AD-A244 668

NAVAL POSTGRADUATE SCHOOL
Monterey, California

THESIS

ENHANCED CONDENSATION OF R-113 ON A
SMALL BUNDLE OF HORIZONTAL TUBES

by

ROBERT WALTER MAZZONE

December, 1991

| Co-Advisor | Paul J. Marto |
| Co-Advisor | Stephen B. Memory |

92-01763

Approved for public release; distribution is unlimited

92 1 21 073

Unclassified

security classification of this page

REPORT DOCUMENTATION PAGE			
1a Report Security Classification Unclassified		1b Restrictive Markings	
2a Security Classification Authority		3 Distribution/Availability of Report	
2b Declassification.Downgrading Schedule		Approved for public release; distribution is unlimited.	
4 Performing Organization Report Number(s)		5 Monitoring Organization Report Number(s)	
6a Name of Performing Organization Naval Postgraduate School	6b Office Symbol (If applicable) 34	7a Name of Monitoring Organization Naval Postgraduate School	
6c Address (city, state, and ZIP code) Monterey, CA 93943-5000		7b Address (city, state, and ZIP code) Monterey, CA 93943-5000	
8a Name of Funding/Sponsoring Organization	8b Office Symbol (If applicable)	9 Procurement Instrument Identification Number	
8c Address (city, state, and ZIP code)		10 Source of Funding Numbers	

		10 Source of Funding Numbers			
		Program Element No	Project No	Task No	Work Unit Accession No

11 Title *(include security classification)* ENHANCED CONDENSATION OF R-113 ON A SMALL BUNDLE OF HORIZONTAL TUBES

12 Personal Author(s) ROBERT WALTER MAZZONE

13a Type of Report Master's Thesis	13b Time Covered From To	14 Date of Report (year, month, day) December, 1991	15 Page Count 235

16 Supplementary Notation The views expressed in this thesis are those of the author and do not reflect the official policy or position of the Department of Defense or the U.S. Government.

17 Cosati Codes			18 Subject Terms *(continue on reverse if necessary and identify by block number)*
Field	Group	Subgroup	Heat Transfer, Condensation, R-113, Inundation, Tube Bundle

19 Abstract *(continue on reverse if necessary and identify by block number)*

Condensation of R-113 was studied using an evaporator/condenser test platform. The condenser section contained four horizontal tubes (nominal outer diameter 15.9 mm) forming a vertical in-line column with a pitch-to-diameter ratio of 2.25 and a condensing length of 1.2 m. The condenser tubes could be operated either individually (ie. as a single tube apparatus) or as a small tube bundle. This allowed investigation of the effects of condensate inundation on different types of condenser tubes. Tubes tested were smooth copper tubes, copper/nickel KORODENSE roped tubes, KORODENSE tubes wrapped with wire, and copper/nickel finned tubes (26 fpi). The outside heat transfer coefficient, h_o, was calculated by experimentally determining the overall heat transfer coefficient, U_o, and then using a modified Wilson Plot procedure. Great care was taken to ensure the results were not vitiated by the presence of noncondensibles.

Results obtained with the smooth copper tubes are in agreement with published data and verify satisfactory operation of the test platform. Furthermore, problems associated with the apparatus encountered by previous workers were successfully overcome. In comparison to the top smooth tubes, the copper/nickel top KORODENSE tube yielded about a 22% increase in h_o. When different diameters of wire were wrapped around the copper/nickel KORODENSE tubes, an optimum pitch-to-wire diameter of 7 was found yielding almost a 90% increase in h_o, compared to the top smooth tube. This marked increase is presumably due to surface tension effects thinning the condensate film. The copper/nickel finned tubes gave the best enhancement (approximately a 7 fold increase). Comparison of the data obtained from the top copper/nickel finned tubes agreed well with the model of Beatty and Katz.

The effect of condensate inundation was to reduce h_o for the lower tubes compared to the top tube in the bundle. Comparison of h_o for the second tube compared to the top tube showed that the effects of condensate inundation are reduced most by wrapping the KORODENSE tubes with fine wire.

20 Distribution/Availability of Abstract ☒ unclassified unlimited ☐ same as report ☐ DTIC users	21 Abstract Security Classification Unclassified	
22a Name of Responsible Individual Paul J. Marto	22b Telephone (Include Area code) (408) 646-2768	22c Office Symbol 54Ss

DD FORM 1473,84 MAR   83 APR edition may be used until exhausted
security classification of this page

All other editions are obsolete

Unclassified

ENHANCED CONDENSATION OF R-113 ON A
SMALL BUNDLE OF HORIZONTAL TUBES

by

ROBERT WALTER MAZZONE
Lieutenant, United States Navy
B.A., University of California, San Diego, 1971
Ph.D., State University of New York at Buffalo, 1975

Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING
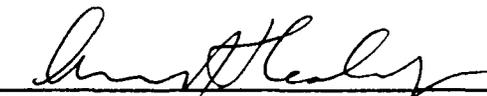
from the

NAVAL POSTGRADUATE SCHOOL
December, 1991

Author: _____
ROBERT WALTER MAZZONE

Approved by: _____
Paul J. Marto, Co-Advisor

_____
Stephen B. Memory, Co-Advisor

_____
Anthony J. Healey, Chairman,
Department of Mechanical Engineering

ABSTRACT

Condensation of R-113 was studied using an evaporator/condenser test platform. The condenser section contained four horizontal tubes (nominal outer diameter 15.9 mm) forming a vertical in-line column with a pitch-to-diameter ratio of 2.25 and a condensing length of 1.2 m. The condenser tubes could be operated either individually (ie. as a single tube apparatus) or as a small tube bundle. This allowed investigation of the effects of condensate inundation on different types of condenser tubes. Tubes tested were smooth copper tubes, copper/nickel KORODENSE roped tubes, KORODENSE tubes wrapped with wire, and copper/nickel finned tubes (26 fpi). The outside heat transfer coefficient, h_o, was calculated by experimentally determining the overall heat transfer coefficient, U_o, and then using a modified Wilson Plot procedure. Great care was taken to ensure the results were not vitiated by the presence of noncondensibles.

Results obtained with the smooth copper tubes are in agreement with published data and verify satisfactory operation of the test platform. Furthermore, problems associated with the apparatus encountered by previous workers were successfully overcome. In comparison to the top smooth tubes, the copper/nickel top KORODENSE tube yielded about a 22% increase in h_o. When different diameters of wire were wrapped around the copper/nickel KORODENSE tubes, an optimum pitch-to-wire diameter of 7 was found yielding almost a 90% increase in h_o, compared to the top smooth tube. This marked increase is presumably due to surface tension effects thinning the condensate film. The copper/nickel finned tubes gave the best enhancement (approximately a 7 fold increase). Comparison of the data obtained from the top copper/nickel finned tubes agreed well with the model of Beatty and Katz.

The effect of condensate inundation was to reduce h_o for the lower tubes compared to the top tube in the bundle. Comparison of h_o for the second tube compared to the top tube showed that the effects of condensate inundation are reduced most by wrapping the KORODENSE tubes with fine wire.

iii

TABLE OF CONTENTS

vi

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

Standard Symbols

A	area, m^2
b	fin spacing, m
C_B	parameter defined in Eq (2-11)
C_p	specific heat, J/kg K
D	diameter, m
e	fin height, m
g	gravitational acceleration, m/s^2
h	heat transfer coefficient, $W/m^2 K$
h_b	heat transfer coefficient in flooded portion of finned tube, $W/m^2 K$
h_h	heat transfer coeffcient in unflooded region of finned tube, $W/m^2 K$
h_f	heat transfer coefficient for fins in unflooded region, $W/m^2 K$
k	thermal conductivity, $W/m K$
L	length, m
$\overline{L}$	average condensing length defined by Eq (2-4)
$\dot{m}$	mass flow rate, kg/s
N	number of tubes in a vertical column
Nu	Nusselt number
Pr	Prandtl number
$\dot{q}$	heat transfer rate, W
q''	heat flux, W/m^2
R	radius, m
R_f	fouling resistance, K/W
R_w	wall resistance, K/W
Re	Reynolds number
Re_f	film Reynolds number
T	temperature, K
t	fin thickness, m
U	overall heat transfer coefficient, $W/m^2 K$

V	velocity, m/s

Greek Symbols

δ	tape thickness, m
η	fin efficiency
ξ	aspect ratio of condensate surface, Eq (2-13)
Θ_m	rotation angle normal to fin surface
μ	dynamic viscosity, kg/m s
ρ	density, kg/m^3
σ	surface tension, N/m
ϕ_f	flooding angle defined in Eq (2-7)

Subscripts

c	coolant
eq	equivalent
ef	effective
f	film
fs	fin sides
ft	fin tips
i	inside or inlet
o	outside or outlet
r	root
sat	saturation

I. INTRODUCTION

A. BACKGROUND

In the last few years, budgetary considerations have led to a reduction in the funding level available for new vessel construction for the US Navy. In construction of ships, weight can often be equated to cost. This has led the Navy to look at various ways by which the weight of auxiliary systems can be reduced.

Of particular interest to the Navy has been the issue of reduced weight and therefore cost of refrigeration systems. Two major components of refrigeration systems are the evaporator and condenser. If the weight of these two components can be reduced, then the overall weight and presumably cost, can also be reduced.

Condenser weight can be significantly reduced by the proper selection of materials for condenser construction. A major weight savings comes from changing the material which is used for construction of the condenser tubes. As an example, the DDG-51 was originally intended to have a refrigeration condenser made of titanium finned tubes. This would provide a substantial weight savings and a reduced cost (presumably). However, the system was not installed on the current flight of ships. Rather, traditional copper/nickel finned tubes were utilized for the refrigeration condenser.[1]

In re-designing refrigeration condensers, changes in condenser tube material based solely on weight consideration should not be the only consideration. The reason for this is that a lightweight material may not have heat transfer characteristics which are comparable to copper or even copper/nickel. Therefore, condenser tube designs have to be developed to maintain or even enhance heat transfer or else the condenser will have to be enlarged. This enlargement is necessary in order to provide the requisite surface area to maintain the required level of heat transfer.

Copper/nickel finned condenser tubes have been used in refrigeration condensers onboard US Navy ships for many years. However, the use of titanium, as was planned for the DDG-51 class ships, holds the promise of providing substantial weight savings (not to mention improved corrosion resistance on the tube side). Unfortunately, titanium has a thermal conductivity which is substantially less than that of copper or copper/nickel. The question which must therefore be addressed is whether titanium

---

[1] According to Mr. John Goodhue, NAVSEA code SEA 56Y15, the reason the units were not installed was that they could not be ready to support proposed construction dates for flight one.

condenser tubes can be constructed in such a manner as to maintain heat transfer (due to enhancement) despite the lower value of thermal conductivity.

A major problem along these lines is how to measure the performance of various types of condenser tubes. Typically, this is done with single tubes or small tube bundles. These tubes tend to be relatively short. Often, the experimental apparatus bares little resemblance to a shipboard condenser. Thus, any results become suspect when they are extended to the real situation.

In order to examine heat transfer characteristics of various designs of condenser tubes, the best means is to construct a test platform composed of an evaporator and condenser with a size comparable to that of a shipboard system. Unfortunately, this is seldom feasible due to costs involved and the size of the plants. However, it is possible to construct a reasonably large scale test platform in which tube performance can be evaluated. This evaluation must include both single tube performance as well as tube bundle operation. The reason for this is that condensate forming on one tube falls onto lower tubes. This effect, known as condensate inundation, can result in decreased heat transfer on lower tubes due to a thickening of the condensate layer. Hence, another aspect which must be considered is whether condenser tube geometry can be changed in such a manner as to reduce the adverse effects of condensate inundation. This could help make up for decreased thermal conductivity associated with lightweight materials such as titanium.

The construction of a suitable condenser/evaporator test platform was begun at the Naval Postgraduate School by Zebrowski [Ref. 1]. The construction and initial operation of the test platform was completed by Mabrey [Ref. 2] in 1988. Mabrey conducted experiments using R-113 and R-114 on smooth copper tubes as well as copper/nickel finned tubes (26 fpi). During the course of his experiments, an orange-peel or dimpled appearance was observed on the smooth copper tubes. Additionally, the copper/nickel finned tubes showed a buildup of "material" between the fins. This material was believed to be ice although this was not proved. The cause of the orange-peel appearance was also never identified.

B. OBJECTIVES.

This thesis concentrates on the condenser section of the test platform with specific interest in the condensation of R-113. R-113 was chosen for this study because of convienence , ie., it is a liquid at atmospheric pressure. The condenser section consists of four horizontal tubes mounted in a vertical in-line column. The nominal diameter of

the tubes is 16 mm with a condensing length of 1.2 m. While the condensing medium used in this study was R-113, the condensation of many different types of fluids can be evaluated. In addition, the condenser can be used to study heat transfer on almost any type of condenser tube subject to the nominal outer diameter and condensing length restrictions indicated.

The main objective of this thesis was to resolve the contamination problem found by Mabrey [Ref. 2], and to make the test platform operational. It was felt that such a system, once operational, could be used to study heat transfer using condenser tubes made of various materials and geometries. In addition, since the tubes are arranged in a vertical in-line column, the effect of condensate inundation could also be studied.

C. SPECIFIC OBJECTIVES

The specific objectives of this study were as follows:

1. Resolve the contamination problem reported by Mabrey [Ref. 2]

2. Establish baseline condensation data by examining the condensation of R-113 on smooth copper tubes.

3. Examine the heat transfer characteristics of R-113 on various tube geometries

4. Investigate the effects of condensate inundation

II. LITERATURE SURVEY

A. INTRODUCTION

Condensation occurs when vapor molecules come into contact with a surface which has a temperature below the vapor saturation temperature. This vapor, coming into contact with the cooler surface, forms a liquid which can accumulate in two ways. If the liquid wets the entire surface, a continuous film of liquid is formed. On the other hand, if the liquid fails to wet the entire surface, droplets are formed. These two types of condensation are referred to as filmwise and dropwise condensation respectively. Dropwise condensation yields larger heat transfer coefficients but is difficult to maintain [Ref. 3]. All condensers today are therefore designed to operate with film condensation conditions.

In 1916, Nusselt [Ref. 4] reasoned that as vapor condenses on a cooler surface, such as a horizontal tube, a continuous film of condensate will form. This film grows in thickness as the condensate flows around the tube under the influence of gravity. Hence, the thickness of the condensate is not uniform around a horizontal tube. Since the condensate acts as a resistance to heat transfer, Nusselt argued that the local heat transfer coefficient will be a maximum at the top of the tube where the film thickness is least and decreases around the tube as the condensate thickness increases.

B. CONDENSATION OF REFRIGERANTS ON SMOOTH TUBES

Goto et al. [Ref. 5] examined film condensation of various refrigerants, including R-113, on smooth copper tubes. Their results showed values for the vapor-side heat transfer coefficient on the order of 1.2 to 1.5 $kW/m^2 K$ for a wall-film temperature difference of 3 to 25 K. Furthermore, their results were about 10% below that predicted by Nusselt theory.

Masuda and Rose [Ref. 6,7] reported that condensation of R-113 on smooth copper tubes yielded an outside heat transfer coefficient of about 1500 $W/m^2 K$ for wall-film temperature differences of 12 to 20 K. Their results showed good agreement with Nusselt theory.

Marto et al. [Ref. 8] showed that for condensation of R-113 on smooth copper tubes, the vapor-side heat transfer coefficient was on the order of 1000 $W/m^2 K$ for a temperature drop across the film of 20 to 25 K. Their values were also in good agreement with Nusselt theory.

4

C. ENHANCEMENT OF FILM CONDENSATION

The most commonly used procedure for enhancement of film condensation is through the use of integral-fin tubes. With this type of condenser tube, two distinct regions exist; namely, a flooded and an unflooded region. Typically, as fluid condenses on a finned surface, it is pulled from the fin tip to the fin root by surface tension forces. Gravity then causes the fluid between fins to drain to the bottom of the tube where it drains off. Hence, condensation on a finned tube involves a combination of surface tension and gravity forces. It is the added surface tension which tends to enhance heat transfer over smooth tubes by thinning the condensate film at the fin tip.

Since the present study uses R-113 as the working fluid, the remainder of this chapter will focus on previous experiments which have used low surface tension fluids such as refrigerants. For additional information involving other fluids, the reader is referred to recent review articles by Marto [Ref. 9,10], Webb [Ref. 11] and Sukhatme [Ref. 12].

1. Condensation on Single Enhanced Tubes: Theoretical Models

The first theoretical model of condensation on finned tubes was published by Beatty and Katz [Ref. 13]. Their theoretical model is given by:

$$h_{BK} = \eta_f h_f \frac{A_f}{A_{ef}} + h_u \frac{A_u}{A_{ef}} \qquad (2-1)$$

Here η_f is the fin efficiency, A_f is the fin surface area, A_u is the unfinned surface area, and A_{ef} is the effective total surface area. The above equation can be simplified to:

$$h_{BK} = 0.689 \left[\frac{k_f^3 \rho_f^2 g h_{fg}}{\mu_f \Delta T D_{eq}} \right]^{\frac{1}{4}} \qquad (2-2)$$

where the equivalent diameter is given by:

$$\left(\frac{1}{D_{eq}} \right)^{\frac{1}{4}} = 1.30 \eta_f \frac{A_f}{A_{ef}} \frac{1}{\overline{L}^{1/4}} + \frac{A_o}{A_{ef}} \frac{1}{D_r^{1/4}} \qquad (2-3)$$

In this equation,

$$\overline{L} = \frac{\pi \left(D_o^2 - D_r^2 \right)}{4 D_o} \qquad (2-4)$$

5

and

$$A_{ef} = \eta_f A_f + A_u \qquad (2-5)$$

The equation for the heat transfer coefficient is a Nusselt-type condensation equation based on the equivalent diameter for a finned tube. This equation predicted the experimental results obtained by Beatty and Katz to within $\pm$ 20%.

There are several important limitations to the Beatty and Katz model [Ref. 13]. It does not distinguish between fin tips and fin flanks, essentially assuming that the fins have no thickness. Therefore the model is not suitable for trapezoidal fins. A second, and perhaps more significant limitation is that the Beatty and Katz model fails to take into account surface tension effects. Instead, the model assumes that the flow of the condensate film is simply a function of gravity. While this may not pose a serious limitation when low surface tension fluids such as refrigerants are examined, this assumption can become critical when condensation of high surface tension fluids such as steam are investigated.

Another limitation of the Beatty and Katz model is that it does not take into account condensate flooding. Rudy and Webb [Ref. 14] argued that the flooded portion of the tube was ineffective in condensation. They therefore proposed that the Beatty and Katz model be modified by:

$$h_{RW} = h_{BK} \frac{\phi_f}{\pi} \qquad (2-6)$$

where ϕ_f, the flooding angle, is defined as:

$$\phi_f = \cos^{-1}\left[\left[\frac{2\sigma \cos \psi}{\rho g b R_o}\right] - 1\right] \qquad (2-7)$$

where R_o is the outside radius, ψ is the fin tip half angle and σ is the fluid surface tension. It is interesting to note that this modification resulted in an underprediction of Rudy and Webb's R-11 condensation data by about 35%.

Webb et al. [Ref. 15] next argued that surface tension drainage must be considered in the model. They derived the equation:

$$h_f = 0.943\left[\frac{k_f^3 \rho h_{fg}}{\mu_f \Delta T}\right]^{\frac{1}{4}}\left[\frac{2\sigma}{e^2}\left(\frac{1}{t} + \frac{1}{b}\right)\right]^{\frac{1}{4}} \qquad (2-8)$$

6

where e is the fin height, t is the fin thickness and b is the width of the space between the fins. Substitution into the Beatty and Katz expression yielded agreement with experimental data of ± 10%.

Owen et al. [Ref. 16] argued that the Rudy and Webb [Ref. 15] model had to be corrected in order to account for heat transfer in the flooded part of the tube. Owen et al. proposed the equation:

$$h_o = h_{BK}\frac{\phi_f}{\pi} + h_b\left(1 - \frac{\phi_f}{\pi}\right) \qquad (2-9)$$

where h_o is the average outside heat transfer coefficient and h_b is the heat transfer coefficient in the flooded region of the tube. This value was estimated using a one-dimensional radial conduction model. This corrected model of Owen et al. resulted in agreement of ± 30% with earlier experimental data.

In 1985, Webb et al. [Ref. 17] proposed a model that included heat transfer in the flooded region as well as surface tension drainage on the fin flanks and gravity drainage on the interfin surface. Their model equation for the average heat transfer coefficient is:

$$h_W = (1 - C_B)\left[h_h\frac{A_r}{A} + h_f\eta_f\frac{A_f}{A}\right] + C_B h_B \qquad (2-10)$$

In this expression, h_h is the heat transfer coefficient for the unflooded root surface between the fins and h_f is the heat transfer coefficient on the fin flanks in the unflooded area, and h_b is the heat transfer coefficient in the condensate flooded region. The term C_B is the fraction of finned tube surface flooded by condensate, or,

$$C_B = 1 - \frac{\phi_f}{\pi} \qquad (2-11)$$

where ϕ_f is the flooding angle given in equation (2-7).

In the Webb et al. [Ref. 17] expression for the average heat transfer coefficient, h_h, h_f and h_b are defined mathematically as follows:

$$h_h = 1.514\left[\frac{\mu_f^2}{k_f^2\rho_f^2 g} Re_f\right]^{-\frac{1}{3}} \qquad (2-12)$$

where Re_f is the condensate film Reynolds number.

7

$$h_f = 2.149 \frac{k_f}{S_m} \left[\frac{\sigma \rho_f h_{fg} \Theta_m S_m}{\mu_f k_f \Delta T} \frac{(\xi + 1)}{(\xi + 2)^3} \right]^{\frac{1}{4}} \qquad (2-13)$$

where S_m is the length of the convex surface and Θ_m is the rotation angle normal to the fin surface [Ref. 18]. The term ξ is a parameter that characterizes the aspect ratio of the condensate surface.

Finally, h_b is defined as:

$$h_b = \frac{\phi k}{e} \qquad (2-14)$$

This complex model of Webb et al. was able to predict heat transfer coefficients to within $\pm$ 20% of experimental data obtained with R-11.

Honda and Nozu [Ref. 19] and Honda et al. [Ref. 20] have proposed the most comprehensive model to date. This model treats the heat transfer problem as one involving vapor-to-coolant heat transfer through a finned tube wall. In this model, trapezoidal fins were divided into flooded (f) and unflooded (u) regions. Equation (2-7) was used to determine the flooding angle. Condensate flow in between the fins was assumed to be driven by gravity whereas for the fin surface, condensate flow was assumed to be driven by both surface tension forces and gravity. The condensate film thickness on the fin surface was arbitrarily divided into a thick region, where no heat transfer occurred, and a thin film region. The equation of the condensate film thickness was solved numerically.

Honda and Nozu [Ref. 19] showed that the average Nusselt number could be written in terms of relevant parameters for the flooded (f) and unflooded (u) regions as:

$$Nu_d = \frac{hd_o}{k_l} = \frac{Nu_{du}\eta_u(1 - \tilde{T}_{wu})\tilde{\phi}_f + Nu_{df}\eta_f(1 - \tilde{T}_{wf})(1 - \tilde{\phi}_f)}{(1 - \tilde{T}_{wu})\tilde{\phi}_f + (1 - \tilde{T}_{wf})(1 - \tilde{\phi}_f)} \qquad (2-15)$$

Here, $\tilde{T}_{wu}$ and $\tilde{T}_{wf}$ are the dimensionless average wall temperatures at the fin root and $\tilde{\phi}_f$ is defined as:

$$\tilde{\phi}_f = \frac{\phi_f}{\pi} \qquad (2-16)$$

It should be pointed out that $\bar{T}_{wu}$ and $\bar{T}_{wf}$ are determined by solving a circumferential wall heat conduction equation. This wall temperature variation must be considered since heat transfer rates through flooded and unflooded regions are very different.

Honda and Nozu [Ref. 19] used their model to predict average heat transfer coefficients. These predictions were compared to experimental data on eleven different fluids and twenty-two tubes. The predicted values matched most experimental values to within ± 20%.

2. Condensation on Single Enhanced Tubes: Experimental Results

The first experiments conducted on enhanced tubes were performed by Katz and colleagues [Ref. 21,22, 13]. These authors looked at condensation of R-12 and R-22 on smooth and finned tubes. Their results showed a significant enhancement in heat transfer for finned tubes.

In 1971, Karkhu and Borovkov [Ref. 23] examined condensation of R-113 on five finned tubes. Their results showed that the rate of condensation for these tubes with trapezoidal fins could be increased by 50 to 100% over smooth tubes.

Arai et al. [Ref. 24] and Kisaragi et al. [Ref. 25] studied the condensation of R-12 and R-113 respectively on finned tubes with spiral grooves cut along the length of the tube. Arai et al. showed that grooves with a pitch of 0.7 mm gave the best enhancement in heat transfer. Kisaragi and co-workers also compared grooved finned tubes with tubes on which the fins had been covered with a porous coating. In comparison to the finned tube, a two fold enhancement was seen with the grooved finned tube while a four fold enhancement was obtained with the finned porous coated tube.

Carnavos [Ref. 26] performed one of the first detailed studies of condensation of R-11 on enhanced copper tubes. In this study, eleven different tubes were tested. The tubes included finned tubes as well as finned tubes which had an axial spiral groove cut along their length. Various sizes and shapes of fins were included. Enhancements of 4 to 6 were seen when compared to smooth tubes.

Rudy [Ref. 27] and Webb et al. [Ref. 15] reported experimental data obtained for R-11 condensing on copper tubes with different fin geometries. Their results showed that increasing the fin density from 748 to 1378 fpm resulted in about a 70% increase in heat transfer presumably due to an increased surface area.

Honda et al. [Ref. 28] studied condensation of R-113 on copper finned tubes of various fin density. Their results showed a 6 to 9 fold enhancement for R-113 condensation for finned tubes when compared to smooth tubes. A similar study by Kabov [Ref. 29] examined the condensation of R-12 and R-22 on tubes with different fin

densities His results also showed that the degree of enhancement is determined by fin spacing.

Masuda and Rose [Ref. 6] examined heat transfer for the condensation of R-113 on low integral-fin copper tubes. Fourteen different fin spacings (0.25 to 20 mm) were examined. The optimal fin spacing was 0.5 mm which resulted in a 7.3 fold enhancement in the vapor-side heat transfer coefficient.

Lin and Berghmans [Ref. 30] examined condensation of R-11, R-113, and R-114 on an enhanced tube. This enhanced tube, known commercially as "EVERFIN", is made of trapezoidal fins (1378 fpm) with a spacing between the fins of 0.064 mm.[2] This tube gave an 8 to 10 fold increase in the vapor-side heat transfer coefficient when compared to a smooth tube.

Condensation of R-11 on a variety of enhanced copper tubes has been reported recently by Sukhatme et al. [Ref. 31]. All tubes had trapezoidal shaped integral fins. The fin density, fin height and fin-tip half angle were systematically varied in order to determine the optimal values which would result in maximum enhancement. The maximum enhancement obtained by these authors was about 10.3 relative to smooth tubes. This enhancement corresponded to a fin spacing of 0.35 mm, a fin height of 1.22 mm and a fin-tip half angle of 10 degrees. In addition, a set of specially enhanced tubes were tested by these authors. These tubes had axial grooves cut into them with the trapezoidal fins present. An enhancement of 12.3 was obtained with these specially enhanced tubes when compared to smooth tubes.

Marto et al. [Ref. 8] studied condensation of R-113 on a variety of copper integral-fin tubes with rectangularly shaped fins. The optimum fin spacing was found to lie between 0.25 and 0.5 mm. The vapor-side heat transfer coefficient was enhanced on the order of 4 to 7 fold.

Michael et al. [Ref. 32] continued the studies of Marto et al. [Ref. 8] by looking at the effect of root diameter on the condensation of R-113 on integral-fin tubes. They found that the root diameter had only a negligible effect on heat transfer enhancement. In addition, these investigators found that the optimal fin spacing was about 0.5 mm. They obtained an enhancement of 5 to 6.

---

2 While the authors quote a fin spacing of 0.064 m, this value seems too small. The schematic diagram of the fins together with other dimensions published by these authors tend to suggest that the correct fin spacing is 0.64 mm. This value is more in line with values quoted by other authors.

In comparing enhancements reported by various authors, a word of caution is required. In calculating values for the vapor-side heat transfer coefficients, the surface area to be used in this calculation must be defined. This area can be based on outside or envelope diameter, root diameter or total fin area. The great majority of authors define this area by the outside diameter of the tube as opposed to the root diameter. In the various sources cited above, this has been the case. That is, the surface area has been defined by the outside tube diameter.

While it is clear that integral-fin tubes can result in an enhancement of the vapor-side heat transfer coefficient, other alternatives exist. As an example, Fujii et al. [Ref. 33] studied the condensation of R-11 on smooth tubes which were wrapped with fine wire. These authors found an optimal pitch-to-wire diameter ratio of 2 which resulted in an enhancement of 2.5 in the vapor-side heat transfer coefficient when compared to copper smooth tubes. These authors argued that the enhancement in the heat transfer coefficient was due to a thinning of the condensate film. The reason that the condensate film thickness is reduced is that the presence of the wire creates a low pressure region in the film near the base of the wires. This low pressure region is due to the concave shape of the condensate film next to the wire. As a result, condensate film is pulled along the tube wall towards the wire. This results in a thinning of the condensate film between the wires.

3. Summary of Single Enhanced Tube Condensation

It is clear from the preceding discussion that theoretical models, despite their degree of sophistication, still only match experimental data to within ± 20%. This suggests that the process of condensation on enhanced tubes is extremely complex and far from being fully understood.

In terms of experimental data involving condensation of refrigerants on enhanced tubes, several comments need to be made. The experiments reported here have shown that integral-fin tubes can give enhancements of the vapor-side heat transfer coefficient on the order of 7 to 12. The magnitude of this enhancement depends upon such factors as fin shape, fin spacing, and fin height. Trapezoidal shaped fins with a spacing of 0.5 mm give the highest enhancements.

Unfortunately, the above experiments were all performed on enhanced tubes made of copper. In Naval refrigeration systems, where seawater is used as the coolant, corrosion is of major concern. This necessitates the need for copper nickel or titanium condenser tubes in order to minimize corrosion. However, as can be seen from the foregoing summary of the literature, essentially no work has been done on the

condensation of refrigerants on copper/nickel or titanium finned tubes. Since both copper/nickel and titanium condenser tubes have a lower thermal conductivity than copper, one must wonder what effect this will have on the vapor-side heat transfer co-efficient. Clearly, experiments using these tubes must be carried out.

4. Condensation on Enhanced Tube Bundles: Theoretical Studies

Condensation in tube bundles presents an entirely different situation than condensation on single tubes. In a tube bundle, two conflicting factors come into play. First, condensate forming on an upper tube drains off this tube onto lower tubes within the bundle. This increases the thickness of the condensate layer on lower tubes which further increases the resistance of these tubes to heat transfer. This effect is referred to as condensate inundation.

Condensate falling from one tube to another can have a second effect. As droplets fall from one tube to the next, ripples and splashing (turbulence) are induced in the condensate film. This can actually enhance heat transfer on the lower tubes.

Vapor velocity can also be a significant factor in tube bundle condensation. As the vapor travels into the tube bundle and condenses, the local velocity decreases. This decrease in local velocity leads to a decrease in the heat transfer coefficient similar to an inundation effect since vapor shear effects that thin the condensate film are reduced.

In analyzing condensation in horizontal tube bundles, Nusselt [Ref. 4] assumed that all condensate flowed from one tube to another as a continuous laminar sheet. He showed that the ratio of the average heat transfer coefficient for a vertical column of horizontal tubes with respect to the top tube could be expressed as:

$$\frac{\bar{h}_N}{h_1} = N^{-\frac{1}{4}}$$

(2 – 17)

where N is the number of tubes in the vertical column. A local coefficient can also be calculated for the Nth tube using Nusselt theory. This coefficient is given by:

$$\frac{h_N}{h_1} = N^{\frac{3}{4}} - (N-1)^{\frac{3}{4}}$$

(2 – 18)

Kern [Ref. 34] proposed a less conservative approach. He argued that droplets of condensate falling from one tube to another causes ripples which disturb the condensate film, thereby diminishing the effect of condensate inundation. His mathematical model took the form:

12

$$\frac{\bar{h}_N}{h_1} = N^{-\frac{1}{6}} \qquad (2-19)$$

The local value of h for the Nth tube can be expressed as:

$$\frac{h_N}{h_1} = N^{\frac{5}{6}} - (N-1)^{\frac{5}{6}} \qquad (2-20)$$

In 1972, Eissenberg [Ref. 35] proposed that condensate may not drain only in a vertical direction, but rather, may be diverted sideways, presumably due to local vapor flow conditions. This so-called "side drainage" model predicts a less severe effect of inundation yields:

$$\frac{\bar{h}_N}{h_1} = 0.60 + N^{-\frac{1}{4}} \qquad (2-21)$$

At the present time, numerous investigations have been carried out in an attempt to better understand the phenomenon of condensate inundation. However, the data are scattered and as a consequence, no successful theoretical model of condensate inundation exists [Ref. 3]. However, for design purposes, Butterworth [Ref. 36] has recommended that the Kern model be used.

5. Condensation in Enhanced Tube Bundles: Experimental Studies

Young and Wohlenberg [Ref. 37] examined the condensation of R-12 on a vertical column of smooth horizontal tubes. These investigators found that the effect of the number of tubes in the vertical column is less than that predicted by Nusselt.

Katz et al. [Ref. 38] investigated the condensation of R-12 on a vertical column of six horizontal finned tubes. Their results also showed a less severe inundation effect than predicted by the model of Nusselt.

Smirnov and Lukanov [Ref. 39] used R-11 to study condensation on 20 horizontal rows of finned tubes arranged in a triangular fashion. Their results showed a dramatic reduction in heat transfer for the first five tubes. However, after this, the heat transfer coefficient remained almost constant for the rest of the tubes in the column.

Honda et al. [Ref. 40] have studied the condensation of R-113 on a vertical column of horizontal low finned tubes. An inundation tube was used to simulate additional rows of tubes. These authors described four modes of condensate flow; namely, droplet mode, column mode, column and sheet mode, and sheet mode. Unfortunately,

13

these authors did not calculate the heat transfer coefficient so that no comparison with theoretical models is possible.

Webb and Murawski [Ref. 41] examined the condensation of R-11 on a column of enhanced condenser tubes. The tubes tested were a standard 1024 fpi tube, a GEWA-SC tube, a Turbo-C tube and a TRED-D tube. The GEWA-SC tube is a 1024 fpi tube but the fins have a Y-shaped cross-section. The Turbo_c and TRED-D tubes are similar to the 1024 fpi tube except that the fins have a saw-tooth shape. Webb and Murawski found that a standard 1024 fpm tube showed virtually no row (inundation) effect. However, when the experiments were conducted on Turbo-C, GEWA-SC and Tred-D tubes, row effects (ie., inundation effects) were seen. The inundation effect was greatest for the Tred-D tube which also had the lowest single tube performance. These investigators found values for the exponent in equations (2-17, 2-19, and 2-21) on the order of 0.12 to 0.26. Recall that Nusselt's theory used a value of 0.25.

More recently, Honda et al. [Ref. 42] studied the condensation of R-113 on an in-line horizontal tube bundle. The tubes tested consisted of either flat-sided (rectangular) annular fins or three dimensional trapezoidal fins. These authors found that the effects of condensate inundation were greatest for the three dimensional fin tubes. Unfortunately, these authors also had vapor velocities on the order of 3.4 m/s or greater. Therefore, it is difficult to compare their results directly to the theoretical models of Nusselt, Kern, or Eissenberg.

An interesting facet of the experiments of Honda et al. [Ref. 42] deserves mention. These investigators compared the effects of condensate inundation on the vapor-side heat transfer coefficient for smooth and finned copper tubes. Their results showed that despite condensate inundation, the heat transfer coefficient for the finned tubes was still substantially above that of the unfinned tubes. In addition, the decrease in the heat transfer coefficient with increasing row number was less with the finned tubes suggesting that the effects of condensate inundation were less severe with finned tubes than with unfinned tubes. Presumably, this result is due to the fact that condensate draining off of a finned tube drains in columns rather than continuous sheets. Lower tubes will therefore not be completely covered with condensate but instead will have unflooded regions.

Honda and Nozu [Ref. 43] examined the effects of condensate inundation on smooth and integral-fin copper tubes using a theoretical model derived recently by Honda et al. [Ref. 44] This model was a single tube model which was extended to include the effects of condensate inundation by realizing that condensate did not flow off tubes

as sheets but rather as columns. The model took account of the fact that regions which received drainage of condensate had different heat transfer coefficients than regions which did not. The results support the more recent experimental results of Honda et al. [Ref. 42]. The effect of inundation was less severe for the finned tubes than for the smooth tubes. The authors suggest that this is due to condensate being channeled on the finned tubes whereas for the smooth tubes, the entire tube surface becomes covered with a condensate film of varying thickness.

6. Summary of Enhanced Tube Bundle Condensation

Experimental data has shown that the Nusselt model for the effects of condensate inundation may be too conservative. Rather, most experimental data seems to fall between the Kern and Eissenberg predictions.

Other experimental data suggests that the effects of condensate inundation are less for finned tubes as opposed to smooth tubes. Presumably, this is due to the manner in which condensate flows from one finned tube to another. The presence of the seems acts to channel the flow of condensate leaving large areas relatively free of flooding from inundation.

As a final comment, the use of wire wrapped tubes provides an interesting alternative to the more conservative finned tube approach. However, the effects of refrigerant condensate inundation on wire wrapped tubes has yet to be examined.

III. EXPERIMENTAL APPARATUS

A. TEST PLATFORM

Details of the design and construction of the condenser/evaporator test platform are provided by Zebrowski [Ref. 1] and Mabrey [Ref. 2]. Therefore, only a brief summary will be presented here.

The test platform is shown schematically in Figure 1 while a photograph of the unit is shown in Figure 2. The unit is made from rolled stainless steel plate (6.35 mm thick) and designed to withstand an absolute pressure of 308 kPa (approximately 3 bar).

1. Condenser Section

The top section is the condenser. It is 1.3 m in length with an internal diameter of 0.61 m. The end plates are made of stainless steel plate (6.35 mm thickness) with a diameter of 0.71 m. They are connected to the condenser shell using a bolted flange assembly. A rubber gasket, 3.22 mm thick, provides a vacuum tight seal. The condenser unit has five viewports which allow the condensation process to be viewed.

Four test condenser tubes are contained in the condenser forming a vertical in-line column with a pitch-to-diameter ratio of 2.25. These are held in place at each end by a nylon block bolted to the end plate. A rubber gasket provides a vacuum tight seal. A stainless steel plate is fitted over the ends of the condenser tubes and bolted to the nylon block. O-rings are placed in between the nylon block and steel plate in order to provide a vacuum tight seal.

In order to provide a means of pressure control during testing of the condenser tubes, the test platform was fitted with five auxiliary condenser tubes. These auxiliary condenser tubes are constructed of 9.53 mm diameter copper tube, wound into coils. Each coil is supported by a stainless steel rod welded to one of the end plates. The auxiliary condenser coils enter and exit through this same condenser end plate.

A stainless steel shroud assembly, shown in Figure 3, is installed in the condenser unit. This shroud surrounds the test condenser tubes as well as the auxiliary condenser coils. A window, installed in the side of the shroud, allows visualization of the test condenser tubes through the viewports. The purpose of this shroud is to funnel vapor from the evaporator below, around the outside of the shroud and then vertically down past the condenser tubes. The velocity of the vapor past the condenser tubes was 0.1 m/s.

THERMOCOUPLE # 1   THERMOCOUPLE # 2

INSTRUMENTED
CONDENSER
TUBES

127 mm VIEWPORTS
1.22 m

INSTRUMENTED
CONDENSER
TUBES

AUXILIARY
CONDENSER →
COILS

VIEWPORTS

0.6 m DIA (INSIDE)

203mm

114 mm

THERMOCOUPLE # 4 & 5

102 mm VIEWPORT
203mm OUTER SUPPORT RING

AUXILIARY HEATERS

16mm INSTRUMENTED HEATER TUBES

203 mm VIEWPORT
305 mm OUTER SUPPORT RING

216 mm

16 mm SIMULATION HEATER TUBES

Figure 1.   Schematic diagram of test platform.

17

Figure 2. Photograph of the evaporator/condenser test platform.
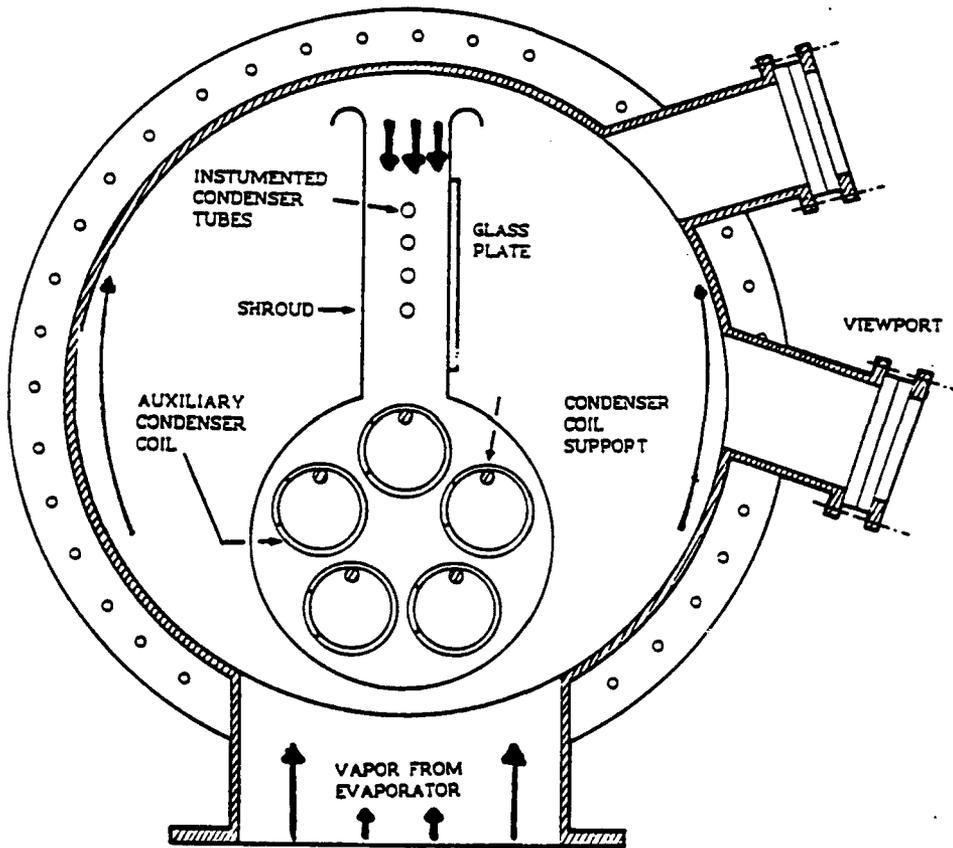
Figure 3. Schematic diagram of condenser showing shroud.

2. Evaporator Section

The flooded evaporator is formed from rolled stainless steel and has an internal diameter of 0.61 m with a width of 0.279 m. Two viewports allow visualization of the boiling process. This unit is connected to the condenser by a collar assembly 0.28 m in diameter and 0.203 m in length. The volume of liquid contained in the evaporator is approximately 0.06 m^3, just sufficient to cover the boiler tubes.

Vapor is produced in the evaporator using a combination of three groups of heaters. The first group is the main bundle and is composed of thirty-five tubes in a triangular pitch (pitch to diameter ratio of 1.2). Fifteen of these have heaters with a nominal rating of 1 kW each. Five of these heated tubes are instrumented with thermocouples to enable bundle boiling data to be obtained. In addition, unheated dummy tubes are used to guide the two phase flow within the bundle.

The second group of tubes is composed of five simulation heaters, each with a nominal rating of 4 kW. These are placed below the main bundle in order to simulate further rows of tubes.

The final group consists of four auxiliary heaters, each with a nominal rating of 4 kW. Two of these are placed on each side of the evaporator in order to keep the liquid pool at saturation conditions; this avoids any subcooling of the liquid. Power to all three sets of heaters is provided by three separate STACO 240V, 23.5kVA rheostat contollers.

For further information on the design and construction of the test platform, the reader is referred to Zebrowski [Ref. 1], Mabrey [Ref. 2], Murphy [Ref. 45], and Chilman [Ref. 46].

3. Coolant System

The coolant is a 54/46 by weight mixture of ethylene glycol and water. The coolant system consists of two main coolant pumps. These pumps are 0.5 hp constant speed pumps which take a suction from the main coolant sump. One pump is used to supply five rotameters. A ball valve, located at the entrance to each rotameter, is used to control the flow rate. Flow to each test condenser tube is supplied by an individual rotameter. The fifth rotameter is used to supply one of the auxiliary condenser coils. A needle valve is used to provide fine control of flow through the coil. This coil is normally used to control saturation pressure during experimental runs.

The second coolant pump is used to supply coolant flow to the remaining four auxiliary condenser coils. Flow through each coil is controlled by a needle valve located at the entrance to each coil. It should be noted that the four auxiliary cooling coils are

seldom needed during the condenser experiments; their primary purpose is for use during evaporator experiments. (see Chilman [Ref. 46]).

In addition to supplying the four auxiliary condenser coils, the second coolant pump also provides coolant flow to a refrigerant storage tank and to a cold trap located at the top of the condenser. The storage tank can be used to store refrigerant during condenser or evaporator tube changeout. The cold trap is placed in-line between the condenser and a vacuum pump. Vapor from the condenser is drawn through the cold trap which is cooled by the ethylene glycol/water mixture. This process is used to remove noncondensibles from the condenser while preventing loss of refrigerant to the atmosphere.

Flow of the coolant mixture from the auxiliary condenser cooling coils, storage tank, and cold trap are returned to the sump tank. Coolant flow from the test condenser tubes empties into a common header which then empties into the same coolant sump tank.

In order to keep the coolant mixture temperature approximately constant, it is cooled by an eight ton refrigeration system located outside the building. This system continually recirculates the coolant using a 0.75 hp pump. Coolant temperature can be maintained at any temperature from ambient temperature (approximately 20 °C) down to -20 °C.

4. Instrumentation

Prior to entering each condenser tube, the coolant flows through a mixing chamber. The inlet temperature is measured using a single copper/constantan type-T thermocouple. Coolant flows from each condenser tube through an outlet mixing chamber. Outlet coolant temperatures are then measured using two thermocouples for each tube. An average outlet temperature is then taken for each tube. All thermocouples had been previously calibrated by Mabrey [Ref. 2].

Refrigerant temperatures were measured at several different locations in the evaporator/condenser test platform using similar single thermocouples. Vapor temperature was measured in two separate locations at the top of the condenser and one location just above the liquid level. Refrigerant liquid temperature was measured at both the top and bottom of the tube bundle. The thermocouples were inserted into stainless steel housings (with copper tips) which were installed in the condenser/evaporator test platform.

B. CLEANING AND DECONTAMINATION PROCEDURES

As mentioned earlier, there was a contamination problem with the test platform during the work of Mabrey [Ref. 2]. Therefore, the experimental apparatus was initially disassembled and cleaned in accordance with the following procedure.

Prior to removing the evaporator, all heaters and thermocouples were disconnected. The lagging around the mating flange was then removed. A Porta-lift was connected to the evaporator mating flange in order to support the weight of the evaporator prior to removing the bolts holding the evaporator and condenser together. After supporting the evaporator, the bolts were removed and the evaporator lowered from the condenser unit.

Referring to Figure 2, the condenser unit is capped at each end by a steel end plate. The left end plate supports the auxiliary condenser units and is therefore removed after the right end plate. The test condenser tubes must be removed prior to the disassembly of the condenser. The assembly which holds the condenser tubes in place is composed of a nylon plate bolted to the condenser end plate. A gasket insures a tight seal between the nylon and condenser end plates. A stainless steel plate is bolted against the nylon plate with O-rings in between which ensure an air tight seal between the environment and the inside of the condenser.

The plates holding the test condenser tubes in place must be removed prior to removal of the condenser end plates. The stainless steel plates at both ends are removed by unbolting the retaining nuts. The nuts holding the nylon plate in place are then removed. A rubber mallet is used to tap out the test condenser tubes from the left end of the platform. This causes the right end nylon plate to move away from the right condenser end plate. Screwdrivers can then be used to pry the nylon end plate away from the condenser.

The condenser end plates can be removed after the test condenser tubes have been removed. As a word of caution, these end plates weigh in excess of 100 pounds. Therefore, they must be supported during the removal process. In these studies, a Porta-lift was used for this purpose. The right condenser end plate was removed first, followed by the left end plate. Prior to removal of the left end plate, the shroud assembly had to be removed. This was facilitated by first removing the thermocouple probes which penetrate the top of the condenser unit. The shroud was then pulled out of the condenser.

If the auxiliary condenser tubes are to be removed, then the the left condenser end plate will have to be removed. This is because the five auxiliary condenser tubes are attached to the left end plate. The location of the test platform in relation to the coolant

sump is such that this end can not be removed with the evaporator/condenser test platform in its present location. Rather, if the left head must be removed, the entire test platform must be rotated 180 degrees about a vertical axis.

Initial inspection of the condenser unit showed only one small area of discoloration which surrounded the middle viewport. No other obvious signs of contamination were evident. The condenser and evaporator were both cleaned using "3-D Supreme Cleaning Agent", a commercially available product which leaves no residue. All components were rinsed with fresh water, dried, then given a final cleaning with acetone. The auxiliary condenser coils were cleaned and decontaminated in the same manner.

The copper condenser tubes also appeared somewhat discolored. These were cleaned using "Shower Power", a commercially available cleaning agent. Prior to installation into the condenser unit, they were given a final cleaning with acetone to ensure removal of any residue left over from the cleaning procedure.

During the reassembly, all gaskets were replaced. Prior to replacement of the evaporator unit, a test blank was installed on the condenser and a vacuum test successfully performed. After this test was completed, the test blank was removed and the evaporator reinstalled. All penetrations were reconnected, the boiling tube bundle reinstalled and another vacuum test performed.

Prior to charging the system with R-113, the ethylene glycol/water coolant pumps were energized and flow started through the test condenser tubes. For this initial test, smooth copper tubes were used. The temperature of the coolant was $-15°C$. Since Mabrey [Ref. 2] had noted that during condensation, a thickening between the fins of copper/nickel finned tubes was occurring which suggested ice formation, it was felt that water vapor might be present in the test platform since it had been open to the atmosphere. Therefore, by running the coolant mixture at this temperature through the condenser tubes, if no condensation was seen, this would rule out the presence of water vapor. During this intitial run, no condensation was seen on any tubes. In addition, the orange peel appearance noted by Mabrey on the smooth copper tubes was not in evidence.

After checking the evacuated system for possible contamination, the test platform was charged with fresh R-113. Coolant flow was again started. This time an orange peel, or dimpled appearance could be seen on the top condenser tube. This suggested some type of contamination layer.

Power ($\sim$ 20 kW) was applied to the heaters while the coolant temperature was raised to $\sim$ 0°C. The orange peel appearance on the top tube was still present. Measurements of the vapor temperature at the top and bottom of the condenser showed approximately a 20 °C differential, signifying the presence of significant amounts of noncondensible gases. These gases enter with the refrigerant and are released into the condenser when the liquid is boiled.

A vacuum pump, connected to the top of the condenser was used to remove these noncondensible gases. When this was done, the temperature differential between the top and bottom of the condenser decreased to less than 0.5°C. The orange peel appearance quickly disappeared and all four condenser tubes demonstrated good filmwise condensation. Based on these observations, it was standard practice to degas the system for at least four hours after the system was open to the atmosphere or fresh R-113 was added to the evaporator.

IV. EXPERIMENTAL PROCEDURES

A. GENERAL PROCEDURES

The experimental procedure was identical for all experiments. The coolant pumps were energized and flow started through one of the auxiliary condensers. Flow was also started through the four test condenser tubes at a nominal coolant velocity of 0.2 m/s. The auxiliary heaters were set at 3 kW, the instrumented tube bundle set at 3.5 kW, and the simulation heaters set at 1.5 kW.

The vapor temperature at the bottom and the top of the condenser was allowed to rise to 47.5 °C which corresponds to the saturation temperature of R-113 at atmospheric pressure. The rate of temperature rise was controlled to approximately 0.3 degrees per minute by adjusting the rate of coolant flow through the auxiliary condenser. During the heat up period, a vacuum pump, connected to the top of the condenser via the cold trap, was run continuously in order to remove any noncondensibles. The absence of noncondensibles was indicated when the vapor temperature difference between the top and bottom of the condenser unit was less than 0.5 °C.

When the vapor temperature was about 45 °C, coolant velocity was adjusted to between 1 and 1.2 m/s for the four condenser tubes. The heater power levels were then readjusted in order to maintain a relatively constant condenser pressure while leaving one auxiliary condenser coil on-line. At lower coolant velocities, condenser pressure was maintained using the auxiliary condenser coils. The power settings for the heaters were not altered during the course of the experiment.

During all experimental runs, the vapor temperature at the top and bottom of the condenser unit was maintained between 46.5 and 48.5 °C which corresponds to a condenser pressure of approximately one atmosphere. Pressure control was accomplished by adjusting coolant flow through the auxiliary condenser tubes.

1. Tube Bundle

Once the vapor temperature was between 46.5 and 48.5 °C , coolant flow was adjusted to the desired coolant velocity for the four test condenser tubes. Here, the desired coolant velocity is an input to the computer program. The program then calculates the required flowmeter setting using the calibration curves (Appendix A). The initial coolant velocity used was 0.2 m/s. After setting the flow rate, the temperature difference between the inlet and outlet of all four condenser tubes was monitored using the com-

puter program, DRPCON7. Approximately ten minutes was required for this temperature differential to stabilize. Once stable, a data set was taken. This procedure was repeated in increments of 0.1 m/s up to a maximum of 1.2 m/s. The procedure was then repeated by decreasing the flow in 0.1 m/s increments down to 0.2 m/s. Two data sets were obtained at each coolant velocity. It should be mentioned that for the copper/nickel finned tubes, the range of flow settings was changed. The minimal flow setting was 0.6 m/s while the maximum flow setting was 2.0 m/s. The reasons for this change are discussed in "3. Enhancement of Outside Heat Transfer Coefficient." on page 30.

2. Single Tube Experiments

After completion of the tube bundle runs, the experiment was repeated for each individual tube starting with the lowest tube (tube D). Flow through the tubes above the test tube was stopped and flow through the tube of interest was set for a velocity of 0.2 m/sec. The vapor pressure was again maintained using an auxiliary condenser coil. During this time, the upper tubes were continuously observed for condensate formation. No single tube data was obtained until condensate formation on the upper tubes had ceased. This typically required twenty to thirty minutes.

Once the coolant temperature rise had stabilized, a data set was obtained. Following this, the coolant flow rate was increased in 0.1 m/s increments up to the maximum flow velocity. A data set was obtained at each flow velocity. Throughout the experiment, tubes above the test tube were visually monitored to ensure that no condensate formation occurred. The flow velocity was then decreased in 0.1 m/s increments down to the minimum coolant flow velocity, with a data set obtained at each flow velocity. This procedure was repeated for each test condenser tube in the bundle.

B. SPECIFIC EXPERIMENTS

1. Baseline Experiments

The purpose of the baseline experiments was to validate the performance of the condenser/evaporator test platform by comparing data to that of Mabrey [Ref. 2]. These initial baseline experiments were performed using smooth copper tubes (inside diameter of 13.26 mm , outside diameter of 15.88 mm, thermal conductivity of 386 W/m K). The condensing length of the tubes was 1.219 m. The tubes were supplied by Wolverine Tube Co.

2. Enhancement of Inside Heat Transfer Coefficient

The main purpose of this study was to obtain accurate values of the outside heat transfer coefficient, h_o. In order to obtain accurate values for h_o, the resistance to heat transfer on the inside of the tube $\frac{1}{h_i A_i}$ needs to be minimized. The reason that this resistance must be minimized is that if the inside resistance is a large part of the overall vapor-to-coolant resistance, then h_o will be become inaccurate. In order to minimize the inside resistance, one must enhance the inside heat transfer coefficient.

The inside heat transfer coefficient is influenced by the type of coolant flow through the tubes. In the test platform, the coolant is a 54/46 percent (by weight) ethylene glycol/water mixture. At temperatures below 0 °C, this fluid is extremely viscous with Reynolds numbers in the range of 300 to 2800, suggesting laminar flow. When the coolant flow is laminar, the inside heat transfer coefficient is low. However, if this laminar flow can be mixed, then the inside resistance can be decreased and the heat transfer coefficient increased. This can be accomplished through the use of radial mixing elements. Radial mixing elements have the effect of mixing the thermal boundary layer (formed on the inside of the tube) with the colder core flow thereby improving heat transfer. It should be noted that a penalty in using such elements is an increased pressure drop which must always be kept in mind in practical applications. However, for the purposes of this study, increased pressure drop was not important.

In this thesis, two types of mixing elements were investigated. The first was a twisted tape insert made of copper [Ref. 47,48]. This tape insert had a thickness of 0.559 mm with a pitch for a 180 degree twist of three times the inner tube diameter.

The second type of insert used in this study was a commercially available product referred to as HEATEX radial mixing elements.[3] These elements are manufactured from stainless steel and consist of a number of small flexible loops (petals) of steel wire attached to a central wire core These flexible petals are placed in the tube such that they face the oncoming flow. Their outside diameter is slightly larger than the inside diameter of the tube and are therefore kept in position with a "pinch" fit. The petals disturb the thermal boundary layer and promote fluid mixing. Figure 4 shows the two types of inserts. It should be noted that two different sizes of HEATEX were used since condenser tubes of two different inside diameters were examined.

---

3 manufactured by CAL GAVIN Birmingham, England.

27
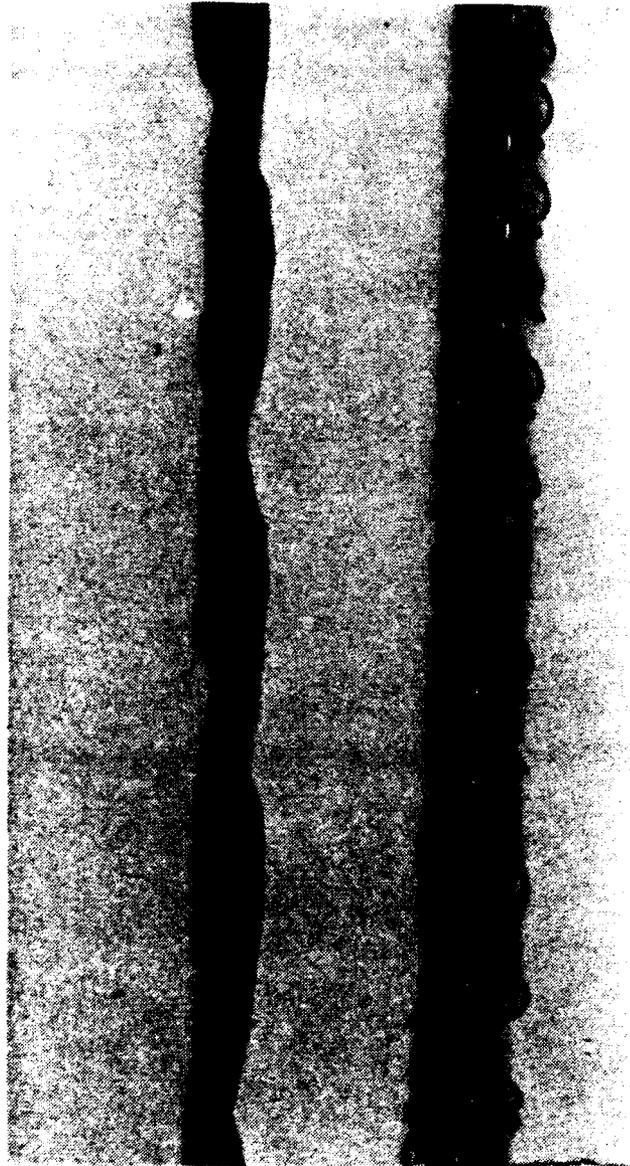
Figure 4.   Twisted tape insert and HEATEX radial mixing element.

As part of the baseline experiments, the smooth copper tubes were also tested with no insert installed. Experiments were repeated on the smooth copper tubes using the twisted tape inserts as well as the HEATEX elements. This data was then used as a baseline for comparison with other types of condenser tubes. It should be mentioned that when no insert is present, the inside tube resistance to heat transfer is very large relative to the outside resistance. In some cases (typically when the coolant flow rate is low), this can produce a negative value for the outside heat transfer coefficient. This is simply the inaccuracy in the calculation of h_o when the inside resistance dominates and clearly demonstrates the need for an insert as mentioned above.

3. Enhancement of Outside Heat Transfer Coefficient.

The overall heat transfer coefficient, U_o, is influenced by the outside as well as the inside heat transfer coefficient. The means by which the outside heat transfer coefficient is enhanced is by altering the outside geometry of the condenser tube.

Experiments to investigate enhancement of the outside heat transfer coefficient utilized three types of condenser tubes. The first type of tube tested was a copper/nickel finned tube (26 fins/inch). This tube has an inside diameter of 10.16 mm with a thermal conductivity of 43 W/m K. The tubes were manufactured by Wolverine Tube Co. These tubes were tested as a tube bundle as well as individually. Experiments were performed with no insert as well as with the twisted tape and HEATEX inserts. These tubes are shown in Figure 5. One point must be mentioned with regard to the finned tubes. The use of fins on the outside of the tube enhances the heat transfer considerably. Consequently, the inside resistance again becomes a large fraction of the overall resistance, even when inserts are used.

One additional point must be discussed with regard to the copper/nickel finned tubes. Because the size of the holes in the nylon end plates is fixed, the outside diameter of any tubes to be tested must be the same. In order to have these finned tubes the same outside diameter, the inside diameter had to be reduced. Since the inside tube diameter was smaller for the finned tubes, higher coolant velocities were obtained.

The second series of experiments used a copper/nickel KORODENSE tube also manufactured by Wolverine Tube Co. This tube, shown in Figure 5, has a shallow spiral groove along the entire condensing length with a pitch of 10 mm. As with the finned copper/nickel tubes, experiments were performed on these KORODENSE tubes with no insert as well as with the twisted tape and HEATEX inserts. The tubes were tested in a tube bundle as well as individually.

29

Fujii and co-workers [Ref. 33] have previously shown that the condensation of refrigerants can be enhanced by wrapping smooth condenser tubes with a small diameter wire. Therefore, a third series of experiments were performed in which copper/nickel KORODENSE tubes were tightly wound with wire as shown in Figure 5. Since these authors showed that there was an optimum wire size which gave the greatest enhancement of heat transfer (0.5 mm at a pitch of 3 mm), the KORODENSE tubes used in this study were wrapped with one of three different sizes of wire. The sizes used were 0.029" (0.736 mm) diameter stainless steel wire, 0.049" (1.244 mm) diameter stainless steel wire, and 0.0675" (1.715 mm) diameter copper wire. The wire lay in the shallow grooves of the KORODENSE tubes and was soldered to the tubes at the ends only. Although the wire was tightly wrapped around the tubes, it was not physically attached except at the ends. Experiments were carried out on each tube individually. Each tube was tested in all four locations within the bundle. This was done to see if position within the bundle (i.e. wake from tubes above) had any effect. Only the HEATEX inserts were used with these tubes.

Once an optimal wire diameter was determined, a fourth series of experiments was conducted. This series used the same four KORODENSE tubes, only this time each was wrapped with the same optimal diameter wire. The tests were done with the HEATEX elements installed. Experiments were conducted on the tubes individually in order to compare with the previously obtained single tube data. In addition, these wire wrapped tubes were tested as a tube bundle in order to investigate whether or not the wire wrap had any effect on condensate innundation.
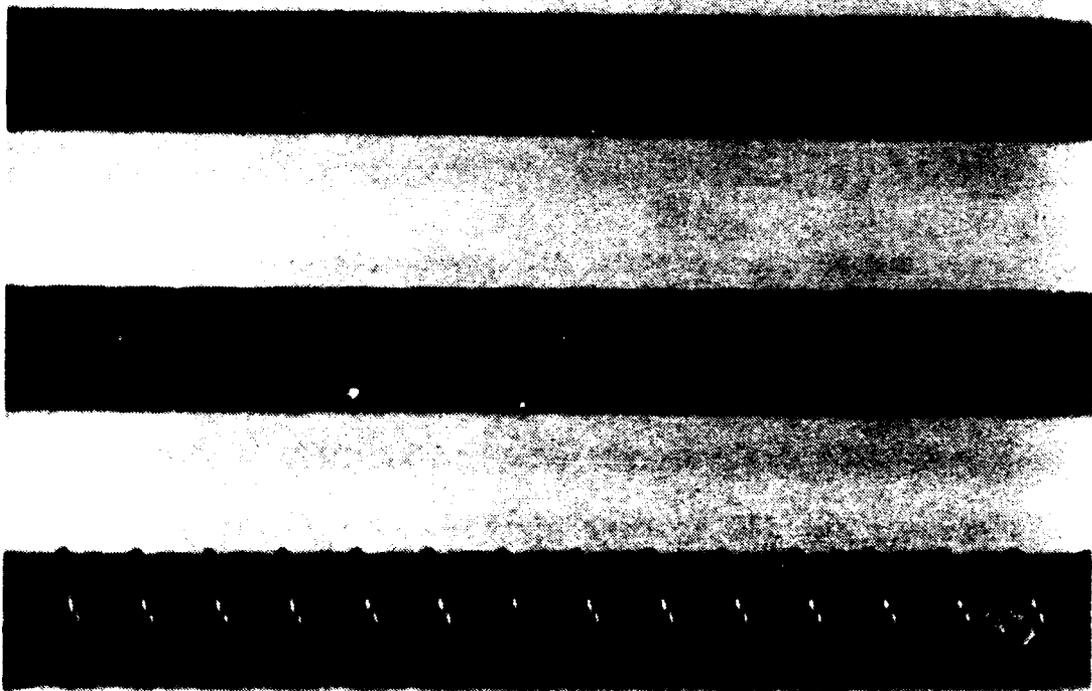
Figure 5. Copper/nickel finned tube, Korodense tube, Wire wrapped tube.

31

V. DATA REDUCTION

A. DATA ACQUISITION

The data acquisition system consisted of a Hewlett-Packard series 9000 computer coupled with a Hewlett-Packard model 3479A Data Acquisition system. The computer program used in this study was origin.'ly written by Mabrey [Ref. 2] but has been substantially modified in the course of this thesis. The data acquisition thermocouple channel assignments are shown in Table 1.

Table 1. THERMCOUPLE CHANNEL ASSIGNMENTS.

Channel	Temperature Measured
0	Vapor (condenser top)
1	Vapor (condenser top)
—2	Vapor (top of liquid)
3	Liquid (top of bundle)
4	Liquid (top of bundle)
5	Liquid (bottom of pool)
6	Inlet tube 1
7	Inlet Tube 2
8	Inlet Tube 3
9	Inlet Tube 4
10, 11	Outlet Tube 1
12, 13	Outlet Tube 2
14, 15	Outlet Tube 3
16, 17	Outlet Tube 4

The computer program allows the user to acquire data, reprocess data, or plot data. During data acquisit the user selects the fluid velocity desired. Based on coolant inlet temperature, the coolant fluid properties and mass flow rates are calculated (see Appendix C). The program then uses equations derived from the flowmeter calibrations (Appendix A) in order to calculate the appropriate flowmeter settings.

Once the flowmeters are set for the required coolant velocity, the computer system displays the vapor temperature at the top and bottom of the condenser as well as the

coolant temperature rise for each condenser tube. When these temperature rises are stable (approximately 10 minutes), the user can terminate the display and acquire the data for that coolant flow velocity. While the temperature rises were stabilizing, coolant flow through an auxiliary condenser tube was adjusted in order to maintain constant pressure.

B. DATA REDUCTION

The main function of the data reduction section of the computer program is to back out the outside heat transfer coefficient from an experimentally determined value of U_o and an assumed inside heat transfer coefficient correlation. This is accomplished in the following manner.

The data acquisition system measures the emf from the thermocouples listed in Table 1. These values of emf are converted to temperatures using correlations obtained from the temperature calibrations both contained in Appendix B. The volume flow rate of the ethylene glycol/water mixture is calculated based on the desired coolant velocity. The density, dynamic viscosity, thermal conductivity, heat capacity, and Prandtl number of the coolant are calculated using inlet temperatures (see Appendix C and Appendix E). From these quantities, the coolant mass flow rate and fluid velocity are calculated. The coolant Reynolds number for each condenser tube is also calculated. The heat transferred to the coolant is then found from

$$\dot{q} = \dot{m}C_p\Delta T \qquad (5-1)$$

where $\dot{m}$ is the mass flow rate and ΔT is the coolant temperature rise. The outside heat flux, q''_o, is calculated by dividing $\dot{q}$ by the surface area of the tube based on the outside tube diameter.

The overall heat transfer coefficient, U_0, is given by:

$$U_0 = \frac{q''_o}{LMTD} \qquad (5-2)$$

where LMTD is the log mean temperature difference defined as:

$$LMTD = \frac{\Delta T}{\ln\left[\dfrac{T_{sat} - T_{c_i}}{T_{sat} - T_{c_o}}\right]} \qquad (5-3)$$

33

The inside heat transfer coefficient (given as a dimensionless Nusselt number, Nu_c) can then be calculated using one of three correlations. The choice of which correlation to use depends upon the type of insert installed inside the condenser tube. If no insert is installed, then Nu_c can be calculated from a correlation provided by Hausen [Ref. 49][4] for developing laminar flow in a tube at constant wall temperature.

$$Nu_c = 3.66 + \left[\frac{0.0668\left(\frac{d_i}{l}\right)Re_{D,c}Pr_c}{1 + 0.04\left[\left(\frac{d_i}{l}\right)Re_{D,c}Pr_c\right]^{0.666}} \right] \qquad (5-4)$$

If a twisted tape insert is installed, a correlation given by Hong and Bergles [Ref. 47] can be used to calculate Nu_c where

$$Nu_c = 5.172\left[1 + 5.484x10^{-3}Pr_c^{0.7}\left(\frac{Re_s}{y}\right)^{1.25} \right]^{0.5} \qquad (5-5)$$

Here, y is the twist ratio (pitch over diameter). Re_s is the coolant Reynolds number, corrected for tape thickness. Re_s is given by:

$$Re_s = \frac{4\dot{m}}{\pi\mu(d_i - 4\delta)} \qquad (5-6)$$

where δ is the tape thickness. This correlation is valid for Re_s up to about 2300.

A number of correlations for determining the inside heat transfer coefficient have been published when using a wire matrix insert [Ref. 50,51,52, 53]. However, these correlations are specific for the type of mixing element used, and are especially dependent upon the loop density of the wire matrix.

The HEATEX mixing elements supplied by CAL GAVIN (nominal diameters of 13.26 and 10.16 mm) did not match the loop densities used by previous investigators. However, CAL GAVIN did supply experimental data for the heat transfer coefficient versus fluid velocity at three different temperatures for both sizes of elements. To convert this data into a usable correlation, the following procedure was developed.

---

4 Calculations were performed for coolant flow through the smooth copper tubes at coolant velocities of 0.2 and 1.2 m/s. These calculations confirmed the coolant flow was not fully developed.

The correlations of Oliver and Aldington [Ref. 51, 52] took the form of

$$Nu = cRe^m Pr^n \qquad (5-7)$$

where c, m, and n are arbitrary constants which are dependent upon the type of insert used. It was assumed that a similar type of equation could be used in the present study by fitting the constants to the data (once non-dimensionalized) provided by CAL GAVIN.

