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TECHNOLOGY FOR SPACE STATION
EVOLUTION WORKSHOP

ADVANCED INTERFACE HEAT EXCHANGERS FOR THE SPACE STATION MAIN THERMAL BUS

NASA CONTRACTS: NAS 9–17810
NAS 9–17989
NAS 9–18167

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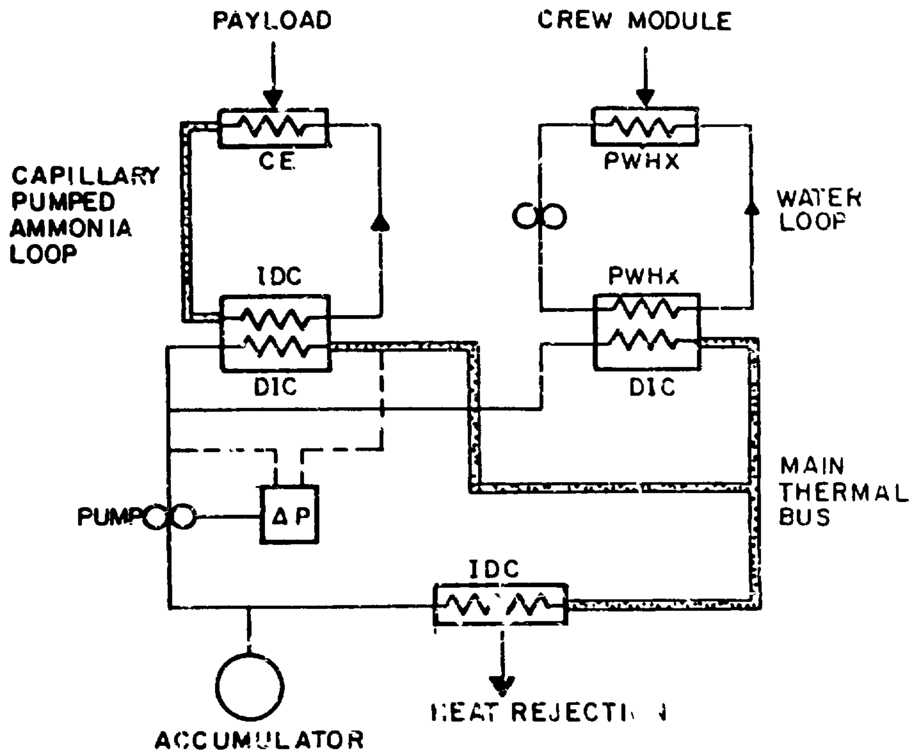
Future evolution and growth of the Space Station will place increasing demands on the thermal management system by the addition of new payloads and from increased activity in the habitat modules. To meet this need, Creare is developing, under the sponsorship of NASA JSC, advanced evaporators, condensers, and single-phase heat exchangers for operation in micro gravity. The objective is to achieve a several-fold increase in the heat flux capability of these components, while operating at the same temperature difference as specified for the present interface heat exchangers. Two prototype interface heat exchangers are presently being developed: one to interface the main thermal bus to a payload two-phase ammonia bus, and the other, to interface with the crew module single-phase water loop. This presentation will review the results achieved to date in the development of these heat exchangers.

PROGRAM OVERVIEW



DIC	DROPLET EVAPORATOR
CE	CAPILLARY EVAPORATOR
IDC	INTERNALLY DRAINED CONDENSER
PWHX	POROUS WALL HEAT EXCHANGER
	VAPOR LINE
	LIQUID LINE

The objectives of this program are to develop advanced heat transfer techniques to allow a several–fold reduction in the size of interface heat exchangers for future space thermal management systems.

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PROGRAM OBJECTIVES

DEVELOP NEW HEAT TRANSFER TECHNIQUES

1. DROPLET IMPINGEMENT COOLING
2. INTERNALLY DRAINED CONDENSER
3. POROUS WALL HEAT EXCHANGER

PROGRAM GOAL

SEVERAL FOLD REDUCTION IN SIZE AND WEIGHT OF SPACE THERMAL MANAGEMENT COMPONENTS

The evaporator uses droplet impingement cooling (DIC) to achieve heat transfer coefficients in excess of 10 W/cm^2–°C. A piezoelectric transducer is used to eject an array of small diameter (100 micron) drops at high velocity (10 m/s). These drops impinge on the heat transfer surface and spread out forming a liquid film only a few microns thick. Because the film is so thin the temperature gradients at the wall are very large, and bubble nucleation is inhibited, even at large values of wall superheat. Heat is conducted through this thin film, and liquid evaporates at the surface of the film removing heat. Since there is no nucleate boiling, the film remains attached to the wall and there is no liquid carryover in the vapor stream. A simple temperature feedback loop regulates the droplet generation rate to maintain the wall at the desired temperature.

Droplet impingement cooling differs from spray cooling in three important respects. First, in DIC drop pattern is highly uniform as compared with the random distribution of drops in a spray. Second, in DIC the drop frequency is regulated so that a given set of drops evaporates completely prior to the arrival of the next set. Finally, in DIC all the liquid evaporates – single phase liquid enters the evaporator and only vapor leaves the evaporator.

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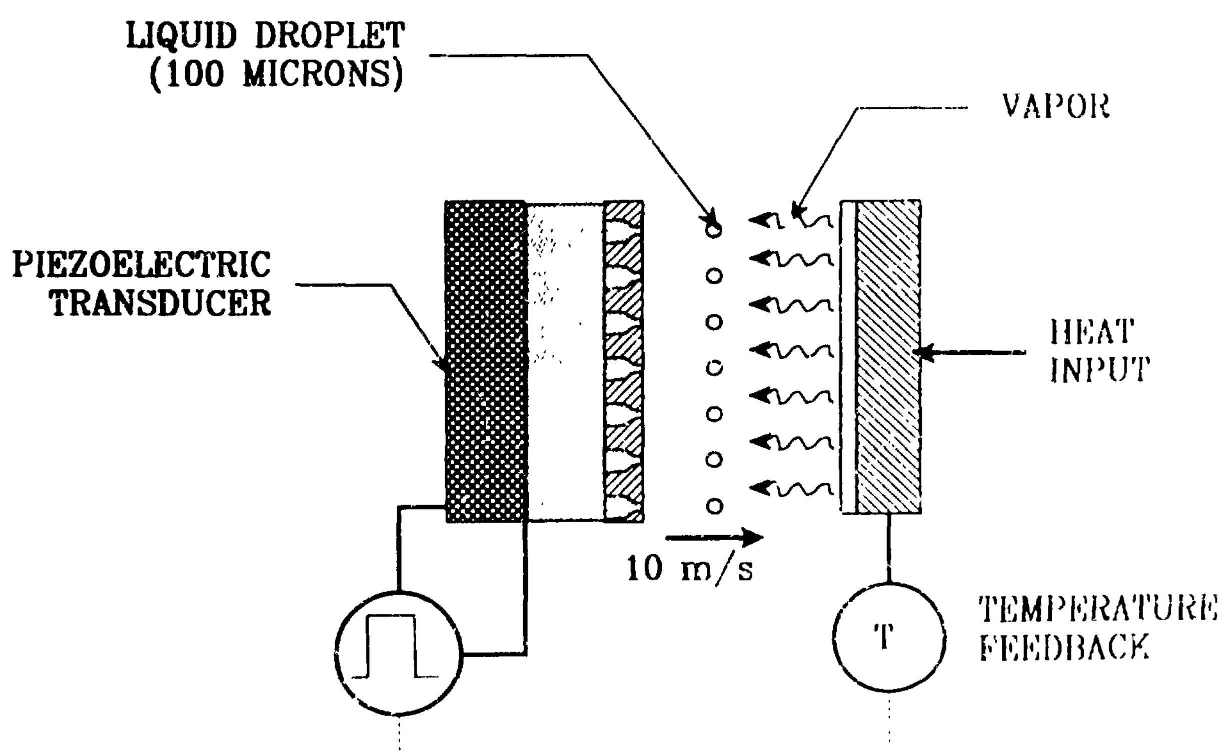
DROPLET IMPINGEMENT COOLING CONCEPT



There are two modes of droplet impingement cooling: film evaporation, and isolated drop evaporation. If the drop packing density is high enough, adjacent drops will merge upon impact forming a continuous film. If the drop packing density is low, they will evaporate as separate lens shaped drops.

The drop spreading time and the liquid thermal diffusion time are orders of magnitude shorter than the drop evaporation time. Hence, the DIC process can be modeled as quasi–steady state conduction through a progressively thinner liquid film or lens.

