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SUBCOOLED FREON-11 FLOW BOILING IN TOP-HEATED FINNED COOLANT CHANNELS WITH AND WITHOUT A TWISTED TAPE

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Alvin Smith and Ronald D. Boyd, Sr.
Department Of Mechanical Engineering
P.O. Box 397
Prairie View A&M University
Prairie View, Texas

ABSTRACT

An experimental study has been conducted in top-heated finned horizontal tubes to study the effect of enhancement devices on flow boiling heat transfer in coolant channels. The objectives of the present work are to: (1) examine the variations in both the mean and local (axial and circumferential) heat transfer coefficients for circular coolant channels with spiral finned walls and/or spiral fins with a twisted tape, and (2) improve the data reduction technique of a previous investigator. The working fluid is freon-11 with an inlet temperature of 22.2°C (approximately 21°C subcooling). The coolant channel's exit pressure and mass velocity are 0.19 M Pa (absolute) and 0.21 Mg/m^2s, respectively. Two tube configurations were examined; i.e., tubes had either 6.52 (small pitch) or 4.0 (large pitch) fins/cm of the circumferential length (26 and 16 fins, respectively). The large pitch fins were also examined with a twisted tape insert. The inside nominal diameter of the copper channels at the root of the fins was 1.0 cm.

The results show that by adding enhancement devices, boiling occurs almost simultaneously at all axial locations. The case of spiral fins with large pitch resulted in larger mean (circumferentially averaged) heat transfer coefficients, h_m, at all axial locations. Finally, when a twisted tape is added to the tube with the large-pitched fins, the power required for the onset of boiling is reduced at all axial and circumferential locations.

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Nomenclature

A_c	Cross sectional area, m^2	N	Number of fins per test section
A_s	Surface area, m^2	p	Static exit pressure, MPa
CHF	Critical heat flux, W/m^2	q	Heat flux W/m^2
D	Tube diameter, cm	r	Radial coordiante for data reduction model, m
D_h	Heated hydraulic diameter, m	R	Tube radius, cm
G	Mass velocity, kg/m^2s	T_f	Bulk temperature of the flowing fluid, ^oC
h	Local heat transfer coefficient, $W/m^{2o}C$	T_m	Local measured outside wall temp. of the tube, ^oC
h_m	Mean heat transfer coefficient, $W/m^{2o}C$	T_s	Local inside wall temp. of the tube, ^oC
H	fin height	T_{sat}	Saturation temperature, ^oC
K	Thermal conductivity, $W/m\ K$	T_∞	Ambient temperature, ^oC
L	Heated length of test section, m	Z	Axial coordinate for heated portin of test section, cm

Greek Letters

ρ Density, kg/m^3; φ Circumferential coordinate; π Half of a full rotation or 180 ^{o}(bottom of test section)

Introduction

Subcooled flow boiling (locally boiling fluid whose mean temperature is below its saturation temperature while flowing over a surface exposed to a high heat flux) has the greatest potential of accommodating high heat fluxes [1]. In space, the active cooling of cold plates, turbine blades, fusion reactor components, chemical reactor components, electronic components, refrigeration systems components, and other systems necessary for the successful commercialization of space will require an efficient heat transfer system. Therefore, the need to understand the behavior of two-phase flow boiling in both the space (zero-g or reduced-g) and earth environments is essential to thermal designers, especially where pumping power requirements must be optimized. Since thermal designers are forced on occasion to use data from uniform heat flux applications for non-uniform heat flux applications, it is important to assess the effect of the non-uniformity on the local and mean heat transfer.

Ideal heat transfer is achieved when the sum of the component thermal resistances is reduced to a minimum. However, flow dynamics through plain tubes rarely provides the ideal conditions needed, because the stationary fluid layer at the tube wall leads to the buildup of a slow moving laminar boundary layer caused by frictional drag forces. This results in heat transfer being principally controlled by fluid conductivity and the thickness of this layer.