The correlations of Oliver and Aldington [Ref. 51, 52] had Prandtl number raised to a power of 0.46. This was primarily to take into account temperature effects. Therefore, the same value was assumed in the present study. Values of $Nu/Pr^{0.46}$ were plotted as functions of Re on a log-log basis. Figure 6 shows the data obtained for the 13.26 mm elements while Figure 7shows the data obtained for the 10.16 mm elements. Based on the appearance of the non-dimensionalized data, a least squares linear regression was performed on each data set. The figures yield the values of the constants m and c (slope and intercept respectively) and provide the following correlations for the 13.26 and 10.16 mm elements respectively:

$$Nu_c = 0.226 Re^{0.65} Pr^{0.46} \qquad (5-8)$$

and

$$Nu_c = 0.065 Re^{0.76} Pr^{0.46} \qquad (5-9)$$

Once Nu_c was obtained from equations (5-4, 5-5 5-8 or 5-9), the inside heat transfer coefficient was calculated from:

$$h_i = \frac{Nu_c k_c}{d_i} \qquad (5-10)$$

35

HEATEX CORRELATION (13.26 MM)

NU = 0.226(RE**.65)(PR**.46)
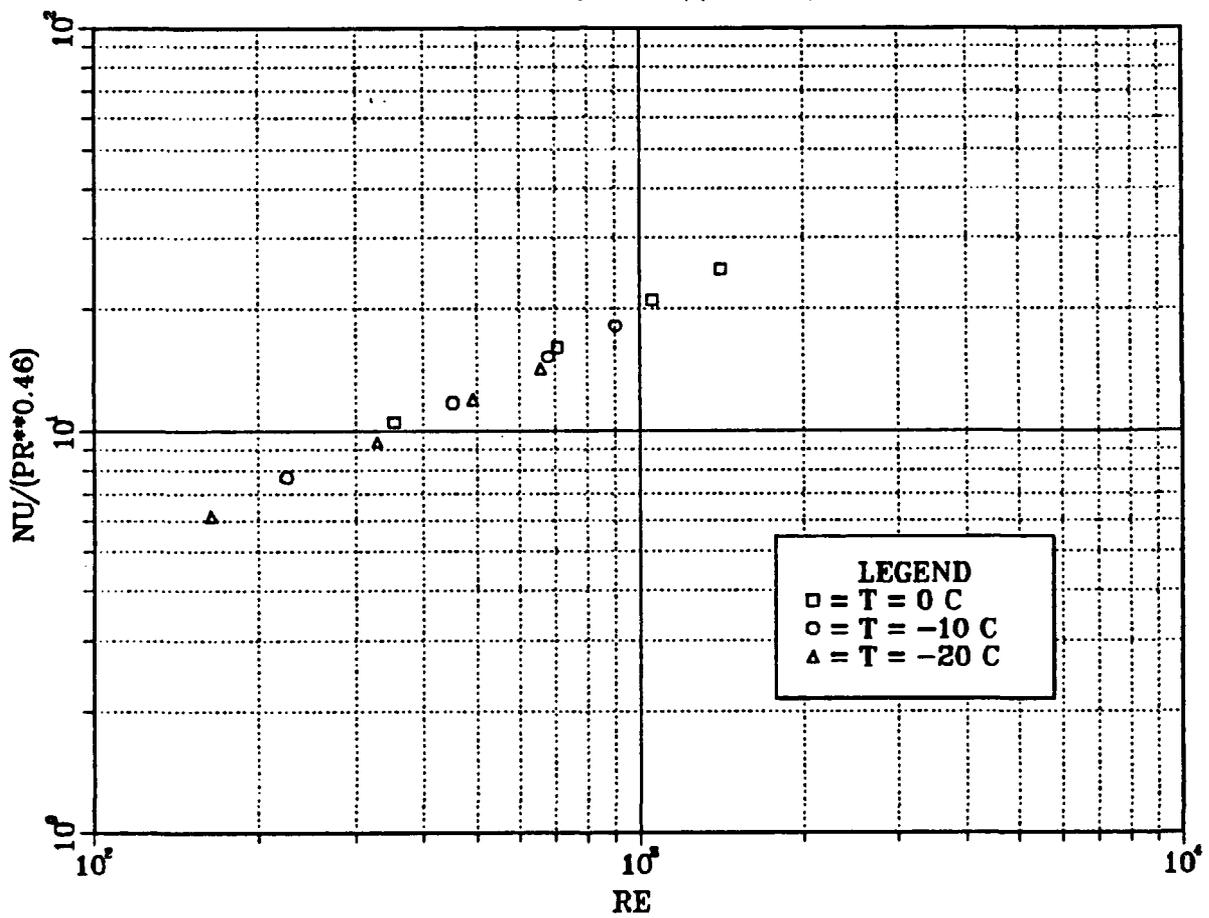
Figure 6.   HITRAN correlation for 13.26 mm elements

HEATEX CORRELATION (10.15 MM)

NU = 0.067(RE**.76)(PR**.46)
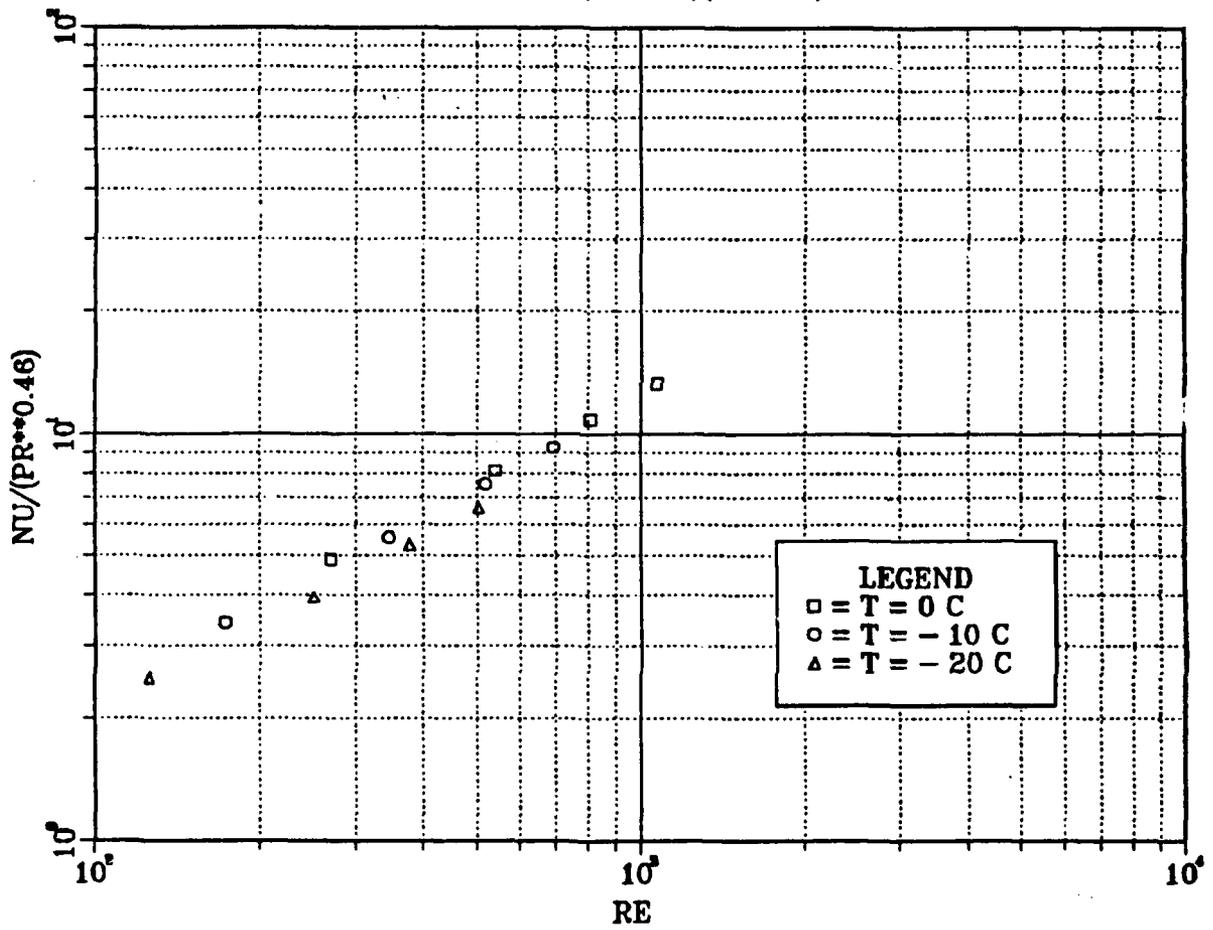
Figure 7. HITRAN correlation for 10.16 mm elements

The overall heat transfer coefficient, U_o is found from the summation of resistances to heat transfer across the whole tube:

$$\frac{1}{U_0 A_0} = \frac{1}{h_o A_o} + R_f + R_w + \frac{1}{h_i A_i} \qquad (5-11)$$

where R_f and R_w are the fouling and wall resistances respectively. In order to find h_o equation (5-11) can be rearranged to give:

$$h_o = \frac{1}{\dfrac{1}{U_0} - R_w A_o - R_f A_o - \dfrac{1}{h_i}\left(\dfrac{A_o}{A_i}\right)} \qquad (5-12)$$

In all experiments, the tubes were meticulously cleaned and therefore, the fouling resistance, R_f, was neglected.

In order to better assess the calculated experimental value of the outside heat transfer coefficient, some form of comparison with prediction must be made. In this case, the experimentally obtained value of h_o was compared with Nusselt theory. However, because the wall temperatures are not known a priori, an iteration must be used to determine this predicted value of h_o. An initial value of h_o was estimated and based on this estimate, a wall temperature was calculated using the equation:

$$T_{wall} = T_{sat} - \frac{q''_o}{h_o} \qquad (5-13)$$

The condensate film properties are then calculated using a reference temperature given by:

$$T_{film} = \frac{1}{3} T_{sat} + \frac{2}{3} T_{wall} \qquad (5-14)$$

Once these condensate fluid properties are known (Appendix D), the theoretical outside heat transfer coefficient can be calculated from Nusselt theory [Ref. 4]:

$$h_o = 0.655 \left[\frac{g k_f^3 \rho_f^2 h_{fg}}{\mu_f d_o q''_o} \right]^{\frac{1}{3}} \qquad (5-15)$$

This calculated value is then compared to the initial guess. The error between the initial guess and the calculated value are compared. If the error is greater than 0.1%, the

"guessed" value for h_o is updated and the iteration continued. This process is repeated until the error between the calculated and guessed values is less than 0.1%.

C. MODIFIED WILSON PLOT

The outside and inside heat transfer coefficients could be determined directly by measuring the vapor temperature, the tube wall temperature, and the coolant temperature rise. Of these, the most difficult temperature to measure directly is the tube wall temperature. This would require the use of instrumented tubes with thermocouples embedded in the tube wall. Unfortunately, the manufacture of such instrumented tubes is both costly and time consuming, especially when large numbers of tubes are to be tested.

An alternative to the use of instrumented tubes is to employ the "Modified Wilson Plot" procedure. This procedure has been outlined in detail by Marto [Ref. 10]. The modified Wilson plot procedure relies upon the overall heat transfer coefficient being reliably measured from experimental data. A summation of heat transfer resistances from vapor-to-coolant is then expressed as:

$$\frac{1}{U_0} = \frac{1}{h_0} + R_f A_0 + R_w A_0 + \frac{1}{h_i}\left(\frac{A_0}{A_i}\right) \qquad (5-16)$$

As mentioned above, the fouling resistance is assumed to be negligible. This summation of resistances can be rearranged into a linear equation:

$$\left[\frac{1}{U_0} - R_w A_0\right] = \left(\frac{A_0}{A_i}\right)\frac{1}{h_i} + \frac{1}{h_o} \qquad (5-17)$$

In looking at the correlations given above for h_o and h_i, it can be seen that they all employ a leading coefficient. As an example, Nusselt theory, (equation 5-15) can be expressed in the form:

$$h_o = \alpha F \qquad (5-18)$$

where:

$$F = \left[\frac{g k_f^3 \rho_f^2 h_{fg}}{\mu_f d_o q''_o}\right]^{\frac{1}{3}} \qquad (5-19)$$

In a similar manner, the equations for the inside heat transfer coefficients (equations 5-4, 5-5, 5-8, and 5-9) can also be transformed into the form:

39

$$h_i = C\Omega \qquad (5-20)$$

where the form of Ω would depend on which type of tube insert was being used. As an example, using equation 5-5, Ω would become:

$$\Omega = \frac{k_c}{d_i}\left[1 + 5.484x10^{-3}Pr_c^{0.7}\left(\frac{Re_s}{y}\right)^{1.25}\right]^{0.5} \qquad (5-21)$$

The linearized version of the summation equation (equation 5-17) can then be rearranged to give:

$$\left[\frac{1}{U_o} - R_wA_o\right]F = \frac{A_oF}{C\Omega A_i} + \frac{1}{\alpha} \qquad (5-22)$$

This is a linear equation with two unknowns, namely, C and α . Letting

$$Y = \left[\frac{1}{U_o} - R_wA_o\right]F \qquad (5-23)$$

and

$$X = \frac{A_oF}{A_i\Omega} \qquad (5-24)$$

we can obtain a simplified linear equation:

$$Y = mX + b \qquad (5-25)$$

where

$$m = \frac{1}{C} \qquad (5-26)$$

and

$$b = \frac{1}{\alpha} \qquad (5-27)$$

Since Ω and F are both temperature dependent, an iterative solution procedure must be used. This is accomplished by applying a least squares fit to the linear equation.

It should be pointed out that the accuracy of the modified Wilson plot procedure relies heavily on the number of data points taken as well as the range of coolant velocities used.

40

VI. RESULTS AND DISCUSSION

A summary of the experimental runs performed in this thesis is presented in Table 2. The abbreviation "SM" is for the smooth copper tubes, "KD" is for the KORODENSE tubes, and "CF" is for the copper/nickel finned tubes.

Table 2.   SUMMARY OF EXPERIMENTAL RUNS

Data Set	Tube type	Insert	Comments
SM01	Copper	Tape	Test run
SM02	Copper	Tape	Test run
SM03	Copper	HEATEX	Test HEATEX correlation
SM05	Copper	HEATEX	Tube A insert bad
SM06	Copper	HEATEX	
SM07	Copper	Tape	
KD01	KORODENSE	Tape	1st test of KORODENSE
KD02	KORODENSE	None	
KD03	KORODENSE	HEATEX	
KD04	KORODENSE	Tape	
KD05	KORODENSE	HEATEX	Wire wrapped
KD06	KORODENSE	HEATEX	Wire wrapped
KD07	KORODENSE	HEATEX	Wire wrapped
KD08	KORODENSE	HEATEX	Wire wrapped
KD09	KORODENSE	HEATEX	0.049" Wire, bad insert
KD10	KORODENSE	HEATEX	0.049" Wire
KD11	KORODENSE	HEATEX	0.049" Wire
KD12	KORODENSE	HEATEX	0.049" Wire
CF01	Cu/Ni fin	Tape	dropwise condensation
CF02	Cu/Ni fin	Tape	Test run
CF03	Cu/Ni fin	Tape	
CF04	Cu/Ni fin	HEATEX	
CF05	Cu/Ni fin	None	

41

A. DECONTAMINATION

Prior to charging the system with R-113, the system was evacuated to approximately 15 inHg and left for 24 hours. For this initial test, smooth copper tubes were used. The ethylene glycol coolant pumps were energized and flow started through the test condenser tubes. The temperature of the ethylene glycol was -15°C. No condensation was seen on any tubes. In addition, the orange peel appearance noted by Mabrey [Ref. 2] was not observed.

After checking the evacuated system for possible contamination, the test platform was charged with fresh R-113. Coolant flow was again started. No orange peel, or dimpled appearance could be seen on the top condenser tube.

Power ($\sim$ 20 kW) was applied to the heaters while the coolant temperature was raised to $\sim$ 0°C. An orange peel appearance was noted for the top tube. Measurements of the vapor temperature at the top and bottom of the condenser showed approximately a 20 °C differential.

A vacuum pump, connected to the top of the condenser, was used to remove any noncondensible gases. When this was done, the temperature differential decreased to less than 0.5°C. The orange peel appearance quickly disappeared from the top condenser tube. This suggests that at least part of the contamination problem reported by Mabrey was due to the presence of noncondensibles.

In a subsequent series of experiments conducted on a set of titanium finned tubes, a foreign substance appeared to build up on the tubes during the experimental run. This occurred despite continuous operation of the vacuum pump. The tubes were removed from the condenser and inspected. The foreign material had an oily feel to it and an odor similar to that of ethylene glycol.

The titanium finned tubes had smooth extensions (6″ in length) attached to their ends using epoxy. These extensions penetrated the nylon block and steel plate and provided the smooth surface necessary for the O-rings to give an airtight seal between the condenser and the atmosphere. When these tubes were removed from the condenser, it was noted that the extensions on two of the tubes had come loose. The condenser was opened for inspection. Upon removal of the end plate and shroud assembly, pools of ethylene glycol were seen inside the condenser shell and on the auxiliary condenser tubes. In addition, ethylene glycol was found around the shroud assembly. The entire condenser and evaporator assembly were therefore dismantled and cleaned in accordance with the procedures outlined in "Experimental Apparatus".

After reassembly of the condenser and evaporator, the smooth copper tubes were cleaned and reinstalled. Operation of the test platform failed to show any orange-peel or dimpled appearance on the tubes. In addition, all tubes demonstrated good filmwise condensation.

It should be mentioned that in subsequent tests, copper/nickel finned tubes were installed in the condenser. However, unlike the titanium finned tubes, the smooth extensions on these tubes were soldered. During system operation, no foreign material was noted between the fins. Condensate retention between the fins was negligible. The tubes demonstrated good film wise condensation.

In summary, it is believed that the contamination problems noted by Mabrey [Ref. 2] were due to two problems. The first problem was the presence of noncondensibles during operation of the condenser. This resulted in the relatively low values for overall heat transfer as well as the fact that the top tube, which would be most affected by the noncondensibles, had the lowest heat transfer. Secondly, while it can not be proved conclusively, the author believes that ethylene glycol may also have been present in the system for certain tubes and that this is what was the material noted in between the fins by Mabrey.

B. OVERALL HEAT TRANSFER COEFFICIENT

In the following sections, the results pertaining to the overall heat transfer coefficient are presented. It is important to keep in mind that this value is experimentally determined and does not rely on any correlations other than for the fluid properties. In addition, it should be mentioned that in the following graphs, tube A is the top tube in the bundle, tube B the second, tube C the third, and tube D the bottom tube.

1. Smooth Copper Tubes

Figure 8 shows the overall heat transfer coefficient, U_o, plotted as a function of coolant velocity. The tubes did not contain any inserts. Note the low values for U_o. These values are almost constant for velocities over the range of 0.2 to 0.7 m/s. In this series of experiments, Reynolds numbers were typically of the order of 300 to 2300 indicating that flow is laminar. Without any insert to promote mixing, one would therefore expect relatively low values for U_o . This is because, in laminar flow, the inside resistance to heat transfer is relatively large and therefore leads to a lower value for heat transfer. Finally, there is essentially no difference between the tubes at low coolant velocities. This suggests that with these relatively low condensation rates, inundation is not a significant factor. At higher flow rate, it can be seen that the values for heat transfer

start increasing and that differences between the tubes become more apparent. As coolant velocity is increasing, flow is beginning to become turbulent. This reduces the inside resistance to heat transfer which leads to an increase in the overall heat transfer. As heat transfer increases, the effects of inundation become more apparent. This is presumably due to the outside resistance becoming more important as the inside resistance becomes less important.

The results obtained when these smooth copper tubes were tested individually is shown in Figure 9. Here, all the tubes behave similarly. In addition, comparison with Figure 8 shows that the values of U_o for the tubes when tested individually are approximately the same as for the top tube during the bundle run as one would expect.

The results for the same run with tape inserts installed are shown in Figure 10. It can be seen that U_o increases as coolant velocity increases. The values are clearly greater than when no insert is used (approximately 50% greater) indicating the advantage gained when enhancing the inside heat transfer coefficient in situations where coolant flow is laminar. The effect of condensate inundation on the overall heat transfer coefficient is also clearly seen. Here, condensate inundation results in approximately a 15% decrease in U_o when the top tube is compared to the second tube. Interestingly, there appears to be little difference in the values for U_o for tubes B, C, and D indicating that the condensate film is of similar thickness for these tubes. The data for a similar run performed by Mabrey [Ref. 2] is shown in Figure 11. The data obtained in the present study is almost twice the values reported by Mabrey. In addition, the present data shows that the top tube has the highest heat transfer which is to be expected assuming noncondensibles are not present. This was not the case in Mabrey's work.

The results for these same tubes when operated individually are shown in Figure 12. All the curves are essentially the same with values for U_o corresponding to that obtained for the top tube during the bundle run. The maximum variation is about 12%.

The next series of experiments were performed on the smooth copper tubes with the HEATEX elements installed. Figure 13 shows the results of these experiments when the tubes were operated as a bundle. As with the experiments performed with the twisted tape inserts, there is an increase in U_o as coolant velocity is increased. In this case the increase is 100% in comparison to the case with no insert! Also, as in the previous experiment, condensate inundation appears to result in about a 15% decrease in U_o for the second tube compared to the top tube. There appears to be little difference in the heat transfer coefficients for tubes B, C, and D, although some effect may be ar-

44

UZERO SMOOTH COPPER TUBES

SM04, R-113, NO INSERT, SINGLE
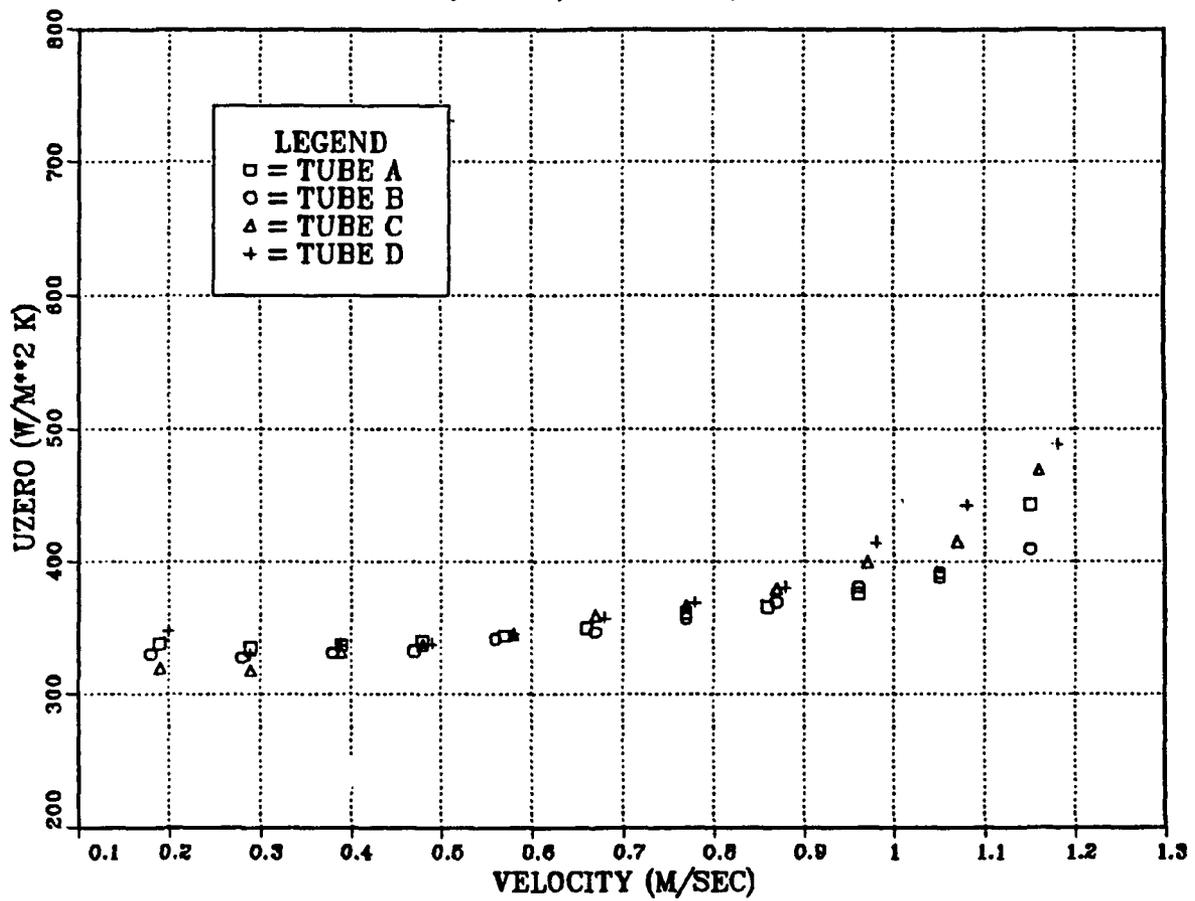
Figure 8. Bundle run on smooth copper tubes without inserts.

45

UZERO SMOOTH COPPER TUBES

SM04, R-113, BUNDLE, NO INSERT
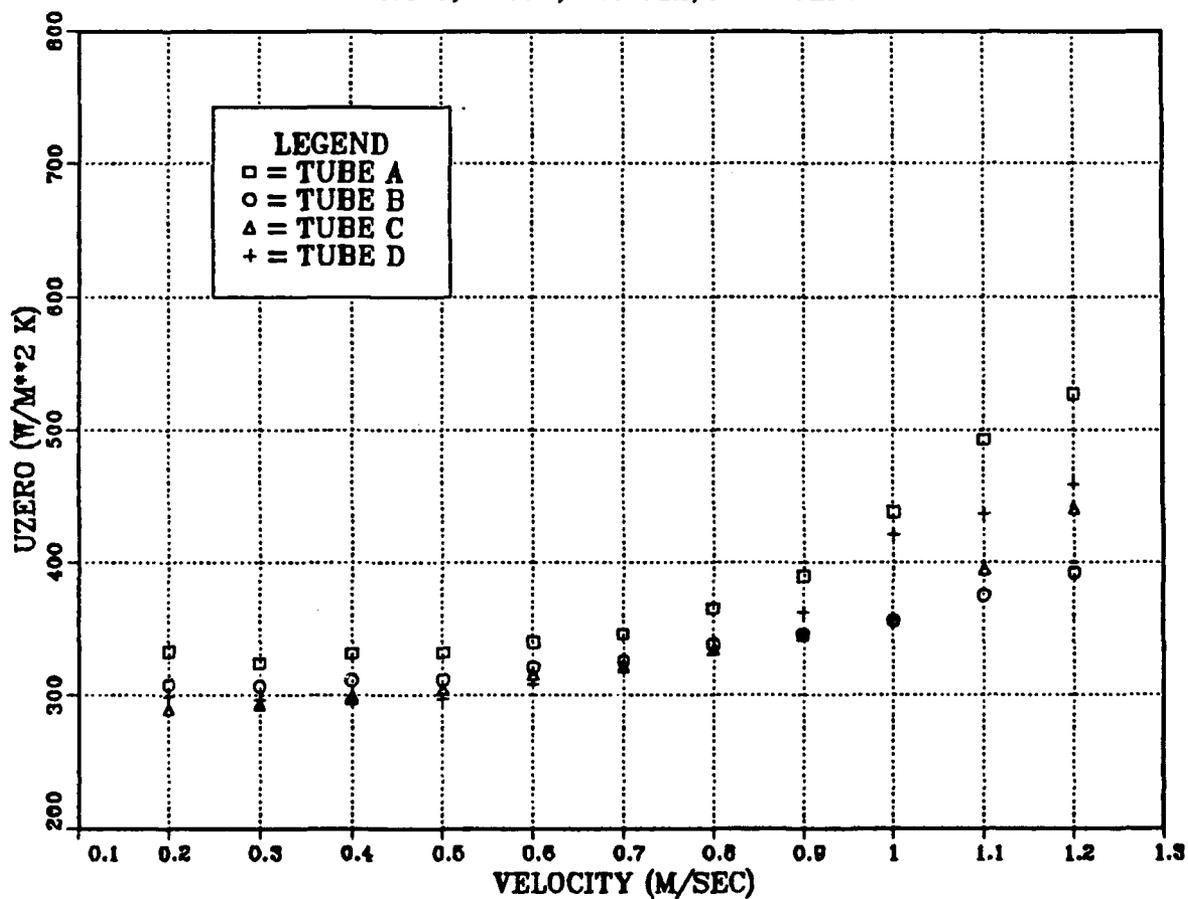

Figure 9. Single tube run on smooth copper tubes without inserts.

UZERO SMOOTH COPPER TUBES

SM07, R-113, BUNDLE, TAPE INSERT
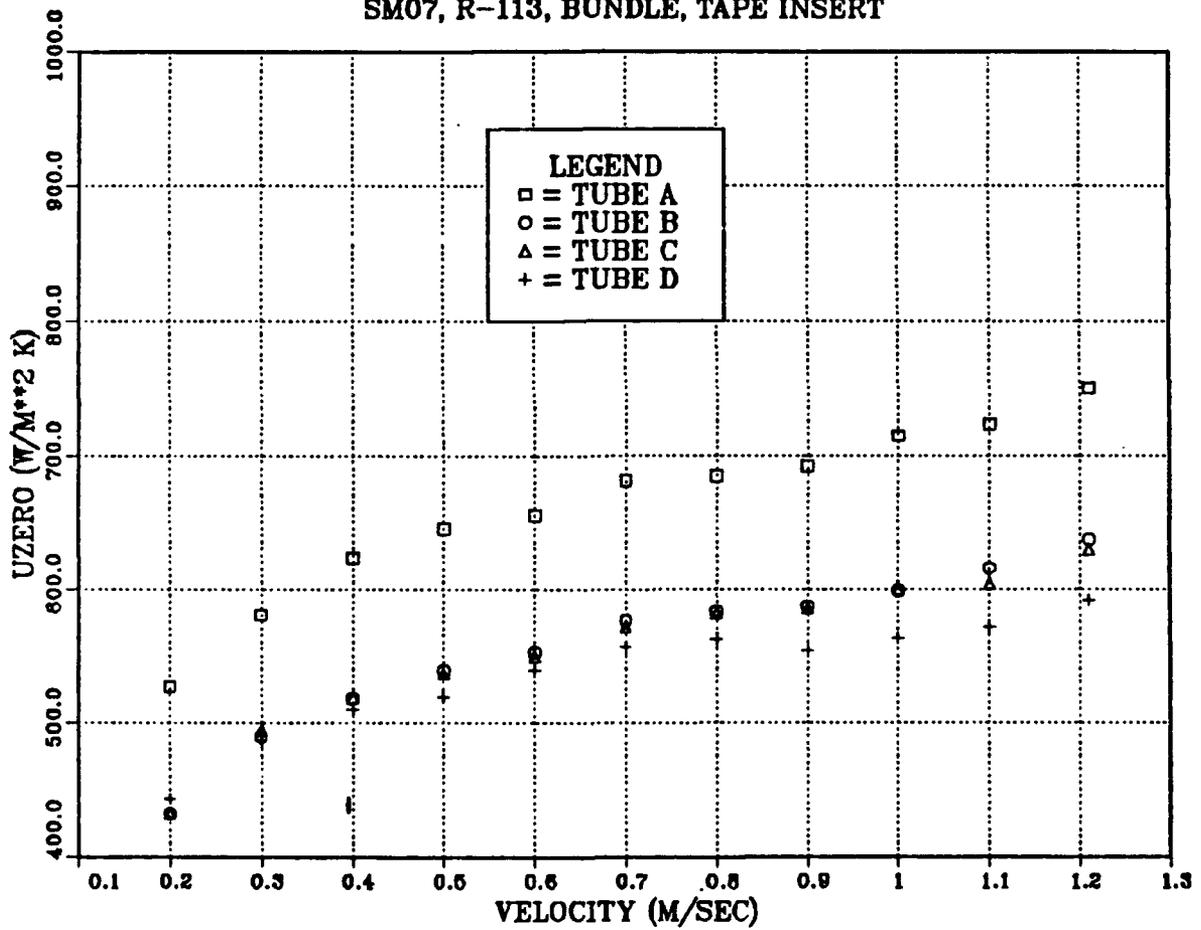

Figure 10.   Smooth copper tubes with tape insert operated as a bundle.
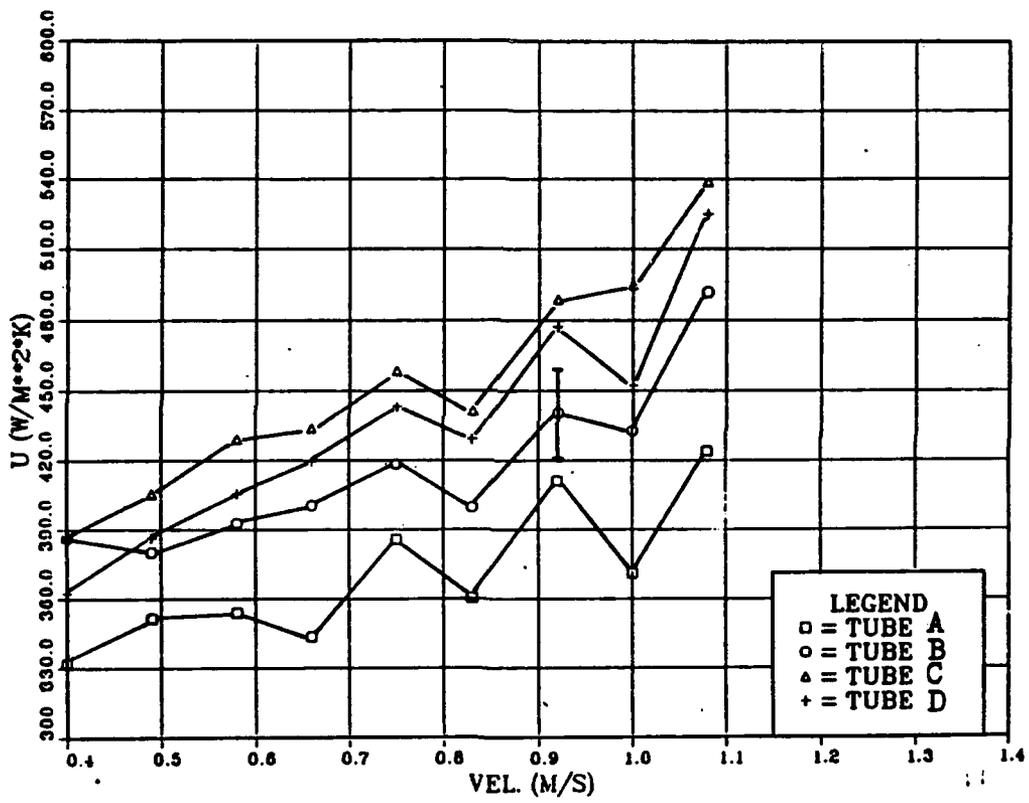
47

Figure 11.   Data from Mabrey [Ref. 2]

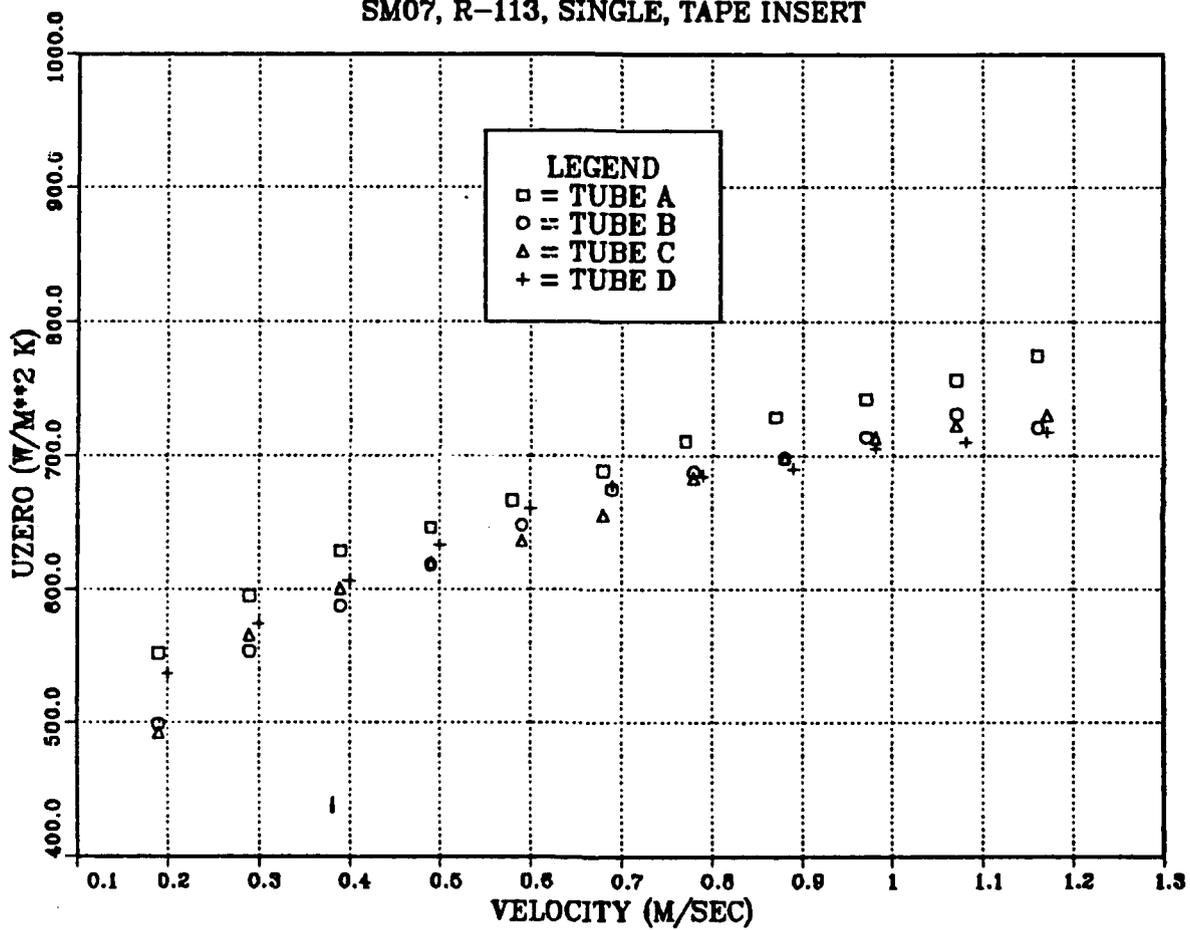# UZERO SMOOTH COPPER TUBES

SM07, R-113, SINGLE, TAPE INSERT

Figure 12. Smooth copper tubes with tape insert operated individually.

gued at the higher coolant velocities. This may reflect the fact that, at the higher coolant velocities, the inside resistance decreases while the outside resistance increases in importance. Anything which causes further increases in this resistance, such as condensate inundation, may cause noticeable effects in heat transfer.

As before, these same tubes with the HEATEX inserts installed were tested individually; the results are shown in Figure 14. The values of U_o for all four tubes are virtually identical and match the values for the top tube obtained during the bundle run (Figure 13). The values appear to be about 10% greater than for the tape insert experiments indicating that the HEATEX provides better tube-side enhancement.

While the preceding figures have shown the effect of inserts and also inundation on the heat transfer coefficient, two important questions must be addressed. First, what is the uncertainty associated with the calculated values for U_o. Secondly, the repeatability of the data must also be examined.

An uncertainty analysis was performed on U_o as shown in Appendix F based on the procedures of Kline and McClintock [Ref. 54]. At the low coolant velocities, the uncertainty was 5.2%. This uncertainty was based primarily on the uncertainty in the mass flow measurement. Here, one must include a scale interpolation term as well as a term to account for the time-wise jitter for the flowmeter. Since the flow rate is low for this coolant velocity, these two terms become more significant than at the higher flow rates. This is especially true if the magnitude of the time-wise jitter is the same for all flow rates which was the case in this study.

At high coolant velocities, the uncertainty in the value for U_o was about 5.6%. Here, the major factor driving this value is the uncertainty in the determination of temperature. The reason for this is that the ΔT between the tube inlet and outlet is smaller than for low coolant velocities. Since the relative uncertainty for the thermocouples remains the same regardless of coolant velocities, uncertainty in the temperature measurement, specifically the LMTD, becomes the driving factor in determining the uncertainty in U_o.

The uncertainty analysis performed here was only for the overall heat transfer coefficient, U_o. In this analysis, any uncertainty in the correlations used to determine the physical properties of the ethylene glycol coolant or R-113 were neglected. Therefore, the level of uncertainty reported here is slightly conservative.

In order to examine the reproducibility of the data, repeat experiments were performed on the smooth copper tubes with the HEATEX elements installed. It is important to mention that during the time between the two runs, the tubes were removed

50

UZERO SMOOTH COPPER TUBES

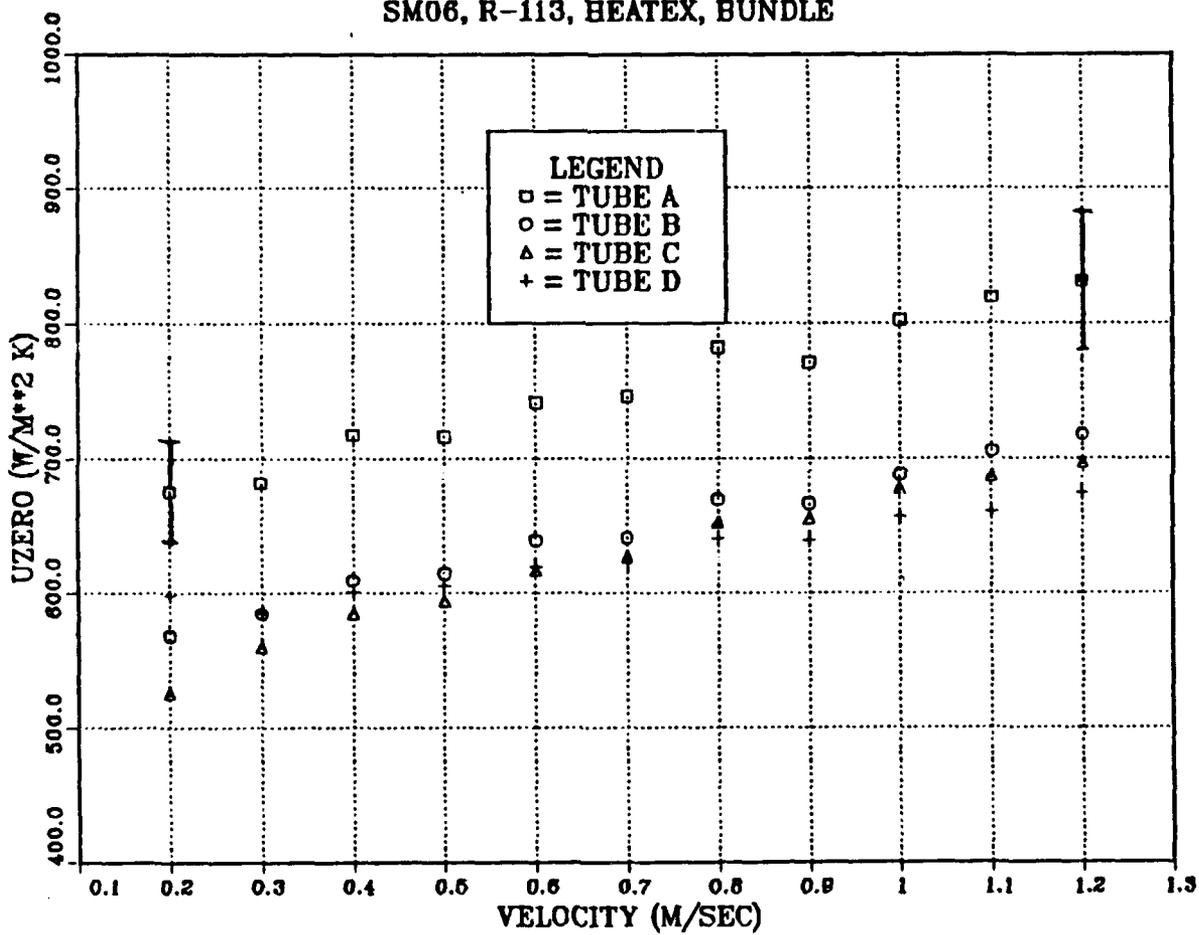## SM06, R-113, HEATEX, BUNDLE

Figure 13.   Smooth copper tubes with HEATEX inserts. Bundle operation.

51

UZERO SMOOTH COPPER TUBES
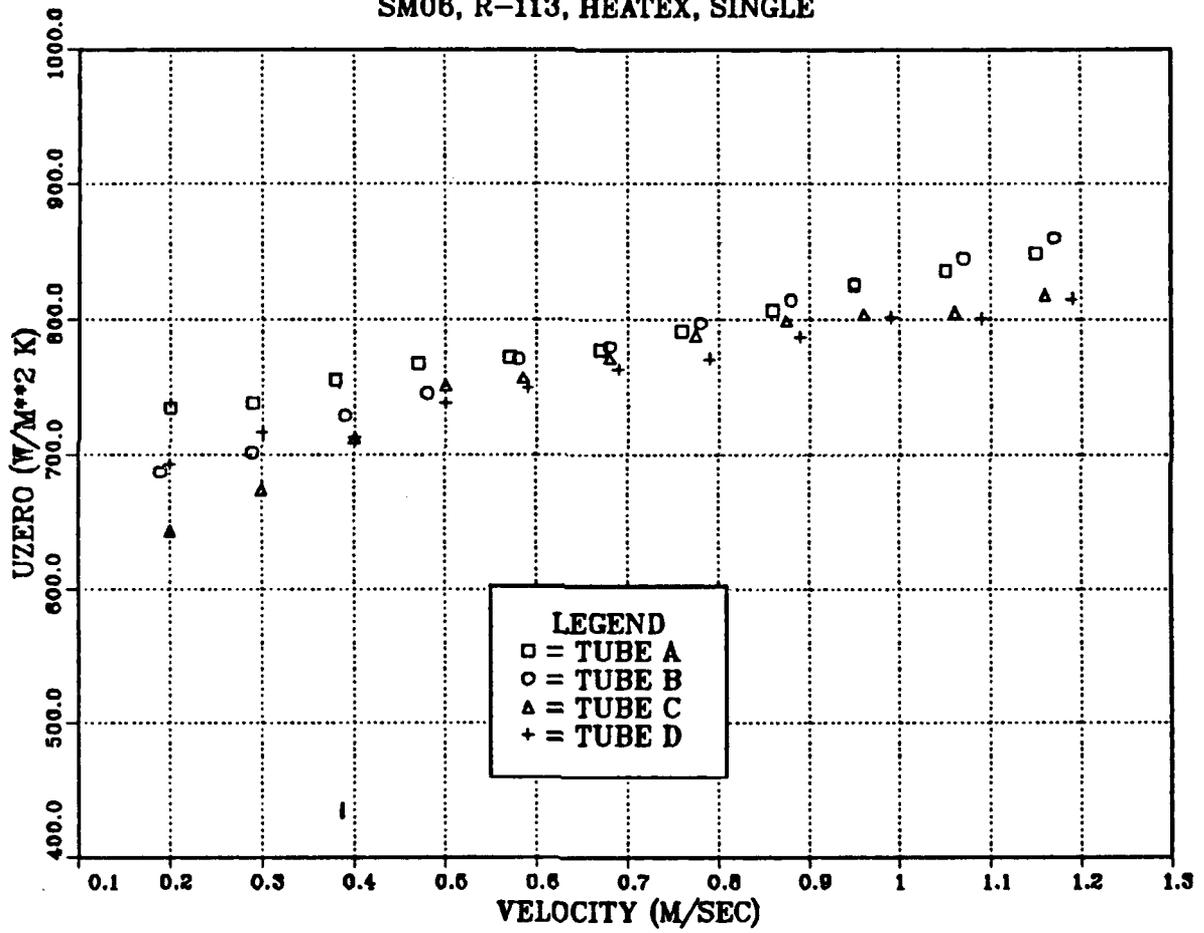
SM06, R-113, HEATEX, SINGLE

Figure 14. Smooth copper tubes with HEATEX installed. Individual operation.

52

from the condenser, cleaned and reinstalled. The results for the duplicate experiment are shown in Figure 15. When these values are compared to Figure 13, differences of the order of 3% is seen in the values of U_o for the top tube. U_o for the lower tubes are slightly larger with a more pronounced difference between the tubes. However, these differences all lie within the 6% uncertainty band indicating that the system gives good reproducibility.

In a similar manner, the results for a repeat experiment on the smooth copper tubes tested individually with the HEATEX installed is shown in Figure 16. When the values for U_o shown here are compared with those shown in Figure 14, no significant differences are seen. Again, this demonstrates the good reproducibility obtained by this system. However, a word of caution is in order. In all the experiments reported here, great care has been taken to control as many variables as possible. Of particular importance is to ensure that the heater power levels are set at the same level for each experiment. This will tend to keep vapor velocity in the condenser at similar levels for each experiment which will result in better repeatability.

The effects of having no insert as opposed to the twisted tape or HEATEX insert is shown in Figure 17. In this figure, the values for U_o for the top tube only have been plotted as a function of coolant velocity for the smooth copper tubes. As previously noted, without any insert, coolant flow is laminar and U_o is relatively constant up to about 0.7 m/s. When either the twisted tape or HEATEX inserts are installed, U_o increases markedly as coolant velocity increases. Interestingly, the HEATEX results in about a 10 to 14% increase in U_o when compared to the twisted tape inserts. Hence, from a heat transfer standpoint, the HEATEX element is the better insert. This is probably because the wire loops which make up the insert, touch much of the inside surface of the tube thereby "scrapping" the thermal boundary layer from the wall. With the twisted tape insert, mixing is provided only by inducing swirling which is probably not as an effective mechanism.

2. Copper/Nickel KORODENSE Tubes

The results obtained for the overall heat transfer coefficient, U_o, for KORODENSE tubes operated as a bundle are shown in Figure 18. This series of experiments was performed without any inserts installed. Note the similarity between these results and those obtained under similar conditions with the smooth copper tubes (Figure 8). Again, values of U_o appear relatively constant for coolant velocities up to 0.7 m/s. As coolant velocity increases further, values of U_o increase dramatically. In addi-

53

UZERO SMOOTH COPPER TUBES
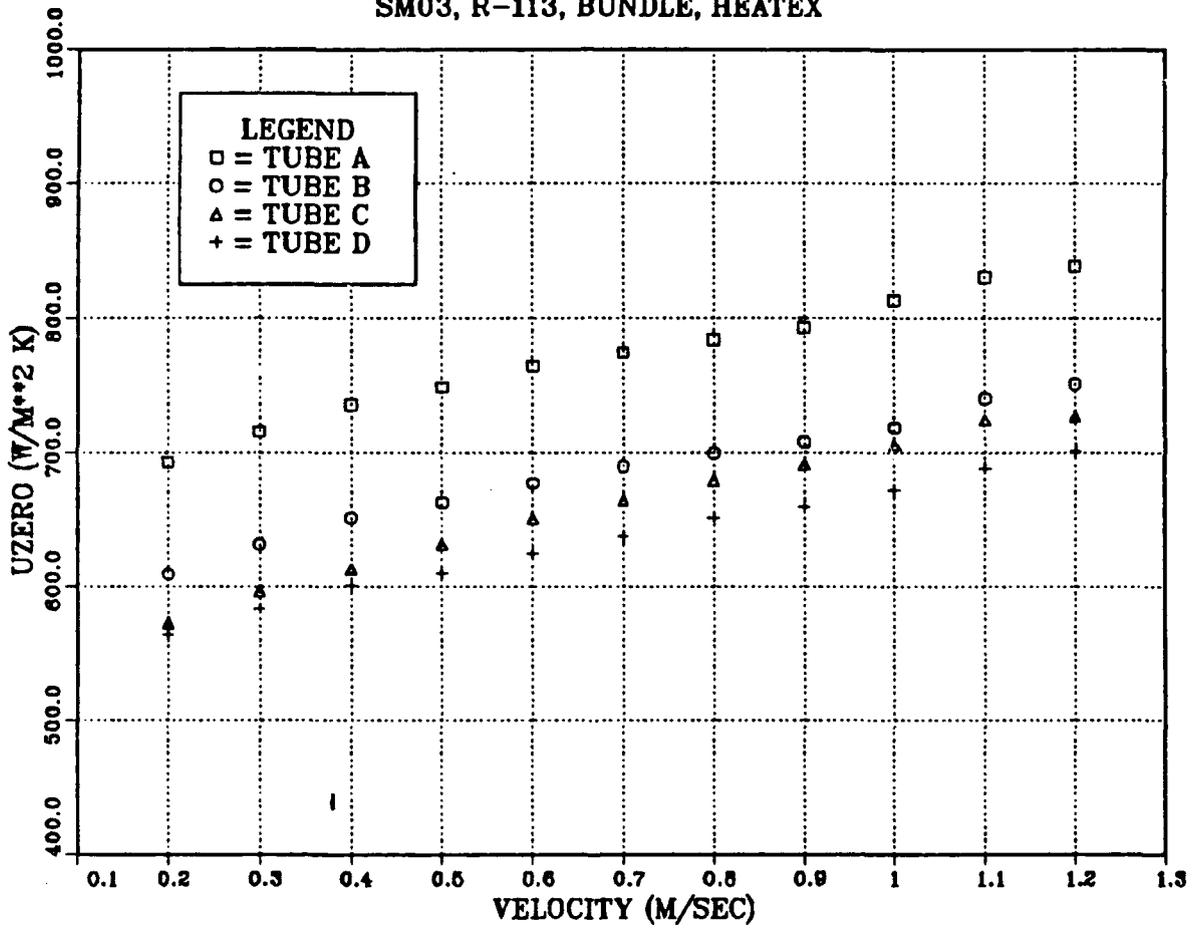
SM03, R-113, BUNDLE, HEATEX

Figure 15.   Repeat run on smooth copper tubes with HEATEX insert.

UZERO SMOOTH COPPER TUBES
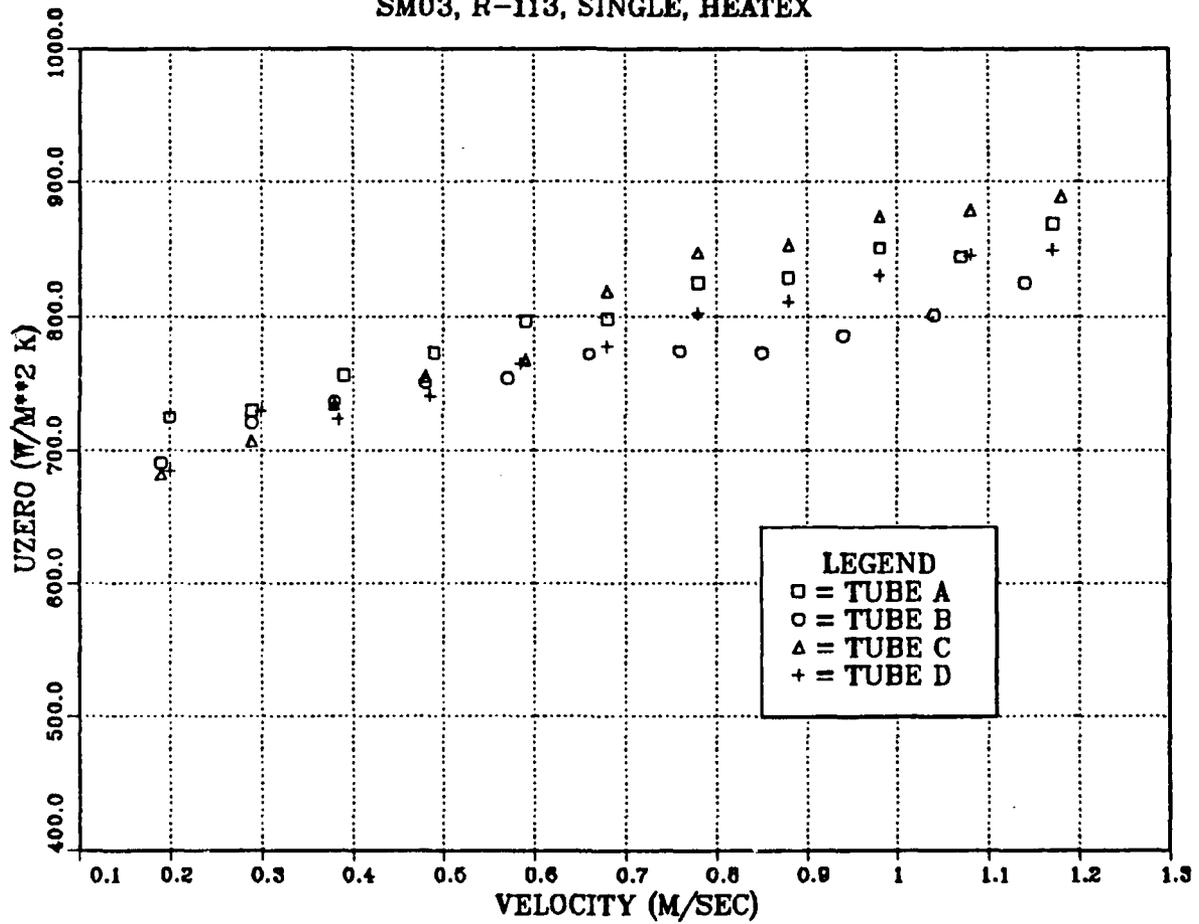
SM03, R-113, SINGLE, HEATEX

Figure 16.   Repeat run on smooth copper tube with HEATEX. Individual runs.
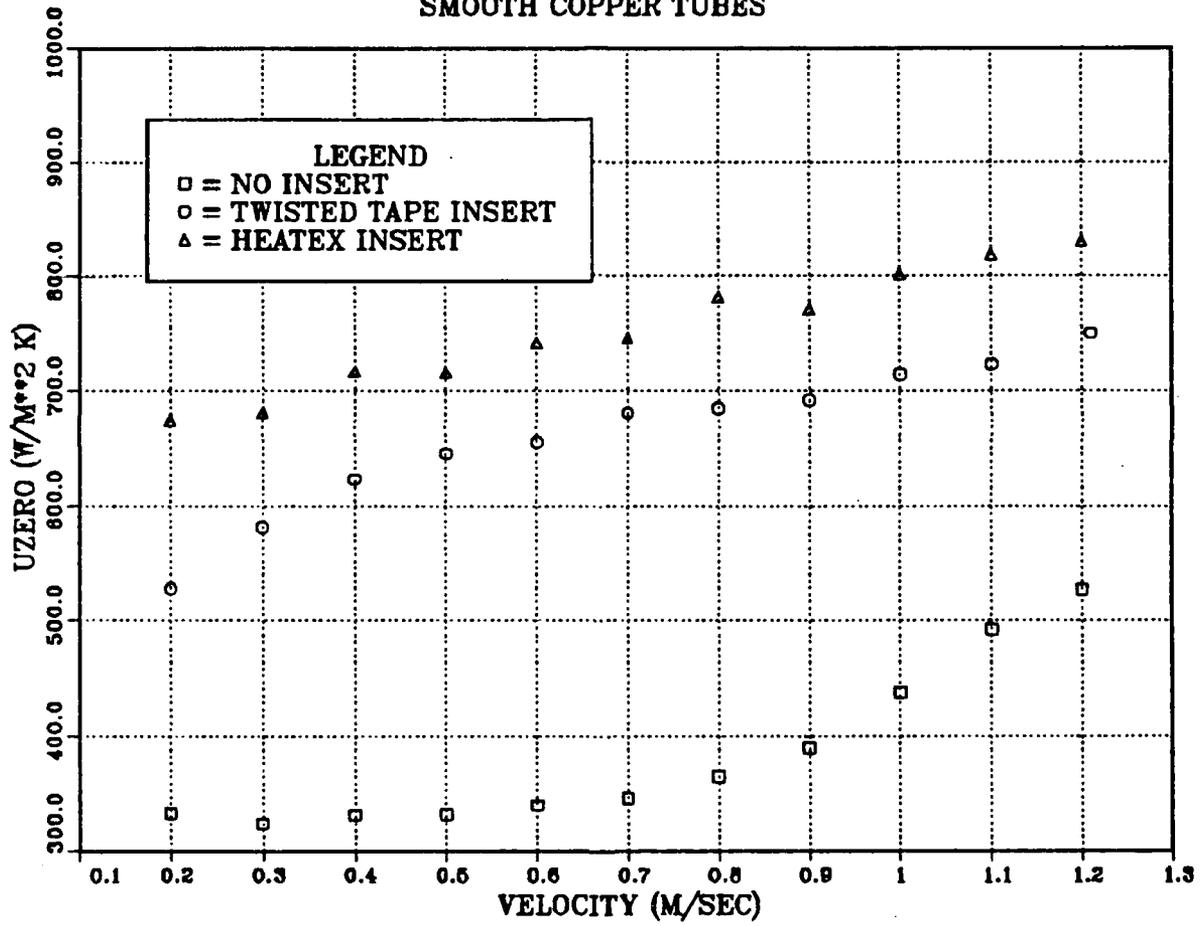
EFFECT OF INSERTS

SMOOTH COPPER TUBES

Figure 17. Effect of tube inserts on U_o

56

tion, an effect of condensate inundation begins to be apparent at these higher coolant velocities.

Results obtained when these same tubes were operated individually are shown in Figure 19. Again, when there is no inundation, all tubes have similar behavior. In addition, the values obtained here match those obtained for the top tube during the bundle experiment (see Figure 18). Interestingly, the relative magnitude of U_o is remarkably similar to that obtained for the smooth copper tubes (Figure 9). This is somewhat surprising. The KORODENSE tubes are corrugated on both the inside as well as the outside. Therefore, one would have expected somewhat higher values for heat transfer with the KORODENSE than with the smooth copper tube.

Values of U_o for the KORODENSE tubes with the twisted tape inserts installed are shown in Figure 20. This data was obtained during bundle operation. As coolant velocity increased, U_o increased steadily. This was true for all four tubes. The effect of inundation reduced U_o by about 15% (tube B relative to tube A). There appeared to be more separation in the values of U_o for tubes B, C, and D than was the case for the smooth copper tubes (see Figure 10) suggesting a greater inundation effect. In comparison with the data obtained for the smooth copper tubes, the KORODENSE tubes resulted in a slight increase (about 6 to 8%) in U_o. This difference may be due to the fact that the insert reduces the inside heat transfer resistance so that the accuracy of the measurement is improved.

The results obtained for these same tubes operated individually are shown in Figure 21. As expected, all four tubes behave similarly when no inundation is present. In addition, the values for U_o match those obtained for the top tube during the bundle operation (Figure 20). In comparison to the smooth copper tube data (Figure 12), the data for the KORODENSE tubes appears somewhat higher (about 6 to 8%). However, given the known uncertainty, this difference is probably not significant.

Results obtained for the KORODENSE tubes during bundle operation with the HEATEX elements installed are shown in Figure 22. Note that U_o again increases markedly for all four tube as coolant velocity increases. An inundation effect can be seen for tubes B, C, and D which is similar to that seen in the twisted tape runs. Tube B shows approximately a 15% decrease in U_o compared to tube A. This effect is magnified at higher coolant velocities (condensation rates) as one would expect. When this data is compared to that obtained for the smooth copper tubes with HEATEX installed (see Figure 13), it can be seen that U_o is greater for the KORODENSE tubes. This suggests that roping of the tube provides some enhancement for the outside (and possibly the

57

UZERO KORODENSE TUBES

KD02, R-113, BUNDLE, NO INSERT
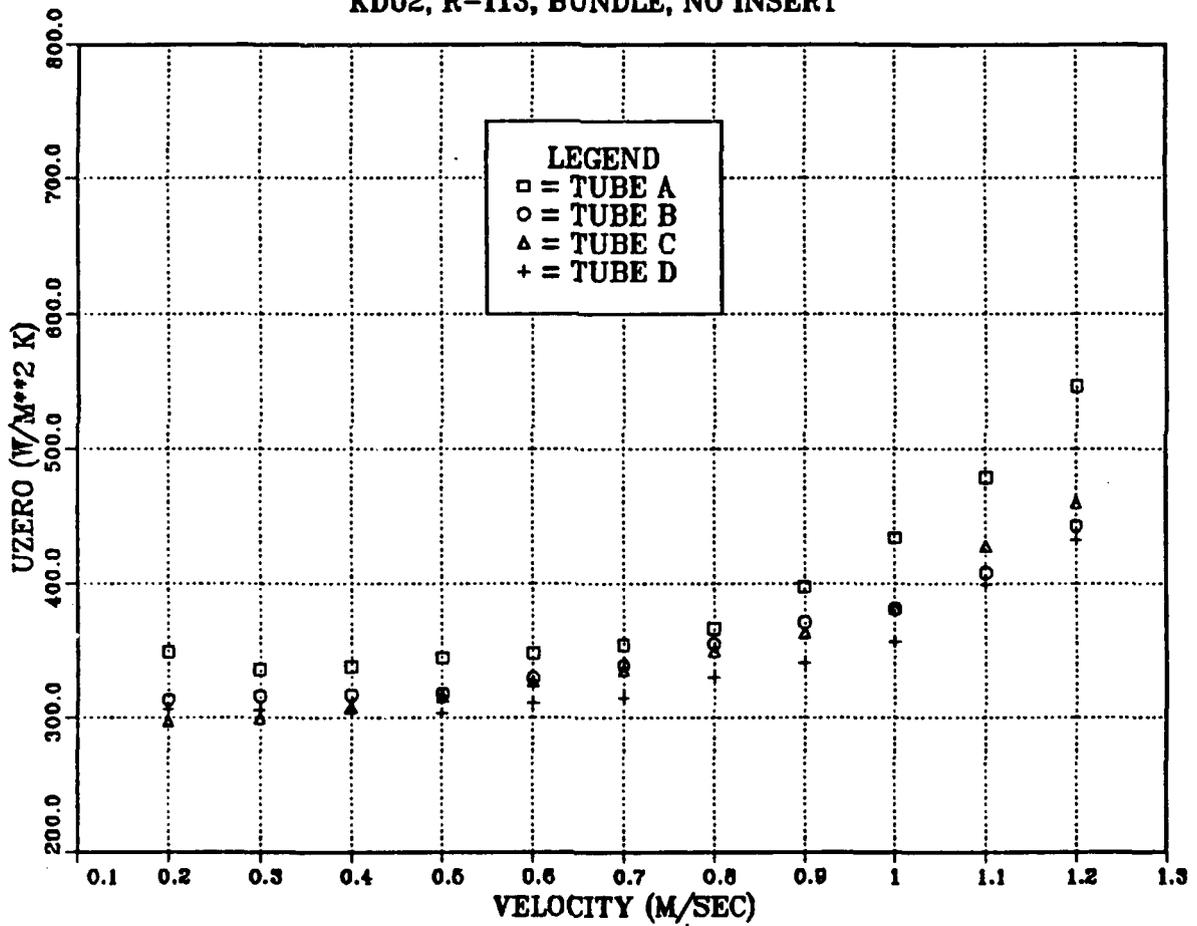
Figure 18.   KORODENSE tubes with no insert, bundle operation.

UZERO KORODENSE TUBES

KD02, R-113, SINGLE, NO INSERT

Figure 19.   KORODENSE tubes operated individually, no inserts installed.

UZERO KORODENSE TUBES

KD04, R-113, BUNDLE, TAPE INSERT

Figure 20.   KORODENSE tubes with tape inserts, bundle operation.

UZERO KORODENSE TUBES

KD04, R-113, SINGLE, TAPE INSERT

Figure 21. KORODENSE tubes with tape inserts, individual operation.

61

inside) heat transfer coefficients. While we expect some small difference, it must be pointed out that the differences observed here are at the bounds of the calculated uncertainty.

Results for these tubes tested individually are shown in Figure 23. As before, the values of U_o are similar for all four tubes and match the values obtained for the top tube during the bundle experiments (see Figure 22). In comparison to the data obtained for the smooth copper tubes, the values of U_o are somewhat higher as expected, but are still at the bounds of the calculated uncertainty.

The effect of using no insert, twisted tape and HEATEX on U_o can be seen in Figure 24. The data are for the top tube only. Note that, as with the smooth copper tubes (Figure 17), the HEATEX results in about a 10 to 14% increase in U_o in comparison to the twisted tape insert. Both provide significant enhancement in U_o when compared to no insert (almost 100% enhancement). As with the smooth copper tubes, when no insert is used, U_o appears relatively constant for coolant velocities over the range of 0.2 to 0.7 m/s indicating laminar flow.

3. Wire Wrapped KORODENSE Tubes

This series of experiments was performed to determine an optimum wire diameter as discussed in "Experimental Procedure". All tubes were tested individually and all had HEATEX mixing elements installed. Figure 25 shows U_o as a function of coolant velocity for the case of an ordinary KORODENSE tube together with tubes wrapped with 0.029", 0.049" and 0.0675" diameter wire, each tested in the top position (tube A). The 0.049" diameter wire gives the biggest enhancement in U_o (about 25%). While the other two wire diameters also provide enhancement in U_o (about 15%), it is below that of the 0.049" wire. The no wire case agrees with the previously obtained data (see Figure 22).

Figure 26 shows similar data but in this case for tube B. Again, the 0.049" diameter wire provides the greatest enhancement in U_o (again about 25%).

Figure 27 shows what happens when the different diameter wires are used on tube C. In this case an interesting fact can be observed. Here, both the 0.049" and 0.029" wire provide virtually the same degree of enhancement for U_o (about 20%) with both being better than the 0.0675" wire. Figure 28 shows the results obtained for tube D. Note that while the 0.049" wire again provides the greatest degree of enhancement for U_o (about 16%), the differences between the three different diameter wires are not as great.

