Research Report 87-9J8-MATHP-R1

OAK RIDGE NATIONAL LABORATORY HEAT PUMP PERFORMANCE PREDICTION COMPUTER CODE ADDITIONS AND VERIFICATION

T. J. Fagan, R. A. Lucheta and D. T. Beecher

Heat Transfer & Fluid Dynamics

November 30, 1987

Westinghouse R&D Center
1310 Beulah Road
Pittsburgh, Pennsylvania 15235

Research Report 87-9J8-MATHP-R1

OAK RIDGE NATIONAL LABORATORY HEAT PUMP PERFORMANCE PREDICTION COMPUTER CODE ADDITIONS AND VERIFICATION

T. J. Fagan, R. A. Lucheta and D. T. Beecher

Heat Transfer & Fluid Dynamics

November 30, 1987

Westinghouse R&D Center
1310 Beulah Road
Pittsburgh, Pennsylvania 15235

ORNL/Sub/79-24712/4D

OAK RIDGE NATIONAL LABORATORY HEAT PUMP PERFORMANCE PREDICTION COMPUTER CODE ADDITIONS AND VERIFICATION

T. J. Fagan
R. A. Lucheta
D. T. Beecher

November 30, 1987

Prepared by

WESTINGHOUSE ELECTRIC CORPORATION
R&D CENTER
1310 Beulah Road
Pittsburgh, PA 15235

under
Subcontract 86X-24712C

for

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37831
operated by
MARTIN MARIETTA ENERGY SYSTEMS, INC.
for the
U.S. DEPARTMENT OF ENERGY
under Contract DE-AC05-84OR21400

ENGINEERING TECHNOLOGY DIVISION

ETD 2303

Report 87-9J8-MATHP-R1 Contract Required

OAK RIDGE NATIONAL LABORATORY HEAT PUMP PERFORMANCE PREDICTION COMPUTER CODE ADDITIONS AND VERIFICATION

Written by T. J. Fagan

T. J. Fagan
Heat Transfer & Fluid Dynamics

Written by R. A. Lucheta

R. A. Lucheta
Acoustic & Noise Control

Written by D. T. Beecher

D. T. Beecher
Heat Transfer & Fluid Dynamics

Approved by J. A. Ciesar

J. A. Ciesar, Manager
Mechanical Systems

Westinghouse Electric Corporation
Research and Development Center
1310 Beulah Road
Pittsburgh, Pennsylvania 15235

NOTICE

CONTENTS

LIST OF TABLES

LIST OF FIGURES

OAK RIDGE NATIONAL LABORATORY HEAT PUMP PERFORMANCE PREDICTION COMPUTER CODE ADDITIONS AND VERIFICATION

T. J. Fagan, R. A. Lucheta, and D. T. Beecher
Heat Transfer & Fluid Dynamics Department

ABSTRACT

Under contract to Oak Ridge National Laboratory, the ORNL heat pump performance prediction computer code was modified to include heat transfer, pressure drop, and internal leakage in refrigerant reversing valves, heat transfer and pressure drop in suction line accumulators with internal heat exchangers, and to account for the effects of the common corrugated fin pattern on air side heat transfer and pressure drop in air-to-refrigerant heat exchangers. Measured and predicted performance were compared for the Westinghouse/DOE advanced electric heat pump for heating and cooling operation at both capacity levels at several different ambient temperatures. This report describes the code modifications and discusses the results of the comparisons between predicted and measured performance.

1. INTRODUCTION

The design of vapor compression refrigeration systems is a highly iterative process with a large number of design variables. Consequently the application of computer aided engineering (CAE) techniques is highly attractive and considerable effort has been devoted toward the development of mathematical models for predicting the cost and performance of vapor compression refrigeration systems and components in both the private and public sectors.

One of the most widely used codes in the public sector was developed by the Oak Ridge National Laboratory.[1] To complement its in-house code development and verification effort, ORNL has contracted with the Westinghouse R&D Center to modify the code to include models of: reversing valve refrigerant heat transfer, pressure drop, and internal leakage; refrigerant heat transfer and pressure drop in suction line accumulators with internal heat exchangers; air side heat transfer and pressure drop effects in air-to-refrigerant heat exchangers due to the common corrugated fin pattern; and to also provide additional verification of the code by comparing program predictions with the extensive system performance data base established during testing of the Westinghouse/DOE advanced dual-stroke electric heat pump.

The reversing valve influences the capacity and energy efficiency of a heat pump in four basic ways:

- High and low pressure side refrigerant pressure drop.
- Internal heat transfer.
- Internal refrigerant leakage.
- External heat transfer.

Two basic types of suction line accumulators are applied to air-to-air residential heat pumps. The most common type, shown schematically in Figure 1-1, is essentially a low side receiver to prevent

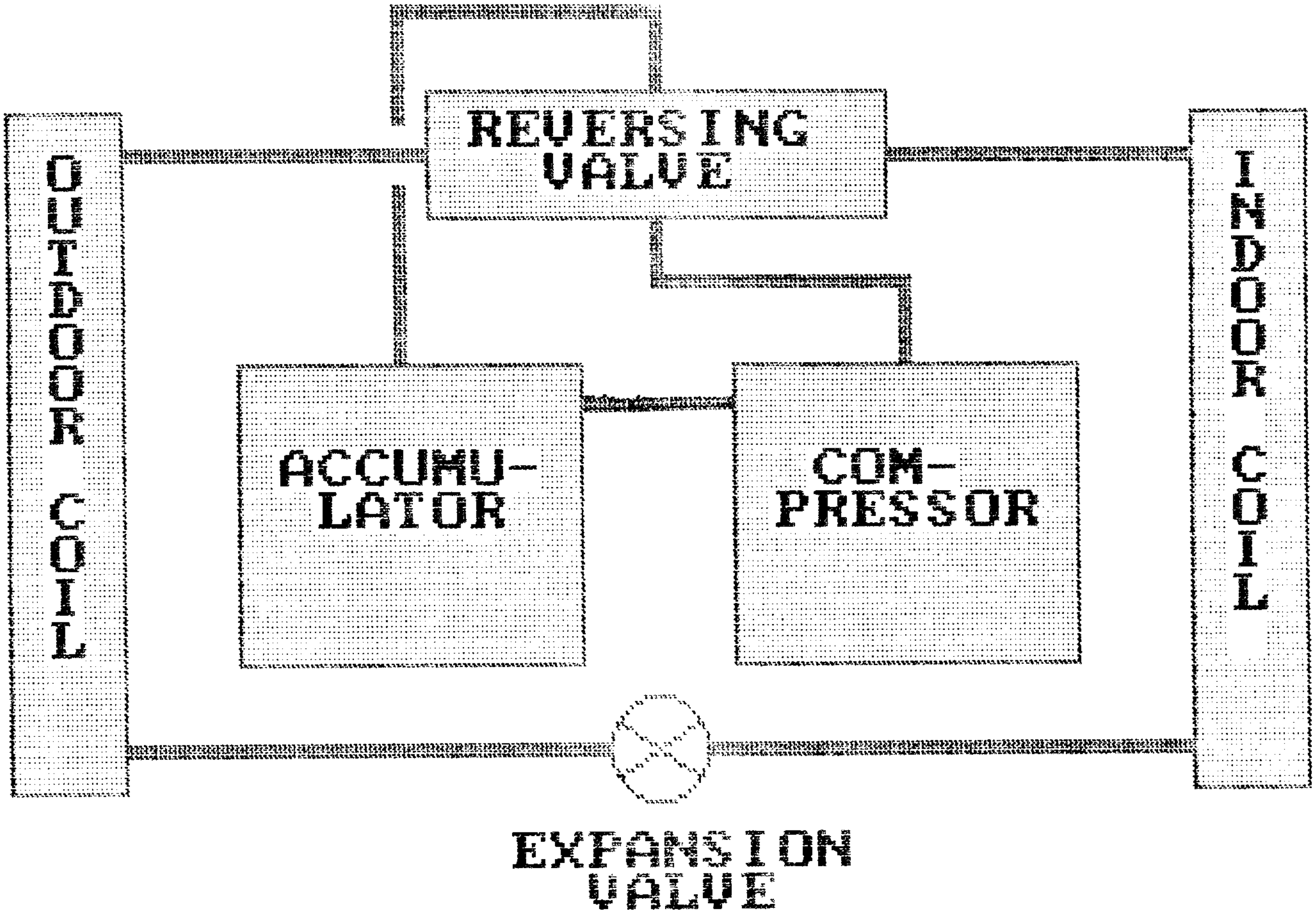


Figure 1-1 — Conventional heat pump circuit.

liquid slugging of the compressor during transients. Normally a venturi section in the discharge line of the accumulator is used to aspirate oil from the bottom of the accumulator and return it to the compressor. This type of accumulator influences system performance due to internal refrigerant pressure drop and heat transfer to ambient. The second type, with an internal heat exchanger, was applied commercially to the Westinghouse Hi-Re-Li heat pump line and was used in the Westinghouse/DOE advanced dual-stroke heat pump, as shown schematically in Figure 1-2. This configuration allows the evaporator to be operated in the "flooded" mode, with a small percentage of liquid in the refrigerant exiting the evaporator.

The warm high pressure liquid refrigerant exiting the condenser is passed through a coil of tubing near the bottom of the accumulator to provide a heat source for vaporizing the liquid refrigerant entering the accumulator from the evaporator to prevent liquid slugging of the compressor. This mode of operation improves system performance since evaporator capacity is maximized by eliminating the superheating section where tube side convection coefficients are relatively low and the refrigerant-to-air temperature difference is rapidly declining and by maximizing the refrigerant pumping capacity of the compressor by maximizing the density of the suction gas. The loss in refrigerant enthalpy change associated with vaporization of the remaining liquid in the evaporator exit stream is compensated by a reduction in the quality of the refrigerant entering the evaporator due to additional subcooling of the high pressure liquid stream in the accumulator heat exchanger. While the internal heat exchanger is not currently used in commercially available residential equipment, it was necessary to incorporate a model in the code to support the verification phase of the project.

The most common form of air-to-refrigerant heat exchanger in residential equipment is the plate-finned tube configuration. Typically some type of patterned fin is used which enhances air side heat transfer via a "spoiler" mechanism in which the boundary layer is periodically separated from the fin surface, taking advantage of the higher convection coefficients associated with developing flow. This increase in air

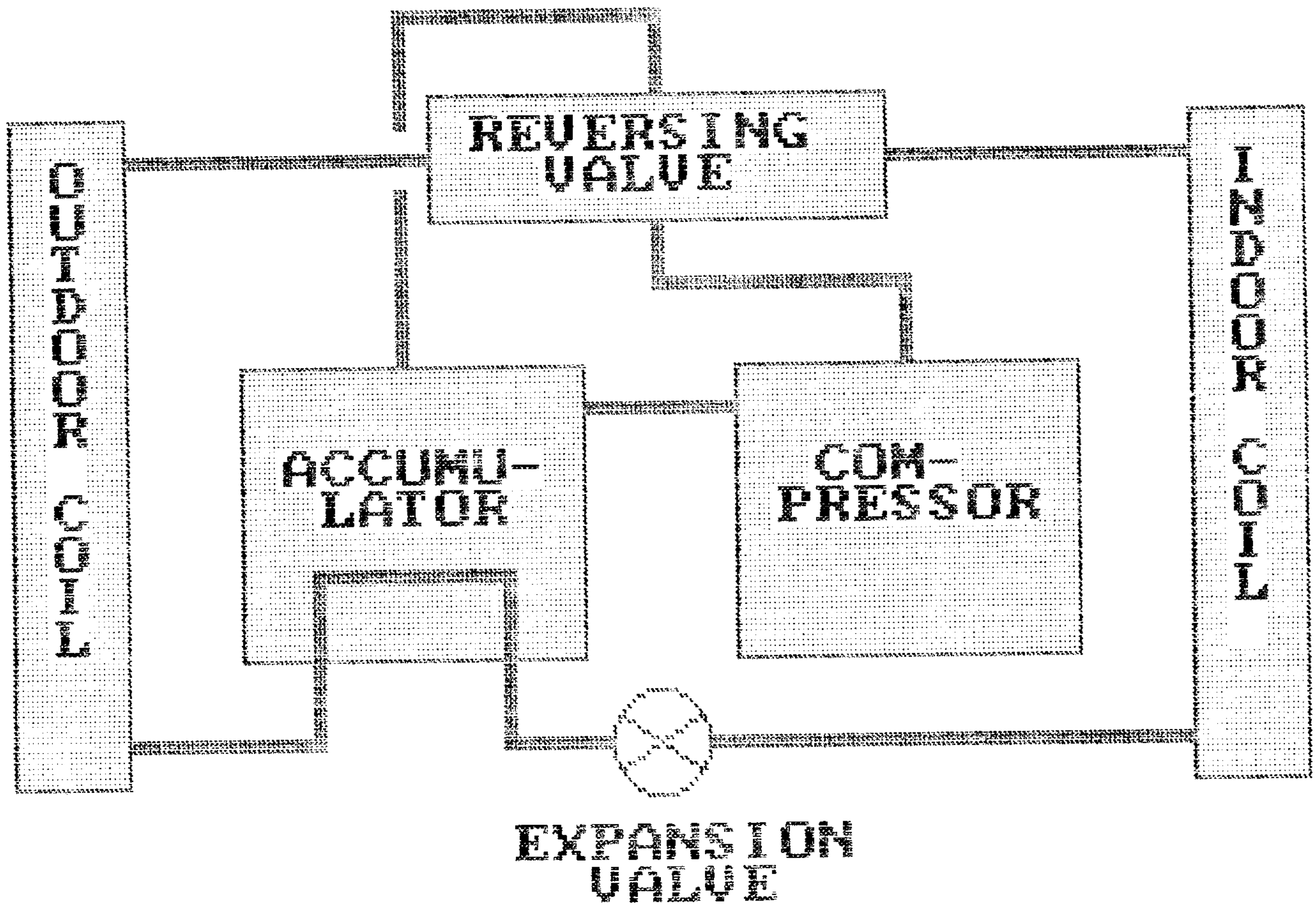


Figure 1-2 — Westinghouse/DOE advanced heat pump circuit.

side heat transfer is inevitably obtained at the expense of increased air side pressure drop and air moving power. The ORNL code currently incorporates a correlation developed by McQuiston[2] for unpatterned fins and applies a constant multiplier to the air side convection coefficient for fins with a wavy or louvered surface, based on the data published by Yoshii,[3] Senshu,[4] and Kirschbaum.[5] However, Westinghouse experience has shown that the level of heat transfer augmentation and the increase in air side pressure drop due to given fin pattern type are not constant and are a function of the pattern geometry, heat exchanger configuration, air velocity, and air inlet conditions. Consequently it was desirable to develop a correlation for the effects of these parameters on the air side heat transfer and pressure drop performance for the fin pattern most commonly used by Westinghouse manufacturing divisions, a simple corrugated fin pattern. The Westinghouse correlation is based on single fin model tests of a "V" type corrugated pattern consisting of short straight sections connected by sharp bends rather than a sinousoidal pattern. A typical configuration is shown in Figure 1-3.

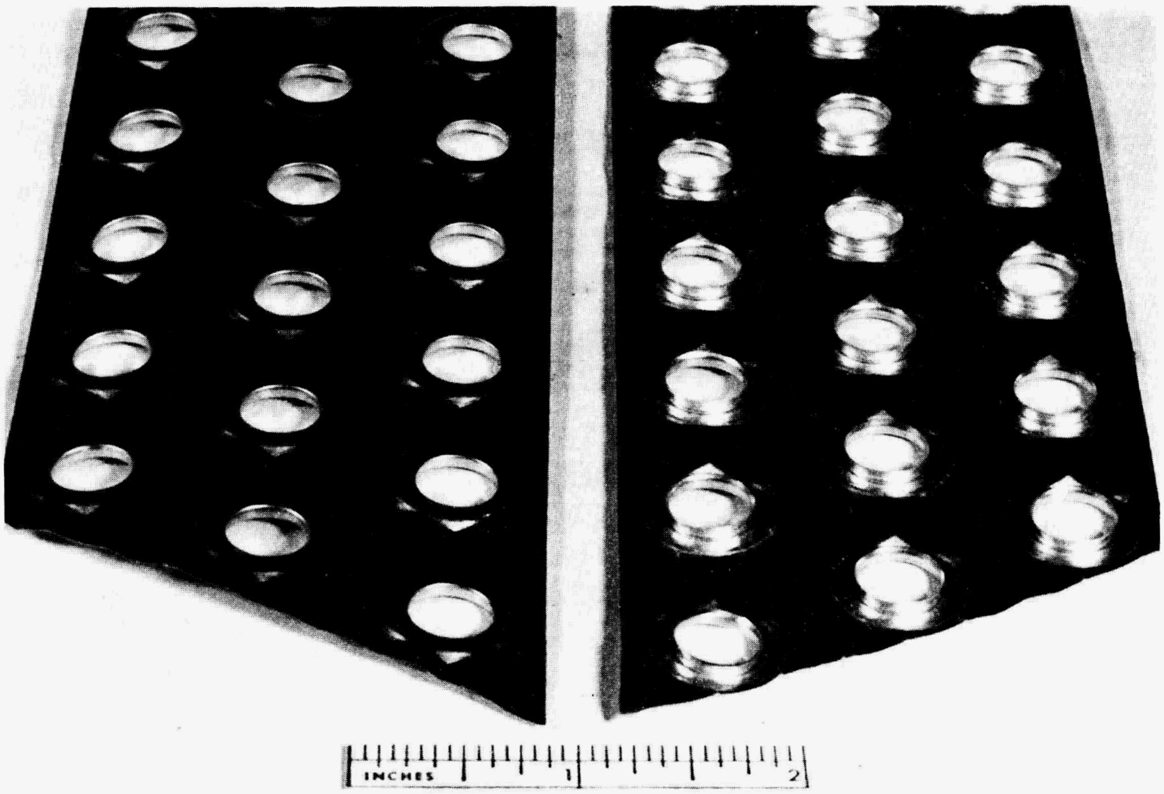


Figure 1-3. Typical fin surface pattern.

2. REVERSING VALVE MODELS

The electrically operated refrigerant reversing valve provides a convenient and reliable means of allowing an air-to-air heat pump to alternate between cooling and heating service and is a key element in the reverse cycle defrost process. However, the presence of the valve results in a decrease in both the capacity and energy efficiency of the system due to internal and external heat transfer, refrigerant pressure drop, and internal refrigerant leakage. There is some controversy in the HVAC industry as to the relative importance of these losses.

The highly superheated high pressure refrigerant gas stream passing from the compressor to the condenser is separated from the low pressure refrigerant flowing from the evaporator to the accumulator, which may be a two-phase mixture, saturated vapor or superheated vapor, by a thin metal or plastic slide. Since a significant temperature difference exists, heat is transferred, cooling the high pressure stream and increasing the temperature or quality of the low pressure stream. Normally the valve body is hotter than the ambient air and heat is transferred to the air by natural convection or, if the valve is located in the evaporator or condenser air stream, by forced convection. In conventional systems, where the refrigerant exiting the evaporator is superheated, the addition of heat to the low pressure refrigerant stream reduces capacity. Heating the refrigerant vapor reduces its density and, since the reciprocating compressor is basically a constant volume flow device, reduces the refrigerant mass flow rate. In flooded evaporator systems such as the Westinghouse Hi-Re-Li, there is no loss in mass flow, but evaporator capacity is slightly reduced since the amount of sub-cooling of the high pressure liquid stream exiting the accumulator heat exchanger is reduced, resulting in an increase in the quality entering the evaporator and consequently a decrease in the maximum quality change. This obviously reduces cooling capacity.

The loss of heat from the high pressure gas stream obviously reduces capacity in heating service. However, this loss is partially offset by the reduction in required evaporator capacity due to the increased inlet quality. For a given evaporator heat transfer area the evaporating temperature will tend to increase, resulting in a decrease in the pressure ratio across the compressor and a slight increase in refrigerant mass flow. Similarly in cooling service the amount of condenser capacity required is reduced, resulting in a small decrease in condenser pressure and a slightly higher refrigerant mass flow rate.

Pressure drop occurs in both the high and low pressure refrigerant streams as they pass through the reversing valve due to a series of contractions, bends and expansions. Refrigerant pressure drop penalizes performance in several ways. Reducing the pressure of the compressor suction gas reduces its density resulting in a decrease in refrigerant mass flow rate. Since the pressure in the condenser is primarily a function of the amount of heat to be rejected, pressure drop between the compressor and condenser will increase the required compressor discharge pressure, reducing the refrigerant mass flow rate and increasing compressor power consumption.

Some form of internal sliding seal is present in all conventional reversing valves. Since no seal of this type is perfect and a significant pressure difference exists, leakage will occur. Most valve manufacturers claim that a leakage rate of 1% to 2% of the total refrigerant mass flow is typical. Our own tests[6] showed that, after repeated cycling, wear of the sealing surfaces can result in leakage rates of 10% or more. Valves from four different vendors were subjected to repeated cycling. Two valves from Manufacturer A showed excessive leakage after 13,000 cycles. A valve from Manufacturer C showed a sharp increase in leakage between 80,000 and 100,000 cycles. A valve provided by Manufacturer B showed a gradual increase in leakage rate, but remained within the manufacturer's specifications after 100,000 cycles. The leakage rate of Valve D, which replaced Valve A after 67,000 cycles, was acceptable after 33,000 cycles when the tests were terminated. The 2-year field test of the advanced electric heat pump[7] showed that in a DOE region IV location the heat pump will experience between 200 and 300 defrosts and approximately 3000 on/off cycles during cooling operation per year. Since the reversing valve was de-energized during heating service and energized during defrost and cooling service, the sum of the number of defrosts and the number of on/off cycles in cooling service equals the number of cycles of the reversing valve. Therefore the life of Valve A would be less than four years. Valve B would have a life of more than 30 years, Valve C would last more than 24 years, and Valve D would have a life of at least 10 years, assuming that cycling is the controlling parameter in valve life. Deterioration of elastomeric sealing materials with age may also be an important factor in determining valve life.

Internal leakage reduces system capacity since the refrigerant flow through the evaporator and condenser are reduced. The hot high pressure gas mixes with the low pressure gas entering the accumulator, and therefore the compressor, reducing its density and consequently the compressor mass flow rate.

2.1 ANALYTICAL HEAT TRANSFER AND PRESSURE DROP MODEL

The primary reversing valve heat transfer and pressure drop model was developed in 1978 at the Westinghouse Fluid Systems Laboratory, formerly located in West Lafayette, Indiana, as part of our in-house program to develop computer codes for analyzing and predicting the performance of HVAC components and systems. This model, described in detail in Appendix A, is based primarily on first principles, with manufacturer's data used to fix the values of some parameters. In its original form, the input parameters were:

- Refrigerant mass flow rate.
- Compressor discharge temperature and pressure.
- Evaporator exit pressure and quality or superheat.
- Valve body ambient air temperature.
- Valve model identifier.

Output parameters included:

- Condenser port refrigerant temperature and pressure.
- Accumulator port refrigerant pressure and quality or superheat.
- Internal heat transfer.
- Heat loss to ambient.

The heat loss to the ambient air was based on natural convection from the valve body.

Since the ORNL code is based on fixed compressor suction superheat and the Westinghouse CRACOP code is based on fixed evaporator exit superheat, it was necessary to alter the code logic to calculate the evaporator port conditions with a fixed set of accumulator port conditions. In addition, it was necessary to substitute the ORNL subroutines for refrigerant thermodynamic and transport properties for the Westinghouse subroutines. A listing of the final version of the subroutine, called VALVER, is presented in Appendix B. In addition to the internal VALVER analytical models for pressure drop and heat transfer, the VALVER subroutine can call a different set of pressure drop and heat transfer subroutines derived empirically from test data from the breadboard version of the advanced electric heat pump. Since the reversing valve used in the breadboard heat pump was from a different manufacturer than the valves used as the basis for the primary model and detailed loss data was not available, this valve could not be directly incorporated into the VALVER analytical model. Note that the preprototype used a reversing valve from a third manufacturer. A description of the empirical model is presented in Section 2.2.

The calling sequence for the VALVER subroutine is:

```
VALVER (IV, PA, TA, XA, PCP, TCP, MDOT, TAMB,
        PDLO, PDHI, QLEAK, QAMB, PRINT)
```

Input parameters include:

IV = Valve Identifier
- = 0 no valve
- = 1 empirical breadboard model
- = 25 Ranco #25 - analytical model
- = 26 Ranco #26 - analytical model
- = 30 Ranco #30 - analytical model

PA = Compressor suction pressure (psia)

TA = Compressor suction temperature (°F)

XA = Compressor suction quality (1.0 = 100%)

PCP = Compressor discharge pressure (psia)

TCP = Compressor discharge temperature (°F)

MDOT = Refrigerant mass flow rate (lbm/hr)

TAMB = Valve ambient air temperature (°F)

PRINT = Logical switch for diagnostic printout
- "F" = no printout
- "T" = printout provided

Output parameters include:

PDLO = Pressure drop - low pressure side (psi)

PDHI = Pressure drop - high pressure side (psi)

QLEAK = Internal heat transfer rate (Btu/hr)

QAMB = Heat transfer rate to ambient air (Btu/hr)

2.2 EMPIRICAL HEAT TRANSFER AND PRESSURE DROP MODEL FOR BREADBOARD UNIT

One of the major objectives of the design, construction, and testing of the breadboard version of the advanced electric heat pump was to verify the CRACOP computer code. A comparison of measured and predicted performance showed that the pressure drop and heat transfer associated with the reversing valve had a significant influence on system performance. It is important to note that while the breadboard unit used the same basic refrigerant circuit as the dual-stroke preprototype unit, the compressor, line sizes and lengths, line couplings, and the reversing valve used in the breadboard unit were different from those used in the preprototype. The Alco reversing valve used in the breadboard unit was of a completely different configuration than the Product Engineering valve used in the preprototype unit and had much higher refrigerant pressure drop. The Alco reversing valve used in the breadboard unit was of a similar configuration but from a different manufacturer than the Ranco valves incorporated into the VALVER code. Since manufacturers' data were not available for the Alco valve, an empirical model was developed from the test data. It was assumed that the significant parameters were compressor suction side pressure drop and internal heat transfer. Compressor discharge side pressure drop and heat transfer to ambient were not considered. Compressor discharge side pressure drop was neglected since we considered suction side pressure drop to be the dominant effect. In theory the refrigerant mass flow rate of a reciprocating compressor is a function of the displacement, clearance volume, pressure ratio, and suction gas density. Pressure drop on the compressor suction side affects both the pressure ratio and the density of the suction gas. Pressure drop on the discharge side of the compressor affects only the pressure ratio. Furthermore, pressure drop on the suction side has a greater effect on the pressure ratio since a given pressure difference represents a larger percentage of the absolute pressure on the low pressure side.

