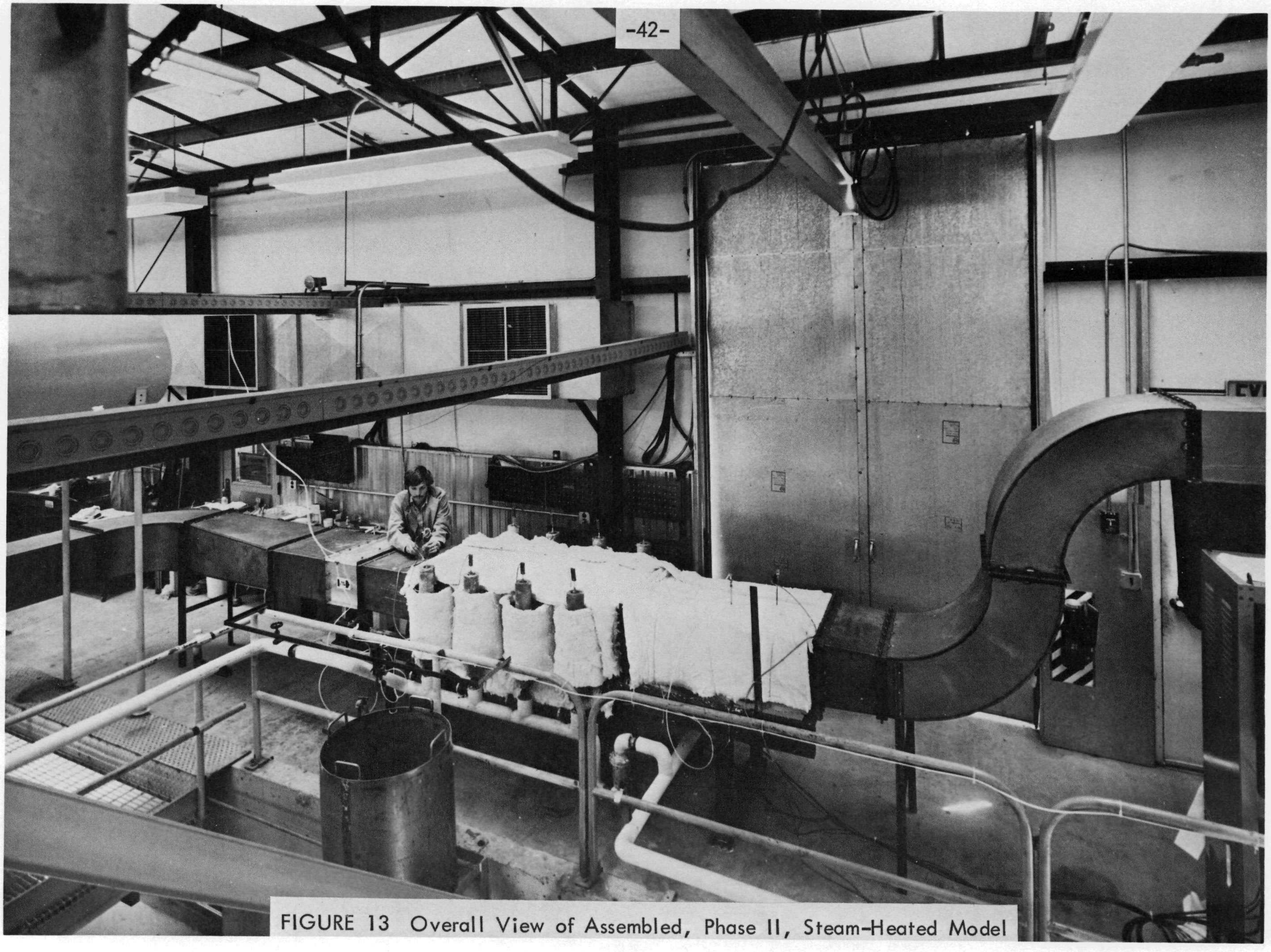

FIGURE 13 Overall View of Assembled, Phase II, Steam-Heated Model

TABLE 5. STEAM MODEL RESULTS

COMPARED TO BASELINE GEOMETRIES

	Δh	ΔP
Dummy Tubes	+ 25%	
Baffles	+ 18%	+ 8%
Screens	+ 0%	+15-20%
Supplemental Fins	+ 83%	+70%
Strips on Tubes	+ 23%	+ 0%

the order of 10%. Problems were also noted with failures of the strips. Fatigue was determined to be the cause. Cooldown with high air flow caused the strips to buckle and vibrate which eventually resulted in failure. The cause of the poor heat transfer performance is still unexplained.