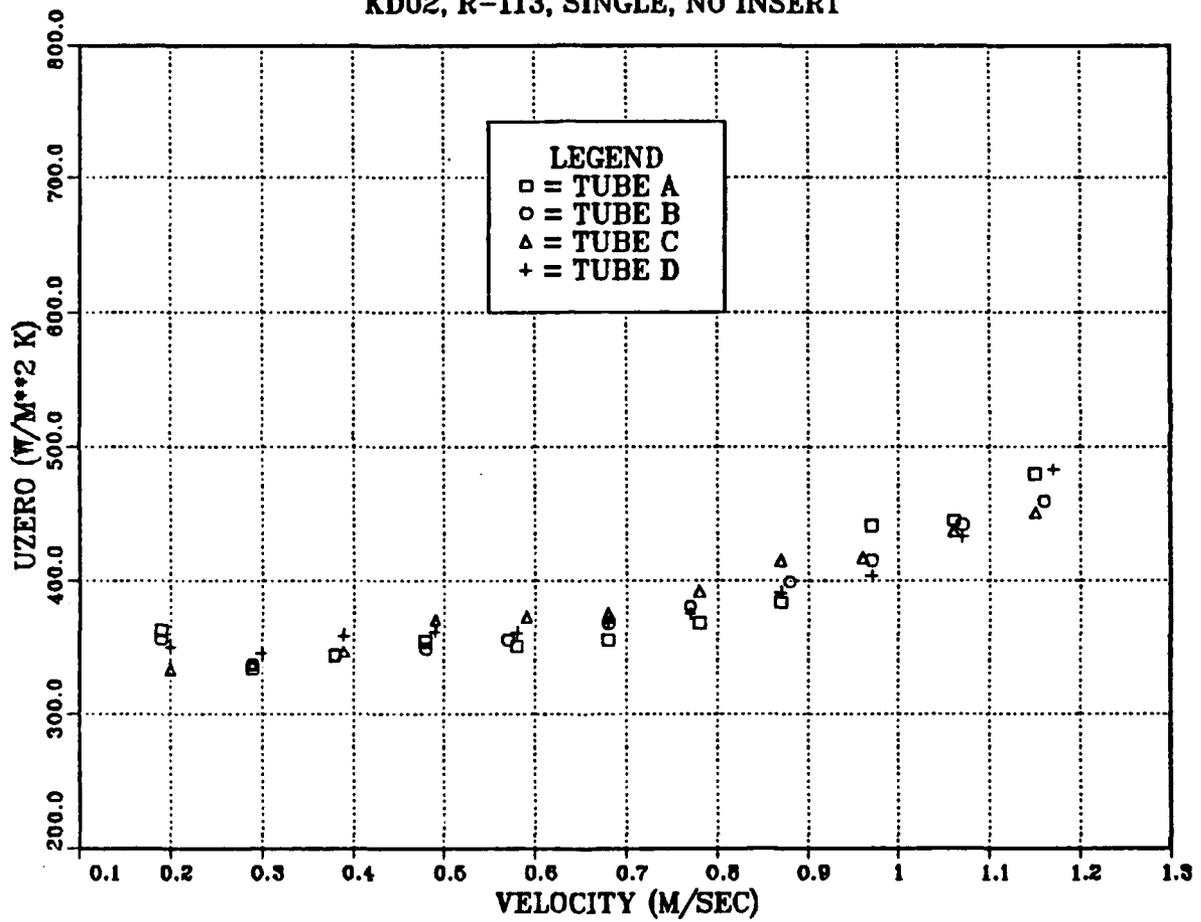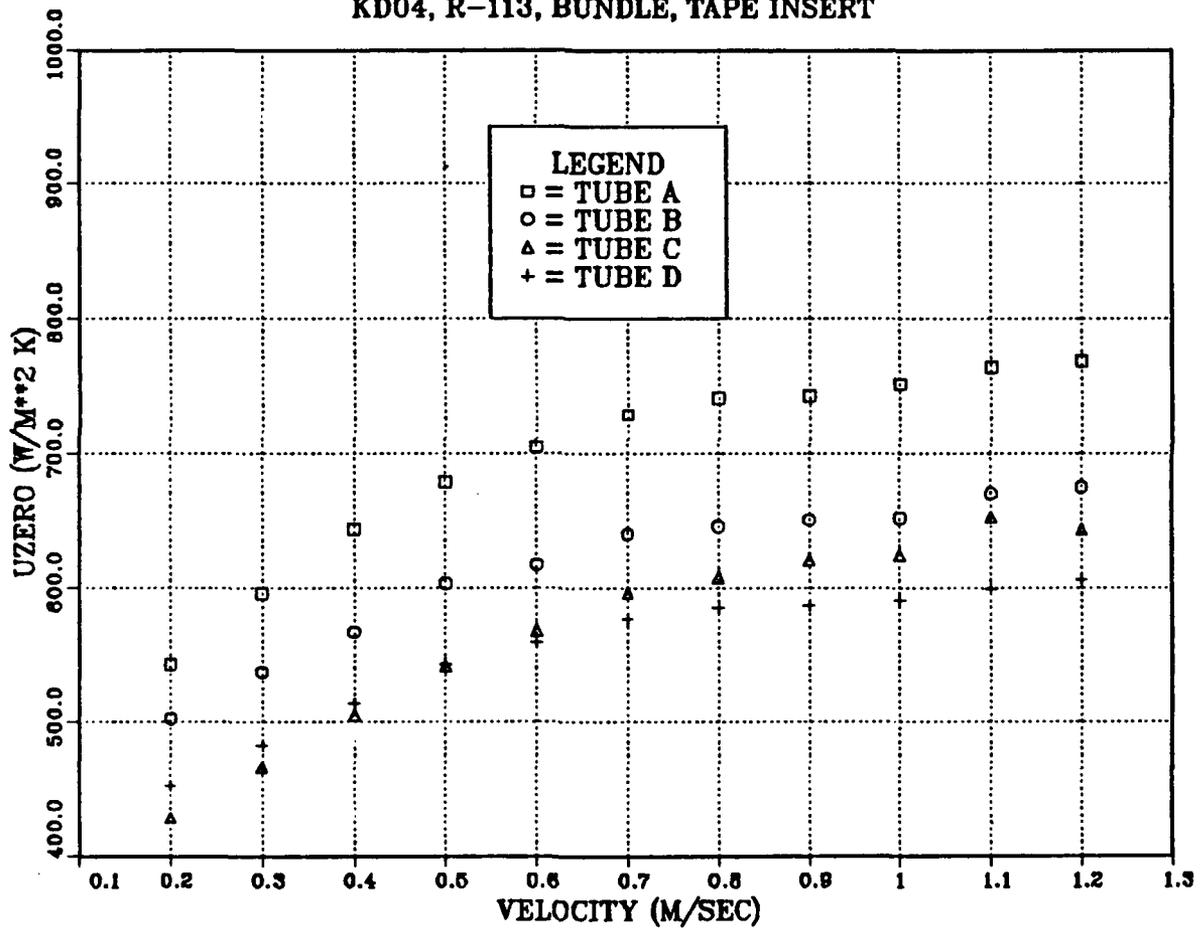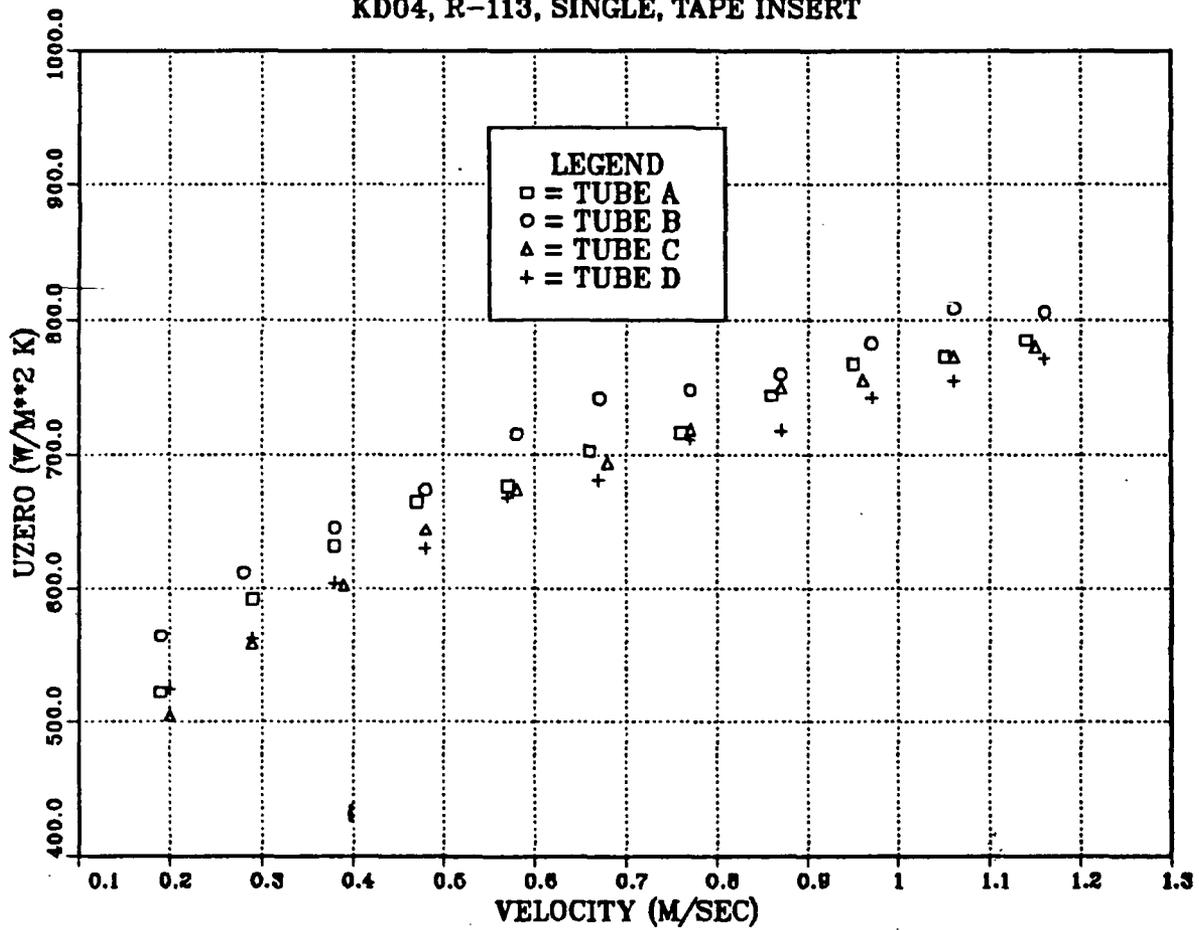# UZERO KORODENSE TUBES

KD03, R-113, BUNDLE, HEATEX
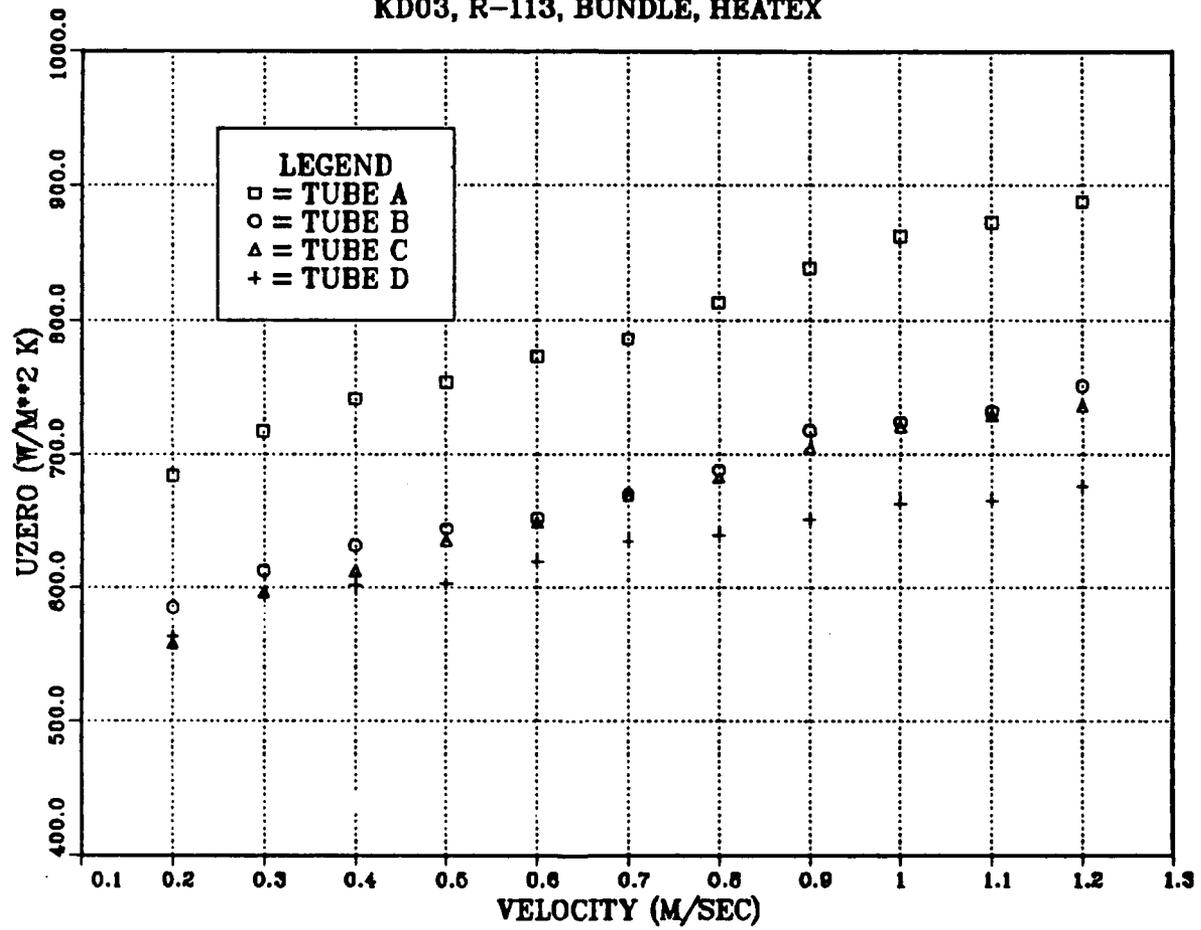
Figure 22. KORODENSE tubes with HEATEX inserts, bundle operation.

UZERO KORODENSE TUBES

KD03, R–113, SINGLE, HEATEX
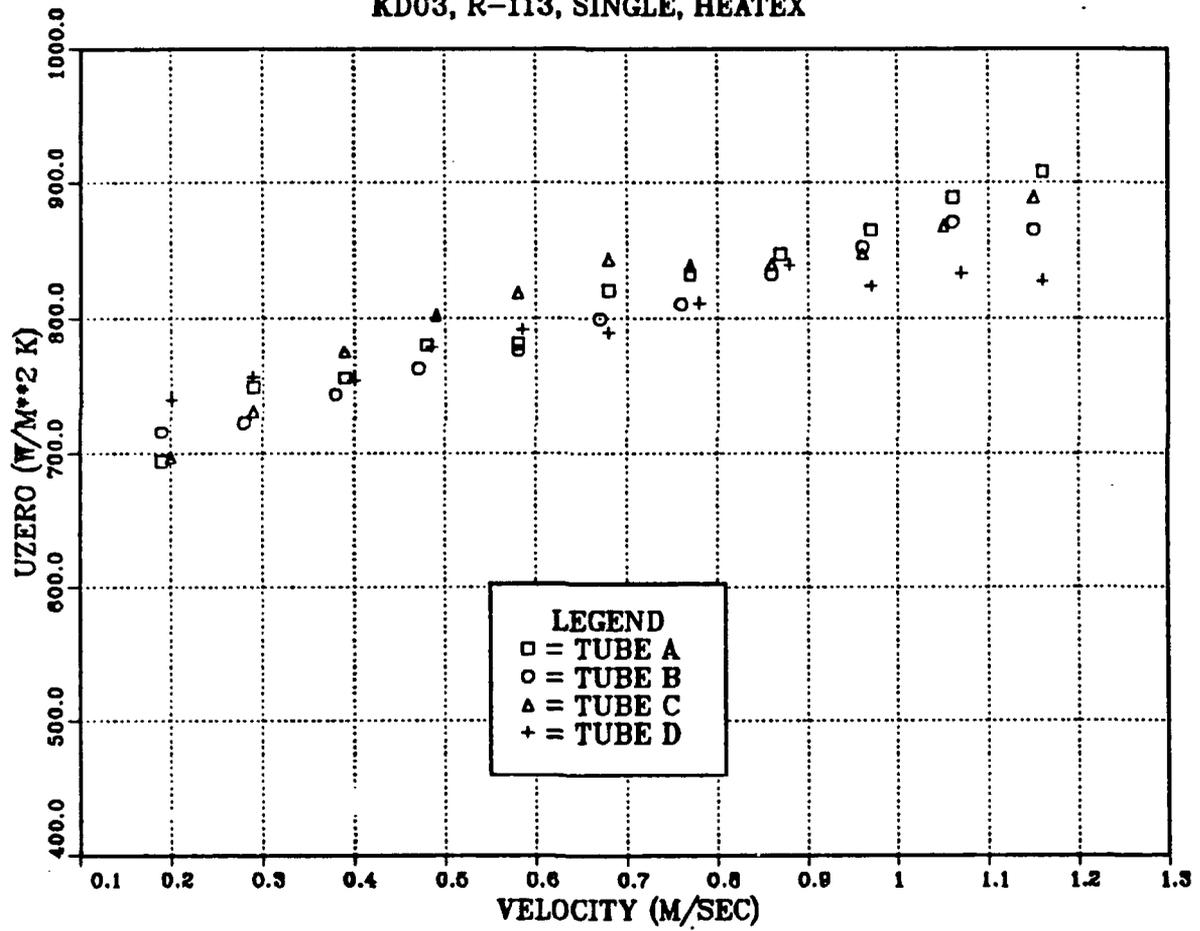
Figure 23. KORODENSE tubes with HEATEX inserts, individual operation.

6-1

EFFECT OF INSERTS

KORODENSE CU/NI TUBES
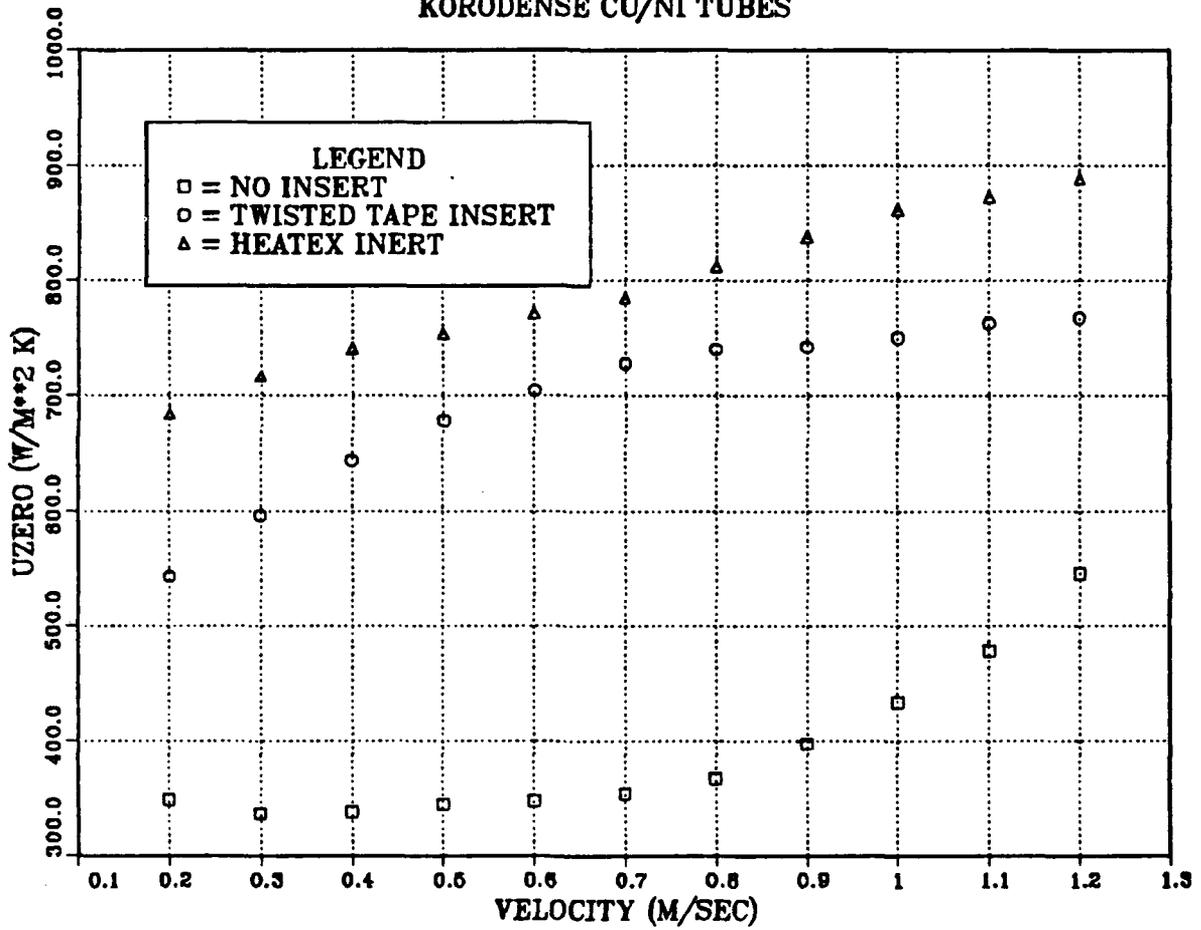
Figure 24. Effect of tube inserts on U_o

UZERO WIRE WRAPPED KORODENSE
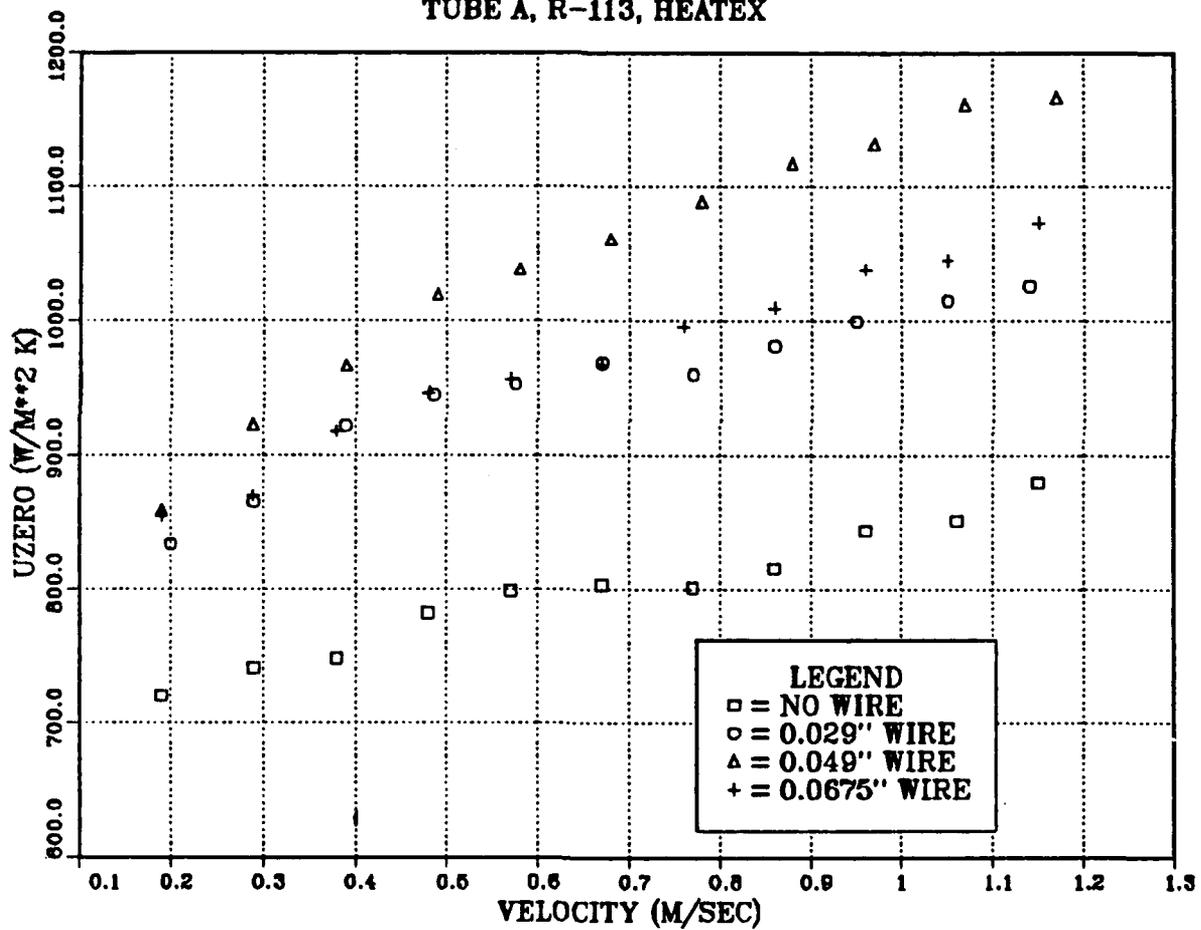
TUBE A, R-113, HEATEX

Figure 25.   Effect of wire diameter on U_o for tube A.

UZERO WIRE WRAPPED KORODENSE
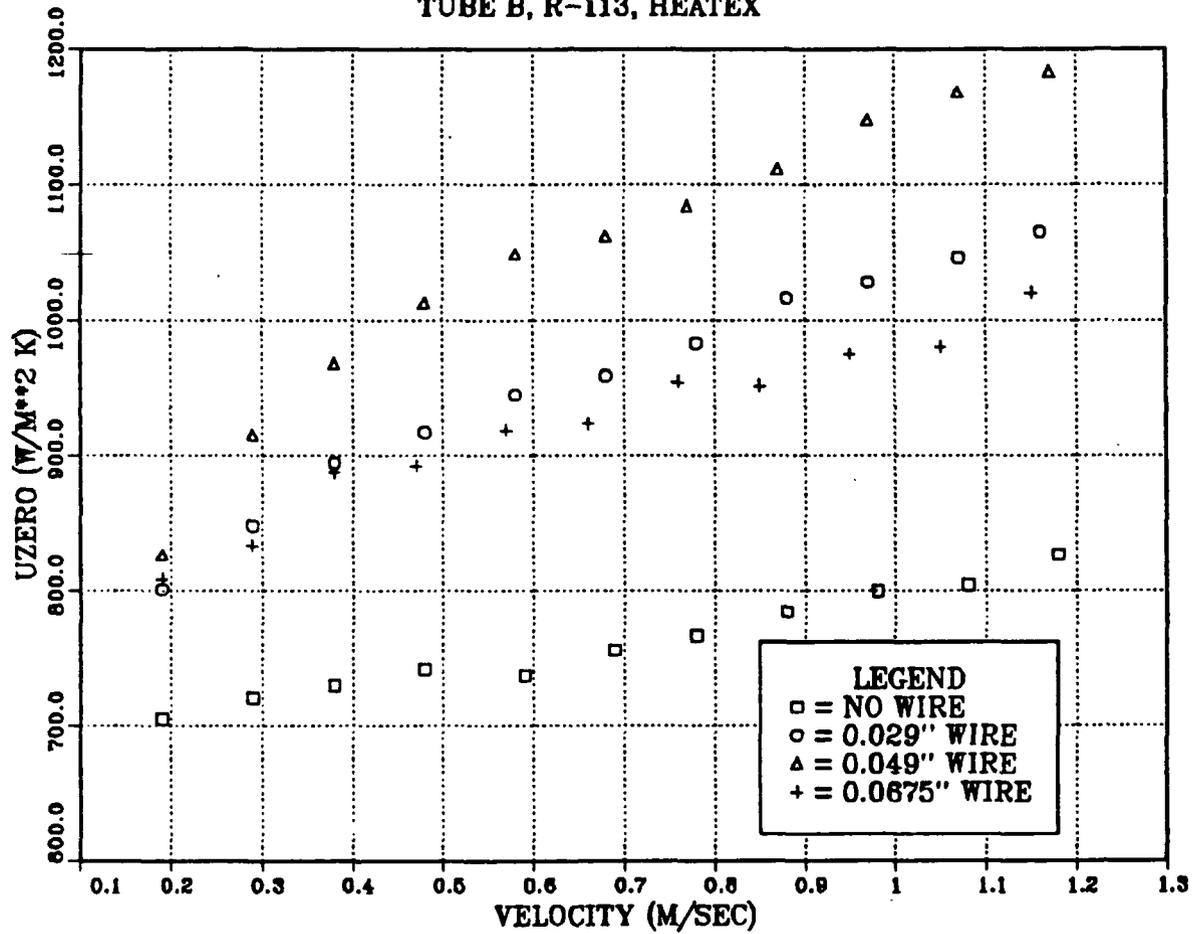
TUBE B, R-113, HEATEX

Figure 26.   Effect of wire diameter on U_o for tube B.

UZERO WIRE WRAPPED KORODENSE
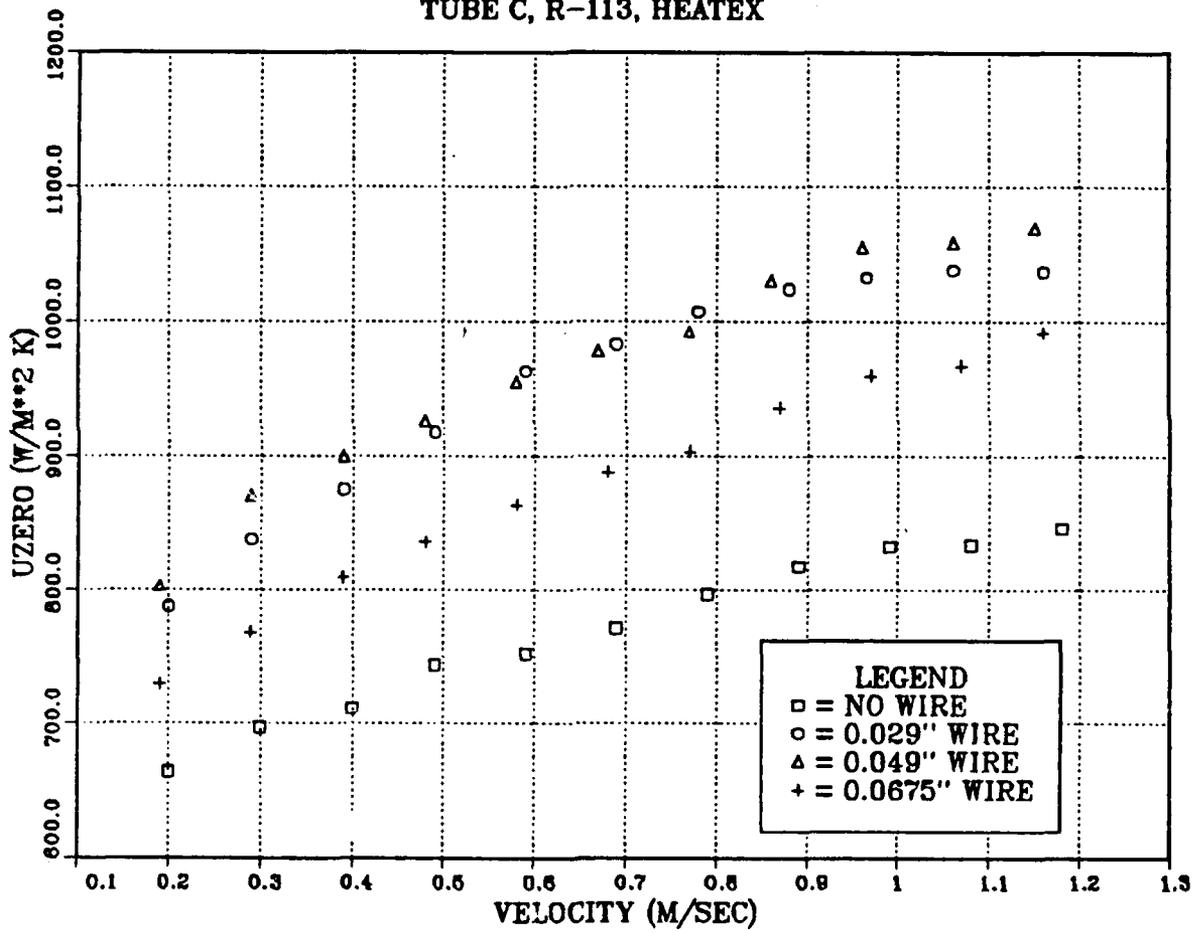
TUBE C, R-113, HEATEX

Figure 27.   Effect of wire diameter on U_o for tube C.

UZERO WIRE WRAPPED KORODENSE

TUBE D, R-113, HEATEX
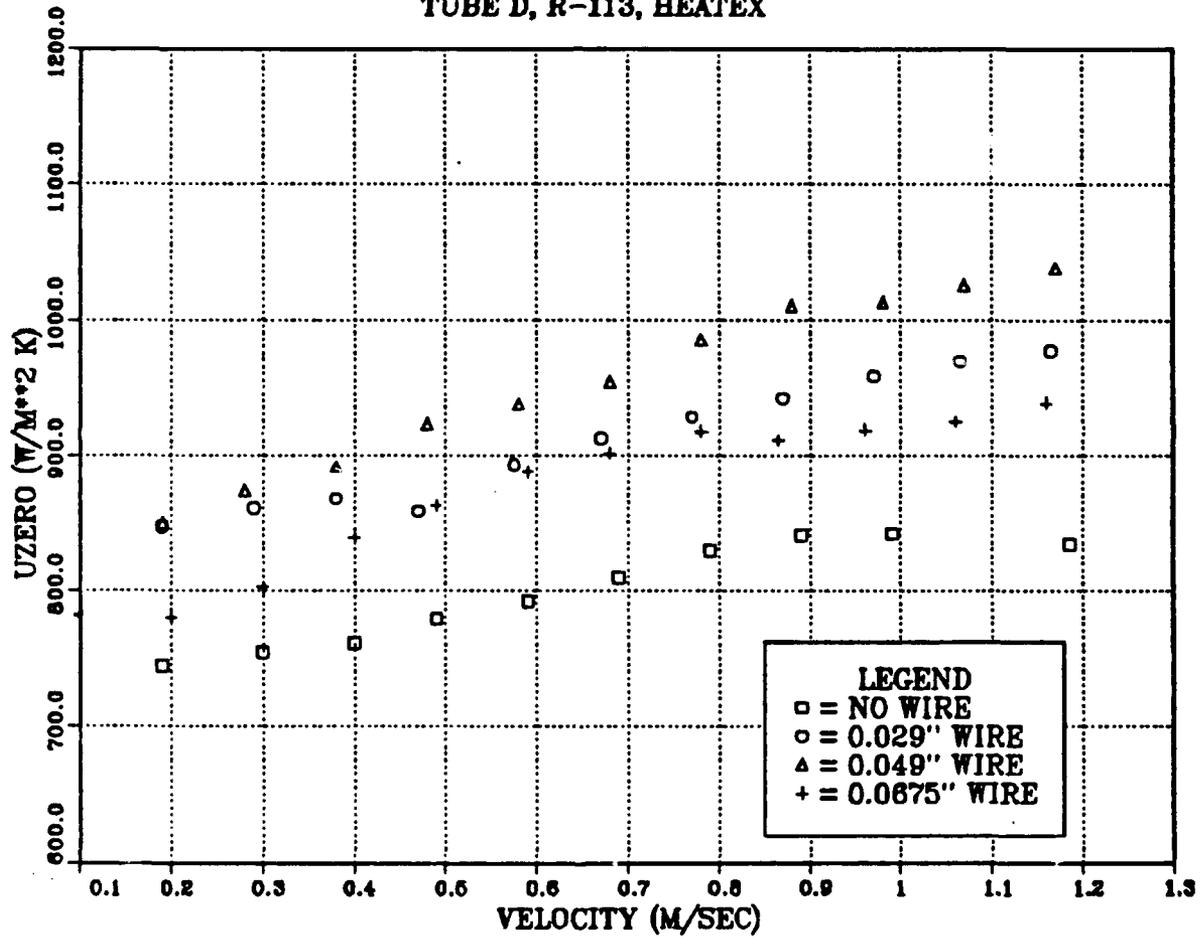
Figure 28. Effect of wire diameter on U_o for tube D.

Figure 27 and Figure 28 suggest that the position of the wire wrapped tube within the bundle may affect the overall heat transfer coefficient. In order to gain further insight on this, the data obtained from these experiments was replotted so that U_o for a given wire diameter was plotted as a function of coolant velocity for all four tube positions. The data for the tube without any wire is shown in Figure 29. The data is all very similar and no distinct trends are evident. Note that comparison of this data with Figure 23 again demonstrates the reproducibility of the system.

Figure 30 (0.029" wire diameter) shows that tubes A, B and C behave similarly. Tube D produces consistently lower values for U_o than the other tubes. Figure 31 (0.049" wire) demonstrates what appears to be a positional effect on U_o. The data for tube A and B is virtually identical and lies above that of tubes C and D which are also virtually identical. Figure 32 suggests that there is also a positional effect for the 0.0675" wire. The top tube seems to provide the greatest enhancement in U_o. The level of enhancement for tubes C and D are approximately the same, both being lower than tubes A and B.

Figure 33 shows U_o plotted as a function of wire diameter for an arbitrarily chosen coolant velocity of 1.0 m/s. Note that the strongest positional affect appears to be for the 0.049" wire. Also, it is clearly evident that the 0.049" wire provides the greatest enhancement for heat transfer.

The possible positional effect may be due to vapor velocity. Although the velocity of the vapor was very small in this study (approximately 0.1 m/s) there may still be an effect. As the vapor penetrates the tube bundle, it slows down. As its velocity is reduced, it can no longer help to strip condensate from the tubes. Hence, heat transfer is decreased. This effect looks much like an inundation effect. An interesting point is why this shows up in these experiments and not with the plain KORODENSE or smooth copper tubes. With the wire wrapped tubes, the condensate layer is presumably thin. Therefore, anything which could help to thin it further would be of importance in improving heat transfer. The loss of this small amount of vapor shearing may be important in this case.

In this study, the 0.049" diameter wire appears to give the best enhancement in heat transfer (see Figure 33). This yields a pitch to wire diameter ratio of 7.2. Fujii et al. [Ref. 33] found that for condensation of R-12 on wire wrapped smooth copper tubes, the optimal pitch-to-wire diameter ratio was on the order of 2. This difference could be due to our use of the KORODENSE tubes. For a pitch-to-wire diameter ratio of 2, we would have had to use wire which was of the order of 5 mm in diameter. This would have

UZERO WIRE WRAPPED KORODENSE

NO WIRE, R-113, HEATEX

Figure 29.   Effect of no wire on U_o for all tubes.

UZERO WIRE WRAPPED KORODENSE

0.029" WIRE, R-113, HEATEX
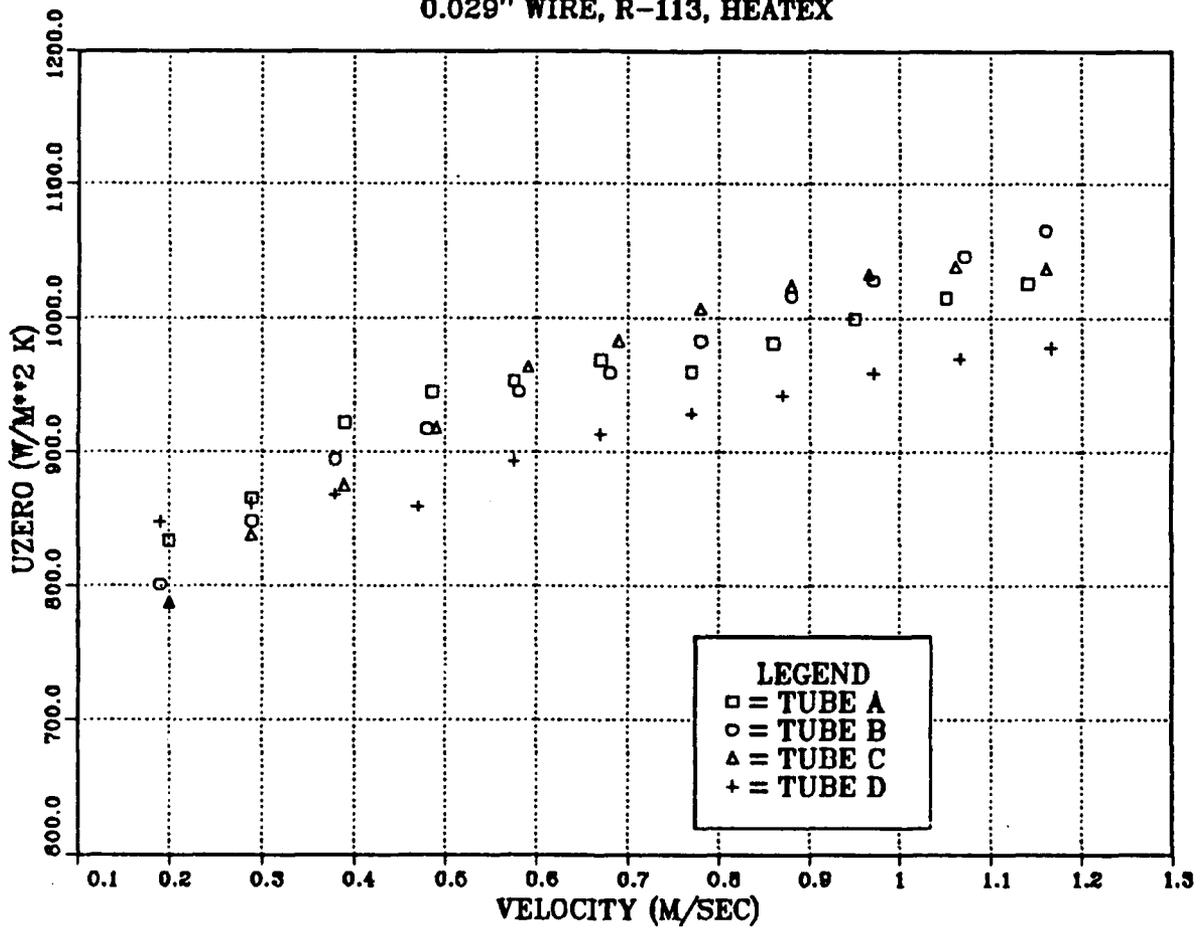
Figure 30. Effect of 0.029" wire on U_o for all tubes.
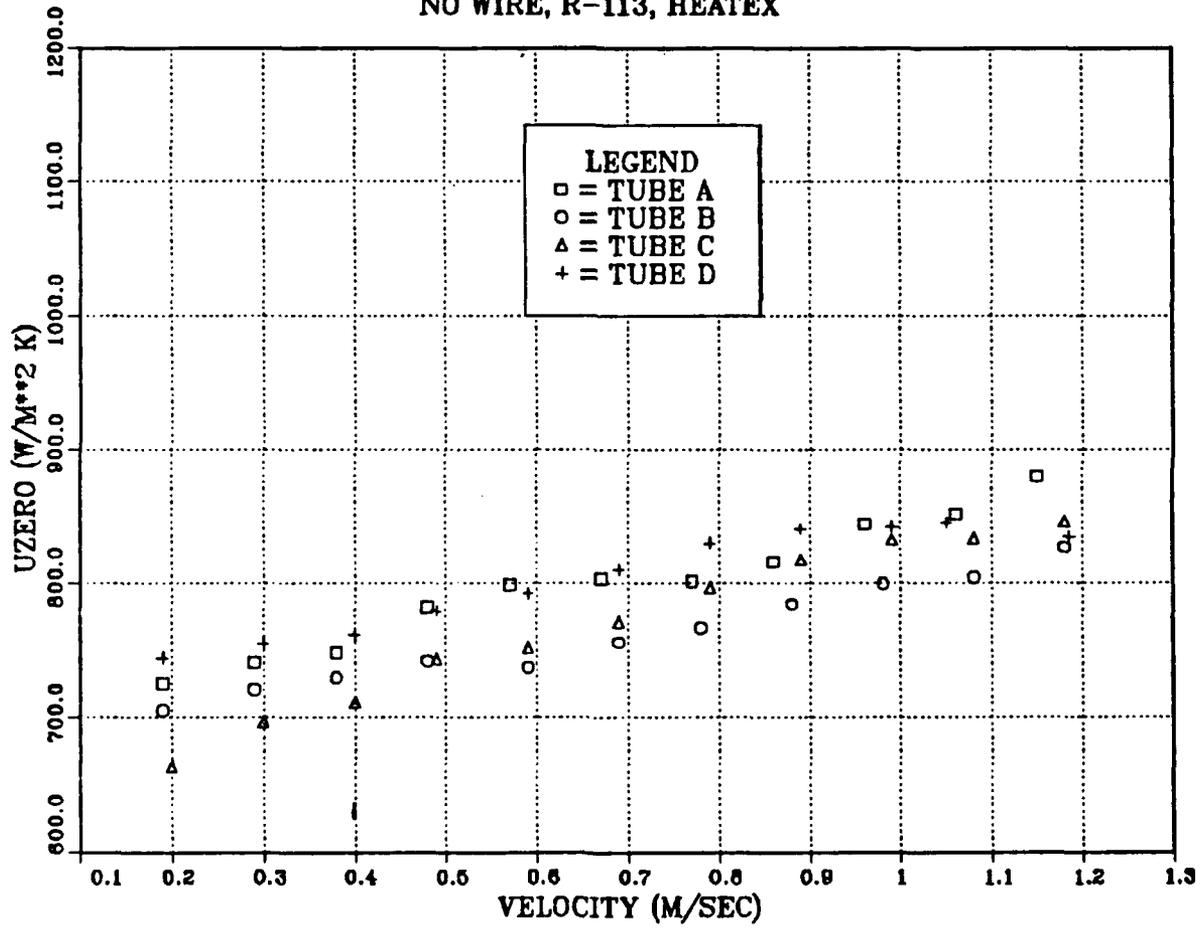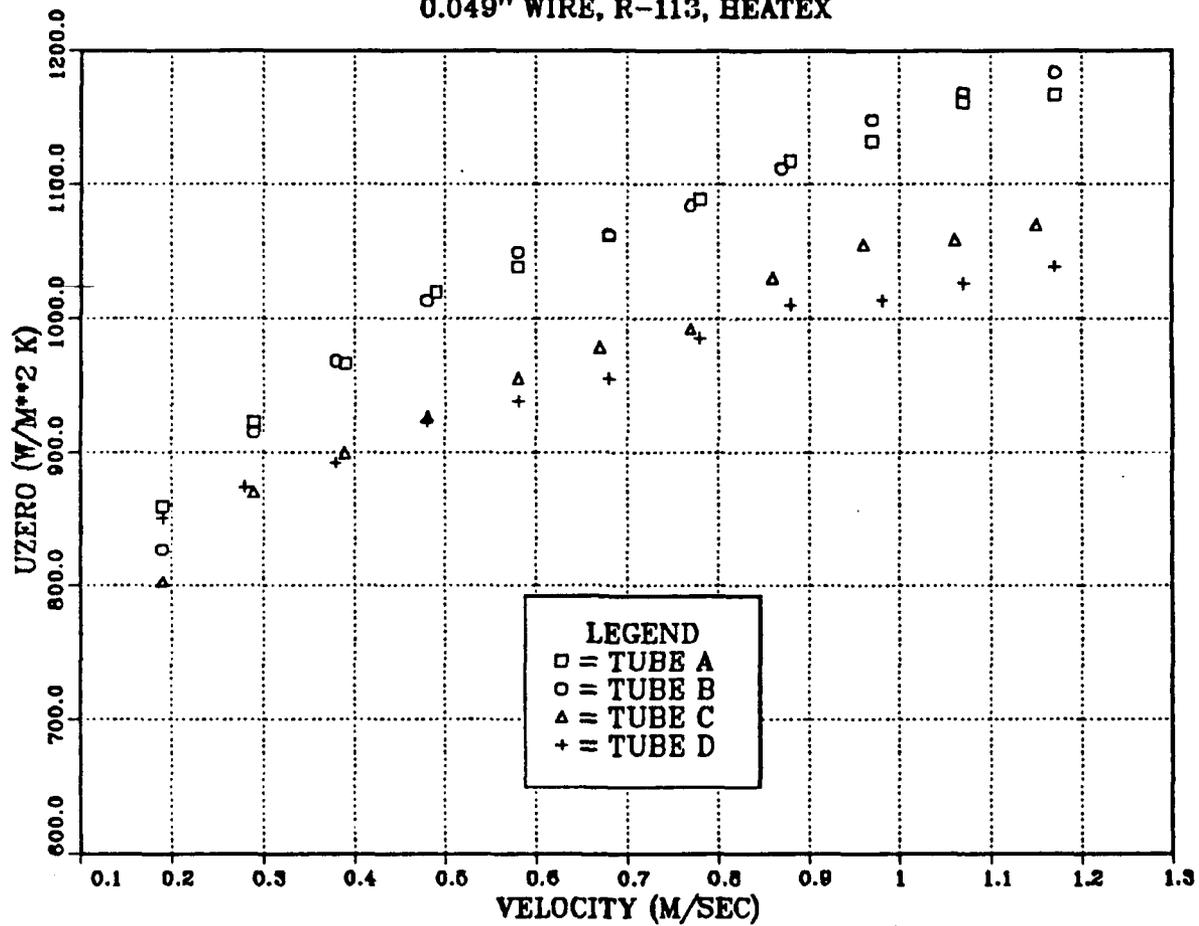
Figure 31. Effect of 0.049" wire on U_o for all tubes.

UZERO WIRE WRAPPED KORODENSE

0.0675" WIRE, R–113, HEATEX
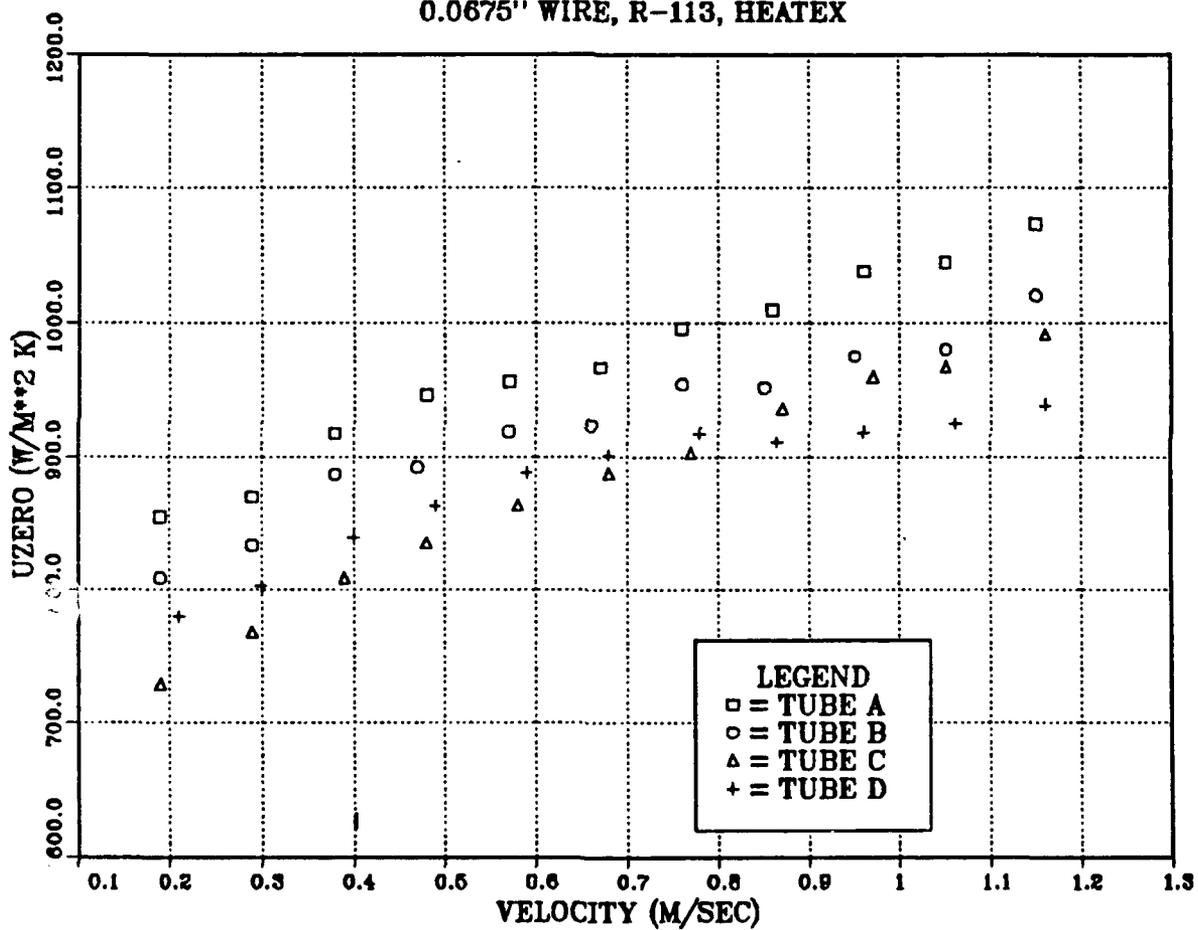
Figure 32.   Effect of 0.0675" wire on U_o for all tubes.

74
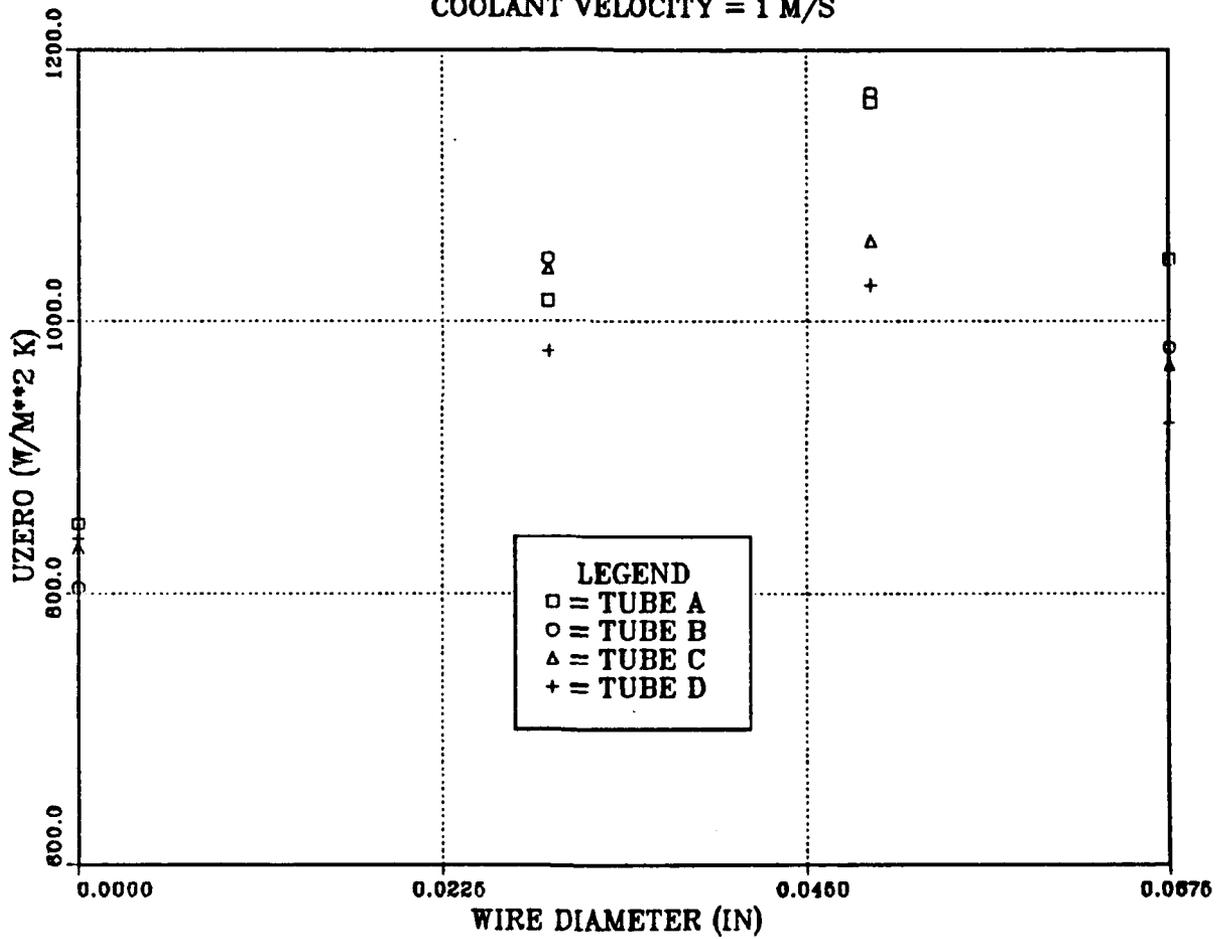
Figure 33. U_o as a function of wire diameter for the four tubes.

been unrealistic for our experiments. It may be that the optimal pitch-to-wire diameter may in fact be about 2. Additional experiments need to be performed with different pitch KORODENSE tubes.

4. KORODENSE Tubes Wrapped with 0.049" Wire

Once the optimal wire diameter had been determined, two identical experiments were performed on the KORODENSE copper/nickel tubes wrapped with the 0.049" diameter wire. The reason that two experiments were performed was to check the repeatability of the data. Figure 34 and Figure 35 show the results from these two experiments for bundle operation of the tubes. The two experiments show excellent repeatability. In comparing these results with the non-wire wrapped tubes (Figure 22), several interesting points can be made. The values of U_o for all four tubes are greater for the wire wrapped tubes than for the plain tubes (approximately 16% for the top tube). In addition, the inundation effect (seen as a lowering of U_o) is reduced for the wire wrapped tubes. Finally, note that the differences between the tubes are more apparent than for the non-wire wrapped tubes. This suggests that the wire wrapping may have an effect upon condensate inundation. This will be addressed in more detail later.

Figure 36 and Figur 37 show the results obtained when the tubes wrapped with the 0.049" diameter wire were operated individually. All the tubes behave similarly and demonstrate good reproducibility between the two runs.

5. Copper/Nickel Finned Tubes

Figure 38 shows the results obtained when no insert was installed and the tubes were operated as a bundle. Comparison of these results with those shown in Figure 8 for the smooth tubes shows that the finned tubes yield higher heat transfer coefficients as expected (approximately 12%). Also, as with the smooth copper tubes, the top tube again displays the highest value of U_o, due to the fact that the lower tubes suffer from condensate inundation. The single tube data for the finned tubes is shown in Figure 39. All four tubes are virtually the same as the top tube during bundle operation.

Figure 40 and Figure 41 show the results obtained for the finned tubes with the tape insert installed during bundle and individual tube operation respectively. In bundle operation, the values of U_o are nearly double those obtained for both the smooth copper tubes (Figure 10) and the KORODENSE tubes (Figure 20) using similar inserts. As with the smooth and KORODENSE tubes, inundation again causes a reduction in U_o. This reduction for the second tube appears to be about 10%. Comparison of the single tube operation (Figure 41) with the bundle operation (Figure 40) shows that all four

0.049" WIRE WRAPPED KORODENSE

KD11, R-113, HEATEX, BUNDLE

LEGEND
□ = TUBE A
○ = TUBE B
△ = TUBE C
+ = TUBE D

UZERO (W/M**2 K)

VELOCITY (M/SEC)
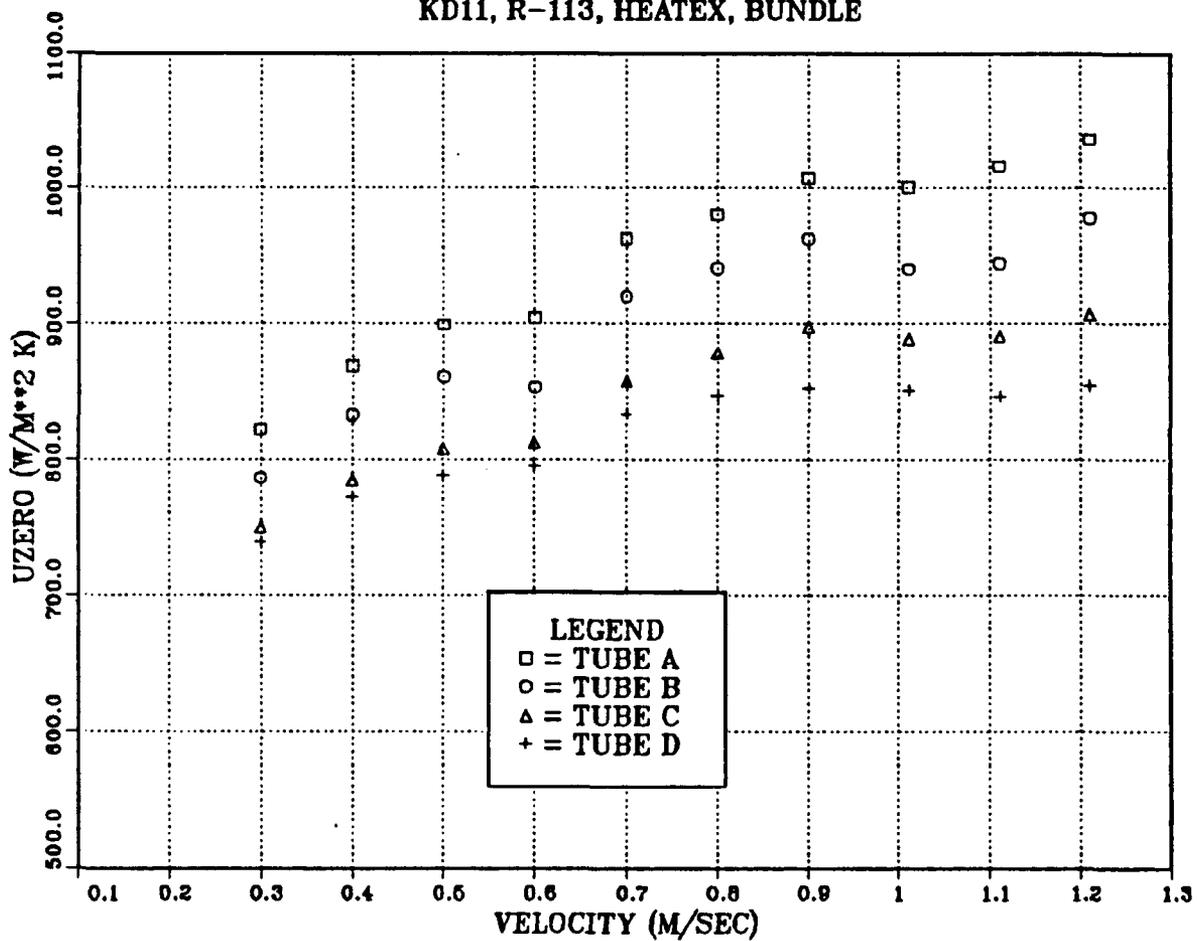
Figure 34.   0.049" wire wrapped KORODENSE tubes, bundle operation, expt. 1.

77

0.049" WIRE WRAPPED KORODENSE

KD12, R-113, HEATEX, BUNDLE
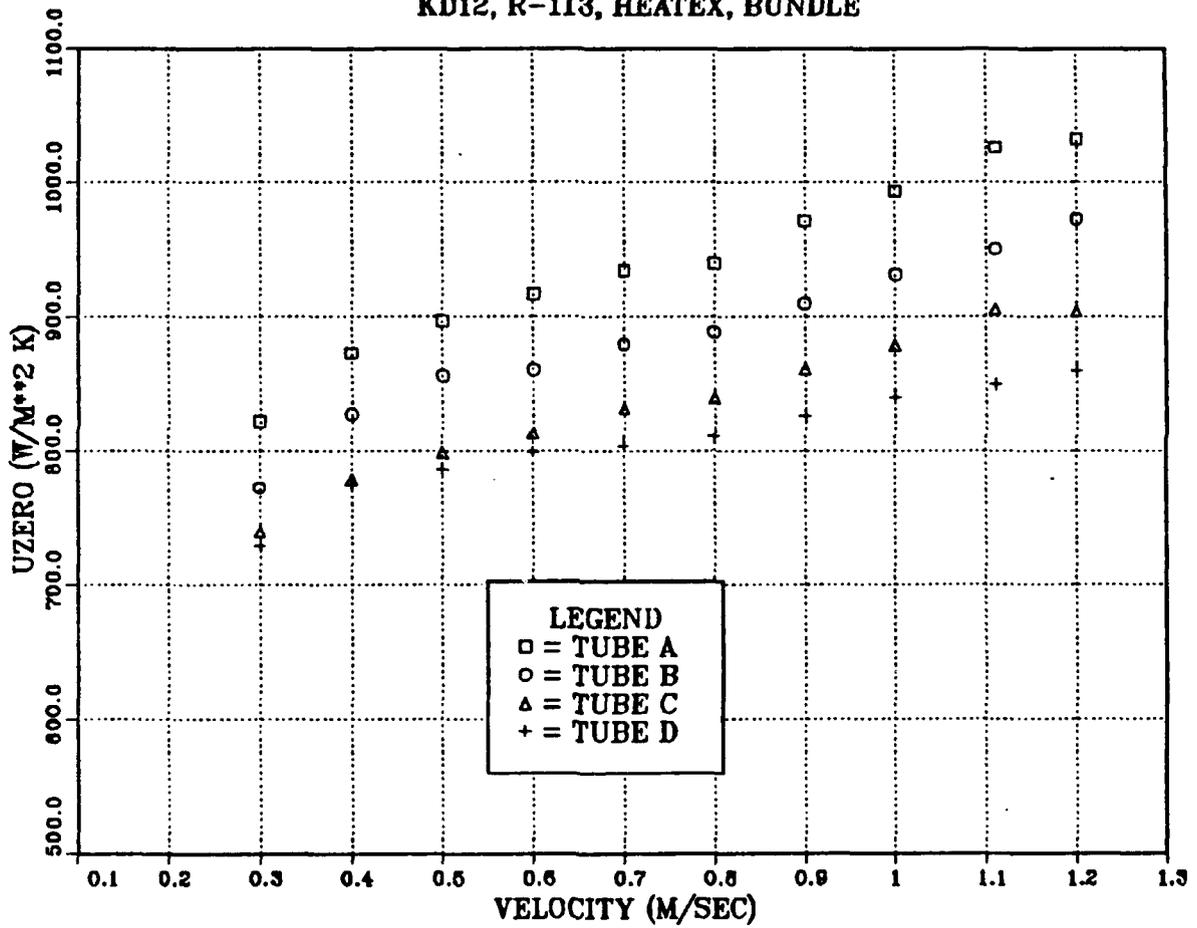
Figure 35.   0.049" wire wrapped KORODENSE tubes, bundle operation, expt. 2.

UZERO WIRE WRAPPED KORODENSE

KD11, R-113, SINGLE, HEATEX

LEGEND
□ = TUBE A
○ = TUBE B
△ = TUBE C
+ = TUBE D
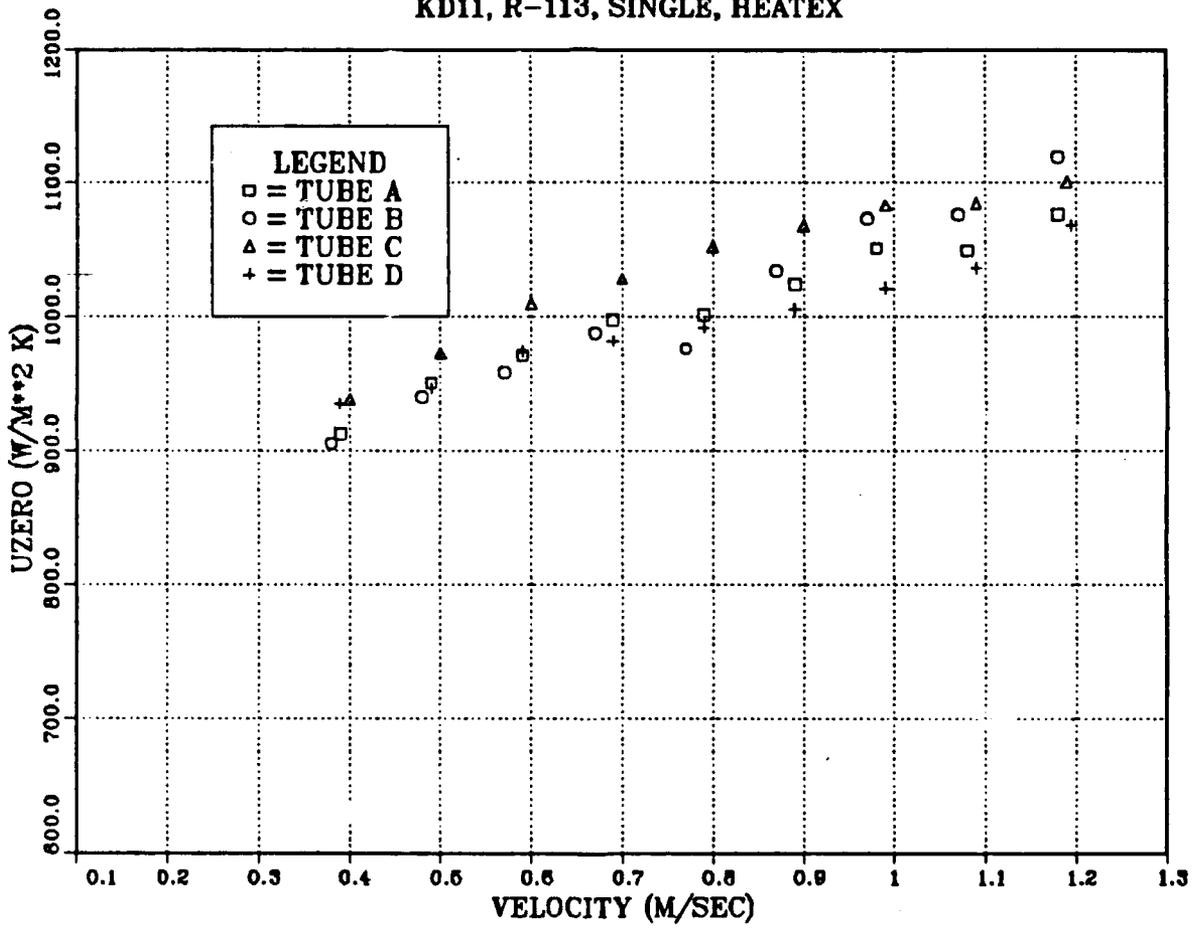
Figure 36.   0.049" wire wrapped KORODENSE tubes, individual operation.

Figure 37. 0.049″ wire wrapped KORODENSE tubes, individual operation.

UZERO CU/NI FINNED TUBES

CF05, R-113, NO INSERT, BUNDLE
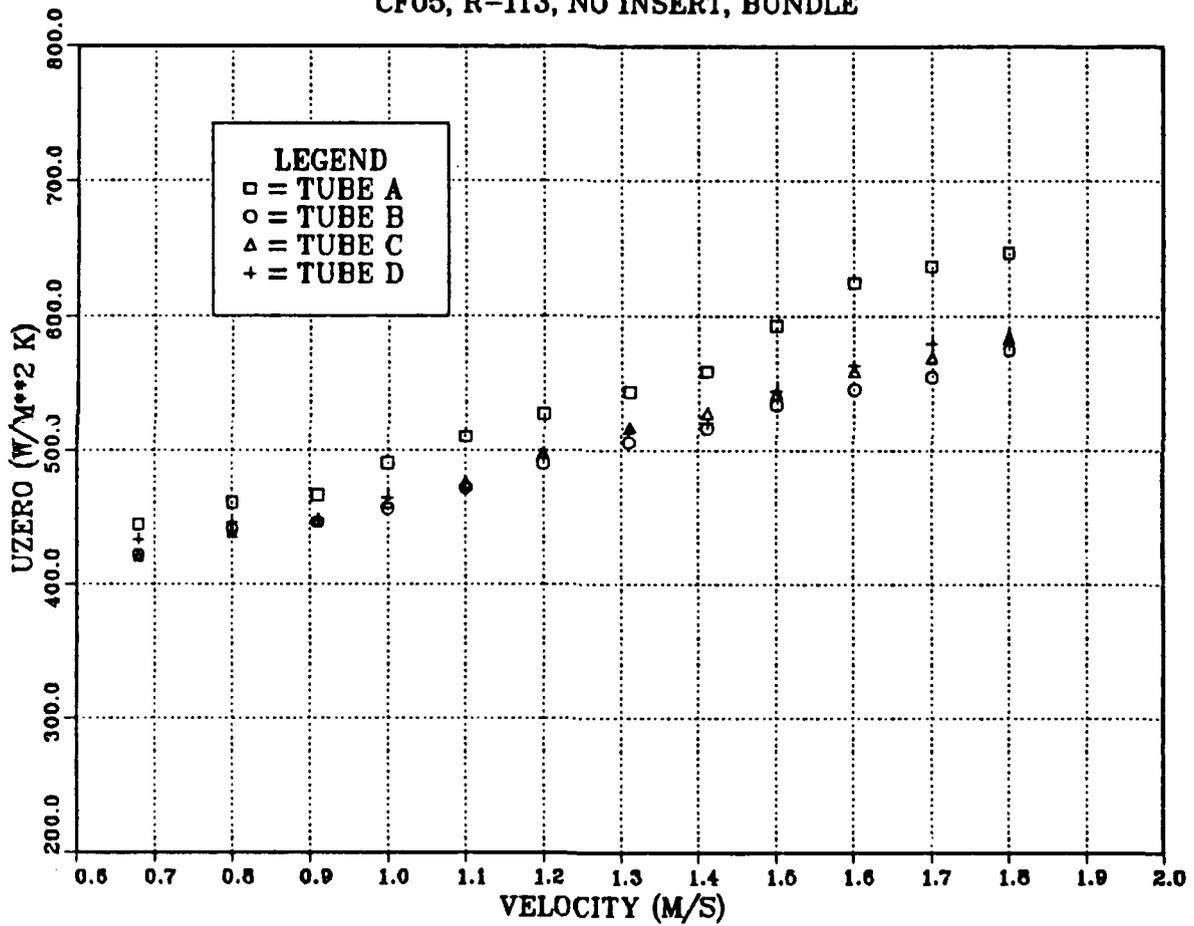
Figure 38.   Copper/nickel finned tubes, no insert, bundle operation.