The correlating parameters used were quite simple. The suction side pressure drop was assumed to be a function of the velocity head of the gas entering the reversing valve:

$$Pd_v = A1(M_r^{\,2}/Rho_s)^{A2} \tag{1}$$

where:

Pd_v = Reversing valve low side pressure drop (psi)
M_r = Refrigerant mass flow rate (lbm/hr)
Rho_s = Reversing valve low side refrigerant inlet density (lbm/cu ft)

The internal heat transfer rate was assumed to be a linear function of the temperature difference between the compressor discharge gas and the accumulator exit gas:

$$Q_{val} = A3 + A4(T_{co} - T_{ao}) \tag{2}$$

where:

Q_{val} = Reversing valve internal heat leak (Btu/hr)
T_{co} = Compressor exit refrigerant temperature (°F)
T_{ao} = Accumulator low side refrigerant exit temperature (°F)

The constants A1 through A4 were evaluated by a conventional least mean squares curve fitting technique applied to the data obtained for the breadboard version of the advanced electric heat pump.

The instrumentation on the breadboard unit was not complete enough to permit separation of the pressure drop in the connecting lines and refrigerant couplings from the reversing valve pressure drop. Therefore, <u>the coefficients obtained include line and coupling pressure drops and all line lengths should be set to zero in the ORNL model when this model is used, except XLRVIC, the equivalent line length of the vapor line between the reversing valve and the indoor coil, and XLRVOC, the equivalent length of the vapor line between the reversing valve and the outdoor coil</u>. The length of the shorter of these two lines should be set to zero and the length of the longer line to a small value, say 1.0 inch. This is necessary since the relative lengths of these lines are used by the MAIN program to determine if the compressor is located in the indoor unit or the outdoor unit. This is required to compensate the heating or cooling capacity for the heat lost from the compressor can.

Two different values of A1 and A2 were obtained for heating and cooling service. In the breadboard unit the compressor was located in the indoor unit. In cooling service the vapor line connecting the evaporator exit and the reversing valve was short and did not contain couplings. Therefore the pressure drop between the evaporator exit and the compressor suction is much lower than in heating service where the full length of the line connecting the indoor and outdoor units plus the line couplings separate the evaporator outlet from the accumulator inlet. The coefficients obtained were:

Heating Service: A1 = 16.977202, A2 = 0.16876337

Cooling Service: A1 = 5.9160745, A2 = 0.0763784

<u>For systems where the compressor is located in the outdoor unit the coefficients must be reversed</u>, since the greater pressure drop will occur during cooling service, when the evaporator exit and accumulator inlet are separated by the lines connecting the indoor and outdoor units and by the line couplings. To identify the reversing valve location, the equivalent lengths of the line connecting the evaporator and the reversing valve XLEQSL, (XLRVIC cooling, XLRVOC heating) and the line connecting the reversing valve to the condenser XLEQDL (XLRVOC cooling, XLRVIC heating) are compared. If the unit is in the cooling mode and XLEQDL is greater than XLEQSL, the compressor is assumed to be located indoors. If the unit is in the heating mode and XLEQSL is greater than XLEQDL, the compressor is assumed to be indoors. If the lengths are equal or zero, an outdoor compressor location is the default value.

The values obtained for the heat transfer equation coefficients were A3 = -105.64513 and A4 = 8.721986. The same coefficients are used for heating and cooling. No distinction is made with respect to compressor location.

The empirical pressure drop model is incorporated in the function REVVPD and the empirical heat transfer model is the basis for the function REVVHT. Both functions are called by the VALVER subroutine if IV is set equal to 1 and are listed in Appendix C.

The calling sequence for function REVVPD is:

REVVPD (TREF, PREF, REFM)

The inputs include:

TREF = Evaporator exit temperature (°F)
PREF = Evaporator exit pressure (psia)
REFM = Refrigerant mass flow rate (lbm/hr)

The output is:

REVVPD = Low pressure side pressure drop (psi)

The calling sequence for function REVVHT is:

REVVHT (TCX, TVX)

Inputs include:

TCX = Compressor discharge temperature (°F)
TVX = Evaporator discharge temperature (°F)

The output is:

REVVHT= Reversing valve internal heat transfer rate (Btu/hr)

2.3 REVERSING VALVE REFRIGERANT LEAKAGE MODEL

The effects of internal refrigerant leakage in the reversing valve are modeled using basic first and second law thermodynamic relationships. The leakage rate is assumed to be a fixed percentage of the compressor mass flow rate. The inputs to the model include: the refrigerant mass flow rate of the compressor; the compressor suction temperature, pressure and enthalpy; and the compressor discharge temperature, pressure, and enthalpy. The reversing valve inlet temperature and enthalpy are then calculated assuming a simple isenthalpic mixing process.

The function FLKRV determines the refrigerant leakage rate. The mixing process is analyzed in the COMPV subroutine if the efficiency and loss based compressor model is specified, and is analyzed in the subroutine COMPMP if the map based compressor model is specified.

The FLKRV function was configured to easily accept more complex leakage models if desired. For example, one could assume that the leakage path is a narrow slot of arbitrary length, width, and height, and then calculate the mass flow rate through the slot using the basic principles of fluid mechanics. For incompressible laminar flow between parallel plates the flow rate is given by:

$$M_{leak} = \frac{2\ P_{dif}\ Rho_a\ g_c b^3 W}{3\ \mu\ L} \tag{3}$$

where:

M_{leak} = Refrigerant leak rate (lbm/sec)
P_{dif} = Pressure difference (lbf/sq ft)
Rho_a = Average refrigerant density (lbm/cu ft)
g_c = Gravitational acceleration (32.174 ft-lbm/lbf-sec^2)
b = Plate spacing (ft)
W = Plate width (ft)
μ = Refrigerant absolute viscosity (lbm/ft-sec)
L = Plate flow length (ft)

The parameters required by the particular leakage model are transmitted via the COMMON block RVLKP. The integer parameter NRVLK selects the type of leakage model; the constant parameters appropriate to the model are contained in the array RVLKP, which is dimensioned to accept up to five parameters. The actual number of parameters to be used for RVLKP for each value of NRVLK is set by the array NLKTB(NRVLK+1), which is filled in the BLOCK DATA RLADP. At present the only valid values of NRVLK are 0, no leakage, and 1, a fixed percentage of the compressor mass flow rate. If NRVLK is 0, NLKTB(1) is 0 — that is, since there is no leakage, no values of RVLKP are required by the leakage model. Note that a value of 0 for NRVLK is entered to DATAIN simply by placing a blank line at the appropriate position in the input data file. If NRVLK is 1, NLKTB(2) is 1; a single value of RVLKP(1) — specifying the leakage fraction — is entered via the DATAIN subroutine.

The calling sequence of the function FLKRV is:

FLKRV (NRVLK, NLPMD, RVLKP, RFLHP, PRFHD, TRFHD,
HRFHD, PRFLD, TRFLD, HRFLD)

The input parameters are:

NRVLK = Switch to identify reversing valve leakage model type (integer)
= 0 no leakage
= 1 fixed fraction of compressor mass flow

NLPDM = Number of parameters in the leakage model [equal to NLKTB(NRVLK+1)]
= 0 if NRVLK = 0
= 1 if NRVLK = 1

RVLKP = Array of parameters of the leakage model (must be specified, but not used if NRVLK = 0)
RVLKP(1) = leak rate as a fraction of the compressor mass flow rate if NRVLK = 1

RFLHD = Compressor mass flow rate (lbm/hr)

PRFHD = Compressor discharge pressure (psia) (reversing valve high side inlet) (must be specified, but not used for NRVLK = 0 or 1)

TRFHD = Compressor discharge temperature (°F) (reversing valve high side inlet) (must be specified, but not used for NRVLK = 0 or 1)

HRFHD = Compressor discharge enthalpy (Btu/lbm) (reversing valve high side inlet) (must be specified, but not used for NRVLK = 0 or 1)

PRFLD = Compressor suction pressure (psia) (reversing valve low side outlet) (must be specified, but not used for NRVLK = 0 or 1)

TRFLD = Compressor suction temperature (°F) (reversing valve low side outlet) (must be specified, but not used for NRVLK = 0 or 1)

HRFLD = Compressor suction enthalpy (Btu/lbm)
(reversing valve low side outlet)
(must be specified, but not used for NRVLK = 0 or 1)

The output is:

FLKRV = Reversing valve internal refrigerant leak rate (lbm/hr)

A more complete discussion of the reversing valve internal refrigerant leakage model and a listing of the function FLKRV are contained in Appendix D.

3. SUCTION LINE ACCUMULATOR MODEL

The refrigerant pressure drops and internal heat transfer rate associated with a suction line accumulator with an internal heat exchanger have been modeled using basic principles of heat transfer and fluid mechanics. Inputs to the model include the dimensions of critical components of the accumulator, the refrigerant mass flow rate, compressor suction temperature, pressure and quality, and the temperature and pressure of the high pressure liquid stream entering the accumulator heat exchanger. The model determines the temperature, pressure, and quality of the low pressure refrigerant stream entering the accumulator, the temperature and pressure of the high pressure liquid refrigerant stream leaving the accumulator heat exchanger and the internal heat transfer rate. This model will be referred to as the "physical model."

A simplified model was also developed, which ignores all details of accumulator construction. This model permits direct specification of accumulator exit superheat, high and low side pressure drops, and heat gain/loss between the accumulator and the ambient. Limitations on funding prevented testing this model, other than checking the syntactical correctness of the program. Because of the highly structured form of the program, we are confident that it will run properly. This model will be referred to as the "performance model."

The energy balancing portion of the model was incorporated into the COMPV and COMPMP subroutines for the efficiency and loss based compressor model, and for the map based compressor model, respectively. The subroutine DHSUL is used to determine the high and low side refrigerant pressure drops in the accumulator and the high and low side refrigerant enthalpy changes using estimated values of low pressure side accumulator inlet pressure and quality provided by COMPV or COMPMP. The COMPV or COMPMP subroutines iterate to find the low pressure inlet conditions to the accumulator which will satisfy the requirement for a balance between the high and low pressure side refrigerant enthalpy change. The parameters required by the model are contained in the COMMON blocks ACPAR and ACTBL.

The model has been constructed to simplify the incorporation of more detailed representations if desired. The model type is specified by the integer parameter NACCU. The fixed parameters are contained in the array ACPAR which is currently dimensioned to accept up to 20 accumulator-specifying parameters. The number of parameters to be used is determined by the array NACTB(NACCU+1) which is dimensioned to accept up to 12 values, corresponding to 12 distinct accumulator models. The values of NACTB are set in BLOCK DATA RLADP. At present three values of NACCU are valid: 0 for no suction line accumulator, 1 for the physical model, and 2 for the performance model. If NACCU is 0, the array ACPAR is not used and no inputs are required. If NACCU is 1, ACPAR requires five geometric parameters, read as data in the DATAIN subroutine. If NACCU is 2, ACPAR requires four performance parameters, read as data by the DATAIN subprogram. A 0 value of NACCU can be entered simply by placing a blank line of data in the appropriate position in the input data file.

The calling sequence for subroutine DHSUL is:

```
DHSUL (NACCU, ACPAR, NACDM, NCORH, PLODM, TLODM, XLODM,
       XLIDM, PHIDM, THIDM, XHIDM, RFLAD, DHLOD, DHHID,
       DPLOD, DPHID)
```

The input parameters are:

NACCU = Switch to identify accumulator model
= 0 no accumulator;
= 1 accumulator with internal heat exchanger;
= 2 accumulator with specified superheat and pressure drop;

ACPAR = Array of dimensional parameters
(required, but not used if NACCU = 0);
If NACCU = 1, as follows:
ACPAR(1) = inside diameter of internal heat exchanger tubing (ft)

ACPAR(2) = outside diameter of internal heat
exchanger tubing (ft)

ACPAR(3) = length of internal heat exchanger tubing (ft)

ACPAR(4) = inside diameter of vapor line (ft)

ACPAR(5) = total length of vapor line (ft)
(inlet + outlet)

If NACCU = 2, as follows:

ACPAR(1) = refrigerant superheat leaving the
accumulator on the suction side (°F)

ACPAR(2) = pressured drop on the low pressure
side of the accumulator (psia)

ACPAR(3) = pressure drop on the high pressure
side of the accumulator (psia)

ACPAR(4) = heat flow from the accumulator shell (Btu/sec)

NACDM = Number of values of NACCU required
= 0 if NACCU = 0
= 5 if NACCU = 1
= 4 if NACCU = 2
(equivalent to NACTB(NACCU+1) in BLOCK DATA RLAPD)

NCORH = Switch to identify operating mode
= 1 cooling mode
= 2 heating mode

PLODM = Low pressure side exit pressure (psia)

TLODM = Low pressure side exit temperature (°F)

XLODM = Low pressure side exit quality (1.0 = 100%)

XLIDM = Estimated low pressure side inlet quality (1.0 = 100%)

PHIDM = High pressure side inlet pressure (psia)

THIDM = High pressure side inlet temperature (°F)

XHIDM = High pressure side inlet quality (1.0 = 100%)

RFLAD = Refrigerant mass flow rate (lbm/hr)

Output parameters include:

DHLOD = Low pressure side refrigerant enthalpy change (Btu/lbm)

DHHID = High pressure side refrigerant enthalpy change (Btu/lbm)

DPLOD = Low pressure side refrigerant pressure drop (psi)

DPHID = High pressure side refrigerant pressure drop (psi)

A more complete discussion of the suction line accumulator model and a listing of the DHSUL subroutine are included in Appendix E.

4. REFRIGERANT THERMODYNAMIC PROPERTIES

The TRIAL subroutine performs an iterative calculation to determine the thermodynamic properties of superheated R22 vapor given the pressure and one additional property, either specific volume, enthalpy or entropy. When this subroutine was called by VALVER, it failed to converge in many cases with small amounts of superheat. In an attempt to circumvent this problem a modified version of TRIAL called TRIAL2 was developed which uses a different iteration scheme. In both versions of the subroutine the refrigerant temperature is the unknown parameter. In TRIAL, a first estimate of the temperature T and a temperature step size DT are input in addition to the known property value ARG. The temperature is then set equal to T + DT and a trial value of the known property ARGN is determined by calling the VAPOR subroutine. If DT is negative, the difference between the input and calculated arguments DIFF is set to ARG - ARGN. If DT is positive, then DIFF = ARGN - ARG. The absolute value of DIFF is compared to the convergence tolerance limit TOL. If DIFF is less than or equal to TOL, the iteration is considered complete. If DIFF is greater than TOL, the temperature estimate is set to T - DT, DT is set equal to DT/2 and the process is repeated. If the required accuracy is not achieved in forty iterations, the subroutine is terminated with an error message. However, the calling program will continue to run with the final estimate of vapor temperature. If the estimated value of T was initially set higher than the required temperature, the subroutine will not converge since no provision is made for changing the sign of DT. Therefore if the amount of superheat is small, the first guess on T should be set equal to the saturation temperature, or a small increment above the saturation temperature. If DT is too large or too small, the subroutine may fail to converge in forty iterations.

In TRIAL2, the initial estimate of T is used to calculate the first value of ARGN and DIFF is set equal to ARGN - ARG. If the absolute value of DIFF is less than or equal to TOL the iteration is consider complete. If the absolute value of DIFF is greater than TOL and DIFF is negative, the flag ITEST is set equal to -1 and the temperature estimate is modified by T + DT. If DIFF is positive, ITEST is set to 1.0 and the temperature estimate is changed to T - DT. If the sign of ITEST is different than the previous value, then DT is set equal to DT/4. If the revised estimate of the temperature is less than the saturation temperature corresponding to the input pressure, it is set equal to the saturation temperature. If convergence is not obtained in 200 iterations an error message is output and the program proceeds using the final temperature estimate.

TRIAL2 gave far fewer convergence errors than TRIAL, but was not "bullet proof." How much of the improvement can be attributed to the altered iteration scheme and how much to the increase in the allowed number of iterations is a moot point.

The calling sequence for TRIAL2 is the same as that of TRIAL. A listing of TRIAL2 is included in Appendix G.

5. FIN PATTERNATION EFFECTS

Correlations for predicting the increases in air side convection coefficients and air side pressure drop due to the common "V" type corrugated fin pattern in plate-finned tube air-to-refrigerant heat exchangers have been incorporated into the ORNL code. These correlations are based on single fin model tests and predict the ratio of the air side heat transfer coefficient and air side pressure drop of a patterned fin to that of a flat fin with the same tube geometry and fin spacing. These correlations are called only when the evaporator fin type identifier FINTYE and/or the condenser fin type identifier FINTYC are set equal to 4.0. In the modified version of the ORNL heat pump model the multiplying factors predicted by these correlations are applied to the flat fin air side convection coefficients and air side pressure drops as determined by the correlations developed by McQuiston[2] and incorporated by ORNL into the HAIR and PDAIR subroutines.

The correlation for predicting the increase in air side convection coefficient due to the corrugated fin pattern was named CORFHA and is called by a modified version of the HAIR subroutine, named HAIR2. The correlation for predicting the increase in air side pressure drop APDCOR is called from a modified version of PDAIR by the COND and EVAPR subroutines. The added parameters are contained in the COMMON block FINPAT. The correlations contained in CORFHA and APDCOR were derived from a series of single fin simulation tests performed at the Westinghouse R&D Center by D. T. Beecher. The derivation of these correlations is described in detail in Appendix F.

5.1 HEAT TRANSFER AUGMENTATION

The increase in air side heat transfer coefficient due to the common corrugated or "zig-zag" fin pattern was correlated as a function of geometric parameters and parameters associated with the air flowing through the heat exchanger. The added geometric parameters include:

* The number of fin patterns per row of tubes in the direction of air flow.
* The depth of the fin pattern from peak to valley.

The remaining geometric input parameters were required by the original ORNL heat pump code:

* The fin spacing.
* The fin thickness.
* The tube diameter.
* The tube spacing perpendicular to the air flow direction.
* The tube spacing in the direction of air flow.
* The number of rows of tubes in the direction of air flow.

The air stream input parameters are the same as those required by the original ORNL heat pump code and include:

* The air velocity based on coil face area.
* The average air density.
* The average air thermal conductivity.
* The average air dynamic viscosity.
* The average air Prandtl number.

The calling sequence of the function CORFHA is:

```
CORFHA (FP, DELTA, DEA, ST, WT, FPD, NTR, NFP, VCF, KPA,
        PRA, MUA, RHOA, IDIAG)
```

The input parameters are:

FP = Fin spacing (fins/ft)
DELTA = Fin thickness (ft)
DEA = Tube outside diameter (ft)
ST = Tube spacing perpendicular to the air flow direction (ft)
WT = Tube spacing in the air flow direction (ft)
FPD = Fin pattern depth (peak to valley) (ft)

NTR = Number of tube rows in the direction of air flow (real value)

NFP = Number of fin patterns per row of tubes in the direction of air flow (integer value)

VCF = Air velocity based on total coil face area (ft/min)

KPA = Average air thermal conductivity (Btu/hr-ft-°F)

PRA = Average air Prandtl number (dimensionless)

MUA = Average air dynamic viscosity (lbm/hr-ft)

RHOA = Average air density (lbm/cu ft)

IDIAG = Switch for diagnostic output (integer value)
- 0 = no diagnostic output
- 1 = diagnostic output printed

The output is:

CORFHA = Air side convection coefficient multiplier (dimensionless)

A listing for the function CORFHA is included in Appendix F.

5.2 AIR PRESSURE DROP AUGMENTATION

As noted in Appendix F, our efforts to obtain an accurate correlation between air pressure drop, heat exchanger geometry, and the properties of the air stream passing through the heat exchanger for the single fin model data were not fully successful. However, a relatively crude correlation of the single fin model data developed previously has been included to provide a rough estimate of the increase in air side pressure drop due to fin patternation. This correlation, which relates the increase in air side pressure drop to the fin pattern depth, number of fin patterns per tube row in the direction of air flow, and the air velocity based on coil face area was incorporated into the function APDCOR, which is called by PDAIR via COND or EVAPR. The calling sequence is:

APDCOR (FPD, NFP, VCF)

The input parameters are:

FPD = Fin pattern depth (peak to valley) (ft)

NFP = Number of fin patterns per row of tubes in the direction of air flow (integer value)

VCF = Air velocity based on coil face area (ft/min)

The output is:

APDCOR = Air pressure drop multiplier (dimensionless)

A listing for APDCOR is included in Appendix F.

6. ADDITIONAL PROGRAM CHANGES

In addition to the new subroutines and functions added to the ORNL code it was necessary to modify several of the existing subroutines to accommodate them. These include the DATAIN subroutine, which accepts and displays the input data, and the CNDNSR subroutine, which calls the efficiency and loss based compressor model or the map based compressor model and the flow balancing subroutine FLOBAL and returns the difference between the calculated and specified condenser exit subcooling or between the compressor and expansion device refrigerant mass flow rates. The COND subroutine, which calculates total condenser heat transfer rate, air and refrigerant properties, and refrigerant and air-side pressure drops for fixed inlet refrigerant conditions and the EVAPR subroutine, which calculates evaporator heat transfer rate, air and refrigerant properties and refrigerant, and air-side pressure drops for fixed exit refrigerant conditions, were modified to include the effects of fin patternation on air side heat transfer and pressure drop. The HAIR subroutine, which evaluates the air side heat transfer coefficient for the air-to-refrigerant heat exchangers, was replaced by the HAIR2 subroutine, which optionally includes the effects of the familiar corrugated or "zig-zag" fin pattern by calling the CORFHA subroutine. The PDAIR subroutine was modified to include an optional call of the APDCOR subroutine which predicts the effects of the corrugated fin pattern on air side pressure drop.

The efficiency and loss based compressor model COMP and the map based compressor model CMPMAP were extensively altered to include calls for the reversing valve heat transfer and pressure drop model, the reversing valve leakage model and the suction line accumulator model. The original subroutines were retained and the altered versions titled COMPV and COMPMP, respectively.

The OUTPUT subroutine, which displays the final program results, was modified to include the added parameters associated with the reversing valve, suction line accumulator and fin patternation models.

Listings for all of the modified subroutines are included in Appendix G.

6.1 DATA INPUT SUBROUTINE

The DATAIN subroutine was modified to accept and display the additional input parameters required by the reversing valve, accumulator and fin pattern augmentation models. Several additional COMMON blocks were added to facilitate transfer of these parameters between program sections.

The first parameter in data line #13 is FINTY, the fin type identifier for the indoor coil. Previously there were three valid values for FINTY, 1.0 for flat fins, 2.0 for wavy fins, and 3.0 for louvered fins. With the addition of the model for the effects of the corrugated fin pattern on air side heat transfer a fourth value, 4.0 for corrugated fins, became valid. If FINTY is input as 4.0, an additional data line must be inserted between line #13 and line #14 containing the following data:

LINE #13A FORMAT (2F10.4)

NFP = Number of fin corrugation per row of tubes in the direction of air flow (integer value)

FPD = depth of the fin corrugation, peak to valley (in.)

If FINTY is input as 1.0, 2.0 or 3.0, data line #13A must not be present.

Similarly, the first parameter on data line #17 is the fin type identifier for the outdoor coil. If FINTY for the outdoor coil is input as 4.0, line #17A, containing the above data for the outdoor coil, must be added. If FINTY for the outdoor coil is input as 1.0, 2.0, or 3.0, line #17A must not be present.

To accommodate the reversing valve refrigerant pressure drop and pressure drop models data line #22 must be added to the input card deck or data file. This line must contain:

LINE #22 FORMAT(I10,F10.3)

NRVALV = Switch to select reversing valve pressure drop and heat transfer model
= 0 no reversing valve pressure drop and heat transfer
= 1 empirical breadboard model
= 25 Ranco #25 valve - analytical model
= 26 Ranco #26 valve - analytical model
= 30 Ranco #30 valve - analytical model

TAMBRV = Temperature of the air surrounding the reversing valve (°F)

If NRVALV is input with a value other than 0, 1, 25, 26, or 30, the program will terminate and an error message will be printed.

The reversing valve leakage model requires the data from data line #23, which must be prepared in accordance with the following format:

LINE #23 FORMAT(I4:F12.5:/3(6F12.4:/))

Notice that if more than one floating-point parameter is to be entered in accordance with NRVLK, additional lines, each containing up to three parameters, must be prepared. However, only the required number of parameters need and should be entered. The parameters have the following significance:

NRVLK = switch to select reversing valve internal leakage model (integer value)
= 0, no reversing valve internal leakage (a blank data line #23 will create this option)
= 1, fixed reversing valve internal leak rate

If NRVLK = 1 then the following data must also be included:

RVLKP(1) = fixed reversing valve leak rate as a fraction of the compressor mass flow rate

The reversing valve leakage model has been constructed in a manner which will easily accommodate more complex models if desired. RVLKP is an array which can include up to five parameters. The number of parameters is a function of NRVLK as specified by the array NLKTB(NRVLK+1) contained in BLOCK DATA RLADP and dimensioned to accept up to five parameters. At present the only valid values of NRVLK are 0 and 1. If NRVLK is set to 0, no values of RVLKP are required. If NRVLK is set to 1, one value of RVLKP is required.