Research related to enhanced surfaces for heat transfer has increased in recent years [2]. The immediate benefit from recent research has led to increasing efficiency of heat exchanges and to the development of the spirally fluted tube [3]. Fins (sometimes called extended surfaces) are often used to increase heat transfer rates to fluids in cases where low heat transfer coefficients exist. If the wall temperature is sufficiently high to cause film boiling and all other resistances are small, fins will cause an increase in the heat transfer rates in a boiling situation [4]. Examples include cooling of nuclear fuel rods, cooling of electrical equipment such as radio-power tubes, and some evaporators. Devices for the establishment of fluid swirl are often used to increase heat transfer in tube flows. The increase in pressure drop, however, is less than the increase in heat transfer in many cases. Twisted tape inserts have been studied by Hong and Bergles [5], who found that in laminar flow, heat transfer could be increased by a factor of as much as 9 whereas pressure drop increased by a factor of less than 4 over empty tube values. In turbulent flow, twisted tape inserts can improve heat transfer by a factor of 2 over empty tube values, but the pressure drop is increased by several orders of magnitude. Brown Fintube Company reported in its literature that increases in heat transfer rate of up to 300 percent have been achieved in viscous fluids with nominal increases in pressure drop with the addition of twisted tapes.

Experimental Investigation

The flow loop (Figure 1) is a closed system designed to control the working fluid (freon-11) and reservoir temperatures, pressures, heating capabilities, and overall system stability. The reader is referred to references 6-8 for detailed descriptions of the experimental flow loop, procedures, and data acquisition.

The loop was designed to study both saturated and subcooled flow boiling, although only subcooled flow boiling was considered here. By allowing both saturated and subcooled flow regimes, the heat rejection can be either isothermal or at a slightly variable axial temperature. The latter case (subcooled flow boiling) will result in enhanced heat rejection capability (above the saturated case) to the extent that the given application will allow a variable axial temperature.

Referring to Figure 1, the reader should be made aware of the possible circulation patterns of the working fluid that can be accomplished by opening and closing various valves within the loop. However, for the tests performed within this investigation, only reservoir #1 was utilized.

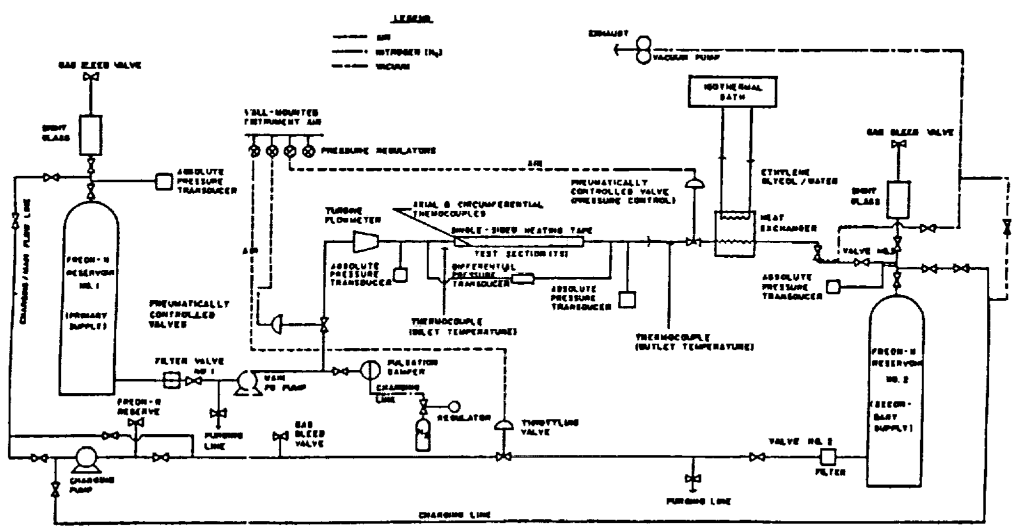

Figure 1. Freon-11 flow loop for both subcooled and saturated flow boiling experiments.