81

UZERO CU/NI FINNED TUBES

CF05, R–113, NO INSERT, SINGLE
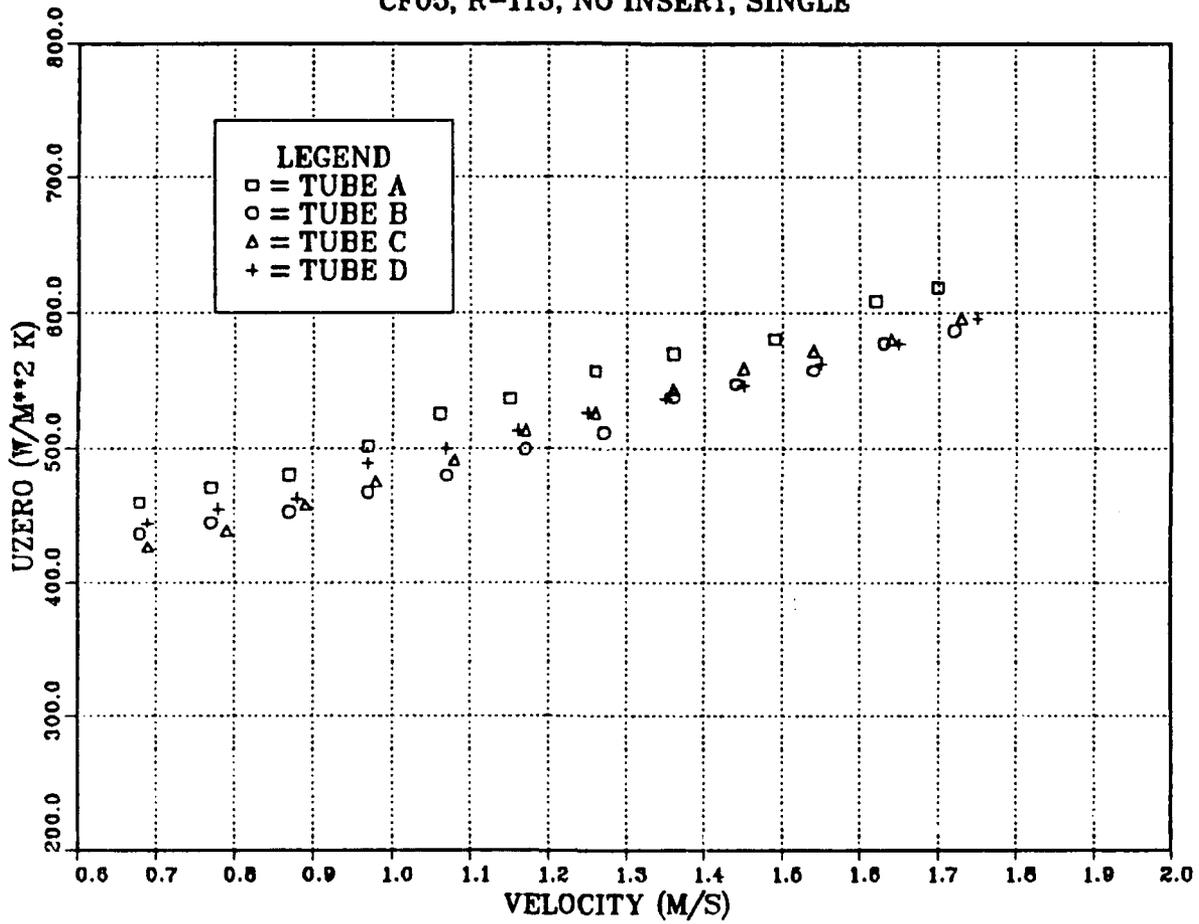
Figure 39.   Copper/nickel finned tubes, no insert, individual operation.

tubes operated individually have approximately the same value of U_o as the top tube during the bundle operation.

Data obtained from experiments on the copper/nickel finned tubes with the HEATEX elements installed are shown in Figure 42 and Figure 43 for bundle and individual tube operation respectively. Note that in comparison with the tape insert experiments (Figure 40 and Figure 41), use of the HEATEX inserts results in about a 20% increase in U_o which again indicates the improvement afforded by using this type of insert. Again the inundation effect results in a decrement of 10 to 14%. During individual tube operation, all four tubes again behave like the top tube during the bundle operation.

As with the smooth copper tubes and the KORODENSE tubes, the effects of using different types of inserts can be summarized. This has been done in Figure 44. This data is for the top tube only. The similarity with data from the other tubes (see Figure 17 and Figure 24) is good with HEATEX yielding approximately a 20% increase in U_o over the twisted tape insert and a remarkable 200% increase over the case with no insert!

C. OUTSIDE HEAT TRANSFER COEFFICIENT

The following figures deal with the outside heat transfer coefficient, h_o. In all cases, except as specifically noted, the outside heat transfer coefficient was determined using the modified Wilson Plot procedure. As before, tube A is the top tube with tube D being the bottom tube in the bundle. In addition, no values were obtained for h_o for experiments which had no insert. The reason for this is that, for the coolant velocities used, without inserts, the inside resistance was simply too large for any accurate results to be obtained for h_o.

1. Smooth Copper Tubes

Figure 45 and Figure 46 show the values obtained for h_o for the smooth copper tubes with the tape insert installed during bundle and individual tube operation respectively. Several features deserve comment. First, values of h_o for the top tube are below those predicted by Nusselt theory. These differences range from about 15% at low coolant velocities to about 7% at the higher coolant velocities. In addition, note the inundation effect (seen as a reduction in h_o for tubes B, C, and D). Reductions in h_o range from 28% to 19% as coolant velocity increases from 0.2 to 1.2 m/s respectively. As was the case for the overall heat transfer coefficient, when the tubes were operated individ-

UZERO COPPER/NICKEL FIN TUBES

CF03, R-113, TAPE INSERT, BUNDLE
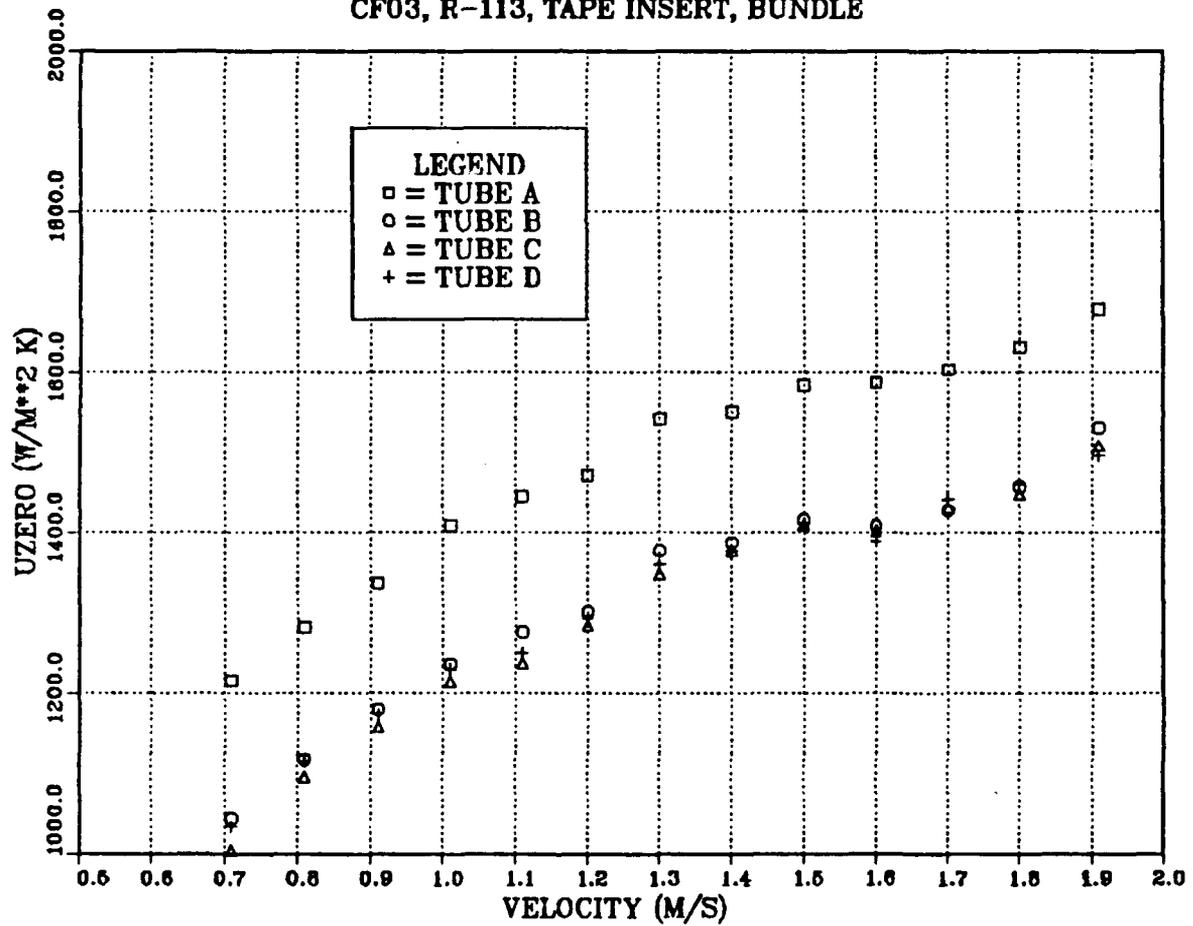
Figure 40.   Copper/nickel finned tubes, tape insert, bundle operation.

UZERO COPPER/NICKEL FIN TUBES

CF03, R-113, TAPE INSERT, SINGLE
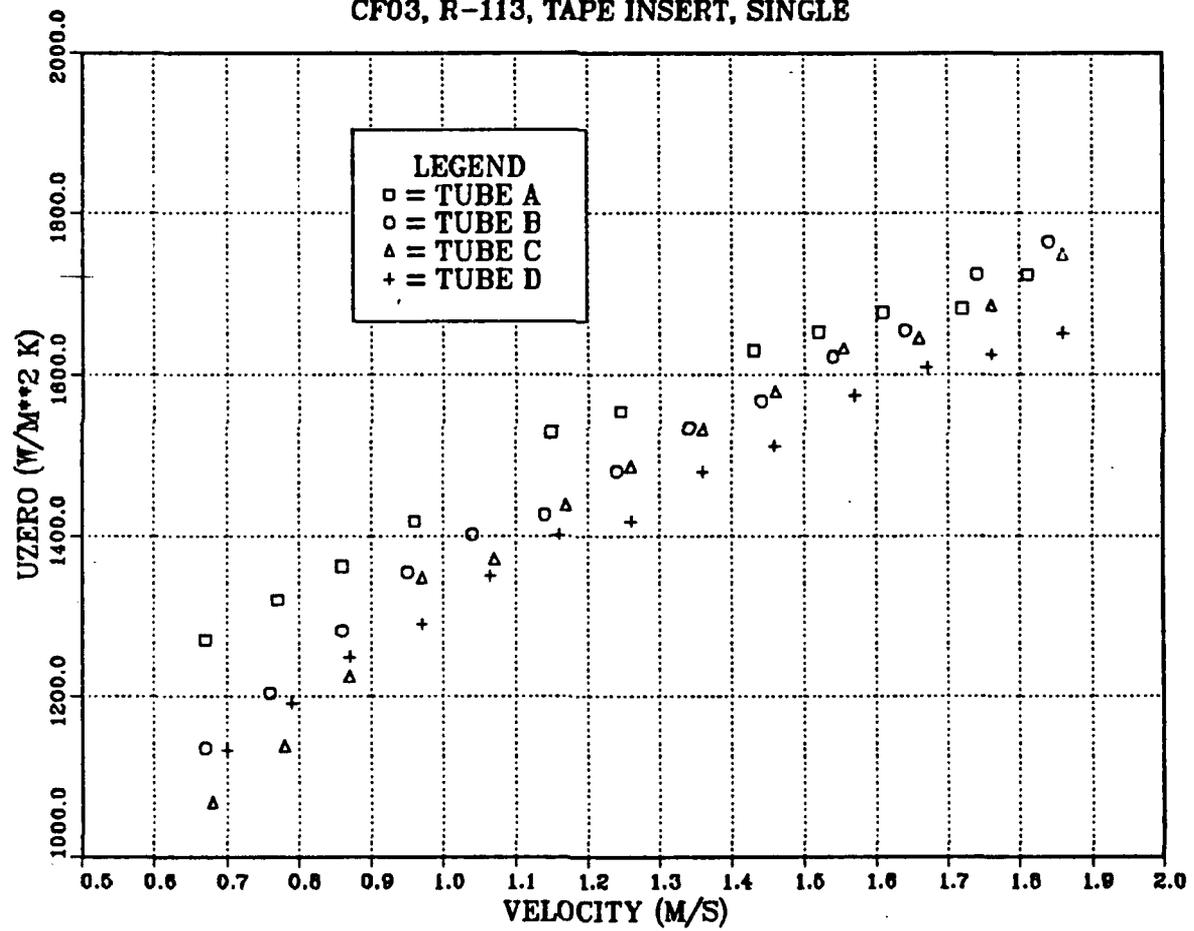
Figure 41. Copper/nickel finned tubes, tape insert, individual operation.

85

UZERO COPPER/NICKEL FIN TUBES

CF04, R-113, HITRAN, BUNDLE
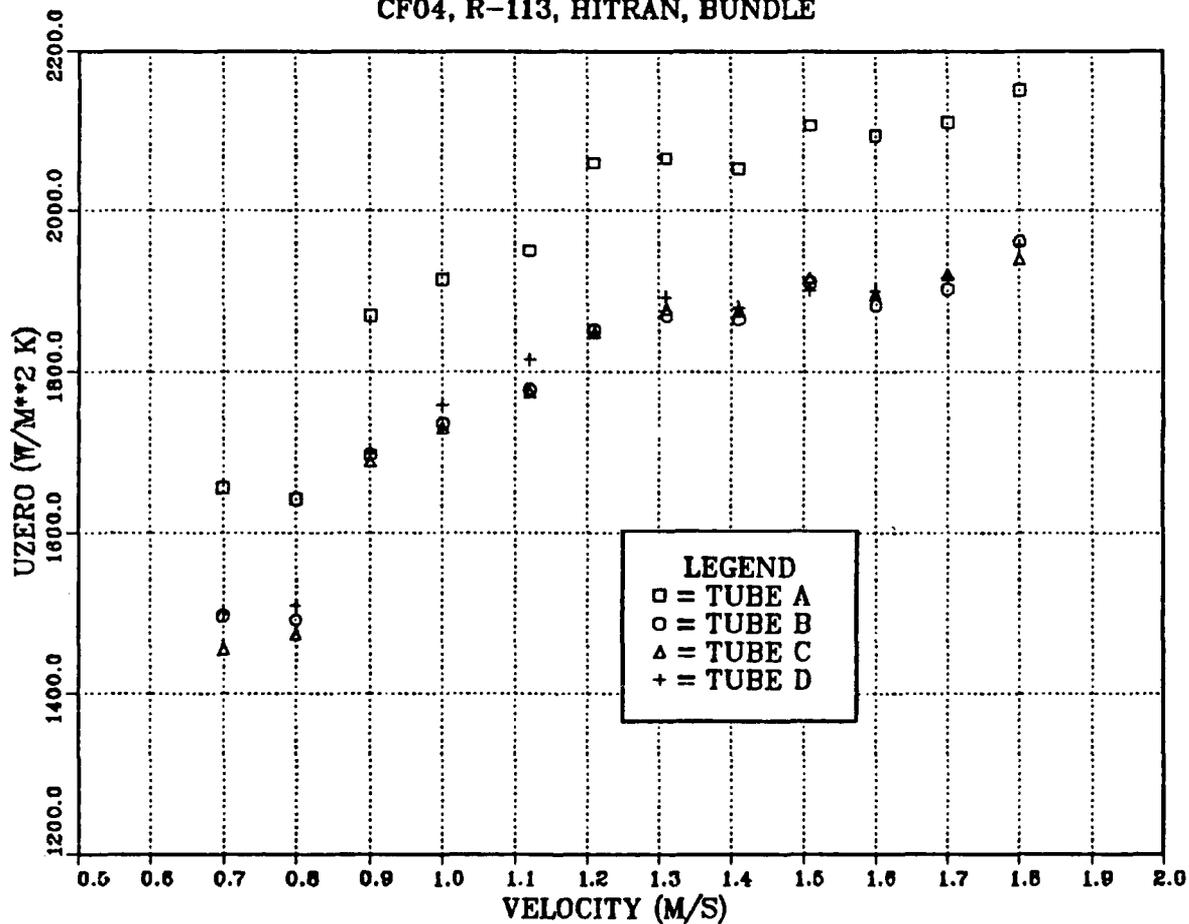
Figure 42. Copper/nickel finned tubes, HEATEX insert, bundle operation.

UZERO COPPER/NICKEL FIN TUBES

CF04, R-113, HITRAN, SINGLE
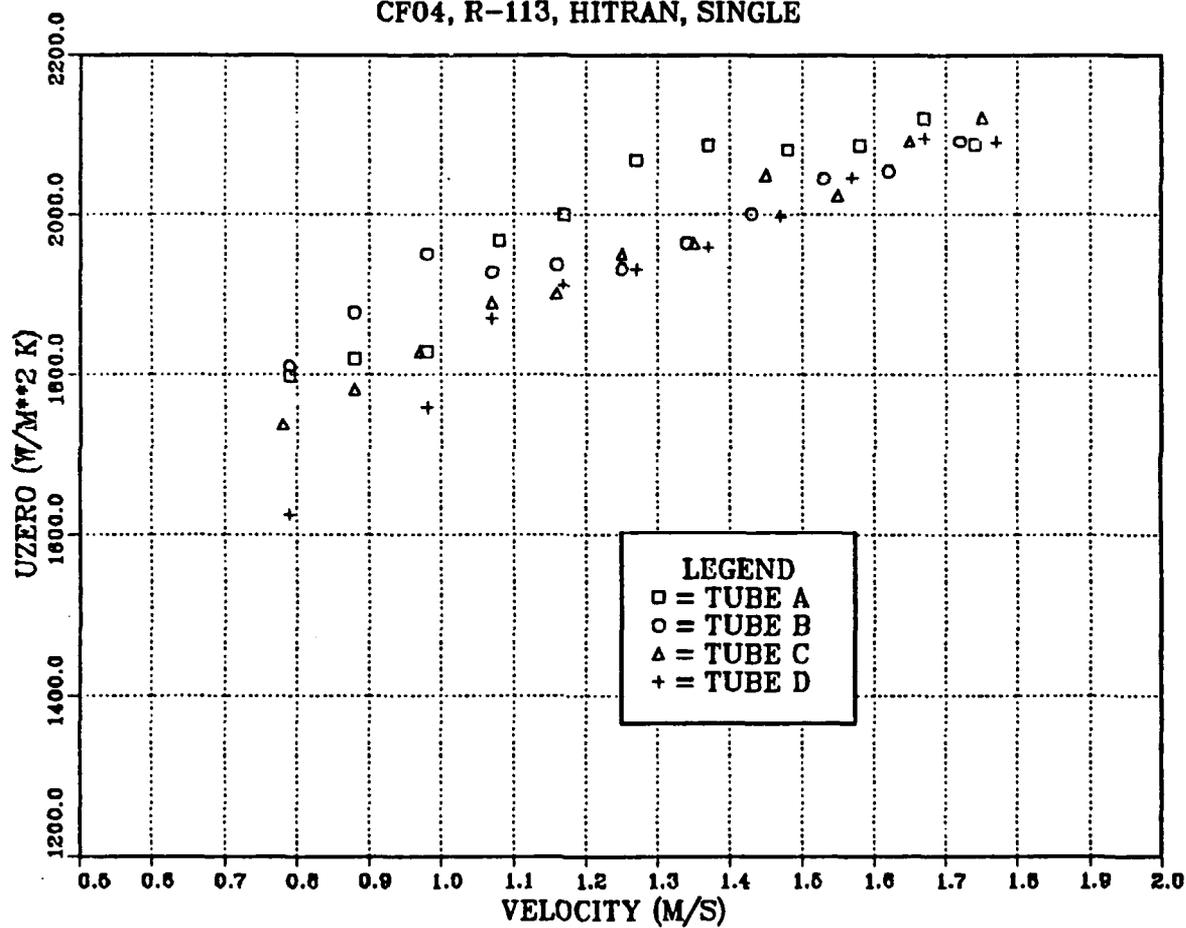
Figure 43. Copper/nickel finned tubes, HEATEX insert, individual operation.

EFFECT OF INSERTS

CU/NI FINNED TUBES
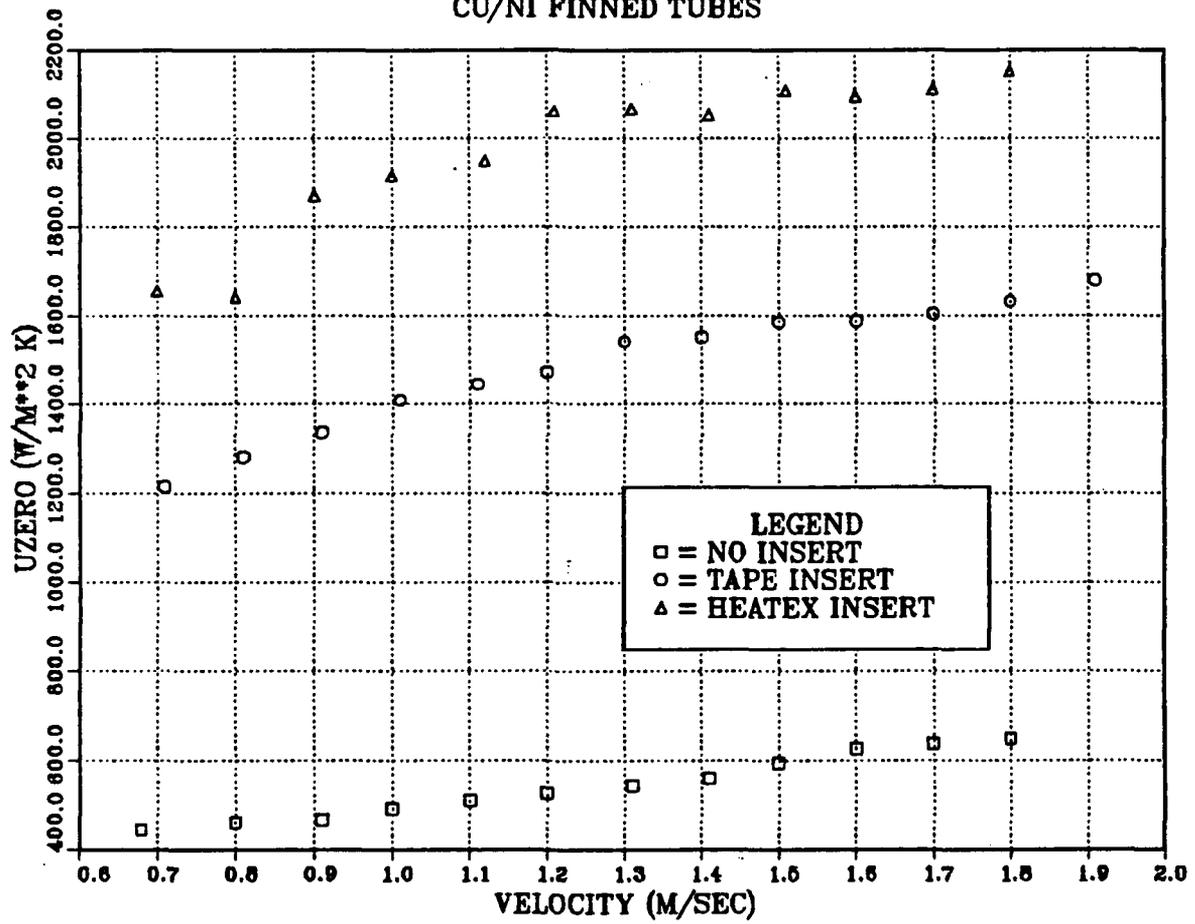
Figure 44.   Summary of effect of tube inserts on U_o.

ually, all had values for h_o which were virtually identical to those obtained for the top tube during bundle operation.

Figure 47 and Figure 48 show values for h_o obtained when the HEATEX elements were installed in the copper tubes during bundle and individual tube operation respectively. As was the case for the tape insert, values of h_o range from about 10 to 14% below that predicted from Nusselt theory. Again, there is a substantial inundation effect which results in a 15 to 18% decrease in h_o when comparing the top tube to the second tube. Note that the values for h_o obtained for the individual tubes agree with those obtained for the top tube during bundle operation. Finally, comparison of the value for h_o obtained when the tape inserts were used as opposed to the HEATEX elements reveal no effect on h_o which is to be expected since we are examining the outside heat transfer coefficient.

It is interesting to note that Goto et al. [Ref. 5] found that values of h_o for R-113 condensing on long, smooth copper tubes were about 10% below that predicted by Nusselt theory. Marto et al. [Ref. 8] found that the vapor-side heat transfer coefficient for R-113 condensing on short (0.133m) smooth copper tubes was in relatively good agreement with Nusselt theory. This result was subsequently confirmed by Michael et al. [Ref. 32]. However, it is important to point out that both the latter reports used very short condenser tubes. The tubes used by Goto et al. were about 0.6 m in length, while the tubes used in this study had a condensing length of 1.2 m. Hence tube length may be an important consideration. The reason for this is as follows. The coolant enters the condenser tube with a given inlet temperature. As the coolant flows through the tube, the coolant temperature increases. The wall termperature is also changing down the length of the tube. At the tube entrance, the temperature difference between the tube wal and the vapor saturation temperature causes the formation of a thick condensate layer. This layer is thinner at the tube outlet due to the smaller wall-vapor saturation temperature difference. This causes condensate to flow in the axial direction. These factors taken together imply that the temperature difference is a function of the length of the tube. This factor is not taken into account in the Nusselt analysis. As coolant velocity increases the difference betwee the coolant inlet and outlet temperatures is reduced. The wall temperature becomes more constant along the tube length so that values for h_o are in better agreement with those determined experimentally.

The effect of using the twisted tape and HEATEX insert on the inside and outside heat transfer coefficients is demonstrated in Figure 49 and Figure 50 respectively. In both cases, the values for h_o are virtually identical as expected. However, the

89

HZERO SMOOTH COPPER TUBES

SM07, R-113, BUNDLE, TAPE INSERT
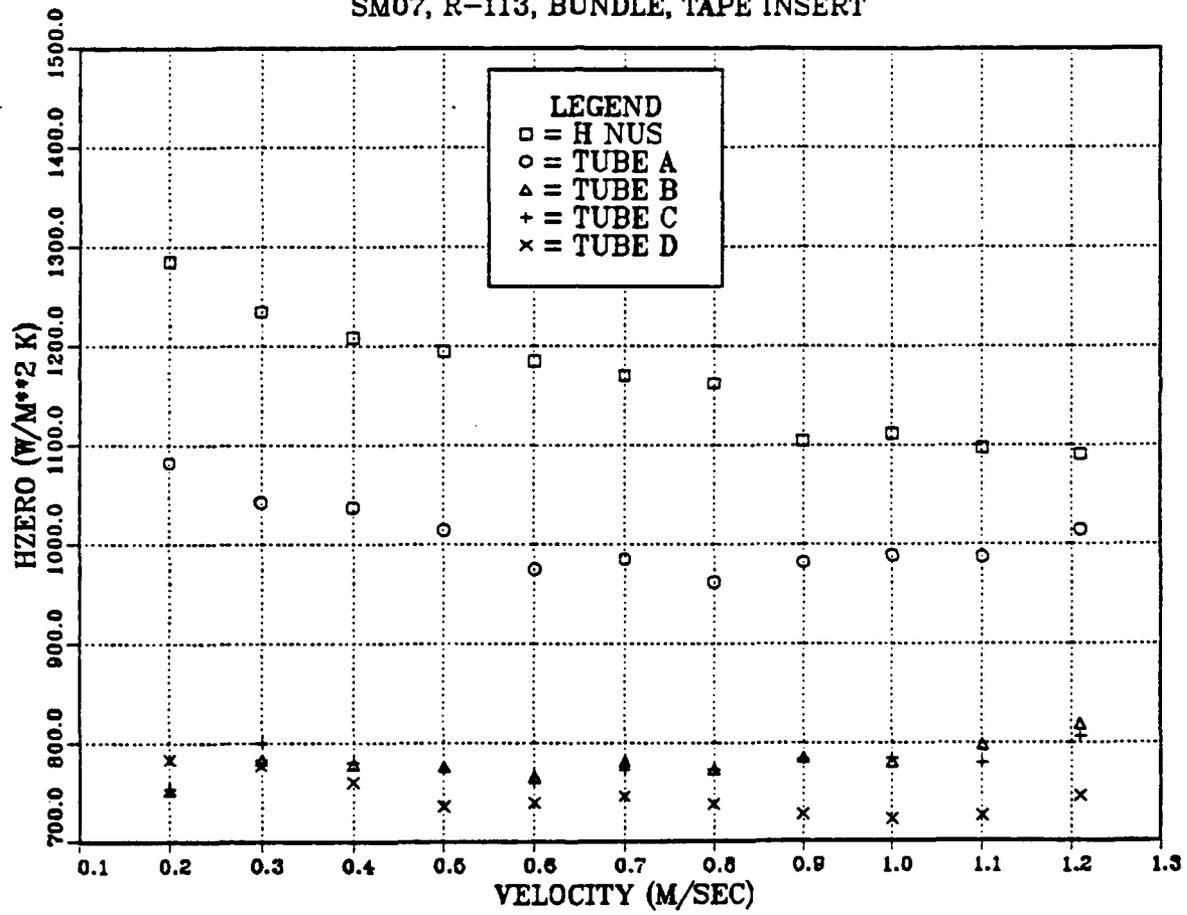
Figure 45.   Smooth Copper tubes, tape insert, bundle operation.

HZERO SMOOTH COPPER TUBES

SM07, R-113, SINGLE, TAPE INSERT
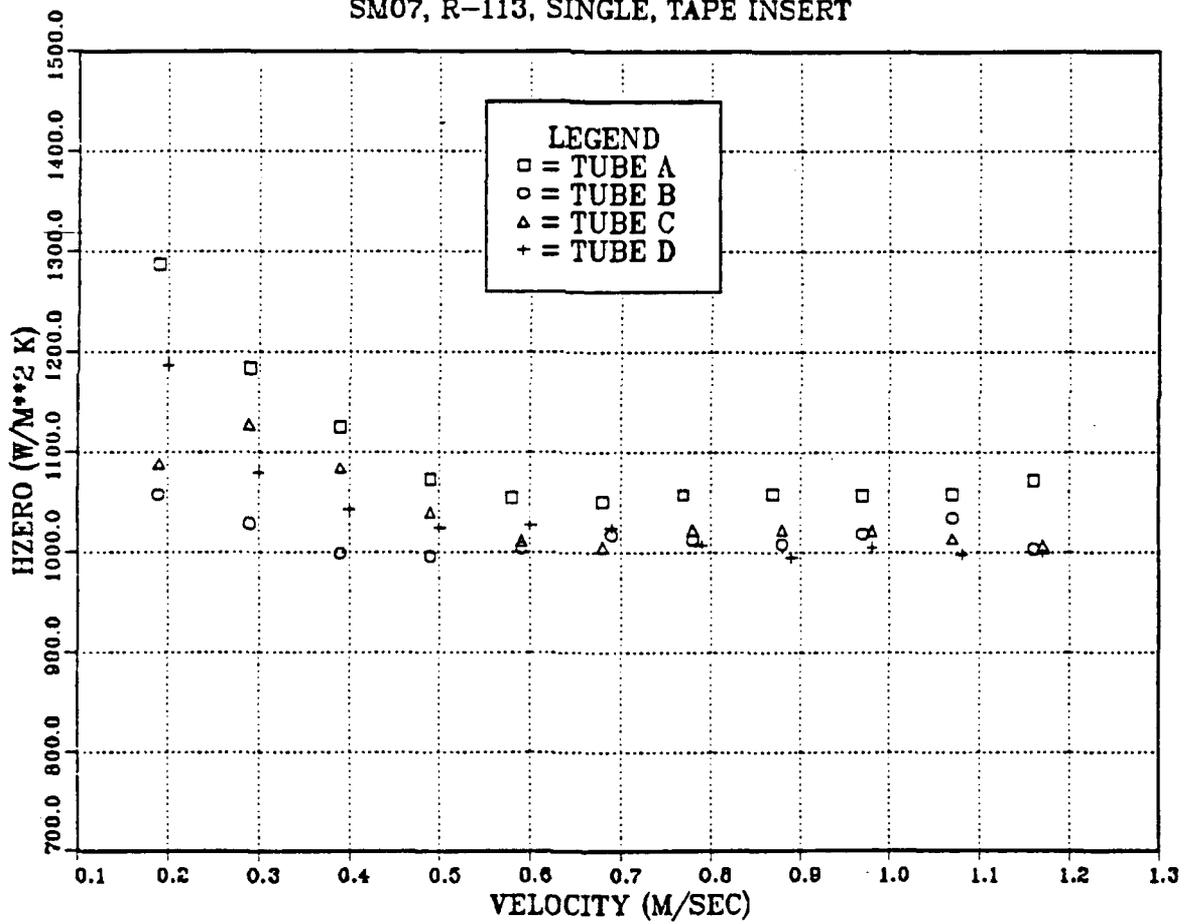
Figure 46.   Smooth Copper tubes, tape insert, individual operation.

91

HZERO SMOOTH COPPER TUBES

SM06, R-113, HEATEX, BUNDLE
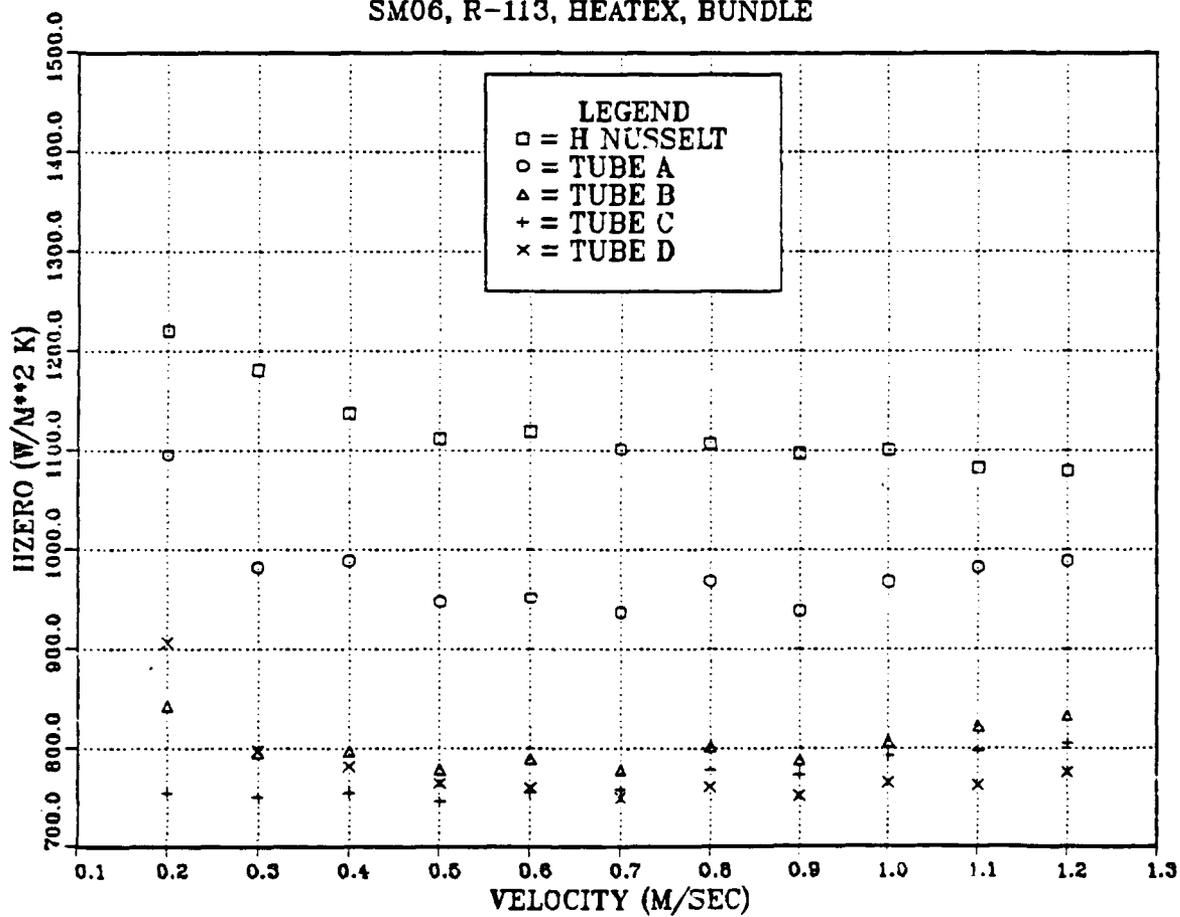
Figure 47. Smooth Copper tubes, HEATEX insert, bundle operation.

HZERO SMOOTH COPPER TUBES

SM06, R-113, HEATEX, SINGLE
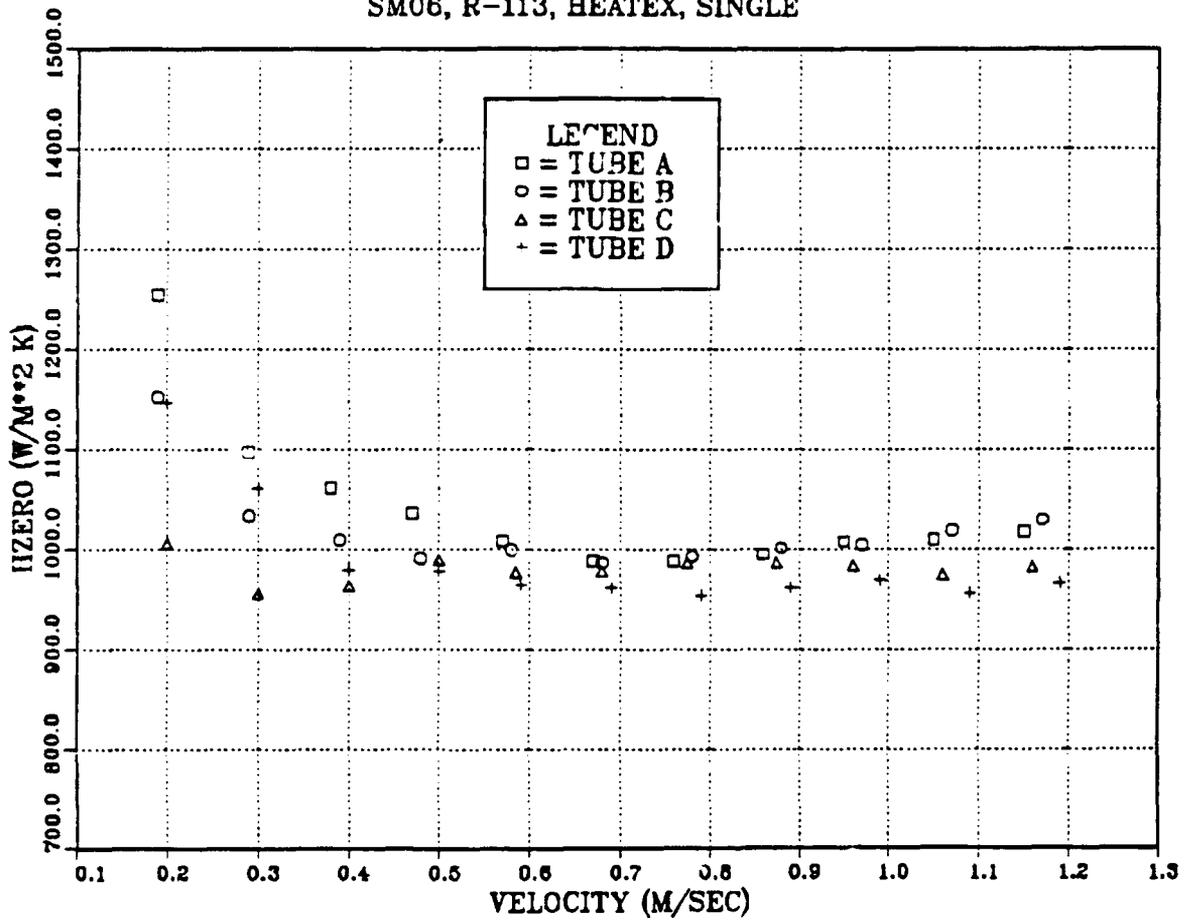
Figure 48. Smooth Copper tubes, HEATEX insert, individual operation.

93

HEATEX elements essentially double the inside heat transfer coefficient. This is what provides the improvement in overall heat transfer coefficient when the HEATEX elements are used.

The effects of inundation were examined by calculating an average and a local heat transfer coefficient. The local h_o is found by dividing the value for h_o for a given tube by the value for the top tube. Values obtained are then compared with Nusselt theory (Eq. 2-18) and the Kern model (Eq. 2-20). The average heat transfer coefficient is determined by taking an average value of h_o for the tube of interest plus the tubes above it and dividing this value by h_o for the top tube. Values obtained are then compared to Nusselt theory (Eq. 2-17), the Kern model (Eq. 2-19), or the Eissenberg model (Eq. 2-12). For both the local and average values, the flatter the curves, the less the effect of condensate inundation.

The effects of inundation on h_o for the smooth copper tubes are shown in Figure 51, Figure 52, Figure 53 and Figure 54 at three arbitrarily chosen coolant velocities. As expected, there is virtually no difference on the local or average values of the heat transfer coefficient when the tape insert is used as opposed to the HEATEX element. In addition, the values for the average heat transfer coefficient lie in between the values predicted by the Kern and Eissenberg models (equations 2-19, and 2-21). Values for the local heat transfer coefficient lie slightly above those predicted by the Kern model (equation 2-20). These results suggest that Nusselt theory is too conservative with regard to the effect of condensate inundation. It is interesting to note that the main inundation effect occurs for the second tube with little additional effect for subsequent tubes in the bundle. This again supports the notion that the smooth copper tubes can only support a limited amount of condensate

2. KORODENSE Tubes

Figure 55 and Figure 56 show values for h_o for the KORODENSE tubes with the tape inserts installed. The figures show data obtained when the tubes were operated as a bundle and individually respectively. The values of h_o for the top tube are above that predicted from Nusselt theory which is due to the small enhancement provided by the roping of the tubes. This enhancement is approximately 15 to 20% in comparison to the smooth copper tubes. As with the smooth copper tubes, there is again a substantial effect of inundation (on the order of 20% for the second tube compared to the top tube). There is also more separation between the values of h_o for tubes B, C, and D than for the smooth copper tubes indicating that the lower tubes may not be "saturated" with

94

SMOOTH COPPER TUBES

TOP TUBE, TAPE INSERT
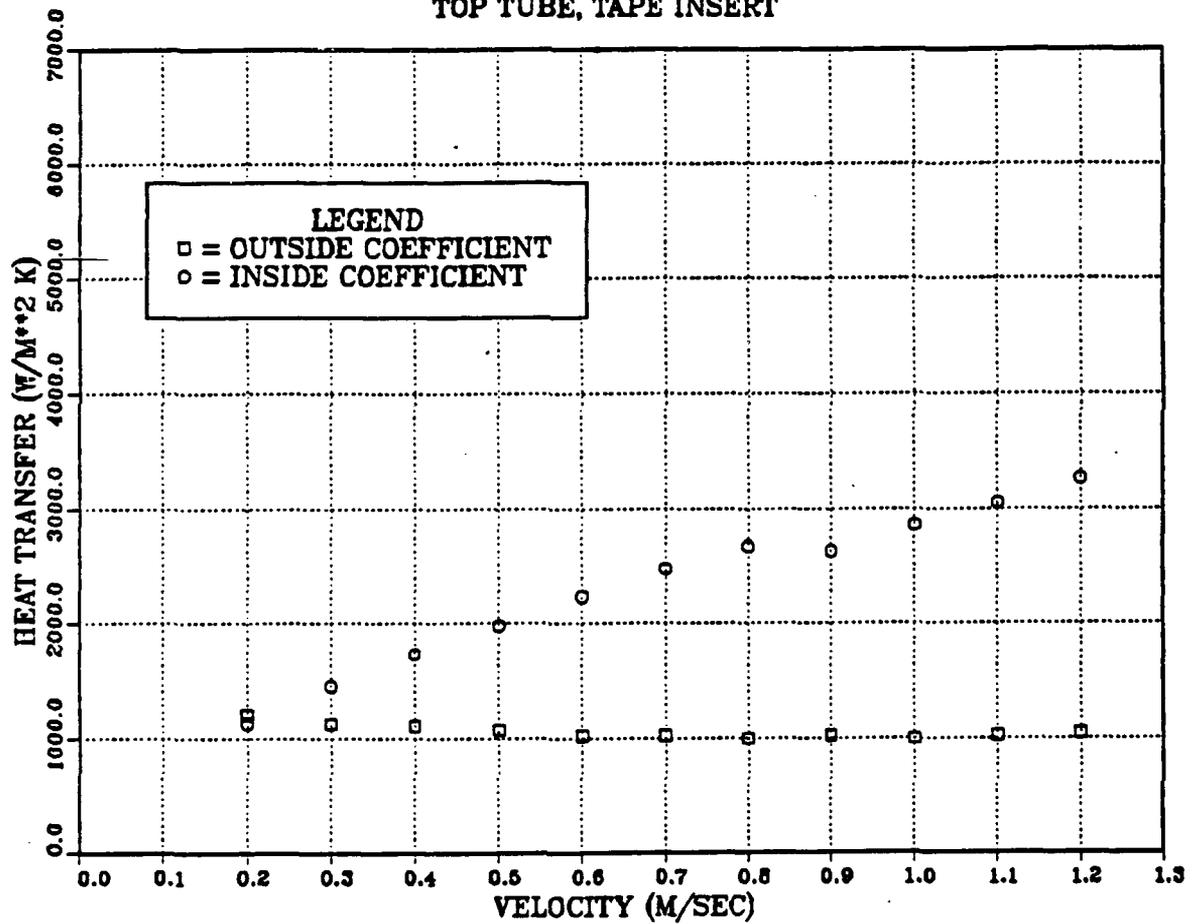
Figure 49.   Effect of tape insert on h_o and h_i
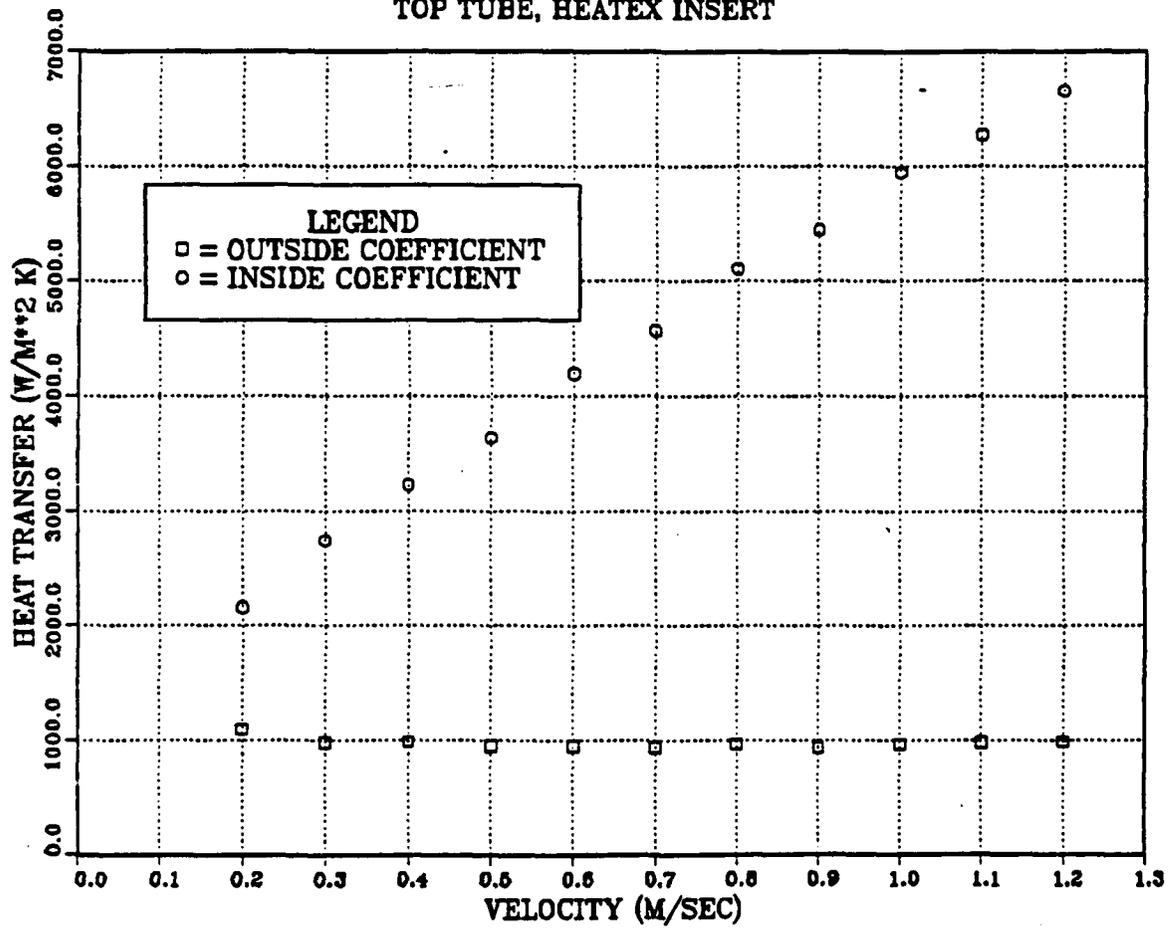
Figure 50. Effect of HEATEX insert on h_e and h_i

HEAT TRANSFER COEFFICIENT

SMOOTH COPPER TUBES, TAPE INSERT

Figure 51.   Local h_o for tape insert.

97
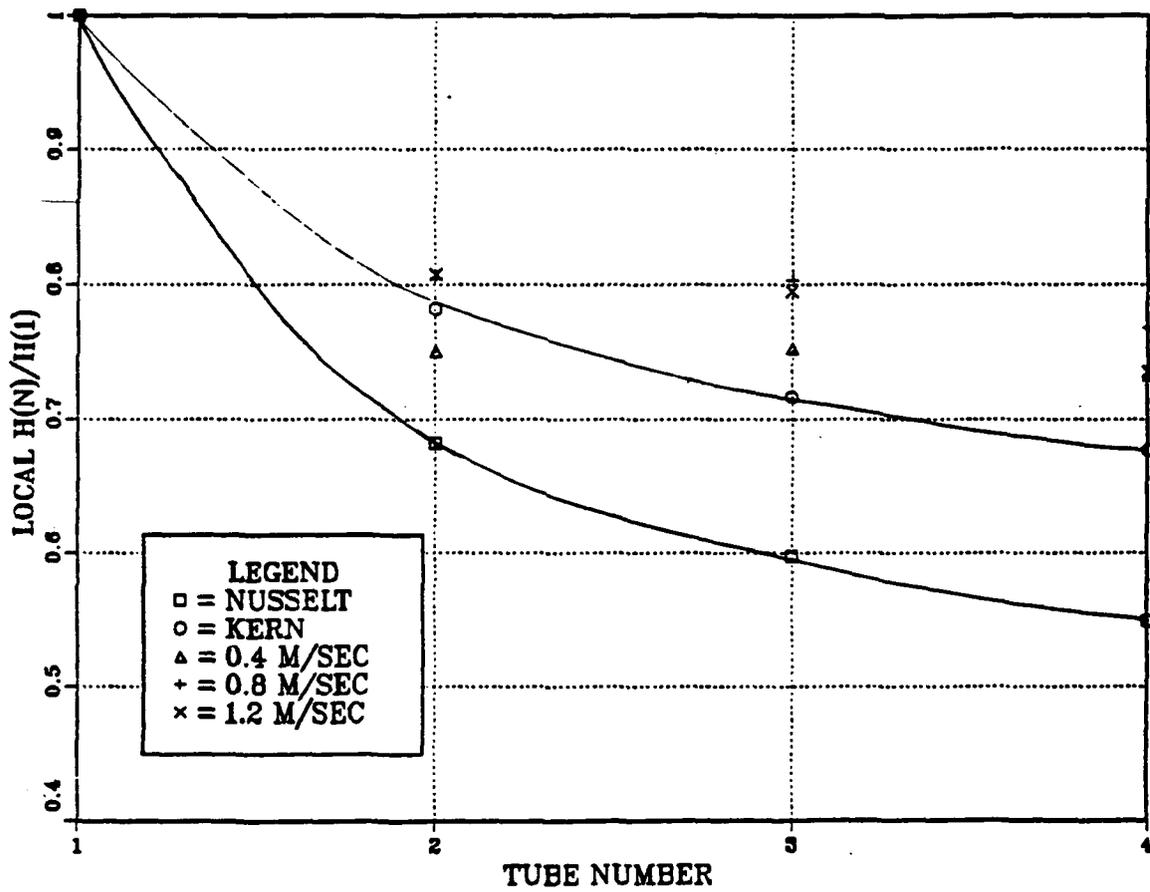
HEAT TRANSFER COEFFICIENT

SMOOTH COPPER TUBES WITH HEATEX
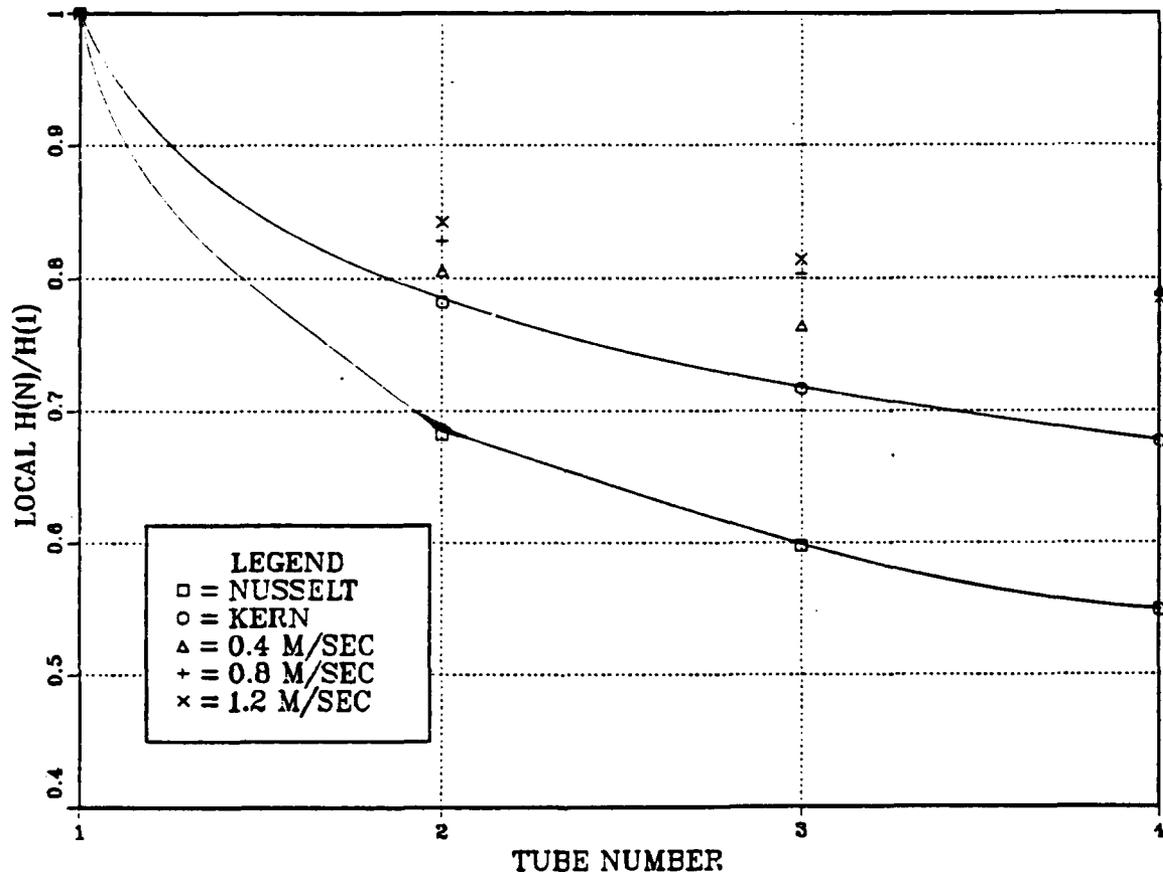
Figure 52.   Local h_o for HEATEX element.

HEAT TRANSFER COEFFICIENT

SMOOTH COPPER TUBES, TAPE INSERT
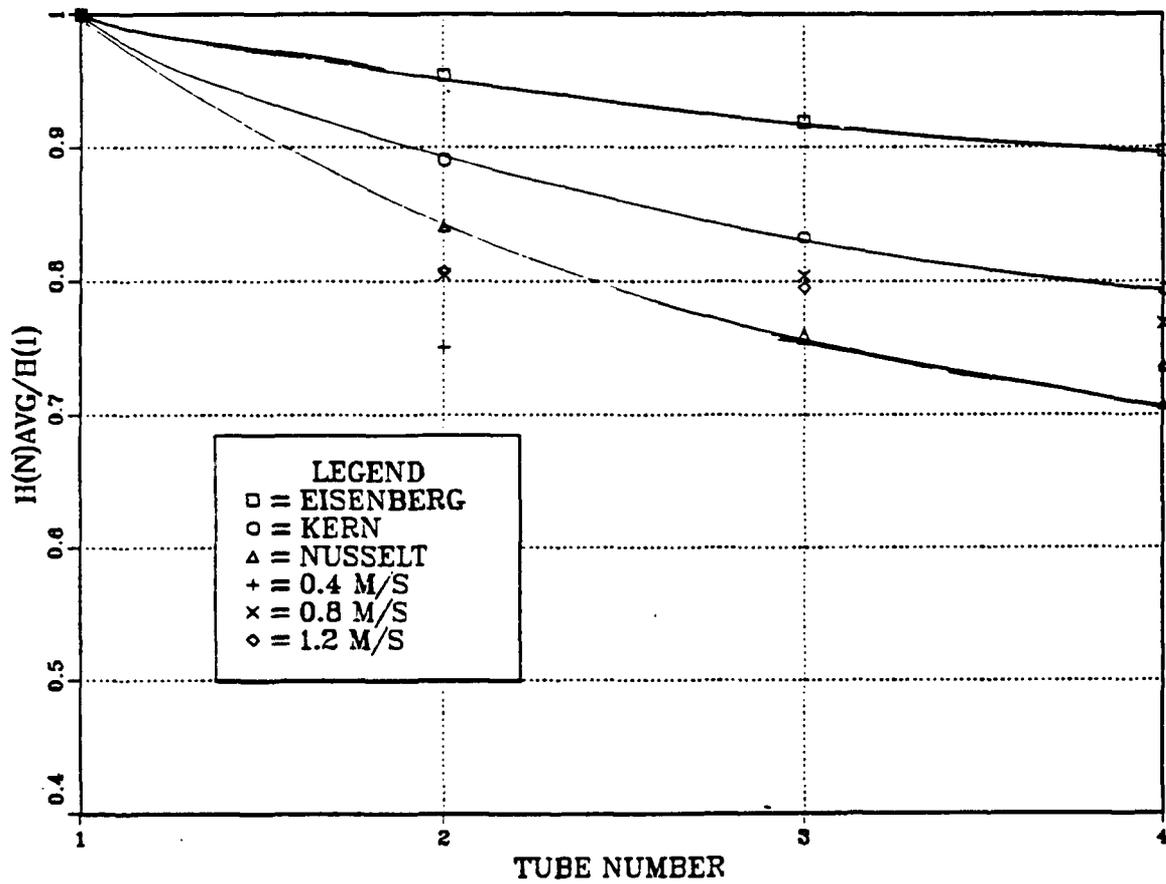
Figure 53.   Average h_o for tape insert.

HEAT TRANSFER COEFFICIENT

SMOOTH COPPER TUBES WITH HEATEX
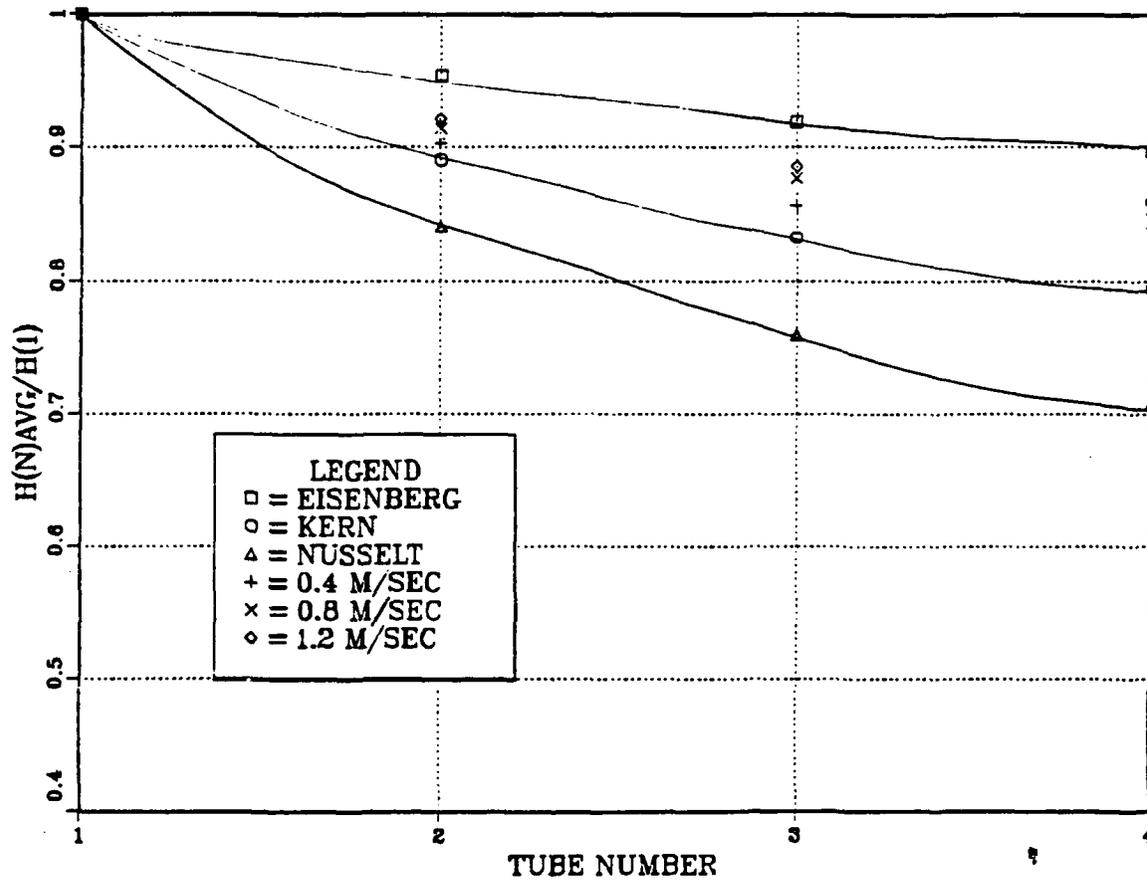
Figure 54.   Average h_o for HEATEX element.

100

condensate. Finally, note that the relative values for h_o for the KORODENSE tubes are about 10 to 15% higher than the corresponding values for the smooth copper tubes.

The data for the KORODENSE tubes operated individually show much more scatter than was seen for the smooth copper tubes especially at low coolant flow rates. Also, rather than being virtually identical with those obtained for the top tube during bundle operation, the values of h_o lay scattered between those for the top tube (bundle operation) and the values predicted by Nusselt theory. Presumably, this is due to either inaccuracies or uncertainty in the inside correlation especially at low coolant flow rates. At higher coolant velocities, the scatter appears to be within about 10%.

Values for h_o for the KORODENSE tubes obtained when the HEATEX elements were installed are shown in Figure 57 and Figure 58 for bundle and individual tube operation respectively. As with the experiments conducted with the tape inserts installed, values of h_o again lie above those predicted by Nusselt theory. The enhancement appears to be approximately 15 to 20%. Again there is a substantial inundation effect which is of the order of 20% for the second tube in comparison to the top tube. The data obtained for the individual tube operation shows less scatter. However, values for all the tubes again lie between those predicted by Nusselt theory and those obtained for the top tube during bundle operation.

As with the smooth copper tubes, the effects of changing inserts on the inside and outside heat transfer coefficients are shown in Figure 59 and Figure 60. The inside heat transfer coefficients are identical to those of the smooth copper tubes (Figure 49 and Figure 50). It was felt that the enhancement provided by the roping would be much less than the enhancement provided by the inserts. Therefore, the same correlations were used to determine the inside heat transfer coefficient. The values for h_o are slightly greater for the KORODENSE tubes than for the smooth copper tubes (approximately 5 to 10%).

Figure 61 and Figure 62 show the local heat transfer coefficient for the four tubes for the tape insert and the HEATEX elements. There is virtually no difference between the two. The values lie above the Kern estimate. Figure 63 and Figure 64 show the average heat transfer coefficient for the four tubes with both inserts. The differences for values using the tape inserts as opposed to the HEATEX elements are again negligible. The average values for h_o also lie just above the Kern model. This is in contrast to the smooth copper tubes where the average values lay near the Nusselt estimate. This suggests that the effect of condensate inundation is indeed less for the KORODENSE tubes.

HZERO KORODENSE TUBES

KD04, R-113, BUNDLE, TAPE INSERT
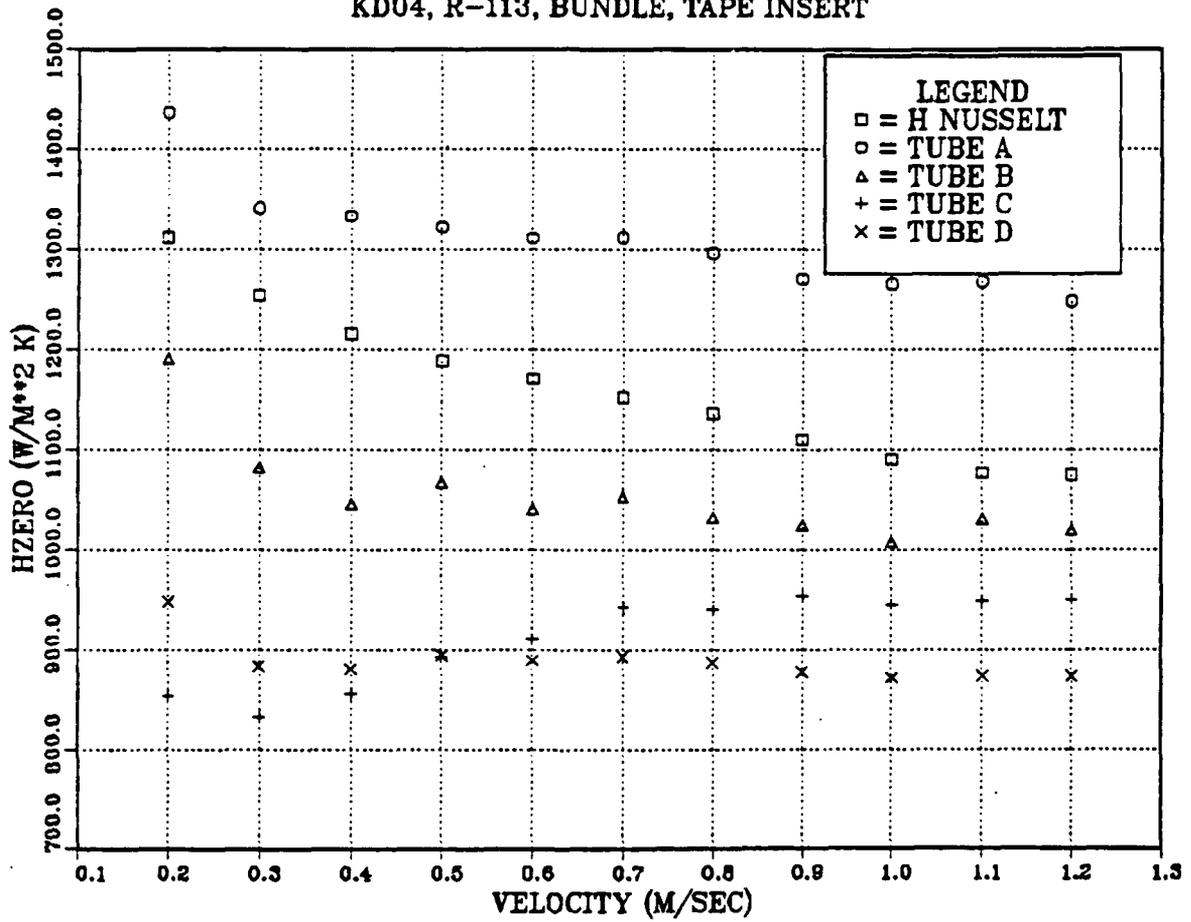
Figure 55.   KORODENSE tubes, tape insert, bundle operation.

102

HZERO KORODENSE TUBES

KD04, R-113, SINGLE, TAPE INSERT

LEGEND
□ = TUBE A
○ = TUBE B
△ = TUBE C
+ = TUBE D

HZERO (W/M**2 K)

VELOCITY (M/SEC)
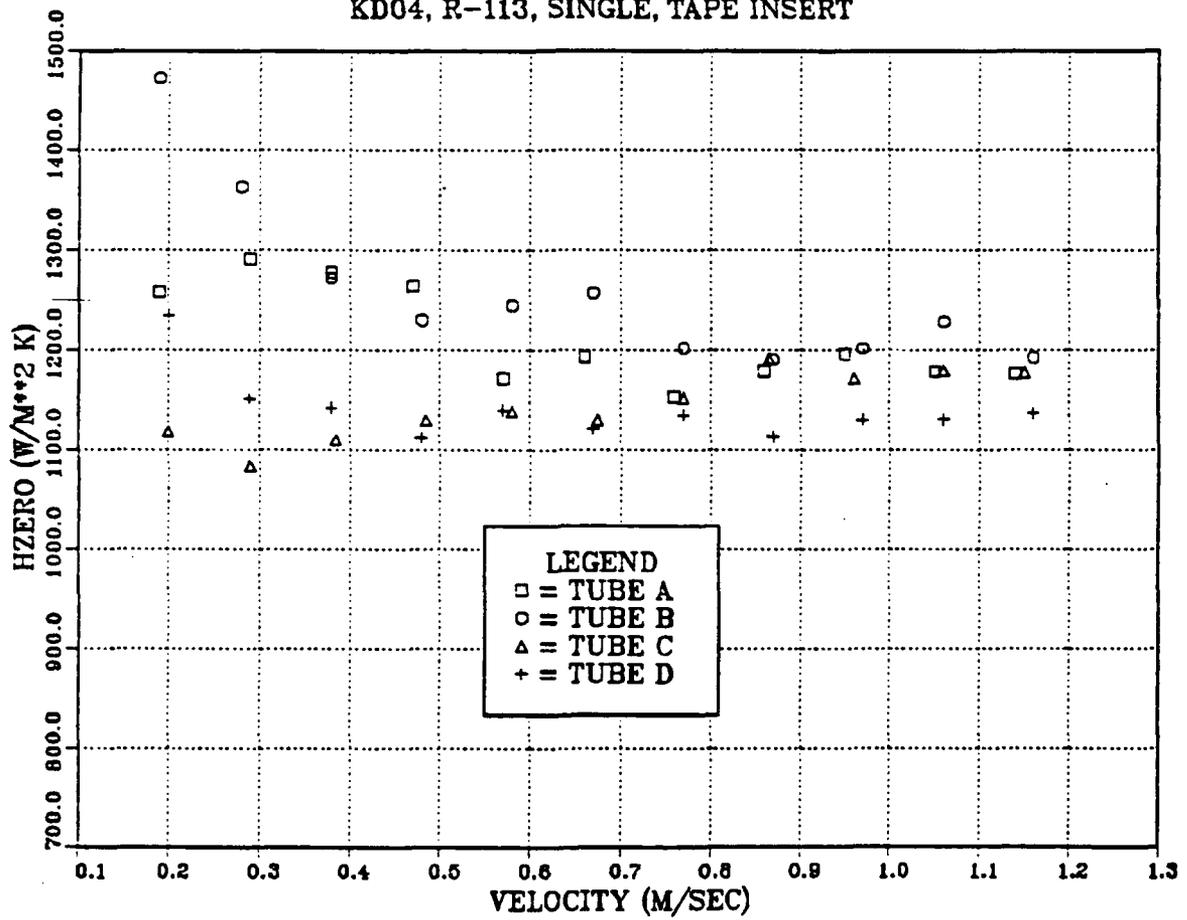
Figure 56.   KORODENSE tubes, tape insert, individual operation.