Testing conducted on extended fin geometries was also successful, but configurations were not tested at LMEC because of cost and schedule constraints. Instead, an improved feature module was constructed at HEDL to more adequately characterize extended fin performance. With the less than expected tape strip performance at LMEC, this activity became a high priority effort.

6.4 Improved Feature Model Test Results

The improved feature model was designed to remove or reduce many of the uncertainties in prior testing. Electrically heated tubes were again selected but the aluminum fill was rejected in favor of sintered copper with pre-tinned tube wall to provide a smooth bond between the heater and the wall. The tubes were nondestructively examined using an infrared technique to ensure the uniformity of the temperature distribution. Finally, the tubes were heated uniformly in a furnace to confirm the accuracy of the thermocouples installed in the tubes.

Evaluation of prior results had indicated that the desired heat removal capability could be achieved by replacing the bottom row of tubes with tubes utilizing extended fins attached by brazing. The fourth row of the model was then equipped with higher powered heaters and was made up of tubes from various brazing suppliers.

The power supplied to the heaters was checked for each row by standards laboratory instrumentation. The flow meter measuring air flow was sent to an independent testing laboratory to confirm the manufacturer's calibration. Air temperatures located between rows had been subject to uncertainty because of radiation from the heated tube rows. Aspirated thermocouples were used to measure these temperatures.

Prior pressure drop measurements had been made with pitot tube rakes. For the improved model, dynamic pressure fields were mapped in detail in order to establish representative locations for the static pressure taps.

Testing was conducted at representative DHX operating points with each row set at pre-calculated temperatures. An iteration procedure was used to readjust row temperatures based on measured performance. Because of the many computations involved in obtaining averaged row temperatures and row powers, an on-line WANG programmable calculator was interfaced to the data acquisition system. Data from the data acquisition system were read out on magnetic tape and subsequently analyzed on a PDP 1140 computing system.

The configuration found to have the best thermal performance employed carbon steel fins, 5.08 cm (2") wide, 7.62 cm (3") high, 1.1mm (.042") thick with a spacing of 4.7/cm (12 per inch). While this configuration achieved the desired thermal rating based on an analysis of test results, it did not achieve the desired design margin. The greatest uncertainty is believed to be in the airside performance of the DHX fan system.

Work on the supplement fins has been suspended following testing; the present project decision is to increase reactor temperatures to be more prototypical of future breeder reactors. This temperature increase, coupled with present construction to add tube bundle baffling around the sides will enable design power levels to be achieved except on very hot days.

7.0 CONSTRUCTION AND MODIFICATION

During the time that the prototype DHX module was being tested at the Liquid Metal Engineering Center, the installation contractor began construction of the DHX units at the FFTF Site. The construction effort was sequenced to meet the needs of the overall Bechtel plant schedule since Bechtel was responsible for completing the piping system to the DHX's and required unrestricted access to the DHX area to accomplish this work. The DHX's were constructed one loop at a time with four units being erected simultaneously within a given pit. Figure 14 shows the three pits in various stages of construction. The structure and insulated panels for each unit were raised to the level where the tube bundles were installed. (Figures 15, 16). At that point, the cruciform structure (structural steel assembly which ties all four units together and supports the secondary sodium piping system) was assembled.

The tube bundles were then installed sequentially as they were received from Struthers Wells. Once the tube bundle was installed in a given unit the remainder of the structure could be installed with relative ease. This was done in three subassemblies: 1) the breech assembly which contains the outlet dampers; 2) the exhaust stack; and 3) the rain hood. Each unit was completed in this fashion. The fan and shroud, motor coupling, preheater and miscellaneous hardware including ladders, railings, ducting, conduit, lighting, weather enclosure, and insulation were then added. As one pit neared completion, the erection of structural steel was begun on the next pit.

The delayed arrival of the final DHX tube bundle and structural steel from LMEC caused a significant schedule problem for the installer at the FFTF site. A work around was developed utilizing an additional set of structural members which allowed the cruciform to be installed and the remaining three modules to be completed. The last module was completed soon after arrival of the last tube bundle.