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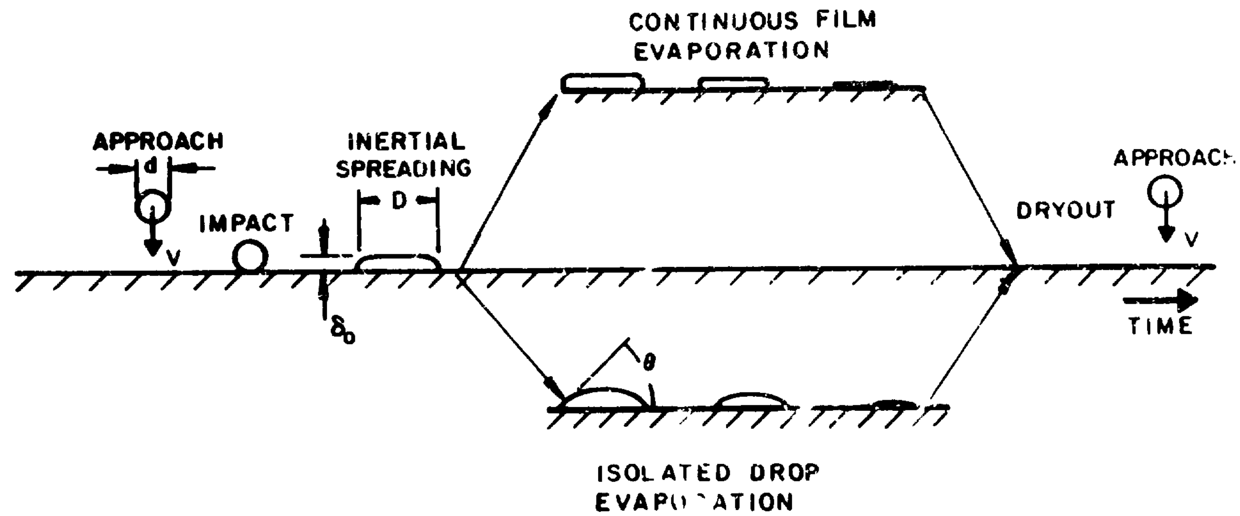
DIC CHARACTERISTIC DIMENSIONS AND TIME SCALES



DROP DIAMETER	$d \cong 100\ \mu m$
DROP VELOCITY	$V \cong 10$ ms
SPOT DIAMETER	$D \cong 500\ \mu m$
INITIAL SPOT THICKNESS	$\delta \cong 3\ \mu m$
DROP FLIGHT TIME	$\tau_v \cong 0.3$ m/s
SPOT FORMATION TIME	$\tau_f \cong 0.05$ m/s
SPOT THERMAL DIFFUSION TIME	$\tau_t \cong 0.04$ m/s
SPOT EVAPORATION TIME	$\tau_c \cong 10$ ms

There are four heat transfer regimes in DIC. For low surface superheats, bubble nucleation is inhibited by the steep temperature gradient in the film, and evaporation only takes place at the surface of the film. Beyond a critical superheat value, which depends on the initial film thickness, nucleate boiling is initiated. In nucleate boiling the bursting bubbles fling liquid away from the surface reducing the cooling capacity of the drops and leading to the critical heat flux (CHF) condition. The heat flux decreases with further increases in wall superheat. At even higher values of wall superheat the Leidenfrost condition is reached, where the droplets no longer contact the surface. Evaporation of the droplets as they approach the surface forms a vapor cushion which causes the drops to rebound. Only a small percentage of the drop volume is evaporated in this heat transfer regime.

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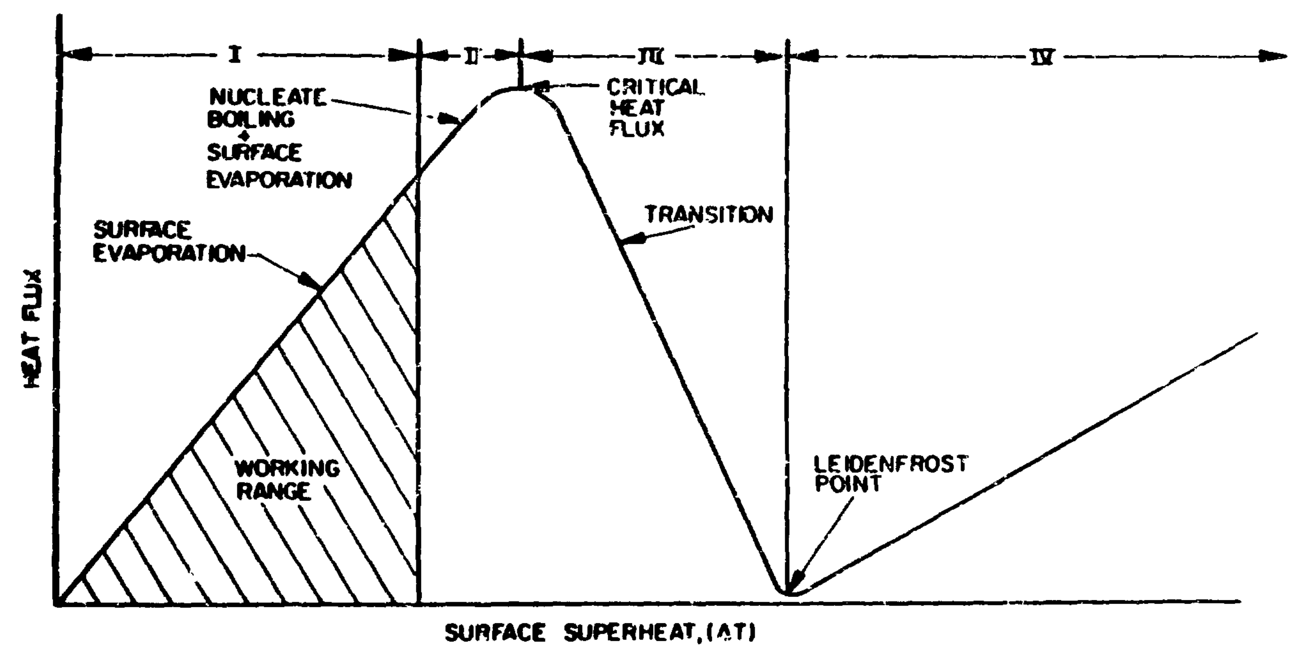
DIC HEAT TRANSFER REGIMES



Proof–of–concept experiments performed in water confirm the shape of the heat transfer regimes curve and demonstrate that very high heat fluxes are possible in DIC. The measured CHF was 330 W/cm^2, more than twice CHF for pool boiling. The heat transfer coefficient is also very high, about 9 W/cm^2–°C.

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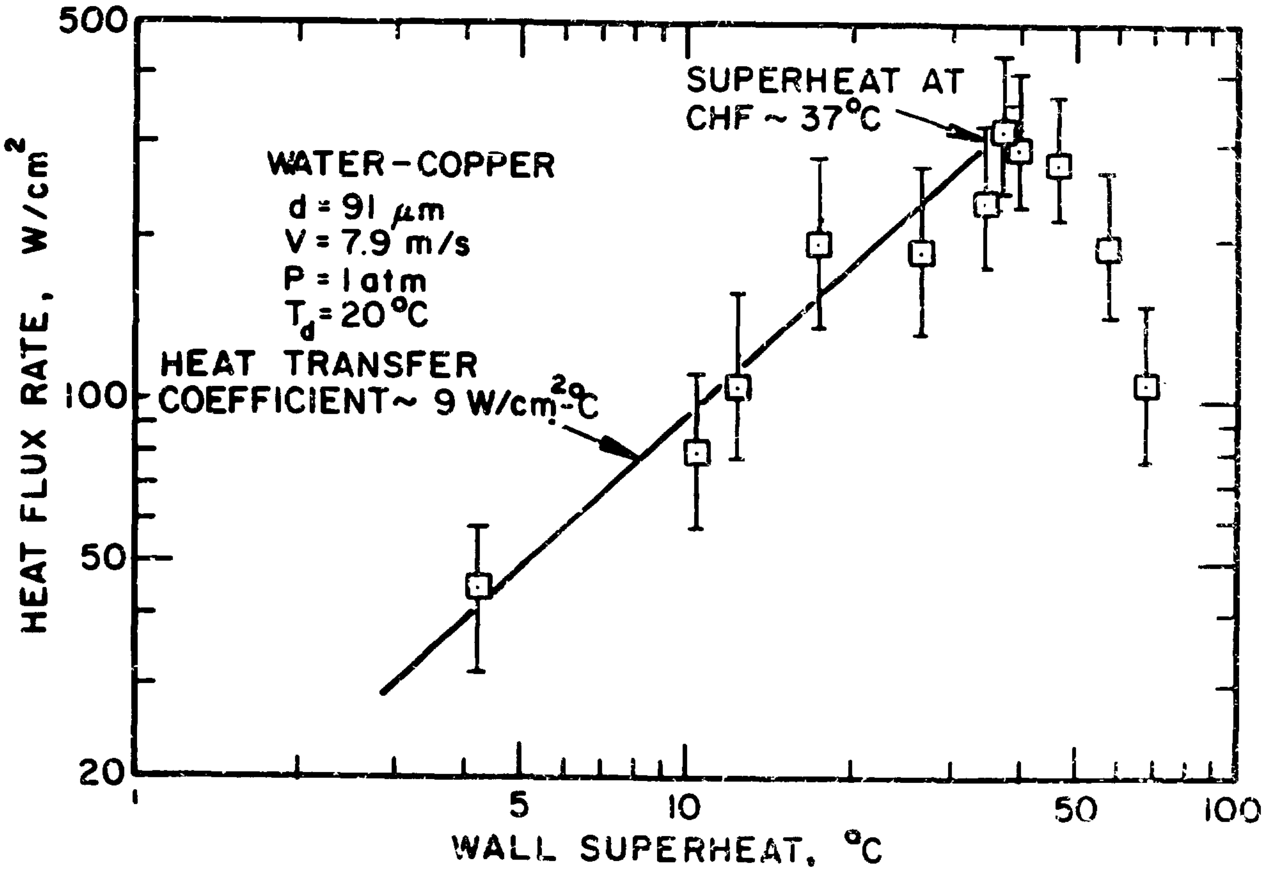
DIC DATA FOR WATER AT 1 ATM