103

HZERO KORODENSE TUBES

KD03, R-113, BUNDLE, HEATEX
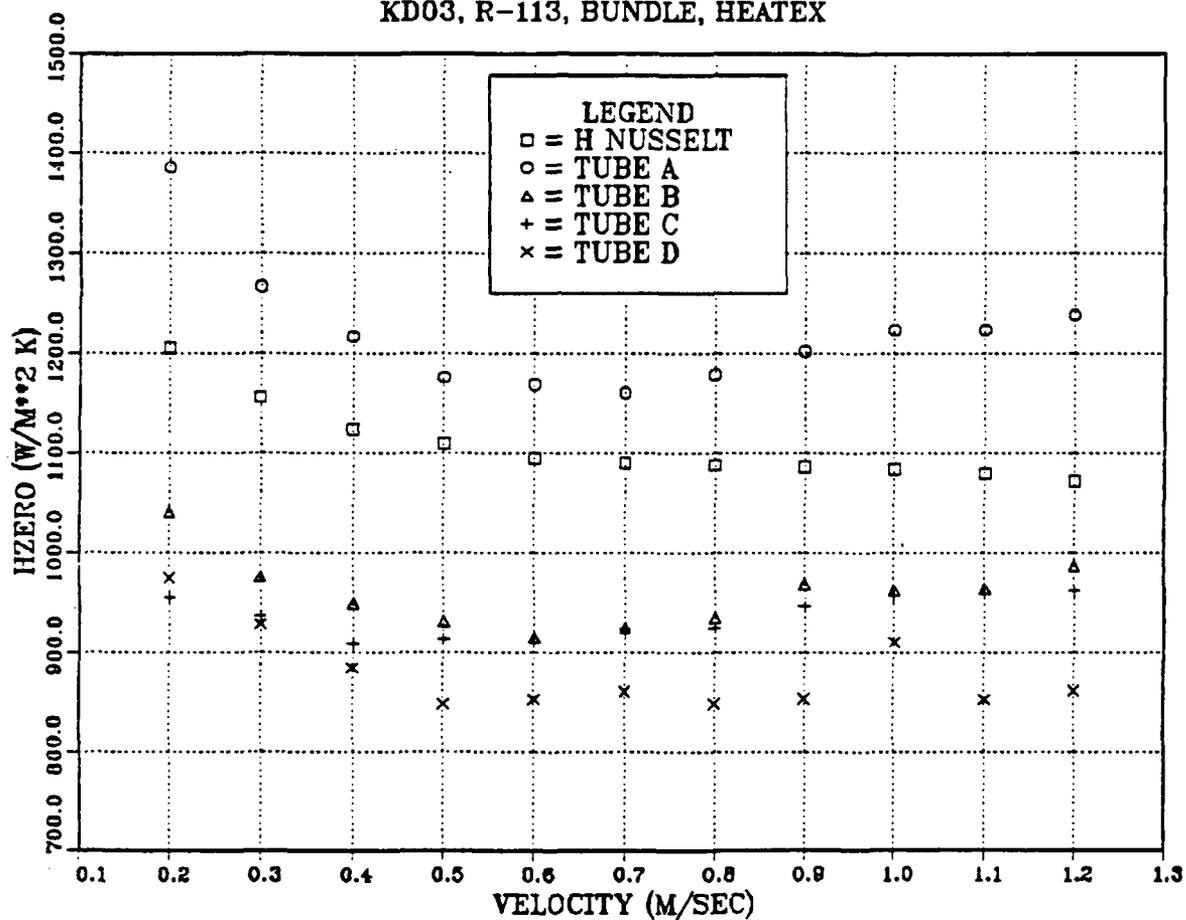
Figure 57.   KORODENSE tubes, HEATEX insert, bundle operation.
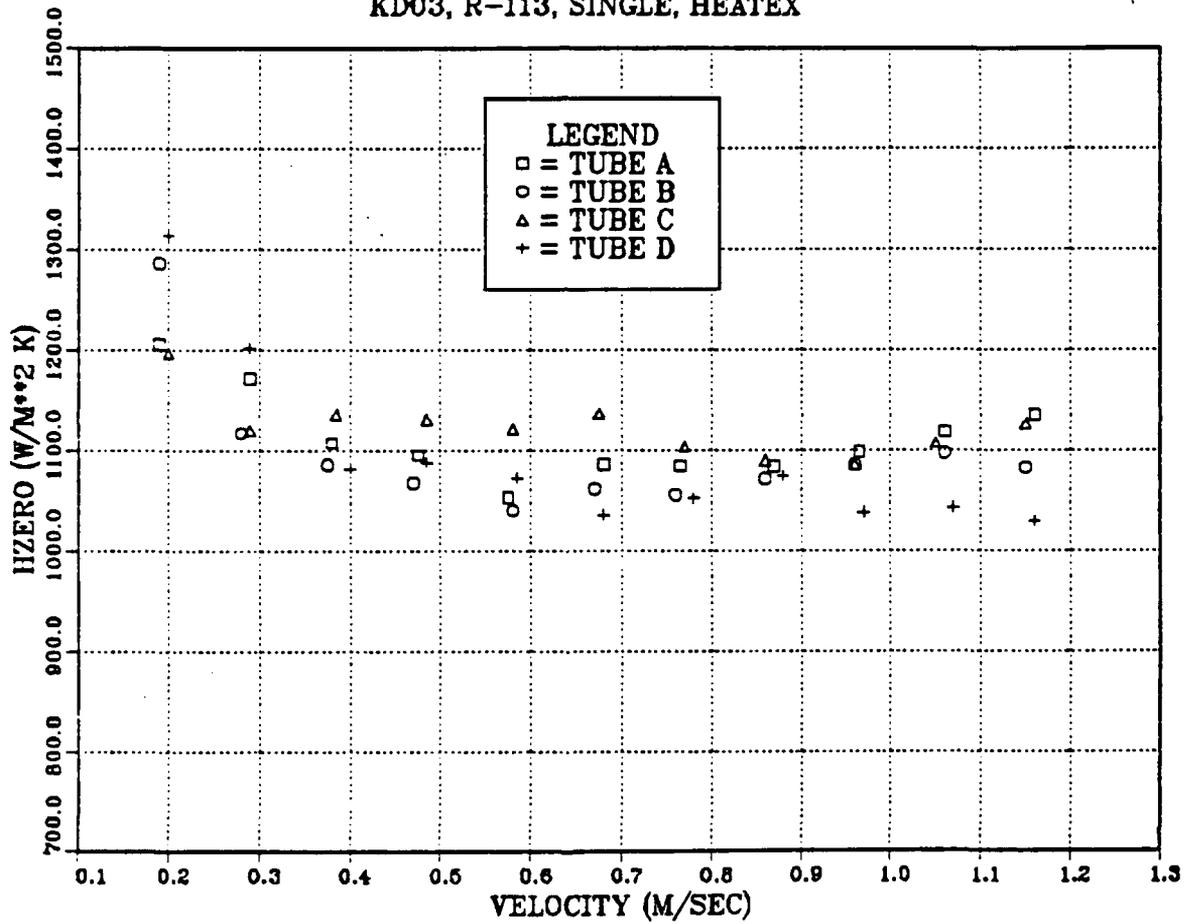
Figure 58.   KORODENSE tubes, HEATEX insert, individual operation.

KORODENSE COPPER/NICKEL TUBES

TOP TUBE, TAPE INSERT
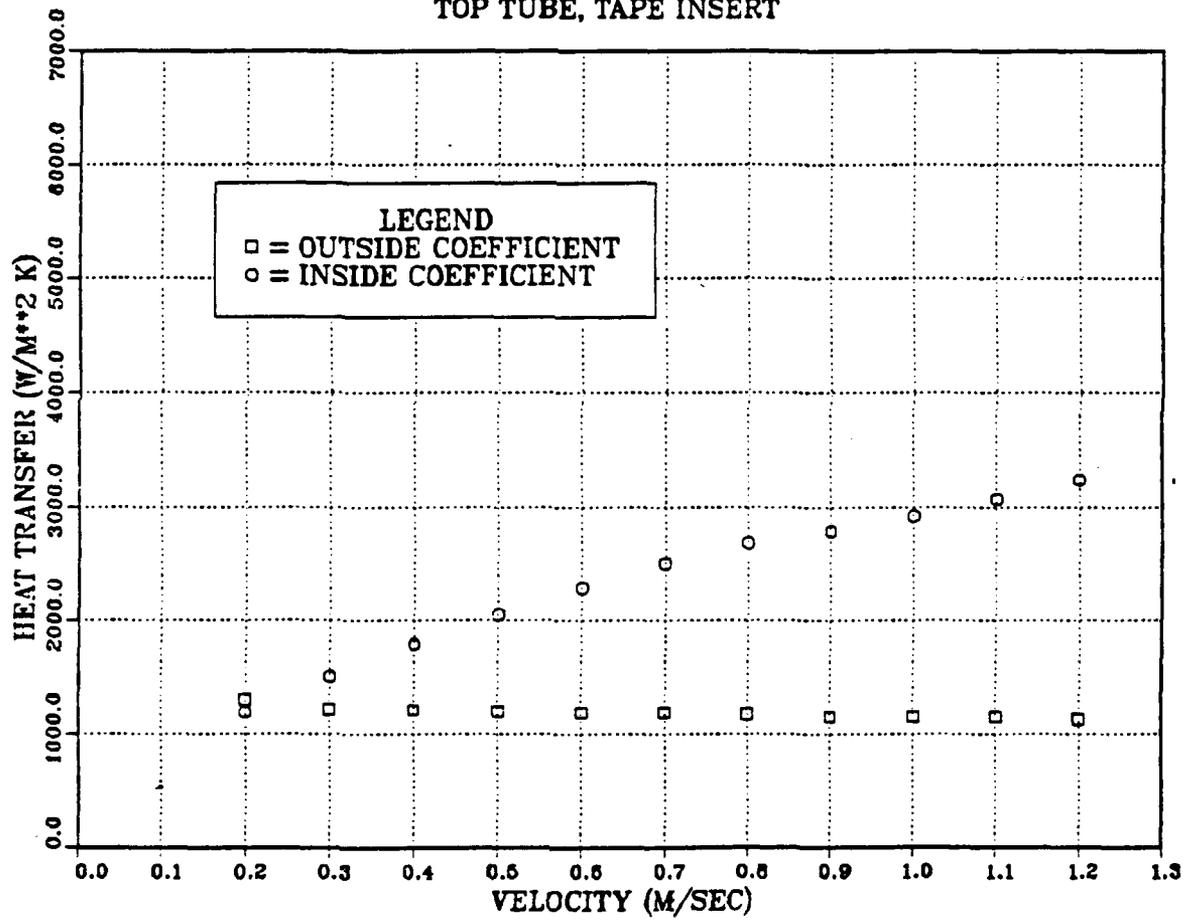
Figure 59. Effect of tape insert on h_o and h_i

KORODENSE COPPER/NICKEL TUBES

TOP TUBE, HEATEX INSERT
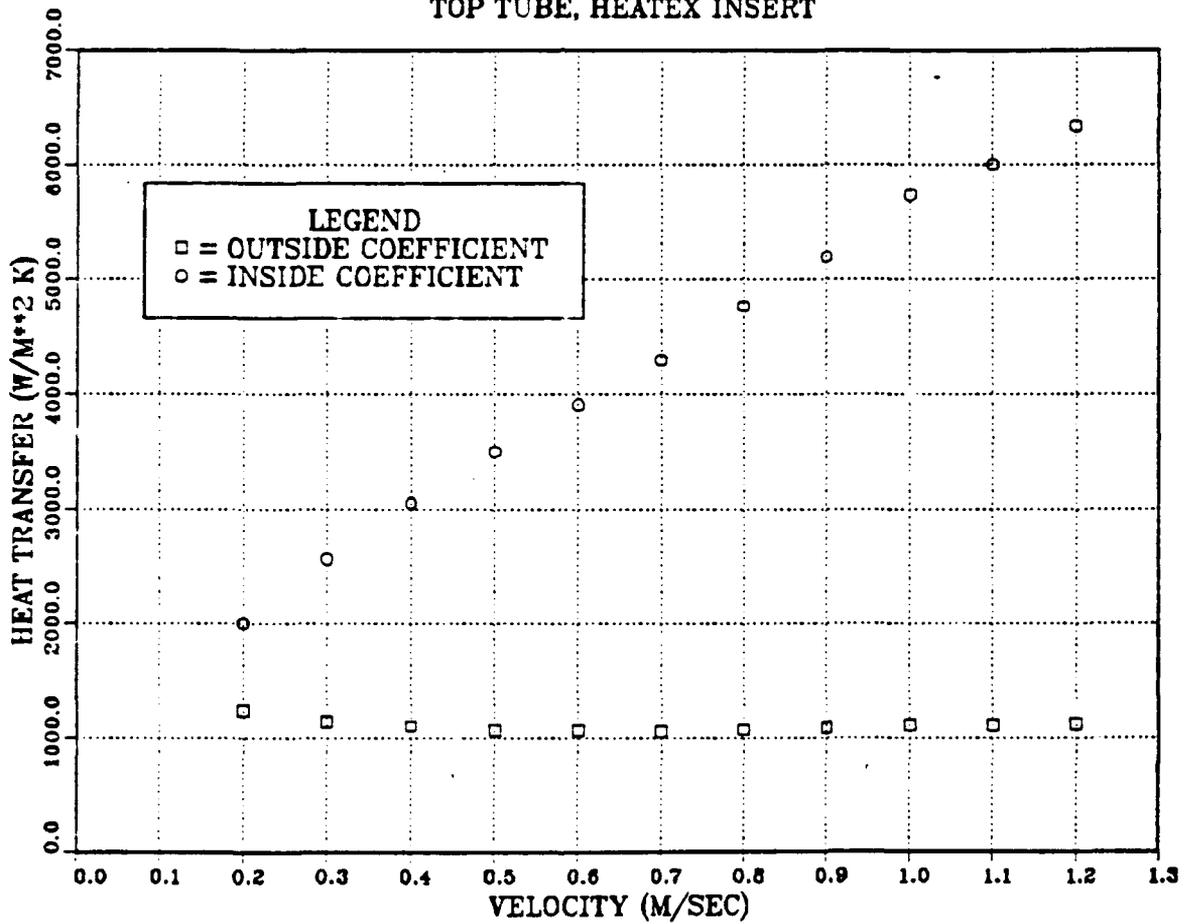
Figure 60. Effect of HEATEX insert on h_o and h_i

HEAT TRANSFER COEFFICIENT

KORODENSE TUBES, TAPE INSERTS
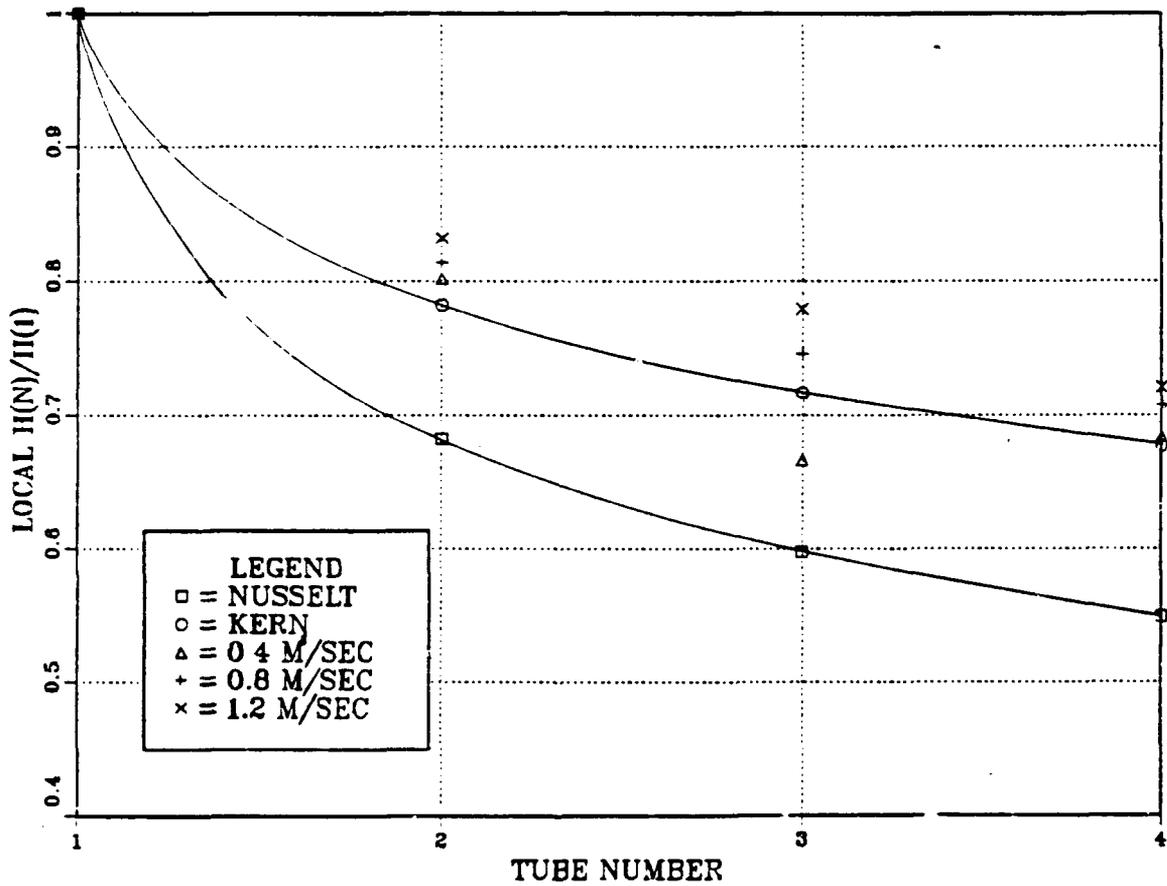
Figure 61. Local h_o for tape insert.
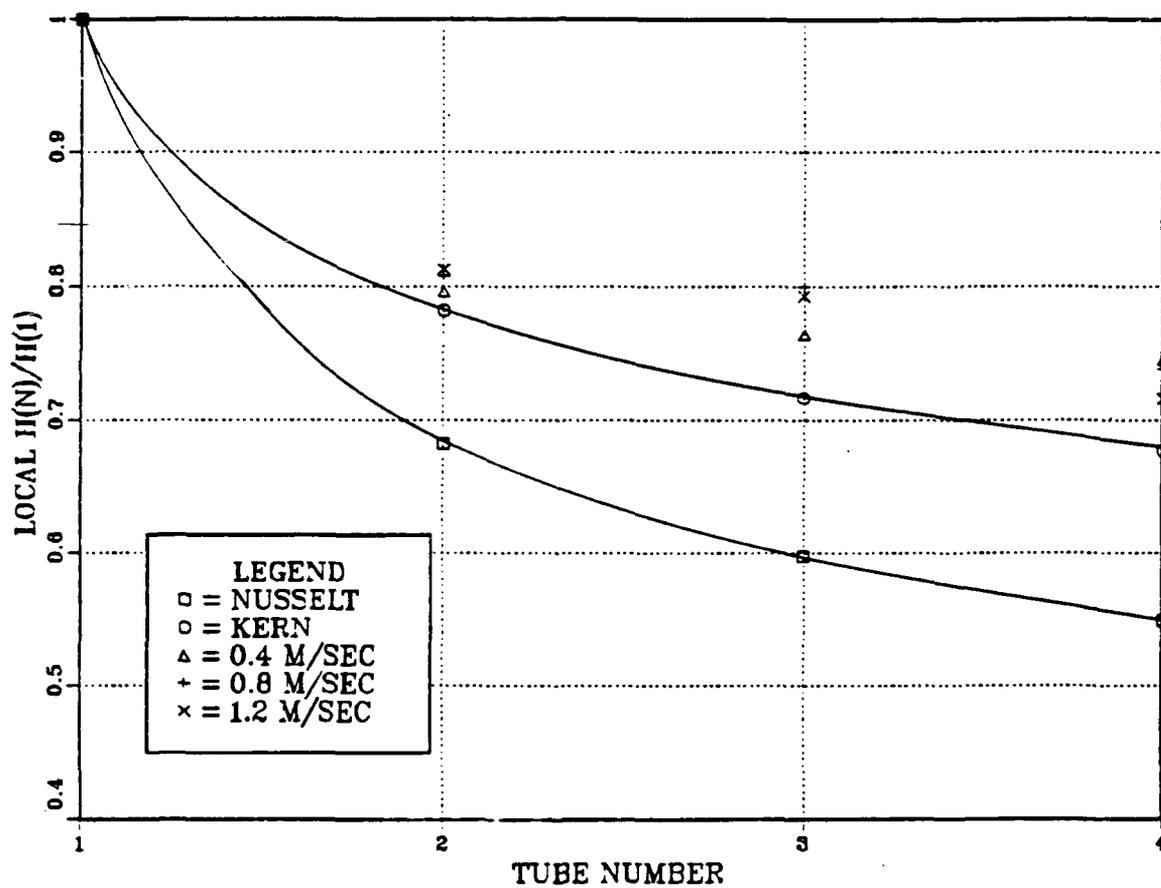
HEAT TRANSFER COEFFICIENT

KORODENSE TUBES WITH HEATEX

Figure 62. Local h_o for HEATEX element.

HEAT TRANSFER COEFFICIENT

KORODENSE TUBES, TAPE INSERT
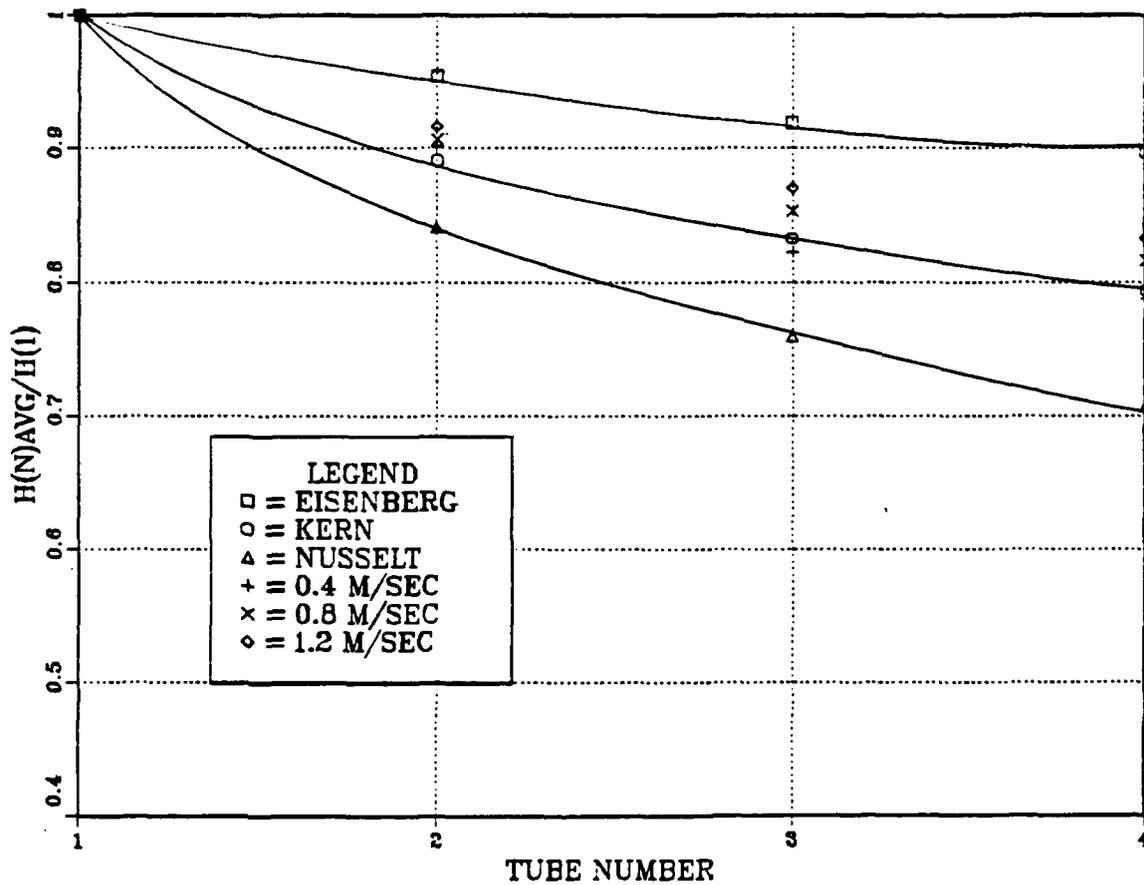
Figure 63.   Average h_o for tape insert.

HEAT TRANSFER COEFFICIENT

KORODENSE TUBES WITH HEATEX
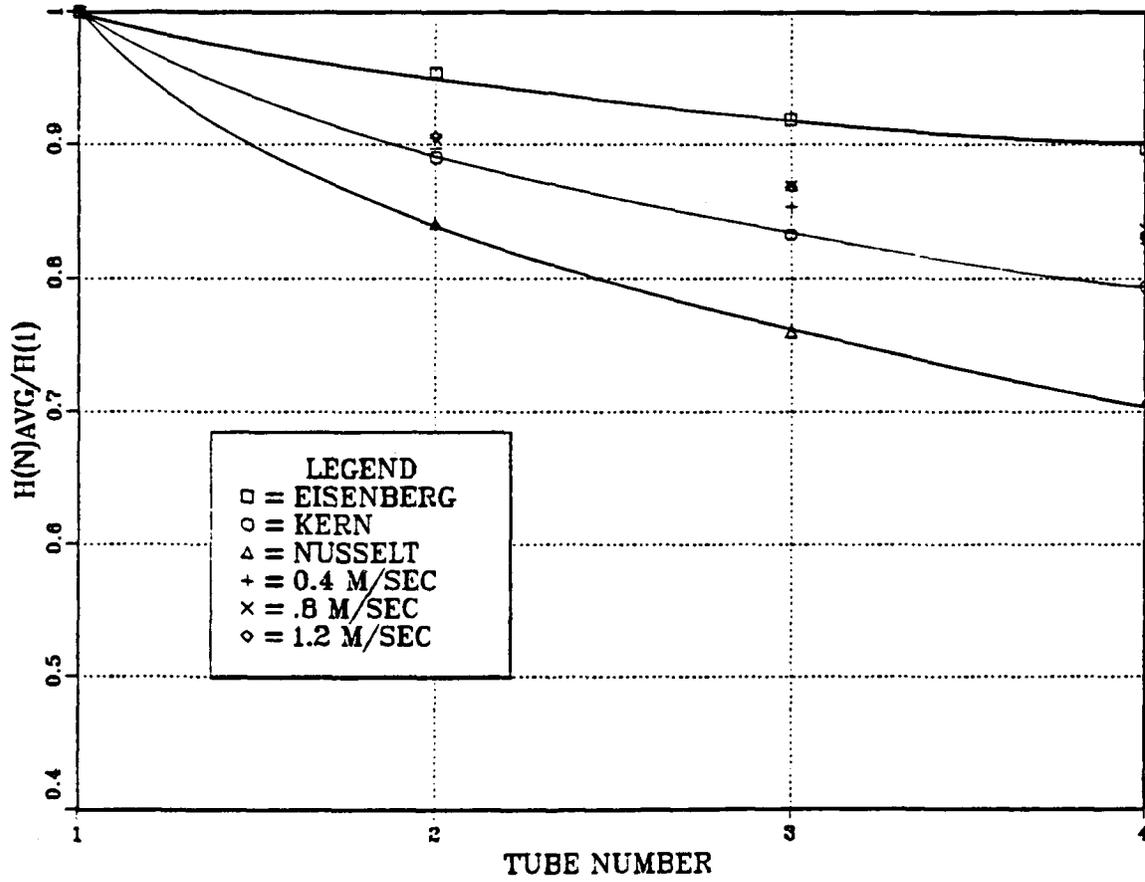
Figure 64. Average h_o for HEATEX element.

3. Wire Wrapped KORODENSE Tubes

The effect on h_o of wrapping wire around the KORODENSE tubes is shown in Figure 65, Figure 66, Figure 67 and Figure 68. In viewing these figures, one should remember that the values for the unwrapped KORODENSE tubes were about 10% greater than predicted from Nusselt theory and about 15% greater than for the smooth copper tubes. From the figures, it appears that the 0.049" diameter wire results in the greatest enhancement in h_o for all four tube positions. However, the relative magnitude of the enhancement decreases as lower tubes are examined. Since this data was obtained during individual tube operation, the results are not due to condensate inundation.

Values for h_o are shown for each wire diameter in all four tube positions in Figure 69, Figure 70, Figure 71 and Figure 72. When no wire is used, the results are the same as those shown in Figure 58. For the 0.029" wire, positions A and B are essentially identical with tube C showing the greatest enhancement and tube D the worst. The magnitude of the deviation is about 20%. The 0.049" diameter wire results in the best enhancement in h_o regardless of position. However, note that position D is not overwhelmingly greater than for the 0.029" wire. The data show that the level of enhancement depends upon bundle position. The top tube positions behave similarly while positions C and D show reduced levels of enhancement Finally, the 0.0675" diameter wire yields values for h_o which are nearly the same although a positional effect appears evident. An interesting comparison are the values for h_o for the 0.049" and 0.0675" wire in the fourth position. Here it appears that the 0.0675" wire gives roughly the same level of enhancement especially at high coolant velocities.

4. KORODENSE Tubes Wrapped with 0.049" Wire

The effects on h_o of wrapping 0.049" diameter wire on KORODENSE tubes on are shown in Figure 73 and Figure 74. These two experiments are identical runs. In both cases, there is a substantial improvement in h_o for all tubes when compared to values predicted from Nusselt theory. It is interesting to note that even though tubes B, C, and D are all being inundated, the values for h_o are still well above Nusselt theory (32% for tube A, 28% for tube B, 22% for tube C and 16% for tube D). In comparison with the non-wire wrapped KORODENSE tubes, wrapping with the 0.049" diameter wire yields an increase of about 70% for tubes A and B and about a 30 to 50% increase for tubes C and D. Comparison with the top smooth tube yields almost a 90% increase in h_o.

The results for the single tube runs are shown in Figure 75 and Figure 76. Both runs demonstrate considerable scatter in the data of about $\pm$ 22%. It is also interesting

112

HZERO WIRE WRAPPED KORODENSE
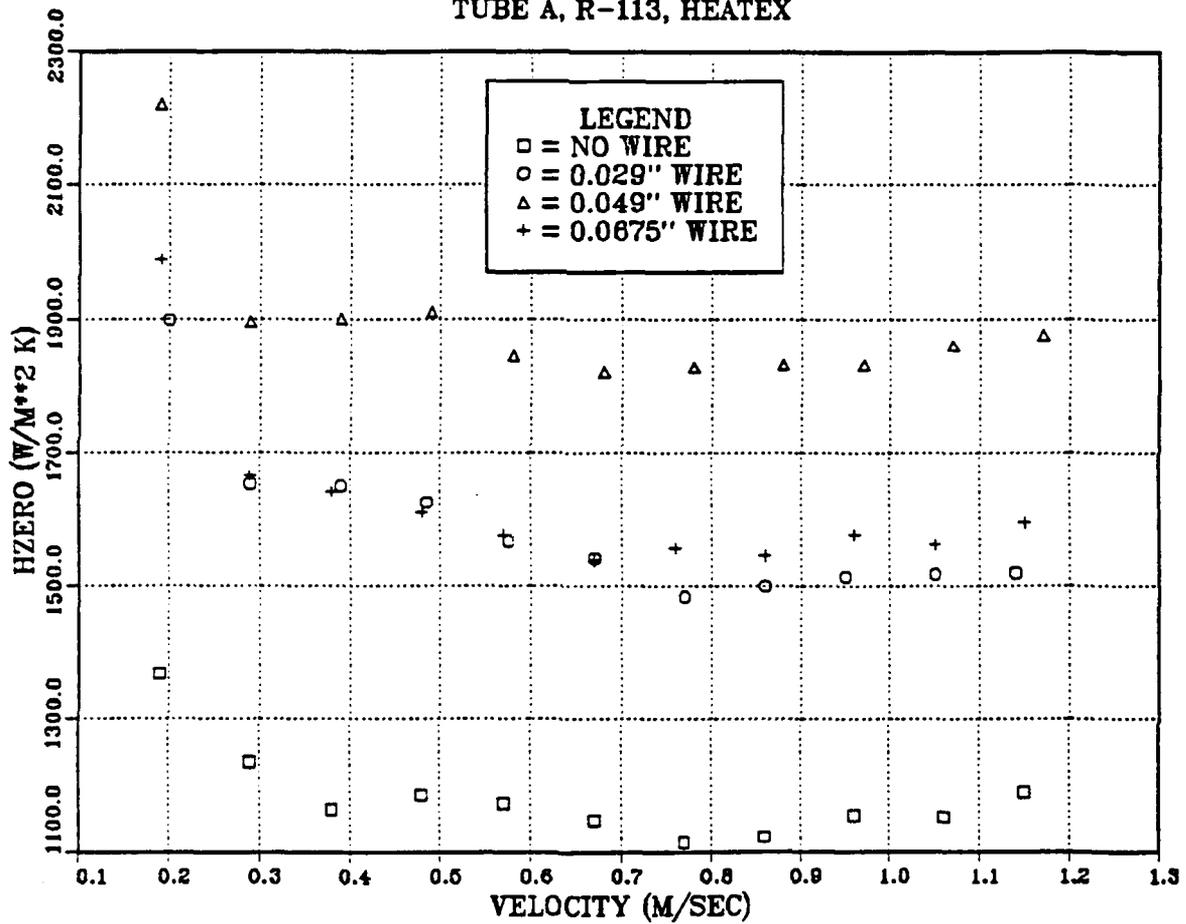
TUBE A, R-113, HEATEX

Figure 65. Effect of wire diameter on h_o for tube A.

HZERO WIRE WRAPPED KORODENSE
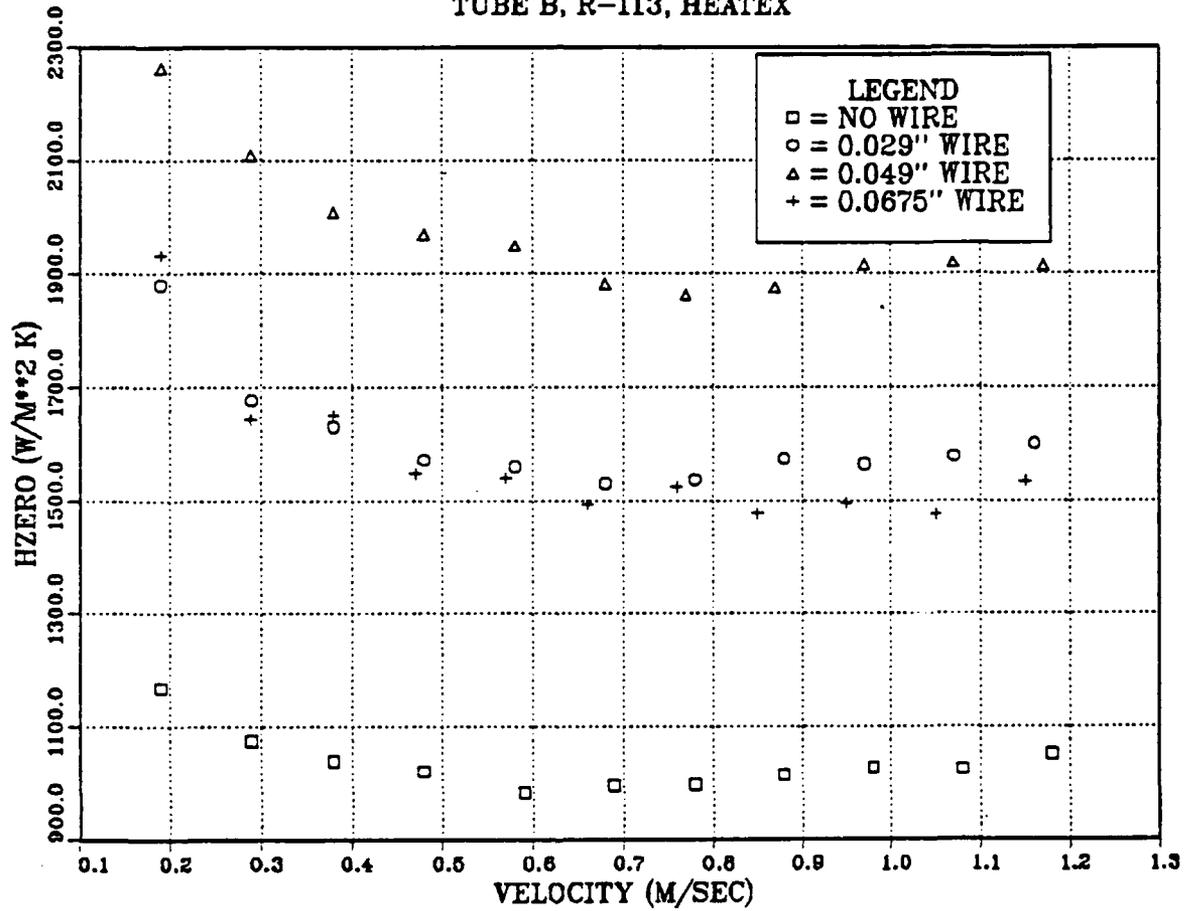
TUBE B, R-113, HEATEX

Figure 66. Effect of wire diameter on h_o for tube B.

114

HZERO WIRE WRAPPED KORODENSE
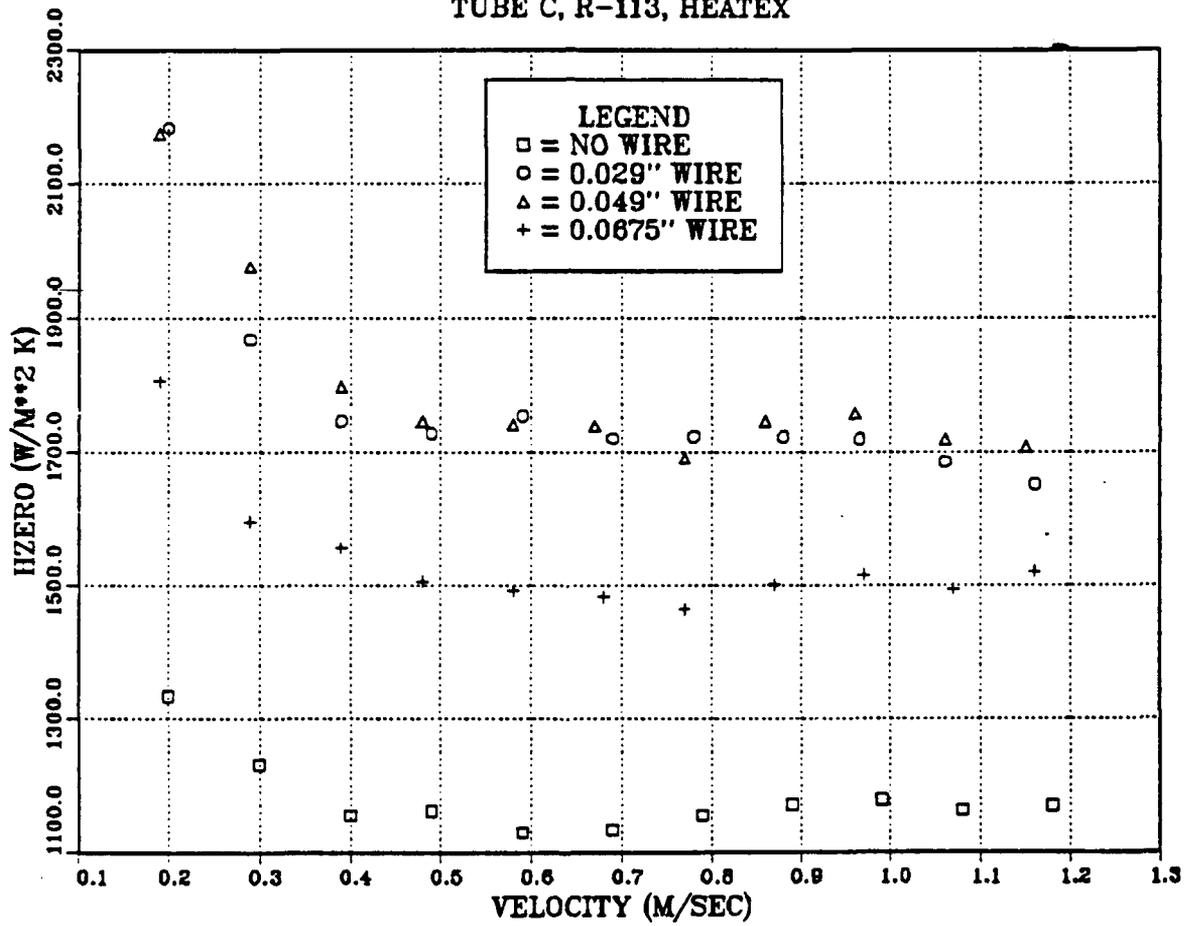
TUBE C, R-113, HEATEX

Figure 67.   Effect of wire diameter on h_o for tube C.

HZERO WIRE WRAPPED KORODENSE

TUBE D, R-113, HEATEX
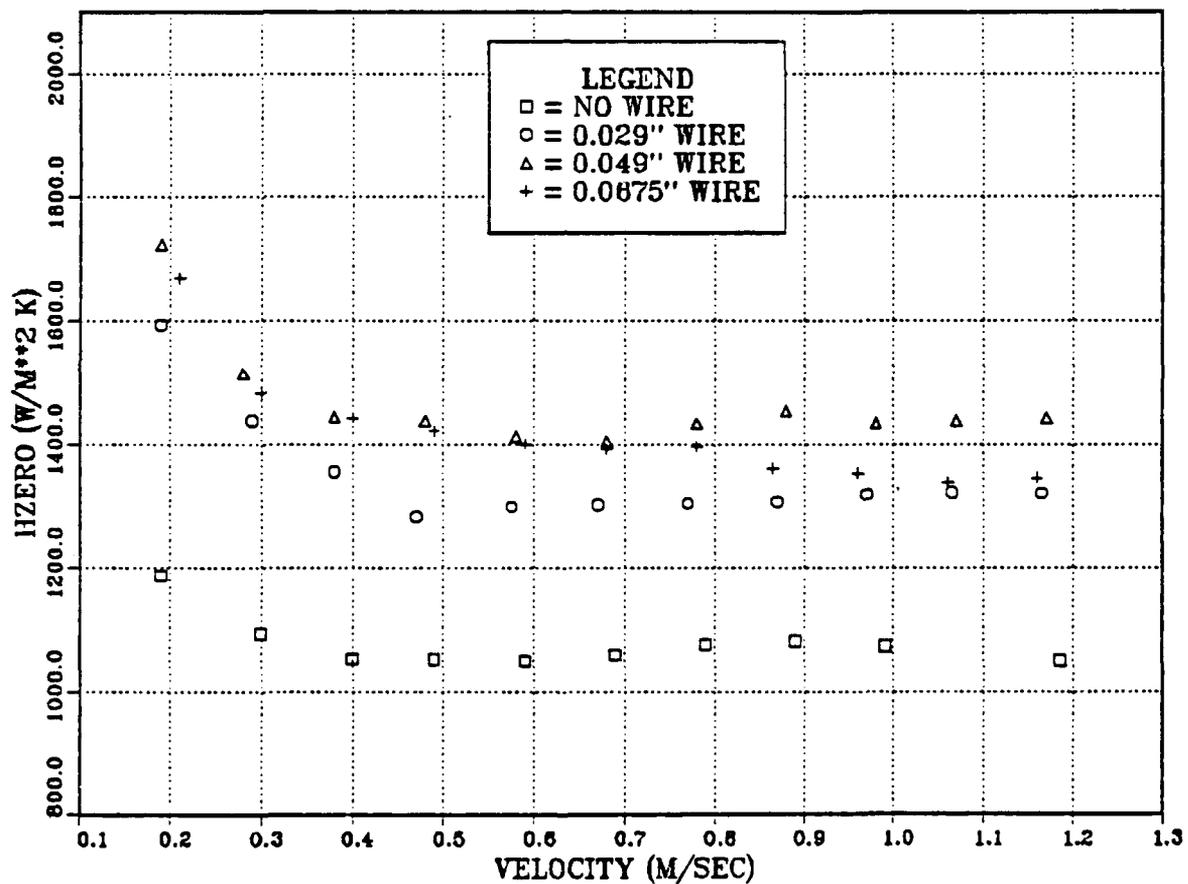
Figure 68. Effect of wire diameter on h_o for tube D.

HZERO WIRE WRAPPED KORODENSE

NO WIRE, HEATEX
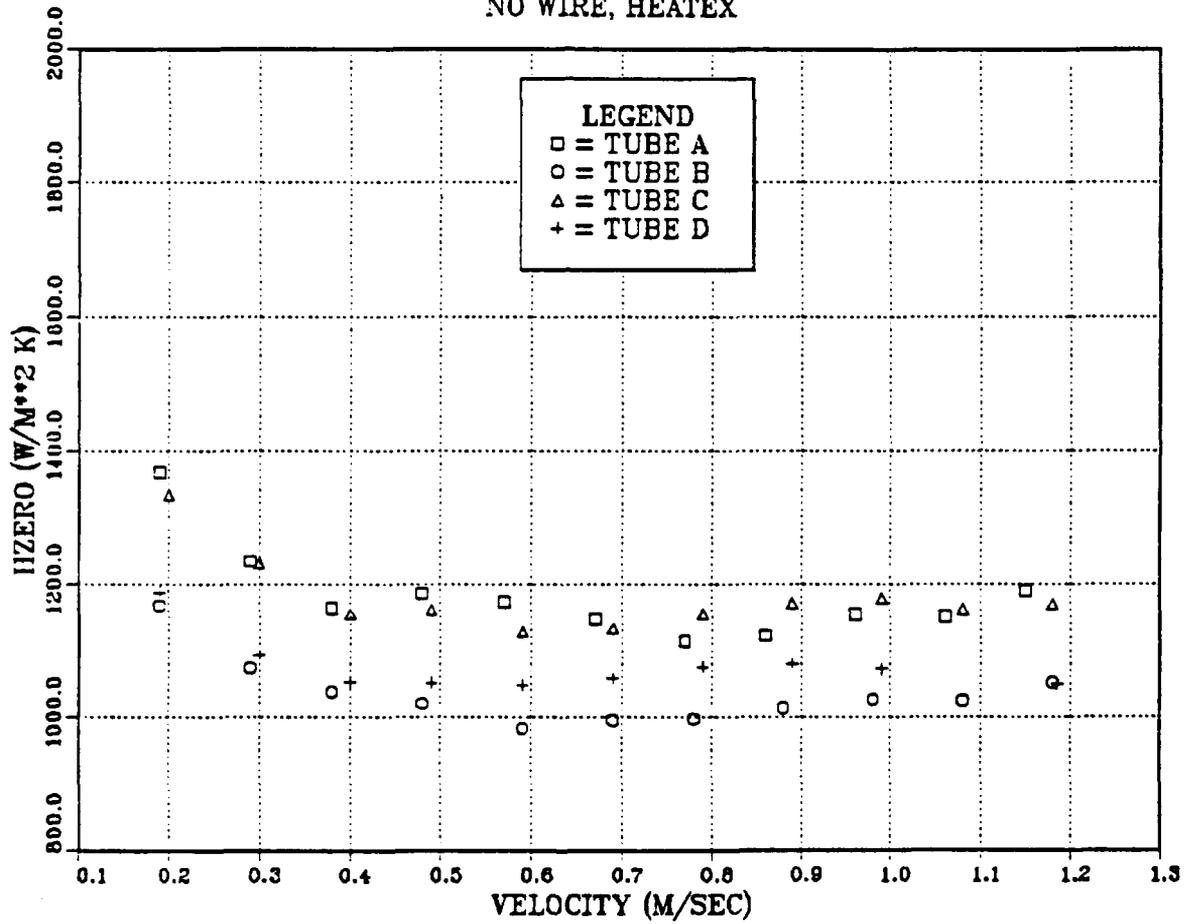
Figure 69. Effect of no wire on h_o for all tubes.
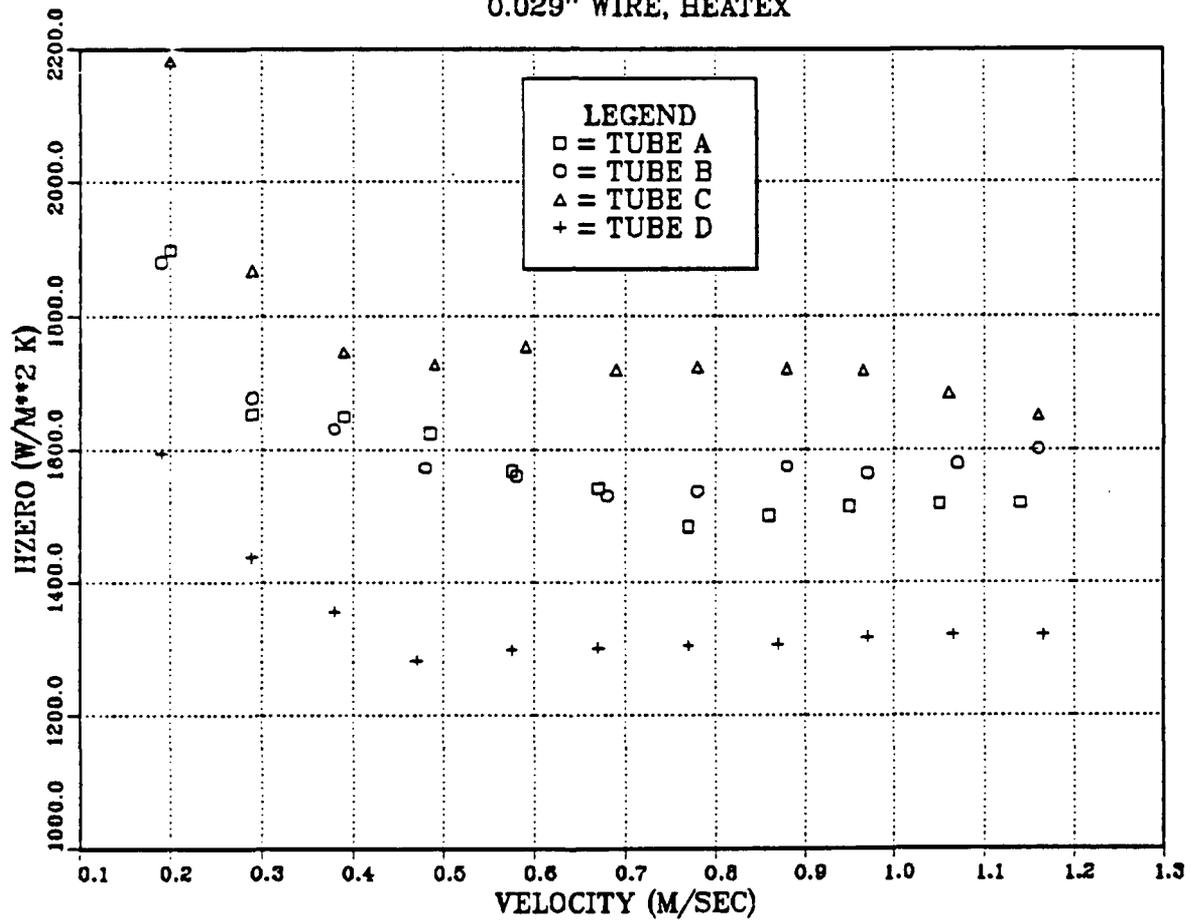
Figure 70. Effect of 0.029" wire on h_o for all tubes.

HZERO WIRE WRAPPED KORODENSE

0.049" WIRE, HEATEX
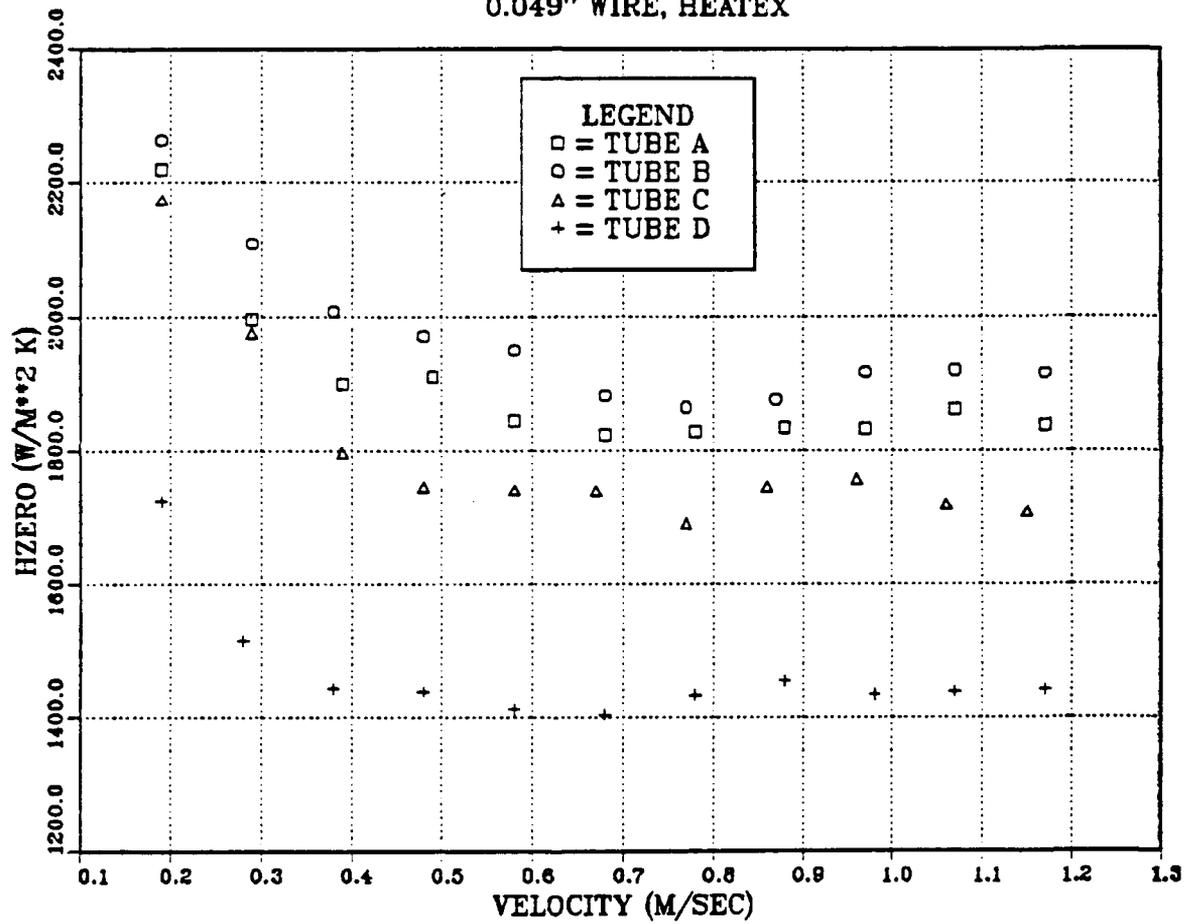
Figure 71.   Effect of 0.049" wire on h_o for all tubes.

HZERO WIRE WRAPPED KORODENSE

0.0675" WIRE, HEATEX
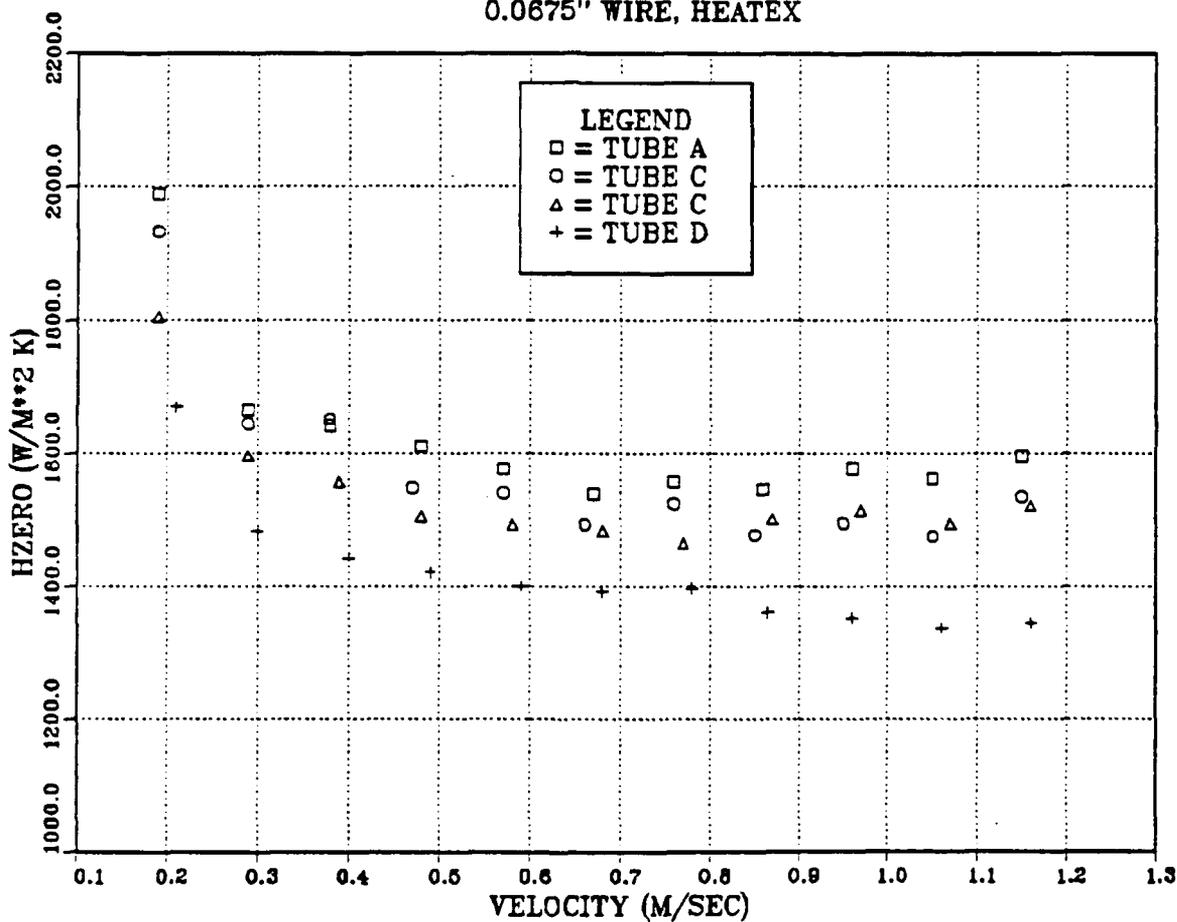
Figure 72.   Effect of 0.0675" wire on h_o for all tubes.

0.049" WIRE WRAPPED KORODENSE

KD11, R-113, HEATEX, BUNDLE
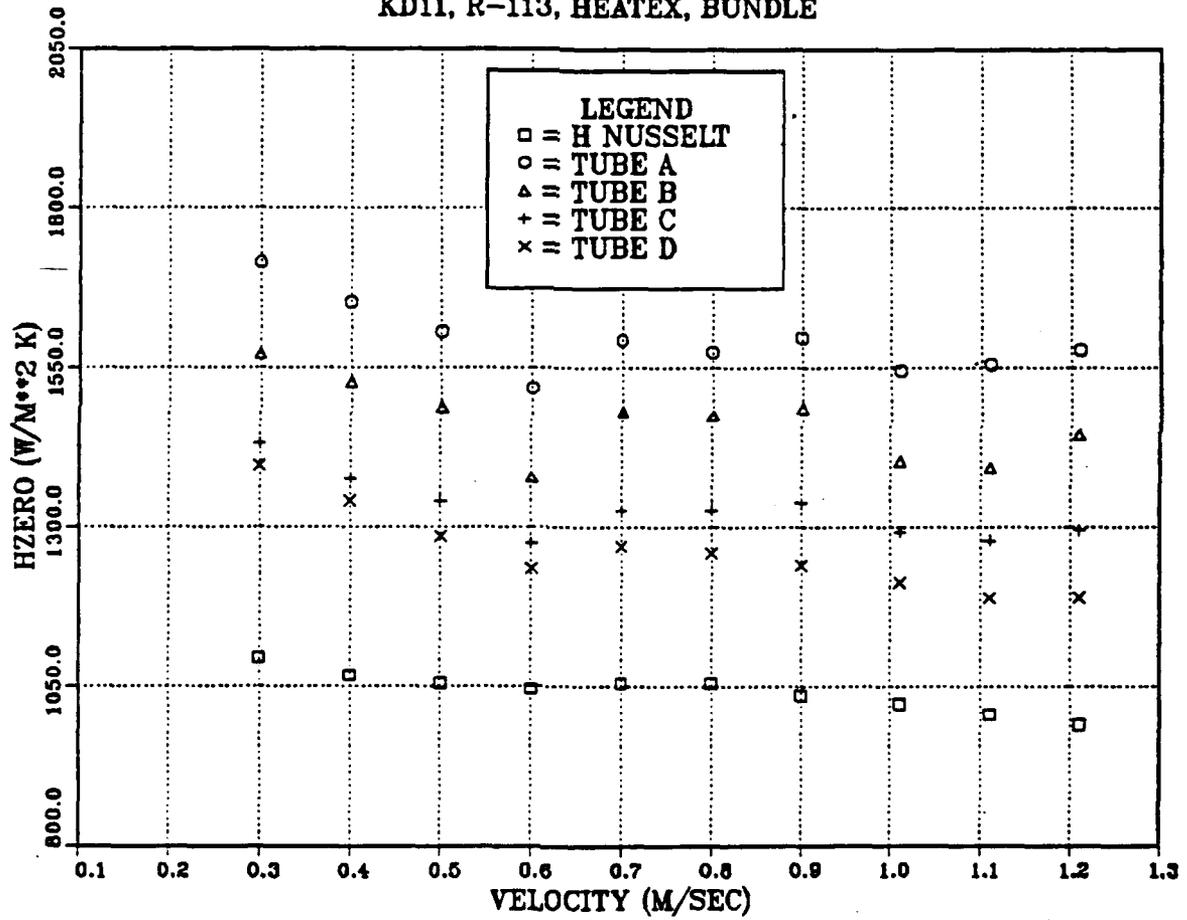
Figure 73. 0.049" wire wrapped KORODENSE tubes, bundle operation, expt. 1.

121

0.049" WIRE WRAPPED KORODENSE

KD12, R-113, HEATEX, BUNDLE
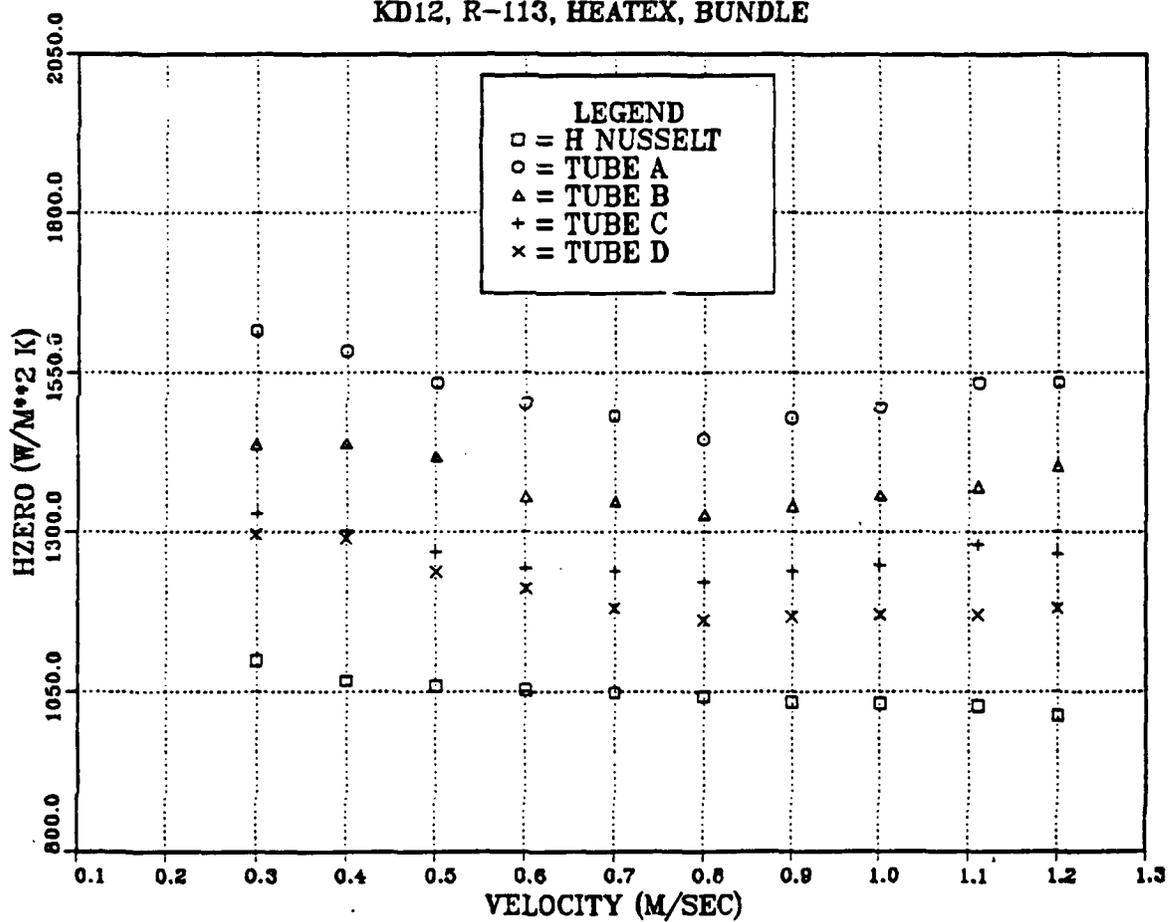
Figure 74.   0.049" wire wrapped KORODENSE tubes, bundle operation, expt. 2.

to note that the data for these single tube runs is somewhat higher than for the bundle runs (of the order of 11%). This may be due to uncertainty in the correlations used for h_i which are reflected in h_o

The values for the average and local vapor-side heat transfer coefficients are shown in Figure 77, Figure 78, Figure 79 and Figure 80 for the two experiments conducted with the 0.049″ wire wrapped KORODENSE tubes. The average vapor-side heat transfer coefficient is equal to that predicted by the Eissenberg model whereas the local vapor-side heat transfer coefficient is substantially above the Kern estimate. Recall that for the plain KORODENSE tubes, the average heat transfer coefficient was above the Kern estimate but below that predicted by the Eissenberg model. The local heat transfer coefficient was just above the Kern estimate. The differences between the plain and wire-wrapped KORODENSE tubes demonstrate that wrapping the tubes with wire substantially reduces the effects of condensate inundation.

5. Copper/Nickel Finned Tubes

The values for h_o obtained from experiments on copper/nickel finned tubes using the tape inserts are shown in Figure 81. The values obtained for the top tube are approximately 7 times greater than predicted by Nusselt theory. Condensate inundation caused about a 35% decrease in h_o for the second tube.

The values for h_o obtained using the same tubes but with the HEATEX elements are shown in Figure 82. In comparison to the values obtained for the tape inserts (see Figure 81), this data appears very scattered with enhancement over Nusselt theory of the order of 10 to 15 fold. This is a large discrepancy. It is possible that the correlation for the small diameter HEATEX element may be suspect. The data supplied by CAL GAVIN was collected on smooth copper tubes with a diameter similar to that of our copper/nickel finned tubes. This data was collected over a lower coolant velocity range than that used in this study. Therefore, the correlation for this HEATEX element may not be valid for the finned tube over the range of coolant velocities used here. This area needs to be studied further, perhaps even to gathering new experimental data which can be used to develop a new correlation.

The degree of inundation can be seen in Figure 83, Figure 84, Figure 85 and Figure 86. The data for the average vapor-side heat transfer coefficient for both HEATEX as well as tape inserts lie between the Nusselt and Kern models. This is also true for the local vapor-side heat transfer coefficients.

An interesting comparison is to look at the average and local heat transfer values for all tubes tested. Regarding the average heat transfer coefficient, for the smooth

HZERO WIRE WRAPPED KORODENSE

KD11, R-113, SINGLE, HEATEX
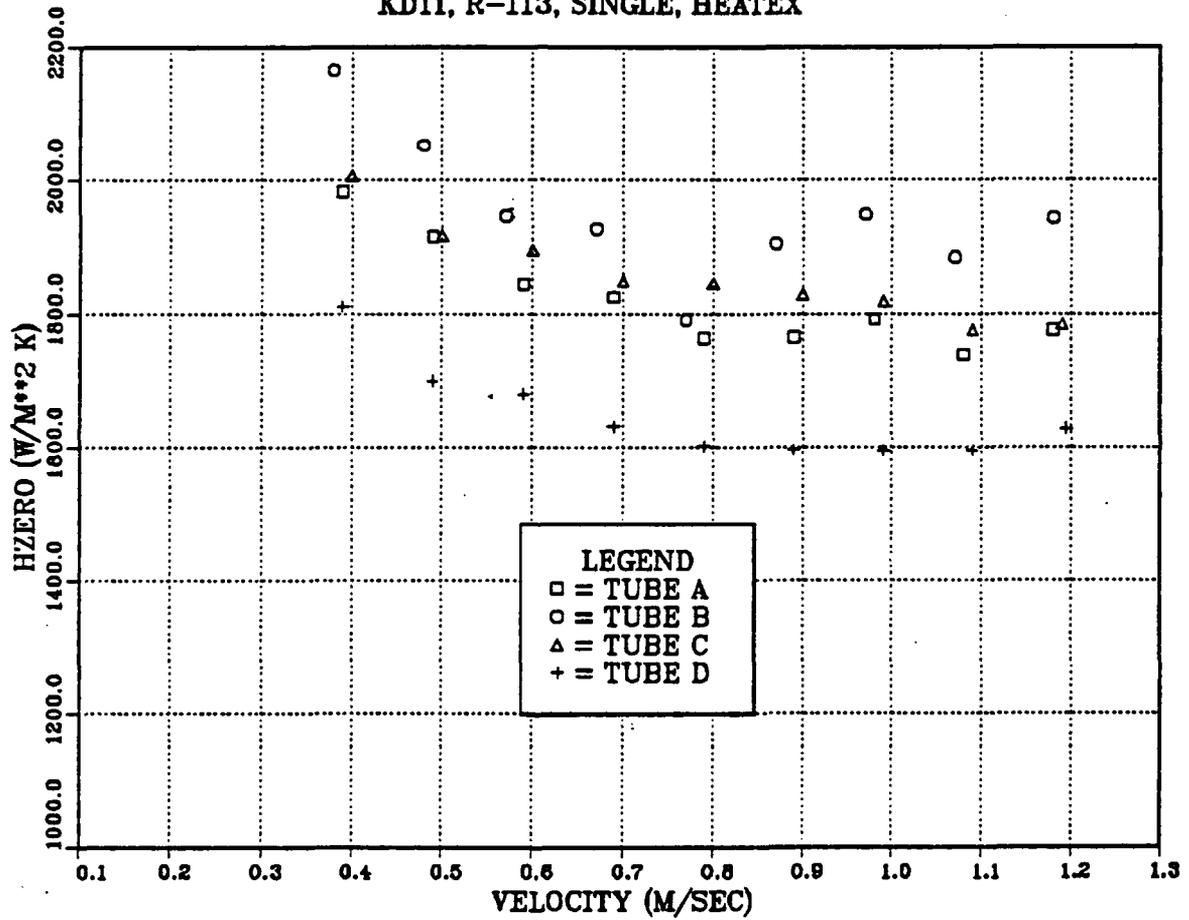
Figure 75.   0.049″ wire wrapped KORODENSE, individual runs, expt. 1.