The input data for the suction line accumulator model is provided by data line #24, which must contain:

LINE #24 FORMAT(I4:F12.5:/3(6F12.4:/))

NACCU = switch to select suction line accumulator heat transfer and pressure drop model (integer value)
- = 0, suction line accumulator heat transfer and pressure drop not included (a blank data line #24 will achieve this)
- = 1 or 2, suction line accumulator heat transfer and pressure drop included

If NACCU = 1 the following data must be included:

ACPAR(1) = inside diameter of accumulator heat exchanger liquid line) tubing (in.)

ACPAR(2) = outside diameter of accumulator heat exchanger (liquid line) tubing (in.)

ACPAR(3) = length of tubing in the accumulator heat exchanger (ft)

ACPAR(4) = inside diameter of accumulator vapor vapor line (in.)

ACPAR(5) = total length of accumulator vapor line (inlet + outlet) in (ft)

If NACCU = 2 the following data must be included:

ACPAR(1) = refrigerant superheat leaving the accumulator on the suction side (°F)

ACPAR(2) = pressure drop on the low pressure side of the accumulator (psia)

ACPAR(3) = pressure drop on the high pressure side of the accumulator (psia)

ACPAR(4) = heat flow from the accumulator shell (Btu/sec)

Note that like the reversing valve leakage model, the accumulator heat transfer and pressure drop subroutine has been configured to easily accept a more complex mathematical representation. ACPAR is an array dimensioned to accept up to 20 parameters. The number of parameters associated with each value of NACCU is defined by the array NACTB(NACCU+1) contained in BLOCK DATA RLADP and dimensioned to accept up to 12 parameters, corresponding to 12 different accumulator models. At present the valid values of NACCU are 0, 1, and 2. If NACCU is 0, NACTB(1) is 0 and no input parameters are required. If NACCU is 1, NACTB(2) is 5 and the five parameters listed above are required. If NACCU is 2, NACTB(3) is 4 and the four corresponding parameters above are required.

The COMMON blocks added include RVALVE containing the parameters associated withe the reversing valve refrigerant pressure drop and heat transfer models, RVLKP containing the parameters of the reversing valve internal refrigerant leakage model, ACPAR and ACTBL containing the suction line accumulator refrigerant heat transfer and pressure drop parameters and FINPAT, which contains the parameters required by models predicting the effects of fin patternation on air side heat transfer and pressure drop.

6.2 CONDENSER BALANCING SUBROUTINE

The only changes required in the CNDNSR subroutine were to substitute a call for the modified version of the loss and efficiency compressor model COMPV for the original model COMP and a call for the modified map based compressor model COMPMP for the original version CMPMAP.

6.3 CONDENSER HEAT TRANSFER SUBROUTINE

The COND subroutine, which calculates the total condenser heat transfer rate, air and refrigerant properties, and refrigerant and air side pressure drop in the condenser for fixed inlet refrigerant conditions, was altered to call the HAIR2 subroutine for evaluating the air side heat transfer coefficient in place of the HAIR subroutine. This change was required to provide access to the correlation for air side heat transfer augmentation due to fin patternation CORFHA, which is called by HAIR2. The call for the air side pressure drop subroutine PDAIR was modified to include the additional parameters required by the model predicting the increase in air side pressure drop due to corrugated fin patternation APDCOR, which is called by PDAIR. The COMMON block FINPAT was added to COND.

6.4 EVAPORATOR HEAT TRANSFER SUBROUTINE

The EVAPR subroutine, which calculates the total evaporator heat transfer rate, air and refrigerant properties, and refrigerant and air side pressure drop in the evaporator for fixed exit refrigerant conditions, was also altered to call the HAIR2 subroutine in place of the HAIR subroutine. The call for the air pressure drop subroutine PDAIR was altered to include the additional parameters required by the APDCOR subroutine. The COMMON block FINPAT was also added to EVAPR.

6.5 EFFICIENCY AND LOSS BASED COMPRESSOR MODEL

The efficiency and loss based compressor model, COMP, computes the refrigerant mass flow rate and power consumption given the compressor suction and discharge conditions. In addition, the evaporator discharge and condenser inlet conditions are computed using the line heat transfer rates input to DATAIN and the refrigerant pressure drops in the lines using the DPLINE subroutine. The heat transfer rates and refrigerant pressure drops associated with the reversing valve and the suction line accumulator, computed by the VALVER and DHSUL subroutines, were included in a similar manner. It was also convenient to call the reversing valve refrigerant leakage subroutine FLKRV at this location. The COMMON blocks RVALVE, ACPAR, and RVLKP were added to transfer in the additional parameters required. Due to the extensive nature of the modifications the modified version was named COMPV and the original version was maintained with no changes.

6.6 MAP BASED COMPRESSOR MODEL

The map based compressor model CMPMAP computes the refrigerant mass flow rate and power consumption given the compressor suction and discharge conditions using second order polynomial curve fits applied to the compressor map data. In addition, the evaporator discharge and condenser inlet refrigerant conditions are calculated using refrigerant line heat transfer rates input via the DATAIN subroutine and refrigerant pressure drops calculated by the DPLINE subroutine. The heat transfer rates and refrigerant pressure drops associated with the reversing valve and the suction line accumulator, as calculated by the VALVER and DHSUL subroutines, are also included. The reversing valve refrigerant leakage subroutine FLKRV is also called from this subroutine. The COMMON blocks RVALVE, ACPAR, and RLVKP were included to transfer in the additional parameters required. Due to the extensive modifications required, this subroutine was renamed COMPMP and the original version was maintained unchanged.

6.7 OUTPUT SUBROUTINE

The OUTPUT subroutine provides a conveniently formatted printout of the final results of the system performance simulation following the completion of all of the iterative loops. The subroutine consists primarily of formats and write statements with the associated parameters transferred in via COMMON blocks. The Westinghouse modification consisted of adding the COMMON blocks RVALVE, ACPAR, RVLKP, and FINPAT and formats and write statements to print selected variables contained in these COMMON blocks. Several additional parameters associated with the reversing valve heat transfer and refrigerant pressure drop model are printed including:

DPLOV = Reversing valve refrigerant pressure drop - low pressure side (psi)

DPHIV = Reversing valve refrigerant pressure drop - high pressure side (psi)

QEXTV = Reversing valve heat transfer to ambient air (Btu/hr)

QINTV = Reversing valve internal heat transfer (Btu/hr)

The additional parameters associated with the effects of fin patternation on air side heat transfer and pressure drop added to the printout include:

XAPE = Air pressure drop multiplying factor due to fin patternation for the evaporator

XAPC = Air pressure drop multiplying factor due to fin patternation for the condenser

XFPE = Air side convection coefficient multiplying factor due to fin patternation for the evaporator

XFPC = Air side convection coefficient multiplying factor due to fin patternation for the condenser

7. VERIFICATION

During the verification phase of the project, the ORNL heat pump code, with the reversing valve heat transfer and pressure drop models, the reversing valve leakage model and the accumulator model incorporated, was run using input data corresponding to the actual test conditions encountered during testing of the first preprototype of the Westinghouse/DOE advanced electric heat pump. The input data was selected to match the test conditions as closely as possible. For example, if the desired indoor air dry bulb temperature and relative humidity were 80°F and 50% and the actual test conditions were 79°F and 51% relative humidity, the actual test conditions were input to the program rather than the desired conditions. The measured indoor and outdoor air flows for each individual test were used rather than the desired values or average values.

Note that the electric expansion valve used in the preprototype was manually controlled during the verification tests. Our ultimate objective was to use refrigerant temperature sensors in combination with a microprocessor based control system to control condenser exit subcooling during heating mode operation and evaporator exit superheat during cooling operation via a feedback loop. At the time these tests were performed, the microprocessor control system was still under development. The breadboard microprocessor proved to have insufficient memory to perform this control function and during the field test[7] the expansion valve was controlled by an algorithm relating expansion valve operating voltage to operating mode and ambient temperature rather than a feedback control loop. Due to the use of manual control of the expansion valve and a tendency for the refrigerant system operating conditions to "hunt," the desired level of subcooling during heating operation and superheat during cooling operation were not achieved in all of the verification tests. In particular, some of the cooling runs have very low evaporator exit superheat and the exit refrigerant may have been slightly wet in some cases.

Verification runs were completed for all four steady state operating modes, high and low capacity heating and high and low capacity cooling. The nominal outdoor ambient temperatures considered were:

* High Capacity Heating; 47°F, 35°F, 17°F and 0°F.
* Low Capacity Heating; 47°F and 35°F.
* High Capacity Cooling; 82°F, 95°F and 106°F.
* Low Capacity Cooling; 76°F, 82°F and 95°F.

For each of these conditions, the program was run with corrections for pressure drop in the connecting lines for all three of the Ranco reversing valve models built into the VALVER subroutine and for the empirical reversing valve heat transfer and pressure drop model with the connecting line lengths set to zero (the empirical model includes line pressure drops).

7.1 INPUT DATA

The input data for the ORNL code includes the number of parallel refrigerant circuits in the indoor and outdoor air-to-refrigerant heat exchangers. Since both the indoor and outdoor air-to-refrigerant heat exchangers used in the advanced electric heat pump have refrigerant circuits that branch or combine, it was necessary to input a "mean" number of parallel refrigerant paths. For example, as shown in Figure 7-1, the indoor coil of the Westinghouse/DOE advanced electric heat pump[6] consists of three parallel sections. In the heating mode, each section has two parallel circuits at the inlet, each containing 12 tubes. These circuits combine into a single circuit containing 10 tubes. Therefore there are 6 parallel circuits containing 72 tubes at inlet and three parallel circuits containing 30 tubes at the coil exit. One of the sections contains an unused hairpin, giving a total of 104 tubes in the heat exchanger. To determine the number of parallel circuits to be input to the ORNL code, a trial-and-error approach was

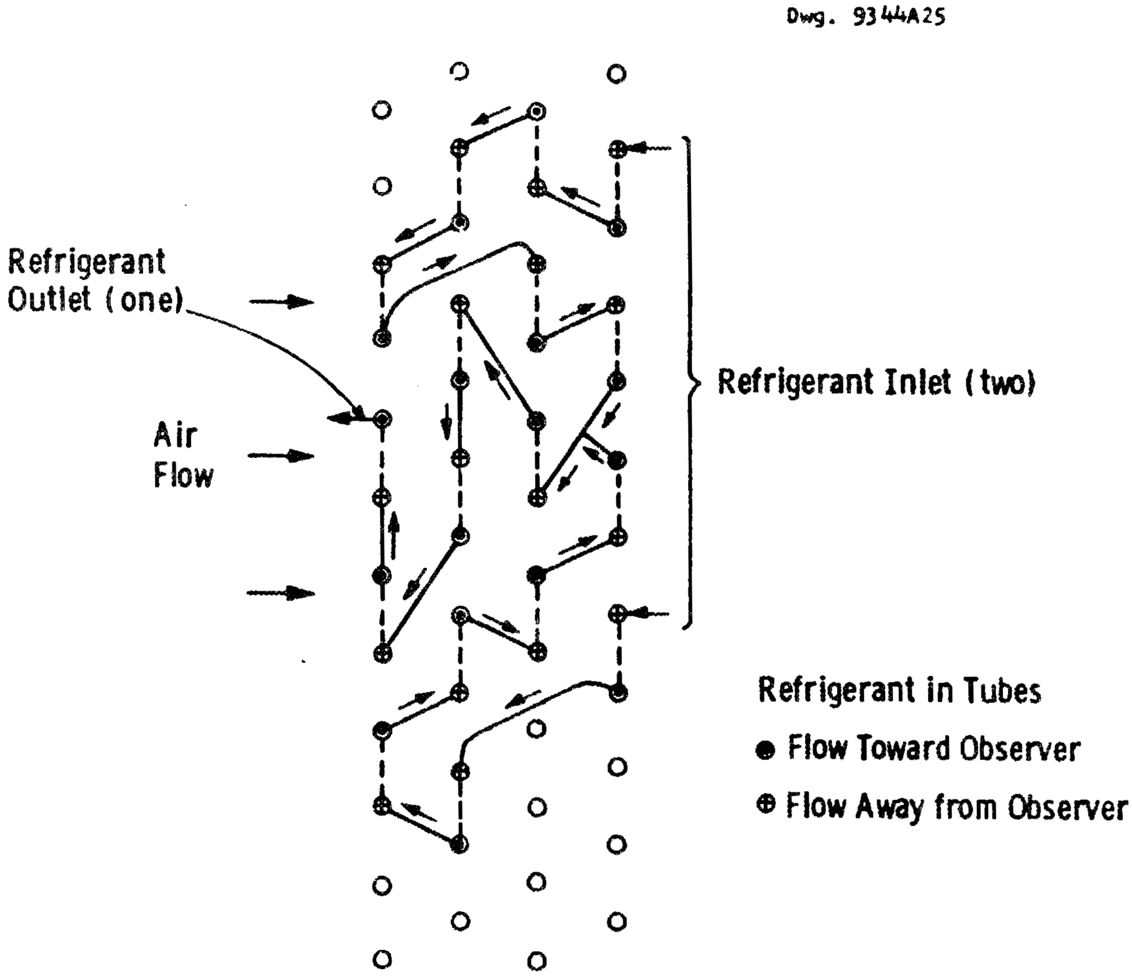


Figure 7-1 — One of three indoor coil circuits (heating mode).

used with high capacity heating performance at 47°F ambient temperature as a reference case. It was found that 4.5 parallel circuits gave a good match between measured and predicted refrigerant pressure drop in the indoor coil. This is not greatly different from the weighted average number of circuits:

$$W_{ai} = [(6 \times 72)+(3 \times 30)]/102 = 5.12$$

The outdoor air to refrigerant heat exchanger consists of four separate coils connected in parallel. The refrigerant circuit for each of the coils in the heating mode, shown in Figure 7-2, consists of a single circuit containing 14 tubes at inlet branching into two circuits of 14 tubes each at the outlet. For the four coils combined there are four parallel circuits containing a total of 72 tubes at inlet and eight parallel circuits containing a total of 112 tubes at the outlet giving a weighted average of:

$$W_{ao} = [(4 \times 72)+(8 \times 112)]/184 = 4.87$$

The trial-and-error method indicated that the number of parallel circuits giving the best match between measured and predicted refrigerant pressure drop was 8.0.

A similar trial-and-error approach was applied in the cooling mode at the high capacity rating condition of 95°F outdoor ambient temperature and 80°F dry bulb, 67°F wet bulb indoors. The number of parallel circuits in the indoor coil giving the best match between predicted and measured refrigerant pressure drop was found to be 4.0 parallel paths. The number of parallel paths in the outdoor coil giving the best match between measured and predicted refrigerant pressure drop was 5.8. In the low capacity cooling mode, one-third of the indoor coil is cut out of the circuit by a solenoid valve to maintain a low enough evaporating temperature to provide a comfortable sensible/total capacity ratio. For the 95°F low capacity cooling mode the number of indoor parallel refrigerant circuits giving the best match between predicted and measured refrigerant pressure drop was found to be 2.9.

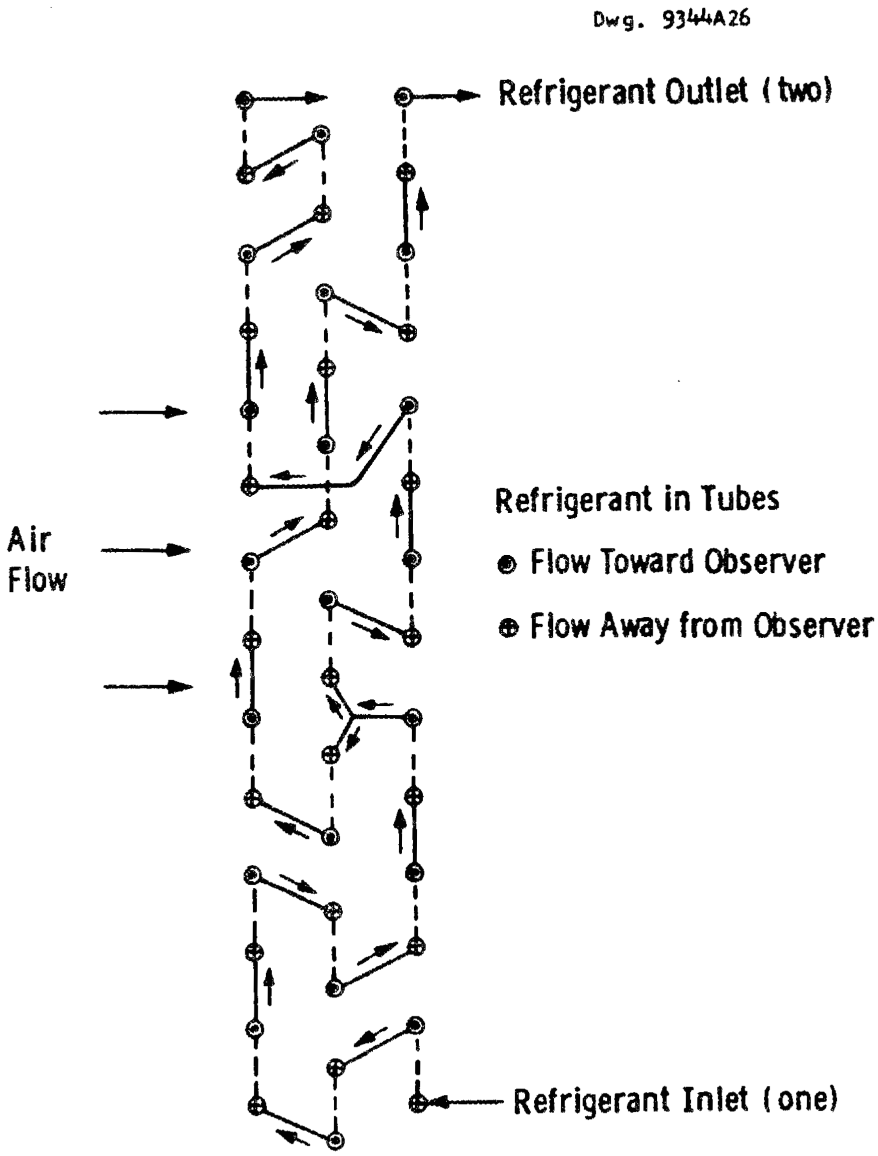


Figure 7-2 — One of four outdoor heat exchanger circuits (heating mode).

An analysis of the sensitivity of the predicted capacity and energy efficiency to the number of refrigerant circuits in the indoor and outdoor heat exchangers was performed for both heating and cooling service and is described later in this report.

Another parameter evaluated empirically was the product of the overall heat transfer coefficient and the heat transfer area for the accumulator heat exchanger. As noted previously, the advanced electric heat pump uses a unique refrigerant circuit which provides for flooded evaporator operation in the heating mode and superheated evaporator exit conditions in cooling service. In the heating mode, the liquid remaining in the evaporator exit stream is vaporized in the accumulator by heat absorbed from the high pressure liquid stream passing from the condenser outlet to the expansion valve inlet via a heat exchanger consisting of a flat spiral of bare tubing near the bottom of the accumulator. During this process the high pressure liquid stream is deeply subcooled resulting in a significant decrease in the quality of the liquid entering the evaporator.

In cooling service the stream passing through the accumulator heat exchanger flows between the expansion valve outlet and the evaporator inlet and little temperature difference exists. Consequently, the heat transfer rate is small and has been neglected in modeling the system.

Two different approaches to modeling the effects of the accumulator heat exchanger were considered. The first approach is to assume that the heat exchanger capacity is always sufficient to evaporate all remaining liquid in the evaporator exit stream. If the evaporator exit conditions are not specified, as in the case of the ORNL code, thermodynamic considerations alone are not sufficient to define the operating conditions. A wide range of evaporator exit qualities and concomitant values of expansion valve inlet subcooling exist which will satisfy the requirements of the first law of thermodynamics. The operating conditions are determined by the refrigerant charge level and the characteristics of the expansion device. Increasing amounts of subcooling entering the expansion device will result in a decrease in the inlet and exit quality of the evaporator and, assuming no great change in the slip factor between the liquid and vapor phases, an increase in the refrigerant charge contained in the evaporator and accumulator.

The second approach, while less rigorous physically, proved to be far more compatible with the internal logic of the ORNL code. The enthalpy change of the refrigerant liquid is given by:

$$Q_{ac} = M_r Cp_r (T_{cx} - T_{vi}) \tag{3}$$

The heat transfer rate is given by:

$$Q_{ac} = U_o A_t L_t [(T_{cx}-T_{ex}) - (T_{vi}-T_{ex})] / \ln[(T_{cx}-T_{ex})/(T_{vi}-T_{ex})] \tag{4}$$

Combining these relationships gives:

$$Q_{ac} = M_r Cp_r \; (T_{cx}-T_{ex}) \; [1.0 - e^{(-U_o A_t L_t / M_r Cp_r)}] \tag{5}$$

If the overall heat transfer coefficient U_o is assumed to be a fixed value, then the primary unknown (assuming M_r, T_{cx}, and T_{ex} are determined primarily by other system considerations) is the length of tubing in the heat exchanger L_t. The value of L_t giving an expansion valve inlet temperature equal to the measured value was determined iteratively for each of the heating runs analyzed. The value of L_t was not a constant, ranging from 30.0 ft to 45.0 ft, indicating that the assumption of a constant overall heat transfer coefficient U_o is an oversimplification of the problem. This is not surprising since we would expect both the inside and outside convection coefficients to be a function of refrigerant mass flow rate, density, quality, and other fluid properties. The values of L_t used for each of the heating runs is shown in Table 7-1. The value of L_t obtained for the 17°F high capacity heating mode is inconsistent with the values obtained for the other high capacity heating operating conditions. The reason for this inconsistency is unclear. It may have been the result of a variation in the refrigerant

Table 7-1. Equivalent Lengths of Tubing in the Accumulator Heat Exchanger

Outdoor Ambient Temperature	Heating Mode High Capacity	Heating Mode Low Capacity
47°F	37.5 ft	30.0 ft
35°F	32.5 ft	32.5 ft
17°F	45.0 ft	
0°F	30.0 ft	

charge level or due to a buildup of frost on the evaporator coil. As shown in Table 7-20, the overall system heating capacity and coefficient of performance are relatively insensitive to the the value used for L_t. The parameter most influenced by L_t is the evaporator exit quality, which is difficult to measure accurately.

Appendix H contains sample input data files for all of the combinations of operating mode and ambient temperature considered in the verification study. The samples shown are for a Ranco #26 reversing valve with 2% refrigerant leakage. The suction line accumulator model is used for the heating cases. For the cooling cases the suction line accumulator model is not called since the evaporator exit refrigerant is assumed to be superheated and heat transfer in the accumulator is assumed to be negligible.

7.2 REVERSING VALVE HEAT TRANSFER AND PRESSURE DROP

7.2.1 Heating Mode

Table 7-2 summarizes the results of the verification runs for the high capacity heating mode at an outdoor ambient temperature of 47°F. The first column shows the test data. The second column shows the results for no reversing valve, but with connecting line losses included. The connecting line diameters and lengths used are shown in Table 7-3 and represent the actual line diameters and lengths present during the tests with equivalent lengths for bends added. The next three columns show the results for the three Ranco reversing valve

Table 7-2. Test 103 High Capacity Heating @ 47 Deg F

	Test	Lines Only	Ranco #25	Ranco #26	Ranco #30	Empri-cal
Compressor						
Mass Flow, lbm/hr	532.2	556.0	554.3	530.1	549.6	539.6
Power Input, Watts	3565	3575	3559	3515	3547	3465
Heat Loss, Btu/hr	2032	2074	2065	2040	2058	2011
Inlet Pressure, psia	66.8	69.6	69.3	67.1	68.9	67.5
Inlet Temperature, deg F	28.4	30.6	30.4	28.6	30.0	28.9
Inlet Superheat, deg F	0.6	0.6	0.6	0.6	0.6	0.6
Inlet Enthalpy, Btu/lbm	107.1	107.4	107.4	107.2	107.3	107.2
Exit Pressure, psia	239.8	242.9	241.5	241.6	241.2	234.4
Exit Temperature, deg F	170.9	170.3	169.8	172.0	170.1	168.0
Exit Superheat, deg F	61.3	59.8	59.8	61.9	60.1	60.1
Exit Enthalpy, Btu/lbm	125.8	125.6	125.6	126.0	125.6	125.4
Lines & Reversing Valve						
Pressure drop, psi	6.49	2.96	3.10	5.96	3.39	0.0
Heat Transfer, Btu/hr	651.2	0.0	560.4	40.8	463.2	1107.3
Condenser						
Inlet Pressure, psia	233.3	239.9	239.4	235.6	237.8	234.4
Inlet Temperature, deg F	163.8	169.8	164.5	170.5	165.3	158.1
Inlet Superheat, deg F	56.3	60.2	55.3	62.3	56.4	50.2
Inlet Enthalpy, Btu/lbm	124.6	125.6	124.5	125.9	124.8	123.4
Exit Pressure, psia	231.6	238.1	236.6	233.9	235.9	232.6
Exit Temperature, deg F	93.9	95.9	95.4	94.7	95.3	94.2
Exit Subcooling, deg F	13.1	13.1	13.1	13.1	13.1	13.1
Exit Enthalpy, Btu/lbm	37.4	38.0	37.8	37.6	37.8	37.5
Lines & Accumulator						
Pressure Drop, psi	?	2.11	2.10	1.93	2.07	1.20
Heat Transfer, Btu/hr	6912	7158	7353	6923	7048	6920
Expansion Valve						
Inlet Pressure, psia	?	234.5	231.9	231.9	233.9	231.4
Inlet Temperature, deg F	50.3	52.6	50.6	50.6	52.1	50.9
Inlet Subcooling, deg F	?	55.8	57.3	56.6	55.7	56.0
Inlet Enthalpy, Btu/lbm	24.4	25.1	24.6	24.5	25.0	24.6
Evaporator						
Inlet Pressure, psia	77.6	74.2	74.2	74.4	74.3	74.8
Inlet Temperature, deg F	35.0	33.6	33.6	33.7	33.6	34.0
Inlet Quality	0.050	0.063	0.056	0.056	0.061	0.056
Inlet Enthalpy, Btu/lbm	24.4	25.1	24.6	24.5	25.0	24.6
Exit Pressure, psia	74.0	71.7	71.8	72.1	71.9	72.5
Exit Temperature, deg F	33.4	31.6	31.7	32.0	31.8	32.3
Exit Quality	0.852	0.854	0.838	0.849	0.845	0.829
Exit Enthalpy, Btu/lbm	94.6	94.5	93.2	94.1	93.7	92.4
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	7.23	0.66	2.44	5.05	2.97	5.01
Heat Transfer, Btu/hr	6652	7158	7855	6940	7476	8027
Heating Capacity, Btu/hr*	46408	48706	48058	46815	47815	46352
Coefficient of Performance**	3.81	3.99	3.96	3.90	3.95	3.92

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-3. Equivalent Lengths and Internal Diameters of Connecting Tubing - Advanced Electric Heat Pump in Test Configuration

Location	Internal Diameter inches	Equivalent Length feet
Liquid line from indoor to outdoor heat exchanger	0.430	40.0
From indoor coil to reversing valve	0.680	3.0
From outdoor coil to reversing valve	0.805	25.0
From reversing valve to compressor inlet	0.680	7.0
From reversing valve to compressor outlet	0.430	10.0

models included in the VALVER subroutine plus the connecting line losses. The last column presents the results for the empirical model with the connecting line lengths set to zero since the empirical model includes line losses. Note that the capacity and COP shown are based on refrigerant side calorimetry and do not include the effects of fan power, compressor losses (oil cooler), controls, etc. Note also that the heat transfer for the lines between the compressor and the condenser and in the high pressure side of the reversing valve (row 13 in Table 7-2) includes heat transfer to the ambient air and is therefore greater than the difference between the heat transfer between the evaporator discharge and the compressor inlet (row 23) and the heat transfer in the high pressure side of the accumulator (row 37).