The measured outside wall temperatures of the test sections were used to calculate the unknown heat transfer coefficient, h. This was achieved through a data reduction technique developed by Turknett and Boyd [7], based on the heated hydraulic diameter, D_h. This approach will result in, at most, a qualitative indication of the local (axial and circumferential) distribution of h. A piece-wise linear profile method was used to evaluate the mean (circumferentially averaged) distribution of h_m. A finite difference technique (although not used for the present work) was developed to evaluate the local h [8].

Results

The results are presented for the case of subcooled freon-11 boiling in top-heated horizontal channels with internal fins and/or twisted tape inserts.

Figure 2 shows the circumferential variations in the heat transfer coefficient at the axial location Z_4 (center of the heated portion of the test section). It is evident that the variations in heat transfer coefficients in the circumferential location are increased by adding more enhancement devices. The spiral fin with large pitch shows no variations in the single phase region located at the bottom ($\frac{3\pi}{4}$ and π) of the tube, and minor variations at the top (0 and $\frac{\pi}{4}$) of the tube in the film boiling region. Adding a twisted tape (spiral fin large pitch tape) decreased variations in heat transfer coefficients in the film boiling region (0 and $\frac{\pi}{4}$), but increased the variations in the single phase region ($\frac{3\pi}{4}$ and π). The spiral fin small pitch shows the largest variation of h in the single phase region ($\frac{3\pi}{4}$ and π) and only a minor variation in the film boiling region (0 and $\frac{\pi}{4}$).

The axial variation of h as a function of power for the three (3) cases is presented in Figure 3. The axial locations Z_1 and Z_7 are excluded due to end losses and the effect of axial conduction. At circumferential location 0 (top of the tube), the onset of boiling appears to occur more uniformly along the length of the channels than at circumferential location $\frac{\pi}{4}$. At circumferential location $\frac{\pi}{4}$, the spiral fin large pitch (3a.2) provides higher heat transfer coefficients than any case presented. By adding a twisted tape to the large pitch tube, the power required for the onset of nucleate boiling was reduced at all axial and circumferential locations, but the average heat transfer coefficients were reduced (Figure 3b.2). The spiral fin small pitch tube at circumferential location $\frac{\pi}{4}$ (Figure 3c.2), provided a more uniform onset of boiling in the axial direction than the tubes with large pitch or large pitch with a twisted tape insert. However, at circumferential location 0, the tube with twisted tape had a more uniform onset of nucleate boiling. The average axial heat transfer coefficients at circumferential location 0 (top of the tube) were higher for the case of the tube with a small pitch than the other cases (spiral fin large pitch and spiral fin large pitch with tape), but were found considerably lower at circumferential location $\frac{\pi}{4}$ for the same cases.

Figure 4 is a plot of the mean (circumferentially averaged) heat transfer coefficient h_m, as a function of power. The averaged h (i.e., h_m) appears to be consistently higher for the case of the spiral fin large pitch (4a) than the cases with a tape (4b) and small pitch (4c). In some cases, the addition of the tape to the spiral fin large pitch reduced the average heat transfer coefficients in the axial direction. However, adding a twisted tape or reducing the pitch of the fins caused boiling to occur more uniformly over the length of the test section. Heat transfer coefficients literally remained unchanged in the spiral pitch tube (4c) in the axial direction when compared with Figure 3.