FIGURE 14 Aerial View of FFTF Site Showing DHXs Under Construction

FIGURE 15 DHX Tube Bundle Installation

FIGURE 16 Lower Structure

A baffling concept based on LMEC testing (Figure 17) is being installed on the units in place. The baffles are designed to be in the optimum proximity to the tubes at the maximum power condition (baffles are fixed and tube moves with respect to the baffle). The addition of the baffles forces the bypass airflow (estimated at 10% of the total flow) into the center of the tube bundle and thus adds approximately 4 to 5% (based on LMEC testing) to the overall performance of the unit.

Access to the tube bundle is being gained by cutting holes in the insulating panels at the level of the tube bundle. The baffles are then attached to the insulating panels after being accurately positioned at the design optimum location. The access holes are then welded closed.

During the entire construction process and DHX modifications, special procedures were and continue to be used to provide protection to the exposed tube bundles.

8.0 PLANT OPERATION AND DECAY HEAT REMOVAL

The original specification for the DHX contained a wide range of operating conditions, allowing fuels experimenters to select desired core operating conditions. The minimum core inlet temperature originally aticipated during the life of the plant was 316°C (600°F) which corresponds to a DHX outlet temperature of 268°C (515°F). At this operating condition, the DHX (as shown through LMEC testing) does not meet the design load of 33.3 MW. The testing of tube bundle modifications to increase heat transfer was aimed at upgrading the unit to the original requirements.

During this development and testing period, a decision was made to increase the core operating temperatures for initial operation to 360°C (680°F) core inlet eliminating the need to make any heat transfer modifications to the tube bundle. At this operating conditions, it is anticipated that the DHX units will operate at the full design capacity. The decision to operate at the higher core temperatures was based on several factors; the most important of which was the need of fuel experimenters to match as closely as possible the core conditions to be expected in follow-on LMFBR plants, specifically the Clinch River Plant. This allows the experimentors to characterize performance and confirm the viability of the reference oxide fuel system prior to operation of the Clinch River plant. The desire to startup FFTF as quickly as possible and the large costs associated with a major tube bundle modification were other major considerations in the decision not to implement any tube bundle heat transfer modifications prior to initial plant operation.