Obviously including reversing valve heat transfer and pressure drop significantly improves agreement between predicted and measured performance. The #25 and #30 valves are of similar construction but different ratings, 9.0 tons and 5.5 tons, respectively. Consequently the refrigerant pressure drop is lower with the #25 valve, but the heat transfer rate is higher due to the larger slide surface area. The #26 valve has a nominal rating of 2.0 tons and significantly higher refrigerant pressure drop than the #25 and #30 valves, but a heat transfer rate more than an order of magnitude lower due to a smaller slide surface area and the use of a plastic (Delrin) slide in place of the metal slides used in the other two valves.

Note that the capacity and COP are lower with the #26 valve than with the #25 and #30 valves. This supports our hypothesis that reversing valve pressure drop has a greater impact on capacity and energy efficiency than internal heat transfer. Note also that the empirical model, which does not include an allowance for pressure drop between the compressor exit and the condenser inlet, is similar in capacity and COP to the #26 valve, which has the largest compressor discharge side pressure drop, supporting the assertion that pressure drop on the compressor suction side penalizes performance far more than pressure drop on the compressor discharge side. Valve #26 gives the best match between predicted capacity and COP of the analytical models. The empirical model more closely matches capacity than the #26 valve, but does not match the COP as accurately.

Table 7-4 summarizes the results for the high capacity heating mode at an ambient temperature of 35°F. The #30 valve case most closely matches the measured capacity, but the #26 valve case more closely predicts the measured COP. In most respects the #30 valve case most closely matches the measured parameters.

The results for the 17°F high capacity heating runs are shown in Table 7-5. The predicted capacities and COPs are higher than the measured values for all of the cases considered. This is primarily the result of the inaccuracy of the compressor map curve fit near or below the lowest compressor suction saturation temperature used in the curve fit. Note that while the #30 valve case quite accurately mirrors the measured compressor suction and discharge pressures, the predicted refrigerant mass flow rate exceeds the measured value by 12.5%. The ORNL code uses a compressor map curve fit of the form:

$$M_r = B1T_cT_c + B2T_c + B3T_eT_e + B4T_e + B5T_cT_e + B6 \tag{6}$$

Our own experience has shown that a fit of the form:

$$M_r = C0 + C1T_e + C2T_eT_e \tag{7}$$

where:

$$C0 = C00 + C01T_c + C02T_cT_c \tag{8}$$

$$C1 = C10 + C11T_c + C12T_cT_c \tag{9}$$

$$C2 = C20 + C21T_c + C22T_cT_c \tag{10}$$

gives a superior match to the map data for the wide range of operating conditions encountered in heat pumps.

For 0°F high capacity heating, the compressor map curve fit gave a poor match to the measured refrigerant mass flow rate and power consumption. Therefore the program was run with a fixed refrigerant mass flow rate and compressor power input to evaluate the performance of the

Table 7-4. Test 109B High Capacity Heating @ 35 Deg F

	Test	Lines Only	Ranco #25	Ranco #26	Ranco #30	Empri-cal
Compressor						
Mass Flow, lb/hr	447.7	453.5	453.0	433.6	448.3	442.8
Power Input, Watts	3197	3166	3141	3113	3137	3077
Heat Loss, Btu/hr	2225	2161	2144	2125	2141	2100
Inlet Pressure, psia	57.5	58.3	58.2	56.4	57.7	56.4
Inlet Temperature, deg F	19.8	20.5	20.4	20.7	20.0	19.2
Inlet Superheat, deg F	0.0	0.0	0.0	0.0	0.0	0.0
Inlet Enthalpy, Btu/lbm	106.3	106.4	106.4	106.3	106.4	106.3
Exit Pressure, psia	218.0	218.1	217.0	217.3	216.7	211.6
Exit Temperature, deg F	166.9	165.2	164.3	166.9	164.9	162.9
Exit Superheat, deg F	64.3	62.7	62.1	64.6	62.8	62.6
Exit Enthalpy, Btu/lbm	125.8	125.5	125.4	125.9	125.5	125.3
Lines & Reversing Valve						
Pressure Drop, psi	5.49	2.26	2.37	4.56	2.58	0.0
Heat Transfer, Btu/hr	654.6	0.0	514.6	43.8	448.7	1142.9
Condenser						
Inlet Pressure, psia	212.6	215.9	214.6	212.8	214.1	211.6
Inlet Temperature, deg F	158.6	164.8	158.3	165.5	159.5	150.2
Inlet Superheat, deg F	57.9	63.0	56.9	64.8	58.3	49.9
Inlet Enthalpy, Btu/lbm	124.4	125.5	124.2	125.8	124.5	122.7
Exit Pressure, psia	210.4	214.4	213.2	211.4	212.7	210.1
Exit Temperature, deg F	95.1	96.5	96.0	95.5	95.9	95.0
Exit Subcooling, deg F	4.8	4.8	4.8	4.8	4.8	4.8
Exit Enthalpy, Btu/lbm	37.7	38.2	38.0	37.9	38.0	37.7
Lines & Accumulator						
Pressure Drop, psi	?	1.26	1.26	1.16	1.28	0.69
Heat Transfer, Btu/hr	6926	6810	6754	6677	7301	7166
Expansion Valve						
Inlet Pressure, psia	?	213.2	211.9	210.2	211.4	209.4
Inlet Temperature, deg F	43.0	45.7	45.6	43.2	40.5	40.0
Inlet Subcooling, deg F	?	55.2	54.9	56.6	59.8	59.6
Inlet Enthalpy, Btu/lbm	22.3	23.2	23.1	22.5	21.7	21.5
Evaporator						
Inlet Pressure, psia	66.9	63.1	63.3	63.3	63.1	63.5
Inlet Temperature, deg F	25.5	24.7	24.8	24.9	24.7	25.0
Inlet Quality	0.055	0.067	0.066	0.059	0.051	0.048
Inlet Enthalpy, Btu/lbm	22.3	23.2	23.1	22.5	21.7	21.5
Exit Pressure, psia	63.8	61.2	61.3	61.6	61.3	61.7
Exit Temperature, deg F	25.3	23.0	23.2	23.4	23.1	23.5
Exit Quality	0.827	0.831	0.820	0.824	0.806	0.787
Exit Enthalpy, Btu/lbm	91.3	91.4	90.5	90.8	89.2	87.5
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	6.31	2.84	3.11	5.21	3.53	4.88
Heat Transfer, Btu/hr	6738	6810	7206	6695	7701	8309
Capacity, Btu/hr*	38816	39591	39049	38113	38778	37638
Coefficient of Performance**	3.56	3.66	3.64	3.59	3.62	3.58

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-5. Test 106 High Capacity Heating @ 17 Deg F

	Test	Lines Only	Ranco #25	Ranco #26	Ranco #30	Emprical
Compressor						
Mass Flow, lb/hr	281.7	319.0	318.9	306.9	316.8	301.0
Power Input, Watts	2695	2695	2701	2669	2683	2609
Heat Loss, Btu/hr	2680	2668	2673	2641	2656	2583
Inlet Pressure, psia	43.2	44.3	43.5	42.2	43.2	41.1
Inlet Temperature, deg F	8.5	8.8	8.8	7.3	8.5	6.0
Inlet Superheat, deg F	3.1	3.1	3.1	3.1	3.1	3.1
Inlet Enthalpy, Btu/lbm	105.5	105.5	105.5	105.4	105.5	105.3
Exit Pressure, psia	194.2	195.3	194.4	194.8	194.3	188.5
Exit Temperature, deg F	176.6	163.5	163.5	165.6	163.4	163.5
Exit Superheat, deg F	82.4	68.9	69.3	71.3	69.2	71.4
Exit Enthalpy, Btu/lbm	128.6	126.0	126.0	126.4	126.0	126.3
Lines & Reversing Valve						
Pressure Drop, psi	3.60	1.32	1.45	2.71	1.52	0.0
Heat Transfer, Btu/hr	591.6	0.0	463.2	52.6	421.9	1289.1
Condenser						
Inlet Pressure, psia	190.6	194.0	193.0	192.1	192.8	188.5
Inlet Temperature, deg F	165.3	163.2	155.9	164.3	156.4	141.8
Inlet Superheat, deg F	72.5	69.2	62.2	70.9	63.0	49.8
Inlet Enthalpy, Btu/lbm	126.5	126.0	124.6	126.3	124.7	122.0
Exit Pressure, psia	188.6	193.2	192.2	191.3	191.9	187.8
Exit Temperature, deg F	85.2	86.8	86.6	86.3	86.4	84.9
Exit Subcooling, deg F	6.9	7.0	6.8	6.8	6.9	6.8
Exit Enthalpy, Btu/lbm	34.7	35.2	35.1	35.0	35.1	34.6
Lines & Accumulator						
Pressure Drop, psi	?	0.82	0.82	0.76	0.80	0.45
Heat Transfer, Btu/hr	4671	5102	5074	5162	5085	5121
Expansion Valve						
Inlet Pressure, psia	?	192.3	191.4	190.5	191.1	187.3
Inlet Temperature, deg F	27.4	31.7	31.8	28.2	31.1	26.0
Inlet Subcooling, deg F	?	61.8	61.3	64.6	61.9	65.6
Inlet Enthalpy, Btu/lbm	17.8	19.2	19.2	18.2	19.0	17.6
Evaporator						
Inlet Pressure, psia	5.14	46.3	46.4	46.5	46.4	46.9
Inlet Temperature, deg F	14.0	8.7	8.9	9.0	8.9	9.4
Inlet Quality	0.053	0.069	0.069	0.058	0.067	0.050
Inlet Enthalpy, Btu/lbm	17.8	19.2	19.2	18.2	19.0	17.6
Exit Pressure, psia	49.4	44.9	45.0	45.3	45.1	45.8
Exit Temperature, deg F	12.0	7.2	7.4	7.6	7.4	8.2
Exit Quality	0.817	0.832	0.819	0.820	0.818	0.770
Exit Enthalpy, Btu/lbm	88.8	89.5	88.4	88.5	88.3	84.0
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	6.20	1.37	1.56	3.09	1.87	4.68
Heat Transfer, Btu/hr	4704	5102	5463	5183	5450	6411
Capacity, Btu/hr*	25860	28969	28542	28006	28393	26303
Coefficient of Performance**	2.78	3.15	3.10	3.07	3.10	2.95

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

remainder of the model. The results are shown in Table 7-6. The #25 and #30 valves give the best match with the measured performance. Since the refrigerant mass flow rates are low, the refrigerant pressure drops are relatively low for all of the valve models. As a result heat transfer becomes more important than at higher mass flow rates and the low heat transfer rate of the #26 valve results in superior system performance. All of the Ranco valve models give pressure drops lower than the measured values. Note also that the empirical model, which greatly overestimates the internal valve heat transfer, underestimates capacity and COP by a significant margin, supporting the conclusion that heat transfer is more significant at low ambient temperatures.

The results for the low capacity heating runs at 47°F are presented in Table 7-7. All three Ranco valve models match the measured capacity fairly accurately, with the #25 valve giving the best match on capacity and refrigerant mass flow rate and the #26 valve more closely matching the measured COP. Due to the low refrigerant mass flow rates compared to high capacity operation, the refrigerant pressure drops are moderate with all three valves. The #26 valve gives lower capacity and COP than the #25 and #30 valves, indicating that refrigerant pressure drop is more significant than internal heat transfer in this operating regime. The empirical model, which gives the highest heat transfer and pressure drop, underestimates the capacity and COP by the widest margin. This is primarily due to the fact that the reversing valve used in the breadboard unit was undersized relative to the compressor capacity and had excessive refrigerant pressure drop.

Low capacity heating results at 35°F outdoor ambient temperature are shown in Table 7-8. Capacity and COP are underestimated for all of the cases considered. Note that while the #25, #26 and #30 valves all predict compressor suction and discharge pressures with reasonable accuracy, the measured refrigerant flow rate is 9 to 12% higher than the predicted flow rate and measured compressor power input is about 4% higher than predicted values. This is the result of the inaccuracy of the six coefficient map curve fit near the lower limit of the suction pressure input data. The more complex nine coefficient model described

Table 7-6. Test 110 High Capacity Heating @ 0 Deg F
Fixed Refrigerant Flow

	Test	Lines Only	Ranco #25	Ranco #26	Ranco #30	Empri-cal
Compressor						
Mass Flow, lbm/hr	157.8	157.8	157.8	157.8	157.8	157.8
Power Input, Watts	2284	2284	2284	2284	2284	2284
Heat Loss, Btu/hr	3701	3820	3820	3820	3820	3820
Inlet Pressure, psia	33.5	33.7	33.7	33.3	33.7	30.8
Inlet Temperature, deg F	-3.6	-3.3	-3.3	-3.9	-3.4	-7.5
Inlet Superheat, deg F	3.1	3.1	3.1	3.1	3.1	3.1
Inlet Enthalpy, Btu/lbm	104.1	104.3	104.3	104.2	104.3	103.9
Exit Pressure, psia	174.9	167.8	167.4	168.1	167.4	164.6
Exit Temperature, deg F	180.5	176.5	176.4	176.2	176.4	173.7
Exit Superheat, deg F	93.7	92.6	92.6	92.1	92.6	91.0
Exit Enthalpy, Btu/lbm	130.0	129.5	129.5	129.4	129.5	129.1
Lines & Reversing Valve						
Pressure drop, psi	2.73	0.44	0.47	0.96	0.51	0.0
Heat Transfer, Btu/hr	522.8	0.0	349.1	62.3	338.5	1496.3
Condenser						
Inlet Pressure, psia	172.1	167.2	166.9	167.1	166.9	164.7
Inlet Temperature, deg F	162.8	176.4	164.8	174.0	165.1	124.5
Inlet Superheat, deg F	77.1	92.7	81.2	90.3	81.5	41.9
Inlet Enthalpy, Btu/lbm	126.7	129.5	127.3	129.0	127.3	119.6
Exit Pressure, psia	170.2	166.9	166.7	166.9	166.7	164.4
Exit Temperature, deg F	81.0	79.6	79.6	79.6	79.6	79.6
Exit Subcooling, deg F	3.9	4.0	3.9	4.0	3.9	4.0
Exit Enthalpy, Btu/lbm	33.4	33.0	33.0	33.0	33.0	32.7
Lines & Accumulator						
Pressure Drop, psi	?	0.20	0.20	0.20	0.20	0.11
Heat Transfer, Btu/hr	3035	2941	2935	2932	2940	3090
Expansion Valve						
Inlet Pressure, psia	?	166.7	166.5	166.7	166.5	164.3
Inlet Temperature, deg F	14.2	14.4	14.5	14.6	14.4	9.9
Inlet Subcooling, deg F	?	69.1	68.9	69.0	69.0	72.6
Inlet Enthalpy, Btu/lbm	14.2	14.4	14.4	14.4	14.4	13.1
Evaporator						
Inlet Pressure, psia	35.9	34.8	34.9	35.0	34.9	35.4
Inlet Temperature, deg F	-3.5	-4.9	-4.8	-4.6	-4.8	-4.2
Inlet Quality	0.050	0.055	0.055	0.056	0.055	0.040
Inlet Enthalpy, Btu/lbm	14.2	14.2	14.4	14.4	14.4	13.1
Exit Pressure, psia	34.6	34.3	34.4	34.6	34.4	35.0
Exit Temperature, deg F	-5.7	-5.6	-5.5	-5.2	-5.4	-4.7
Exit Quality	0.784	0.808	0.791	0.806	0.790	0.692
Exit Enthalpy, Btu/lbm	83.4	85.7	84.1	85.5	84.0	74.8
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	6.05	0.60	0.66	1.30	0.79	4.20
Heat Transfer, Btu/hr	3299	2941	3191	2958	3206	4586
Heating Capacity, Btu/hr*	14720	15229	14882	15157	14890	13713
Coefficient of Performance**	1.89	1.95	1.91	1.94	1.91	1.76

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-7. Test 107 Low Capacity Heating @ 47 Deg F

	Test	Lines Only	Ranco #25	Ranco #26	Ranco #30	Emprical
Compressor						
Mass Flow, lb/hr	350.1	344.1	345.2	335.5	343.8	332.0
Power Input, Watts	1934	1917	1910	1892	1907	1873
Heat Loss, Btu/hr	837	851	847	840	846	831
Inlet Pressure, psia	78.6	78.5	78.5	77.5	78.4	76.0
Inlet Temperature, deg F	38.2	38.2	38.2	37.5	38.1	36.4
Inlet Superheat, deg F	1.5	1.5	1.5	1.5	1.5	1.5
Inlet Enthalpy, Btu/lbm	108.1	108.1	108.1	108.1	108.1	108.0
Exit Pressure, psia	210.1	215.9	215.1	215.5	215.1	209.2
Exit Temperature, deg F	151.6	160.7	160.0	161.4	160.2	159.7
Exit Superheat, deg F	51.8	58.9	58.5	59.7	58.7	60.2
Exit Enthalpy, Btu/lbm	123.0	124.7	124.6	124.8	124.6	124.7
Lines & Reversing Valve						
Pressure Drop, psi	4.14	1.34	1.42	2.81	1.55	0.0
Heat Transfer, Btu/hr	428.8	0.0	366.3	37.6	332.5	981.9
Condenser						
Inlet Pressure, psia	205.9	214.6	213.7	212.7	213.5	209.2
Inlet Temperature, deg F	144.7	160.5	154.5	160.3	155.1	145.1
Inlet Superheat, deg F	46.4	59.1	53.5	59.5	54.1	45.6
Inlet Enthalpy, Btu/lbm	121.8	124.7	123.5	124.8	123.6	121.8
Exit Pressure, psia	204.2	213.7	212.8	211.9	212.6	208.4
Exit Temperature, deg F	95.8	99.1	98.7	98.4	98.7	97.3
Exit Subcooling, deg F	2.0	2.0	2.0	2.1	2.0	2.0
Exit Enthalpy, Btu/lbm	37.9	39.0	38.9	38.8	38.9	38.4
Lines & Accumulator						
Pressure Drop, psi	?	0.76	0.76	0.72	0.75	0.37
Heat Transfer, Btu/hr	3763	4423	4373	4175	4387	4390
Expansion Valve						
Inlet Pressure, psia	?	212.9	212.1	211.2	211.9	208.0
Inlet Temperature, deg F	60.0	56.1	56.3	56.7	56.0	52.8
Inlet Subcooling, deg F	?	44.7	44.2	43.5	44.5	46.3
Inlet Enthalpy, Btu/lbm	27.2	26.1	26.2	26.3	26.1	25.2
Evaporator						
Inlet Pressure, psia	83.0	80.2	80.3	80.3	80.4	81.0
Inlet Temperature, deg F	39.8	37.9	38.0	38.0	38.0	38.5
Inlet Quality	0.074	0.061	0.061	0.063	0.060	0.048
Inlet Enthalpy, Btu/lbm	27.2	26.1	26.2	26.3	26.1	25.2
Exit Pressure, psia	81.4	79.3	79.4	79.4	79.4	80.2
Exit Temperature, deg F	38.7	37.3	37.3	37.4	37.4	37.9
Exit Quality	0.877	0.855	0.847	0.858	0.846	0.814
Exit Enthalpy, Btu/lbm	97.5	95.3	94.6	95.6	94.5	91.8
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	2.79	0.74	0.88	1.95	1.09	4.21
Heat Transfer, Btu/hr	3726	4423	4686	4192	4679	5372
Capacity, Btu/hr*	29468	29849	29213	28835	29143	27670
Coefficient of Performance**	4.45	4.51	4.48	4.47	4.48	4.33

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-8. Test 109A Low Capacity Heating @ 35 Deg F

	Test	Lines Only	Ranco #25	Ranco #26	Ranco #30	Emprical
Compressor						
Mass Flow, lb/hr	285.2	258.8	259.6	253.9	258.6	245.6
Power Input, Watts	1766	1691	1702	1690	1690	1661
Heat Loss, Btu/hr	2156	2164	2178	2163	2163	2126
Inlet Pressure, psia	65.3	65.2	65.2	64.6	65.1	62.7
Inlet Temperature, deg F	29.2	29.2	29.4	28.7	29.1	27.1
Inlet Superheat, deg F	2.7	2.7	2.7	2.7	2.7	2.7
Inlet Enthalpy, Btu/lbm	107.4	107.4	107.4	107.4	107.4	107.2
Exit Pressure, psia	195.9	195.5	194.9	195.4	194.9	189.7
Exit Temperature, deg F	153.4	140.3	140.4	141.3	140.1	140.5
Exit Superheat, deg F	58.6	45.7	46.0	46.7	45.7	48.0
Exit Enthalpy, Btu/lbm	124.0	121.4	121.4	121.6	121.4	121.7
Lines & Reversing Valve						
Pressure Drop, psi	2.96	0.83	0.68	1.77	0.96	0.0
Heat Transfer, Btu/hr	499.8	0.0	302.9	35.0	283.4	906.2
Condenser						
Inlet Pressure, psia	192.9	194.7	194.1	193.7	193.9	189.7
Inlet Temperature, deg F	144.0	140.1	134.4	140.3	134.5	122.2
Inlet Superheat, deg F	50.3	45.8	40.3	46.3	40.4	29.7
Inlet Enthalpy, Btu/lbm	122.2	121.4	120.2	121.4	121.4	118.0
Exit Pressure, psia	191.1	194.1	193.5	192.7	193.3	189.1
Exit Temperature, deg F	88.2	89.3	89.0	88.9	88.9	87.4
Exit Subcooling, deg F	4.9	4.9	4.9	4.8	4.9	4.9
Exit Enthalpy, Btu/lbm	35.6	35.9	35.9	35.8	35.8	35.3
Lines & Accumulator						
Pressure Drop, psi	?	0.46	0.46	0.45	0.46	0.22
Heat Transfer, Btu/hr	3697	3230	3176	3257	3170	3302
Expansion Valve						
Inlet Pressure, psia	?	193.6	193.0	192.7	192.9	188.9
Inlet Temperature, deg F	44.3	46.8	47.5	45.2	47.3	41.4
Inlet Subcooling, deg F	?	47.1	46.3	48.4	46.4	50.8
Inlet Enthalpy, Btu/lbm	22.6	23.5	23.6	23.0	23.6	21.9
Evaporator						
Inlet Pressure, psia	69.4	66.4	66.5	66.7	66.6	67.3
Inlet Temperature, deg F	29.8	27.5	27.6	27.7	27.6	28.2
Inlet Quality	0.049	0.062	0.064	0.056	0.063	0.043
Inlet Enthalpy, Btu/lbm	22.6	23.5	23.6	23.0	23.6	21.9
Exit Pressure, psia	68.3	65.8	65.9	66.1	65.9	66.7
Exit Temperature, deg F	29.0	26.9	27.0	27.2	27.0	27.7
Exit Quality	0.854	0.865	0.857	0.860	0.856	0.809
Exit Enthalpy, Btu/lbm	94.2	94.9	94.2	94.5	94.2	90.1
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	2.98	0.54	0.64	1.42	0.79	4.04
Heat Transfer, Btu/hr	3766	3230	3429	3273	3426	4208
Capacity, Btu/hr*	24699	22108	21904	21737	21836	20288
Coefficient of Performance**	4.10	3.83	3.77	3.77	3.79	3.58

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

previously also does not fit the low capacity compressor map data as accurately as it fits the high capacity data, particularly at the lower end of the suction pressure range. This may be due to the effects of the relatively large clearance volume of the dual stroke compressor in the short stroke mode on volumetric efficiency, particularly at moderate to high pressure ratios. At low suction pressures the mass flow rate curves are close together and the power consumption curves cross over, making accurate curve fitting quite difficult. One approach to improving the match between measured and predicted compressor performance would be to split the compressor map into two or more regions and apply separate curve fits to each region.

7.2.2 Cooling Mode

The measured and predicted results for high capacity cooling at an outdoor ambient temperature of 95°F are shown in Table 7-9. Clearly the #26 valve model gives the most accurate estimate of capacity, refrigerant mass flow rate and energy efficiency ratio (EER). As in the mild temperature heating cases, reversing valve pressure drop has a greater impact on capacity and efficiency than internal heat transfer. The empirical model, which has the highest compressor suction side pressure drop and the highest internal heat transfer rate, underestimates capacity by 4.5%, but overestimates EER by about 1%.