The DIC heat transfer coefficient depends primarily on the initial liquid film thickness (film thickness immediately following drop impact). For conditions of interest, the initial film thickness will range between 5 and 10 microns. Heat transfer coefficients in ammonia can be as high as 15 W/cm^2°C.

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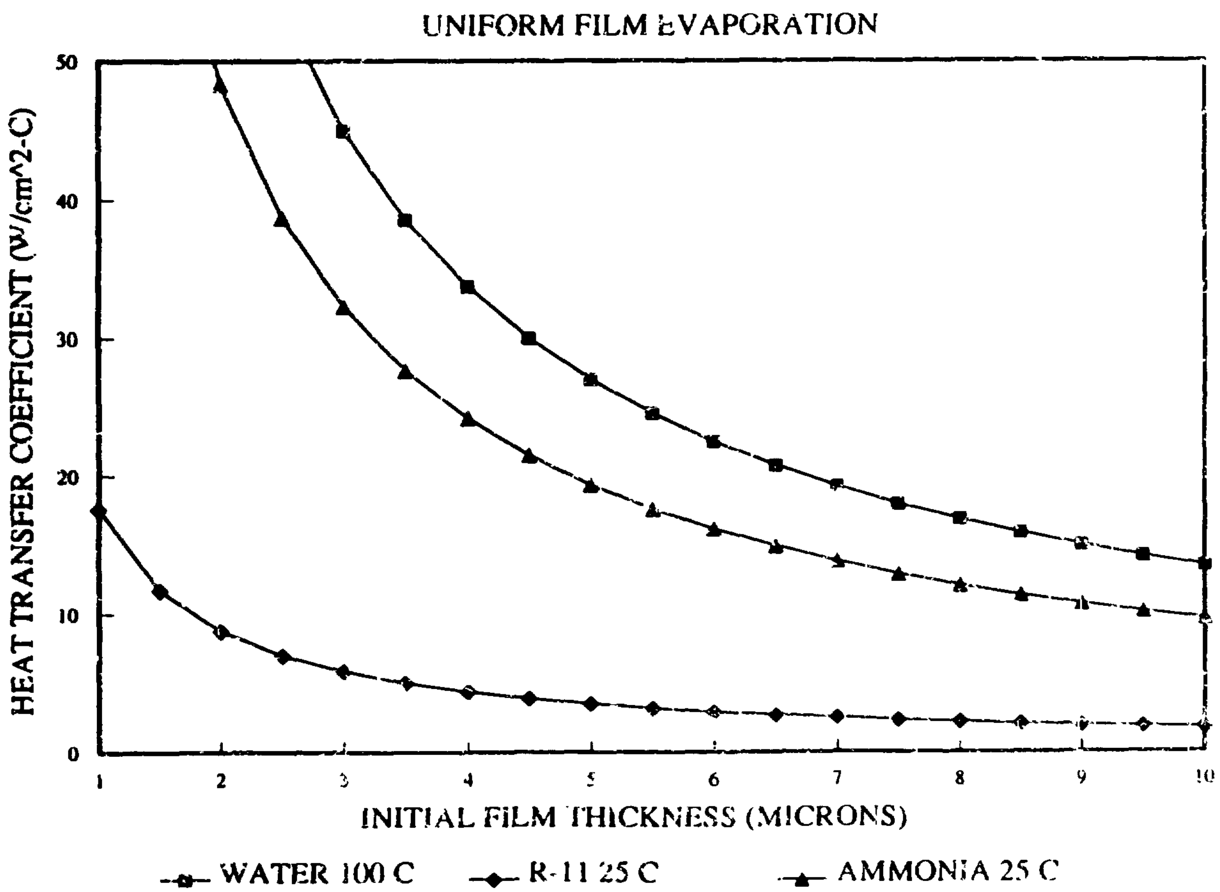
PREDICTED DIC HEAT TRANSFER COEFFICIENT



The condenser uses Gregorig fins in conjunction with an internal drainage network to achieve heat transfer coefficients comparable to those of the droplet evaporator. The main innovation in this condenser is that the condensate is drained through ducts embedded in the wall itself. The condensate only travels a short distance (less than a millimeter) over the surface of the condenser before being removed from the surface. Hence, the entire capillary pressure gradient available can be used to drive the condensate over this short distance. The resulting liquid velocities are high, leading to condensate films only a few microns thick.

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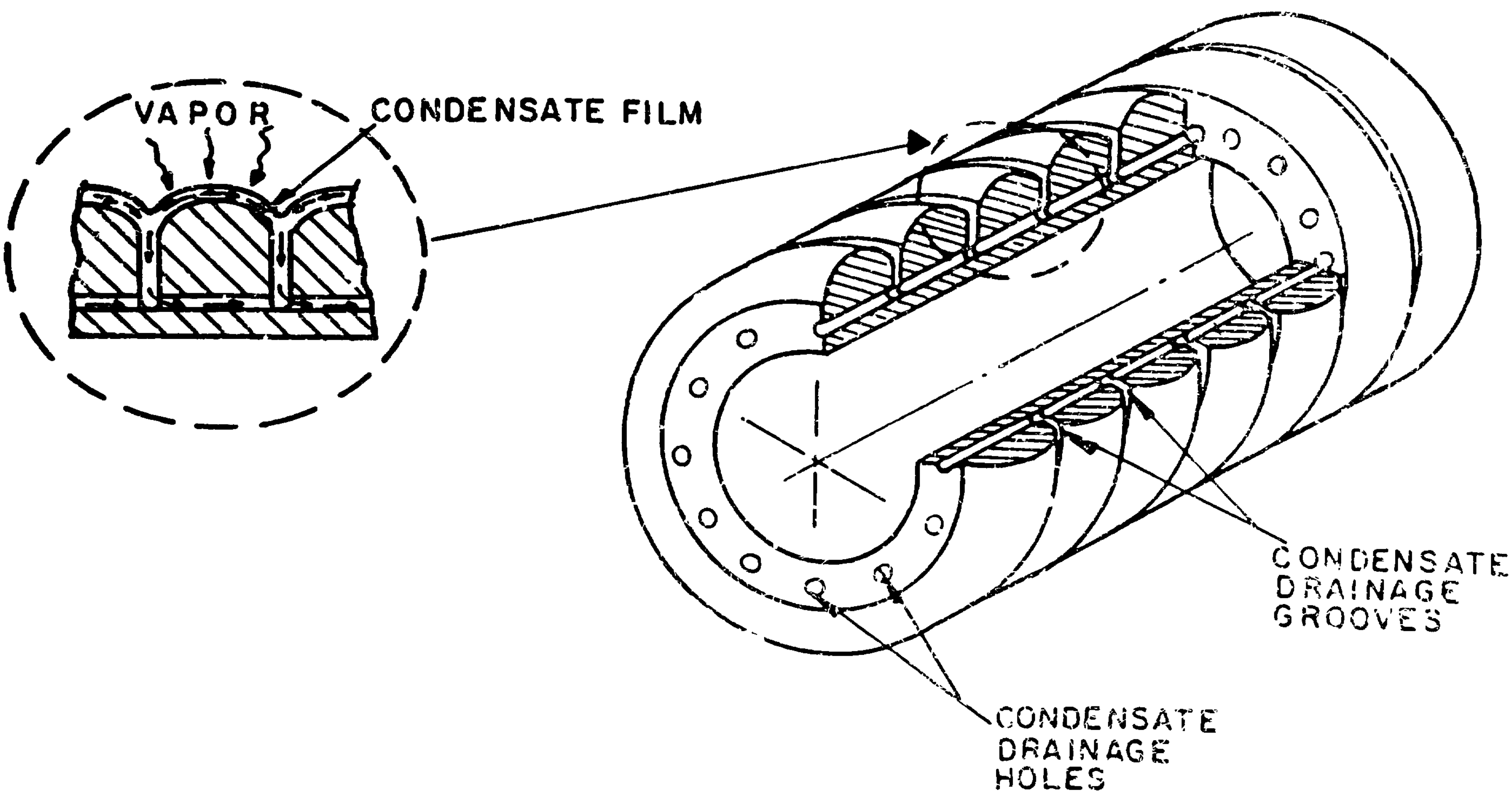
INTERNALLY DRAINED CONDENSER CONCEPT



By optimizing the shape of the fins very high capillary pressure gradients can be obtained. For a 1 mm wide fin, the capillary pressure gradient can be 20 times higher than gravitational pressure gradient in 1g. The high pressure gradient results in extremely thin condensate films and hence high heat transfer coefficients.