HZERO WIRE WRAPPED KORODENSE

KD12, R-113, SINGLE, HEATEX
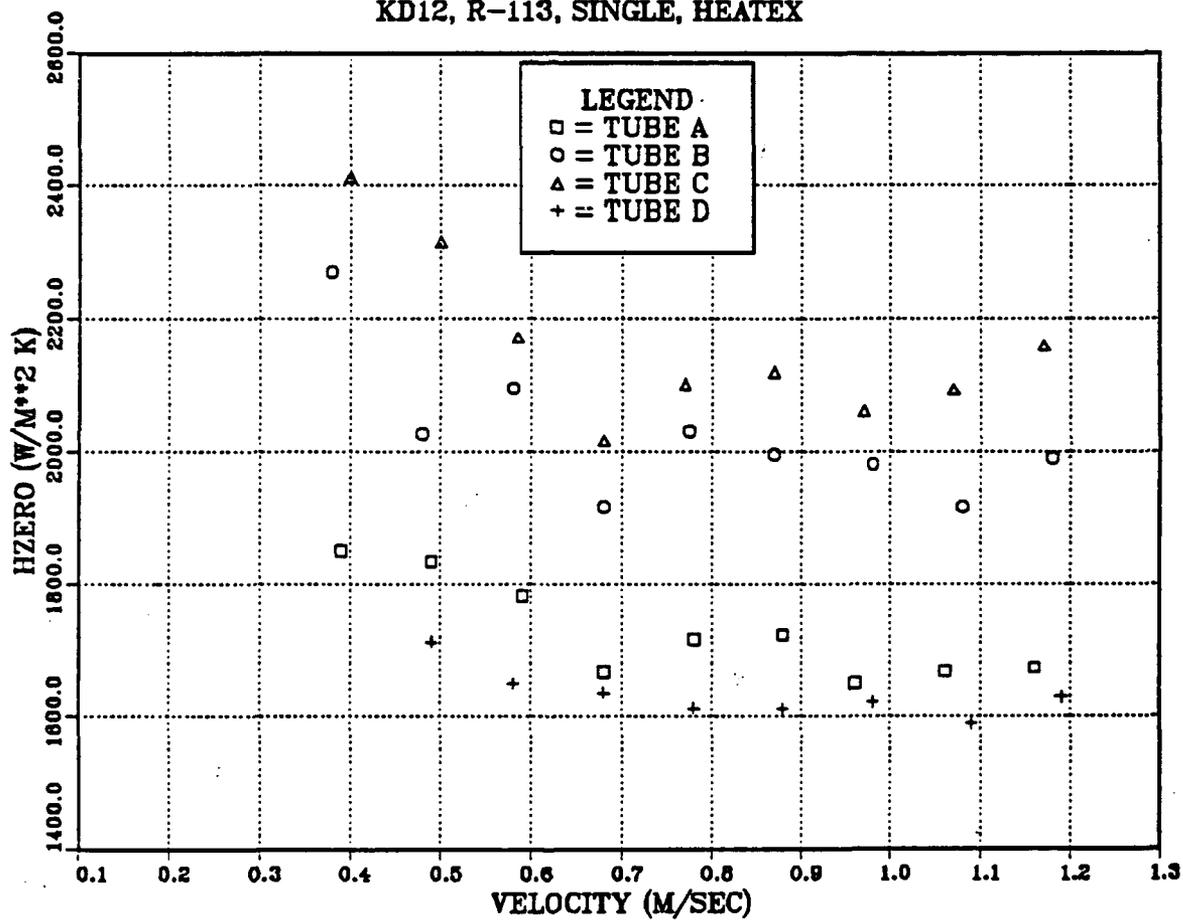
Figure 76.   0.049″ wire wrapped KORODENSE, individual runs, expt. 2.

HEAT TRANSFER COEFFICIENT

KD11, 0.049" WIRE, KORODENSE
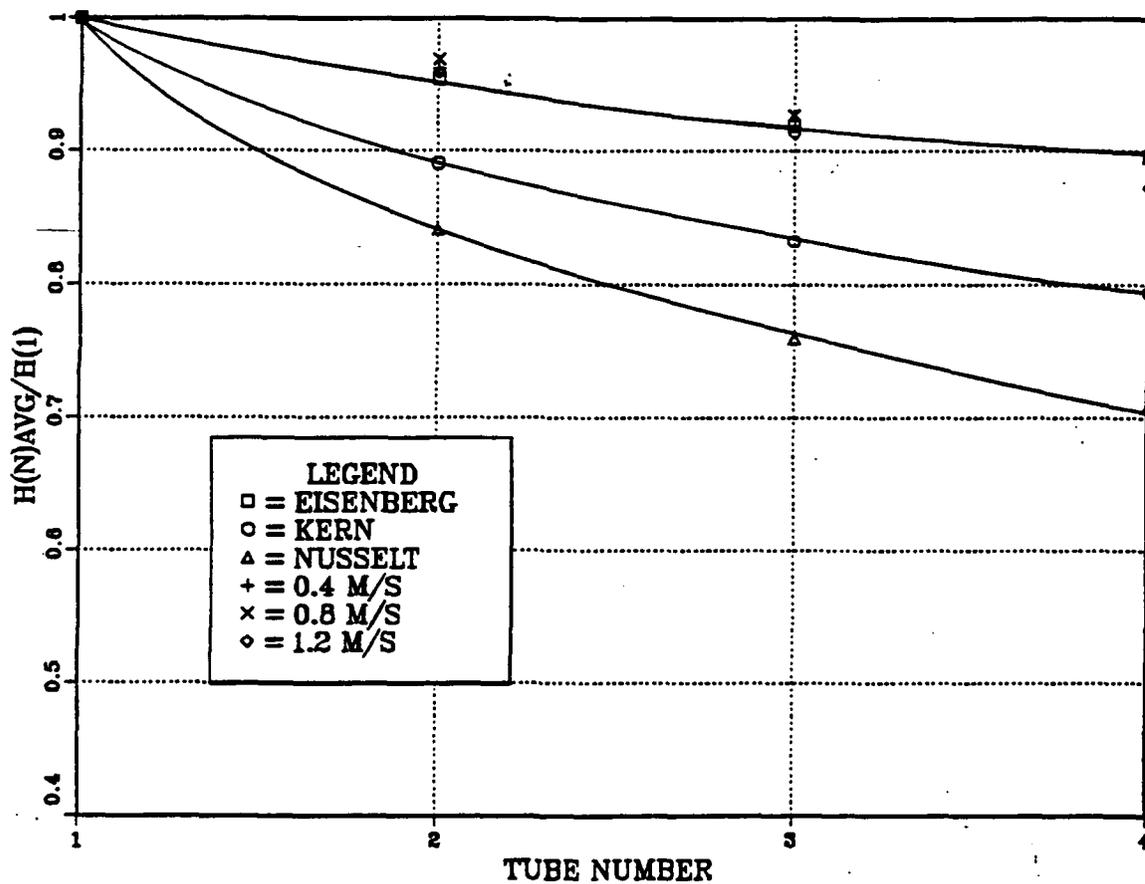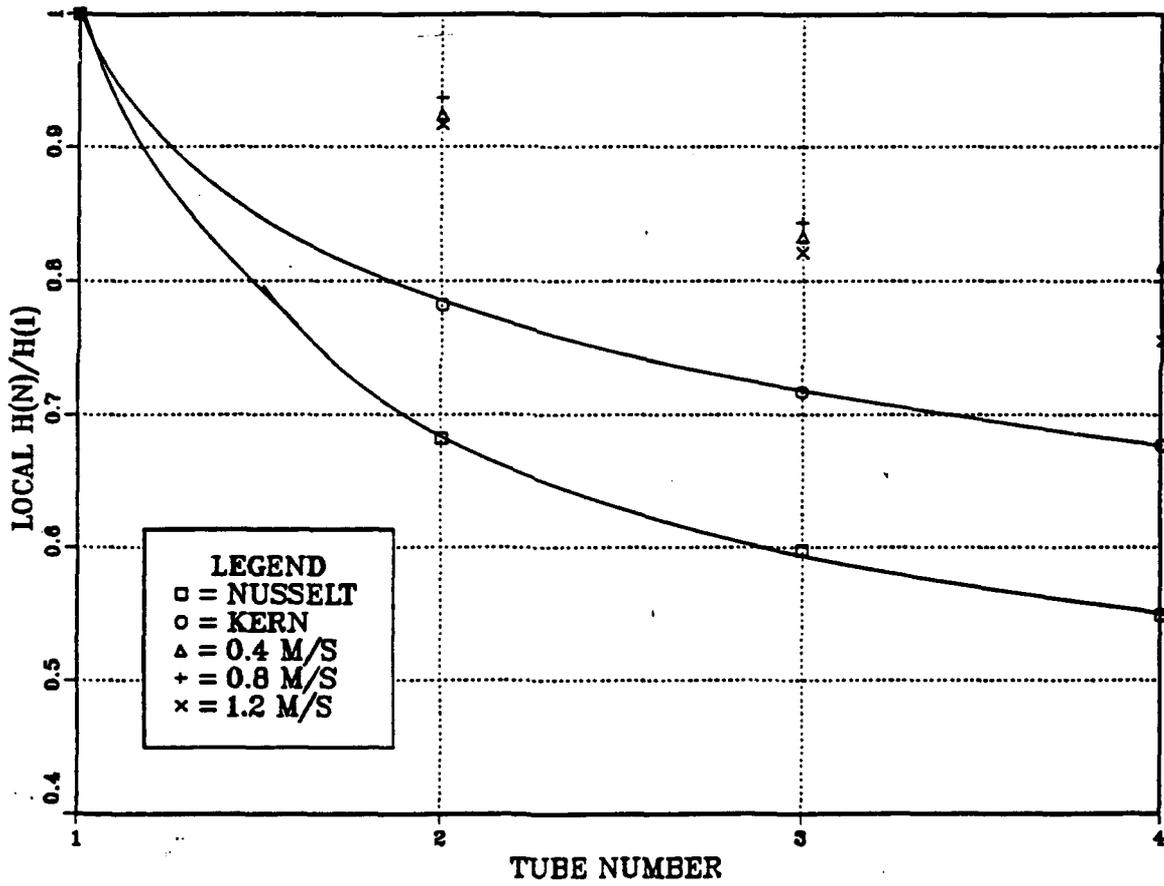
Figure 77.   Average h_o wire wrapped KORODENSE, expt. 1.

HEAT TRANSFER COEFFICIENT

KD12, 0.049" WIRE, KORODENSE

Figure 78.   Average h, wire wrapped KORODENSE, expt. 2.

Figure 79.   Local h_o wire wrapped KORODENSE, expt. 1.

HEAT TRANSFER COEFFICIENT

KD12, 0.049" WIRE, KORODENSE
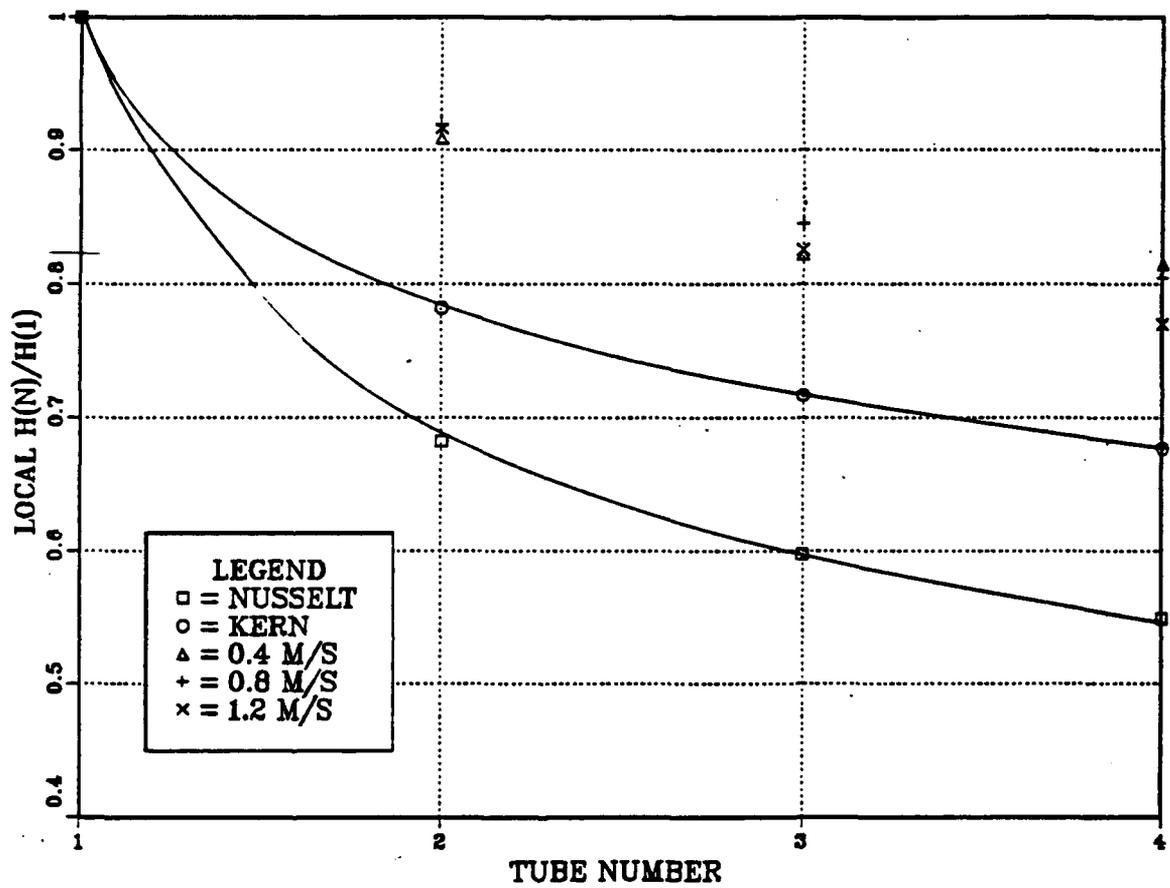
Figure 80.   Local h_e wire wrapped KORODENSE, expt. 2.

HZERO COPPER/NICKEL FIN TUBES

CF03, R-113, TAPE INSERT, BUNDLE
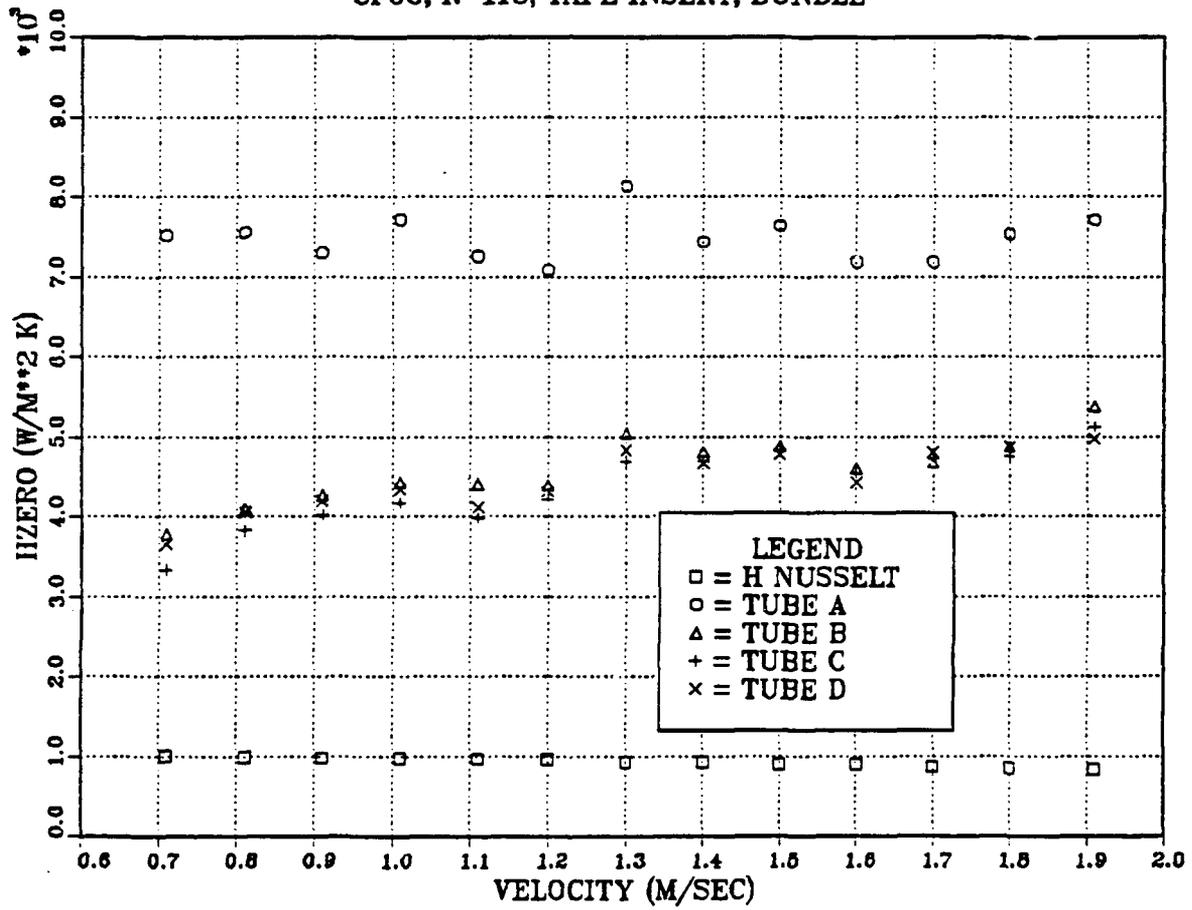
Figure 81. h_o for copper/nickel finned tubes, tape insert.

HZERO COPPER/NICKEL FIN TUBES

CF04, R-113, HITRAN, BUNDLE

Figure 82. h_o for copper/nickel finned tubes, HEATEX insert.

131
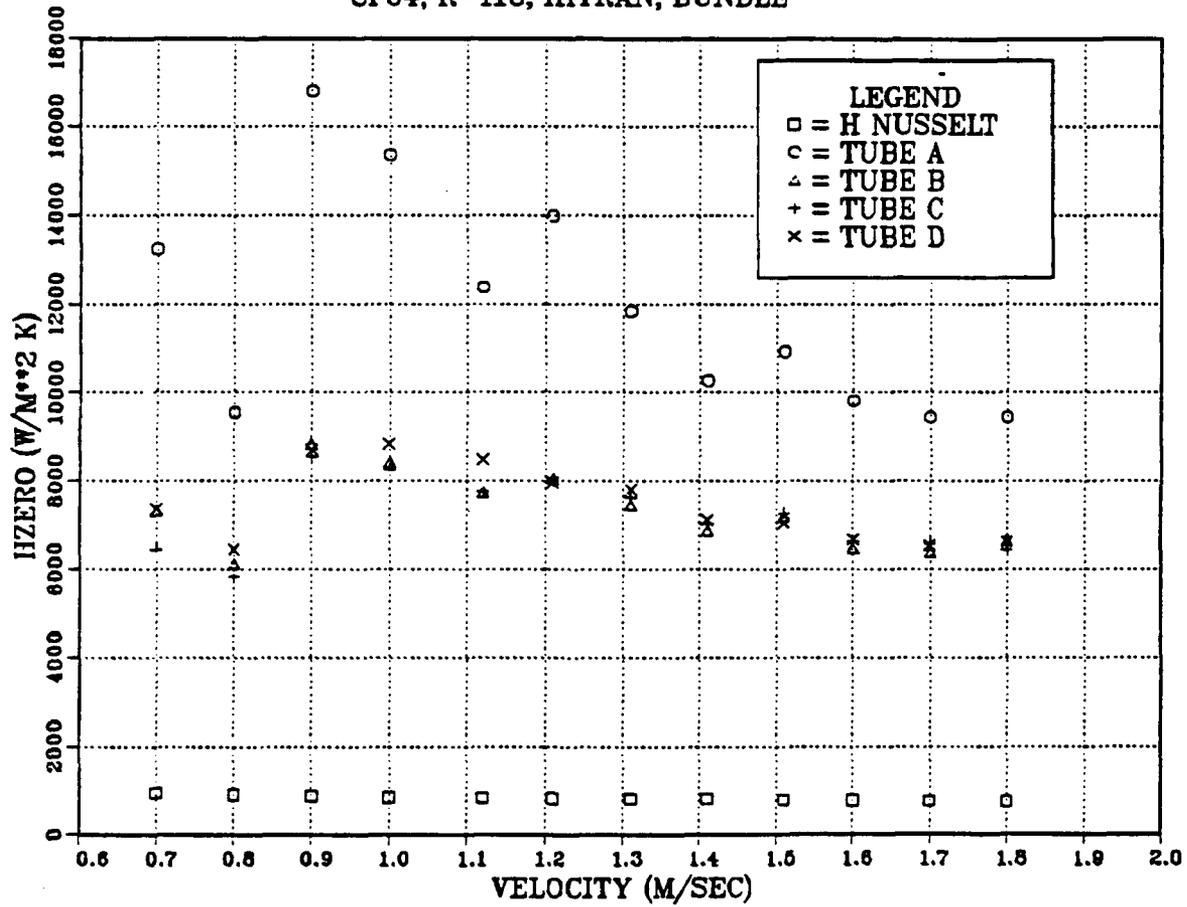
copper tubes, the values lay between the Kern and Eissenberg estimates. The plain KORODENSE tubes had values similar to those for the smooth copper tubes while the 0.049″ wire-wrapped KORODENSE had values which matched estimates based on Eissenberg's model. The copper/nickel finned tubes, on the other hand, had values which were between the Kern and Nusselt estimates. Since the lower the values the more condensate effect present, this data would imply that the effect of inundation is worse on the finned tubes and least on the wire-wrapped tubes. If the data for the local heat transfer coefficients is analyzed in a similar manner, the same conclusions are reached.

The wire wrapped KORODENSE tubes have the least effect of condensate inundation presumably because the wire serves to create a low pressure area at the base of the wire. This causes condensate to be pulled along the tube which enhances heat transfer. The values for the finned tubes are not in agreement with those of Honda et al. [Ref. 42]. These investigators found that the degree of inudation effect was substantially less than that reported here. There are two apparent reasons for this. The pitch-to-diameter ratio used by Honda et al. was 1.37 whereas in the present study it is 2.25. This difference may cause differences in the way condensate drains from one tube onto another. Alternatively, the correlations used to determine the heat transfer coefficient may be suspect. Hong and Bergles [Ref. 47] have reported that their correlation for the twisted tape has an uncertainty of about 15%. Therefore, it is possible that the large effect of condensate inundation is due more to uncertainty in the correlations than to a real phenomenon.

In order to assess the validity of the data obtained on the copper/nickel finned tubes, a comparison was made with the model of Beatty and Katz [Ref. 13]. In order to do this comparison, the fin efficiency had to be determined using equations from Incropera and DeWitt [Ref. 55]. It was assumed that the fins could be approximated as rectangular fins with adiabatic tips. The fin efficiency was then approximated by:

$$\eta_f = \frac{\tanh(mL_c)}{mL_c} \qquad (6-1)$$

According to Incropera and DeWitt, the term mL_c can be approximated by [Ref. 55]:

$$mL_c = \left(\frac{2h}{kA_p} \right)^{\frac{1}{2}} L_c^{\frac{3}{2}} \qquad (6-2)$$

HEAT TRANSFER COEFFICIENT

CU/NI FINNED TUBES, TAPE INSERT
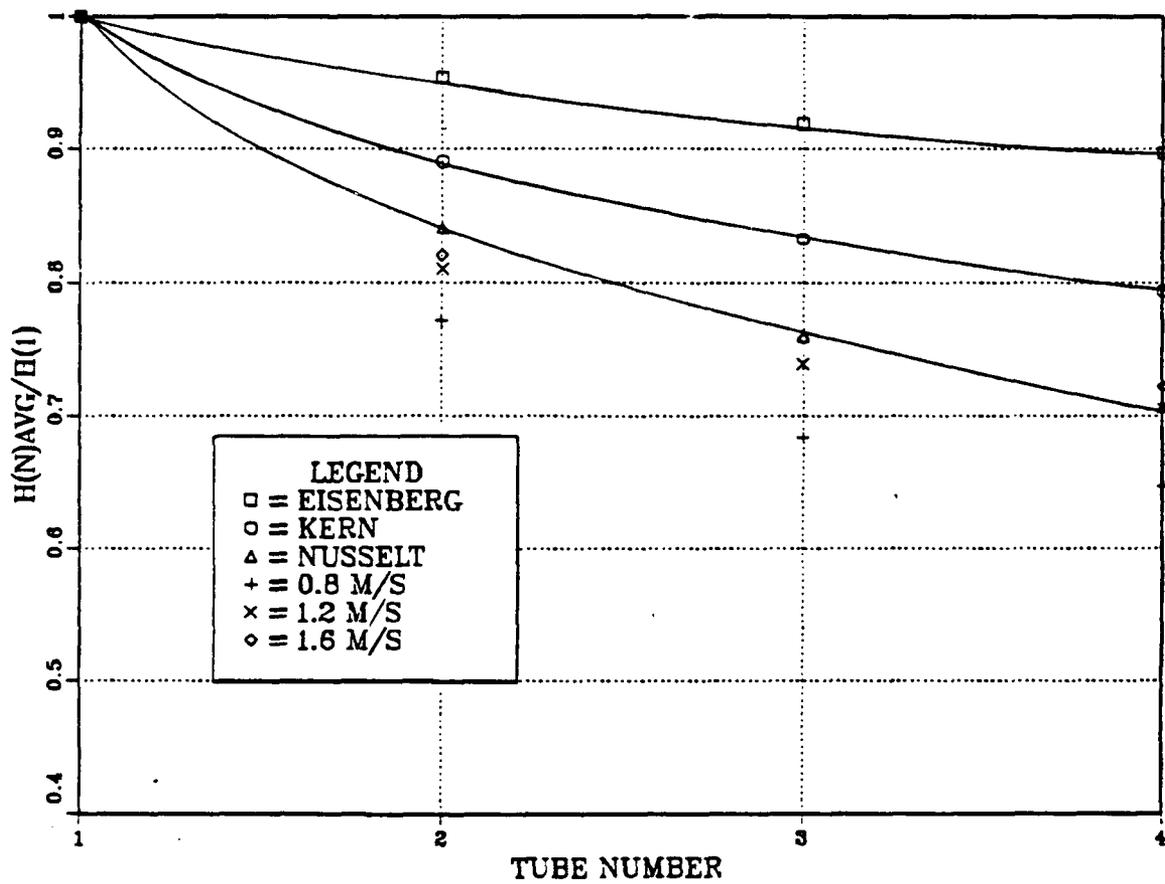
Figure 83.   Average h_o for Cu/Ni finned tubes, tape insert.

HEAT TRANSFER COEFFICIENT

CU/NI FINNED TUBES, HEATEX
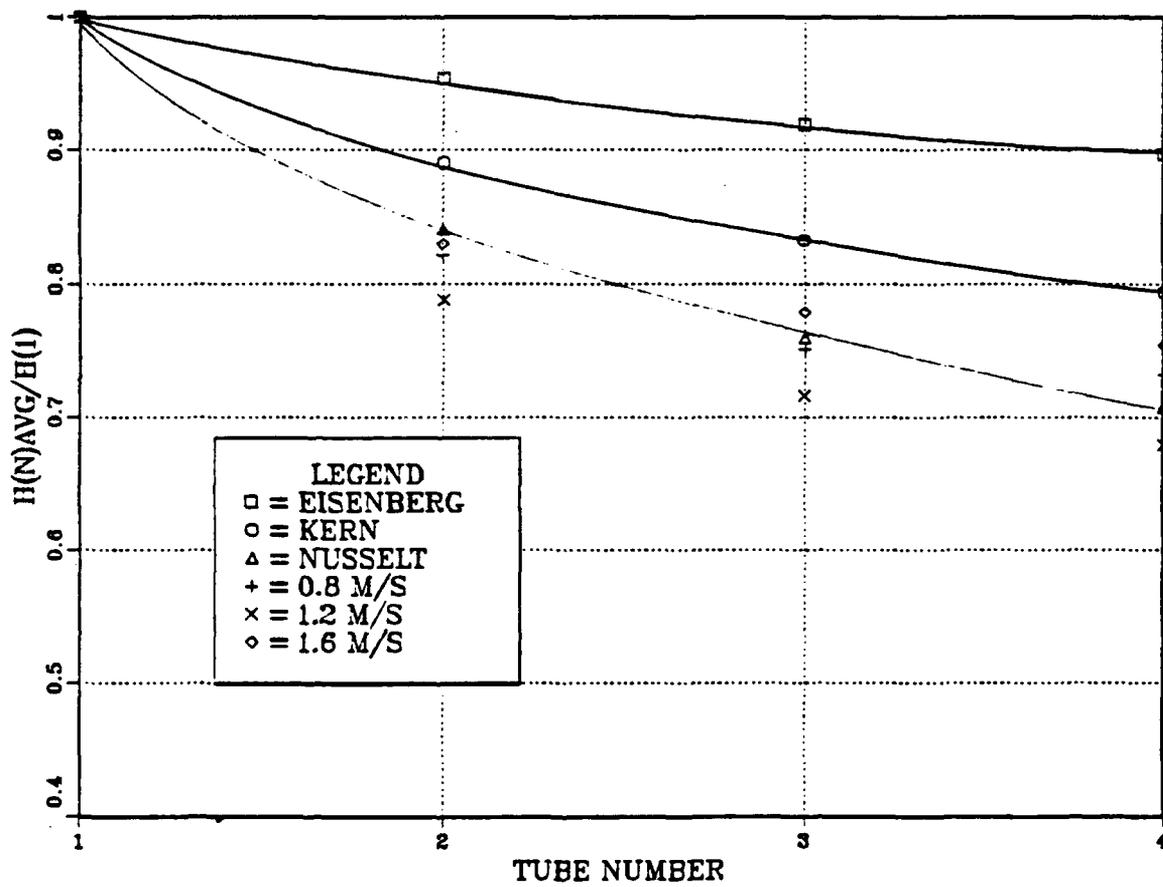
Figure 84.   Average h_o for Cu/Ni finned tubes, HEATEX insert.

HEAT TRANSFER COEFFICIENT
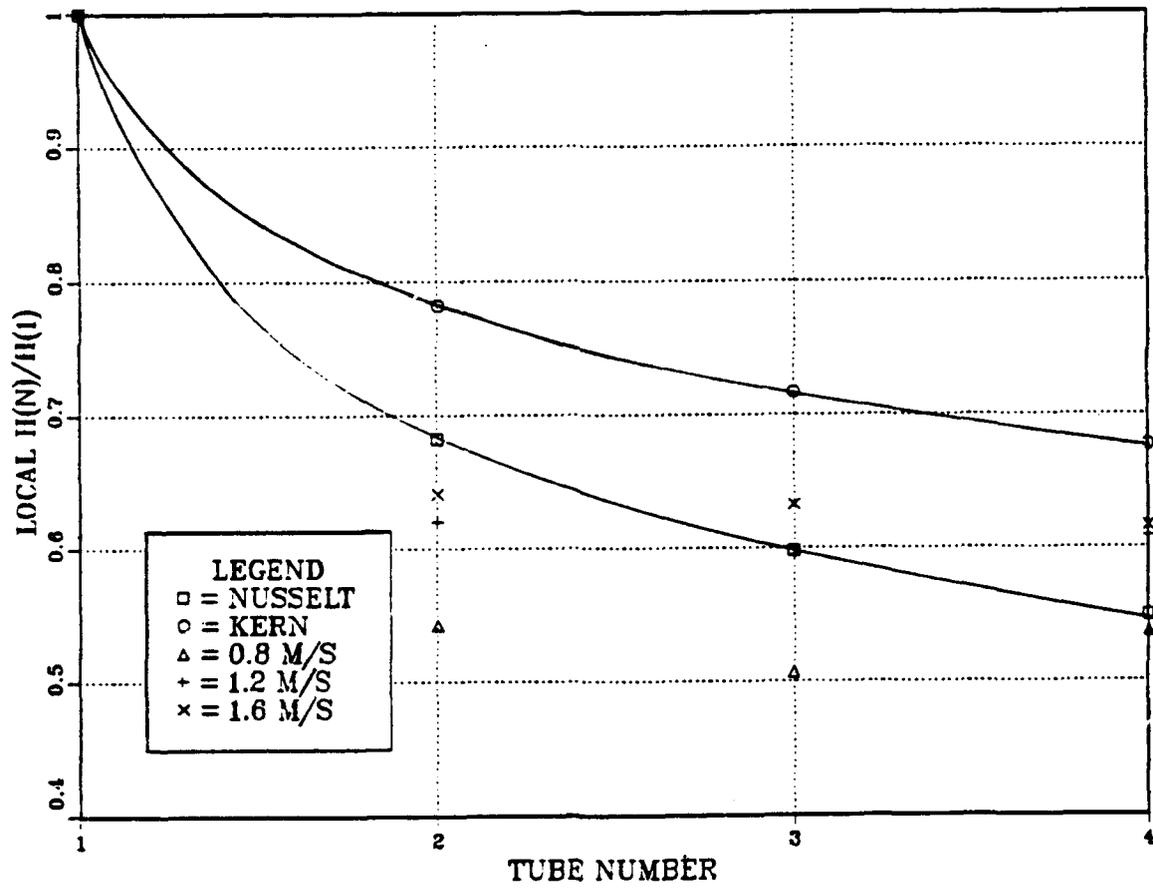
CU/NI FINNED TUBES, TAPE INSERT

LEGEND
□ = NUSSELT
○ = KERN
△ = 0.8 M/S
+ = 1.2 M/S
× = 1.6 M/S

Figure 85.   Local h_o for Cu/Ni finned tubes, tape insert.

HEAT TRANSFER COEFFICIENT

CU/NI FINNED TUBES, HEATEX

LEGEND
□ = NUSSELT
○ = KERN
△ = 0.8 M/S
+ = 1.2 M/S
× = 1.6 M/S
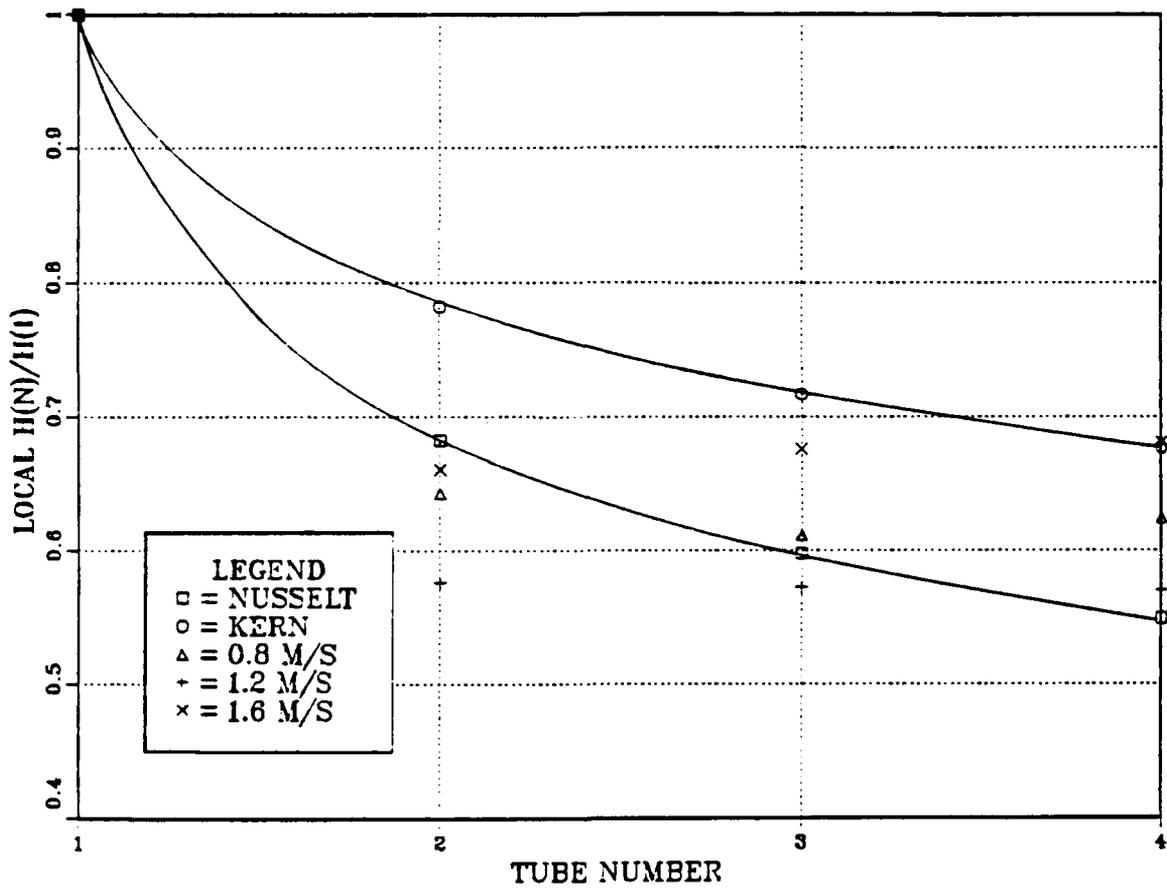
Figure 86.   Local h_o for Cu/Ni finned tubes, HEATEX insert.

where L_c is defined as:

$$L_c = L + \frac{t}{2} \tag{6-3}$$

Here, L is defined as the height of the fin and t is the fin thickness. The term A_p is defined as:

$$A_p = L_c t \tag{6-4}$$

In order to calculate a value on the fin efficiency, η_f, an iterative procedure was used. First, a value for h was guessed. Based on this value, η_f was calculated using equation (6-1). An effective area was then calculated from the following equation (Marto [Ref. 3]):

$$A_{ef} = \eta_f A_{fs} + \eta_f A_{ft} + A_u \tag{6-5}$$

where A_{fs} is the surface area of the fins sides, A_{ft} is the surface area of the fin tips and A_u is the interfin surface area. Using these values, the effective diameter, D_e was determined from (Marto [Ref. 3]):

$$\left[\frac{1}{D_e} \right]^{0.25} = 1.30 \eta_f \frac{A_{fs}}{A_{ef}} \frac{1}{\overline{L}^{0.25}} + \eta_f \frac{A_{ft}}{A_{ef}} \frac{1}{D_o^{0.25}} + \frac{A_u}{A_{ef}} \frac{1}{D_r^{0.25}} \tag{6-6}$$

where $\overline{L}$ is defined as:

$$\overline{L} = \frac{\pi(D_o^2 - D_r^2)}{4 D_o} \tag{6-7}$$

The heat transfer coefficient is then calculated from a modified version of the Beatty and Katz model (Eq. 2-2) which uses heat flux rather then the film temperature difference, ΔT.

$$h_o = .6085 \left(\frac{k^3 \rho^2 g h_{fg}}{\mu D_e q''_o} \right)^{\frac{1}{3}} \tag{6-8}$$

The results of these calculations for the copper/nickel finned tubes with the tape inserts are shown in Figure 87. In this figure, the values of h_o predicted from the Beatty and Katz model have been expressed in terms of the smooth tube area, that is $\pi d_o L$. Also note that values for h_o have been calculated for copper, copper/nickel and titanium. It is important to note that the raw data lies below the predicted values for copper/nickel

137

finned tubes. The data obtained after reprocessing with the modified Wilson plot procedure lies somewhat above the prediction. However, both lie within ± 20% which is within the range of validity for the Beatty and Katz model. Therefore, it would appear that the data obtained in the present study for the copper/nickel finned tubes with the tape inserts are reasonable.

The results for the copper/nickel finned tubes with the HEATEX elements installed are not as good. These results are shown in Figure 88. This figure shows the tremendous amount of scatter in the data irrespective of whether the raw data or the data obtained from the modified Wilson plot procedure is used. In either case, it is apparent that the data do not match values predicted from the model of Beatty and Katz. The only difference in these two experiments is the type of insert used. The values for h_o are based on an experimentally determined value for U_o and a value for h_i which is obtained from a correlation. In this study, the correlations for the HEATEX elements were based on empirical data supplied from the manufacturer. This data was obtained on smooth copper tubes having the same diameter as the copper/nickel finned tubes used in this study. Another point which must be considered is that the range of velocity for which the empirical data was supplied was from 0.2 to 1.2 m/s. However, for the copper/nickel finned tubes used in this study, a coolant velocity of 0.7 to 2 m/s was used. These factors would suggest that the correlation used for the small diameter HEATEX element is in error. Further research will have to verify whether or not this is the case.

D. IMPLICATIONS OF PRESENT WORK

This study used R-113 as the refrigerant and a refrigerated ethylene glycol/water mixture as the coolant. However, Naval refrigeration systems use R-114 for the refrigerant and sea water as the condensing medium. Therefore, one must ask whether the results obtained from the present work can be extended to Naval applications. In order to perform the following analysis, the dimensions of the smooth copper tubes and copper/nickel finned tubes (26 fpi) used in this study were assumed. The inside heat transfer coefficient was calculated using the Dittus-Boelter correlation. For the smooth tubes, h_o was calculated using the Nusselt equation, modified to use heat flux rather than the film temperature difference, ΔT. For the finned tube, h_i was again calculated using the Dittus-Boelter correlation. However, h_o was calculated using the modified Beatty and Katz equation (Eq. 6-8). The fin efficiency was assumed to be 1.0. The condenser design parameters were modeled after the requirements for the DDG-51. Hence, the refrigeration load was fixed at 242.7 tons. The number of condenser tubes was fixed at 340 tubes. Coolant inlet temperature was assumed to be 65 F (18.3 C) while the saturation temperature of R-114 was 107.8 F (42.1 C). The coolant velocity was held constant at 2 m, s.

138
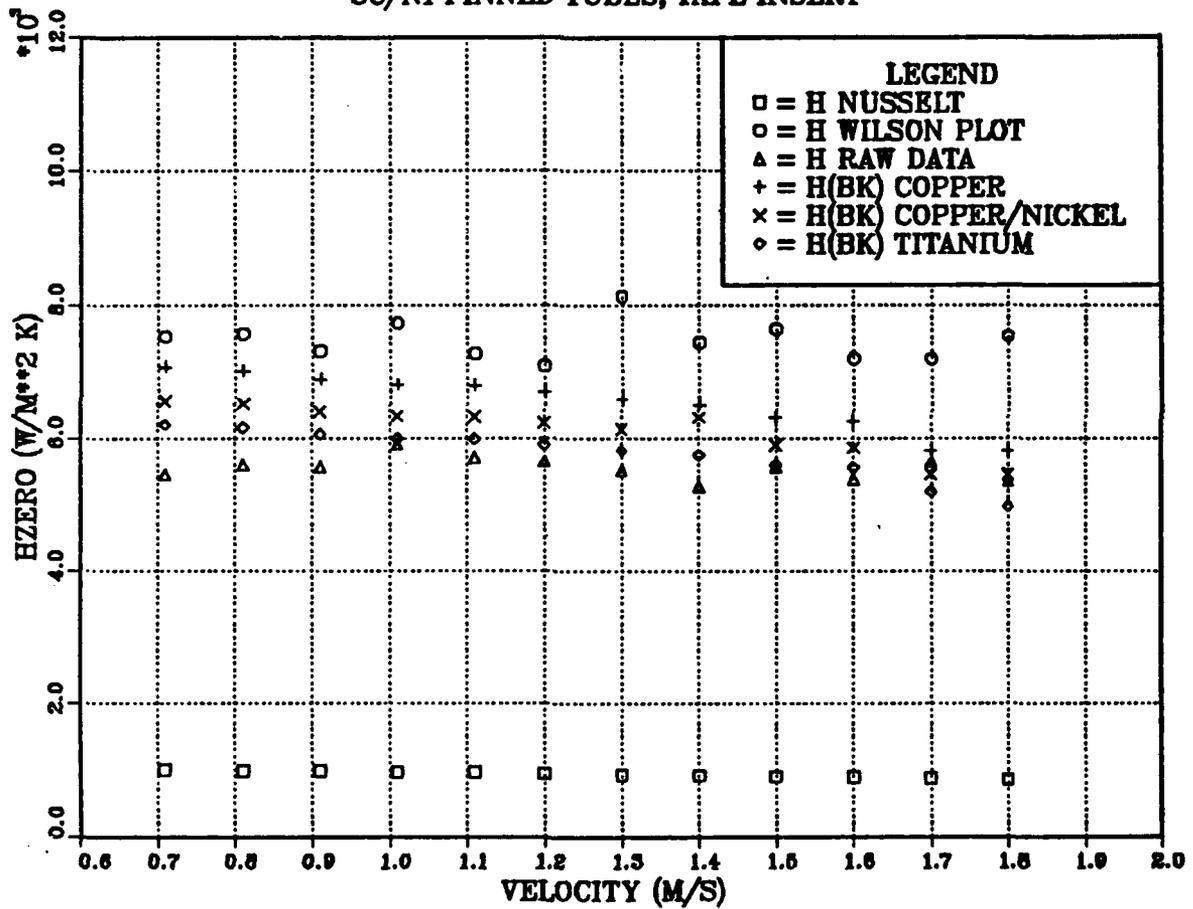
Figure 87.   Experimental data (tape insert) vs. Beatty/Katz model.

139

HEAT TRANSFER COEFFICIENT

CU/NI FINNED TUBES, HEATEX
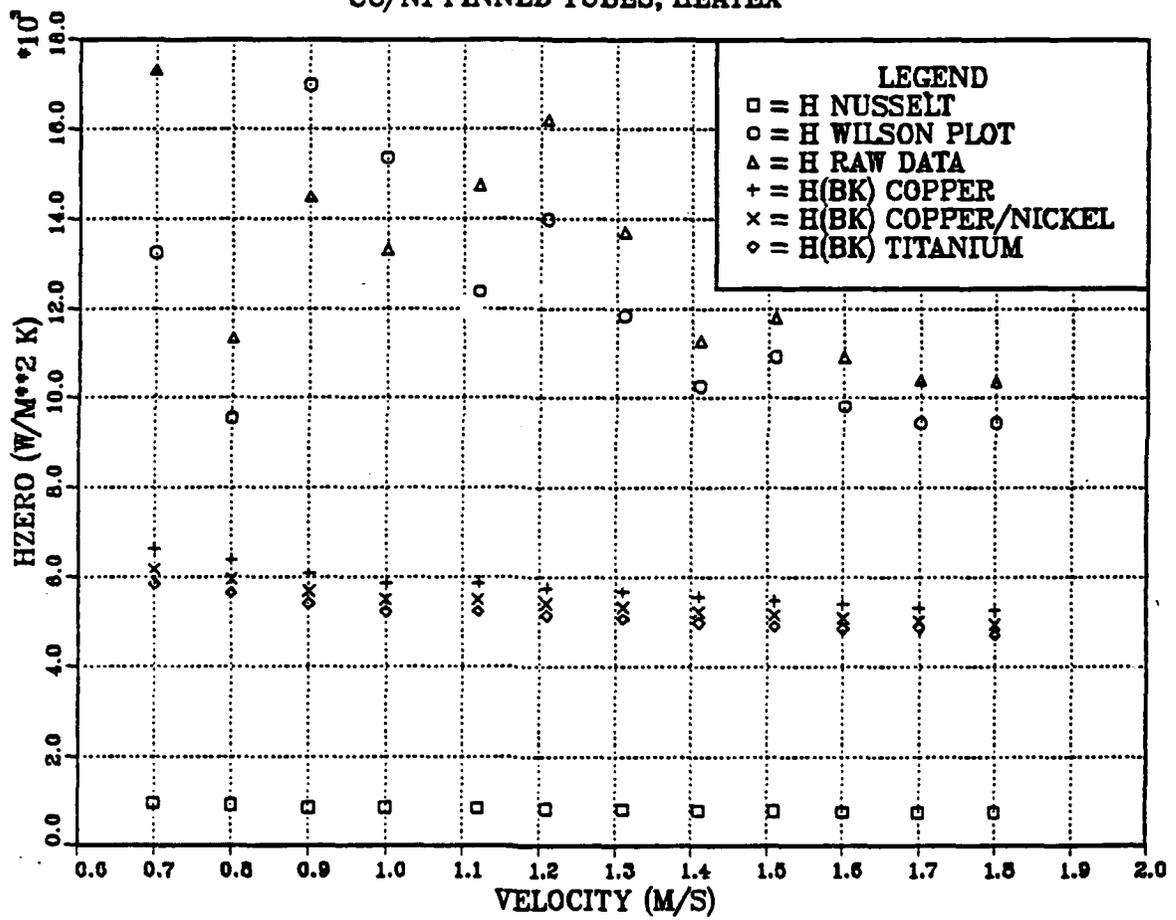
Figure 88.   Experimental data (HEATEX insert) vs. Beatty/Katz model.

Ratios of the surface areas for finned to smooth tube and a finned KORODENSE tube to smooth tube were calculated. In order to carry out this calculation for the KORODENSE tube, some type of correlation must be used to calculate h_i. However, as shown by Cunningham and Ben Boudinar [Ref. 56] in their work on steam condensation on roped tubes, the enhancement due to the roping depends upon the type of groove cut into the tube. In this analysis, an enhancement of 1.6 for h_i was assumed for the KORODENSE tube over the finned tube. Once the area ratio is obtained, a condenser tube length can be calculated. This is done as follows. The area ratio can be expressed in terms of length by:

$$\frac{A_{o,f}}{A_{o,s}} = \frac{Na_f L_f}{Npid_o L_s}$$

(6 — 9)

where $A_{o,f}$ is the area of the finned tube, $A_{o,s}$ is the area for the smooth tube, and N is the number of tubes. The term a_f is given by:

$$a_f = (A_f + A_o)N$$

(6 — 10)

where A_f and A_o are the finned and unfinned areas respectively.

Table 3 shows the results of this calculation. For the finned tube, the area ratio of 1.75 leads to a decrease in the condenser length of about 50%. On the other hand, if a roped KORODENSE tube could be manufactured which had fins on the outside surface, the reduction in length of the condenser tubes would be 58%. This could lead to even greater weight savings for refrigeration systems.

Table 3.   SUMMARY OF AREA RATIO CALCULATIONS

	Smooth tube	Finned Tube	KORODENSE + Fins
$\dfrac{A_{o,f}}{A_{o,s}}$	1.0	1.75	1.51
$\dfrac{L_f}{L_s}$	1.0	0.497	0.42

VII. CONCLUSIONS

The following conclusions can be made based on the results obtained in this study.

A. GENERAL

1. The condenser/evaporator test platform is fully operational and shows no signs of the contamination problems reported by Mabrey [Ref. 2].

2. Values for outside heat transfer coefficient obtained using smooth copper tubes appear to be reasonable and agree with other published data.

B. OVERALL HEAT TRANSFER COEFFICIENT

1. KORODENSE tubes provide enhancements of 6 to 8% over smooth copper tubes regardless of the type of insert used.

2. For the KORODENSE tubes used in this study, wire wrapping results in an enhancement in heat transfer.

3. The optimal pitch-to-wire diameter was about 7 for the KORODENSE tubes used in the present work.

4. Wrapping the KORODENSE tubes with 0.049" diameter wire resulted in an enhancement of 16% over the non wire-wrapped KORODENSE tubes and a 22 to 24% enhancement over the smooth copper tubes.

5. Copper/nickel finned tubes (26 fpi) yielded a doubling of the heat transfer in comparison to smooth copper tubes.

6. The use of HEATEX radial mixing elements yielded a 10 to 15% increase in comparison to the use of twisted tape inserts.

C. OUTSIDE HEAT TRANSFER COEFFICIENT

1. The smooth tubes yielded values of h_o which were 7 to 15% below those estimated from Nusselt theory.

2. The KORODENSE tubes gave values of h_o which were about 15% greater than those estimated from Nusselt theory and approximately 25% greater than values obtained for the smooth copper tubes.

3. Wrapping the KORODENSE tube with 0.049" diameter wire resulted in a 90% increase in the outside heat transfer coefficient in comparison to the smooth copper tubes.

4. The copper/nickel finned tubes yielded a 7 fold increase in h_o over the smooth copper tubes.

D. EFFECTS OF CONDENSATE INUNDATION

1. For smooth copper tubes, condensate inundation resulted is a 15 to 18% decrease in h_o for the second tube compared to the top tube.

142

2. KORODENSE tubes showed a 20% decrease in h_o for this same comparison.

3. Wrapping the KORODENSE tubes with 0.049" diameter wire resulted in a decrease of only 5 to 10% in h_o for the second tube compared to the top tube.

4. For the copper/nickel finned tubes, condensate inundation resulted in almost a 35% decrease in h_o for the second tube relative to the top tube. This large decrease is presumably due to uncertainty in the correlations used to determine the heat transfer coefficient.

VIII. RECOMMENDATIONS

Based on the results of this study, the following recommendations are made:

A. GENERAL RECOMMENDATIONS

1. The ball valves which control coolant flow to the flowmeters should be replaced with needle valves. This will provide better control of coolant flow.

2. The flowmeters should be replaced with larger capacity meters. This will allow investigation of heat transfer at higher coolant velocities and therefore higher condensation rates. This change will have to be accompanied by an increase in the diameter of the coolant return header in order to accommodate the increased coolant flow.

3. Extra lines from the top ends of the condenser to the cold trap need to be added. This will ensure that no isolated pockets of noncondesibles exist in the condenser.

B. EXPERIMENTAL RECOMMENDATIONS

1. The correlation for the small diameter HEATEX element must be re-investigated. This may require obtaining additional data from CAL GAVIN or designing an experiment at the Naval Postgraduate School which can be used to obtain the required data.

2. The correlation used for the twisted tape insert also needs to be re-evaluated in light of the large inundation effects. As with the HEATEX, this may require that additional experiments be carried out at the Naval Postgraduate School.

3. Once a reliable correlation is obtained, the experiments conducted on the copper/nickel finned tubes need to be repeated.

4. Experiments need to be conducted using both titanium finned tubes as well as titanium KORODENSE tubes. This data can then be compared to that obtained with the copper/nickel finned tubes and the copper/nickel KORODENSE tubes to determine whether titanium tubes can deliver the required performance.

5. Smooth copper tubes should be wrapped with different wire diameters at different pitches in order to determine an optimal pitch-to-wire diameter ratio.

6. Experiments need to be conducted on wire wrapping of KORODENSE tubes which have a different pitch. These experiments should be based on results obtained from the smooth copper tube experiments.

7. Experiments need to be repeated using R-114 and possible alternate refrigerants that the Navy is interested in such as R-24.

8. A KORODENSE type tube which is roped on the inside and finned on the outside needs to be developed and tested since calculations have shown that this type of tube can lead to smaller condensers.

APPENDIX A. FLOWMETER CALIBRATION

The flowmeters used to measure flow through the main condenser tubes were calibrated in the following manner. The meters were disconnected from the condenser tubes. An empty 55 gallon drum was placed on a scale (resolution 0.5 lbs). The condenser pump was started with flow through the bypass.

Flowmeters were calibrated at 0 °C and 24 °C. Each flowmeter was initially opened to maximum flow. The flow was then throttled back to 10%. The weight of fluid in the drum was recorded and the timer started. The weight was then recorded at the end of the calibration time. This procedure was repeated in 10% increments up to 100% flow. Flowmeter number 2 was only calibrated up to 90% flow since a mechanical stop prevented it from going to 100%. The calibration runs were then repeated from the maximum flow back to 10% flow in 10% increments. The mass flow rate was then calculated in kg/sec based on an average of the two readings. The data for the flowmeter calibrations as a function of nominal flow rate are shown in Table 4.

Table 4.   MASS FLOW RATE (KG/SEC) AT 0 AND 24 C.

Nominal Flow Rate (%)	Flow Meter A		Flow Meter B		Flow Meter C		Flow Meter D	
	0°C	24°C	0°C	24°C	0°C	24°C	0°C	24°C
10	0.0149	0.0172	0.0127	0.0181	0.0141	0.0179	0.0253	0.0318
20	0.0363	0.0403	0.0308	0.0408	0.0374	0.0391	0.0513	0.0613
30	0.0544	0.0621	0.0531	0.0626	0.0566	0.0616	0.0798	0.0894
40	0.0794	0.0851	0.0739	0.0844	0.0793	0.0899	0.1129	0.1216
50	0.1003	0.112	0.0953	0.1102	0.1000	0.1103	0.1406	0.1555
60	0.1179	0.1339	0.1134	0.1295	0.1211	0.1318	0.1728	0.1823
70	0.1388	0.1556	0.1379	0.1506	0.1427	0.1552	0.1975	0.2114
80	0.1563	0.1769	0.1533	0.1715	0.1639	0.1716	0.2261	0.2409
90	0.1846	0.2028	0.1799	0.1912	0.1813	0.1975	0.2461	0.2667
100	0.2019	0.2151			0.2034	0.2183	0.2788	0.2924

A least squares linear regression was performed for each flowmeter at each of the two temperatures for which the calibration runs were carried out. The linear regression equation takes the following form:

$$\dot{m} = a + bN$$

where a is the y-intercept in kg/s, b is the slope in kg/s/% and N is the flowmeter setting in %. The results are shown in Table 5.

Table 5.   FLOWMETER CALIBRATION REGRESSION RESULTS.

	Flow Meter A		Flow Meter B		Flow Meter C		Flow Meter D	
	0 °C	24 °C	0 °C	24 °C	0 °C	24 °C	0 °C	24 °C
Slope	0.00206	0.00225	0.00204	0.00216	0.00208	0.0022	0.0028	0.0029
Y intercept	-0.00414	-0.00329	-0.00615	-0.00138	-0.0036	-0.00211	-0.00122	0.0048

In order to obtain flowrates at any intermediate temperatures, a simplified interpolation procedure was used. The results of the calibrations for the four flowmeteres are shown in Figure 89, Figure 90, Figure 91, and Figure 92.

FLOWMETER A
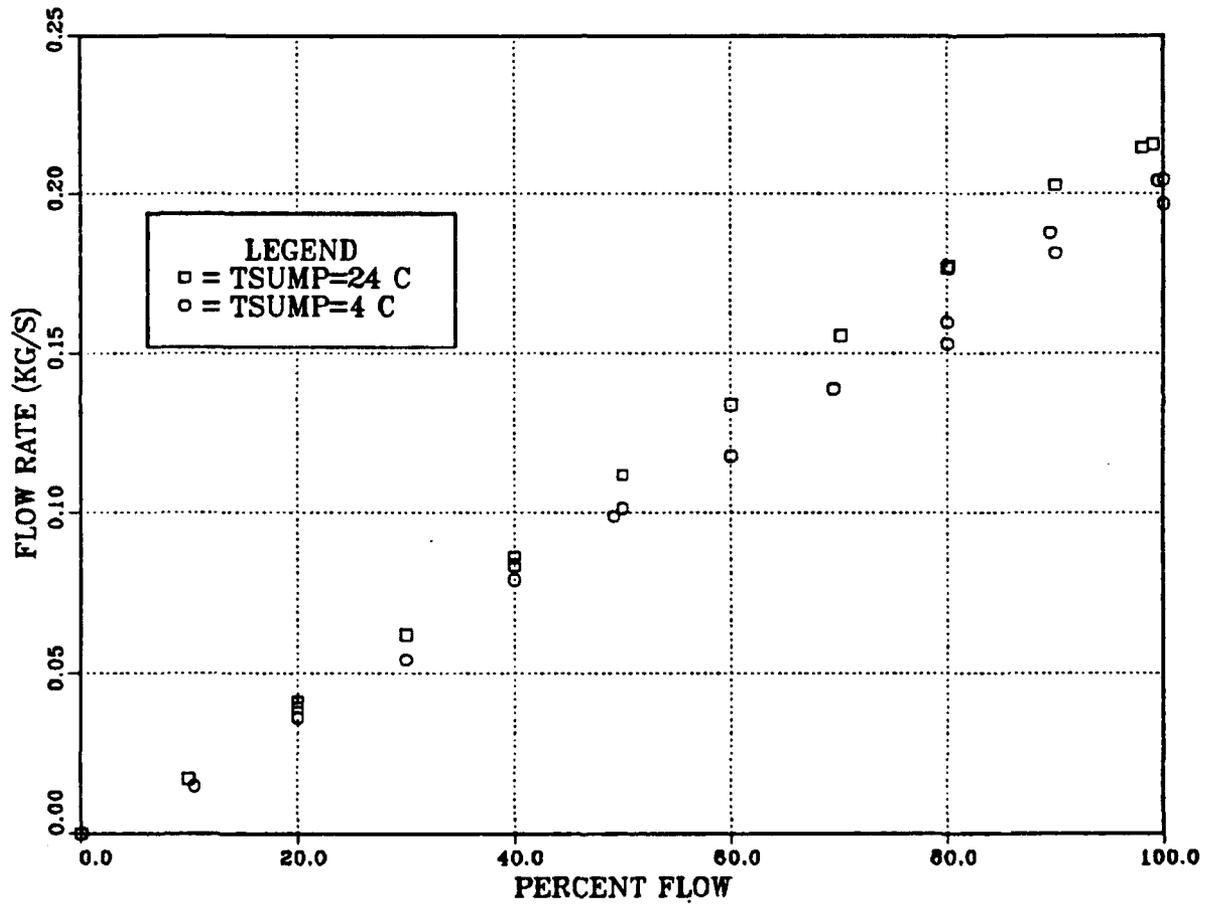
Figure 89.   Calibration for flowmeter A.

147
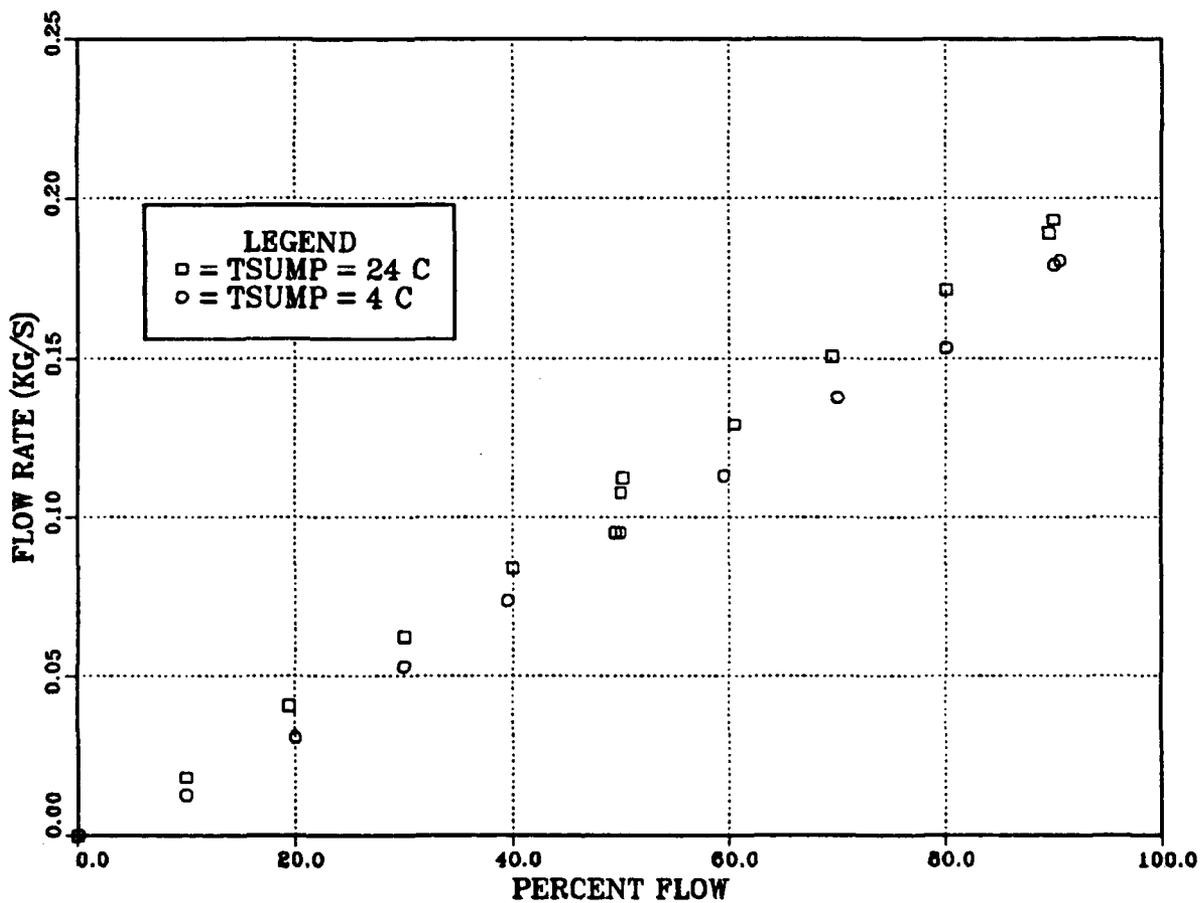
FLOWMETER B

Figure 90. Calibration for flowmeter B.

FLOWMETER C
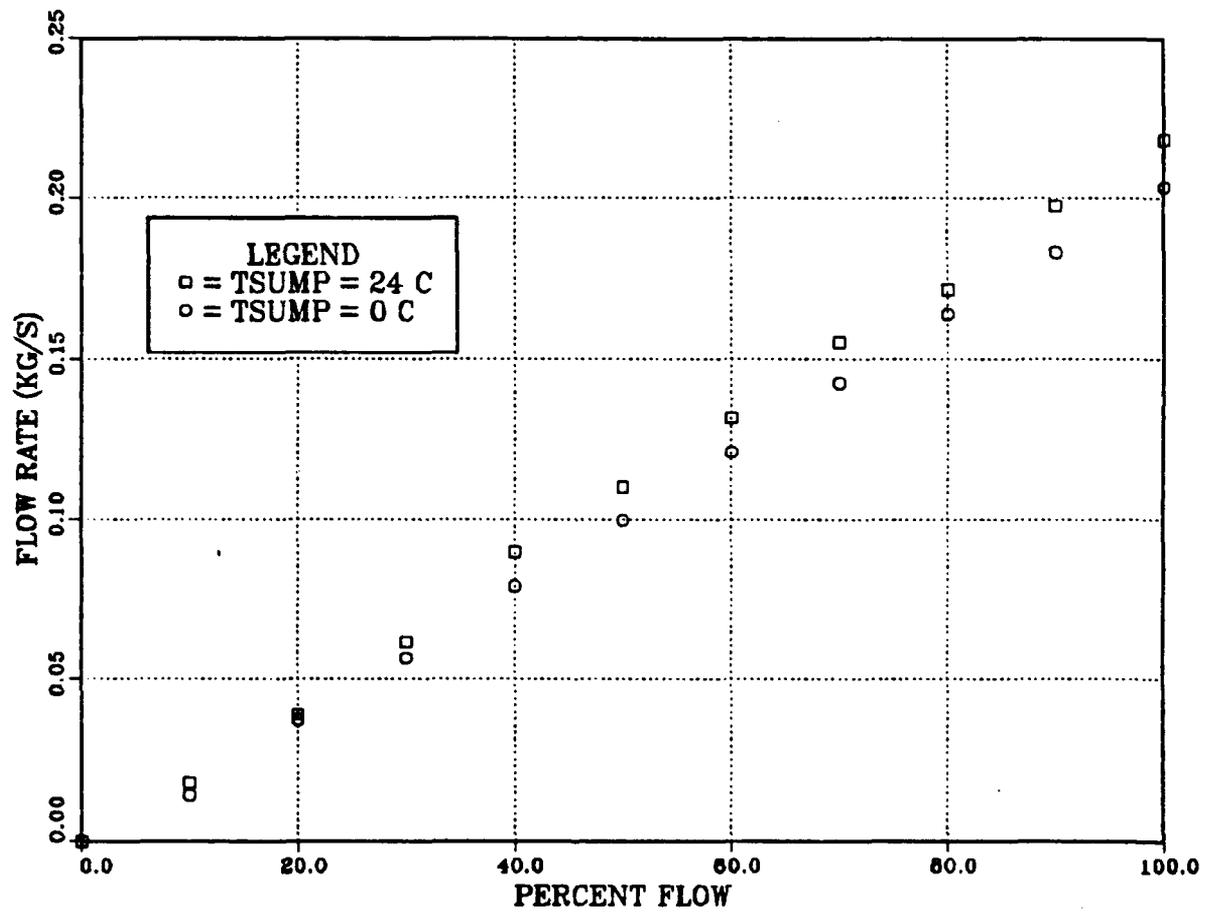
Figure 91. Calibration for flowmeter C.

FLOWMETER D
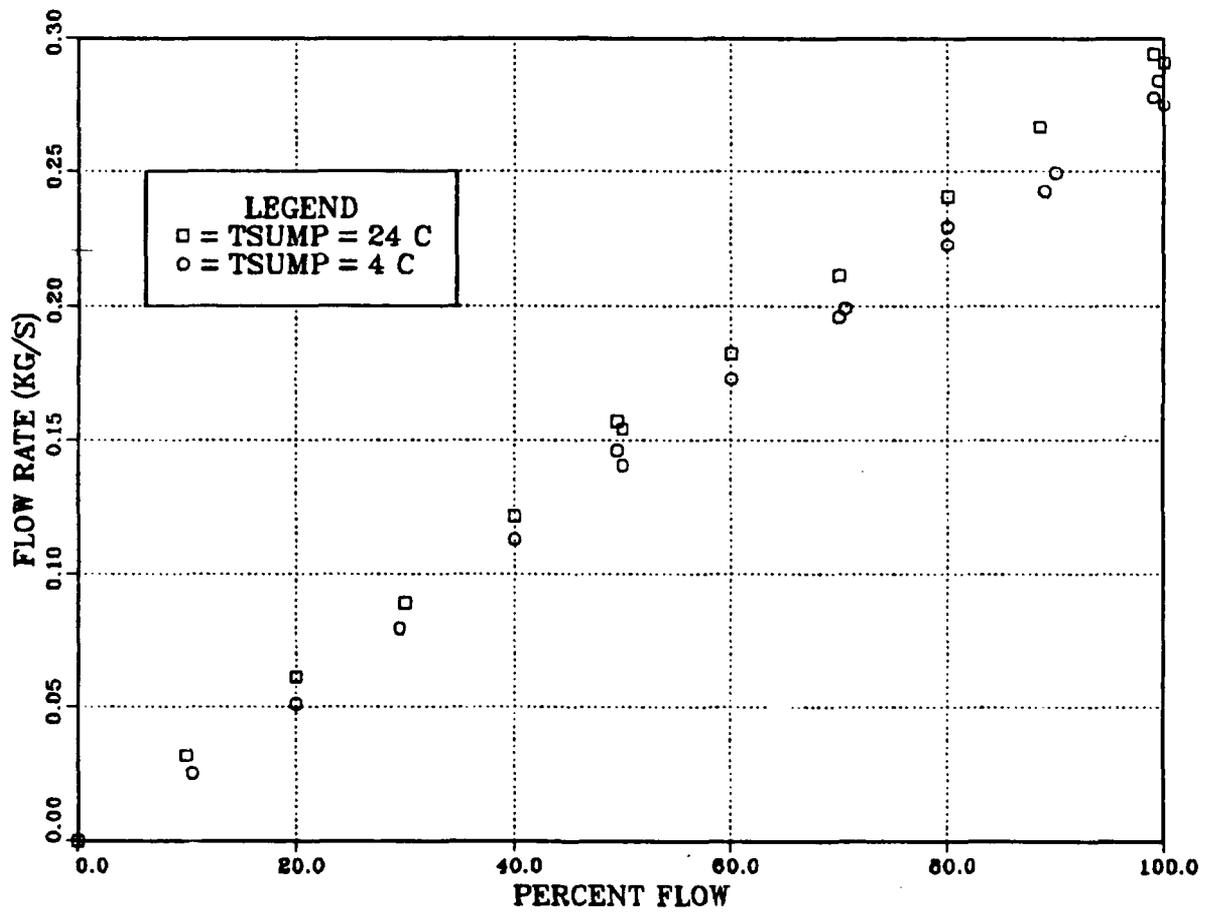
Figure 92.   Calibration for flowmeter D.

APPENDIX B. EMF VERSUS TEMPERATURE

The EMF read from the thermocouples must be converted into a temperature for prupoes of calculation. This is accomplished using a 7th order polynomial which has the general form:

$$T = C(0) + C(I)V^I$$

where V is the EMF reading in volts, and I has values from 1 to 7. The leading coefficients are given in Table 6.

Figure 93 shows the relationship between EMF and temperature.