Note that the voltage supplied to the electric expansion valve was controlled manually during these tests. While we attempted to maintain a low level of superheat in order to maximize performance, the temperatures and pressures in the refrigeration loop tended to "hunt" during the tests and the value of superheat shown in the tables is the average of several scans of the data over a period of 30 minutes or more. As a result, the levels of superheat obtained during several of the cooling tests were very low and it is possible that superheat was not actually obtained and the refrigerant exiting the evaporator may have been slightly "wet" in some of the cooling runs.

High capacity cooling results for 82°F outdoor ambient temperature are shown in Table 7-10. The #25 and #30 valves most closely match

Table 7-9. Test 111 High Capacity Cooling @ 95 Deg F

	Test	Lines Only	Ranco #25	Ranco #26	Ranco #30	Empri-cal
Compressor						
Mass Flow, lb/hr	736.8	770.0	769.9	741.6	764.2	723.0
Power Input, Watts	4389	4330	4328	4320	4319	4155
Heat Loss, Btu/hr	3767	3990	3988	3981	3980	3829
Inlet Pressure, psia	88.7	91.3	91.6	89.1	91.1	86.8
Inlet Temperature, deg F	44.8	46.6	46.6	45.0	46.2	43.5
Inlet Superheat, deg F	1.1	1.1	1.1	1.1	1.1	1.1
Inlet Enthalpy, Btu/lbm	108.6	108.8	108.8	108.7	108.8	108.5
Exit Pressure, psia	278.9	280.3	280.1	282.3	280.2	270.3
Exit Temperature, deg F	163.6	164.4	164.1	166.5	164.5	162.5
Exit Superheat, deg F	42.4	42.8	42.7	44.3	42.9	43.7
Exit Enthalpy, Btu/lbm	122.7	122.9	122.8	123.2	122.8	122.8
Line & Reversing Valve						
Pressure drop, psi	9.83	4.85	5.09	9.41	5.52	0.0
Heat Transfer, Btu/hr	1296.5	0.0	570.5	33.4	457.0	933.1
Condenser						
Inlet Pressure, psia	269.1	275.4	275.0	272.9	274.6	270.3
Inlet Temperature, deg F	153.7	163.4	159.9	164.4	160.7	156.6
Inlet Superheat, deg F	35.3	43.2	39.8	44.9	40.7	37.8
Inlet Enthalpy, Btu/lbm	120.9	122.8	122.1	123.1	122.2	121.5
Exit Pressure, psia	266.3	272.5	272.1	270.1	271.7	267.6
Exit Temperature, deg F	113.2	115.0	114.9	114.3	114.8	113.6
Exit Subcooling, deg F	4.4	4.4	4.4	4.4	4.4	4.4
Exit Enthalpy, Btu/lbm	43.5	44.1	44.0	43.8	44.0	43.6
Evaporator						
Inlet Pressure, psia	109.8	111.9	112.1	111.7	112.0	112.0
Inlet Temperature, deg F	56.4	57.6	57.7	57.5	57.7	57.6
Inlet Quality	0.204	0.212	0.211	0.209	0.210	0.206
Inlet Enthalpy, Btu/lbm	43.2	44.1	44.1	43.6	44.0	43.6
Exit Pressure, psia	92.6	92.7	93.1	94.3	93.4	96.0
Exit Temperature, deg F	47.0	47.1	46.1	47.2	46.7	48.3
Exit Superheat, deg F	0.9	0.8	0.0	0.0	0.0	0.0
Exit Enthalpy, Btu/lbm	108.7	108.8	108.1	108.6	108.2	107.2
Lines & Reversing Valve						
Pressure Drop, psi	3.82	1.16	1.60	5.14	2.37	9.16
Heat Transfer, Btu/hr	57.9	0.0	524.3	14.1	420.3	933.1
Capacity, Btu/hr*	48247	49830	49331	48052	49075	46069
EER, Btu/W-hr**	11.0	11.6	11.4	11.1	11.4	11.1

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-10. Test 114 High Capacity Cooling @ 82 Deg F

	Test	Lines Only	Ranco #25	Ranco #26	Ranco #30	Empri-cal
Compressor						
Mass Flow, lb/hr	758.0	779.1	776.6	749.1	772.6	733.3
Power Input, Watts	3750	3847	3838	3850	3837	3667
Heat Loss, Btu/hr	3340	3939	3929	3942	3939	3754
Inlet Pressure, psia	89.0	89.4	89.2	87.0	88.8	84.6
Inlet Temperature, deg F	44.4	44.7	44.5	43.0	44.3	41.5
Inlet Superheat, deg F	0.5	0.5	0.5	0.5	0.5	0.5
Inlet Enthalpy, Btu/lbm	108.5	108.5	108.6	108.4	108.5	108.3
Exit Pressure, psia	233.6	239.0	238.4	241.9	238.8	228.5
Exit Temperature, deg F	142.1	144.8	144.7	147.0	144.9	142.0
Exit Superheat, deg F	34.5	35.4	35.5	36.7	35.6	36.0
Exit Enthalpy, Btu/lbm	120.1	120.4	120.3	120.7	120.4	120.2
Lines & Reversing Valve						
Pressure Drop, psi	9.45	5.68	5.93	10.89	6.46	0.0
Heat Transfer, Btu/hr	277.4	0.0	495.5	26.2	392.5	770.7
Condenser						
Inlet Pressure, psia	224.2	233.3	232.5	231.0	232.3	228.5
Inlet Temperature, deg F	133.3	143.6	140.4	144.6	141.2	137.1
Inlet Superheat, deg F	28.8	36.1	33.1	37.8	34.0	31.1
Inlet Enthalpy, Btu/lbm	118.6	120.4	119.7	120.7	119.9	119.2
Exit Pressure, psia	221.0	229.7	228.9	227.6	228.8	225.2
Exit Temperature, deg F	98.3	101.0	100.9	100.4	100.8	99.6
Exit Subcooling, deg F	5.3	5.3	5.3	5.3	5.3	5.3
Exit Enthalpy, Btu/lbm	38.7	39.6	39.5	39.4	39.5	39.1
Evaporator						
Inlet Pressure, psia	106.9	109.9	109.8	109.8	110.0	110.0
Inlet Temperature, deg F	54.8	56.5	56.4	56.4	56.5	56.6
Inlet Quality	0.155	0.161	0.161	0.159	0.160	0.156
Inlet Enthalpy, Btu/lbm	38.7	39.6	39.5	39.4	39.5	39.1
Exit Pressure, psia	92.3	90.6	90.8	92.3	91.3	93.9
Exit Temperature, deg F	47.0	45.0	45.1	46.0	45.4	47.0
Exit Superheat, deg F	1.0	0.0	0.1	0.0	0.0	0.0
Exit Enthalpy, Btu/lbm	108.7	108.6	108.0	108.4	108.0	107.3
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	3.27	1.21	1.66	5.35	2.42	9.23
Heat Transfer, Btu/hr	277.4	0.0	459.0	12.0	363.9	771.5
Capacity, Btu/hr*	53102	53752	53133	51716	51971	49953
Coefficient of Performance**	14.16	13.97	13.85	13.43	13.81	13.62

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

the measured capacity. All of the models, except the empirical model, overestimate the compressor power input by about 100 W leading to an underestimate of the EER. The empirical model underestimates capacity by about 6% and compressor power input by 2.2% resulting in a 4% underestimate of the EER.

Table 7-11 summarizes the results for the high capacity cooling mode at an outdoor ambient of 105°F. All of the models overestimate the cooling capacity. The #26 valve gives the best match between predicted and measured EER. The predicted refrigerant mass flow rates exceed the measured mass flow rate by 9 to 16%. Note that for the #26 valve model the predicted compressor suction pressure is within 2 psi of the measured value and the predicted compressor discharge pressure is 13.3 psi higher than the measured value. Therefore we would expect the predicted refrigerant mass flow rate to be lower than the measured mass flow rate and the converse is true. Here again we observe inaccuracies in the map curve fit near the limits of the range of the input data. In this case we are near or above the upper limit of the range of compressor discharge pressure data.

Predicted and measured low capacity cooling performances at 76°F outdoor ambient temperature are compared in Table 7-12. All of the predicted capacities and energy efficiencies are lower than the measured values, with the no reversing valve case giving the best match to the measured capacity and EER. The #25 and #30 valve models give a good match to the measured refrigerant mass flow rate and give capacities within 3% of the measured value. Compressor suction pressure is predicted with good accuracy, but predicted compressor discharge pressures are about 7 psi higher than the measured value. Predicted compressor power consumption is within 1.2% of the measured value for these two cases. The #26 valve gives lower capacity and EER than the #25 and #30 valves, showing that refrigerant pressure drop penalizes performance more than internal heat transfer in this operating regime.

Table 7-13 compares predicted and measured low capacity cooling performance at an outdoor ambient temperature of 82°F. All of the predicted capacities and energy efficiency ratios are again less than the

Table 7-11. Test 117 High Capacity Cooling @ 106 Deg F

	Test	Lines Only	Ranco #25	Ranco #26	Ranco #30	Empri-cal
Compressor						
Mass Flow, lb/hr	638.5	737.5	738.9	710.3	733.4	693.8
Power Input, Watts	4532	4961	4963	4904	4948	4755
Heat Loss, Btu/hr	4716	5079	5081	5021	5066	4869
Inlet Pressure, psia	89.2	93.4	93.5	90.9	93.0	88.7
Inlet Temperature, deg F	50.6	53.3	53.4	51.8	53.1	50.3
Inlet Superheat, deg F	6.6	6.6	6.6	6.6	6.6	6.6
Inlet Enthalpy, Btu/lbm	109.6	109.9	109.9	109.8	109.9	109.6
Exit Pressure, psia	326.1	338.9	338.8	339.4	338.6	329.6
Exit Temperature, deg F	189.4	189.2	189.1	190.6	189.3	187.6
Exit Superheat, deg F	55.6	52.3	52.3	53.6	52.5	53.3
Exit Enthalpy, Btu/lbm	126.5	126.0	125.9	126.2	126.0	126.0
Lines & Reversing Valve						
Pressure Drop, psi	9.93	3.90	4.00	7.50	4.40	0.0
Heat Transfer, Btu/hr	532.0	0.0	629.8	40.4	512.7	109.2
Condenser						
Inlet Pressure, psia	316.1	335.0	334.8	331.9	334.2	329.6
Inlet Temperature, deg F	177.5	188.5	184.7	189.0	185.5	180.7
Inlet Superheat, deg F	46.3	52.6	48.8	53.9	49.7	46.4
Inlet Enthalpy, Btu/lbm	124.2	126.0	125.1	126.2	125.3	124.4
Exit Pressure, psia	314.3	333.9	333.7	330.9	333.1	327.5
Exit Temperature, deg F	113.7	118.6	118.4	117.8	118.3	117.0
Exit Subcooling, deg F	17.1	17.1	17.1	17.1	17.1	17.1
Exit Enthalpy, Btu/lbm	43.6	45.3	45.2	45.0	45.1	44.7
Evaporator						
Inlet Pressure, psia	108.2	103.6	104.1	104.0	104.1	105.4
Inlet Temperature, deg F	56.1	52.9	53.2	53.2	53.2	54.0
Inlet Quality	0.197	0.240	0.238	0.236	0.238	0.230
Inlet Enthalpy, Btu/lbm	42.5	45.3	45.2	45.0	45.1	44.7
Exit Pressure, psia	91.9	94.5	95.0	95.7	95.2	97.7
Exit Temperature, deg F	47.0	53.5	49.6	53.2	50.3	49.4
Exit Superheat, deg F	1.3	6.1	1.9	5.1	2.4	0.0
Exit Enthalpy, Btu/lbm	108.8	109.9	109.1	109.7	109.2	108.1
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	2.70	1.10	1.50	4.80	2.20	9.02
Heat Transfer, Btu/hr	510.8	0.0	569.2	24.1	466.2	1092.0
Capacity, Btu/hr*	42333	47643	47216	45956	47011	43987
Coefficient of Performance**	9.34	9.60	9.51	9.37	9.50	9.25

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-12. Test 118 Low Capacity Cooling @ 76 Deg F

	Test	Lines Only	Ranco #25	Ranco #26	Ranco #30	Empri-cal
Compressor						
Mass Flow, lb/hr	505.9	503.3	504.1	491.6	501.5	471.8
Power Input, Watts	1995	2018	2109	2020	2017	1960
Heat Loss, Btu/hr	2459	3099	3100	3103	3098	3010
Inlet Pressure, psia	91.1	91.6	91.6	90.7	91.4	87.2
Inlet Temperature, deg F	46.5	46.8	46.8	46.2	46.7	43.9
Inlet Superheat, deg F	1.2	1.2	1.2	1.2	1.2	1.2
Inlet Enthalpy, Btu/lbm	108.8	108.8	108.8	108.8	108.8	108.6
Exit Pressure, psia	185.4	192.3	192.2	194.3	192.4	186.5
Exit Temperature, deg F	114.1	114.9	114.9	116.1	115.0	113.7
Exit Superheat, deg F	23.2	21.4	21.5	21.9	21.5	22.4
Exit Enthalpy, Btu/lbm	116.5	116.2	116.3	116.5	116.4	116.4
Lines & Reversing Valve						
Pressure Drop, psi	5.88	2.93	3.07	5.81	3.34	0.0
Heat Transfer, Btu/hr	720.8	0.0	268.1	15.7	266.7	502.9
Condenser						
Inlet Pressure, psia	179.6	189.4	189.2	188.5	189.0	186.5
Inlet Temperature, deg F	105.6	114.2	111.6	114.8	112.0	108.5
Inlet Superheat, deg F	17.0	21.8	19.3	22.5	19.8	17.2
Inlet Enthalpy, Btu/lbm	115.1	116.3	115.8	116.5	115.9	115.3
Exit Pressure, psia	177.2	187.4	187.2	186.7	187.1	184.8
Exit Temperature, deg F	83.6	87.6	87.5	87.3	87.3	86.5
Exit Subcooling, deg F	4.1	4.1	4.1	4.1	4.1	4.1
Exit Enthalpy, Btu/lbm	34.2	35.4	35.4	35.3	35.3	35.1
Evaporator						
Inlet Pressure, psia	102.3	105.5	105.8	105.9	105.7	106.4
Inlet Temperature, deg F	51.8	54.0	54.2	54.2	54.1	54.5
Inlet Quality	0.111	0.119	0.118	0.117	0.118	0.114
Inlet Enthalpy, Btu/lbm	34.1	35.4	35.4	35.3	35.3	35.1
Exit Pressure, psia	91.6	92.1	92.4	93.2	92.5	95.2
Exit Temperature, deg F	46.6	47.0	46.1	47.0	46.1	47.8
Exit Superheat, deg F	1.1	1.1	0.0	0.4	0.0	0.0
Exit Enthalpy, Btu/lbm	108.8	108.8	108.3	108.8	108.4	107.5
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	0.48	0.52	0.73	2.48	1.08	7.90
Heat Transfer, Btu/hr	2.4	0.0	248.5	8.8	213.2	502.5
Capacity, Btu/hr*	37767	36946	36768	36103	36631	34172
Coefficient of Performance**	18.93	18.31	18.21	17.87	18.16	17.43

* Not corrected for blower work and compressor losses

** Does not include fan, blower and control power

Table 7-13. Test 115 Low Capacity Cooling @ 82 Deg F

	Test	Lines Only	Ranco #25	Ranco #28	Ranco #30	Emprical
Compressor						
Mass Flow, lb/hr	511.0	491.2	491.9	480.7	489.6	461.6
Power Input, Watts	2098	2106	2107	2104	2104	2041
Heat Loss, Btu/hr	2635	3234	3235	3232	3231	3135
Inlet Pressure, psia	93.9	93.3	93.3	92.5	93.1	89.1
Inlet Temperature, deg F	47.8	47.4	47.4	47.0	47.3	44.8
Inlet Superheat, deg F	0.8	0.8	0.8	0.8	0.8	0.8
Inlet Enthalpy, Btu/lbm	108.8	108.8	108.8	108.8	108.8	108.6
Exit Pressure, psia	200.9	208.8	208.7	210.6	208.8	203.4
Exit Temperature, deg F	118.9	121.5	121.5	122.6	121.6	120.4
Exit Superheat, deg F	22.4	22.1	22.1	22.6	22.2	22.9
Exit Enthalpy, Btu/lbm	116.7	116.9	116.9	117.0	116.9	116.9
Lines & Reversing Valve						
Pressure Drop, psi	5.83	2.58	2.71	5.15	2.95	0.0
Heat Transfer, Btu/hr	759.2	0.0	286.8	17.3	245.8	553.9
Condenser						
Inlet Pressure, psia	195.0	206.3	206.0	205.5	205.9	203.4
Inlet Temperature, deg F	110.4	120.9	118.0	121.2	118.5	114.7
Inlet Superheat, deg F	15.9	22.4	19.6	23.0	20.1	17.2
Inlet Enthalpy, Btu/lbm	115.2	116.9	116.3	117.0	116.4	115.7
Exit Pressure, psia	193.1	204.9	204.3	203.8	204.2	201.9
Exit Temperature, deg F	91.8	96.0	95.9	95.7	95.9	95.0
Exit Subcooling, deg F	1.9	1.9	1.9	1.9	1.9	1.9
Exit Enthalpy, Btu/lbm	36.7	38.0	38.0	37.9	38.0	37.7
Evaporator						
Inlet Pressure, psia	106.5	106.7	106.9	107.0	106.8	107.8
Inlet Temperature, deg F	54.3	54.7	54.8	54.9	54.8	55.3
Inlet Quality	0.121	0.148	0.147	0.146	0.147	0.142
Inlet Enthalpy, Btu/lbm	35.7	38.0	38.0	37.9	38.0	37.7
Exit Pressure, psia	94.7	93.8	94.0	94.8	94.1	97.0
Exit Temperature, deg F	48.3	47.5	47.1	47.6	47.2	48.9
Exit Superheat, deg F	0.8	0.5	0.0	0.0	0.0	0.0
Exit Enthalpy, Btu/lbm	108.8	108.8	108.3	108.8	108.3	107.4
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	0.78	0.49	0.69	2.34	1.02	7.83
Heat Transfer, Btu/hr	21.0	0.0	266.1	9.6	230.1	553.9
Capacity, Btu/hr*	37363	34786	34589	34051	34458	32178
Coefficient of Performance**	17.81	16.52	16.42	16.18	16.38	15.77

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

measured value. The predicted refrigerant mass flow rates are 4 to 10% less than the measured flow rate. The #25 and #30 valve cases predict the compressor suction pressure with good accuracy, but overestimate the compressor discharge pressure by about 8 psi. The #26 valve gives lower capacity and EER than the #25 and #30 valves, indicating that refrigerant pressure drop effects dominate heat transfer effects for this case.

Predicted and measured low capacity cooling performance at 95°F outdoor ambient temperature are presented in Table 7-14. The #25 and #30 valves give a reasonable match between measured and predicted capacity and EER, overestimating capacity by about 3% and EER by less than 2%. The #26 valve gives the best match to the measured capacity (within 2%) and EER (within 0.6%). All of the Ranco valve model cases overestimate the compressor suction pressure by 3 to 4 psi and the compressor discharge pressure by 5 to 7 psi.

7.3 SENSITIVITY ANALYSIS

7.3.1 Refrigerant Circuitry

In order to evaluate the sensitivity of the model to the number of refrigerant circuits in the heat exchangers, a limited number of computer runs were made using the number of circuits best matching the measured refrigerant pressure drop in the cooling mode for heating service and the number of circuits best matching the measured refrigerant pressure drop in the heating mode for cooling service with the #26 reversing valve model.

The high capacity heating results for outdoor ambient temperatures of 47°F and 17°F are shown in Tables 7-15 and 7-16, respectively. Surprisingly, the runs made using cooling circuitry in the high capacity heating mode more closely match the measured performance than the runs made using heating circuitry in many respects. The changes in capacity, refrigerant mass flow rate, and energy efficiency ratio are relatively small, less than 2%.

Table 7-17 compares predicted and measured low capacity heating performance at 47°F outdoor ambient temperature for the #26 reversing valve model with cooling circuitry and heating circuitry. Here again

Table 7-14. Test 113 Low Capacity Cooling @ 95 Deg F

	Test	Lines Only	Ranco #25	Ranco #26	Ranco #30	Emprical
Compressor						
Mass Flow, lb/hr	432.0	452.6	454.1	443.4	452.3	424.0
Power Input, Watts	2229	2265	2259	2253	2264	2172
Heat Loss, Btu/hr	3221	3093	3084	3076	3091	2965
Inlet Pressure, psia	92.6	96.1	96.3	95.3	96.1	92.0
Inlet Temperature, deg F	47.7	49.9	50.0	49.4	49.9	47.3
Inlet Superheat, deg F	1.5	1.5	1.5	1.5	1.5	1.5
Inlet Enthalpy, Btu/lbm	108.9	109.1	109.1	109.0	109.1	108.9
Exit Pressure, psia	241.1	245.8	245.6	247.0	245.7	240.5
Exit Temperature, deg F	138.7	141.5	141.2	142.3	141.5	140.5
Exit Superheat, deg F	28.7	30.0	29.8	30.5	30.0	30.7
Exit Enthalpy, Btu/lbm	119.0	119.4	119.3	119.5	119.4	119.4
Lines & Reversing Valve						
Pressure Drop, psi	6.24	1.93	2.04	3.89	2.22	0.0
Heat Transfer, Btu/hr	824.5	0.0	328.8	23.5	288.5	707.2
Condenser						
Inlet Pressure, psia	234.9	243.9	243.6	243.1	243.5	240.5
Inlet Temperature, deg F	128.6	141.1	137.5	141.2	138.1	132.8
Inlet Superheat, deg F	20.5	30.2	26.7	30.6	27.3	23.0
Inlet Enthalpy, Btu/lbm	117.1	119.4	118.6	119.4	118.7	117.7
Exit Pressure, psia	233.3	242.7	242.4	242.0	242.3	239.4
Exit Temperature, deg F	103.9	106.8	106.7	106.6	106.7	105.8
Exit Subcooling, deg F	3.7	3.7	3.7	3.7	3.7	3.7
Exit Enthalpy, Btu/lbm	40.5	41.4	41.4	41.4	41.4	41.1
Evaporator						
Inlet Pressure, psia	105.4	107.5	107.8	107.8	107.8	108.7
Inlet Temperature, deg F	54.2	55.2	55.3	55.3	55.3	55.8
Inlet Quality	0.178	0.188	0.187	0.186	0.186	0.182
Inlet Enthalpy, Btu/lbm	40.4	41.4	41.4	41.4	41.4	41.1
Exit Pressure, psia	93.4	96.5	96.8	97.3	96.9	99.5
Exit Temperature, deg F	47.7	50.1	48.8	50.1	48.9	50.5
Exit Superheat, deg F	1.0	1.4	0.0	1.0	0.0	0.0
Exit Enthalpy, Btu/lbm	108.8	109.1	108.5	109.0	108.5	107.2
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	0.78	0.41	0.58	1.98	0.86	7.60
Heat Transfer, Btu/hr	67.7	0.0	289.3	11.5	265.0	707.2
Capacity, Btu/hr*	29523	30630	30452	30012	30356	28033
Coefficient of Performance**	13.25	13.52	13.48	13.32	13.41	12.91

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-15. Test 103 High Capacity Heating @ 47 Deg F
Ranco #26 Reversing Valve

	Test	Heating Circuitry	Cooling Circuitry
Compressor			
Mass Flow, lbm/hr	532.2	530.1	523.8
Power Input, Watts	3565	3515	3484
Heat Loss, Btu/hr	2032	2040	2021
Inlet Pressure, psia	66.8	67.1	66.4
Inlet Temperature, deg F	28.4	28.8	28.0
Inlet Superheat, deg F	0.6	0.6	0.6
Inlet Enthalpy, Btu/lbm	107.1	107.2	107.2
Exit Pressure, psia	239.8	241.6	239.4
Exit Temperature, deg F	170.9	172.0	171.6
Exit Superheat, deg F	61.3	61.9	62.1
Exit Enthalpy, Btu/lbm	125.8	126.0	126.0
Lines & Reversing Valve			
Pressure drop, psi	6.49	5.96	5.88
Heat Transfer, Btu/hr	651.2	40.9	41.4
Condenser			
Inlet Pressure, psia	233.3	235.6	233.6
Inlet Temperature, deg F	163.8	170.5	170.2
Inlet Superheat, deg F	56.3	62.3	62.5
Inlet Enthalpy, Btu/lbm	124.6	125.9	125.9
Exit Pressure, psia	231.6	233.9	231.1
Exit Temperature, deg F	93.9	94.7	93.7
Exit Subcooling, deg F	13.1	13.1	13.1
Exit Enthalpy, Btu/lbm	37.4	37.6	37.3
Lines & Accumulator			
Pressure Drop, psi	?	1.93	1.88
Heat Transfer, Btu/hr	6912	6923	6655
Expansion Valve			
Inlet Pressure, psia	?	231.9	229.3
Inlet Temperature, deg F	50.3	50.6	50.8
Inlet Subcooling, deg F	?	56.6	55.4
Inlet Enthalpy, Btu/lbm	24.4	24.5	24.6
Evaporator			
Inlet Pressure, psia	77.6	74.4	77.0
Inlet Temperature, deg F	35.0	33.7	35.6
Inlet Quality	0.050	0.056	0.050
Inlet Enthalpy, Btu/lbm	24.4	24.5	24.6
Exit Pressure, psia	74.0	72.1	71.4
Exit Temperature, deg F	33.4	32.0	31.4
Exit Quality	0.852	0.849	0.853
Exit Enthalpy, Btu/lbm	94.6	94.1	94.4
Lines, Reversing Valve & Accumulator			
Pressure Drop, psi	7.23	5.05	5.00
Heat Transfer, Btu/hr	6652	6940	6800
Heating Capacity, Btu/hr*	46408	46815	46416
Coefficient of Performance**	3.81	3.90	3.90

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-16. Test 106 High Capacity Heating @ 17 Deg F
Ranco #26 Reversing Valve