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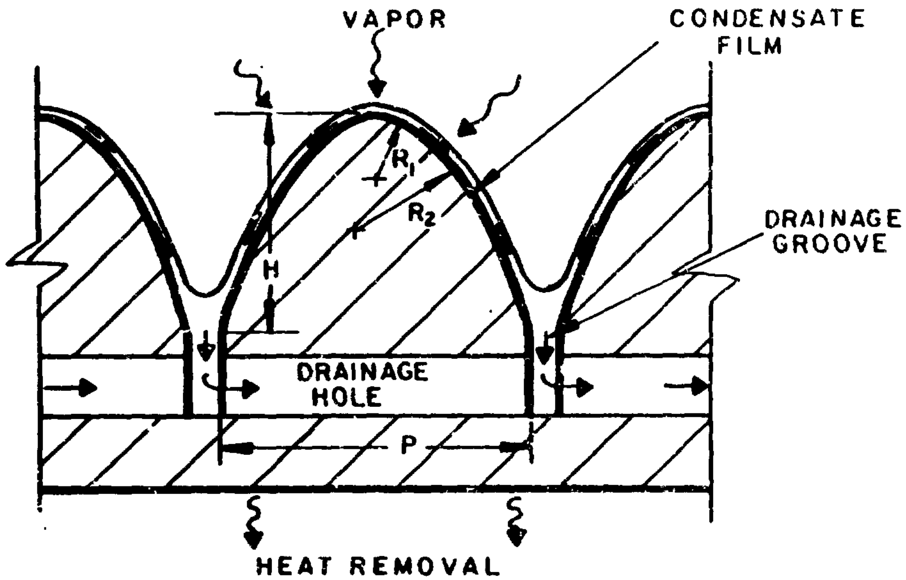
IDC OPERATION



Capillary Drainage

$$\left.\frac{dp}{ds}\right|_c = \sigma \frac{d}{ds}\left|\frac{1}{R}\right| = \frac{3\,\sigma}{P^2}$$

Gravitational Drainage

$$\left.\frac{dp}{ds}\right|_g = \frac{H\rho g}{S} = 0.6\,\rho g$$

Ammonia, P = 1 mm

$$\frac{\left.\frac{dp}{ds}\right|_c}{\left.\frac{dp}{ds}\right|_g} = 20$$

The heat transfer coefficient in the internally drained condenser (IDC) depends on the suction level applied to the drainage ducts. At suction levels below the drainage groove capillary pressure, the fins are partially flooded and the heat transfer coefficient varies from close to zero to a maximum value. For suction levels between one and four times the capillary pressure of the drainage groove, the heat transfer coefficient remains constants. At higher suction levels, vapor is pulled into the drainage ducts. By controlling the suction level, the IDC can be used as a variable conductance element. Because the grooves are so small, very little liquid is required to vary the heat transfer coefficient over a wide range. This would reduce the size of the accumulator required for temperature control of the thermal bus.

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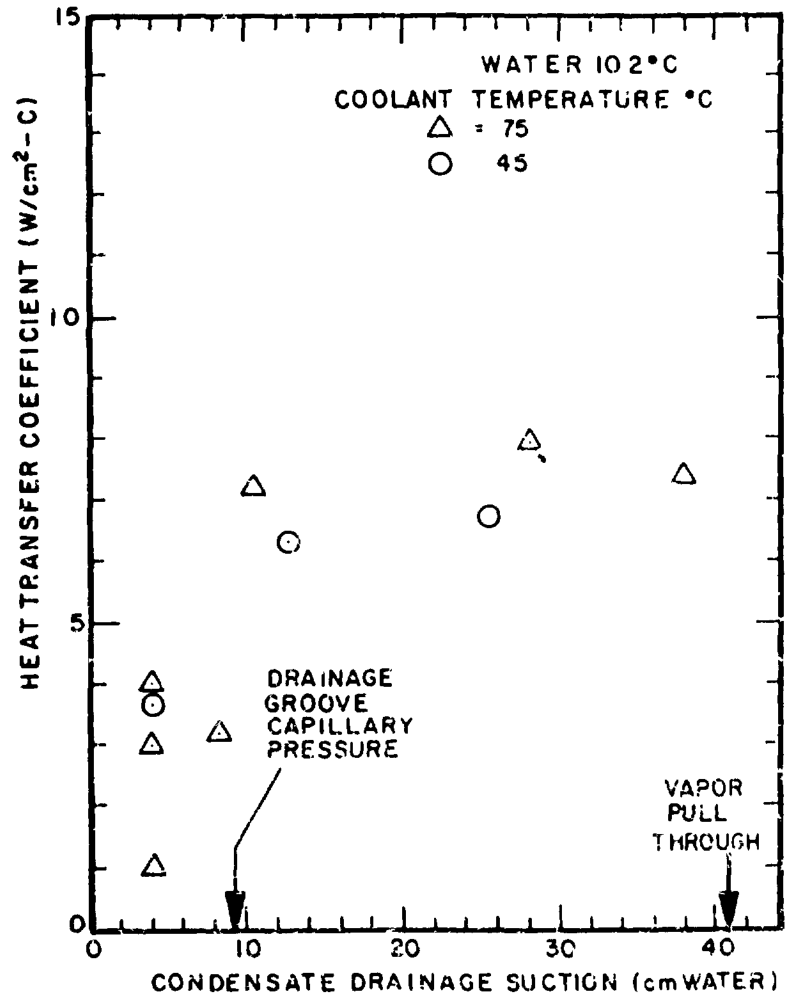
EFFECT OF SUCTION LEVEL ON IDC PERFORMANCE



The heat transfer coefficient in the IDC is very high, about eight times higher than that of a smooth tube operating at equal heat flux. The IDC contoured fins provides about twice as much enhancement as rectangular fins in a 1 G environment. The performance of the IDC can be well predicted using analytical models.

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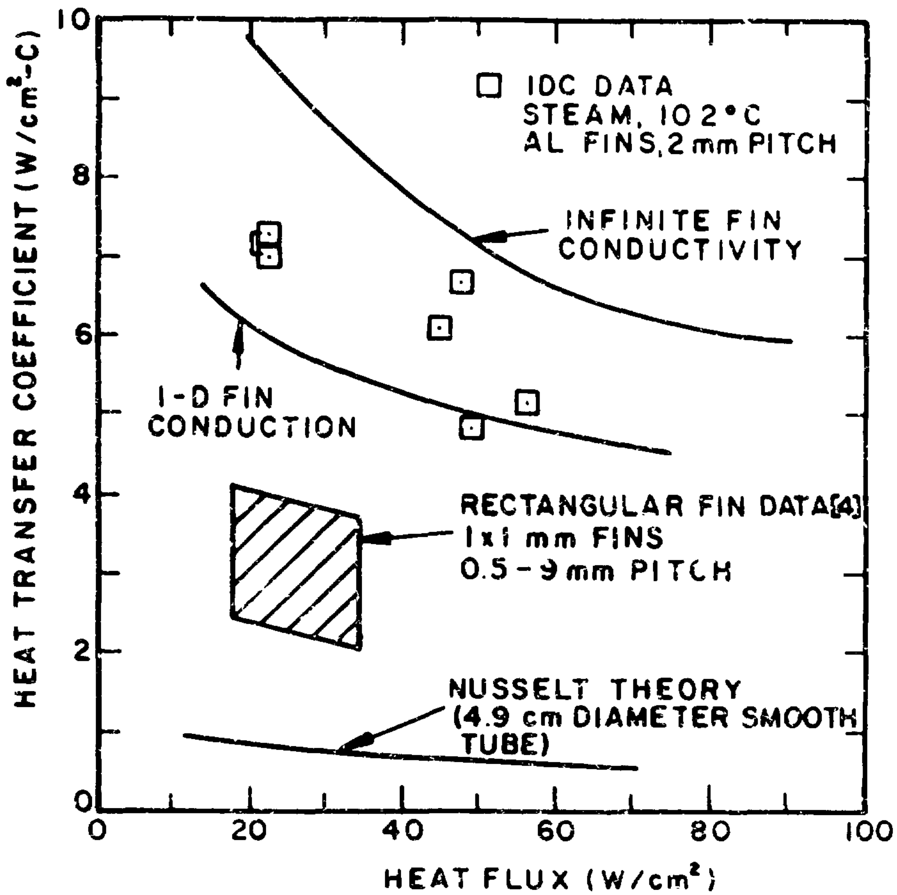
IDC MEASURED HEAT TRANSFER COEFFICIENT



High heat transfer coefficients can also be achieved in ammonia. A surface suocooling of only 1°C will result in a heat flux of about 9 W/cm^2–°C.