Table 6.   **LEADING COEFFICIENTS FOR CONVERSION OF EMF TO TEMPERATURE.**

Leading Coefficient	Value
C(0)	0.10086
C(1)	25727.944
C(2)	-767345.8
C(3)	78025595.8
C(4)	-9247486589
C(5)	$6.977x10^{11}$
C(6)	$-2.66x10^{13}$
C(7)	$3.9x10^{14}$

EMF VERSUS TEMPERATURE
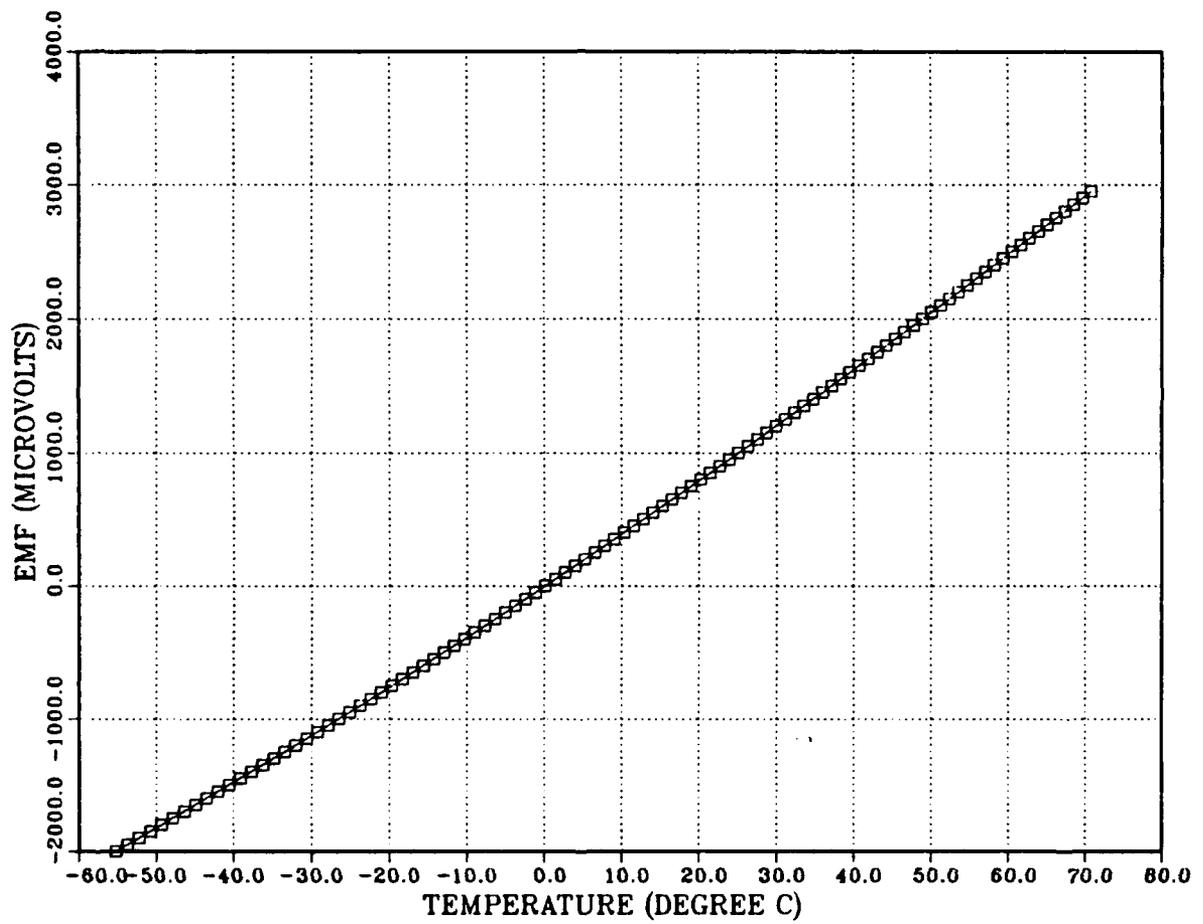
Figure 93.   EMF versus temperature.

APPENDIX C.   PROPERTIES OF ETHYLENE GLYCOL/WATER MIXTURE

The physical and thermodynamic properties of the ethylene glycol/water mixture were determined using equations and figures from Cragoe [Ref. 56] and Gallant [Ref. 57].

The density of the ethylene glycol/water mixture is calculated in lines 2490 to 2530 of the computer program. Density as a function of temperature for various concentrations of ethylene glycol are shown in Figure 94.

The kinematic viscosity of the ethylene glycol/water mixture is calculated in lines 2370 to 2420. Figure 95 shows the kinematic viscosity plotted as a function of temperature for various ethylene glycol concentrations.

The heat capacity is calculated in lines 2430 to 2480. Figure 96 is a plot of heat capacity as a function of temperature for different ethylene glycol concentrations.

The thermal conductivity is calculated in lines 2560 to 2610 of the program. Values of the thermal conductivity as a function of temperature are plotted for various ethylene glycol concentrations in Figure 97.

DENSITY
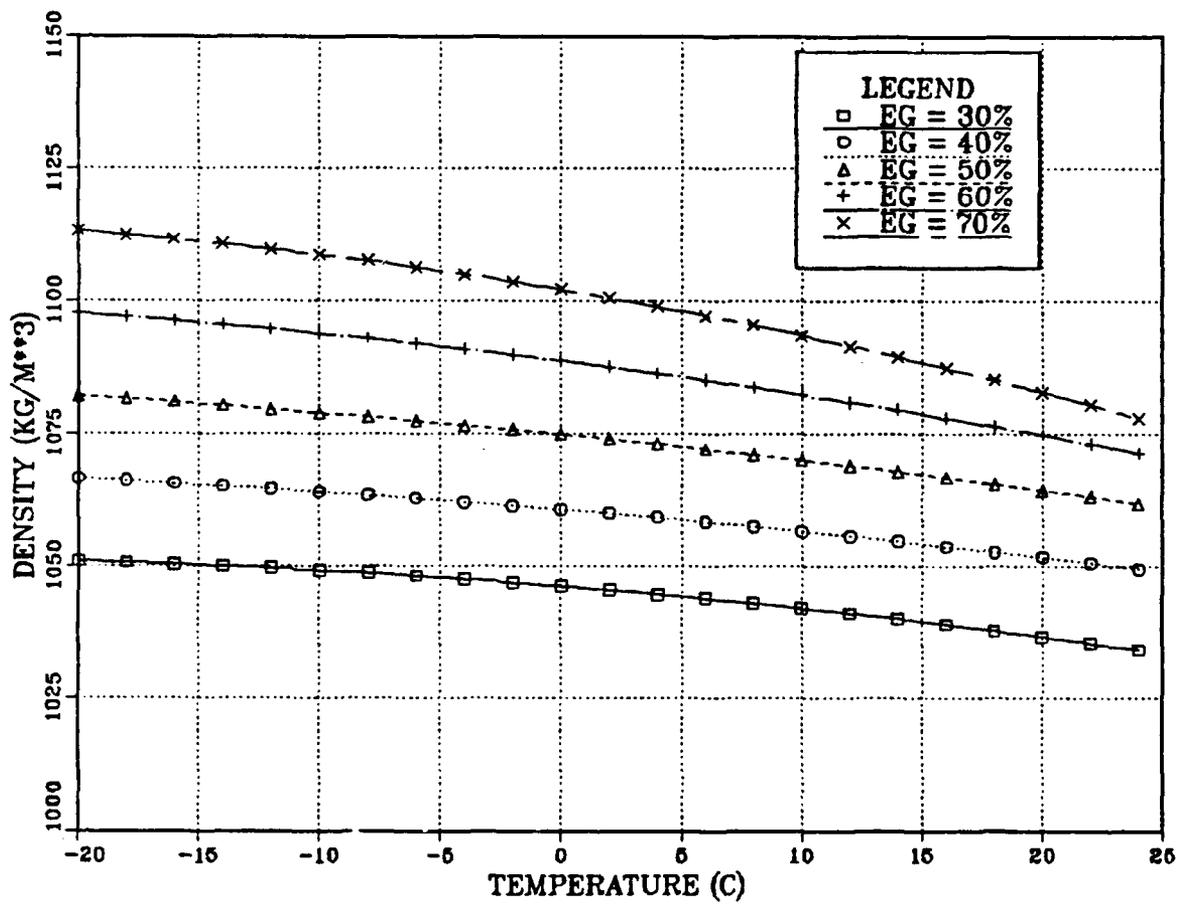
Figure 94.   Density of ethylene glycol/water mixture.

154

KINEMATIC VISCOSITY
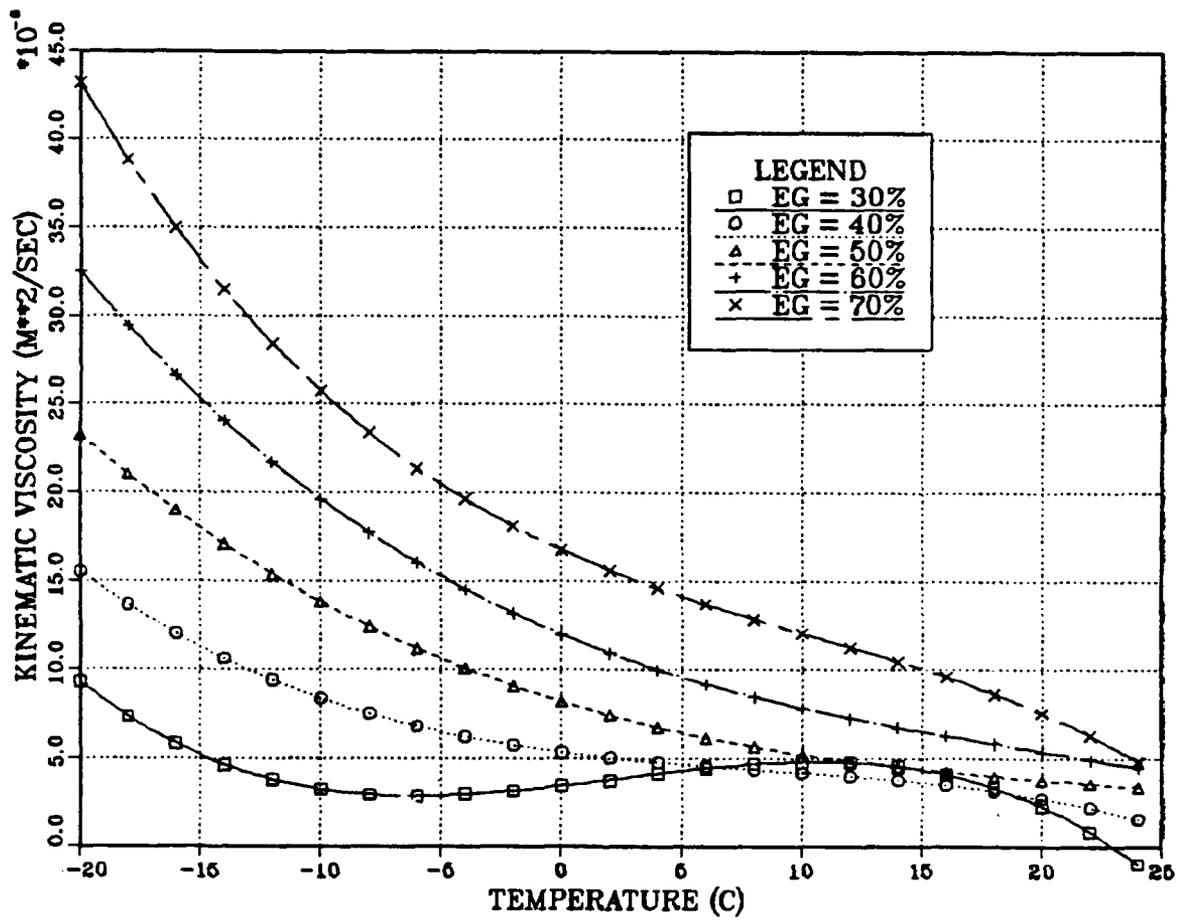
Figure 95. Kinematic viscosity of ethylene glycol/water mixture.

HEAT CAPACITY
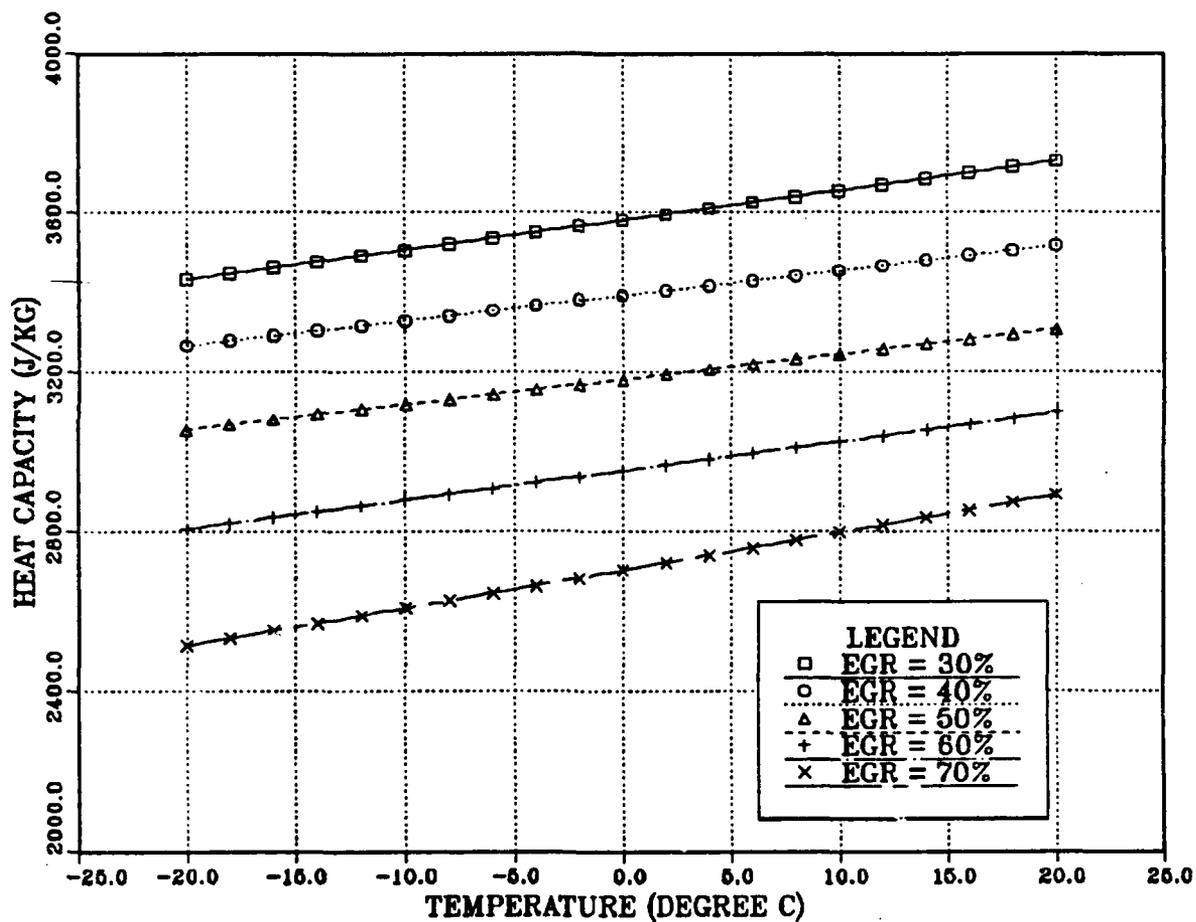
Figure 96.   Thermal heat capacity of ethylene glycol/water mixture.

156

THERMAL CONDUCTIVITY
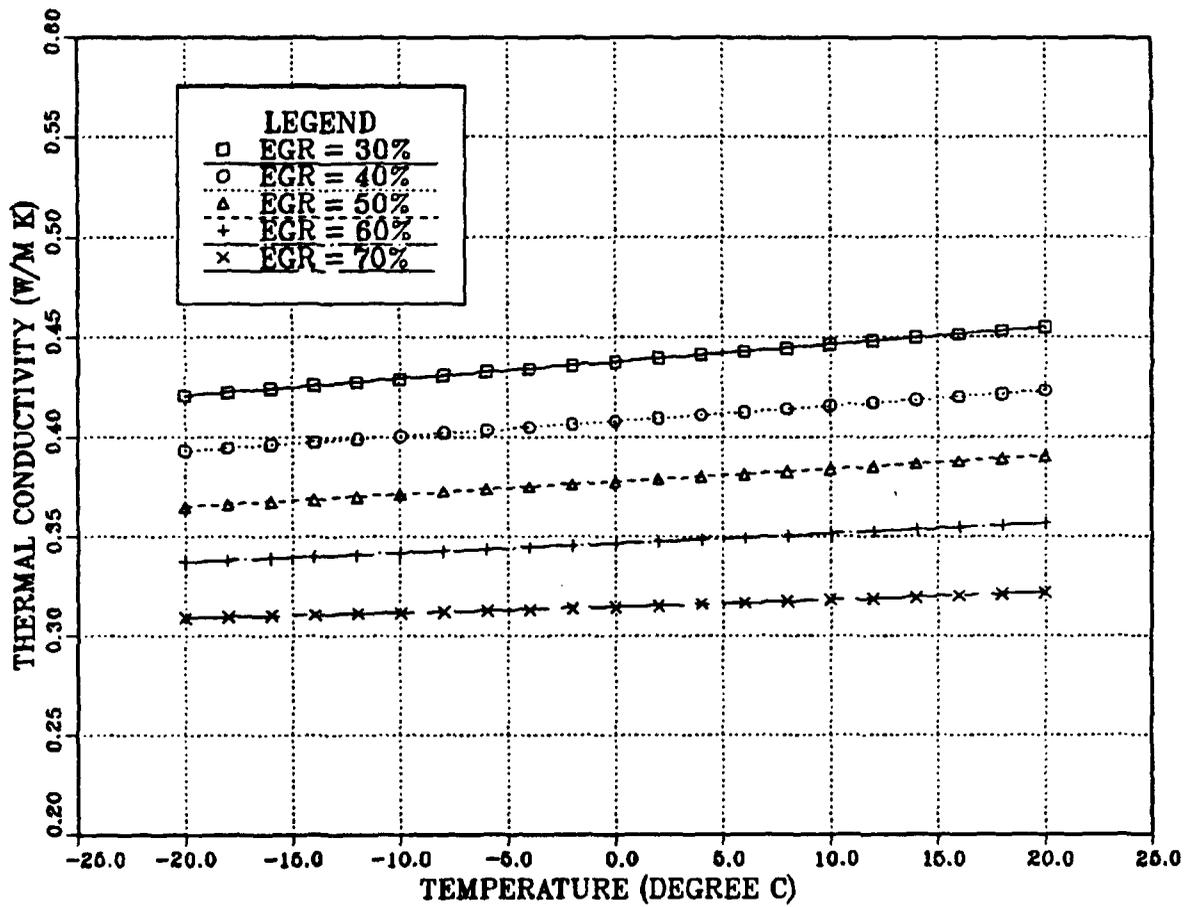
Figure 97.   Thermal conductivity of ethylene glycol/water mixture.

APPENDIX D.   PHYSICAL AND THERMODYNAMIC PROPERTIES OF R-113

The physical and thermodynamic properties of R-113 were calculated based on equations supplied by Chapman [Ref. 58].

The heat capacity, C_p, of R-113 was calculated from the equation:

$$C_p = (929.0 + 1.03T)$$

where T is in $°C$ and C_p is in kJ/kg K. Figure 98 shows C_p as a function of temperature for R-113.

The dynamic viscosity for R-113 is calculated from the equation:

$$\mu = 1.34x10^{-5} \times 10^{\frac{503}{(T-2.15)}}$$

where T is in K and μ is given in kg/m s. The relationship between μ and temperature is shown in Figure 99.

The thermal conductivity of R-113 can be calculated from:

$$k = 1.7308(4.846x10^{-2} - T \times 6.57x10^{-5})$$

where T is in $°F$ and K is in W/m K. Figure 100 shows the thermal conductivity as a function of temperature for R-113.

The density of R-113 can be calculated from:

$$\rho = 1.62x10^3 - T \times (2.219 + T \times 2.358x10^{-3})$$

where T is in $°C$ and ρ is in kg/m^3. Density of R-113 is shown as a function of temperature in Figure 101.

The latent heat of R-113 can be calculated from the equation:

$$h_{fg} = 2326.1(70.55 - t \times (0.04838 + 1.26x10^{-4}T))$$

where T is in $°F$ and h_{fg} is in J/kg. Figure 102 shows the relationship between latent heat and temperature for R-113.

HEAT CAPACITY OF R–113
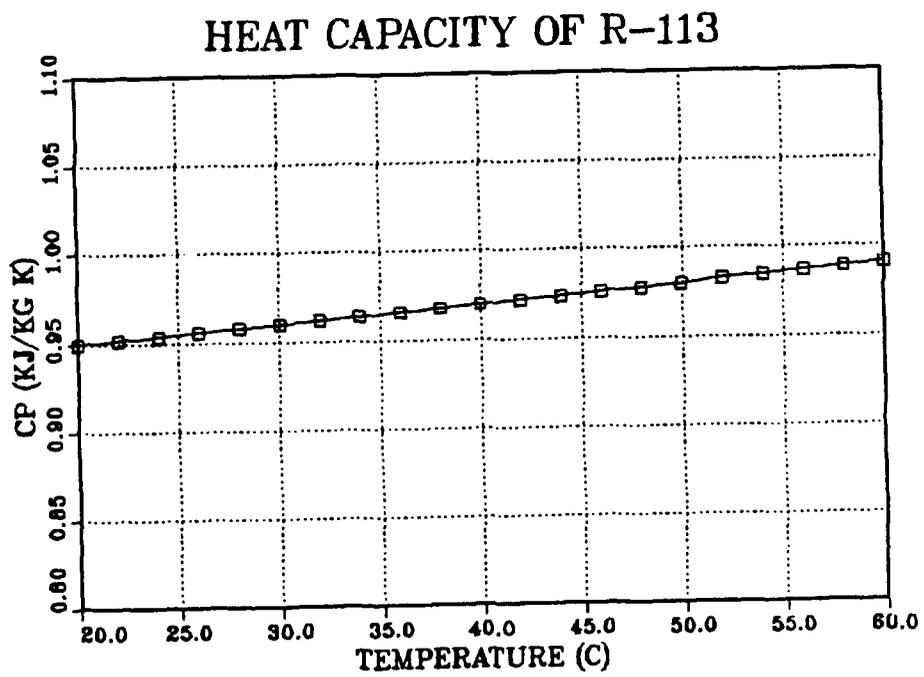
Figure 98.   Heat capacity versus temperature for R-113.
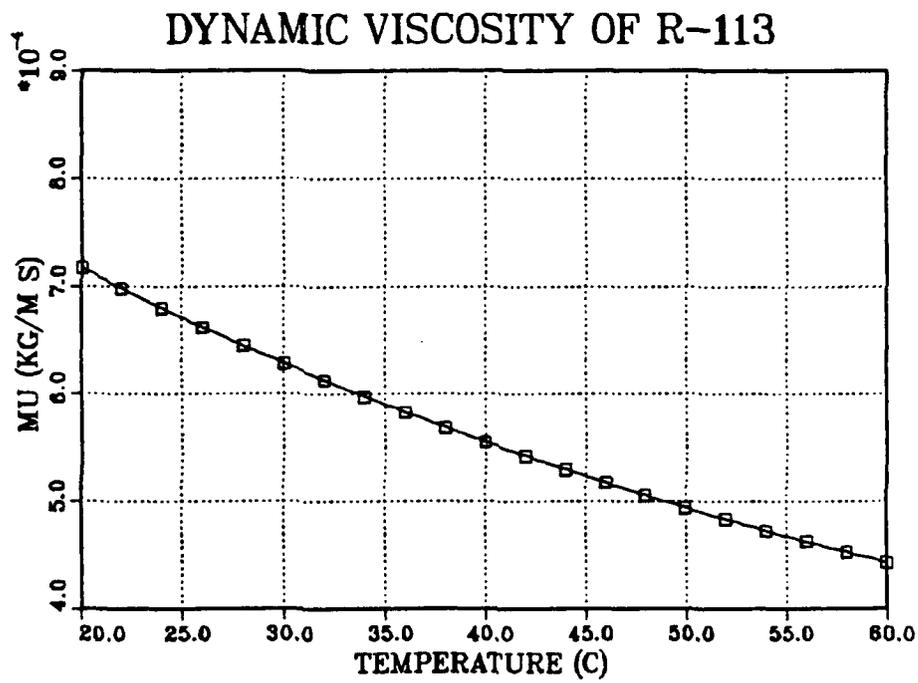
Figure 99. Viscosity as a function of temperature for R-113.

THERMAL CONDUCTIVITY OF R-113
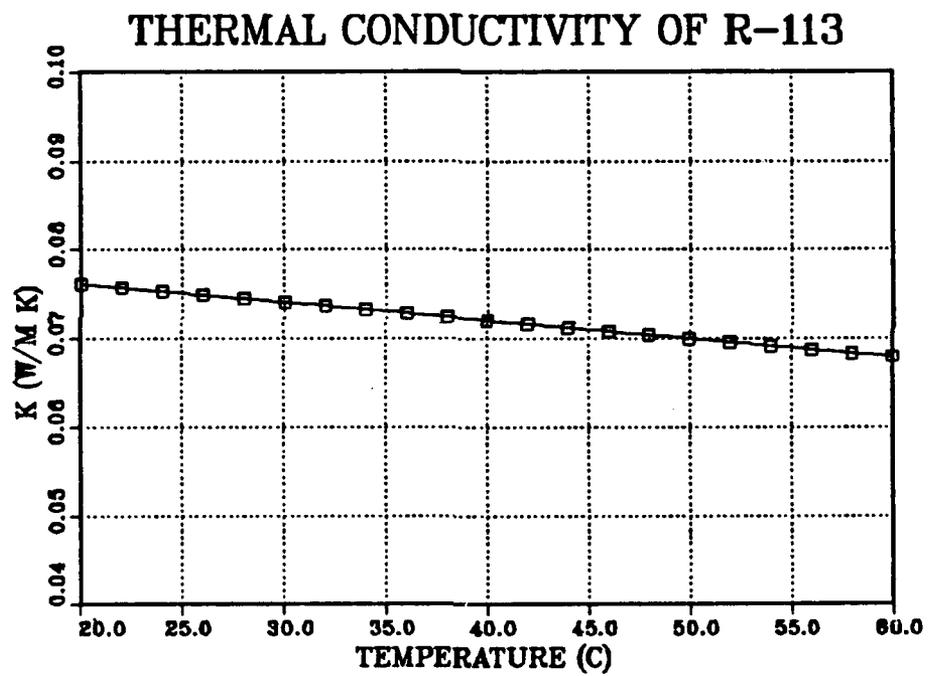
Figure 100. Thermal Conductivity of R-113 as function of temperature.
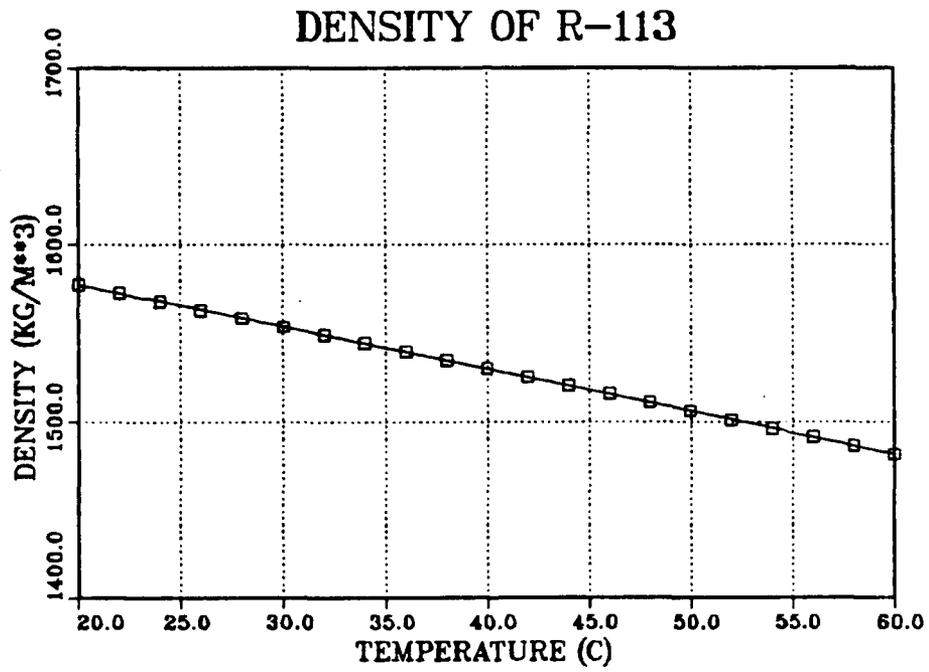
Figure 101.   Density as a function of temperature for R-113.

162
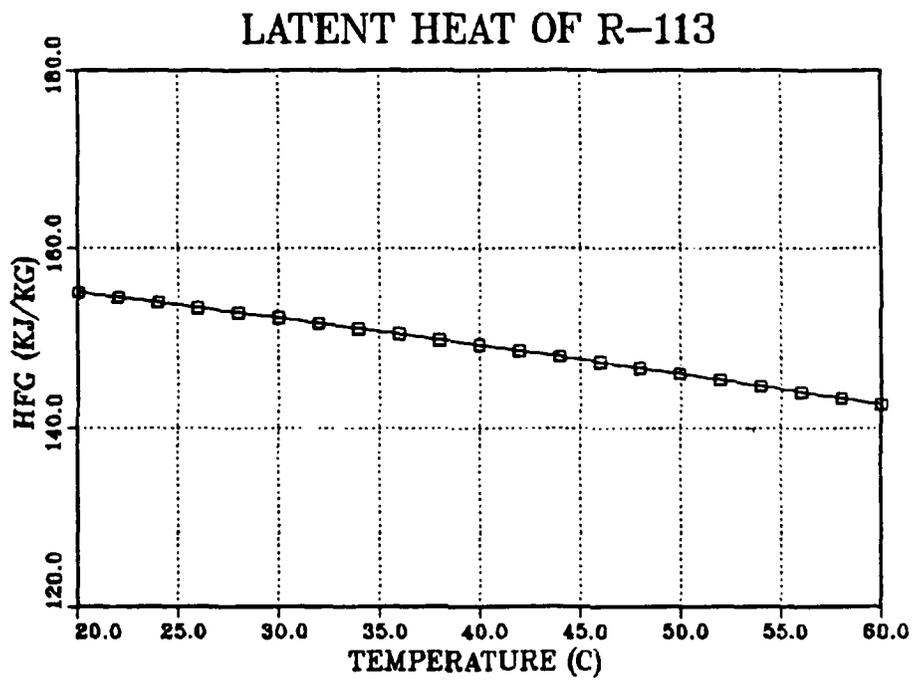
LATENT HEAT OF R−113

Figure 102. Latent heat versus temperature for R-113.

APPENDIX E.   SAMPLE CALCULATION

A sample calculation was conducted on run 1 of data set SM06. This run was carried out on smooth copper tubes with the HEATEX insert installed. The sample calculation was performed for the top tube only.

Temperatures were recorded as emf's using copper/constantan thermocouples. EMF values were converted to temperatures using the equation shown in Appendix B. Table 7 summarizes the results.

Table 7.   EMF AND TEMPERATURES FOR RUN 1 DATA SET SM06.

Thermocouple	EMF (microvolts)	Temperature (C)
T(0)	1908	46.83
T(1)	1895	46.54
T(2)	1909	46.86
T(3)	1922	47.16
T(4)	1918	47.06
T(5)	1919	47.099
T(6)	367	9.44
T(7)	369	9.496
T(8)	370	9.52
T(9)	377	9.7
T(10)	893	22.52
T(11)	893	22.53
T(12)	831	20.99
T(13)	831	20.99
T(14)	798	20.18
T(15)	798	20.18
T(16)	853	21.54
T(17)	853	21.54

For tube A, the temperature differential is calculated by:

$$\Delta T = \frac{T(10) + T(11)}{2} - T(6) = 13.086 \ C$$

The average temperature is then calculated from:

$$T_{avg} = T(6) + \Delta T \times 0.5 = 15.98 \ C$$

Based on the ethylene glycol/water mixture physical properties (Appendix C), and using and average temperature of 15.98 C and an ethylene glycol concentration of 54%, the following fluid properties are obtained:

$$\rho_c = 1071.3 \ kg/m^3$$
$$\mu_c = 0.0051 \ kg/m \ s$$
$$C_{p,c} = 3196.99 \ J/kg \ K$$
$$k_c = 0.3742 \ W/m \ k$$
$$Pr_c = 43.57$$

Based on the calibration curves for flowmeter A, the mass flowrate of coolant corresponding to a setting of 15.74 can be calculated. A simplified interpolation procedure is used to account for temperature differences. Based on this calculation, we obtain a mass flow rate of 0.2979 kg/s for this run.

The coolant velocity can be calculated from:

$$V_c = \frac{\dot{m}}{\rho_c A_i}$$

where the inside cross-sectional area is given by:

$$A_i = \frac{\pi}{4} d_i^2$$

$$A_i = \frac{\pi}{4} (0.013259)^2 = 0.000138 \ m^2$$

and

$$V_c = \frac{0.02979}{(1071.3)(0.000138)} = 0.201 \ m/s$$

The coolant Reynolds number, Re_c, is calculated from:

$$Re_c = \frac{V_c d_i}{\nu_c}$$

$$Re_c = \frac{(0.201)(0.013259)}{4.76x10^{-6}} = 559.89$$

The heat transferred to the coolant is calculated from:

$$\dot{q} = \dot{m}C_{p,c}\Delta T = (0.02979)(3196.99)(13.085)$$

$$\dot{q} = 1246.19 \ W$$

and the heat flux is then calculated from :

$$q''_o = \frac{\dot{q}}{A_o}$$

where

$$A_o = \pi d_o L = \pi(0.015875)(1.2192) = 0.0608 \ m^2$$

where L is the condensing length of the tube. Based on this equation, we obtain:

$$q''_o = \frac{1246.19}{0.0608} = 20494.9 \ W/m^2$$

In order to calculate the inside and outside heat transfer coefficients, the physical and thermodynamic properties of R-113 must be calculated (Appendix D). These fluid properties are based on a reference or condensate film temperature. In order to obtain this film temperature, the wall temperature is needed. To obtain this, an iteration is carried out with an initial value of h_o assumed (in this case 1000 $W/m^2 K$). The wall temperature is then calculated from:

$$T_{wall} = T_{sat} - \frac{q''_o}{h_o}$$

$$T_{wall} = 46.925 - \frac{20494.9}{1000} = 26.43°C$$

The film temperature is then calculated from:

$$T_f = \frac{T_{sat}}{3} + \frac{2}{3}\, T_{wall}$$

or

$$T_f = \frac{46.925}{3} + \frac{2}{3}\,(26.43) = 33.26°C$$

Based on the equations of fluid properties for R-113 (Appendix D), the following values were obtained for the condensate:

$$h_{fg} = 147036.5\ J/kg$$
$$k = 0.0734\ W/m\ K$$
$$\rho = 1544.34\ kg/m^3$$
$$\mu = 6.029x10^{-4}\ kg/m\ s$$

Using these fluid properties together with equation (5-15), we obtain:

$$h_o = 1907.06\ W/m^2 K$$

Using an iterative procedure, we adjust the initial guess of h_o by taking the average of the calculated and guessed value of h_o. The wall and film temperatures are then recalculated and new fluid properties for R-113 are obtained. A new value of h_o is then calculated. This procedure continues until the newly calculated and previosuly calculated values are within 0.1%. For this run, we obtained:

$$h_o = 1251\ W/m^2\ K$$

The next step is to find a value for the overall heat transfer coefficient, U_o, where:

$$U_o = \frac{q''_o}{LMTD}$$

where, for this run,

$$LMTD = 30.48°C$$

and

$$U_o = \frac{20494.9}{30.48} = 672.5 \ W/m^2 \ K$$

The inside heat transfer coefficient, h_i, can be obtained from a correlation for the Nusselt number for single phase coolant flow in a tube. This Nusselt number is obtained from a correlation (in this case) for the HEATEX element where:

$$Nu_c = 0.226(Re_c)^{0.65}(Pr_c)^{0.46}$$

$$Nu_c = 0.226(559.89)^{.65}(43.75)^{.46} = 78.41$$

and

$$h_i = \frac{Nu_c k_c}{d_i} = \frac{(78.41)(0.3742)}{0.013259}$$

$$h_i = 2213.07 \ W/m^2 \ K$$

The outside heat transfer coefficient is calculated from equation (5-12) where:

$$h_o = \frac{1}{\dfrac{1}{U_o} - \dfrac{d_o}{d_i h_i} - R_m A_o}$$

The wall resistance, R_m, is defined as:

$$R_m = \frac{\ln\left(\dfrac{d_o}{d_i}\right)}{2Lk}$$

$$R_m = \frac{\ln\left(\dfrac{0.015875}{0.013259}\right)}{2(1.2192)(386)} = 1.91x10^{-4} \ K/W$$

Therefore,

$$h_o = \frac{1}{\dfrac{1}{672.49} - \dfrac{0.015875}{(0.013259)(2213.07)} - (1.91x10^{-4})(0.0608)}$$

$$h_o = 1070.2 \ W/m^2 \ K$$

Figure 103 is a copy of the computer printout for this run. The small differences seen in the computer calculations versus these sample calculations are the result of roundoff error.

Data set number = 1

RESULTS FOR TUBE NO. 1

Mass flow rate   = 2.980E-02
Inlet temperature   = 9.443E+00
Saturation temp (Deg C) = 4.693E+01
DELT temp Dif.   = 1.307E+01
Log. Mean temp Dif.   = 3.048E+01
Heat flux   = 2.048E+04
Resistance of Tube Metal = 1.913E-04
Prandtl number   = 4.359E+01
Reynolds number   = 5.608E+02
Reynolds number H&B S   = 6.745E+02
Inside h.t.c.   = 2.216E+03
Inside NUSSULT NO.   = 7.852E+01
OVERALL H.t.c. (Uo)   = 6.720E+02

TUBE	FM	VEG	DELT	Uo	HO	HNUS
#	(%)	(m/s)	(K)	(W/m^2.K)		
1	15.74	0.20	13.074	6.720E+02	1.068E+03	1.251E+03

Figure 103.   Computer printout of results for run 1 data set SM06.

APPENDIX F.   UNCERTAINTY ANALYSIS FOR U_O

Run number 1 of data set SM06 was also chosen for the uncertainty analysis. The analysis was only conducted on U_o. The measured and calculated parameters found in the sample calculation (Appendix E) were therefore used in this section. In order to assess the effects of coolant velocity on the uncertainty in the measurement of U_o , an uncertainty analysis was also performed on run number 11 of data set SM06. This run had a coolant velocity of 1.2 m/s as opposed to 0.2 m/s in run number 1.

The uncertainty analysis utilizes the procedure suggested by Kline and McClintock [Ref. 54]. This procedure states that if

$$R = R(x_1, x_2, x_3, \ldots\ldots, x_n)$$

then the uncertainty in R, namely, δR is given by the equation:

$$\delta R = \left[\left(\frac{\partial R}{\partial x_1} \delta x_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} \delta x_2 \right)^2 + \ldots\ldots + \left(\frac{\partial R}{\partial x_n} \delta x_n \right)^2 \right]^{0.5}$$

where x_n is the measured variable and δx_n is the uncertainty in the measured variable.

The uncertainty in the temperature measurements is due to uncertainty in the voltage measured by the thermocouples. For the thermocouples used corresponds to 0.1 °C. The uncertainty in in this thesis, this value was assumed to be 4 microvolts or 0.1 C. The uncertainty in ΔT, T_{sat} and coolant inlet and outlet temperatures, T_{ci} and T_{co} , can be calculated as follows:

$$T_{co} = \frac{T(10) + T(11)}{2}$$

Taking the partial derivatives we obtain:

$$\frac{\partial T_{co}}{\partial T(10)} = \frac{1}{2}$$

likewise,

$$\frac{\partial T_{co}}{\partial T(11)} = \frac{1}{2}$$

The uncertainty in the measurement of T_{co} is thus:

$$\delta T_{co} = \left[\left(\frac{\partial T_{co}}{\partial T(10)} \, \delta T \right)^2 + \left(\frac{\partial T_{co}}{\partial T(11)} \, \delta T \right)^2 \right]^{0.5}$$

Based on the thermocouple uncertainty, $\delta T = 0.1$ C, we obtain:

$$\delta T_{co} = 0.0778 \; C$$

The coolant inlet temperature is calculated from a single thermocouple, number 6 for tube A. The uncertainty in this measurement is calculated as:

$$\frac{\partial T_{ci}}{\partial T(6)} = 1$$

and,

$$\delta T_{ci} = \left[\left(\frac{\partial T_{ci}}{\partial T(6)} \, \delta T \right)^2 \right]^{0.5}$$

or,

$$\delta T_{ci} = 0.1 \; C$$

Finally, the uncertainty in ΔT is calculated from:

$$\delta \Delta T = \left[(\delta T_{co})^2 + (\delta T_{ci})^2 \right]^{0.5}$$

and,

$$\delta \Delta T = 0.1347 \; C$$

Coolant velocity is calculated from:

$$V_c = \frac{\dot{m}}{\rho_c A_l}$$

In this run, $\dot{m}$ was calculated to be 0.02979 kg/s. The uncertainty in the reading of the flowmeter is defined as the scale interpolation. This value corresponds to one-half the value of the smallest marked increment. In this case, the scale interpolation factor, δ_{si},

172

is 0.5% which corresponds to 0.00101 kg/s resolution. In addition, a time-wise jitter, δ_{tj}, of 0.5% was noted during the runs. Therefore, the uncertainty in $\dot{m}$ is given by:

$$\frac{\delta \dot{m}}{\dot{m}} = \left[\left(\frac{\delta \dot{m}}{\dot{m}} \delta_{si} \right)^2 + \left(\frac{\delta \dot{m}}{\dot{m}} \delta_{tj} \right)^2 \right]^{\frac{1}{2}}$$

or

$$\frac{\delta \dot{m}}{\dot{m}} = 0.0479$$

The uncertainty in the cross-sectional area of the tubes was estimated from:

$$\frac{\delta A_i}{A_i} = \left[2 \times \left(\frac{\delta d_i}{d_i} \right)^2 \right]^2$$

where δd_i is given as 0.1 mm based on tolerances supplied from the manufacturer. Therefore,

$$\frac{\delta A_i}{A_i} = 0.0107 = 1.07 \%$$

The uncertainty in the coolant velocity is now given by:

$$\frac{\delta V_c}{V_c} = \left[\left(\frac{\delta \dot{m}}{\dot{m}} \right)^2 + \left(\frac{\delta A_i}{A_i} \right)^2 \right]^{0.5}$$

or

$$\frac{\delta V_c}{V_c} = 0.049 = 4.9 \%$$

173

The uncertainty in the calculation of coolant Reynolds number is given by:

$$\frac{\delta Re_c}{Re_c} = \left[\left(\frac{\delta V_c}{V_c} \right)^2 + \left(\frac{\delta d_i}{d_i} \right)^2 \right]^{0.5}$$

or substituting appropriate numbers we find

$$\frac{\delta Re_c}{Re_c} = 0.0497 = 4.97 \%$$

The uncertainty in the calculation of the heat transferred to the coolant is related to the uncertainty in the measurements of coolant mass flow rate and coolant temperature rise where:

$$\frac{\delta \dot{q}}{\dot{q}} = \left[\left(\frac{\delta \dot{m}}{\dot{m}} \right)^2 + \left(\frac{\delta \Delta T}{\Delta T} \right)^2 \right]^{0.5}$$

or ___

$$\frac{\delta \dot{q}}{\dot{q}} = 0.0489 = 4.89 \%$$

The uncertainty in the heat flux is then:

$$\frac{\delta q''_o}{q''_o} = \left[\left(\frac{\delta \dot{q}}{\dot{q}} \right)^2 + \left(\frac{\delta A_o}{A_o} \right)^2 \right]^{0.5}$$

where the uncertainty in A_o is given by:

$$\frac{\delta A_o}{A_o} = \left[\left(\frac{\delta d_o}{d_o} \right)^2 + \left(\frac{\delta L}{L} \right)^2 \right]^{0.5}$$

Here, the uncertainty in the length of the tube, δL , is assumed to be 5 mm while δd_o is assumed to be 0.1 mm (again from given tolerances). Substitution of the appropriate numbers yields:

$$\frac{\delta q''_o}{q''_o} = 0.0496$$

174

Finally, since U_o is given by:

$$U_o = \frac{q''_o}{LMTD}$$

the uncertainty in the LMTD must be calculated. The LMTD is given by:

$$LMTD = \frac{\Delta T}{\ln\left[\dfrac{T_{sat} - T_{c,i}}{T_{sat} - T_{c,o}}\right]}$$

In order to calculate the uncertainty in the measurement of the LMTD, the following equations were used:

$$\frac{\partial LMTD}{\partial \Delta T} = \frac{1}{\ln\left[\dfrac{T_{sat} - T_{c,i}}{T_{sat} - T_{c,o}}\right]}$$

$$\frac{\partial LMTD}{\partial T_{sat}} = \frac{\Delta T[(T_{sat} - T_{c,o}) - (T_{sat} - T_{c,i})]}{\left[\ln\left[\dfrac{T_{sat} - T_{c,i}}{T_{sat} - T_{c,o}}\right]\right]^2 (T_{sat} - T_{c,i})(T_{sat} - T_{c,i})}$$

$$\frac{\partial LMTD}{\partial T_{c,o}} = -\frac{\Delta T}{\left[\ln\left[\dfrac{T_{sat} - T_{c,i}}{T_{sat} - T_{c,o}}\right]\right]^2 (T_{sat} - T_{c,o})}$$

$$\frac{\partial LMTD}{\partial T_{c,i}} = \frac{\Delta T}{\left[\ln\left[\dfrac{T_{sat} - T_{c,i}}{T_{sat} - T_{c,o}}\right]\right]^2 (T_{sat} - T_{c,i})}$$

The uncertainty in the LMTD is therefore given by:

$$\delta LMTD = \left[\left(\frac{\partial LMTD}{\partial \Delta T}\delta\Delta T\right)^2 + \left(\frac{\partial LMTD}{\partial T_{sat}}\delta T_{sat}\right)^2 + \left(\frac{\partial LMTD}{\partial T_{c,o}}\delta T_{c,o}\right)^2 + \left(\frac{\partial LMTD}{\partial T_{c,i}}\delta T_{c,i}\right)^2\right]^{\frac{1}{2}}$$

For the sample run with a coolant velocity of 0.2 m/s, the uncertainty in the measurement of the LMTD is:

$$\delta LMTD = \left[[(2.329)(0.1347)]^2 + [(-1.0157)(0.0778)]^2 + [(-2.9097)(0.0778)]^2 + [(1.894)(0.1)]^2\right]^{\frac{1}{2}}$$

or

$$\delta LMTD = 0.446\ K$$

and

$$\frac{\delta LMTD}{LMTD} = \frac{0.446}{30.48} = 0.0146 = 1.46\ \%$$

The uncertainty in the overall heat transfer coefficient, U_o, is then given by:

$$\frac{\delta U_o}{U_o} = \left[\left(\frac{\delta q''_o}{q''_o} \right)^2 + \left(\frac{\delta LMTD}{LMTD} \right)^2 \right]^{\frac{1}{2}}$$

or

$$\frac{\delta U_o}{U_o} = \left[(0.0496)^2 + (0.0146)^2 \right]^{\frac{1}{2}}$$

and

$$\frac{\delta U_o}{U_o} = 0.052 = 5.2\ \%$$

Table 7 summarizes the results of the uncertainty analysis for the smooth copper tubes with the HEATEX insert for coolant velocities of 0.2 and 1.2 m/s respectively.

The uncertainty analysis neglects any uncertainty in the physical properties of the coolant or the R-113. For the most part, this is probably a reasonable assumption. The exception to this may be the viscosity due to its sensitivity to temperature. Hence, the uncertainty reported here is an underestimate.

It is curious that the relative magnitudes of the uncertainty for the low and high coolant velocities are essentially the same. Two different mechanisms dominate in the calculated uncertainty at low and high coolant velocities. In terms of the mass flow rate, the time-wise jitter and scale interpolation terms are the same for both coolant velocities. However, the actual mass flow rate is less at the lower flow rates. Hence, the uncertainty in the mass flow rate calculation is more significant at the lower coolant velocities. At higher flow rates the uncertainty in the calculation of the LMTD becomes the dominant term. This is because the coolant temperature rise is smaller at high coolant velocities but the uncertainty in the temperature measurements remains the same. The mass flow

Table 8. SUMMARY OR RESULTS FOR SAMPLE UNCERTAINTY ANALYSIS.

Variable	$V_c = 0.2$ m/s	$V_c = 1.2$ m/s
$\delta T_{c,i}$	$0.1°C$	$0.1°C$
$\delta T_{c,o}$	$0.078°C$	$0.078°C$
δT_{sat}	$0.078°C$	$0.078°C$
$\delta \Delta T$	$0.135°C$	$0.135°C$
$\dfrac{\delta m}{m}$	0.048	0.008
$\dfrac{\delta A_i}{A_i}$	0.011	0.011
$\dfrac{\delta V_c}{V_c}$	0.049	0.013
$\dfrac{\delta Re_c}{Re_c}$	0.05	0.015
$\dfrac{\delta q}{q}$	0.05	0.015
$\dfrac{\delta A_o}{A_o}$	0.008	0.008
$\dfrac{\delta q''_o}{q''_o}$	0.05	0.017
$\dfrac{\delta LMTD}{LMTD}$	0.015	0.054
$\dfrac{\delta U_o}{U_o}$	0.052	0.057

rate goes up so that the uncertainty in the coollant mass flow rate calculation decreases. Thus, at the higher coolant velocities, the uncertainty in the calculation of the LMTD is the dominant term.

APPENDIX G.   PROGRAM LISTING

```
1000 !  FILE:    DRPCON7
1005 !  PURPOSE: This program collects and processes condensation data for the t
ube bundle.
1015 !  CREATED: NOVEMBER 2, 1988
1020 !  UPDATE1: APRIL 25 1991 (S. MEMORY)
1021 !  UPDATE2: May 17 1991 (S. MEMORY)
1022 !  UPDATE3: JULY 26, 1991 (R.W. MAZZONE)
1023 !  UPDATE4: 13 AUGUST 1991(R.W. MAZZONE)
1024 !  UPDATE5: 10 OCTOBER 1991 (R.W. MAZZONE)
1026 ! CHANGE HOD=3 AND USING CALCULATED CI FROM WILSON
1030     BEEP
1035     PRINTER IS 1
1040     PRINT USING "4X,""SELECT OPTION"""
1045     PRINT USING "6X,""0  TAKING DATA OR REPROCESING PREVIOUS DATA"""
1050     PRINT USING "6X,""1  PLOTTING H VS DELTA-T"""
1055     PRINT USING "6X,""2  PLOTTING HRAT VS N"""
1060     PRINT USING "6X,""3  PLOTTING WILSON"""
1065     PRINT USING "6X,""4  PURGE FILES"""
1070     PRINT USING "6X,""5  XYREAD"""
1075     PRINT USING "6X,""6  NUSSELT ESTIMATE"""
1080     PRINTER IS 701
1085     INPUT Icall
1090     IF Icall=0 THEN CALL Main
1095     IF Icall=1 THEN CALL Plot2
1100     IF Icall=2 THEN CALL Plot1
1105     IF Icall=3 THEN CALL Plot3
1110     IF Icall=4 THEN CALL Purge
1115     IF Icall=5 THEN CALL Xyread
1120     IF Icall=6 THEN CALL Nusselt
1125     END
1130     SUB Main
1135     COM /Cc/ C(7)
1140     COM /Fld/ Ift
1145     COM /Nus/ Iin,Tsat,Qdp1,Hnus,Kf,Rhof,Hfg,Muf,Do,Itube
1150     COM /Wil1/ Doa(4),Dia(4),Kma(4),Iact,Droot(4)
1155     COM /Wil2/ Delta,Isat,Nsets,Hod,Cia(3,3),Alpaa(3)
1160     DIM Emf(17),Tp(4),T(17),Ho(4),Qdp(4),Uo(4),Pc(4)
1215     DATA 1.0,1.0,1.0,1.0
1216     DATA 5.172,5.172,5.172,5.172
1217     DATA 0.226,0.226,0.226,0.226
1218     DATA 0.063,0.063,0.063,0.063
1220     READ Cia(*)
1225     DATA 0.10086091,25727.94369,-767345.8295,78025595.81
1230     DATA -9247486589,6.97688E11,-2.66192E13,3.94078E14
1235     READ C(*)
1240     DATA 0.015875,0.014000,0.015798,0.0159,0.0
1245     DATA 0.013259,0.010160,0.012776,0.01346,0.0
```

```
1250      DATA 386.0,42.975,42.975,21.9,0.0
1251      DATA 0.0,0.014,0.0,.0141,0.0
1255      DATA 0.0005588,3
1260      READ Doa(*),Dia(*),Kma(*),Droot(*),Delta,Hod   ! Hod=H/Di
1265      L=1.2192   ! Condensing length
1270      Jset=0
1271      Okaccpt=1
1275      BEEP
1280      INPUT "ENTER MONTH, DATE AND TIME (MM:DD:HH:MM:SS)",Dtg$
1285      OUTPUT 709;"TD";Dtg$
1290      BEEP
1295      INPUT "SELECT OPTION (0=DAQ, 1=FILE)",Im
1300      Ihard=1
1305      BEEP
1310      INPUT "WANT A HARDCOPY PRINTOUT (1=DEF=YES,0=N)",Ihard
1315      BEEP
1320      INPUT "SELECT (0=R-114,1=STEAM,2=R-113,3=EG)",Ift
1325      Iin=1
1330      Isat=2
1335      BEEP
1340      INPUT "SELECT SAT TEMP MODE (0=LIQ,1=VAP,2=(LIQ+VAP)/2=DEF)",Isat
1345      IF Ihard=1 THEN PRINTER IS 701
1350     ─IF Im=0 THEN
1355         BEEP
1360         INPUT "GIVE A NAME FOR THE NEW DATA FILE",File$
1365         CREATE BDAT File$,20
1370         ASSIGN @File TO File$
1375         BEEP
1380         INPUT "ENTER TUBE CODE (0=SMOOTH CU,1=FINNED CU,2=KORODENSE,3=TIT FIN
          )",Itube
1381         BEEP
1382         INPUT "0=NO INSERT, 1=TWISTED TAPE, 2=HITRAN1, 3=HITRAN2",Insert
1385         BEEP
1390         INPUT "ENTER EG CONCENTRATION (WT PERCENT)",Egrat
1395         ENTER 709;Dtg$
1400         OUTPUT @File;Dtg$
1405         OUTPUT @File;Itube,Egrat,Dd1,Dd2,Dd3,Dd4,Dd5
1410         Iact=10
1415         BEEP
1420         INPUT "SELECT (0=TOP,1=SECOND,.....,10=BUNDLE=DEFAULT)",Iact
1425         PRINT
1430         PRINT USING "10X,""FILE NAME:    "",12A";File$
1435         PRINT
1440      ELSE
1445         BEEP
1450         INPUT "ENTER NAME OF EXISTING FILE",File$
1455         BEEP
1460         INPUT "ENTER NUMBER OF DATA SETS STORED",Nsets
1465         Iwil=1
1470         BEEP
```

```
1475        INPUT "WANT TO CALL WILSON? (1=DEFAULT=YES,0=NO)",Iwil
1480        IF Iwil=1 THEN
1485           BEEP
1490           INPUT "WHICH TUBE (0=TOP,1=SECOND,...,10=BUNDLE)",Iact
1491           INPUT "0=NO INSERT, 1=TWISTED TAPE, 2=HITRAN1, 3=HITRAN2",Insert
1493           CALL Wilson
1494           IF Iact=10 THEN
1495           FOR Icount=0 TO 3
1496           Cia(Insert,Icount)=Cia(Insert,0)
1497           Alpaa(Icount)=Alpaa(0)
1498           NEXT Icount
1499           END IF
1500           IF Ihard=1 THEN PRINTER IS 701
1505           PRINT
1510           PRINT USING "10X,""FILE NAME:    "",12A";File$
1515           PRINT
1520           IF Iact>9 THEN
1521           FOR I1=0 TO 3
1522           PRINT USING "10X,""RESULTS FOR TUBE NO."",3D";I1+1
1523           PRINT
1525              PRINT USING "10X,""INSIDE COEFFICIENTS: "",DD.3D";Cia(Insert,I1
)
1530              PRINT USING "10X,""ALPHAS:                  "",4(1X,MZ.3DE,2X)";Alp
aa(I1)
1531              NEXT I1
1535           ELSE
1540              PRINT USING "10X,""INSIDE COEFFICIENT FOR TUBE "",D,"": "",DD.3
D";Iact+1,Cia(Insert,Iact)
1545              PRINT USING "10X,""ALPHA FOR TUBE "",D,"":              "",DD.3
D";Iact+1,Alpaa(Iact)
1550              END IF
1555           ELSE
1560              BEEP
1561              INPUT "0=TOP, 1=SECOND, .....10=BUNDLE",Iact
1562              INPUT "0=SMOOTH CU, 1=FIN CU, 2=KOROD, 3=FIN TITANIUM",Itube
1563              INPUT "0=NO INSERT, 1=TWISTED TAPE, 2=HITRAN1, 3=HITRAN2",Insert
1565              INPUT "ENTER CI VALUE ",Cia(Insert,I1)
1570           END IF
1575           ASSIGN @File TO File$
1580           ENTER @File;Dtg$
1585           ENTER @File;Itube,Egrat,Dd1,Dd2,Dd3,Dd4,Dd5
1586        INPUT "0=SMOOTH CU, 1=CUNI FIN, 2=KORODENSE, 3=FIN TITAN ",Itube
1590        END IF
1595        Iout=1
1600        BEEP
1605        INPUT "WANT TO CREATE AN OUTPUT FILE? (1=DEF=YES,0=NO)",Iout
1610        IF Iout=1 THEN
1615           BEEP
1620           INPUT "ENTER A NAME FOR OUTPUT FILE",Fout$
1625           CREATE BDAT Fout$,5
```

```
1630          ASSIGN @Fout TO Fout$
1635      END IF
1640  !
1645      Do=Doa(Itube)
1646      Do1=Doa(Itube)
1650      Di=Dia(Itube)
1655      Km=Kma(Itube)
1660      Ax=PI*Di^2/4   ! Cross-sectional area
1665      Ao=PI*Do*L
1670  !   Rm=Do*LOG(Do,Di)/(2*Km)
1671          IF Itube=1 THEN
1672              Do1=Droot(Itube)
1673          END IF
1674          IF Itube=3 THEN
1675              Do1=Droot(Itube)
1676          END IF
1677          Ao=PI*Do*L
1679      Rm=LOG(Do1/Di)/(2*L*Km)
1680      A1=PI*Do1*L
1682  !
1683      IF Im=0 THEN
1685          PRINTER IS 1
1690          BEEP
1691          INPUT "WHAT IS REQUIRED NOMINAL COOLANT VELOCITY IN m/s (MAX. = 1.4 m
/s)?",Vegst
1694          OUTPUT 709;"AR AF6 AL9 VR5"
1695          FOR I=6 TO 9
1696              OUTPUT 709;"AS SA"
1697              Vsum=0
1698              FOR J=1 TO 5
1699                  ENTER 709;E
1700                  Vsum=Vsum+E
1701              NEXT J
1702              Emf(I)=Vsum/5
1703              T(I)=FNTvsv(Emf(I))
1704          NEXT I
1705          Tavg1=(T(6)+T(7)+T(8)+T(9))/4
1706          IF Tavg1<0 OR Tavg1>24 THEN
1707              PRINT "COOLANT TEMP. NOT IN RANGE 0 TO 24 C"
1708              GOTO 1691
1709          END IF
1711          Rhoeg1=FNRhoeg(Tavg1,Egrat)
1712          Mdot1=Rhoeg1*Ax*Vegst
1713          Pca0=(Mdot1+.00414)/.00206
1714          Pca24=(Mdot1+.00329)/.00225
1715          Pcb0=(Mdot1+.00615)/.00204
1716          Pcb24=(Mdot1+.00138)/.00216
1717          Pcc0=(Mdot1+.0036)/.00208
1718          Pcc24=(Mdot1+.00301)/.00223
1719          Pcd0=(Mdot1+.00122)/.0028
```

```
1720        Pcd24=(Mdot1-.0048)/.0029
1721        Pc(0)=Pca0+(Tavg1*(Pca24-Pca0))/24
1722        Pc(1)=Pcb0+(Tavg1*(Pcb24-Pcb0))/24
1723        Pc(2)=Pcc0+(Tavg1*(Pcc24-Pcc0))/24
1724        Pc(3)=Pcd0+(Tavg1*(Pcd24-Pcd0))/24
1725        PRINT "SET FLOWMETER READINGS CORRESPONDING TO:"
1726        PRINT USING "6X,""% OF METER A = "",DDD.D";Pc(0)
1727        PRINT USING "6X,""% OF METER B = "",DDD.D";Pc(1)
1728        PRINT USING "6X,""% OF METER C = "",DDD.D";Pc(2)
1729        PRINT USING "6X,""% OF METER D = "",DDD.D";Pc(3)
1730        PRINT
1731        PRINT "HIT CONTINUE WHEN READY"
1733        PAUSE
1734 Repeat:     !
1735 !      PRINTER IS 701
1736        Dtld=47.5    ! Desired temperature of liquid
1737        Ido=1
1738        ON KEY 0,15 RECOVER 1734
1739        PRINT USING "4X,""SELECT OPTION """
1740        PRINT USING "6X,""0=TAKE DATA """
1741        PRINT USING "6X,""1=SET Tsat (DEFAULT) """
1742        PRINT USING "4X,""NOTE: KEY 0 = ESCAPE"""
1743 !      Ido = desired option
1744        BEEP
1745        INPUT Ido
1746 !      Set default value for input
1747        IF Ido>1 THEN Ido=1
1748 !      Take data option
1749        IF Ido=0 THEN 1797
1750 !      Loop to check saturation temperature
1751        IF Ido=1 THEN
1752           INPUT "ENTER DESIRED Tsat (DEFAULT=47.5 C - R-113)",Dtld
1753 !         INPUT "ENTER DESIRED Tsat (DEFAULT=2.2 C - R-114)",Dtld
1754           Nn=1
1755           Nrs=Nn MOD 15
1756           Nn=Nn+1
1757           IF Nrs=1 THEN
1758           PRINT USING "4X,"" DTsat     Tld1     Tvab     Tvat     Del1     D
el2     Del3     Del4     """
1759           END IF
1760 !         Read thermocouple voltages
1761           OUTPUT 709;"AR AF0 AL17 VR5"
1762           Nend=17
1763           FOR I=0 TO 5
1764              OUTPUT 709;"AS SA"
1765              Vsum=0
1766              FOR J=1 TO 5
1767                 ENTER 709;E
1768                 Vsum=Vsum+E
1769              NEXT J
```

183

```
1770              Emf(I)=Vsum/5
1771              T(I)=FNTvsv(Emf(I))
1772          NEXT I
1773        FOR I=6 TO Nend
1774              OUTPUT 709;"AS SA"
1775              Vsum=0
1776              FOR J=1 TO 20
1777                  ENTER 709;E
1778                  Vsum=Vsum+E
1779              NEXT J
1780              Emf(I)=Vsum/20
1781              T(I)=FNTvsv(Emf(I))
1782        NEXT I
1783 !      Compute average temperature of liquid
1784        Tliq=(T(3)+T(4))/2
1785        Tvap=(T(0)+T(1)+T(2))/3
1786        Tvat=(T(0)+T(1))/2
1787        Tliq1=T(5)
1788        Del1=((T(10)+T(11))/2)-T(6)
1789        Del2=((T(12)+T(13))/2)-T(7)
1790        Del3=((T(14)+T(15))/2)-T(8)
1791        Del4=((T(16)+T(17))/2)-T(9)
1792        PRINT USING "4X,8(MDD.DD,3X)";Dt1d,T(3),T(2),Tvat,Del1,Del2,Del3,De
14
1793        WAIT 2
1794        GOTO 1755
1795      END IF
1796 !    TAKE DATA IF Im=0 LOOP
1797      PRINTER IS 701
1798      OUTPUT 709;"AR AF0 AL17 VR5"
1799      Nend=17    ! INCREASE TO 20 IF FIVE TUBES IN BUNDLE
1800      FOR I=0 TO 5
1801          OUTPUT 709;"AS SA"
1802          Vsum=0
1803          FOR J=1 TO 5
1804              ENTER 709;E
1805              Vsum=Vsum+E
1806          NEXT J
1807          Emf(I)=Vsum/5
1808      NEXT I
1809      FOR I=6 TO Nend
1810          OUTPUT 709;"AS SA"
1811          Vsum=0
1812          FOR J=1 TO 20
1813              ENTER 709;E
1814              Vsum=Vsum+E
1815          NEXT J
1816          Emf(I)=Vsum/20
1820      NEXT I
1825    ELSE
```

```
1830         ENTER @File;Pc(*),Emf(*),Tp(*)
1835     END IF
1840 !
1845 !   DATA ANALYSIS
1850 !
1855     Nend=17
1860     FOR I=0 TO Nend
1865         T(I)=FNTvsv(Emf(I))
1870     NEXT I
1875     Tvap=(T(0)+T(1)+T(2))/3
1880     Tliq=(T(3)+T(4))/2
1881     Tliq1=T(5)
1885     IF Isat=0 THEN
1890         Tsat=Tliq1
1895     END IF
1896     IF Isat=1 THEN
1900         Tsat=Tvap
1905     END IF
1906     IF Isat=2 THEN
1907         Tsat=(Tvap+Tliq)/2
1908     END IF
1925     Jset=Jset+1
1926     IF Okaccpt=0 THEN Jset=Jset-1
1930     PRINT
1935     PRINT USING "10X,""Data set number  = "",DD";Jset
1940     PRINT
1945     Ibeg=0
1950     Iend=3
1955     IF Iact<10 THEN
1960         Ibeg=Iact
1965         Iend=Iact
1970     END IF
1975     FOR I=Ibeg TO Iend
1980 !       Grad=FNGrad(Emf(I+6))
1981         Tout=(T(2*I+10)+T(2*I+11))/2
1985         Delt=Tout-T(I+6)
1990         Tavg=T(I+6)+Delt*.5
1991 !       IF I=Iend THEN
1992 !           Delt=T(10)-T(9)
1994 !           Tavg=T(10)-T(9)/2
1995 !       END IF
1996         Rhoeg=FNRhoeg(Tavg,Egrat)
2000         Nueg=FNNueg(Tavg,Egrat)
2005         Mueg=Nueg*Rhoeg
2010         Cpeg=FNCpeg(Tavg,Egrat)
2015         Keg=FNKeg(Tavg,Egrat)
2020         Preg=Cpeg*Mueg/Keg
2025         Mdot=FNFmcal(I,T(I+6),Pc(I))
2030         Veg=Mdot/(Rhoeg*Ax)
2035         Reeg=Veg*Di/Nueg
```

```
2040          Res=4*Mdot/(PI*Mueg*(Di-4*Delta))
2045          Qdot=Mdot*Cpeg*Delt
2050          Qdp(I)=Qdot/Ao
2055          IF I=0 OR I=Iact THEN
2060              Qdp1=Qdp(I)
2065              CALL Nusselt
2070          END IF
2075          Lmtd=Delt/LOG((Tsat-T(I+6))/(Tsat-T(I+6)-Delt))
2080          Uo(I)=Qdp(I)/Lmtd
2081          Cfeg=1
2085          IF Reeg<4000 THEN
2086          IF Insert=0 THEN
2088              Bb1=.0668*(Di/L)*Reeg*Preg
2089              Bb2=1+.04*((Di/L)*Reeg*Preg)^.6666
2090              Nueg=3.66+(Bb1/Bb2)
2091          END IF
2092              IF Insert=1 THEN
2094              Nueg=1.86*(Preg*Reeg)^.333333*(Di/L)^.33333*Cfeg
2095              Nueg=Cia(Insert,I)*(1+5.484E-3*Preg^.7*(Res/Hod)^1.25)^.5
2096              END IF
2097              IF Insert=2 THEN
2098              Nueg=Cia(Insert,I)*(Reeg^.65)*Preg^.46
2099    ──       END IF
2100              IF Insert=3 THEN
2101              Nueg=Cia(Insert,I)*(Reeg^.76)*Preg^.46
2102              END IF
2104          ELSE
2105              BEEP
2106              PRINT USING "10X,""TURBULENT CORRELATION"""
2110              Nueg=.027*Reeg^.8*Preg^.3333*Cfeg
2115          END IF
2120          Hi=Nueg*Keg/Di
2125          Ho(I)=1/(1/Uo(I)-Do/(Di*Hi)-Rm*A1)
2130          IF I=0 OR I=Iact THEN
2131              PRINT
2132              PRINT
2134              PRINT USING "10X,""RESULTS FOR TUBE NO.""",3D";I+1
2135              PRINT
2136              PRINT USING "10X,""Mass flow rate          = "",MZ.3DE";Mdot
2137!             PRINT USING "10X,""Inside Tube Dia. (m.)    = "",MZ.3DE";Dia(Itube
)
2138!             PRINT USING "10X,""180 DEG OVER Dia. (HOD) = "",MZ.3DE";Hod
2140              PRINT USING "10X,""Inlet temperature        = "",MZ.3DE";T(I+6)
2145              PRINT USING "10X,""Saturation temp (Deg C) = "",MZ.3DE";Tsat
2146!             PRINT USING "10X,""DELTA Tape Thickness     = "",MZ.3DE";Delta
2147              PRINT USING "10X,""DELT  temp  Dif.         = "",MZ.3DE";Delt
2148              PRINT USING "10X,""Log. Mean temp  Dif.     = "",MZ.3DE";Lmtd
2150              PRINT USING "10X,""Heat flux                = "",MZ.3DE";Qdp(I)
2151!             PRINT USING "10X,""Conductivity E.G.        = "",MZ.3DE";Keg
2152!             PRINT USING "10X,""Conductivity Tube Metal = "",MZ.3DE";Km
```