	Test	Heating Circuitry	Cooling Circuitry
Compressor			
Mass Flow, lbm/hr	281.7	306.9	303.5
Power Input, Watts	2695	2669	2639
Heat Loss, Btu/hr	2680	2641	2612
Inlet Pressure, psia	43.2	42.2	41.7
Inlet Temperature, deg F	8.5	7.3	6.7
Inlet Superheat, deg F	3.1	3.1	3.1
Inlet Enthalpy, Btu/lbm	105.5	105.4	105.3
Exit Pressure, psia	194.2	194.8	192.9
Exit Temperature, deg F	176.6	165.6	165.0
Exit Superheat, deg F	82.4	71.3	71.3
Exit Enthalpy, Btu/lbm	128.6	126.4	126.4
Lines & Reversing Valve			
Pressure drop, psi	3.60	2.71	2.68
Heat Transfer, Btu/hr	591.6	52.6	51.0
Condenser			
Inlet Pressure, psia	190.6	192.1	190.2
Inlet Temperature, deg F	165.3	164.3	163.6
Inlet Superheat, deg F	72.5	70.9	70.9
Inlet Enthalpy, Btu/lbm	126.5	126.3	126.2
Exit Pressure, psia	188.6	191.3	189.2
Exit Temperature, deg F	85.2	86.3	85.5
Exit Subcooling, deg F	6.9	6.8	6.8
Exit Enthalpy, Btu/lbm	34.7	35.0	34.8
Lines & Accumulator			
Pressure Drop, psi	?	0.76	0.74
Heat Transfer, Btu/hr	4761	5162	5094
Expansion Valve			
Inlet Pressure, psia	?	190.5	188.4
Inlet Temperature, deg F	27.4	28.2	27.5
Inlet Subcooling, deg F	?	64.6	64.5
Inlet Enthalpy, Btu/lbm	17.8	18.2	18.0
Evaporator			
Inlet Pressure, psia	51.4	46.5	47.8
Inlet Temperature, deg F	14.0	9.0	10.4
Inlet Quality	0.053	0.058	0.052
Inlet Enthalpy, Btu/lbm	17.8	18.2	18.0
Exit Pressure, psia	49.4	45.3	44.7
Exit Temperature, deg F	12.0	7.6	7.1
Exit Quality	0.817	0.820	0.820
Exit Enthalpy, Btu/lbm	88.8	88.5	88.5
Lines, Reversing Valve & Accumulator			
Pressure Drop, psi	6.20	3.09	3.08
Heat Transfer, Btu/hr	4704	5183	5115
Heating Capacity, Btu/hr*	25860	28006	27756
Coefficient of Performance**	2.78	3.07	3.08

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-17. Test 107 Low Capacity Heating @ 47 Deg F
Ranco #26 Reversing Valve

	Test	Heating Circuitry	Cooling Circuitry
Compressor			
Mass Flow, lbm/hr	350.1	335.5	335.6
Power Input, Watts	1934	1892	1884
Heat Loss, Btu/hr	837	840	836
Inlet Pressure, psia	78.6	77.5	77.4
Inlet Temperature, deg F	38.2	37.5	37.4
Inlet Superheat, deg F	1.5	1.5	1.5
Inlet Enthalpy, Btu/lbm	108.1	108.1	108.1
Exit Pressure, psia	210.1	215.5	214.8
Exit Temperature, deg F	151.6	161.4	160.8
Exit Superheat, deg F	51.8	59.7	59.4
Exit Enthalpy, Btu/lbm	123.0	124.8	124.7
Lines & Reversing Valve			
Pressure drop, psi	4.14	2.81	2.82
Heat Transfer, Btu/hr	428.8	37.6	37.3
Condenser			
Inlet Pressure, psia	205.9	212.7	211.9
Inlet Temperature, deg F	144.7	160.3	159.7
Inlet Superheat, deg F	46.4	59.5	59.2
Inlet Enthalpy, Btu/lbm	121.8	124.8	124.6
Exit Pressure, psia	204.2	211.9	210.7
Exit Temperature, deg F	95.8	98.4	93.7
Exit Subcooling, deg F	2.0	2.1	2.1
Exit Enthalpy, Btu/lbm	37.9	38.8	38.6
Lines & Accumulator			
Pressure Drop, psi	?	0.72	0.72
Heat Transfer, Btu/hr	3763	4175	4388
Expansion Valve			
Inlet Pressure, psia	?	211.2	210.0
Inlet Temperature, deg F	60.0	56.7	54.1
Inlet Subcooling, deg F	?	43.5	45.7
Inlet Enthalpy, Btu/lbm	27.2	26.3	25.6
Evaporator			
Inlet Pressure, psia	83.0	80.3	81.5
Inlet Temperature, deg F	39.8	38.0	38.8
Inlet Quality	0.074	0.063	0.051
Inlet Enthalpy, Btu/lbm	27.2	26.3	25.6
Exit Pressure, psia	81.4	79.4	79.3
Exit Temperature, deg F	38.7	37.4	37.3
Exit Quality	0.877	0.858	0.851
Exit Enthalpy, Btu/lbm	97.5	95.6	94.9
Lines, Reversing Valve & Accumulator			
Pressure Drop, psi	2.79	1.95	1.95
Heat Transfer, Btu/hr	3726	4192	4404
Heating Capacity, Btu/hr*	29368	28835	28857
Coefficient of Performance**	4.45	4.47	4.49

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

the use of cooling circuitry in the heating mode actually improves the match between measured and predicted performance in many respects.

High capacity cooling performance with the #26 reversing valve model for both cooling and heating refrigerant circuitry at outdoor ambient temperatures of 82°F and 95°F are shown in Tables 7-18 and 7-19. Clearly the use in the cooling mode of refrigerant circuitry matching the refrigerant pressure drop in the heating mode results in a poorer match between predicted and measured performance in most respects. However, as in the heating cases, the changes in capacity, refrigerant mass flow rate and energy efficiency ratio are relatively small (in the 1 to 2% range).

It should be noted that the output of the transducers used to measure the refrigerant pressure drop in the heat exchangers had a poor signal-to-noise ratio, resulting in a relatively large standard deviation. This was not the result of inaccuracy of the instruments per se, but was primarily due to the pressure pulses associated with the reciprocating compressor. Since the differential pressure transducer used had a very high frequency response rate, these pulses were reflected in the transducer output. Therefore, accurately matching the measured and predicted refrigerant pressure drops is probably not justified in view of the limited accuracy of the data. Using the weighted average number of refrigerant circuits or the number of parallel circuits entering the condenser and exiting the evaporator should give acceptable results.

7.3.2 Accumulator Heat Exchanger Length

As noted previously, the length of tubing in the accumulator heat exchanger was determined empirically and varied over a fairly wide range. Table 7-20 shows the sensitivity of the model to accumulator heat exchanger length for high capacity heating at 17°F ambient temperature. While including the model obviously improves agreement between predicted and measured performance, the capacity, refrigerant mass flow rate and coefficient of performance are all rather insensitive to the length of tubing in the heat exchanger. Increasing the heat exchanger tubing length from 30 ft to 45 ft increases the amount of subcooling entering the expansion valve from 35.7°F to 64.6°F and lowers the evaporator inlet

Table 7-18. Test 111 High Capacity Cooling @ 95 Deg F
Ranco #26 Reversing Valve
Cooling Circuitry vs Heating Circuitry

	Test	Cooling Circuitry	Heating Circuitry
Compressor			
Mass Flow, lb/hr	736.8	741.6	756.6
Power Input, Watts	4389	4320	4379
Heat Loss, Btu/hr	3767	3981	4035
Inlet Pressure, psia	88.7	89.1	90.8
Inlet Temperature, deg F	44.8	45.0	46.1
Inlet Superheat, deg F	1.1	1.1	1.1
Inlet Enthalpy, Btu/lbm	108.6	108.7	108.7
Exit Pressure, psia	278.9	282.3	286.4
Exit Temperature, deg F	163.6	166.5	167.2
Exit Superheat, deg F	42.4	44.3	43.9
Exit Enthalpy, Btu/lbm	122.7	123.2	123.2
Line & Reversing Valve			
Pressure drop, psi	9.83	9.41	9.82
Heat Transfer, Btu/hr	1296.5	33.4	33.3
Condenser			
Inlet Pressure, psia	269.1	272.9	276.8
Inlet Temperature, deg F	153.7	164.4	165.1
Inlet Superheat, deg F	35.3	44.9	44.5
Inlet Enthalpy, Btu/lbm	120.9	123.1	123.1
Exit Pressure, psia	266.3	270.1	275.7
Exit Temperature, deg F	113.2	114.3	115.8
Exit Subcooling, deg F	4.4	4.4	4.5
Exit Enthalpy, Btu/lbm	43.5	43.8	44.3
Evaporator			
Inlet Pressure, psia	109.8	111.7	108.7
Inlet Temperature, deg F	56.4	57.5	55.8
Inlet Quality	0.204	0.209	0.220
Inlet Enthalpy, Btu/lbm	43.2	43.8	44.3
Exit Pressure, psia	92.6	94.3	96.0
Exit Temperature, deg F	47.0	47.2	48.4
Exit Superheat, deg F	0.9	0.0	0.0
Exit Enthalpy, Btu/lbm	108.7	108.6	108.7
Lines & Reversing Valve			
Pressure Drop, psi	3.82	5.14	5.23
Heat Transfer, Btu/hr	57.9	14.1	14.4
Capacity, Btu/hr*	48247	48052	48716
EER, Btu/W-hr**	11.0	11.1	11.1

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-19. Test 115 High Capacity Cooling @ 82 Deg F
Ranco #26 Reversing Valve
Cooling Circuitry vs Heating Circuitry

	Test	Cooling Circuitry	Heating Circuitry
Compressor			
Mass Flow, lb/hr	511.0	480.7	480.9
Power Input, Watts	2098	2104	2113
Heat Loss, Btu/hr	2635	3232	3245
Inlet Pressure, psia	93.9	92.5	92.8
Inlet Temperature, deg F	47.8	47.0	47.1
Inlet Superheat, deg F	0.8	0.8	0.8
Inlet Enthalpy, Btu/lbm	108.8	108.8	108.8
Exit Pressure, psia	200.9	210.6	212.2
Exit Temperature, deg F	118.9	122.6	123.2
Exit Superheat, deg F	22.4	22.6	22.6
Exit Enthalpy, Btu/lbm	116.7	117.0	117.1
Line & Reversing Valve			
Pressure drop, psi	5.83	6.15	5.12
Heat Transfer, Btu/hr	759.2	17.3	17.8
Condenser			
Inlet Pressure, psia	195.0	205.5	207.0
Inlet Temperature, deg F	110.4	121.2	121.8
Inlet Superheat, deg F	15.9	23.0	23.0
Inlet Enthalpy, Btu/lbm	115.2	117.0	117.0
Exit Pressure, psia	193.1	203.8	206.4
Exit Temperature, deg F	91.8	95.7	96.7
Exit Subcooling, deg F	1.9	1.9	1.9
Exit Enthalpy, Btu/lbm	36.7	37.9	38.2
Evaporator			
Inlet Pressure, psia	106.5	107.0	108.2
Inlet Temperature, deg F	54.3	54.9	54.4
Inlet Quality	0.121	0.146	0.151
Inlet Enthalpy, Btu/lbm	35.7	37.9	38.2
Exit Pressure, psia	94.7	94.8	95.1
Exit Temperature, deg F	48.3	47.6	47.8
Exit Superheat, deg F	0.8	0.0	0.0
Exit Enthalpy, Btu/lbm	108.8	108.8	108.8
Lines & Reversing Valve			
Pressure Drop, psi	0.78	2.34	2.34
Heat Transfer, Btu/hr	21.0	9.6	9.1
Capacity, Btu/hr*	37363	34051	33935
EER, Btu/W-hr**	17.81	16.18	16.06

* Not corrected for blower work or compressor losses
** Does not include fan, blower and control power

Table 7-20. Test 106 High Capacity Heating @ 17 Deg F
Ranco #26 Reversing Valve

			Accumulator Heat Exchanger Length - ft		
	Test	None	30.0	37.0	45.0
Compressor					
Mass Flow, lb/hr	281.7	310.2	307.1	307.1	306.9
Power Input, Watts	2695	2677	2670	2670	2669
Heat Loss, Btu/hr	2680	2649	2643	2643	2641
Inlet Pressure, psia	43.2	42.6	42.2	42.2	42.2
Inlet Temperature, deg F	8.5	7.8	7.3	7.3	7.3
Inlet Superheat, deg F	3.1	3.1	3.1	3.1	3.1
Inlet Enthalpy, Btu/lbm	105.5	105.4	105.4	105.4	105.4
Exit Pressure, psia	194.2	195.3	194.9	194.9	194.8
Exit Temperature, deg F	176.6	165.1	165.6	165.6	165.6
Exit Superheat, deg F	82.4	70.6	71.2	71.2	71.3
Exit Enthalpy, Btu/lbm	128.6	126.3	126.4	126.4	126.4
Lines & Reversing Valve					
Pressure Drop, psi	3.60	2.54	2.71	2.71	2.71
Heat Transfer, Btu/hr	591.6	50.3	51.0	51.0	52.6
Condenser					
Inlet Pressure, psia	190.6	192.8	192.1	192.1	192.1
Inlet Temperature, deg F	165.3	163.8	164.3	164.3	164.3
Inlet Superheat, deg F	72.5	70.2	70.9	70.9	70.9
Inlet Enthalpy, Btu/lbm	126.5	126.2	126.3	126.3	126.3
Exit Pressure, psia	188.6	191.8	191.4	191.4	191.3
Exit Temperature, deg F	85.2	86.3	86.2	86.2	86.3
Exit Subcooling, deg F	6.9	7.0	6.9	6.9	6.8
Exit Enthalpy, Btu/lbm	34.7	35.0	35.0	35.0	35.0
Lines & Accumulator					
Pressure Drop, psi	?	0.30	0.61	0.68	0.76
Heat Transfer, Btu/hr	4761	0.0	2728	3965	5162
Expansion Valve					
Inlet Pressure, psia	?	191.5	190.8	190.7	190.5
Inlet Temperature, deg F	27.4	86.3	56.2	42.1	28.2
Inlet Subcooling, deg F	?	6.8	36.7	49.8	64.6
Inlet Enthalpy, Btu/lbm	17.8	35.0	26.1	22.1	18.2
Evaporator					
Inlet Pressure, psia	51.4	46.9	46.8	46.7	46.5
Inlet Temperature, deg F	14.0	9.4	9.3	9.2	9.0
Inlet Quality	0.053	0.239	0.142	0.100	0.058
Inlet Enthalpy, Btu/lbm	17.8	35.0	26.1	22.1	18.2
Exit Pressure, psia	49.4	45.1	45.3	45.3	45.3
Exit Temperature, deg F	12.0	8.4	7.7	7.7	7.6
Exit Quality	0.817	1.000	0.905	0.862	0.820
Exit Enthalpy, Btu/lbm	88.8	105.4	96.4	92.4	88.5
Lines, Reversing Valve & Accumulator					
Pressure Drop, psi	6.20	2.56	3.14	3.12	3.09
Heat Transfer, Btu/hr	4704	21.1	2749	3986	5183
Capacity, Btu/hr*	25850	28272	28032	28032	28006
Coefficient of Performance**	2.76	3.09	3.08	3.08	3.07

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

quality from 14.2% to 5.8%, but the changes in capacity, refrigerant mass flow rate, compressor suction and discharge conditions, condenser inlet and exit conditions, and evaporator inlet and exit temperature are quite small. Therefore, it does not appear to be necessary to accurately predict the amount of subcooling entering the expansion valve for the purposes of these validation comparisons in order to obtain an accurate prediction of overall system performance, provided that the condenser exit subcooling and compressor inlet superheat are specified directly. If one of the expansion device models were being used this would not be the case.

7.4 REVERSING VALVE LEAKAGE

The effects of internal refrigerant leakage from the high pressure side of the reversing valve to the low pressure side were investigated for heating and cooling modes at both capacity levels. The #26 reversing valve model was used in all cases. For the heating cases the program was run with fixed compressor suction superheat. It was assumed that the refrigerant flow control mechanisms (expansion valve plus accumulator heat exchanger) would maintain a flooded evaporator in the heating mode regardless of the amount of reversing valve leakage. For the cooling mode both fixed compressor suction superheat and fixed evaporator exit superheat cases were considered since the unit was designed to operate with evaporator superheat in the cooling mode.

7.4.1 Heating Mode

Table 7-21 summarizes the results for the high capacity heating mode at 47°F outdoor ambient temperature for reversing valve leakage rates equal to 0%, 1%, 2%, 5%, and 10% of the compressor refrigerant mass flow rate. Valve manufacturers claim leakage rates in the 1% to 2% range, and our tests[6] confirm that for new valves, leakage rates in this range are typical. The effects of leakage on system capacity and COP are surprisingly small. A 2% leak rate reduces capacity by 1.09% and

Table 7-21 Test 103 High Capacity Heating @ 47 Deg F
Ranco #26 Reversing Valve

	Test	No Leak	1% Leak	2% Leak	5% Leak	10% Leak
Compressor						
Mass Flow, lbm/hr	532.2	530.1	532.6	535.2	543.0	556.3
Power Input, Watts	3565	3515	3515	3414	3513	3517
Heat Loss, Btu/hr	2032	2040	2040	2039	2038	2040
Inlet Pressure, psia	66.8	67.1	67.3	67.5	68.1	69.1
Inlet Temperature, deg F	28.4	28.6	28.8	28.9	29.4	30.2
Inlet Superheat, deg F	0.6	0.6	0.6	0.6	0.6	0.6
Inlet Enthalpy, Btu/lbm	107.1	107.2	107.2	107.2	107.3	107.4
Exit Pressure, psia	239.8	241.6	241.1	240.5	238.8	235.8
Exit Temperature, deg F	170.9	172.0	171.6	171.1	169.6	165.9
Exit Superheat, deg F	61.3	61.9	61.6	61.2	60.3	59.1
Exit Enthalpy, Btu/lbm	125.8	126.0	125.9	125.8	125.6	125.2
Lines & Reversing Valve						
Pressure drop, psi	6.49	5.96	5.98	60.2	6.11	6.27
Heat Transfer, Btu/hr	651.2	40.8	704.7	1359.7	3450.2	7012.3
Condenser						
Inlet Pressure, psia	233.3	235.6	235.1	234.4	232.7	229.5
Inlet Temperature, deg F	163.8	170.5	170.1	169.6	168.1	165.9
Inlet Superheat, deg F	56.3	62.3	62.0	61.7	60.8	59.5
Inlet Enthalpy, Btu/lbm	124.6	125.9	125.8	125.8	125.5	125.2
Exit Pressure, psia	231.6	233.9	233.4	232.8	231.0	228.0
Exit Temperature, deg F	93.9	94.7	94.4	94.3	93.7	92.8
Exit Subcooling, deg F	13.1	13.1	13.1	13.1	13.1	13.0
Exit Enthalpy, Btu/lbm	37.4	37.6	37.5	37.5	37.3	37.0
Lines & Accumulator						
Pressure Drop, psi	?	1.93	1.93	1.94	1.95	1.96
Heat Transfer, Btu/hr	6912	6923	6837	6783	6494	6035
Expansion Valve						
Inlet Pressure, psia	?	231.9	231.4	230.8	229.1	226.0
Inlet Temperature, deg F	50.3	50.6	50.6	50.6	51.2	52.1
Inlet Subcooling, deg F	?	56.6	56.3	56.2	55.0	53.1
Inlet Enthalpy, Btu/lbm	24.4	24.5	24.6	24.5	24.7	25.0
Evaporator						
Inlet Pressure, psia	77.6	74.4	74.6	74.7	75.3	76.0
Inlet Temperature, deg F	35.0	33.7	33.8	33.9	34.3	34.9
Inlet Quality	0.050	0.056	0.055	0.055	0.056	0.057
Inlet Enthalpy, Btu/lbm	24.4	24.5	24.6	24.5	24.7	25.0
Exit Pressure, psia	74.0	72.1	72.3	72.5	73.1	74.1
Exit Temperature, deg F	33.4	32.0	32.1	32.3	32.7	33.4
Exit Quality	0.852	0.849	0.846	0.843	0.838	0.822
Exit Enthalpy, Btu/lbm	94.6	94.1	93.9	93.6	93.0	91.9
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	7.23	5.05	5.03	5.02	4.98	4.92
Heat Transfer, Btu/hr	6652	6940	7518	8120	10263	13700
Heating Capacity, Btu/hr*	48408	46815	46571	46306	45514	44140
Coefficient of Performance**	3.81	3.90	3.88	3.86	3.80	3.68

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

COP by 1.03%. A 10% leak rate results in a reduction in capacity of 5.71% and a 5.64% decrease in COP. The cases with 1% to 2% leakage match the measured performance more accurately than the zero leakage example, supporting our own test data and the claim by the valve manufacturers that a leakage rate in this range is typical. The refrigerant mass flow rate shown is the flow rate through the compressor. The flow rate through the heat exchangers and expansion valve is equal to the compressor mass flow rate minus the leakage flow.

Table 7-22 presents the results for the high capacity heating mode at an outdoor ambient temperature of 17°F. The results are quite similar to the 47°F cases. A 2% leak rate reduces capacity by 1.35% and COP by 1.24%. The effects seem to be slightly greater than at 47°F.

Low capacity heating results at 47°F are shown in Table 7-23. The reduction in capacity due to reversing valve refrigerant leakage is quite similar to the high capacity results at the same ambient temperature. The impact on COP is slightly larger than for the high capacity cases.

The reversing valve leakage rate tests performed by Westinghouse[6] indicated that for the reversing valve used in the preprototype heat pump the leakage mass flow rate varies as approximately the 0.8 power of the absolute pressure ratio across the valve. Therefore reversing valve leakage will represent a larger fraction of the refrigerant mass flow rate at high pressure ratios due to a higher leak rate combined with a decrease in the refrigerant mass flow delivered by the compressor. If we assume that in the high capacity heating mode at 47°F the reversing valve leakage is equal to 2% of the compressor mass flow rate, and that the leak rate varies as the 0.8 power of the absolute pressure ratio, the relationship between the reversing valve leakage and ambient temperature shown in Table 7-24 is obtained.

7.4.2 Cooling Mode

The effects of reversing valve leakage on high capacity cooling performance at an outdoor ambient temperature of 95°F with fixed

Table 7-22. Test 106 High Capacity Heating @ 17 Deg F
Ranco #26 Reversing Valve

	Test	No Leak	1% Leak	2% Leak	5% Leak	10% Leak
Compressor						
Mass Flow, lb/hr	281.7	306.9	308.0	309.1	312.5	318.0
Power Input, Watts	2695	2669	2665	2666	2665	2680
Heat Loss, Btu/hr	2680	2641	2638	2639	2637	2652
Inlet Pressure, psia	43.2	42.2	42.3	42.4	42.7	43.1
Inlet Temperature, deg F	8.5	7.3	7.4	7.5	7.9	8.7
Inlet Superheat, deg F	3.1	3.1	3.1	3.1	3.1	3.1
Inlet Enthalpy, Btu/lbm	105.5	105.4	105.4	105.4	105.4	105.5
Exit Pressure, psia	194.2	194.8	194.4	194.1	193.0	191.1
Exit Temperature, deg F	176.6	165.6	165.1	164.8	163.5	162.2
Exit Superheat, deg F	82.4	71.3	70.8	70.6	69.8	69.2
Exit Enthalpy, Btu/lbm	128.6	126.4	126.4	126.4	126.1	125.9
Lines & Reversing Valve						
Pressure Drop, psi	3.60	2.71	2.72	2.72	2.74	2.78
Heat Transfer, Btu/hr	591.6	52.6	439.7	831.0	2019.5	4052.8
Condenser						
Inlet Pressure, psia	190.6	192.1	191.7	191.3	190.2	188.2
Inlet Temperature, deg F	165.3	164.3	163.8	163.4	162.2	160.8
Inlet Superheat, deg F	72.5	70.9	70.5	70.3	69.5	68.8
Inlet Enthalpy, Btu/lbm	126.5	126.3	126.2	126.1	126.1	125.7
Exit Pressure, psia	188.6	191.3	191.0	190.6	189.5	187.6
Exit Temperature, deg F	85.2	86.3	86.1	85.9	85.5	84.9
Exit Subcooling, deg F	6.9	6.8	6.9	6.9	6.9	6.8
Exit Enthalpy, Btu/lbm	34.7	35.0	35.0	34.9	34.8	34.6
Lines & Accumulator						
Pressure Drop, psi	?	0.76	0.76	0.76	0.76	0.76
Heat Transfer, Btu/hr	4761	5162	5126	5048	4805	4151
Expansion Valve						
Inlet Pressure, psia	?	190.5	190.2	189.9	188.7	186.9
Inlet Temperature, deg F	27.4	28.2	28.0	28.4	29.7	35.0
Inlet Subcooling, deg F	?	64.6	64.7	64.2	62.5	56.5
Inlet Enthalpy, Btu/lbm	17.8	18.2	18.1	18.2	18.6	20.1
Evaporator						
Inlet Pressure, psia	51.4	46.5	46.6	46.6	46.8	47.2
Inlet Temperature, deg F	14.0	9.0	9.0	9.1	9.3	9.7
Inlet Quality	0.053	0.058	0.057	0.058	0.061	0.076
Inlet Enthalpy, Btu/lbm	17.8	18.2	18.1	18.2	18.6	20.1
Exit Pressure, psia	49.4	45.3	45.3	45.4	45.7	46.1
Exit Temperature, deg F	12.0	7.6	7.7	7.8	8.1	8.6
Exit Quality	0.817	0.820	0.816	0.813	0.806	0.803
Exit Enthalpy, Btu/lbm	88.8	88.5	88.1	87.9	87.2	87.0
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	6.20	3.09	3.08	3.07	3.04	2.99
Heat Transfer, Btu/hr	4704	5183	5588	5952	7048	8637
Capacity, Btu/hr*	25860	28008	27818	27629	27053	26087
Coefficient of Performance**	2.78	3.07	3.06	3.04	2.97	2.85

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-23. Test 1Ø7 Low Capacity Heating @ 47 Deg F
Ranco #26 Reversing Valve