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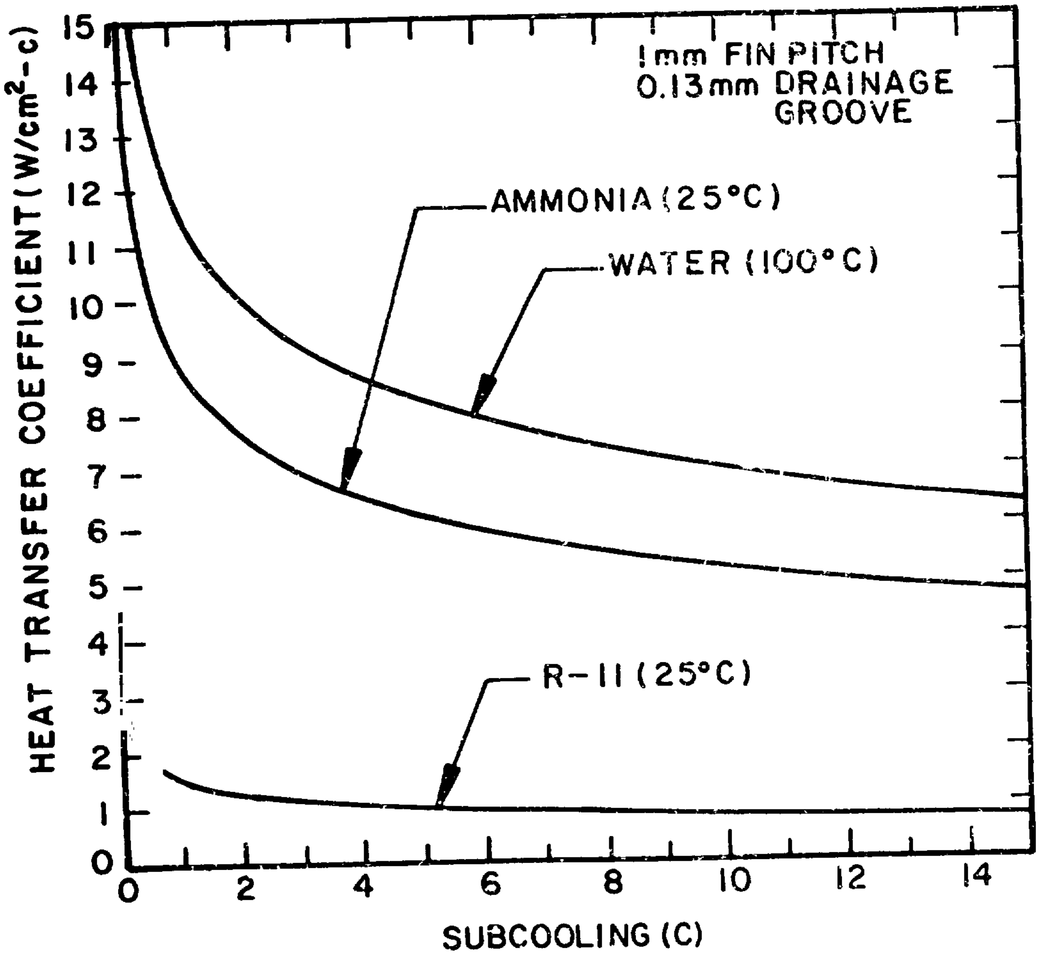
PREDICTED CONDENSATION HEAT TRANSFER COEFFICIENT AS A FUNCTION OF SUBCOOLING



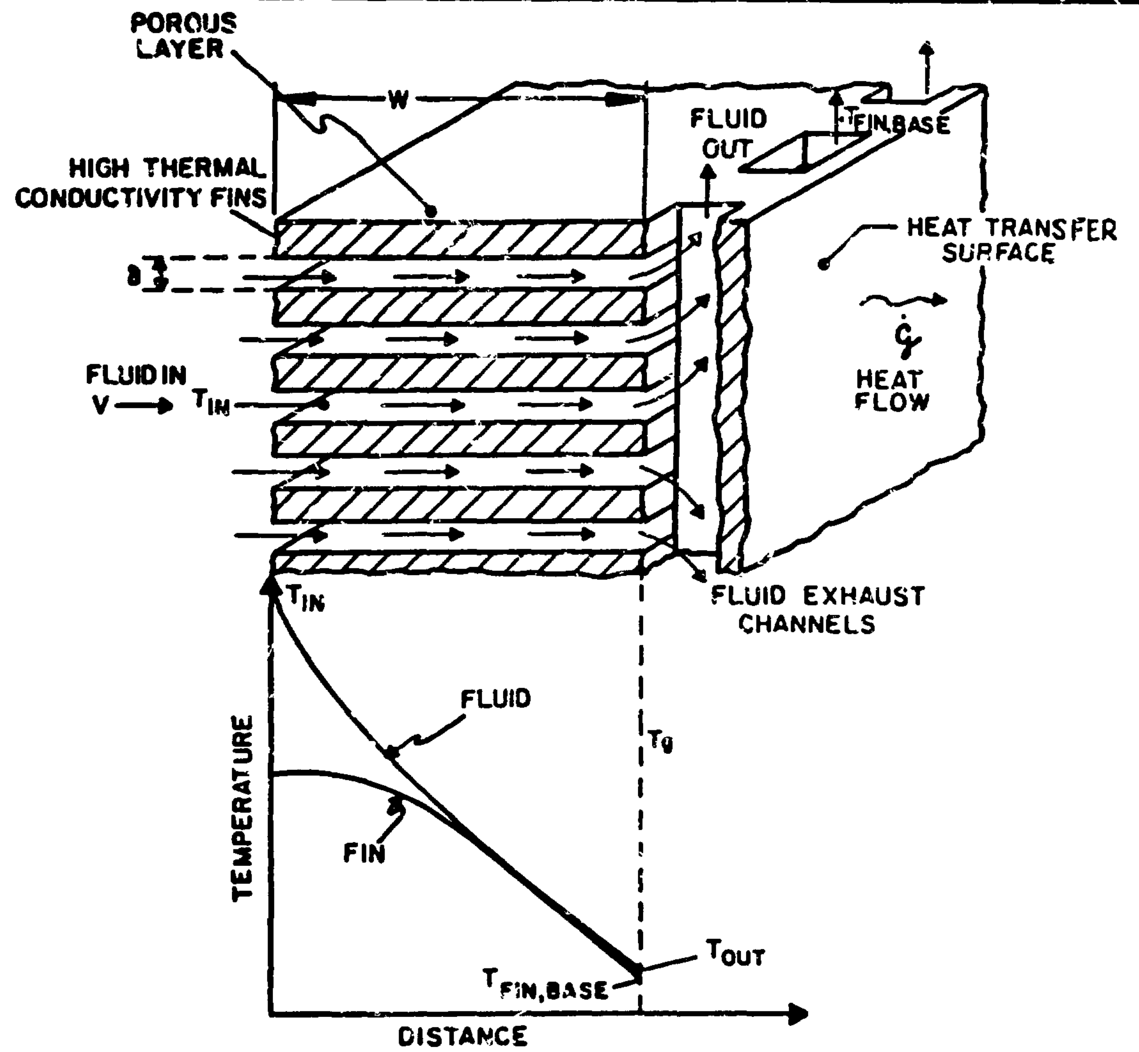
The single–phase heat exchanger is perhaps the most challenging component from a performance standpoint. Heat transfer coefficients in conventional single–phase heat exchangers are so much lower than those in two–phase systems that the water side drives the size and performance of the heat exchanger. Creare has developed[1] a new single–phase heat exchanger concept which can achieve heat fluxes comparable to those of the droplet evaporator with high effectiveness and low pressure drop.

The heat exchanger consists of a layer of closely spaced fins in good thermal contact with the heat transfer surface. The fins form a "porous layer" though which the fluid flows. The fin spacing is small, typically 0.1 mm or less. The fluid is therefore in excellent thermal communication with the fins. The fluid leaves the heat exchanger through an array of small diameter ducts located at the interface between the "porous layer" and the heat transfer surface.

The main difference between the porous wall heat exchanger concept and conventional finned plate heat exchangers is the direction of the flow. In the PWHX the fluid flows in a direction normal to the heat transfer surface, whereas in a conventional heat exchanger, the fluid flows parallel to the surface. Normal flow aligns the directions of the temperature gradients in the fins and the fluid and results in high effectiveness even at high heat fluxes. The fins are very short and, therefore, pressure drops in the PWHX are quite small.