186

```
2153          PRINT USING "10X,""Resistance of Tube Metal= "",MZ.3DE";Rm
2155          PRINT USING "10X,""Prandtl number          = "",MZ.3DE";Preg
2160          PRINT USING "10X,""Reynolds number         = "",MZ.3DE";Reeg
2161          PRINT USING "10X,""Reynolds number H&B S   = "",MZ.3DE";Res
2165          PRINT USING "10X,""Inside h.t.c.           = "",MZ.3DE";Hi
2166          PRINT USING "10X,""Inside NUSSULT NO.      = "",MZ.3DE";Nueg
2167          PRINT USING "10X,""OVERALL H.t.c. (Uo)     = "",MZ.3DE";Uo(I)
2170          PRINT
2171!         PRINT USING "10X,""KF                       = "",MZ.3DE";Kf
2172!         PRINT USING "10X,""RHOF                     = "",MZ.3DE";Rhof
2173!         PRINT USING "10X,""HFG                      = "",MZ.3DE";Hfg
2174!         PRINT USING "10X,""MUF                      = "",MZ.3DE";Muf
2175          PRINT
2177          PRINT USING "10X,""TUBE       FM      VEG       DELT      Uo
     HO          HNUS"""
2180          PRINT USING "10X,""  #       (%)     (m/s)      (K)             (W/m^2.
K)"""
2185          PRINT USING "10X,3D,4X,3D.DD,3X,Z.DD,4X,DD.3D,2X,MZ.3DE,3X,MZ.3DE
,3X,MZ.3DE,3X";I+1,Pc(I),Veg,Delt,Uo(I),Ho(I),Hnus
2190       ELSE
2195          PRINT USING "10X,3D,4X,3D.DD,3X,Z.DD,4X,DD.3D,2X,MZ.3DE,3X,MZ.3DE
,3X";I+1,Pc(I),Veg,Delt,Uo(I),Ho(I)
2200       END IF
2205    NEXT I
2210    IF Im=0 THEN
2215       Okaccpt=1
2220       BEEP
2225       INPUT "OK TO ACCEPT THIS SET (1=DEFAULT=YES, 0=NO)?",Okaccpt
2230       IF Okaccpt=1 THEN OUTPUT @File;Pc(*),Emf(*),Tp(*)
2235    END IF
2240    IF (Okaccpt=1 OR Im=1) AND Iout=1 THEN
2245       FOR I=0 TO 2
2250          OUTPUT @Fout;Ho(I),Qdp(I)
2255       NEXT I
2260    END IF
2265    IF Im=0 THEN
2270       Okrpt=1
2275       BEEP
2280       INPUT "WILL THERE BE ANOTHER RUN (1=YES=DEFAULT,0=NO)",Okrpt
2285       IF Okrpt=1 THEN 1683
2290    ELSE
2295       IF Jset<Nsets THEN 1683
2300    END IF
2305    ASSIGN @File TO *
2310    IF Iout=1 THEN ASSIGN @File TO *
2315    SUBEND
2320    DEF FNGrad(T)
2325    Grad=-3.877857E-5-2*4.7142857E-8*T
2330    RETURN Grad
2335    FNEND
```

```
2340    DEF FNKcu(T)
2345 !  OFHC COPPER 250 TO 300 K
2350    Tk=T+273.15
2355    K=434-.112*Tk
2360    RETURN K
2365    FNEND
2370    DEF FNNueg(Tc,Egr)
2375 !  RANGE OF VALIDITY:  -20 TO 20 DEG C
2380    Tk=Tc+273.15
2385    Nu1=7.1196507E-3-Tk*(7.4863347E-5-Tk*(2.6294943E-7-Tk*3.0833329E-10))
2390    Nu2=4.9237638E-3-Tk*(4.9213912E-5-Tk*(1.6437534E-7-Tk*1.8333331E-10))
2395    Nu3=8.6586293E-3-Tk*(8.8837902E-5-Tk*(3.0495032E-7-Tk*3.4999996E-10))
2400    A2=(Nu3-2*Nu2+Nu1)/200
2405    A1=(Nu2-Nu1-940*A2)/10
2410    A0=Nu1-42*A1-1764*A2
2415    Nu=A0+Egr*(A1+Egr*A2)
2420    RETURN Nu
2425    FNEND
2430    DEF FNCpeg(Tc,Egr)
2435 !  RANGE OF VALIDITY: 0 TO 20 DEG C
2440    Tk=Tc+273.15
2445    Cp1=1.6701550E+3+Tk*6.3
2450    Cp2=1.4748125E+3+Tk*6.25
2455    Cp3=9.5800500E+2+Tk*7.3
2460    A2=(Cp3-2*Cp2+Cp1)/200
2465    A1=(Cp2-Cp1-900*A2)/10
2470    A0=Cp1-40*A1-1600*A2
2475    Cp=A0+Egr*(A1+Egr*A2)
2480    RETURN Cp
2485    FNEND
2490    DEF FNRhoeg(T,Egr)
2495    Ro1=1.0607093E+3-T*(3.7031283E-1+T*4.0837183E-3)
2500    Ro2=1.0748272E+3-T*(4.4266195E-1+T*4.0939706E-3)
2505    Ro3=1.0885934E+3-T*(5.7355653E-1+T*6.1281405E-3)
2510    A2=(Ro3-2*Ro2+Ro1)/200
2515    A1=(Ro2-Ro1-900*A2)/10
2520    A0=Ro1-40*A1-1600*A2
2525    Ro=A0+Egr*(A1+Egr*A2)
2530    RETURN Ro
2535    FNEND
2540    DEF FNPreg(T,Egr)
2545    Pr=FNCpeg(T,Egr)*FNNueg(T,Egr)*FNRoeg(T,Egr)/FNKeg(T,Egr)
2550    RETURN Pr
2555    FNEND
2560    DEF FNKeg(Tc,Egr)
2565 !  RANGE OF VALIDITY:  -20 TO 20 DEG C
2570    Tk=Tc+273.15
2575    K1=2.2824708E-1+Tk*(5.5989286E-4+Tk*3.5714286E-7)
2580    K2=2.5846616E-1+Tk*(2.3978571E-4+Tk*7.1428571E-7)
2585    K3=3.2138932E-1-Tk*(3.0042857E-4-Tk*1.4285714E-6)
```

```
2590      A2=(K3-2*K2+K1)/200
2595      A1=(K2-K1-900*A2)/10
2600      A0=K1-40*A1-1600*A2
2605      K=A0+Egr*(A1+Egr*A2)
2610      RETURN K
2615      FNEND
2620      DEF FNTanh(X)
2625      P=EXP(X)
2630      Q=1/P
2635      Tanh=(P-Q)/(P+Q)
2640      RETURN Tanh
2645      FNEND
2650      DEF FNTvsv(V)
2655      COM /Cc/ C(7)
2660      T=C(0)
2665      FOR I=1 TO 7
2670          T=T+C(I)*V^I
2675      NEXT I
2680      RETURN T
2685      FNEND
2690      DEF FNBeta(T)
2695      Rop=FNRho(T+.1)
2700      Rom=FNRho(T-.1)
2705      Beta=-2/(Rop+Rom)*(Rop-Rom)/.2
2710      RETURN Beta
2715      FNEND
2720      DEF FNPsat(Tc)
2725    ! 0 TO 80 deg F CURVE FIT OF Psat
2730      Tf=1.8*Tc+32
2735      Pa=5.945525+Tf*(.15352082+Tf*(1.4840963E-3+Tf*9.6150671E-6))
2740      Pg=Pa-14.7
2745      IF Pg>0 THEN        ! +=PSIG,-=in Hg
2750          Psat=Pg
2755      ELSE
2760          Psat=Pg*29.92/14.7
2765      END IF
2770      RETURN Psat
2775      FNEND
2850      SUB Xyread
2855      BEEP
2860      INPUT "ENTER FILE NAME",File$
2865      BEEP
2870      INPUT "ENTER NUMBER OF X,Y PAIRS",N
2875      ASSIGN @File TO File$
2880      FOR I=1 TO N
2885          ENTER @File;X,Y
2890          PRINT X,Y
2895      NEXT I
2900      SUBEND
2905      SUB Purge
```

189

```
2910      BEEP
2915      INPUT "ENTER FILE NAME TO BE DELETED",File$
2920      PURGE File$
2925      GOTO 2910
2930      SUBEND
2935      SUB Wilson
2940      COM /Will/ Doa(4),Dia(4),Kma(4),Iact,Droot(4)
2945      COM /Wil2/ Delta,Isat,Nsets,Hod,Cia(3,3),Alpaa(3)
2950      DIM Emf(17),T(17),Xa(100),Ya(100),Pc(4),Tp(4)
2955      BEEP
2960      INPUT "PLEASE RE-ENTER NAME OF FILE",File$
2965      ASSIGN @File TO File$
2970      INPUT "0=SMOOTH COPPER TUBE, 1=CUNI FIN, 2=KORODENSE, 3=FIN TITAN",Itube
2975      BEEP
2976      INPUT "0=NO INSERT, 1= TWISTED TAPE INSERT, 2=HITRAN1, 3=HITRAN2",Insert
2977      BEEP
2980      INPUT "GIVE A NAME FOR XY FILE",Xy$
2985      CREATE BDAT Xy$,20
2990      ASSIGN @Xy TO Xy$
2995      L=1.2192
3000      Do=Doa(Itube)
3005      Di=Dia(Itube)
3010      Km=Kma(Itube)
3015      Ax=PI*Di^2/4   ! Cross-sectional area
3020      Ao=PI*Do*L
3025      Rm=Do*LOG(Do/Di)/(2*Km)
3030 !
3035 !    Initial values
3040      Tf=Tsat
3045      Alpa=.655
3046      IF Insert=0 THEN
3050      Ci=1.0
3051      END IF
3052      IF Insert=1 THEN
3053      Ci=5.172
3054      END IF
3055      IF Insert=2 THEN
3056      Ci=.226
3057      END IF
3058      IF Insert=3 THEN
3059      Ci=.063
3060      END IF
3062      G=9.81
3063      Ibeg=0
3065      Iend=0   !CHANGE TO 4, IF FIVE TUBES IN BUNDLE
3070      IF Iact<10 THEN
3075          Ibeg=Iact
3080          Iend=Iact
3085      END IF
3090 !
```

```
3095     FOR I=Ibeg TO Iend
3100         Sx=0
3105         Sy=0
3110         Sxs=0
3115         Sxy=0
3120         Jset=0
3125         ASSIGN @File TO File$
3130         ENTER @File;Dtg$,Itube,Egrat,Dd1,Dd2,Dd3,Dd4,Dd5
3135         ENTER @File;Pc(*),Emf(*),Tp(*)
3140         FOR J=0 TO 17
3145             T(J)=FNTvsv(Emf(J))
3150         NEXT J
3155         Tvap=(T(0)+T(1)+T(2))/3
3160         Tliq=(T(3)+T(4))/2
3161         Tliq1=T(5)
3165         IF Isat=0 THEN
3170             Tsat=Tliq1
3175         ELSE
3180             Tsat=Tvap
3185         END IF
3190 !       Grad=FNGrad(T(I+6))
3191         Tout=(T(2*I+10)+T(2*I+11))/2
3195         Delt=Tout-T(I+6)
3200         Tavg=T(I+6)+Delt*.5
3201 !       IF I=Iend THEN
3202 !          Delt=T(10)-T(9)
3204 !          Tavg=(T(10)-T(9))/2
3205 !       END IF
3206 !
3210 !       Water/Ethylene Glycol Mixture Properties
3215         Rhoeg=FNRhoeg(Tavg,Egrat)
3220         Nueg=FNNueg(Tavg,Egrat)
3225         Mueg=Nueg*Rhoeg
3230         Cpeg=FNCpeg(Tavg,Egrat)
3235         Keg=FNKeg(Tavg,Egrat)
3240         Preg=Cpeg*Mueg/Keg
3245 !
3250         Mdot=FNFmcal(I,T(I+6),Pc(I))
3255         Veg=Mdot/(Rhoeg*Ax)
3260         Reeg=Veg*Di/Nueg
3265         Res=4*Mdot/(Mueg*(PI*Di-4*Delta))
3270         Qdot=Mdot*Cpeg*Delt
3275         Qdp=Qdot/Ao
3280         Lmtd=Delt/LOG((Tsat-T(I+6))/(Tsat-T(I+6)-Delt))
3285         Uo=Qdp/Lmtd
3286         IF Insert=0 THEN
3287         Bb1=.0668*(Di/L)*Reeg*Preg
3288         Bb2=1+.04*((Di/L)*Reeg*Preg)^.6666
3289         Omega=3.66+(Bb1/Bb2)
3290         END IF
```

191

```
3291          IF Insert=1 THEN
3293          Omega=(1+5.484E-3*Preg^.7*(Res/Hod)^1.25)^.5
3294          END IF
3295          IF Insert=2 THEN
3296          Omega=(Reeg^.65)*(Preg^.46)
3297          END IF
3298          IF Insert=3 THEN
3299          Omega=(Reeg^.76)*(Preg^.46)
3300          END IF
3302  !
3303  !       R-114/R113 Properties
3305          Hfg=FNHfg(Tsat)
3310          Kf=FNK(Tf)
3315          Rhof=FNRho(Tf)
3320          Muf=FNMu(Tf)
3325  !
3330          F=(Kf^3*Rhof^2*G*Hfg/(Muf*Do*Qdp))^.33333
3335          Ho=Alpa*F
3340          Two=Tsat-Qdp/Ho
3345          Tf=Tsat/3+2*Two/3
3350          Y=(1/Uo-Rm)*F
3355          X=Do*F/(Keg*Omega)
3360  !---    PRINT "OMEGA=";Omega;"F=";F;"X=";X;"Y=";Y
3365          Xa(Jset)=X     !   INEFFICIENT (MODIFY LATER)
3370          Ya(Jset)=Y
3375          Sx=Sx+X
3380          Sy=Sy+Y
3385          Sxs=Sxs+X*X
3390          Sxy=Sxy+X*Y
3395          Jset=Jset+1
3400          IF Jset<Nsets THEN 3135
3405          ASSIGN @File TO *
3410          Slope=(Nsets*Sxy-Sx*Sy)/(Nsets*Sxs-Sx^2)
3415          Intcpt=(Sy-Slope*Sx)/Nsets
3420          Cic=1/Slope
3425          Alpac=1/Intcpt
3430          Cerr=ABS((Ci-Cic)/Cic)
3435          Aerr=ABS((Alpac-Alpa)/Alpac)
3440          IF Cerr>.001 OR Aerr>.001 THEN
3445             Alpa=(Alpa+Alpac)*.5
3450             Ci=(Ci+Cic)*.5
3455  !          PRINT "CIC=";Cic;"ALPA=";Alpa
3460             GOTO 3100
3465          END IF
3470          BEEP
3475          BEEP
3480          PRINTER IS 1
3485          PRINT "CIC=";Cic;"ALPA=";Alpa
3490          Cia(Insert,I)=Cic
3495          Alpaa(I)=Alpac
```

```
3500        PRINTER IS 701
3505        FOR J=0 TO Nsets-1
3510            OUTPUT @Xy;Xa(J),Ya(J)
3515        NEXT J
3520        PRINTER IS 1
3525    NEXT I
3530    ASSIGN @Xy TO *
3535    SUBEND
3540    SUB Nusselt
3545    COM /Nus/ Iin,Tsat,Qdp,Hoc,Kf,Rhof,Hfg,Muf,Do,Itube
3550 !  Do=.0159
3555    Ho=1000
3560    IF Iin=0 THEN
3565        BEEP
3570        INPUT "ENTER TSAT AND HEAT FLUX",Tsat,Qdp
3575    END IF
3580    Hfg=FNHfg(Tsat)
3585    Two=Tsat-Qdp/Ho
3590    Tf=Tsat/3+2*Two/3
3595    Kf=FNK(Tf)
3600    Rhof=FNRho(Tf)
3605    Muf=FNMu(Tf)
3610    Hoc=.655*(Kf^3*Rhof^2*9.81*Hfg/(Muf*Do*Qdp))^.333333
3615    IF ABS((Ho-Hoc)/Hoc)>.001 THEN
3620        Ho=(Ho+Hoc)*.5
3625        GOTO 3585
3630    END IF
3635    IF Iin=0 THEN PRINT "HO=";Hoc
3640    SUBEND
3645    SUB Plot1
3650    DIM Yaa(4)
3655    PRINTER IS 705
3660    Idv=1
3665    BEEP
3670    INPUT "OK TO USE DEFAULT VALUES (1=DEF=Y,0=N)",Idv
3675    IF Idv=1 THEN
3680        Itn=2
3685        Xmin=1
3690        Xmax=5
3695        Xstep=1
3700        Ymin=0
3705        Ymax=2.0
3710        Ystep=.5
3715    ELSE
3720        BEEP
3725        INPUT "ENTER MINIMUM AND MAXIMUM X-VALUES",Xmin,Xmax
3730        BEEP
3735        INPUT "ENTER MINIMUM AND MAXIMUM Y-VALUES",Ymin,Ymax
3740        BEEP
3745        INPUT "ENTER STEP SIZE FOR X-AXIS",Xstep
```

```
3750        BEEP
3755         INPUT "ENTER STEP SIZE FOR Y-AXIS",Ystep
3760     END IF
3765     PRINT "IN;SP1;IP 2300,1800,8300,6800;"
3770     PRINT "SC 0,100,0,100;TL 2,0;"
3775     Sfx=100/(Xmax-Xmin)
3780     Sfy=100/(Ymax-Ymin)
3785     PRINT "PU 0,0 PD"
3790     FOR Xa=Xmin TO Xmax STEP Xstep
3795         X=(Xa-Xmin)*Sfx
3800         PRINT "PA";X,",0; XT;"
3805     NEXT Xa
3810     PRINT "PA 100,0;PU;"
3815     PRINT "PU PA 0,0 PD"
3820     FOR Ya=Ymin TO Ymax STEP Ystep
3825         Y=(Ya-Ymin)*Sfy
3830         PRINT "PA 0,";Y,"YT"
3835     NEXT Ya
3840     PRINT "PA 0,100 TL 0 2"
3845     FOR Xa=Xmin TO Xmax STEP Xstep
3850         X=(Xa-Xmin)*Sfx
3855         PRINT "PA";X,",100; XT"
3860     NEXT Xa
3865     PRINT "PA 100,100 PU PA 100,0 PD"
3870     FOR Ya=Ymin TO Ymax STEP Ystep
3875         Y=(Ya-Ymin)*Sfy
3880         PRINT "PD PA 100,",Y,"YT"
3885     NEXT Ya
3890     PRINT "PA 100,100 PU"
3895     PRINT "PA 0,-2 SR 1.5,2"
3900     FOR Xa=Xmin TO Xmax STEP Xstep
3905         X=(Xa-Xmin)*Sfx
3910         PRINT "PA";X,",0;"
3915         PRINT "CP -2,-1;LB";Xa;""
3920     NEXT Xa
3925     PRINT "PU PA 0,0"
3930     FOR Ya=Ymin TO Ymax STEP Ystep
3935         Y=(Ya-Ymin)*Sfy
3940         PRINT "PA 0,";Y,""
3945         PRINT "CP -4,-.25;LB";Ya;""
3950     NEXT Ya
3955     BEEP
3960     INPUT "SELECT MODE (0=HN/H1,1=HN(avg)/H1)",Ism
3965     Ism=Ism+1
3970     IF Idv=1 THEN
3975         IF Ism=1 THEN Ylabel$="HN/H1"
3980         IF Ism=2 THEN Ylabel$="HN(avg)/H1"
3985         Xlabel$="Tube Number"
3990     ELSE
3995         BEEP
```

```
4000        INPUT "ENTER X-LABEL",Xlabel$
4005        BEEP
4010        INPUT "ENTER Y-LABEL",Ylabel$
4015    END IF
4020    PRINT "SR 1.5,2;PU PA 50,-10 CP";-LEN(Xlabel$)/2;"0;LB";Xlabel$;""
4025    PRINT "PA -11,50 CP 0,";-LEN(Ylabel$)/2*5/6;"DI 0,1;LB";Ylabel$;""
4030    PRINT "CP 0,0 DI"
4035    Okp=1
4040    BEEP
4045    INPUT "WANT TO PLOT DATA FROM A FILE (1=DEF=Y,0=N)?",Okp
4050    IF Okp=1 THEN
4055        BEEP
4060        INPUT "ENTER THE NAME OF THE DATA FILE",Dfile$
4065        ASSIGN @File TO Dfile$
4070        BEEP
4075        INPUT "ENTER THE BEGINNING RUN NUMBER",Md
4080        BEEP
4085        INPUT "ENTER THE NUMBER OF X-Y PAIRS STORED",Nsets
4090        BEEP
4095        INPUT "SELECT A SYMBOL FOR THE PLOTTER (1=*,2=+,3=c,4=o,5=^)",Sy
4100        PRINT "PU DI"
4105        IF Sy=1 THEN PRINT "SM*"
4110        IF Sy=2 THEN PRINT "SM+"
4115        IF Sy=3 THEN PRINT "SMc"
4120        IF Sy=4 THEN PRINT "SMo"
4125        IF Sy=5 THEN PRINT "SM^"
4130        FOR I=1 TO Nsets
4135            FOR J=0 TO 3
4140                ENTER @File;Yaa(J),D
4145                IF J=0 THEN Ytop=Yaa(0)
4150                Yaa(J)=Yaa(J)/Ytop
4155            NEXT J
4160            FOR J=0 TO 3
4165                X=(J+1-Xmin)*Sfx
4170                Y='Yaa(J)-Ymin)*Sfy
4175                PRINT "PA",X,Y,""
4180            NEXT J
4185        NEXT I
4190        BEEP
4195        ASSIGN @File TO *
4200        GOTO 4040
4205    END IF
4210    BEEP
4215    INPUT "LIKE TO PLOT THE NUSSELT RELATION (1=Y,0=N)?",Oknus
4220    PRINT "PU;SM"
4225    IF Oknus=1 THEN
4230        FOR Xa=Xmin TO Xmax STEP Xstep/50
4235            X=(Xa-Xmin)*Sfx
4240            IF Ism=1 AND Xa>Xmin THEN Ya=Xa^.75-(Xa-1)^.75
4245            IF Ism=2 AND Xa>Xmin THEN Ya=Xa^(-.25)
```

```
4250            IF Xa=Xmin THEN Ya=1
4255            Y=(Ya-Ymin)*Sfy
4260            PRINT "PA",X,Y,"PD"
4265        NEXT Xa
4270        BEEP
4275        PRINT "PU"
4280        INPUT "MOVE THE PEN TO LABEL THE NUSSELT LINE",Ok
4285        PRINT "LBNusselt"
4290    END IF
4295    IF Ism=2 THEN
4300        BEEP
4305        INPUT "LIKE TO PLOT EXPTL CURVE (1=Y,0=N)",Okex
4310        Nq=0
4315        IF Okex=1 THEN
4320            BEEP
4325            INPUT "ENTER THE EXPONENT",Ex
4330            FOR Xa=Xmin TO Xmax STEP Xstep/10
4335                Nq=Nq+1
4340                Ya=Xa^(-Ex)
4345                X=(Xa-Xmin)*Sfx
4350                Y=(Ya-Ymin)*Sfy
4355                IF Nq MOD 2=0 THEN
4360                    PRINT "PA",X,Y,"PD"
4365                ELSE
4370                    PRINT "PA",X,Y,"PU"
4375                END IF
4380            NEXT Xa
4385            PRINT "PU"
4390            BEEP
4395            INPUT "MOVE PEN TO LABEL AND HIT ENTER",Ok
4400            PRINT "LBs=0"
4405            PRINT "PR -1 0"
4410            PRINT "LB";Ex;""
4415            GOTO 4300
4420        END IF
4425    END IF
4430    GOTO 4510
4435    BEEP
4440    INPUT "LIKE TO PLOT KERN RELATIONSHIP (1=Y,0=N)?",Yes
4445    IF Yes=1 THEN
4450        FOR Xa=Xmin TO Xmax STEP Xstep/20
4455            Ya=Xa^(-1/6)
4460            X=(Xa-Xmin)*Sfx
4465            Y=(Ya-Ymin)*Sfy
4470            PRINT "PA",X,Y,"PD"
4475        NEXT Xa
4480        PRINT "PU"
4485        BEEP
4490        INPUT "MOVE THE PEN TO LABEL KERN RELATIONSHIP",Ok
4495        PRINT "LBKern:PU"
```

```
4500     END IF
4505     PRINT "PU PA 0,0"
4510     BEEP
4515     INPUT "LIKE TO PLOT EISSENBERG RELATION (1=Y,0=N)?",Okei
4520     IF Okei=1 THEN
4525         FOR Xa=Xmin TO Xmax STEP Xstep/10
4530             Ya=.6+.42*Xa^(-.25)
4535             X=(Xa-Xmin)*Sfx
4540             Y=(Ya-Ymin)*Sfy
4545             PRINT "PA",X,Y,"PD"
4550         NEXT Xa
4555         PRINT "PU"
4560         BEEP
4565         INPUT "MOVE THE PEN TO LABEL THE EISSENBERG LINE",Ok
4570         PRINT "LBEissenberg;PU"
4575     END IF
4580     PRINT "PU SP0"
4585     SUBEND
4590     DEF FNPvst(Tc)
4595     COM /Fld/ Ift
4600     DIM K(8)
4605     IF Ift=0 THEN
4610         BEEP
4615         PRINT "PVST CORRELATION NOT AVAILABLE FOR R-114"
4620         STOP
4625     END IF
4630     IF Ift=1 THEN
4635         DATA -7.691234564,-26.08023696,-168.1706546,64.23285504,-118.9646225
4640         DATA 4.16711732,20.9750676,1.E9,6
4645         READ K(*)
4650         T=(Tc+273.15)/647.3
4655         Sum=0
4660         FOR N=0 TO 4
4665             Sum=Sum+K(N)*(1-T)^(N+1)
4670         NEXT N
4675         Br=Sum/(T*(1+K(5)*(1-T)+K(6)*(1-T)^2))-(1-T)/(K(7)*(1-T)^2+K(8))
4680         Pr=EXP(Br)
4685         P=22120000*Pr
4690     END IF
4695     IF Ift=2 THEN
4700         Tf=Tc*1.8+32+459.6
4705         P=10^(33.0655-4330.98/Tf-9.2635*LGT(Tf)+2.0539E-3*Tf)
4710         P=P*101325/14.696
4715     END IF
4720     IF Ift=3 THEN
4725         A=9.394685-3066.1/(Tc+273.15)
4730         P=133.32*10^A
4735     END IF
4740     RETURN P
4745     FNEND
```

```
4750    DEF FNHfg(T)
4755    COM /Fld/ Ift
4760    IF Ift=0 THEN
4765        Tf=T*1.8+32
4770        Hfg=6.1451558E+1-Tf*(6.951079E-2+Tf*(1.3988688E-4+1.9607843E-7*Tf))
4775        Hfg=Hfg*2326
4780    END IF
4785    IF Ift=1 THEN
4790        Hfg=2477200-2450*(T-10)
4795    END IF
4800    IF Ift=2 THEN
4805        Tf=T*1.8+32
4810        Hfg=7.0557857E+1-Tf*(4.838052E-2+1.2619048E-4*Tf)
4815        Hfg=Hfg*2326.1
4816!       Hfg=(1.611-.0031*T)*1.0E+5
4820    END IF
4825    IF Ift=3 THEN
4830        Tk=T+273.15
4835        Hfg=1.35264E+6-Tk*(6.38263E+2+Tk*.747462)
4840    END IF
4845    RETURN Hfg
4850    FNEND
4855    DEF FNMu(T)
4860    COM /Fld/ Ift
4865    IF Ift=0 THEN
4870        Tk=T+273.15
4875        Mu=EXP(-4.4636+1011.47/Tk)*1.E-3
4880    END IF
4885    IF Ift=1 THEN
4890        A=247.8/(T+133.15)
4895        Mu=2.4E-5*10^A
4900    END IF
4905    IF Ift=2 THEN
4910!       Mu=8.9629819E-4-T*(1.1094609E-5-T*5.566829E-8)
4911        Tk=T+273.15
4913        Mu=1.34E-5*10.0^(503.0/(Tk-2.15))
4915    END IF
4920    IF Ift=3 THEN
4925        Tk=1/(T+273.15)
4930        Mu=EXP(-11.0179+Tk*(1.744E+3-Tk*(2.80335E+5-Tk*1.12661E+8)))
4935    END IF
4940    RETURN Mu
4945    FNEND
4950    DEF FNVvst(Tt)
4955    COM /Fld/ Ift
4960    IF Ift=0 THEN
4965        BEEP
4970        PRINT "VVST CORRELATION NOT AVAILABLE FOR R-114"
4975        STOP
4980    END IF
```

```
4985      IF Ift=1 THEN
4990         P=FNPvst(Tt)
4995         T=Tt+273.15
5000         X=1500/T
5005         F1=1/(1+T*1.E-4)
5010         F2=(1-EXP(-X))^2.5*EXP(X)/X^.5
5015         B=.0015*F1-.000942*F2-.0004882*X
5020         K=2*P/(461.52*T)
5025         V=(1+(1+2*B*K)^.5)/K
5030      END IF
5035      IF Ift=2 THEN
5040         Tf=Tt*1.8+32
5045         V=13.955357-Tf*(.16127262-Tf*5.1726190E-4)
5050         V=V/16.018
5055      END IF
5060      IF Ift=3 THEN
5065         Tk=Tt+273.15
5070         P=FNPvst(Tt)
5075         V=133.95*Tk/P
5080      END IF
5085      RETURN V
5090      FNEND
5095      DEF FNCp(T)
5100      COM /Fld/ Ift
5105      IF Ift=0 THEN
5110         Tk=T+273.15
5115         Cp=.40118+Tk*(1.65007E-3+Tk*(1.51494E-6-Tk*6.67853E-10))
5120      END IF
5125      IF Ift=1 THEN
5130         Cp=4.21120858-T*(2.26826E-3-T*(4.42361E-5+2.71428E-7*T))
5135      END IF
5140      IF Ift=2 THEN
5145!        Cp=9.2507273E-1+T*(9.3400433E-4+1.7207792E-6*T)
5146         Cp=(929.0+1.03*T)/1000.
5150      END IF
5155      IF Ift=3 THEN
5160         Tk=T+273.15
5165         Cp=4.1868*(1.6884E-2+Tk*(3.35083E-3-Tk*(7.224E-6-Tk*7.61748E-9)))
5170      END IF
5175      RETURN Cp*1000
5180      FNEND
5185      DEF FNRho(T)
5190      COM /Fld/ Ift
5195      IF Ift=0 THEN
5200         Tk=T+273.15
5205         X=1-(1.8*Tk/753.95)
5210         Ro=36.32+61.146414*X^(1/3)+16.418015*X+17.476838*X^.5+1.119828*X^2
5215         Ro=Ro/.062428
5220      END IF
5225      IF Ift=1 THEN
```

```
5230        Ro=999.52946+T*(.01269-T*(5.482513E-3-T*1.234147E-5))
5235    END IF
5240    IF Ift=2 THEN
5245        Ro=1.6207479E+3-T*(2.2186346+T*2.3578291E-3)
5246|       Ro=1./((.617+.000647*T^1.1)*1.0E-3)
5250    END IF
5255    IF Ift=3 THEN
5260        Tk=T+273.15-338.15
5265        Vf=9.24848E-4+Tk*(6.2796E-7+Tk*(9.2444E-10+Tk*3.057E-12))
5270        Ro=1/Vf
5275    END IF
5280    RETURN Ro
5285    FNEND
5290    DEF FNPr(T)
5295    Pr=FNCp(T)*FNMu(T)/FNK(T)
5300    RETURN Pr
5305    FNEND
5310    DEF FNK(T)
5315    COM /Fld/ Ift
5320    IF Ift=0 THEN K=.071-.000261*T
5325    IF Ift=1 THEN
5330        X=(T+273.15)/273.15
5335        K=-.92247+X*(2.8395-X*(1.8007-X*(.52577-.07344*X)))
5340    END IF
5345    IF Ift=2 THEN
5350|       K=8.2095238E-2-T*(2.2214286E-4+T*2.3809524E-8)
5351|       K=.0802-.000203*T
5352|       K=.071-.000261*T
5353        Tf=T*1.8+32
5354        K=4.8461905E-2-Tf*6.5714286E-5
5355        K=K*1.7308
5357    END IF
5360    IF Ift=3 THEN
5365        Tk=T+273.15
5370        K=4.1868E-4*(519.442+.320920*Tk)
5375    END IF
5380    RETURN K
5385    FNEND
5390    DEF FNHf(T)
5395    COM /Fld/ Ift
5400    IF Ift=0 THEN
5405        BEEP
5410        PRINT "HF CORRELATION NOT FOR R-114"
5415        STOP
5420    END IF
5425    IF Ift=1 THEN
5430        Hf=T*(4.203849-T*(5.88132E-4-T*4.55160317E-6))
5435    END IF
5440    IF Ift=2 THEN
5445        Tf=T*1.8+32
```

```
5450        Hf=8.2078571+Tf*(.19467857+Tf*1.3214286E-4)
5455        Hf=Hf*2.326
5460     END IF
5465     IF Ift=3 THEN
5470        Hf=250   ! TO BE VERIFIED
5475     END IF
5480     RETURN Hf*1000
5485     FNEND
5490     SUB Plot2
5495     COM /Dr1/ Star,Sym,Icon
5500     COM /Fld/ Ift
5505     DIM C(9),Xya(7),Doa(4)
5510     DATA 0.0158,0.0158,0.0158,0.0158
5515     READ Doa(*)
5520     Fw=1
5525     PRINTER IS 1
5530     BEEP
5535     PRINT USING "4X,""Select Option X-Y Limits:"""
5540     PRINT USING "6X,""0 Use default values"""
5545     PRINT USING "6X,""1 Use new values"""
5550     INPUT Okd
5555     BEEP
5560     INPUT "ENTER TUBE CODE",Icode
5565     Do=Doa(Itube)
5570     Iht=2
5575     BEEP
5580     PRINT USING "4X,""Select option:"""
5585     PRINT USING "6X,""0 h versus q"""
5590     PRINT USING "6X,""1 q versus Delta-T"""
5595     PRINT USING "6X,""2 h versus Delta-T (default)"""
5600     INPUT Iht
5605     PRINTER IS 705
5610     IF Okd=0 THEN !AXIS DEFAULT VALUES
5615        IF Iht=0 THEN !(h vs q)
5620           Ymin=0
5625           Ymax=60
5630           Ystep=10
5635           Xmin=.2
5640           Xmax=1.4
5645           Xstep=.2
5650        END IF
5655        IF Iht=1 THEN !(q vs t)
5660           Xmin=0
5665           Ymin=0
5670           Ymax=.5
5675           Xmax=15
5680           Xstep=3
5685           Ystep=.1
5690        END IF
5695        IF Iht=2 THEN !(h vs t)
```

```
5700            Xmin=0
5705            Ymin=0
5710            Xmax=50
5715            Ymax=6
5720            Xstep=10
5725            Ystep=1
5730         END IF
5735      END IF
5740      IF Ord=1 THEN
5745         BEEP
5750         INPUT "ENTER MINIMUM AND MAXIMUM X-VALUES",Xmin,Xmax
5755         BEEP
5760         INPUT "ENTER MINIMUM AND MAXIMUM Y-VALUES",Ymin,Ymax
5765         BEEP
5770         INPUT "ENTER STEP SIZE FOR X-AXIS",Xstep
5775         BEEP
5780         INPUT "ENTER STEP SIZE FOR Y-AXIS",Ystep
5785      END IF
5790      BEEP
5795      PRINT "IN;SP1;IP 2300,1800,8300,6800;"
5800      PRINT "SC 0,100,0,100;TL 2,0;"
5805      Sfx=100/(Xmax-Xmin)
5810      Sfy=100/(Ymax-Ymin)
5815      BEEP
5820      Icg=0
5825      INPUT "LIKE TO BY-PASS CAGE (1=Y,0=N=DEFAULT)?",Icg
5830      IF Icg=1 THEN 6175
5835      PRINT "PU 0,0 PD"
5840      FOR Xa=Xmin TO Xmax STEP Xstep
5845         X=(Xa-Xmin)*Sfx
5850         PRINT "PA";X,",0; XT;"
5855      NEXT Xa
5860      PRINT "PA 100,0;PU;"
5865      PRINT "PU PA 0,0 PD"
5870      FOR Ya=Ymin TO Ymax STEP Ystep
5875         Y=(Ya-Ymin)*Sfy
5880         PRINT "PA 0,";Y,"YT"
5885      NEXT Ya
5890      PRINT "PA 0,100 TL 0 2"
5895      FOR Xa=Xmin TO Xmax STEP Xstep
5900         X=(Xa-Xmin)*Sfx
5905         PRINT "PA";X,",100; XT"
5910      NEXT Xa
5915      PRINT "PA 100,100 PU PA 100,0 PD"
5920      FOR Ya=Ymin TO Ymax STEP Ystep
5925         Y=(Ya-Ymin)*Sfy
5930         PRINT "PD PA 100,";Y,"YT"
5935      NEXT Ya
5940      PRINT "PA 100,100 PU"
5945      PRINT "PA 0,-2 SR 1.5,2"
```

```
5950    FOR Xa=Xmin TO Xmax STEP Xstep
5955        X=(Xa-Xmin)*Sfx
5960        PRINT "PA";X,",0:"
5965        IF Xa<1 AND Xa<>0 THEN PRINT "CP -1.5,-1:LB0;PR -1,0:LB";Xa;""
5970        IF Xa=0 THEN PRINT "CP -.5,-1:LB0" .
5975        Xin=0
5980        IF Xa MOD 1=0 THEN Xin=1
5985        IF Xa>=10 THEN PRINT "CP -2,-1:LB";Xa;""
5990        IF Xa>1 AND Xa<10 AND Xin=1 THEN PRINT "CP -1.25,-1:LB";Xa;""
5995        IF Xa>1 AND Xin=0 THEN PRINT "CP -2,-1:LB";Xa;""
6000        IF Xa=1 THEN PRINT "CP -1,-1:LB1.0"
6005    NEXT Xa
6010    PRINT "PU PA 0,0"
6015    FOR Ya=Ymin TO Ymax STEP Ystep
6020        Y=(Ya-Ymin)*Sfy
6025        PRINT "PA 0,";Y,""
6030        IF Iht=0 AND Ya>0 THEN PRINT "PR 2,0"
6035        IF Ya<1 AND Ya<>0 THEN PRINT "CP -4,-.25:LB0;PR -2,0:LB";Ya;""
6040        IF Ya=0 THEN PRINT "CP -2,-.25:LB0"
6045        IF Ya>1 AND Iht<2 THEN PRINT "CP -5,-.25:LB";Ya;""
6050        IF Ya>9 AND Iht=2 THEN PRINT "CP -4,-.25:LB";Ya;""
6055        IF Ya>0 AND Ya<10 AND Iht=2 THEN PRINT "CP -3,-.25:LB";Ya;""
6060  |     IF Ya=1 THEN PRINT "CP -4,-.25:LB1.0"
6065    NEXT Ya
6070    IF Ord=1 THEN
6075        BEEP
6080        INPUT "ENTER X-LABEL",Xlabel$
6085        BEEP
6090        INPUT "ENTER Y-LABEL",Ylabel$
6095    END IF
6100    IF Iht<>1 THEN
6105        PRINT "SR 1.5,2:PU PA -12,35:DI 0,1:LBh:PR 1,0.5:LBo:PR -1,0.5:LB/(
kW/m"
6110        PRINT "PR -1,0.5:SR 1,1.5:LB2:SR 1.5,2:PR .5,.5:LB.:PR .5,0:LBK)"
6115    ELSE
6120        PRINT "PA -12,39:DI 0,1:LBq/(MW/m:SR 1,1.5:PR -1,0.5:LB2:SR 1.5,2:P
R 1.0,0.5:LB)"
6125    END IF
6130    IF Iht=0 THEN
6135        PRINT "SR 1.5,2:PU PA 40,-10:DI:LBq/(MW/m:PR 0.5,1:SR 1,1.5:LB2:SR
1.5,2:PR .5,-1:LB)"
6140    ELSE
6145        PRINT "DI PA 38,-10:LB(T:PR .5,-1:LBs;PR .5,1:LB-T:PR -2.4,3 PD PR
 2.0 PU:PR .5,-4:"
6150        PRINT "LBwo:PR .5,1:LB)/K"
6155    END IF
6160    PRINT "CP 0,0 DI"
6165    Xlg=1.E+6
6170    Xug=-1.E+6
6175    Xal=50
```

```
6180    Yal=95
6185    Nrun=0
6190    BEEP
6195    INPUT "WANT TO PLOT DATA FROM A FILE (1=Y,0=N)?",Ok
6200    Xll=1.E+6
6205    Xul=-1.E+6
6210    Okp=0
6215    IF Ok=1 THEN
6220        BEEP
6225        INPUT "ENTER THE NAME OF THE PLOT DATA FILE",D_file$
6230        ASSIGN @File TO D_file$
6235        IF Icomb<>0 THEN 6265
6240        Sx=0
6245        Sy=0
6250        Sx2=0
6255        Sxy=0
6260        Md=1
6265        BEEP
6270        INPUT "ENTER THE BEGINNING RUN NUMBER (DEF=1)",Md
6275        Npairs=9
6280        BEEP
6285        INPUT "ENTER THE NUMBER OF X-Y PAIRS STORED (DEF=9)",Npairs
6290    --  Nrun=Nrun+Npairs
6295        PRINTER IS 1
6300        BEEP
6305        PRINT USING "4X,""Select a symbol:"""
6310        PRINT USING "4X,"" 1  Star    2  Plus sign"""
6315        PRINT USING "4X,"" 3  Circle  4  Square"""
6320        PRINT USING "4X,"" 5  Rombus"""
6325        PRINT USING "4X,"" 6  Right-side-up triangle"""
6330        PRINT USING "4X,"" 7  Up-side-down triangle"""
6335        INPUT Sym
6340        BEEP
6345        INPUT "ENTER TUBE NUMBER FOR PLOTTING (0=TOP,1=SECOND,...)",Itube
6350        PRINTER IS 705
6355        IF Sym=1 THEN PRINT "SM*"
6360        IF Sym=2 THEN PRINT "SM+"
6365        IF Sym=3 THEN PRINT "SMo"
6370        IF Md>1 THEN
6375            FOR I=1 TO (Md-1)
6380                ENTER @File;Xya(*)
6385            NEXT I
6390        END IF
6395        FOR I=1 TO Npairs
6400            ENTER @File;Xya(*)
6405            Ya=Xya(Itube*2)
6410            Xa=Xya(Itube*2+1)
6415            Yc=LOG(Xa)
6420            Xc=LOG(Xa/Ya)
6425            Sx=Sx+Xc
```

```
6430              Sy=Sy+Yc
6435              Sx2=Sx2+Xc^2
6440              Sxy=Sxy+Xc*Yc
6445              IF Iht=0 THEN
6450                  Xt=Ya
6455                  Ya=Ya/Xa
6460                  Xa=Xt
6465                  IF Xa/1.E+6>Xul THEN Xul=Xa/1.E+6
6470                  IF Xa/1.E+6<Xll THEN Xll=Xa/1.E+6
6475              END IF
6480              IF Iht=0 THEN
6485                  X=(Xa*1.E-6-Xmin)*Sfx
6490                  Y=(Ya*1.E-3-Ymin)*Sfy
6495              END IF
6500              IF Iht=1 THEN
6505                  X=(Xa-Xmin)*Sfx
6510                  Y=(Ya*1.E-6-Ymin)*Sfy
6515              END IF
6520              IF Iht=2 THEN
6525                  X=(Xa/Ya-Xmin)*Sfx
6530                  Y=(Ya*1.E-3-Ymin)*Sfy
6535              END IF
6540              IF Y>100 OR Y<0 THEN 6585
6545              IF Sym>3 THEN PRINT "SM"
6550              IF Sym<4 THEN PRINT "SR 1.4,2.4"
6555              PRINT "PA",X,Y,""
6560              IF Sym>3 THEN PRINT "SR 1.2,1.6"
6565              IF Sym=4 THEN PRINT "UC2,4,99,0,-8,-4,0,0,8,4,0;"
6570              IF Sym=5 THEN PRINT "UC3,0,99,-3,-6,-3,6,3,6,3,-6;"
6575              IF Sym=6 THEN PRINT "UC0,5.3,99,3,-8,-6,0,3,8;"
6580              IF Sym=7 THEN PRINT "UC0,-5.3,99,-3,8,6,0,-3,-8;"
6585          NEXT I
6590          BEEP
6595          INPUT "WANT TO LABEL (1=Y,0=N)?",Ilbl
6600          IF Ilbl=1 THEN
6605              IF Sym>3 THEN PRINT "SM"
6610              IF Sym<4 THEN PRINT "SR 1.4,2.4"
6615              PRINT "PA",Xal,Yal,""
6620              IF Sym>3 THEN PRINT "SR 1.2,1.6"
6625              IF Sym=4 THEN PRINT "UC2,4,99,0,-8,-4,0,0,8,4,0;"
6630              IF Sym=5 THEN PRINT "UC3,0,99,-3,-6,-3,6,3,6,3,-6;"
6635              IF Sym=6 THEN PRINT "UC0,5.3,99,3,-8,-6,0,3,8;"
6640              IF Sym=7 THEN PRINT "UC0,-5.3,99,-3,8,6,0,-3,-8;"
6645              PRINT "SM"
6650              IF Sym<4 THEN PRINT "PR 2,0"
6655              PRINT "PR 2,-1.0;SR 1.0,1.8;LB";D_file$;""
6660              Yal=Yal-5
6665              BEEP
6670              INPUT "WANT TO ADD ANOTHER STRING (1=Y,0=N)?",Ias
6675              IF Ias=1 THEN
```

205

```
6680            BEEP
6685            INPUT "ENTER THE STRING",Label$
6690            PRINT "PR 2,0;SR 1.0,1.8;LB";Label$;""
6695            GOTO 6665
6700          END IF
6705        END IF
6710      BEEP
6715      INPUT "WANT TO COMBINE ANOTHER FILE? (1=Y,0=N)",Icomb
6720      ASSIGN @File TO *
6725      Xll=5
6730      Xul=45
6735      IF Icomb<>0 THEN 6220
6740      BEEP
6745      INPUT "WANT TO PLOT A LEAST-SQUARES LINE (1=Y,0=N)",Ils
6750      IF Ils=1 THEN
6755        BEEP
6760        INPUT "SELECT EXPONENT: 0=COMPUTE, 1=0.75)",Iexp
6765        BEEP
6770        INPUT "SELECT CURVE TYPE (0=SOLID,1=DASHED)",Ilt
6775        Ilt=Ilt+1
6780        PRINT "SM"
6785        IF Iexp=0 THEN
6790            Bb=(Nrun*Sxy-Sy*Sx)/(Nrun*Sx2-Sx^2)
6795        ELSE
6800            Bb=.75
6805        END IF
6810        Aa=(Sy-Bb*Sx)/Nrun
6815        Aa=EXP(Aa)
6820        PRINTER IS 1
6825        PRINT USING "10X,""a = "",Z.4DE";Aa
6830        PRINT USING "10X,""n = "",Z.4DE";Bb
6835        PRINTER IS 705
6840        In=0
6845        IF Iht=0 THEN Xxstep=Xstep/40
6850        IF Iht>0 THEN Xxstep=Xstep/10
6855        FOR Xa=Xll TO Xul STEP Xxstep
6860            IF Xa>.99*Xmax THEN 6995
6865            IF Iht=1 THEN Ya=Aa*Xa^Bb
6870            IF Iht=0 THEN Ya=Aa^(1/Bb)*(Xa*1.E+6)^((Bb-1)/Bb)
6875            IF Iht=2 THEN Ya=Aa*Xa^(Bb-1)
6880            IF Iht=0 THEN
6885                Y=(Ya*1.E-3-Ymin)*Sfy
6890                X=(Xa-Xmin)*Sfx
6895            END IF
6900            IF Iht=1 THEN
6905                Y=(Ya*1.E-6-Ymin)*Sfy
6910                X=(Xa-Xmin)*Sfx
6915            END IF
6920            IF Iht=2 THEN
6925              . Y=(Ya*1.E-3-Ymin)*Sfy
```

```
6930                   X=(Xa-Xmin)*Sfx
6935               END IF
6940               IF Y<0 THEN Y=0
6945               IF Y>100 THEN 6990
6950               IF Ilt=1 THEN
6955                   PRINT "PA",X,Y,"PD"
6960               ELSE
6965                   In=In+1
6970                   Ir=In MOD Ilt
6975                   IF Ir=1 THEN PRINT "PA",X,Y,"PD"
6980                   IF Ir=0 THEN PRINT "PA",X,Y,"PU"
6985               END IF
6990           NEXT Xa
6995           PRINT "PU"
7000       END IF
7005       Icomb=0
7010       GOTO 6185
7015   END IF
7020   PRINT "PU SM"
7025   BEEP
7030   INPUT "WANT TO PLOT NUSSELT LINE (1=Y,0=N)?",Inp
7035   IF Inp=0 THEN 7125
7040   BEEP
7045   INPUT "ENTER TSAT (DEFAULT=18 DEG C)",Tsat
7050   Hfg=FNHfg(Tsat)
7055   Xll=5
7060   Xul=45
7065   FOR Xa=Xll TO Xul STEP Xstep/50
7070       Tfilm=Tsat-Xa*.5
7075       Kf=FNK(Tfilm)
7080       Rhof=FNRho(Tfilm)
7085       Muf=FNMu(Tfilm)
7090       Ya=.728*(Kf^3*Rhof^2*9.81*Hfg/(Muf*Do*Xa))^.25
7095       X=(Xa-Xmin)*Sfx
7100       Y=(Ya*1.E-3-Ymin)*Sfy
7105       PRINT "PA",X,Y,"PD"
7110   NEXT Xa
7115   PRINT "PU PA 0,0"
7120   PRINT "PU PA 0,0 SP0"
7125   SUBEND
7130   SUB Plot3
7135   COM /Dr1/ Star,Sym,Icon
7140   COM /Fld/ Ift
7145   DIM C(9)
7150   Fw=1
7155   PRINTER IS 1
7160   BEEP
7165   PRINT USING "4X,""Select Option X-Y Limits:"""
7170   PRINT USING "6X,""0 Use default values"""
7175   PRINT USING "6X,""1 Use new values"""
```

```
7180      INPUT Okd
7185      PRINTER IS 705
7190      IF Okd=0 THEN
7195          Xmin=0
7200          Ymin=0
7205          Xmax=15
7210          Ymax=15
7215          Xstep=3
7220          Ystep=3
7225      ELSE
7230          BEEP
7235          INPUT "ENTER MINIMUM AND MAXIMUM X-VALUES",Xmin,Xmax
7240          BEEP
7245          INPUT "ENTER MINIMUM AND MAXIMUM Y-VALUES",Ymin,Ymax
7250          BEEP
7255          INPUT "ENTER STEP SIZE FOR X-AXIS",Xstep
7260          BEEP
7265          INPUT "ENTER STEP SIZE FOR Y-AXIS",Ystep
7270      END IF
7275      BEEP
7280      PRINT "IN;SP1;IP 2300,1800,8300,6800;"
7285      PRINT "SC 0,100,0,100;TL 2,0;"
7290      Sfx=100/(Xmax-Xmin)
7295      Sfy=100/(Ymax-Ymin)
7300      BEEP
7305      Icg=0
7310      INPUT "LIKE TO BY-PASS CAGE (1=Y,0=N=DEFAULT)?",Icg
7315      IF Icg=1 THEN 7620
7320      PRINT "PU 0,0 PD"
7325      FOR Xa=Xmin TO Xmax STEP Xstep
7330          X=(Xa-Xmin)*Sfx
7335          PRINT "PA";X,",0; XT;"
7340      NEXT Xa
7345      PRINT "PA 100,0;PU;"
7350      PRINT "PU PA 0,0 PD"
7355      FOR Ya=Ymin TO Ymax STEP Ystep
7360          Y=(Ya-Ymin)*Sfy
7365          PRINT "PA 0,";Y,"YT"
7370      NEXT Ya
7375      PRINT "PA 0,100 TL 0 2"
7380      FOR Xa=Xmin TO Xmax STEP Xstep
7385          X=(Xa-Xmin)*Sfx
7390          PRINT "PA";X,",100; XT"
7395      NEXT Xa
7400      PRINT "PA 100,100 PU PA 100,0 PD"
7405      FOR Ya=Ymin TO Ymax STEP Ystep
7410          Y=(Ya-Ymin)*Sfy
7415          PRINT "PD PA 100,",Y,"YT"
7420      NEXT Ya
7425      PRINT "PA 100,100 PU"
```

```
7430        PRINT "PA 0,-2 SR 1.5,2"
7435        FOR Xa=Xmin TO Xmax STEP Xstep
7440            X=(Xa-Xmin)*Sfx
7445            PRINT "PA";X,",0;"
7450            IF Xa<1 AND Xa<>0 THEN PRINT "CP -1.5,-1;LB0;PR -1,0;LB";Xa;""
7455            IF Xa=0 THEN PRINT "CP -.5,-1;LB0"
7460            Xin=0
7465            IF Xa MOD 1=0 THEN Xin=1
7470            IF Xa>=10 THEN PRINT "CP -2,-1;LB";Xa;""
7475            IF Xa>1 AND Xa<10 AND Xin=1 THEN PRINT "CP -1.25,-1;LB";Xa;""
7480            IF Xa>1 AND Xin=0 THEN PRINT "CP -2,-1;LB";Xa;""
7485            IF Xa=1 THEN PRINT "CP -1,-1;LB1.0"
7490        NEXT Xa
7495        PRINT "PU PA 0,0"
7500        Iht=2      ! MODIFY
7505        FOR Ya=Ymin TO Ymax STEP Ystep
7510            Y=(Ya-Ymin)*Sfy
7515            PRINT "PA 0,";Y,""
7520            IF Iht=0 AND Ya>0 THEN PRINT "PR 2,0"
7525            IF Ya<1 AND Ya<>0 THEN PRINT "CP -4,-.25;LB0;PR -2,0;LB";Ya;""
7530            IF Ya=0 THEN PRINT "CP -2,-.25;LB0"
7535            IF Ya>1 AND Iht<2 THEN PRINT "CP -5,-.25;LB";Ya;""
7540            IF Ya>9 AND Iht=2 THEN PRINT "CP -4,-.25;LB";Ya;""
7545            IF Ya>0 AND Ya<10 AND Iht=2 THEN PRINT "CP -3,-.25;LB";Ya;""
7550  !        IF Ya=1 THEN PRINT "CP -4,-.25;LB1.0"
7555        NEXT Ya
7560        IF Okd=0 THEN
7565            Xlabel$="X"
7570            Ylabel$="Y"
7575        ELSE
7580            BEEP
7585            INPUT "ENTER X-LABEL",Xlabel$
7590            BEEP
7595            INPUT "ENTER Y-LABEL",Ylabel$
7600        END IF
7605        PRINT "SR 1.5,2 PU PA 50,-10 CP";-LEN(Xlabel$)/2;"0;LB";Xlabel$;""
7610        PRINT "PA -11,50 CP 0,";-LEN(Ylabel$)/2*5/6;"DI 0,1;LB";Ylabel$;""
7615        PRINT "CP 0,0 DI"
7620        Nrun=0
7625        BEEP
7630        INPUT "WANT TO PLOT DATA FROM A FILE (1=Y,0=N)?",Ok
7635        Okp=0
7640        IF Ok=1 THEN
7645            BEEP
7650            INPUT "ENTER THE NAME OF THE PLOT DATA FILE",D_file$
7655            ASSIGN @File TO D_file$
7660            IF Icomb<>0 THEN 7690
7665            Sx=0
7670            Sy=0
7675            Sx2=0
```

209

```
7680        Sxy=0
7685        Md=1
7690        BEEP
7695        INPUT "ENTER THE BEGINNING RUN NUMBER (DEF=1)",Md
7700        Npairs=9
7705        BEEP
7710        INPUT "ENTER THE NUMBER OF X-Y PAIRS STORED (DEF=9)",Npairs
7715        BEEP
7720        INPUT "SELECT TUBE NUMBER (0=TOP,1=SECOND,.....)",Itube
7725        Nrun=Nrun+Npairs
7730        PRINTER IS 1
7735        BEEP
7740        PRINT USING "4X,""Select a symbol:"""
7745        PRINT USING "4X,"" 1  Star    2  Plus sign"""
7750        PRINT USING "4X,"" 3  Circle  4  Square"""
7755        PRINT USING "4X,"" 5  Rombus"""
7760        PRINT USING "4X,"" 6  Right-side-up triangle"""
7765        PRINT USING "4X,"" 7  Up-side-down triangle"""
7770        INPUT Sym
7775        PRINTER IS 705
7780        IF Sym=1 THEN PRINT "SM*"
7785        IF Sym=2 THEN PRINT "SM+"
7790   ——   IF Sym=3 THEN PRINT "SMo"
7795        Md=Itube*9
7800        IF Md>1 THEN
7805           FOR I=0 TO (Md-1)
7810              ENTER @File;X,Y
7815           NEXT I
7820        END IF
7825        FOR I=1 TO Npairs
7830           ENTER @File;Xa,Ya
7835           Sx=Sx+Xa
7840           Sy=Sy+Ya
7845           Sx2=Sx2+Xa^2
7850           Sxy=Sxy+Xa*Ya
7855           X=(Xa-Xmin)*Sfx
7860           Y=(Ya-Ymin)*Sfy
7865           IF Y>100 OR Y<0 THEN 7910
7870           IF Sym>3 THEN PRINT "SM"
7875           IF Sym<4 THEN PRINT "SR 1.4,2.4"
7880           PRINT "PA",X,Y,""
7885           IF Sym>3 THEN PRINT "SR 1.2,1.6"
7890           IF Sym=4 THEN PRINT "UC2,4,99,0,-8,-4,0,0,8,4,0;"
7895           IF Sym=5 THEN PRINT "UC3,0,99,-3,-6,-3,6,3,6,3,-6;"
7900           IF Sym=6 THEN PRINT "UC0,5.3,99,3,-8,-6,0,3,8;"
7905           IF Sym=7 THEN PRINT "UC0,-5.3,99,-3,8,6,0,-3,-8;"
7910        NEXT I
7915        BEEP
7920        INPUT "WANT TO LABEL (1=Y,0=N)?",Ilbl
7925        IF Ilbl=1 THEN
```

```
7930        IF Sym>3 THEN PRINT "SM"
7935        IF Sym<4 THEN PRINT "SR 1.4,2.4"
7940        PRINT "PA",Xal,Yal,""
7945        IF Sym>3 THEN PRINT "SR 1.2,1.6"
7950        IF Sym=4 THEN PRINT "UC2,4,99,0,-8,-4,0,0,8,4,0;"
7955        IF Sym=5 THEN PRINT "UC3,0,99,-3,-6,-3,6,3,6,3,-6;"
7960        IF Sym=6 THEN PRINT "UC0,5.3,99,3,-8,-6,0,3,8;"
7965        IF Sym=7 THEN PRINT "UC0,-5.3,99,-3,8,6,0,-3,-8;"
7970        PRINT "SM"
7975        IF Sym<4 THEN PRINT "PR 2,0"
7980        PRINT "PR 2,-1.0;SR 1.0,1.8;LB";D_file$;""
7985        Yal=Yal-5
7990        BEEP
7995        INPUT "WANT TO ADD ANOTHER STRING (1=Y,0=N)?",Ias
8000        IF Ias=1 THEN
8005           BEEP
8010           INPUT "ENTER THE STRING",Label$
8015           PRINT "PR 2,0;SR 1.0,1.8;LB";Label$;""
8020           GOTO 7990
8025        END IF
8030     END IF
8035     BEEP
8040     INPUT "WANT TO COMBINE ANOTHER FILE? (1=Y,0=N)",Icomb
8045     ASSIGN @File TO *
8050     IF Icomb<>0 THEN 7645
8055     Ils=1
8060     BEEP
8065     INPUT "WANT TO PLOT A LEAST-SQUARES LINE (1=DEF=YES,0=NO)",Ils
8070     IF Ils=1 THEN
8075        BEEP
8080        INPUT "SELECT CURVE TYPE (0=SOLID,1=DASHED)";Ilt
8085        Ilt=Ilt+1
8090        PRINT "SM"
8095        IF Iexp=0 THEN
8100           Bb=(Nrun*Sxy-Sy*Sx)/(Nrun*Sx2-Sx^2)
8105        ELSE
8110           Bb=.75
8115        END IF
8120        Aa=(Sy-Bb*Sx)/Nrun
8125        PRINTER IS 1
8130        PRINT USING "10X,""a = "",MZ.3DE";Aa
8135        PRINT USING "10X,""n = "",MZ.3DE";Bb
8140        PRINTER IS 705
8145        In=0
8150        FOR Xa=Xmin TO Xmax STEP (Xmax-Xmin)
8155           Ya=Aa+Xa*Bb
8160           Y=(Ya-Ymin)*Sfy
8165           X=(Xa-Xmin)*Sfx
8170           IF Y<0 THEN Y=0
8175           IF Y>100 THEN GOTO 8220
```

```
8180              IF Ilt=1 THEN
8185                  PRINT "PA",X,Y,"PD"
8190              ELSE
8195                  In=In+1
8200                  Ir=In MOD Ilt
8205                  IF Ir=1 THEN PRINT "PA",X,Y,"PD"
8210                  IF Ir=0 THEN PRINT "PA",X,Y,"PU"
8215              END IF
8220          NEXT Xa
8225          PRINT "PU"
8230      END IF
8235      Icomb=0
8240      GOTO 7620
8245  END IF
8250  PRINT "PU PA 0,0"
8255  PRINT "PU PA 0,0 SP0"
8260  SUBEND
8270  DEF FNFmcal(I,T,Pc)
8280  IF I=0 THEN
8290      Mdot0=.00206*Pc-.00414
8300      Mdot24=.00225*Pc-.00329
8320  END IF
8330  IF I=1 THEN
8340      Mdot0=.00204*Pc-.00615
8350      Mdot24=.00216*Pc-.00138
8370  END IF
8380  IF I=2 THEN
8390      Mdot0=.00208*Pc-.0036
8400      Mdot24=.00223*Pc-.00301
8420  END IF
8430  IF I=3 THEN
8440      Mdot0=.0028*Pc-.00122
8450      Mdot24=.0029*Pc+.0048
8460  END IF
8470  Mdt=(T*(Mdot24-Mdot0)/24)+Mdot0
8480  RETURN Mdt
8490  FNEND
```

LIST OF REFERENCES

1. Zebrowski, D.S., *Condensation Heat Transfer Measurements of Refrigerants on Externally Enhanced Surfaces* , M.S. Thesis, Naval Postgraduate School, Monterey, California, September 1987.

2. Mabrey, B.D., *Condensation of Refrigerants on Small Tube Bundles* , M.S. Thesis, Naval Postgraduate School, Monterey, California, December 1988.

3. Marto, P.J., *Heat Transfer in Condensation* , Boilers, Evaporators and Condensers, ed. by S. Kakal, John Wiley and Sons, pp. 525-570, 1991.

4. Nusselt, W., *The Condensation of Steam on Cooled Surfaces* , Z.d. Ver. Deut. Ing. , Vol. 60, pp. 541-546 and pp. 569-575, 1916. (Translated into English by F. Fullarton in Chem. Eng. Fund., Vol. 1, No. 2, pp. 6-19, 1982).

5. Goto, M., Hotta, H. and S. Tezuka, *Film Condensation of Refrigerant Vapors on a Horizontal Tube*, Int. J. of Refrig., Vol. 3, No. 3, pp. 161-166, 1980.

6. Masuda, H. and J.W. Rose, *An Experimental Study of Condensation of Refrigerant 113 on Low Integral-fin Tubes*, Proc. Int. Symp. on Heat Transfer, Beijing, PRC, Vol. 2, paper 32, Tsinghua Univ. Press, Oct. 1985.

7. Masuda, H., *Film Condensation Heat Transfer on Low Integral-Fin Tube*, Ph.D Thesis, Queen Mary College, University of London, July 1985.

8. Marto, P.J., Zebrowski, D., and A.S. Wanniarachchi, *An Experimental Study of R-113 Film Condensation on Horizontal Integral-Fin Tubes*, J. Heat Transfer, Vol. 112, pp. 758-767, 1990.

9. Marto, P.J., *An Evaluation of Film Condensation on Horizontal Integral-fin Tubes* , Journal of Heat Transfer, Vol.110, No.4B, pp.1287-1305, 1988.

10. Marto, P.J., *Recent Progress in Enhancing Film Condensation Heat Transfer on Horizontal Tubes*, Heat Trans. Engrg., Vol. 7, pp.61-71, 1986.

11. Webb, R.L., *Enhancement of Film Condensation*, Int. Comm. Heat Mass Transfer, Vol. 15, pp. 475-507, 1988.

12. Sukhatme, S.P., *Condensation on Enhanced Surface Horizontal Tubes*, Proc. 9th Int. Heat Transfer Conference, Vol. 1, pp. 305-327, 1990.

13. Beatty, K.D. and D.L. Katz, *Condensation of Vapors on Outside Finned Tubes* , Chem. Eng. Prog., Vol. 44, No. 1, pp. 55-70, 1948.

14. Rudy, T.M. and R.L. Webb, *Condensate Retention of Horizontal Integral-finned Tubing*, Advances in Enhanced Heat Transfer, HTD-Vol. 18, pp. 35-41, 20th National Heat Transfer Conference, Milwawkee, Wisconsin, August, 1981.

15. Webb, R.L., Keswani, S.T. and T.M. Rudy, *Investigation of Surface Tension and Gravity Effects in Film Condensation*, Proc. 7th Int. Heat Transfer Conf., Vol. 5, pp. 175-180, 1982.

16. Owen, R.G., Sardesai, R.G., Smith, R.A. and W.C. Lee, *Gravity Controlled Condensation on Low Integral-Fin Tubes*, Proc. Symp. on Condensers: Theory and Practice, I. Chem. E. Symp. Ser. No. 75, pp. 415-428, 1983.

17. Webb, R.L., Rudy, T.M. and M.A. Kedzierski, *Prediction of the Condensation Coefficient on Horizontal Integral-Fin Tubes*, J. Heat Transfer, Vol. 107, pp. 369-376, 1985.

18. Adamek, T., *Rechenmodell der Filmkondensation an engberippten Kondensatorrohren*, Warme-und Stoffubertragung, Vol. 19, pp. 145-157, 1985.

19. Honda, H. and S. Nozu, *A Prediction Method for Heat Transfer During Film Condensation on Horizontal Low Integral-Fin Tubes*, ASME J. Heat Transfer, Vol. 109, pp. 218-225, 1987.

20. Honda, H., Nozu, S. and B. Uchima, *A Generalized Prediction Method for Heat Transfer During Film Condensation on a Horizontal Low Finned Tube*, Proc. 2nd ASME-JSME Thermal Engineering Joint Conference, P.J. Marto and I. Tanasawa, eds., JSME, Vol. 4, pp. 385-392, 1987.

21. Katz, D.L., Hope, R.E., Datsko, S.C., and D.B. Robinson, *Condensation of Freon-12 with Finned Tubes, Part I. Single Horizontal Tubes*, J. Am. Soc. Refrigerating Engineers, Vol. 53, pp. 211-217, 1947.

22. Katz, D.L., Hope, R.E., Datsko, S.C., and D.B. Robinson, *Condensation of Freon-12 with Finned Tubes, Part II. Multitube Condensers*, J. Am. Soc. Refrigerating Engineers, Vol. 53, pp. 315-319, 352-354, 1947.

23. Karkhu, V.A. and V.P. Borovkov, *Film Condensation of Vapor at Finely Finned Horizontal Tubes*, Heat Transfer-Soviet Research, Vol. 3, pp. 183-191, 1971.

24. Arai, N., Fukushima, T., Arai, A., Nakajima, T., Fujie, K., and Y. Nakayama, *Heat Transfer Tubes Enhancing Boiling and Condensation in Heat Exchangers of a Refrigerating Machine*, Trans. ASHRAE, Vol. 83, pp. 58-70, 1977.

25. Kisaragi, t., Enya, S., Ochiai, J., Kuwahara, K., and I. Tanasawa, *On the Improvement of Condensation Heat Transfer on Horizontal Tubes*, JSME paper No. 780-1, pp. 1-5, 1978.

26. Carnavos, T.C., *An Experimental Study: Condensing R-11 on Augmented Tubes*, ASME Paper No. 80-HT-54, 1980.

27. Rudy, T.M., *A Theoretical and Experimental Study of Condensation on Single, Integral-Fin Tubes*, Ph.D. Thesis, The Pennsylvania State University, University Park, Pa, 1982.

28. Honda, H., Nozu, S. and K. Mitsumori, *Augmentation of Condensation on Horizontal Finned Tubes by Attaching a Porous Drainage Plate*, Proc. ASME-JSME Thermal Eng. Joint Conf., Vol. 3, pp. 289-296, 1983.

29. Kabov, O.A., *Film Condensation of Vapor on a Horizontal Finned Cylinder*, Heat Transfer-Soviet Research, Vol. 16, pp. 76-83, 1984.

30. Lin, C.C. and J. Berghmans, *R-11, R-113 and R-114 Condensation on an Enhanced Tube*, Proc. 4th Miami Int. Symp. on Multi-phase Transport and Particulate Phenomena, 15-17 December, 1986.

31. Sukhatme, S.P., Jagadish, B.S. and P. Prabhakaran, *Film Condensation of R-11 Vapor on SIngle Horizontal Enhanced Condenser Tubes*, J. Heat Transfer, Vol. 112, pp. 229-234, 1990.

32. Michael, A.G., Marto, P.J., Wanniarachchi, A.S. and J.W. Rose, *Filmwise Condensation of R-113 on Horizontal Integral-Fin Tubes of Different Diameters* , Proc. 9th Int. Heat Transfer Conf., Vol. 3, pp. 15-20, 1990.

33. Fujii, T., Wang, W. Ch., Koyama, Sh. and Y. Shimizu, *Heat Transfer Enhancement for Gravity Controlled Condensation on a Horizontal Tube by a Coiled Wire*, Proc. Int. Symp. on Heat Transfer, Beijing, China, 1985.

34. Kern, D.Q., *Mathematical Development of Loading in Horizontal Condensers* , Journal of the American Institute of Chem. Eng., Vol.4, pp. 157-160, 1958.

35. Eissenberg, D.M., *An Investigation of the Variables Affecting Steam Condensation of the Outside of a Horizontal Tube Bundle* , Ph.D. Thesis, Univ. of Tennessee, Knoxville, December, 1972.

36. Butterworth, D., *Developments in the Design of Shell and Tube Condensers* , ASME Winter Meeting, Atlanta, ASME Prepreint 77-WA/HT-24, 1977.

37. Young, F.L. and W.J. Wohlenberg, *Condensation of Saturated Freon-12 Vapor on a Bank of Horizontal Tubes*, Trans. ASME., Vol. 64, pp. 787-794, 1942.

38. Katz, D.L. and J.M. Geist, *Condensation on Six Finned Tubes in a Vertical Row*, Trans. ASME, Vol. 70, pp. 907-914, 1948.

216

39. Smirnov, G.F. and I.I. Lukanov, *Study of Heat Transfer from Freon-11 Condensing on a Bundle of Finned Tubes*, Heat Transfer-Soviet Research, Vol. 4, pp. 51-56, 1972.

40. Honda, H., S. Nozu and Y. Takeda, *Flow Characteristics of Condensate on a Vertical Column of Horizontal Low Finned Tubes*, Proc. 2nd ASME-JSME Thermal Eng. Joint Conf., Vol. 1, pp. 517-524, 1987.

41. Webb, R.L. and C.G. Murawski, *Row Effect for R-11 Condensation on Enhanced Tubes*, Trans. ASME, Vol. 112, pp. 768-776, 1990.

42. Honda, H., Uchima, B., Nozu, S., Nakata, H. and E. Torgoe, *Film Condensation of R-113 on In-line Bundles of Horizontal Finned Tubes*, J. Heat Transfer, Vol. 113, pp. 479-486, 1991.

43. Honda, H. and S. Nozu, *Effects of Bundle Depth and Working Fluid on the Optimal Fin Geometry of a Horizontal Low Finned Condenser Tube*, Proc. 2nd Int. Symp. Condensers and Condensation, HTFS, pp. 407-416, 1990.

44. Honda, H., Nozu, S. and Y. Takeda, *A Theoretical Model of Film Condensation in a Bundle of Horizontal Low Finned Tubes*, J. Heat Transfer, Vol. 111, pp. 525-532, 1989.

45. Murphy, T.J., *Pool Boiling of R-114/Oil Mixtures from Single Tubes and Tube Bundles*, M.S. Thesis, Naval Postgraduate School, Monterey, California, September 1987

46. Chilman, S.V., *Nucleate Boiling Characteristics of R-113 in a Small Enhanced Tube Bundle*, M.S. Thesis, Naval Postgraduate School, Monterey, California, September 1991.

47. Hong, S.W. and A.E. Bergles, *Augmentation of Laminar Flow Heat Transfer in Tubes by Means of Twisted Tape Insert*, J. Heat Transfer, Vol. 98, No. 2, 1976, pp. 251-256.

48. Saha, S.K., U.N. Gaitonoe and A.W. Date, *Heat Transfer and Pressure Drop Characteristics of Laminar Flow in a Circular Tube fitted with Regularly Spaced Twisted Tape Elements* , Experimental Heat Transfer, Fluid Mechanics and Thermodynamics, ed. by R.K. Shah, E.N. Ganic and K.T. Yang, Elsevier Science Publishing Company, pp. 511-518, 1988.

49. Hausen, H., *Darstellung des Wamneuberganges in Rohren durch verallgemeinerte Potenzbeziehungen* , VDI Z. , No. 4, p. 91, 1943.

50. Gough, M.J. and J.V. Rodgers, *Reduced Fouling by Enhanced Heat Transfer using Wire Matrix Radial Mixing Elements* , A. I. Ch. Eng., Vol. 83, No. 257, pp. 16-21, 1987.

51. Olvier, D.R. and R.W.J. Aldington, *Enhancement of Laminar Flow Heat Transfer using Wire MAtrix Turbulators* , 8th International Heat Transfer Conference, Vol. 6, pp. 2897-2902, 1986.

52. Oliver, D.R. and R.W.J. Aldington, *Heat Transfer Enhancement in Round Tubes using Wire Matrix Turbulators: Newtonian and Non-Newtonian Liquids* , Chem. Eng. Res. Des., Vol.66, pp. 555-565, 1988.

53. Gough, M.J., J.V. Rodgers, and G.M.B. Russell, *The Development of a Practical Answer to the Improvement of Tube-side Heat Transfer* , 12th HTFS Res. Symp., Univ. Warwick, United Kingdom, pp. 123-134, 1982.

54. Kline, S.J. and F.A. McClintock, *Describing Uncertainties in Single Sample Experiments*, Mechanical Engineering, Vol. 78, pp. 3-8, 1953.

55. Incropera, F.P. and D.P. DeWitt, *Introduction to Heat Transfer*, 2nd edition, published by John Wiley and Sons, New York, pp. 127-133, 1990.

56. Cragoe, C.S., *Properties of Ethylene Glycol and its Aqueous Solutions* , Cooperative Research Council, New York, Unpublished.

57. Gallant, R. W., *Ethylene Glycols* , Physical Properties of Hydrocarbons, Golf Publishing Co., Houston, Texas, Chapter 13, pp. 109-123, 1968.

58. Chapman, R.H., *Saturated Pool and Flow Boiling Studies with Freon-113 and Water at Atmospheric Pressure* , ORNL-4987, pp. 217-220, November, 1974.

INITIAL DISTRIBUTION LIST