	Test	No Leak	1% Leak	2% Leak	5% Leak	1Ø% Leak
Compressor						
Mass Flow, lb/hr	35Ø.1	335.5	337.2	338.9	344.4	352.8
Power Input, Watts	1934	1892	19Ø1	1897	191Ø	1919
Heat Loss, Btu/hr	837	84Ø	843	842	847	851
Inlet Pressure, psia	78.6	77.5	77.6	77.7	78.2	78.7
Inlet Temperature, deg F	38.2	37.5	37.5	37.6	37.9	38.3
Inlet Superheat, deg F	1.5	1.5	1.5	1.5	1.5	1.5
Inlet Enthalpy, Btu/lbm	1Ø8.1	1Ø8.1	1Ø8.1	1Ø8.1	1Ø8.1	1Ø8.1
Exit Pressure, psia	21Ø.1	215.5	215.1	214.7	213.5	211.1
Exit Temperature, deg F	151.6	161.4	161.3	16Ø.5	159.8	157.9
Exit Superheat, deg F	51.8	59.7	59.7	59.2	58.8	57.7
Exit Enthalpy, Btu/lbm	123.Ø	124.8	124.8	124.7	124.6	124.3
Lines & Reversing Valve						
Pressure Drop, psi	4.14	2.81	2.83	2.84	2.89	2.96
Heat Transfer, Btu/hr	428.8	37.6	458.2	882.3	2181.7	4419.9
Condenser						
Inlet Pressure, psia	2Ø5.9	212.7	212.3	211.8	21Ø.6	2Ø8.2
Inlet Temperature, deg F	144.7	16Ø.3	16Ø.2	159.5	158.7	156.8
Inlet Superheat, deg F	46.4	59.5	59.6	59.1	58.7	57.6
Inlet Enthalpy, Btu/lbm	121.8	124.8	124.7	124.6	124.5	124.2
Exit Pressure, psia	2Ø4.2	211.9	211.5	211.Ø	2Ø9.8	2Ø7.4
Exit Temperature, deg F	95.8	98.4	98.3	98.1	97.7	97.Ø
Exit Subcooling, deg F	2.Ø	2.1	2.Ø	2.Ø	2.Ø	1.9
Exit Enthalpy, Btu/lbm	37.9	38.8	38.7	38.7	38.6	38.3
Lines & Accumulator						
Pressure Drop, psi	?	Ø.72	Ø.72	Ø.72	Ø.72	Ø.73
Heat Transfer, Btu/hr	3763	4175	4361	4253	4Ø56	3811
Expansion Valve						
Inlet Pressure, psia	?	211.2	21Ø.7	21Ø.3	2Ø9.1	2Ø6.7
Inlet Temperature, deg F	6Ø.Ø	56.7	54.5	55.2	56.2	56.7
Inlet Subcooling, deg F	?	43.5	45.6	44.7	43.3	41.9
Inlet Enthalpy, Btu/lbm	27.2	26.3	25.7	25.9	26.2	26.3
Evaporator						
Inlet Pressure, psia	83.Ø	8Ø.3	8Ø.4	8Ø.5	8Ø.9	81.4
Inlet Temperature, deg F	39.8	38.Ø	38.1	38.1	38.4	38.7
Inlet Quality	Ø.Ø74	Ø.Ø63	Ø.Ø55	Ø.Ø57	Ø.Ø6Ø	Ø.Ø61
Inlet Enthalpy, Btu/lbm	27.2	26.3	25.7	25.9	26.2	26.3
Exit Pressure, psia	81.4	79.4	79.6	79.7	8Ø.1	8Ø.6
Exit Temperature, deg F	38.7	37.4	37.5	37.5	37.8	38.2
Exit Quality	Ø.877	Ø.858	Ø.848	Ø.847	Ø.841	Ø.827
Exit Enthalpy, Btu/lbm	97.5	95.6	94.7	94.6	94.1	93.Ø
Lines, Reversing Valve & Accumulator						
Pressure Drop, psi	2.79	1.95	1.95	1.94	1.94	1.93
Heat Transfer, Btu/hr	3726	4192	4839	5291	643Ø	8634
Capacity, Btu/hr*	29368	28835	28694	2853Ø	281Ø7	27256
Coefficient of Performance**	4.45	4.47	4.42	4.41	4.31	4.16

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-24. Projected Reversing Valve Leak Rates for the Preprototype Heat Pump in Heating Service

	Compressor Mass Flow Rate lbm/hr	Leakage Mass Flow Rate lbm/hr	Percent Leakage
High Capacity - 47°F	532.2	10.6	2.0%*
High Capacity - 35°F	447.7	11.1	2.5%
High Capacity - 17°F	218.7	12.7	3.8%
High Capacity - 0°F	157.8	14.4	9.1%
Low Capacity - 47°F	350.1	8.5	2.4%
Low Capacity - 35°F	285.2	9.2	3.2%

* Assumed value.

compressor suction superheat are shown in Table 7-25. A 2% leak rate results in a capacity reduction of 1.34% and a decrease in COP of 1.72%. The effects appear to be slightly greater than in heating operation. Low capacity cooling results for 82°F outdoor ambient temperature and fixed compressor suction superheat are shown in Table 7-26 and are quite similar to the 95°F high capacity cooling results. Based on the assumptions used to generate Table 7-24, the projected reversing valve leakage rates are 1.3% of the compressor mass flow rate for the 95°F high capacity cooling case and 1.4% for the 82°F low capacity cooling case.

7.4.3 Discussion of Reversing Valve Leakage Effects

The results for all of the leakage cases run with fixed compressor suction superheat are summarized in Table 7-27. The percentage loss in capacity and COP due to leakage is generally 50% to 70% of the leak rate. Part of the reason for the relatively small impact of leakage is that the refrigerant mass flow rate through the compressor is higher when leakage occurs. Since the heat exchanger loading is reduced by refrigerant leakage in the reversing valve, the average evaporator

Table 7-25. Test 111 High Capacity Cooling @ 95 Deg F
Ranco #26 Reversing Valve

	Test	No Leak	1% Leak	2% Leak	5% Leak	10% Leak
Compressor						
Mass Flow, lb/hr	736.8	741.6	745.1	749.0	760.8	780.0
Power Input, Watts	4389	4320	4335	4337	4332	4354
Heat Loss, Btu/hr	3767	3981	3995	3997	3992	4012
Inlet Pressure, psia	88.7	89.1	89.5	89.8	90.8	92.4
Inlet Temperature, deg F	44.8	45.0	45.3	45.5	46.1	47.1
Inlet Superheat, deg F	1.1	1.1	1.1	1.1	1.1	1.1
Inlet Enthalpy, Btu/lbm	108.6	108.7	108.7	108.7	108.7	108.8
Exit Pressure, psia	278.9	282.3	282.0	281.6	280.6	278.8
Exit Temperature, deg F	163.6	166.5	166.4	166.1	165.0	163.8
Exit Superheat, deg F	42.4	44.3	44.3	44.1	43.3	42.6
Exit Enthalpy, Btu/lbm	122.7	123.2	123.2	123.1	122.9	122.7
Lines & Reversing Valve						
Pressure Drop, psi	9.83	9.41	9.45	9.50	9.60	9.83
Heat Transfer, Btu/hr	1296.5	33.4	951.0	1876.6	4708.0	9604.8
Condenser						
Inlet Pressure, psia	269.1	272.9	272.6	272.2	271.0	269.0
Inlet Temperature, deg F	153.7	164.4	164.4	164.0	163.0	161.7
Inlet Superheat, deg F	35.3	44.9	44.9	44.7	44.0	43.3
Inlet Enthalpy, Btu/lbm	120.9	123.1	123.1	123.1	122.9	122.7
Exit Pressure, psia	266.3	270.1	269.8	268.0	268.4	266.4
Exit Temperature, deg F	113.2	114.3	114.2	114.2	113.8	113.2
Exit Subcooling, deg F	4.4	4.4	4.4	4.4	4.4	4.4
Exit Enthalpy, Btu/lbm	43.5	43.8	43.8	43.8	43.7	43.5
Evaporator						
Inlet Pressure, psia	109.8	111.7	111.8	111.8	112.0	112.2
Inlet Temperature, deg F	56.4	57.5	57.5	57.5	57.7	57.8
Inlet Quality	0.204	0.209	0.209	0.209	0.207	0.204
Inlet Enthalpy, Btu/lbm	43.2	43.8	43.8	43.8	43.7	43.5
Exit Pressure, psia	92.6	94.3	94.6	94.9	95.9	97.4
Exit Temperature, deg F	47.0	47.2	47.5	47.7	48.2	49.2
Exit Superheat, deg F	0.9	0.0	0.0	0.0	0.0	0.0
Exit Enthalpy, Btu/lbm	108.7	108.6	108.5	108.4	108.0	107.3
Lines & Reversing Valve						
Pressure Drop, psi	3.82	5.14	5.12	5.11	5.08	5.02
Heat Transfer, Btu/hr	57.9	14.1	931.5	1857.9	4690.2	9586.4
Capacity, Btu/hr*	48247	48052	47730	47410	46540	44763
Coefficient of Performance**	11.0	11.1	11.0	10.9	10.7	10.3

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-26. Test 115 Low Capacity Cooling @ 82 Deg F
Ranco #26 Reversing Valve

	Test	No Leak	1% Leak	2% Leak	5% Leak	10% Leak
Compressor						
Mass Flow, lb/hr	511.0	480.7	481.6	484.4	494.7	507.2
Power Input, Watts	2098	2104	2103	2106	2126	2125
Heat Loss, Btu/hr	2635	3232	3230	3235	3264	3263
Inlet Pressure, psia	93.9	92.5	92.5	92.8	93.9	95.0
Inlet Temperature, deg F	47.8	47.0	47.0	47.1	47.8	48.5
Inlet Superheat, deg F	0.8	0.8	0.8	0.8	0.8	0.8
Inlet Enthalpy, Btu/lbm	108.8	108.8	108.8	108.8	108.9	108.9
Exit Pressure, psia	200.9	210.6	210.3	210.2	209.9	208.9
Exit Temperature, deg F	118.9	122.6	122.4	122.3	122.0	121.1
Exit Superheat, deg F	22.4	22.6	22.5	22.4	22.3	21.7
Exit Enthalpy, Btu/lbm	116.7	117.0	117.0	117.0	116.9	116.8
Lines & Reversing Valve						
Pressure Drop, psi	5.83	5.15	5.15	5.18	5.30	5.40
Heat Transfer, Btu/hr	759.2	17.3	580.6	1150.8	2909.6	5939.7
Condenser						
Inlet Pressure, psia	195.0	205.5	205.2	205.0	204.6	203.5
Inlet Temperature, deg F	110.4	121.2	121.0	120.9	120.6	119.6
Inlet Superheat, deg F	15.9	23.0	22.9	22.8	22.7	22.1
Inlet Enthalpy, Btu/lbm	115.2	117.0	117.0	117.0	116.9	116.7
Exit Pressure, psia	193.1	203.8	203.5	203.4	203.0	202.0
Exit Temperature, deg F	91.8	95.7	95.7	95.6	95.4	95.1
Exit Subcooling, deg F	1.9	1.9	1.8	1.8	2.0	1.9
Exit Enthalpy, Btu/lbm	36.7	37.9	37.9	37.9	37.8	37.7
Evaporator						
Inlet Pressure, psia	106.5	107.0	106.8	107.0	107.6	107.9
Inlet Temperature, deg F	54.3	54.9	54.8	54.8	55.2	55.4
Inlet Quality	0.121	0.146	0.147	0.146	0.144	0.142
Inlet Enthalpy, Btu/lbm	35.7	37.9	37.9	37.9	37.8	37.7
Exit Pressure, psia	94.7	94.8	94.9	95.1	96.2	97.3
Exit Temperature, deg F	48.3	47.6	47.6	47.8	48.5	49.1
Exit Superheat, deg F	0.8	0.0	0.0	0.0	0.0	0.0
Exit Enthalpy, Btu/lbm	108.8	108.8	108.7	108.6	108.4	108.0
Lines & Reversing Valve						
Pressure Drop, psi	0.78	2.34	2.33	2.33	2.33	23.1
Heat Transfer, Btu/hr	21.0	9.6	573.0	1142.5	2900.9	5931.1
Capacity, Btu/hr*	37363	34051	33740	33574	33177	32089
Coefficient of Performance**	17.81	16.18	16.04	15.94	15.61	15.10

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-27 Effect of Reversing Valve Refrigerant Leakage On Capacity and Energy Efficiency

		1% Leak	2% Leak	5% Leak	10% Leak
			% Decrease		
47°F High Capacity Heating					
Reference Capacity, Btu/hr	46815	0.52	1.09	2.78	5.71
Reference COP	3.90	0.51	1.03	2.56	5.64
17°F High Capacity Heating	28006	0.67	1.35	3.40	6.85
Reference Capacity, Btu/hr	3.07	0.53	1.24	3.26	7.23
Reference COP					
47°F Low Capacity Heating					
Reference Capacity, Btu/hr	28835	0.49	1.06	2.52	5.48
Reference COP	4.47	0.96	1.32	3.44	6.81
95°F High Capacity Cooling					
Reference Capacity, Btu/hr	48052	0.67	1.34	3.15	6.84
Reference EER, Btu/W-hr	11.1	1.01	1.72	3.41	7.57
82°F Low Capacity Cooling					
Reference Capacity, Btu/hr	34051	0.91	1.40	2.57	5.76
Reference EER, Btu/W-hr	16.2	0.87	1.49	4.57	6.69

pressure rises slightly and the average condenser temperature drops somewhat, resulting in a decrease in the pressure ratio across the compressor, which results in an increase in compressor mass flow rate. This increase is obviously always less than the leak rate, so the flow through the heat exchangers and expansion valve is always reduced by refrigerant leakage in the reversing valve.

For a system with flooded evaporator operation the assumption of constant compressor suction superheat is reasonable for moderate leak rates. For air conditioning service, where the evaporator is not flooded, the assumption of fixed compressor suction superheat is somewhat unrealistic. We would expect the expansion device to maintain constant evaporator exit superheat and that as the leak rate increased, the compressor suction superheat would increase due to mixing of the relatively cold stream entering the accumulator with the hot gas leaking from the compressor discharge stream. While the logic of the ORNL code is based on fixed compressor suction conditions, we can simulate this operating mode by varying the amount of compressor suction superheat to maintain a constant evaporator exit superheat. This method is highly iterative and only a single example, high capacity cooling at 95°F was considered. The results are shown in Table 7-28. As expected, the effect of leakage is greater than for the fixed compressor suction superheat case, but remains relatively small. A 5% leak rate reduces capacity and EER by 3.2%.

7.5 FIN PATTERNATION EFFECTS

The ORNL code utilizes the McQuiston[2] correlation for air side heat transfer in plate-finned-tube air-to-refrigerant heat exchangers with smooth fins and adds correction factors for wavy and louvered fins. These corrections are in the form of a constant multiplying factor which is independent of heat exchanger geometry or air properties. The air side heat transfer coefficient is multiplied by 1.45 for wavy fins[3] or 1.75 for louvered fins.[4] The correlation added by Westinghouse for corrugated fins also generates a correction factor which is applied to the smooth fin correlation, which in turn is a function of both heat exchanger geometry and air velocity and transport properties.

Table 7-28 Test 111 High Capacity Cooling @ 95 Deg F
Ranco #26 Reversing Valve
Fixed Evaporator Exit Superheat

	No Leak	1% Leak	2% Leak	5% Leak	10% Leak
Compressor					
Mass Flow, lb/hr	722.7	723.3	725.8	733.5	745.5
Power Input, Watts	4332	4330	4341	4334	4381
Heat Loss, Btu/hr	3992	3990	4000	3993	4037
Inlet Pressure, psia	88.5	88.7	89.0	89.9	91.5
Inlet Temperature, deg F	53.8	54.4	55.5	58.7	64.5
Inlet Superheat, deg F	10.0	10.8	11.6	14.2	19.0
Inlet Enthalpy, Btu/lbm	110.2	110.4	110.5	111.1	112.0
Exit Pressure, psia	281.6	281.0	280.6	279.5	277.5
Exit Temperature, deg F	175.4	175.8	176.5	177.9	181.5
Exit Superheat, deg F	53.4	54.0	54.8	56.5	60.7
Exit Enthalpy, Btu/lbm	125.2	125.3	125.4	125.8	126.6
Line & Reversing Valve					
Pressure drop, psi	9.27	9.27	9.31	9.43	9.64
Heat Transfer, Btu/hr	35.4	941.3	1856.4	4648.6	9475.8
Condenser					
Inlet Pressure, psia	272.3	271.7	271.3	270.0	267.9
Inlet Temperature, deg F	173.5	173.9	174.6	175.9	179.6
Inlet Superheat, deg F	54.1	54.7	55.5	57.2	61.5
Inlet Enthalpy, Btu/lbm	125.1	125.3	125.4	125.7	126.6
Exit Pressure, psia	269.7	269.1	268.7	267.6	265.5
Exit Temperature, deg F	113.6	113.4	113.4	113.0	112.4
Exit Subcooling, deg F	5.0	5.0	5.0	5.0	5.0
Exit Enthalpy, Btu/lbm	43.6	43.5	43.5	43.4	43.2
Evaporator					
Inlet Pressure, psia	110.1	109.9	109.9	110.1	110.3
Inlet Temperature, deg F	56.6	56.5	56.5	56.6	56.7
Inlet Quality	0.209	0.209	0.290	0.207	0.204
Inlet Enthalpy, Btu/lbm	43.6	43.5	43.5	43.4	43.2
Exit Pressure, psia	93.6	93.7	94.0	95.0	96.5
Exit Temperature, deg F	55.2	55.4	55.6	56.2	57.0
Exit Superheat, deg F	8.4	8.5	8.5	8.5	8.4
Exit Enthalpy, Btu/lbm	110.2	110.2	110.2	110.3	110.4
Lines & Reversing Valve					
Pressure Drop, psi	5.10	5.07	5.06	5.03	4.97
Heat Transfer, Btu/hr	14.5	920.0	1835.0	4626.4	9452.9
Capacity, Btu/hr*	48128	47724	47429	46583	45041
EER, Btu/W-hr**	11.11	11.02	10.93	10.75	10.28

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Two independent correlations are used by the ORNL code for air-side pressure drop in plate-finned-tube heat exchangers. The first (from Reference 2) is used exclusively for smooth fins. The second is used for wavy and louvered fins. The correlation used for louvered fins is the wavy fin correlation[5] with a constant multiplying factor of 1.1[4] applied. The Westinghouse correlation for corrugated fins generates a multiplying factor which is applied to the smooth fin correlation. This multiplying factor is a function of the number of fin patterns per tube row in the air flow direction, the fin pattern depth and the air velocity at the coil inlet.

To determine the sensitivity of the model to the type of fin pattern selected, the program was run with all four options for two cases, high capacity heating at 47°F ambient temperature and high capacity cooling at 95°F ambient temperature. For the corrugated fin pattern cases the actual fin geometry of two patterns per tube row with a pattern depth of 0.046 in. was used. The terms wavy fin and corrugated fin have been applied to distinguish the ORNL model (wavy fins), which assumes fixed multiplying factors for fin patternation, from the Westinghouse model (corrugated fins), which calculates the multiplying factor as a function of fin geometry, air properties, and air side velocity. The fin pattern used in the single fin model tests is similar to the "wavy" pattern used by the two leading OEM suppliers of air to refrigerant heat exchangers. However, one of the most commonly used "wavy" patterns has three patterns per tube row with a pattern depth of 0.038 in., giving a greater increase in both air side heat transfer and air side pressure drop than the fin pattern used in the preprototype. The ORNL report[1] does not give detailed information on the fin geometry used as a basis for the multiplying factors used.

The results for the high capacity heating case are presented in Table 7-29. As expected, the smooth fin case gives the lowest capacity and COP and the louvered fin case the highest. The performance predicted for the corrugated fin case was higher than for the smooth fin case,

Table 7-29 Test 103 High Capacity Heating @ 47 Deg F
Ranco #26 Reversing Valve

	Test	Smooth Fins	Wavy Fins	Louvered Fins	Corrugated Fins
Compressor					
Mass Flow, lbm/hr	532.2	514.7	530.1	537.1	519.2
Power Input, Watts	3565	3562	3515	3508	3551
Heat Loss, Btu/hr	2032	2066	2040	2035	2060
Inlet Pressure, psia	66.8	66.2	67.1	67.6	66.5
Inlet Temperature, deg F	28.4	27.9	28.6	29.0	28.1
Inlet Superheat, deg F	0.6	0.6	0.6	0.6	0.6
Inlet Enthalpy, Btu/lbm	107.1	107.1	107.2	107.2	107.2
Exit Pressure, psia	239.8	249.9	241.6	238.5	247.7
Exit Temperature, deg F	170.9	177.1	172.0	170.2	175.7
Exit Superheat, deg F	61.3	64.4	61.9	61.0	63.7
Exit Enthalpy, Btu/lbm	125.8	126.7	126.0	125.7	126.5
Lines & Reversing Valve					
Pressure drop, psi	6.49	5.50	5.96	6.16	5.62
Heat Transfer, Btu/hr	651.2	43.2	40.8	40.3	42.6
Condenser					
Inlet Pressure, psia	233.3	244.4	235.6	232.3	242.0
Inlet Temperature, deg F	163.8	175.8	170.5	168.7	174.3
Inlet Superheat, deg F	56.3	64.7	62.3	61.5	64.0
Inlet Enthalpy, Btu/lbm	124.6	126.7	125.9	125.7	126.5
Exit Pressure, psia	231.6	242.9	233.9	230.5	240.4
Exit Temperature, deg F	93.9	97.5	94.7	93.6	96.7
Exit Subcooling, deg F	13.1	13.1	13.1	13.0	13.1
Exit Enthalpy, Btu/lbm	37.4	38.5	37.6	37.3	38.2
Lines & Accumulator					
Pressure Drop, psi	?	1.83	1.93	1.98	1.86
Heat Transfer, Btu/hr	6912	7287	6923	6828	7191
Expansion Valve					
Inlet Pressure, psia	?	241.0	231.9	228.5	238.6
Inlet Temperature, deg F	50.3	49.8	50.6	50.7	50.0
Inlet Subcooling, deg F	?	60.2	56.6	55.3	59.2
Inlet Enthalpy, Btu/lbm	24.4	24.3	24.5	24.6	24.4
Evaporator					
Inlet Pressure, psia	77.6	73.2	74.4	75.0	73.6
Inlet Temperature, deg F	35.0	32.8	33.7	34.1	33.1
Inlet Quality	0.050	0.056	0.056	0.055	0.056
Inlet Enthalpy, Btu/lbm	24.4	24.3	24.5	24.6	24.4
Exit Pressure, psia	74.0	71.1	72.1	72.7	71.4
Exit Temperature, deg F	33.4	31.2	32.0	32.4	31.4
Exit Quality	0.852	0.837	0.849	0.853	0.840
Exit Enthalpy, Btu/lbm	94.6	93.0	94.1	94.5	93.3
Lines, Reversing Valve & Accumulator					
Pressure Drop, psi	7.23	4.85	5.05	5.13	4.91
Heat Transfer, Btu/hr	6652	7305	6939	6846	7209
Heating Capacity, Btu/hr *	46408	45386	46815	47476	45804
Coefficient of Performance**	3.81	3.73	3.90	3.97	3.78

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

but lower than the wavy and louvered fin cases. The calculated heat transfer multiplying factors for the corrugated fin case were 1.10 for the condenser and 1.09 for the evaporator. In the single fin model tests, the fins were heated and no condensed moisture was present. In the absence of data for wet fins, it has been assumed that the multiplying factors for air side heat transfer and pressure drop are the same for wet and dry fins. The match between predicted and measured capacity and COP is best for the corrugated and wavy fin cases, which attests to the accuracy of the model.

The results for the high capacity cooling case, summarized in Table 7-30, follow the same pattern as the heating results with the smooth fin case giving the lowest capacity and EER and the louvered fin case the highest. The performance of the corrugated fin case is higher than the smooth fin case and lower than the wavy fin case. The calculated heat transfer multiplying factors for the corrugated fin case were 1.08 for the condenser and 1.10 for the evaporator. As in the case of the heating tests, the multiplying factors for wet fins have been assumed to be the same as for dry fins since no wet fin data were available. The corrugated and wavy fin cases give the best match to the measured capacity and EER.

The predicted values of air pressure drop for the four different fin models are shown in Table 7-31. The indoor unit pressure drops are based on a duct diameter DDUCT of 8.0 inches and a house heating load FIXCAP of 42000.0 Btu/hr in all cases. It is interesting to note that due to the use of different correlations for smooth fins and wavy and louvered fins, the predicted air pressure drops for the wavy and louvered fin cases are less than for the smooth fin cases, an obvious inconsistency. The calculated air pressure drop multipliers for the corrugated fin in the high capacity heating mode were 1.40 for the condenser and 1.44 for the evaporator. In the high capacity cooling mode with a corrugated fin, the calculated air pressure drop multiplying factors were 1.42 for the evaporator and 1.41 for the condenser.