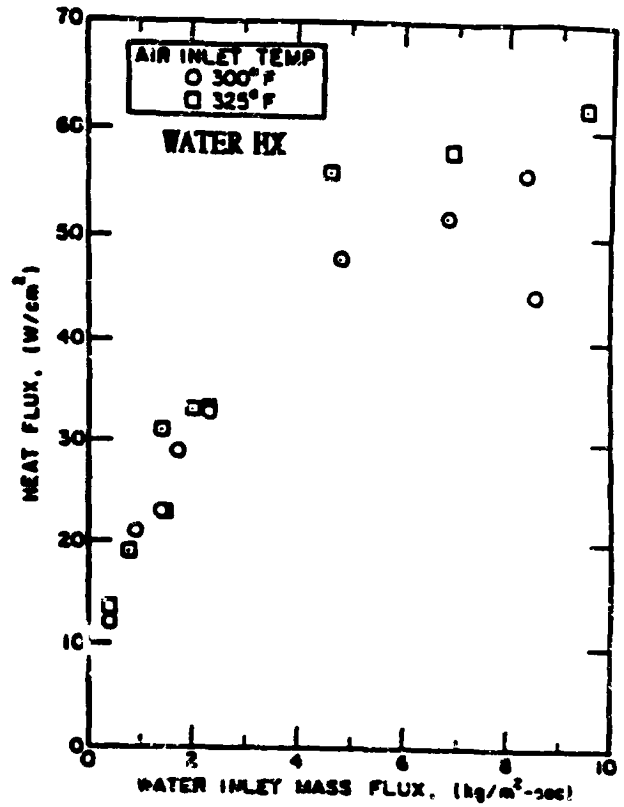
[1] Patent pending.

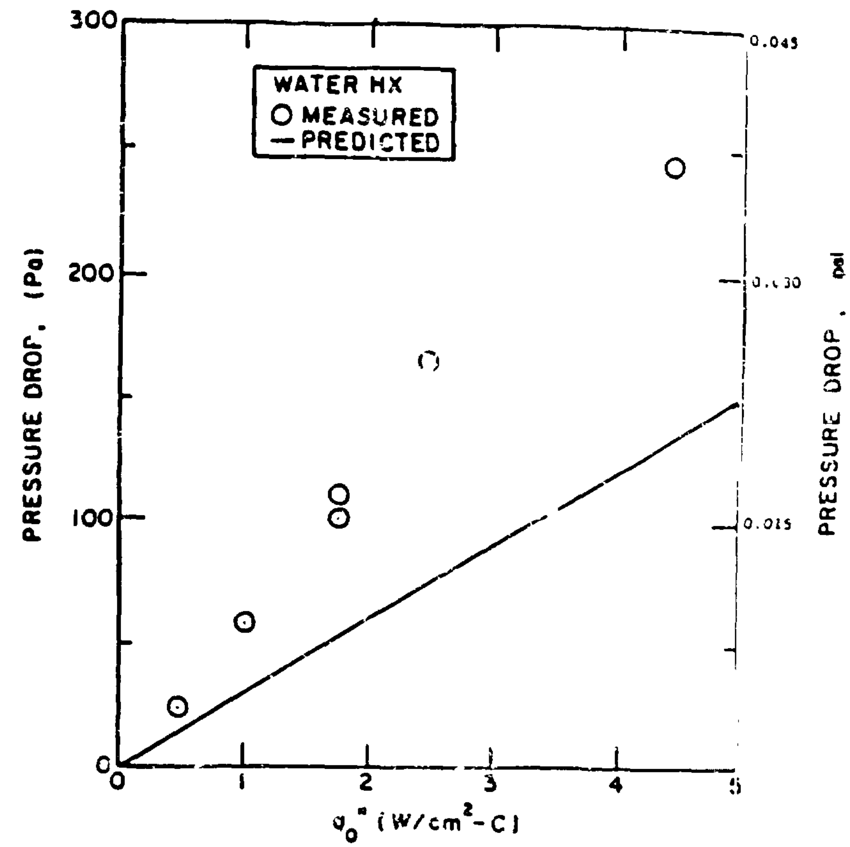
PORQUS WALL HEAT EXCHANGER CONCEPT
PWHX



Proof–of–concept experiments performed in water achieved heat fluxes in excess of 60 W/cm^2–C with a pressure drop of only 250 Pa (0.04 psi).

MEASURED HEAT FLUX AND PRESSURE DROP





The PWHX heat transfer coefficients are extremely high for a single phase heat exchanger. At a heat flux of 60 W/cm^2 the heat transfer coefficient was 4 W/cm^2—°C. The effectiveness was also high, about 70%. The heat transfer coefficient and effectiveness are lower than predicted because of flow maldistribution resulting from uneven fin spacing. Later experiments performed in air and helium have shown good agreement with the analytical models.

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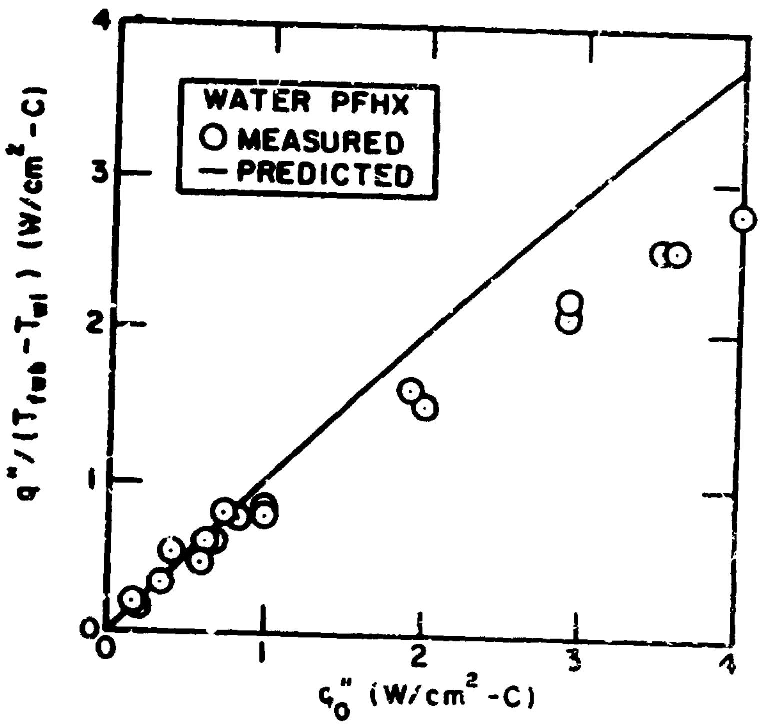
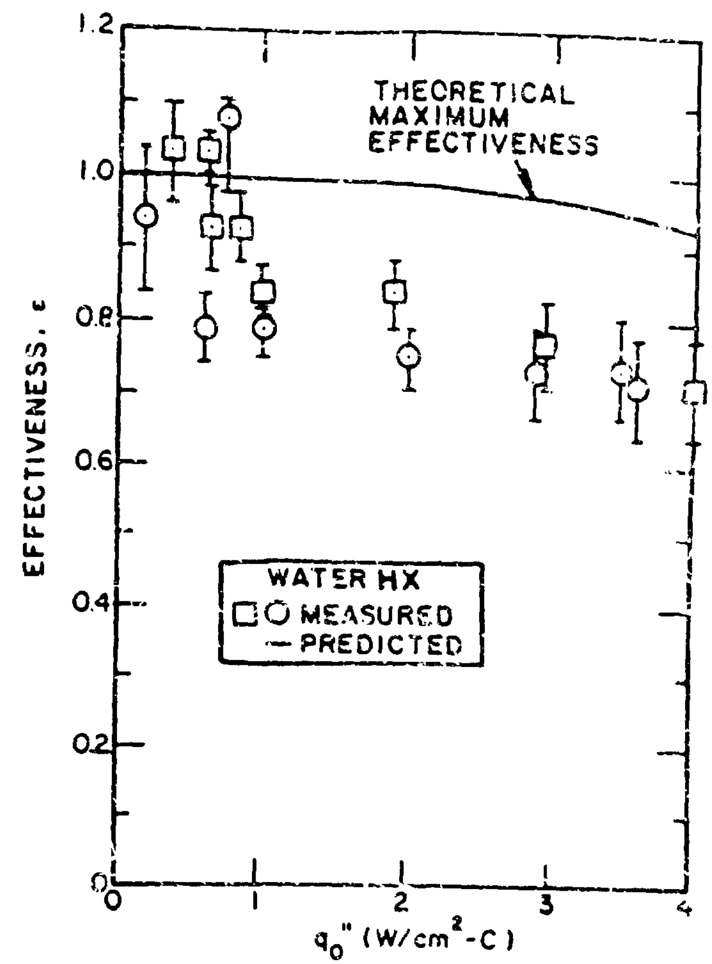
MEASURED HEAT TRANSFER COEFFICIENT AND EFFECTIVENESS





The PWHX performance depends strongly on the fin spacing. Smaller fin spacing leads to higher heat transfer coefficients and higher effectiveness. For a fin spacing of 0.05 mm, heat transfer coefficients of 20 W/cm^2–°C can be achieved with effectiveness in excess of 80%.