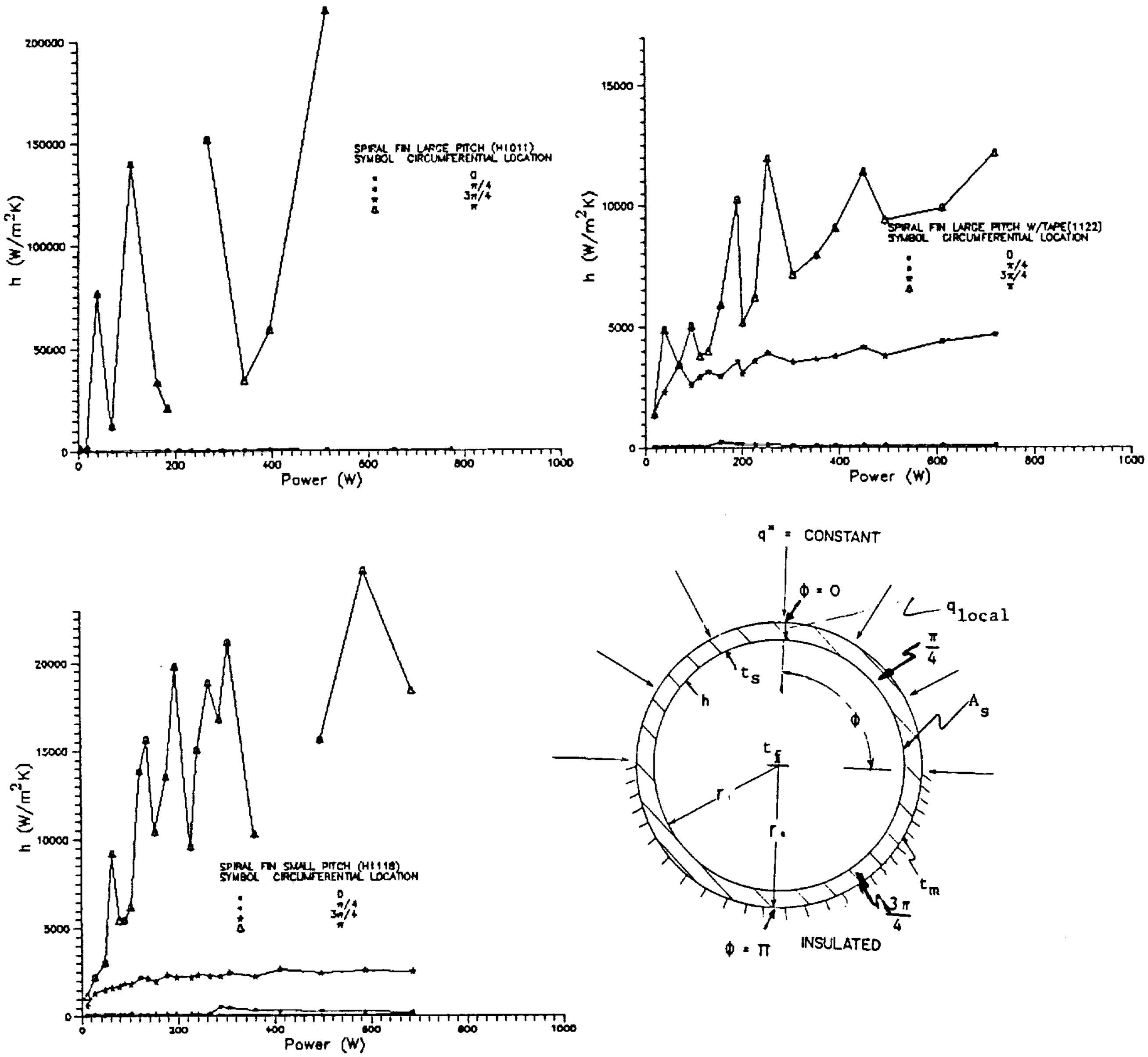

Figure 2. Heat transfer coefficient versus power at axial location $Z = Z_4$ (center of heated section of the test section) for top-heated finned tubes at 0.19 MPa and 0.210 Mg/m^2s. Channel diameter = 1.27 cm, Heated length = 1.22 m.