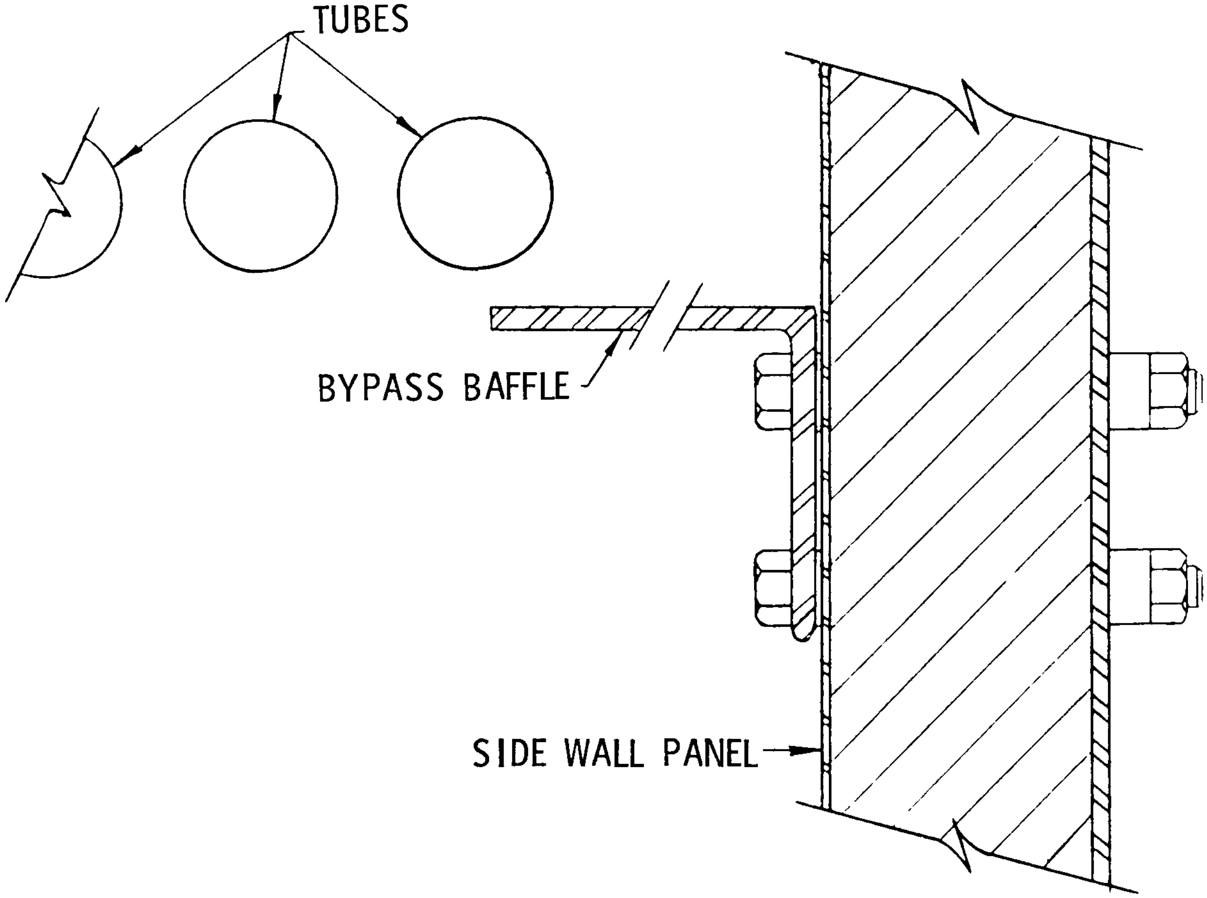

FIGURE 17 HTS/DHX Bypass Baffle Arrangement

HEDL 7710-126.9

Just as important as the ability to dissipate the full reactor heat load is the ability to remove reactor decay heat while maintaining and controlling system temperatures. The anticipated mode of operating for decay heat removal is in the post scram condition. The main pump motors will be off and the pumps will be operating on the auxiliary or "pony" motors. The DHX fans will also be off. Cooling is maintained through natural convection air circulation while temperature is controlled by modulation of the outlet dampers. As the decay heat falls off, the DHXs are closed up and the preheaters turned on, if necessary, to maintain system temperatures.

The most critical condition for decay heat removal is the case of total loss of electrical power for an extended period of time. Under this circumstance, the pumps are completely shut down and the heat transfer system operates under natural circulation. The DHX operates as previously discussed, in a natural convection mode. The problem arises in the long term analysis as the reactor decay heat dies off. Since no power is available, the only source of heat input to the system is the reactor itself (other sources normally available with electrical power include pump heat, trace heat, and DHX preheaters). In this situation, the DHX must minimize heat losses to the environment to maintain system temperatures. The extremely large surface area of uninsulated tubing becomes an undesirable design feature. For this reason, the DHX is provided with an insulated plenum and inlet/outlet dampers to reduce the amount of heat loss. As the decay heat continues to drop off, the DHX modules are manually valved off and allowed to freeze sequentially in an attempt to maintain system temperatures and to prevent the main lines from freezing, which would eliminate the decay heat removal path. When only one DHX remains operative, it is sealed as tightly as possible by closing inlet doors and outlet dampers, and will operate in that condition until freezing occurs. At that time, reactor decay heat is dissipated totally by primary system heat losses to the surrounding containment structures.

9.0 CONCLUSIONS

The dump heat exchanger program in its entirety involved many organizations. One of the national objectives of the United States Liquid Metal Fast Breeder Program is to establish an industrial base technology capable of fabricating equipment to the demanding standards of the reactor program. The DHX is an example of the progress that can be made in a co-operative atmosphere with the involvement and commitment of both industrial concerns and government agencies.

The early design activities and problems identified and corrected in the test program point up the need for a well conceived feature test program in any area where existing data do not encompass the design parameters. While the application for air cooled units in future LMFBR's is probably limited to emergency decay heat removal systems and test loops, much of the technology, design methods and criteria developed during the design process will be applicable to other components.

The prototype testing program was found to be extremely valuable. A number of minor problems were identified and resolved. The potentially large problem of reduced heat removal capability was identified early enough in the project to allow corrective actions to be evaluated and implemented prior to operation.

The feature testing program undertaken to evaluate tube bundle modifications also achieved some important advancements. An ASME Code case has been approved which will permit the use of commercially available finned tubing with extended surface. The test data obtained by feature testing also have extended the data base for low finned tube heat exchangers with wide row spacing. This data has been used to predict performance of the smaller 2.3 MWt Closed Loop Dump Heat Exchangers installed at FFTF.

The FFTF is scheduled for sodium fill in August of 1978. After a period of isothermal testing, core loading will be initiated with criticality scheduled for August 1979. Because of the extensive design and testing to which the dump heat exchangers have been subjected, we expect them to operate successfully over the full range of plant conditions.