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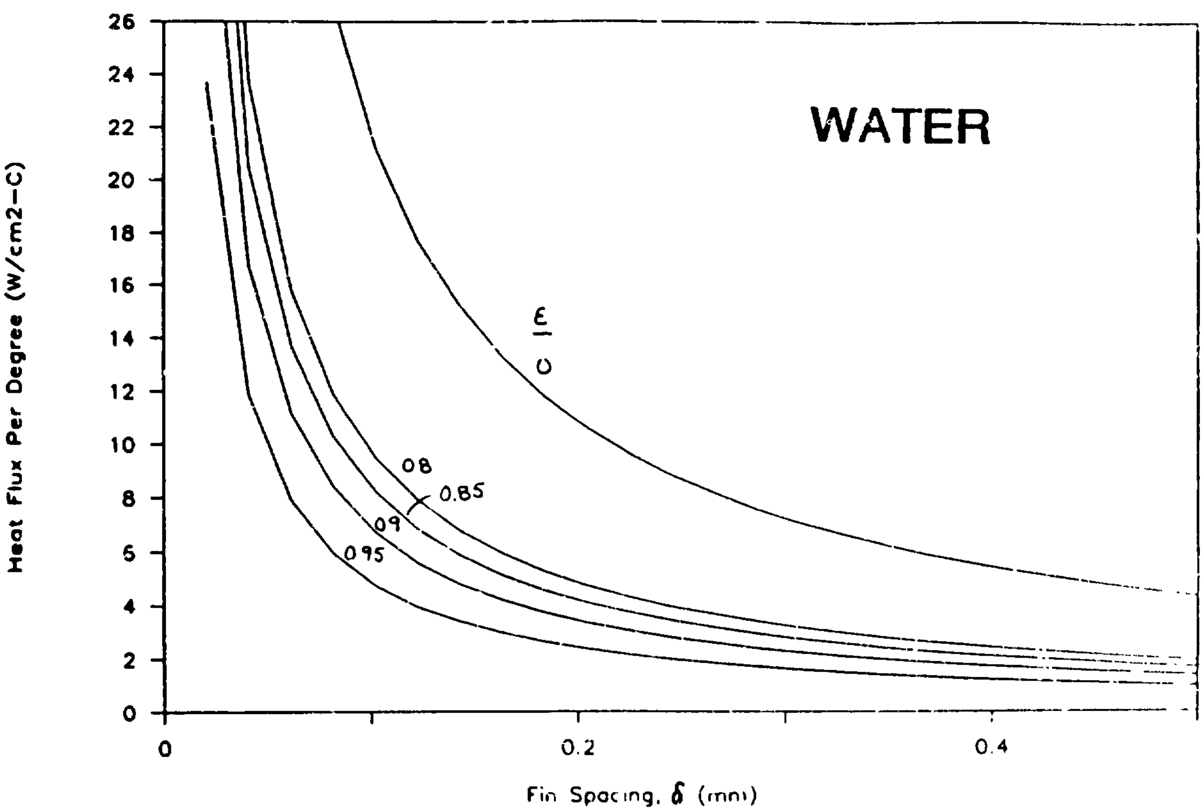
PREDICTED PWHX HEAT TRANSFER COEFFICIENT



The PWHX pressure drops are very small, typically less than a tenth of a psi.

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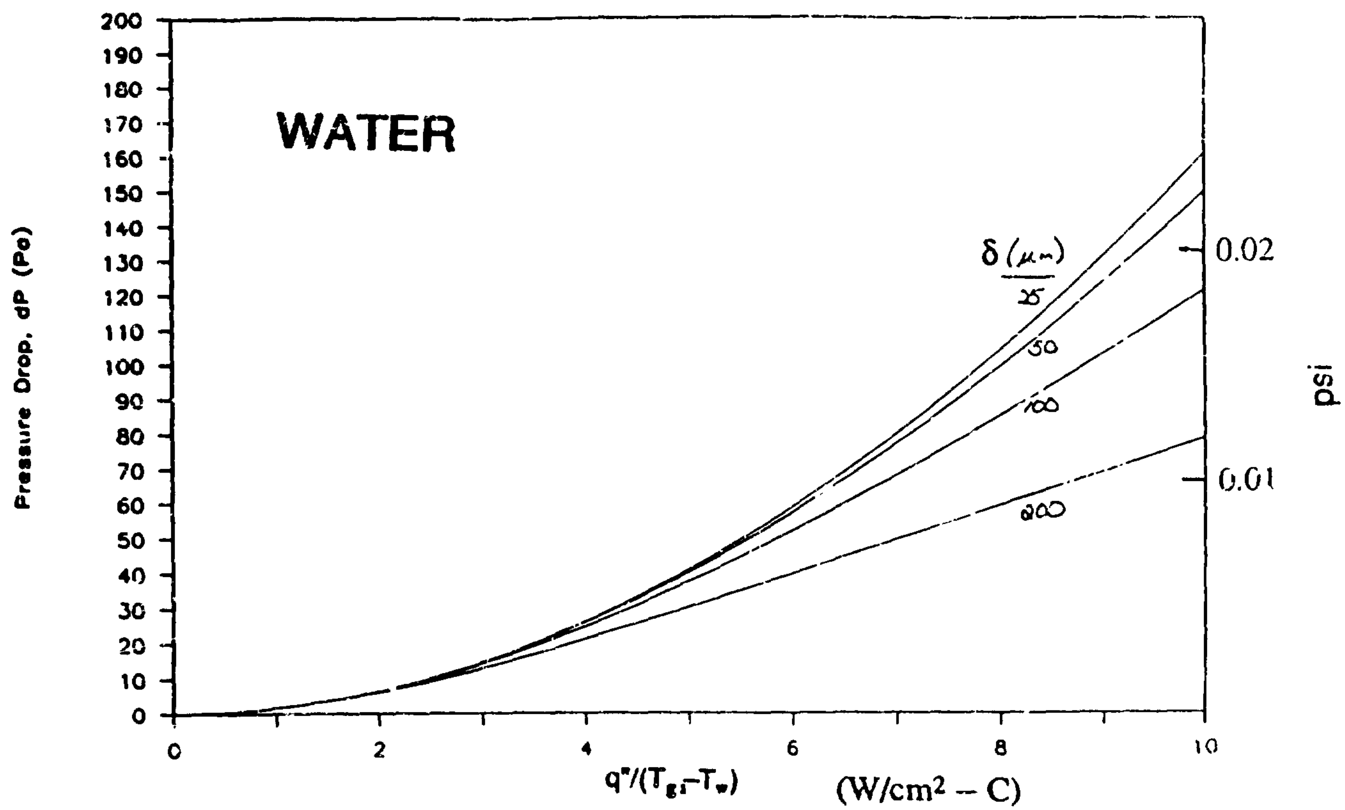
PREDICTED POROUS LAYER PRESSURE DROP



We are presently developing a payload interface heat exchanger which combines droplet impingement cooling on the evaporator side and the internally drained condenser on the condenser side. The heat exchanger will have a nominal capacity of 2 kW with an overall temperature difference of 5 °C or less.