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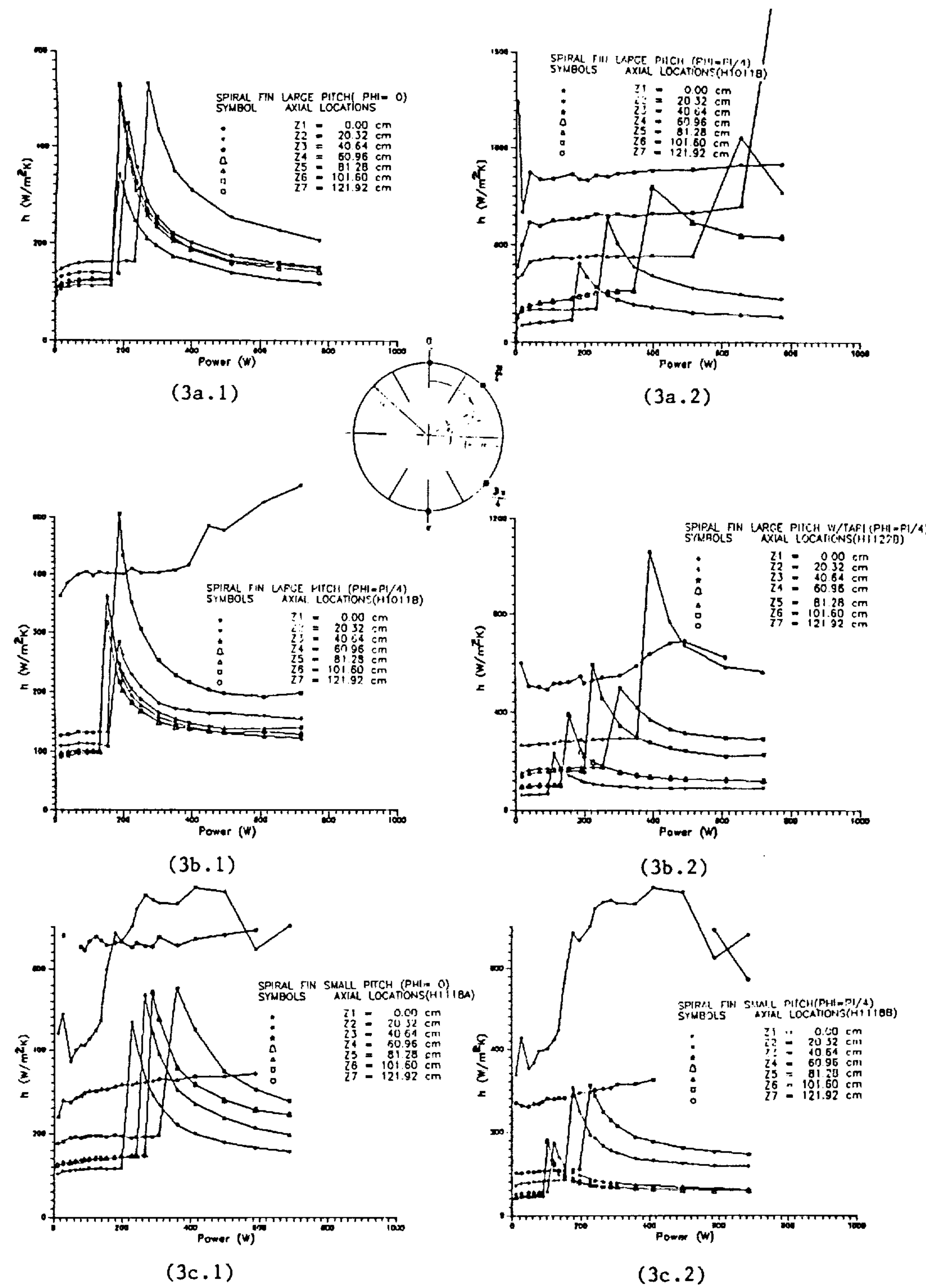

Figure 3. Axial variations of h versus power at circumferential locations 0 and $\frac{\pi}{4}$ for spiral fin large pitch (3a), spiral fin large pitch with tape (3b), and spiral fin small pitch (3c).

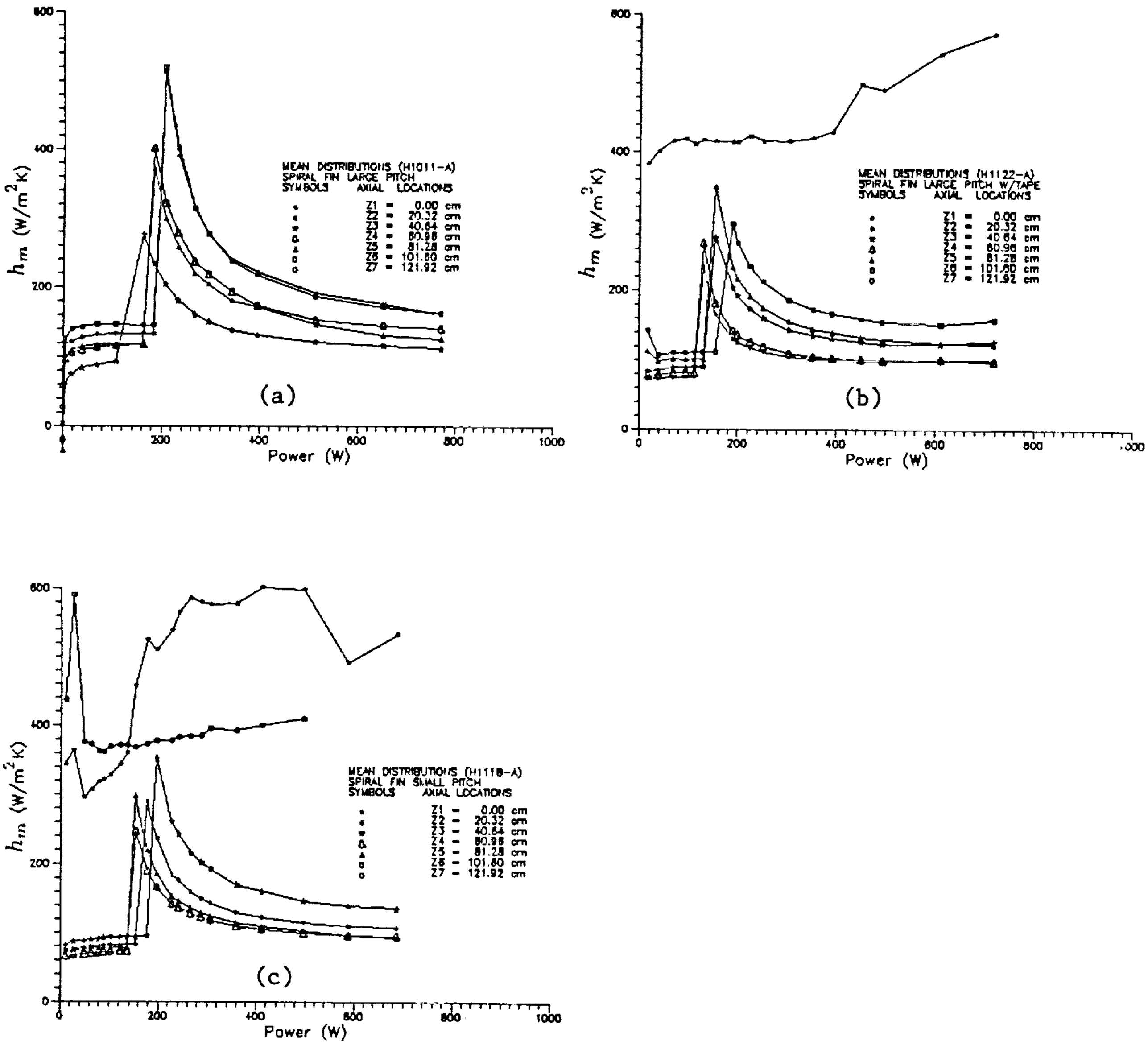

Figure 4. Mean heat transfer coefficient versus power for spiral fin large pitch (4a), spiral fin large pitch with tape (4b), and spiral fin small pitch (4c).

Acknowledgments

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