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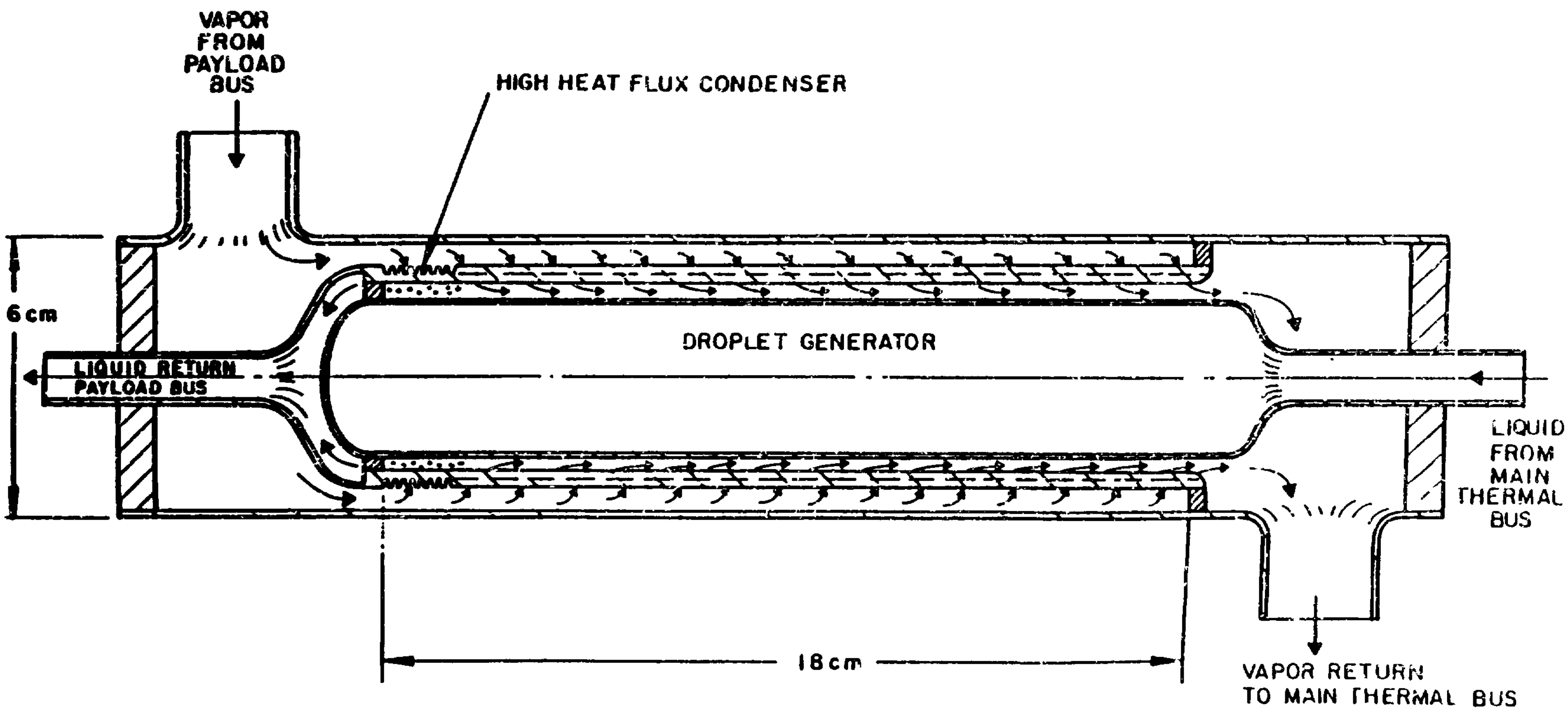
CONCEPTUAL DESIGN OF PAYLOAD CTB–IF–HX



We are also developing a habitat interface heat exchanger which combines droplet impingement cooling on the ammonia side with the PWHX on the water side. The heat exchanger will also have a nominal capacity of 2 kW, an overall effectiveness of 73%, and a water side pressure drop of 0.1 psi.

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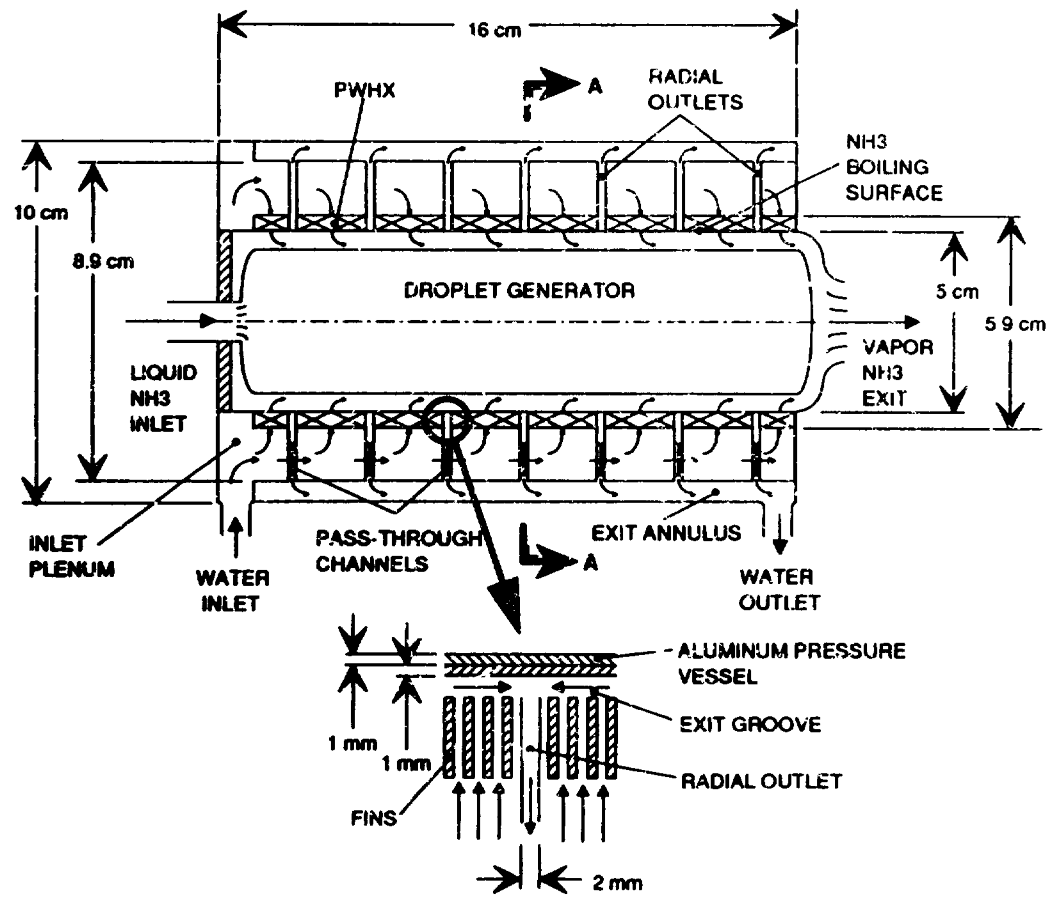
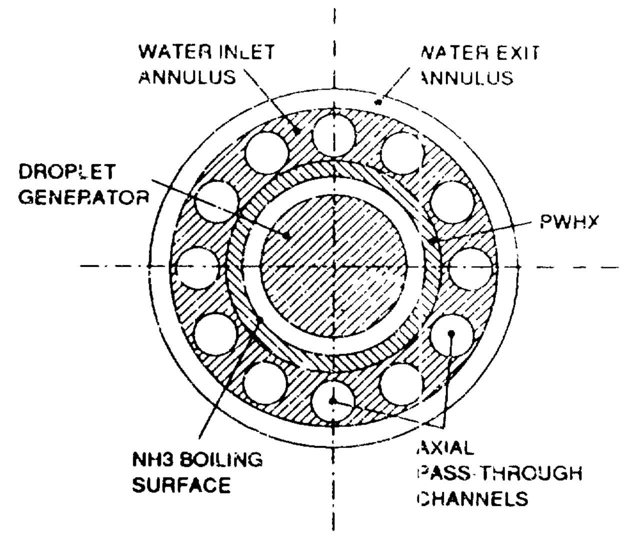
CONCEPTUAL DESIGN OF HABITAT CTB–IF–HX





SECTION A - A

The performance goals for these heat exchanger breadboards represent a several fold increase in the heat flux of present interface heat exchangers.

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PERFORMANCE GOALS

1. PAYLOAD CTB–IF–HX

- h_{evap} = 10 W/cm^2–°C
- h_{cond} = 10 W/cm^2–°C
- q" = 10 W/cm^2

2. HABITAT CTB–IF–HX

- h_{evap} = 10 W/cm^2–°C
- h_{water} = 2 W/cm^2–°C
- q" = 10 W/cm^2
- EFFEC = 73%
- ΔP_{water} = 150 Pa (0.1 psi)

The technology development issues in this program involve a combination of heat transfer modeling and optimization, coupled with the development of suitable fabrication techniques. All three heat transfer concepts require flow passages with small dimensions and extremely tight tolerances.

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TECHNOLOGY DEVELOPMENT ISSUES

1. DROPLET EVAPORATOR
   - 1.1 DROPLET IMPINGEMENT HEAT TRANSFER
   - 1.2 MULTIORIFICE DROPLET GENERATOR
     - – Nozzle Drilling
     - – Piezoelectric Transducer Design

2. INTERNALLY DRAINED CONDENSER
   - 2.1 SURFACE SHAPE OPTIMIZATION
   - 2.2 FABRICATION TECHNIQUES
     - – Surface Shape
     - – Drainage Grooves
     - – Drainage Ducts

3. POROUS WALL HEAT EXCHANGER
   - 3.1 HEAT TRANSFER AND PRESSURE DROP MODELS
   - 3.2 FABRICATION TECHNIQUES
     - – Fins
     - – Cu/Al Bonding
     - – Flow Ducts

We have completed the design work for the evaporator and condenser. We will be testing the condenser and evaporator by themselves by mid 1990, and combined into an interface heat exchanger by the end of the year. The PWHX heat exchanger will be tested as a separate component in the second half of 1990, and integrated with the evaporator in early 1991. The next step in the development of this technology would be to integrate these components into a thermal bus.

PROGRAM SCHEDULE