Table 7-30

Test 111 High Capacity Cooling @ 95 Deg F
Ranco #26 Reversing Valve

	Test	Smooth Fins	Wavy Fins	Louvered Fins	Corrugated Fins
Compressor					
Mass Flow, lb/hr	736.8	721.8	741.8	748.1	727.5
Power Input, Watts	4389	4327	4306	4294	4350
Heat Loss, Btu/hr	3767	3988	3968	3957	4008
Inlet Pressure, psia	88.7	87.7	89.2	89.6	88.2
Inlet Temperature, deg F	44.8	44.1	45.1	45.3	44.4
Inlet Superheat, deg F	1.1	1.1	1.1	1.1	1.1
Inlet Enthalpy, Btu/lbm	108.6	108.6	108.7	108.7	108.6
Exit Pressure, psia	278.9	287.1	282.1	280.1	286.0
Exit Temperature, deg F	163.6	168.9	166.2	165.2	168.7
Exit Superheat, deg F	42.4	45.4	44.1	43.6	45.5
Exit Enthalpy, Btu/lbm	122.7	123.5	123.1	123.0	123.5
Line & Reversing Valve					
Pressure drop, psi	9.83	8.83	9.41	9.61	8.98
Heat Transfer, Btu/hr	1296.5	33.9	32.6	32.9	33.5
Condenser					
Inlet Pressure, psia	269.1	278.3	272.7	270.5	277.0
Inlet Temperature, deg F	153.7	167.0	164.2	163.1	166.7
Inlet Superheat, deg F	35.3	46.0	44.7	44.3	46.0
Inlet Enthalpy, Btu/lbm	120.9	123.5	123.1	122.9	123.5
Exit Pressure, psia	266.3	275.7	269.9	267.7	274.4
Exit Temperature, deg F	113.2	115.9	114.2	113.6	115.6
Exit Subcooling, deg F	4.4	4.4	4.5	4.4	4.4
Exit Enthalpy, Btu/lbm	43.5	44.4	43.8	43.6	44.2
Evaporator					
Inlet Pressure, psia	109.8	109.6	111.7	112.4	110.2
Inlet Temperature, deg F	56.4	56.3	57.5	57.9	56.7
Inlet Quality	0.204	0.219	0.209	0.206	0.217
Inlet Enthalpy, Btu/lbm	43.2	44.4	43.8	43.6	44.2
Exit Pressure, psia	92.6	92.7	94.3	94.8	93.2
Exit Temperature, deg F	47.0	46.3	47.3	47.6	46.6
Exit Superheat, deg F	0.9	0.0	0.0	0.0	0.0
Exit Enthalpy, Btu/lbm	108.7	108.6	108.6	108.7	108.6
Lines & Reversing Valve					
Pressure Drop, psi	3.82	4.97	5.14	5.19	5.02
Heat Transfer, Btu/hr	57.9	14.4	14.1	13.5	13.8
Capacity, Btu/hr*	48247	46346	48097	48669	46809
EER, Btu/W-hr**	11.0	10.7	11.2	11.3	10.8

* Not corrected for blower work and compressor losses
** Does not include fan, blower and control power

Table 7-31. Predicted Air Pressure Drops in Inches of H_2O

	Smooth Fins	Wavy Fins	Louvered Fins	Corrugated Fins
	High Capacity Heating @ 47°F			
Condenser Side(1)	0.6706	0.6493	0.6584	0.7170
Evaporator Side	0.0580	0.0550	0.0610	0.0800
	High Capacity Cooling @ 95°F			
Condenser Side	0.0537	0.0454	0.0499	0.0764
Evaporator Side(1)	0.6670	0.7380	0.7580	0.7200

(1) Includes pressure drop in ductwork, cabinet, etc. DDUCT = 8.0 inches, FIXCAP = 42,000 Btu/hr

8. SUMMARY

The primary objective of the ORNL heat pump performance computer code is to provide the user with a rapid and economical means of predicting the approximate performance of a heat pump or air conditioner of a given configuration prior to constructing and testing a prototype. The potential for significant savings in time and money is obvious.

The code is also highly useful for constructing a complete performance map for a given system from a limited number of test points. This ability is particularly useful in providing a data base for methods of predicting seasonal and annual performance of heat pump and air conditioning systems.

The code has proven highly suitable for performing both of these tasks. Under most operating conditions, it can predict the capacity and energy efficiency of a properly defined system with an accuracy approaching that of the laboratory methods used to measure these parameters. In addition, the code provides a detailed prediction of the operating parameters of the refrigeration cycle that is sufficiently accurate to aid in refining and optimizing a system with respect to a given set of objectives.

The addition of models for predicting the effects of reversing valve refrigerant pressure drop, heat transfer and internal leakage, suction line accumulator refrigerant pressure drop and internal heat transfer and of fin patternation on air side heat transfer and pressure drop has enhanced the accuracy and flexibility of the code without sacrificing "user friendliness" to a significant degree. The number of additional user supplied input parameters is minimal and the added features are all optional.

The verification phase of the project significantly increased our confidence in the ability of the code to accurately predict system performance in the range of ambient temperatures where the bulk of system operation occurs. At ambient temperature extremes, very low temperatures in the heating mode and very high temperatures in the cooling mode, the accuracy of the model declines for cases using the map based compressor model, primarily due to inaccuracies in the mass flow and power input curve fits near the limits of the input data. The accuracy of the program can be improved for these cases by relatively minor changes in the code or in the method of application. In particular, the substitution of map-based compressor model with a larger number of parameters, such as the model used by Westinghouse which uses nine coefficients in place of the six used by the ORNL model, would allow more accurate matching of the compressor characteristics over a wider range of operating conditions. Alternately, the existing map-based model could be used with multiple sets of coefficients for various operating regimes.

The refrigerant pressure drop, heat transfer, and internal leakage associated with the reversing valve have a significant impact on system performance and energy efficiency. The addition of these models significantly improves the ability of the code to predict heat pump performance.

The addition of the suction line accumulator heat transfer and pressure drop model significantly improved the ability of the code to predict the thermodynamic cycle points of the Westinghouse/DOE advanced electric heat pump in the heating mode. However, the impact on system capacity and energy efficiency are relatively small and for systems with a conventional accumulator, without an internal heat exchanger, accumulator effects can be ignored without significantly compromising the accuracy of the results.

The addition of the models for predicting the effects of the corrugated fin pattern on air side heat transfer and pressure drop make the model more flexible and realistic, since patterned fins are far more common than smooth fins in the industry and patternation significantly influences both air side heat transfer and air side pressure drop.

Table 8-1 compares the measured performance of the preprototype advanced electric heat pump with the predicted performance using the

Table 8-1. Comparison of Measured and Predicted Performance with Ranco #26 Reversing Valve with Leakage

Ambient Temp. °F	Valve Leakage %	Measured Capacity Btu/hr	Predicted Capacity Btu/hr	Error %	Measured COP	Predicted COP	Error
			High Capacity Heating				
47	2.0	46408	46306	0.2	3.81	3.86	1.3
17	5.0	25860	27053	4.6	2.78	2.97	6.8
			Low Capacity Heating				
47	2.0	29368	28530	2.9	4.45	4.41	0.9
			High Capacity Cooling				
95	1.0	48247	47730	1.1	3.22	3.19	0.9
			Low Capacity Cooling				
82	1.0	37363	33574	10.1	5.22	4.67	10.5

modeling parameters which, in our opinion, give the best overall match to the measured performance of the cases investigated in detail, the Ranco #26 reversing valve, with the approximate leakage rates projected in Table 7-24 and a wavy fin pattern. The accumulator model was used for the heating runs and was not included in the cooling runs. This comparison was completed prior to the implementation of the corrugated fin model and project resources did not permit a full evaluation of the corrugated fin model, which may give an better match between predicted and measured performance than the wavy fin model.

This combination of modeling parameters results in an excellent match between predicted and measured performance in the both heating and cooling modes at the ARI rating conditions, 47°F outdoor ambient temperature heating and 95°F outdoor ambient temperature cooling. Both capacity and COP are predicted with a accuracy well within the experimental error limits permitted by the ARI rating method.

For the high capacity heating mode at 17°F outdoor ambient temperature, the predicted capacity and COP are higher than the measured values. As noted in Section 7.2.1, this discrepancy is primarily due to the difference between the measured refrigerant mass flow rate and the mass flow rate predicted by the map based compressor model at high pressure ratios.

In the low capacity cooling mode at 82°F ambient temperature the predicted cooling capacity and COP exceed the measured values by 10% to 11%. Most of this discrepancy is due to the map based compressor model. The map based model does not accurately match the performance of the dual-stroke compressor in the low capacity mode. The large clearance volume of the dual-stoke compressor in the low capacity mode results in characteristics which are not typical of reciprocating compressors and are difficult to represent mathematically. It should also be noted that the evaporator exit superheat deduced from the test data was less than 1°F and due to the accuracy limitations of the instrumentation, the actual evaporator exit conditions may have been slightly wet. If the evaporator exit conditions were wet, the actual capacity and COP would be less than that calculated assuming a low level of superheat.

9. CONCLUSIONS

1. The pressure drop and heat transfer in the refrigerant reversing valve has a significant impact on the capacity and energy efficiency of heat pumps. The analytical reversing valve model predicts a loss in heating capacity of 1.5 to 5.0% and a reduction in COP of 0.5 to 2.5% in the high capacity mode. The empirical model predicts a capacity reduction of 5.0 to 9.0% and a COP reduction of 1.5 to 6.0% in the high capacity heating mode. For the high capacity cooling mode the analytical model predicts a loss of both capacity and COP of 1.0 to 4.0%. The empirical model predicts a capacity loss of 7 to 8% and a reduction in COP of 2.5 to 3.5% for the high capacity cooling mode.

2. In the range of ambient temperatures where the bulk of heat pump operation occurs, reversing valve refrigerant pressure drop on the compressor suction side has a greater influence on system performance and efficiency than high side pressure drop, internal and external heat transfer and internal refrigerant leakage.

3. At extreme operation conditions (low temperature heating and high temperature cooling), where refrigerant flow rates are low, internal heat transfer in the reversing valve has a significant influence on system performance.

4. Internal refrigerant leakage in the reversing valve has a minor impact on system capacity and energy efficiency when the valve functions properly and the leakage rate is 1 to 2% of the total refrigerant flow.

5. If the reversing valve operates at large pressure ratios, or is defective or worn so than the leakage rate exceeds 5% of the total refrigerant flow, system capacity and efficiency are compromised. For the preprototype unit operating in the high capacity heating mode at 0°F, the projected reversing valve leak rate exceeds 9% of the compressor mass flow rate, assuming a 2% leak rate at 47°F ambient temperature.

6. If a flooded evaporator system with an internal heat exchanger in the accumulator is used, the heat transfer effects must be modeled to obtain an accurate prediction of the refrigerant cycle points.

7. The impact of the suction line accumulator on overall system capacity and energy efficiency is small.

8. It is difficult to represent the performance of a reciprocating compressor over a wide range of operating conditions using a single curve fit and the use of multiple curve fits or a table look-up approach should be considered.

9. There appear to be inconsistencies between the air side pressure drop correlations used for smooth fins and for wavy and louvered fins that should be investigated.

10. The ORNL code provides accurate predictions of heat pump and air conditioning system performance for most operating conditions, is highly flexible and relatively "user friendly."

10. ACKNOWLEDGMENT

The authors acknowledge the following individuals who contributed to the work reported: S. Veyo, initial Westinghouse program manager; M. Hogan, reversing valve heat transfer and pressure drop modeling; A. Swiss, N. Wilds and A. Venturino, operation of the heat pump test facility; A. Montali, heat pump test data reduction; C. Kapral, preparation of the tables for the verification section; P. Malloy, text processing.

The authors acknowledge the help and encouragement given by personnel of the Oak Ridge National Laboratory in the administration of this research project. In particular we recognize P. Fairchild and V. Baxter, Program Managers, and K. Rice for his assistance in understanding the internal logic of the ORNL heat pump performance code.

11. REFERENCES

1. S. K. Fischer and C . K. Rice, "The Oak Ridge Heat Pump Models: I. A Steady-State Model for Air-to-Air Heat Pumps," ORNL/CON-80/R1, August 1983.

2. F. C. McQuiston, "Correlation of Heat, Mass and Momentum Transport Coefficients for Plate-Fin-Tube Heat Transfer Surfaces with Staggered Tubes," ASHRAE Transactions, Vol. 84, Part 1, 1978.

3. T. Yoshii, "Transient Testing Technique for Heat Exchanger Fin," Reito, 47:531, 1972, pp. 23-29.

4. T. Senshu et al., "Surface Heat Transfer Coefficient of Fins Utilized in Air-Cooled Heat Exchangers," Reito, 54:615, 1979, pp. 11-17.

5. H. S. Kirschbaum and S. E. Veyo, "An Investigation of Methods to Improve Heat Pump Performance in a Northern Climate," Vol. I-III, Electric Power Research Institute, EPRI EM-319, January 1977.

6. S. E. Veyo and T. J. Fagan, "Advanced Electric Heat Pump Dual-Stroke Compressor and System Development," ORNL/Sub/79-24712/3, December 1983.

7. T. J. Fagan, R. R. Young, R. A. Lucheta, and S. E. Veyo, "Engineering Field Evaluation of the Westinghouse/DOE Dual-Stroke Advanced Electric Heat Pump," ORNL/Sub/79-24712/4D, To be released.

12. NOMENCLATURE

Symbol	Meaning	Units
A_t	Heat transfer area per unit length of tubing in the suction line accumulator	ft
A1	Constant term in reversing valve pressure drop curve fit	psi
A2	Exponent in reversing valve pressure drop curve fit	dimensionless
A3	Constant term in reversing valve heat leak curve fit	Btu/hr
A4	Linear term in reversing valve heat leak curve fit	Btu/hr-°F
b	Plate spacing	ft
B1	Second order term in compressor mass flow rate curve fit	lbm/hr-°F^2
B2	Linear term in compressor mass flow rate curve fit	lbm/hr-°F
B3	Second order term in compressor mass flow rate curve	lbm/hr-°F^2
B4	Linear term in compressor mass flow rate curve fit	lbm/hr-°F
B5	Second order term in compressor mass flow rate curve	lbm/hr-°F^2
B6	Constant term in compressor mass flow rate curve fit	lbm/hr
Cp_r	Refrigerant liquid specific heat	Btu/lbm-°F
C0	Constant term in compressor mass flow rate curve fit	lbm/hr
C00	Constant term in compressor mass flow rate curve fit	lbm/hr
C01	Linear term in compressor mass flow rate curve fit	lbm/hr-°F
C02	Second order term in compressor mass flow rate curve	lbm/hr-°F^2
C10	Constant term in compressor mass flow rate curve fit	lbm/hr-°F
C11	Linear term in compressor mass flow rate curve fit	lbm/hr-°F^2
C12	Second order term in compressor mass flow rate curve fit	lbm/hr-°F^3
C20	Constant term in compressor mass flow rate curve fit	lbm/hr-°F^2
C21	Linear term in compressor mass flow rate curve fit	lbm/hr-°F^3
C22	Second order term in compressor mass flow rate curve	lbm/hr-°F^4

g_c	Gravitational acceleration	lbm-ft/lbf-sec^2
L	Plate flow length	ft
L_t	Length of tubing in accumulator heat exchanger	ft
M_{leak}	Refrigerant leakage rate	lbm/sec
M_r	Refrigerant mass flow rate	lbm/hr
μ	Refrigerant absolute viscosity	lbm/ft-sec
P_{dif}	Pressure difference	lbf/sq ft
Pd_v	Reversing valve low side refrigerant pressure drop	psi
Q_{ac}	Suction line accumulator heat flux	Btu/hr
Q_{val}	Reversing valve internal heat flux	Btu/hr
Rho_a	Average refrigerant density	lbm/cu ft
Rho_s	Reversing valve low pressure side refrigerant inlet density	lbm/cu ft
T_{ao}	Accumulator low pressure side refrigerant exit temperature	°F
T_c	Compressor exit refrigerant saturation temperature	°F
T_{co}	Compressor exit refrigerant temperature	°F
T_{cx}	Condenser exit refrigerant temperature	°F
T_e	Compressor suction refrigerant saturation temperature	°F
T_{ex}	Evaporator exit refrigerant temperature	°F
T_{vi}	Expansion valve inlet refrigerant temperature	°F
U_o	Overall heat transfer coefficient of suction line accumulator heat exchanger	Btu/hr-sq ft-°F
W	Plate width	ft
W_{ai}	Weighted average number of refrigerant circuits for indoor coil	dimensionless
W_{ao}	Weighted average number of refrigerant circuits for outdoor coil	dimensionless

APPENDIX A

REVERSING VALVE HEAT TRANSFER
AND PRESSURE DROP

78-1E9-HPUMP-R1

Proprietary Class 2

Distribution only within the company Copies of this document cannot be given or shown o anyone outside the company

Destroy by tearing into several pieces

A MATHEMATICAL MODEL OF A HEAT PUMP REVERSING VALVE

M. R. Hogan

May, 1978

Downgraded to

UNRESTRICTED

Mgr., Engrg. Division
R&D Center

Date

APPROVED:

G. W. Wiener

G. W. Wiener, Manager
Energy Systems Division

Westinghouse R&D Center
Fluid Systems Laboratory
West Lafayette, Indiana

COMPUTER PROGRAM

Record of Transmittal within Westinghouse

Westinghouse Research Laboratories

Date May 1978

* To: (see attached distribution list)

This shipment contains:

The computer program V25, V26, and V30
(title)

and/or the following documents describing the above or other computer programs:

Research No.	Author	Title	Proprietary Class
78-1E9-HPUMP-R1	Hogan, M. R.	"A Mathematical Model of a Heat Pump Reversing Valve"	2

If you wish to send this material outside of Westinghouse, you must obtain approval from the Supervisor, Computer Center. This approval will be in accordance with the policy set forth in Chairman's Letter No. 3 dated August 11, 1969. For purposes of this policy, the Research Laboratories considers this program to be in Class A .†
(A, B, C, or D)

Sender M. R. Hogan

Address Fluid Sys. Lab. Phone WIN 556-1634

* One copy of this sheet must accompany every copy of the program or document transmitted outside the Research Labs. In addition, send one copy to E. J. Duckett, 801-3rd, and one to N.A. Mascia, 401-2B5.

† For definitions of classes, the sender is directed to Research Laboratories Operation Manual, Section 5.2.7. If assistance is needed, contact the Computer Center Supervisor, N.A. Mascia (WIN 236-7741).

Research Report 78-1E9-HPUMP-R1
Proprietary Class 2 May, 1978

A MATHEMATICAL MODEL OF A HEAT PUMP REVERSING VALVE

M. R. Hogan
Heat Transfer and Fluid Dynamics

ABSTRACT

The heat transfer between the high and low pressure circuits, the static pressure loss of both paths, and the natural convection heat loss for three geometrically similar reversing valves is modeled. Performance predictions, for parametric variations of inlet mass quality, as a function of refrigerant flowrate are included. The effect of a change in return bend thermal conductivity on valve performance is also examined.

TABLE OF CONTENTS

Research Report 77-1E9-HPUMP-R
Proprietary Class 2

A MATHEMATICAL MODEL OF A HEAT PUMP REVERSING VALVE

M. R. Hogan
Heat Transfer and Fluid Dynamics

I. INTRODUCTION

Current practice in heat pump control requires that the unit switch from the cooling to heating mode of operation quickly and automatically. This entails the switching of the refrigerant condenser from outside to inside the air conditioned space, and the evaporator vice versa. This is accomplished in Westinghouse heat pumps by the use of the so-called reversing valve. A schematic of such a valve is shown in Figure 1. Note that there are four paths from the valve, each corresponding to a major heat pump component. For the cooling mode the condenser would be outside, whereas the evaporator would be the inside coil. Upon reversal to the heating mode, the sliding return bend would move, by the hydraulic action of a pressure difference across the two pistons as shown in Figure 2, from left to right as shown in Figure 1. This would change the outside coil from a condenser to an evaporator, and the inside coil from an evaporator to a condenser.

It was the objective of this work to obtain a coded model of three Ranco reversing valves, designated V25, V26, and V30, for inclusion in a large, proprietary heat pump code. Specifically, it was desired that the code estimate the condenser and accumulator port exit state given the refrigerant, the mass flowrate, the local ambient temperature, and the compressor and evaporator port static pressure. As a result, the heat transfer between the high and low pressure circuits, the pressure losses of both paths, and the natural convection heat losses from the valve body were modeled.

Dwg. 6260A87

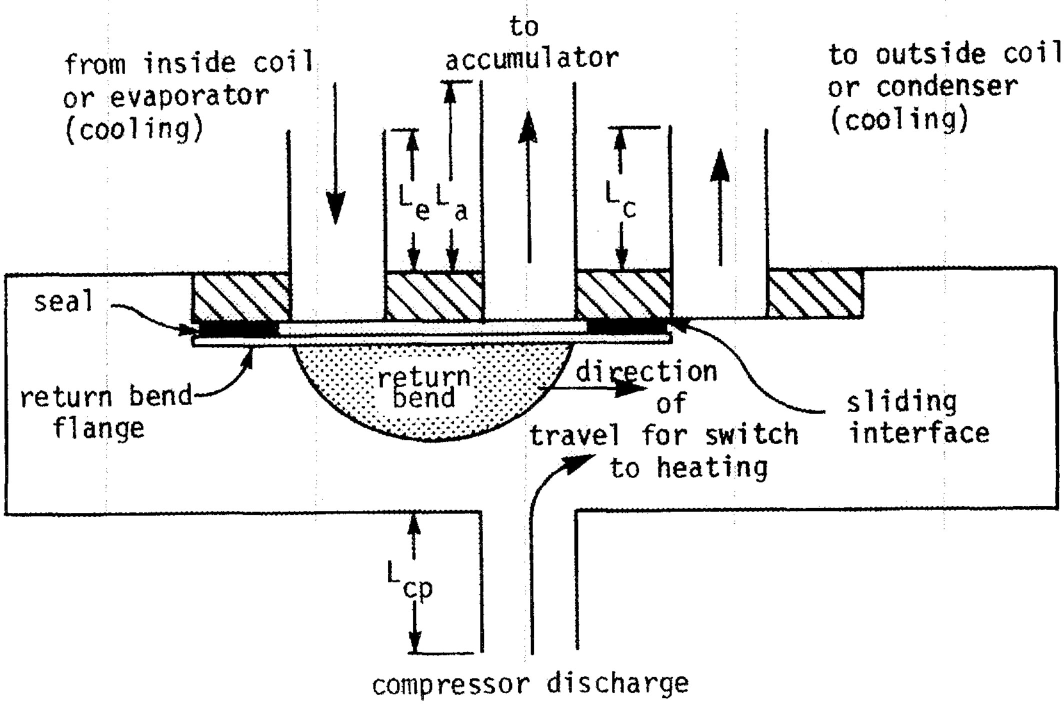


Figure 1. A Schematic of a Typical Ranco Reversing Valve. The Valve can be Divided into the High and Low Pressure Sections. The Low Side Consists of the Flow Path Required to Direct the High Quality, Low Pressure Vapor from the Evaporator to the Accumulator. Hot, High Pressure Gas is Routed from the Compressor to the Condenser via the High Side.

Dwg. 6260A88

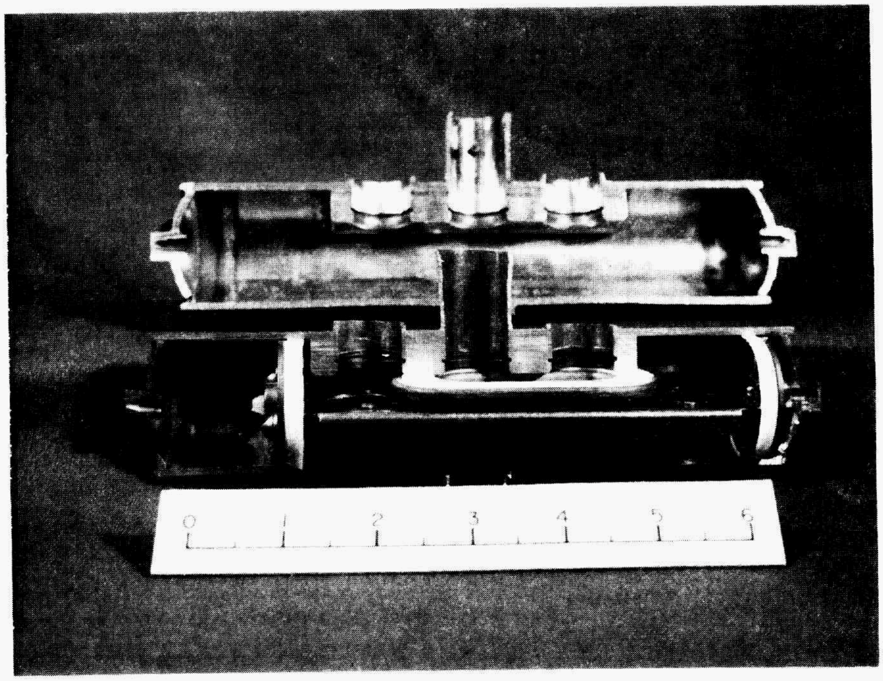


Figure 2. A Photograph of the Ranco V25 Reversing Valve. In the Foreground One Can See the Piston Rings Across Which a Pressure Difference Forces the Return Bend Assembly to Slide. Also Shown is The Return Bend Sealing Surface Pressing Against the Valve Body.

II. RESULTS AND RECOMMENDATIONS

A mathematical model has been developed which predicts the refrigerant exit state from three Ranco reversing valves. The model was coded in Fortran IV and cast into subroutine form for inclusion in a larger heat pump program. As a result of experimental data provided by the manufacturer, it was possible to establish the value of an empirical constant necessary to reproduce the low pressure circuit pressure drop for 100% inlet mass quality. Although additional data are not available for comparison, the remaining segments of the model appear to produce reasonable results.

For a more complete model, more accurate results, and additional confidence in the model, it is recommended that additional analytical and experimental work be performed. Specifically, the bend area treated as adiabatic should be included as a fin conducting heat from the flat bend surface; an external bend surface heat transfer coefficient multiplier should be obtained and used in lieu of an estimated area reduction fraction; and the bend low pressure side should incorporate a heat transfer coefficient correlation applicable for single- or two-phase flows in a return bend. Useful but of less importance would be pressure drop measurements as a function of flow rate for the high pressure circuit.

III. MODEL DEVELOPMENT

A. High Side Pressure Losses

Examination of Figure 2 shows that the hot gas must enter the valve (from below), expand, turn, pass through an orifice in the slide frame, and finally exit. The process was analyzed as a sharp edge expansion followed by a sharp edge contraction. The irreversible pressure loss can, therefore, be expressed as the sum of the two components, i.e.,

$$\Delta P\text{internal} = -\frac{\rho_{cp}}{2g} \left(C1 V_{cp}^2 + C2 V_c^2\right) \tag{1}$$

where the subscripts cp and c denote the compressor and condenser ports respectively as shown in Figure 1. C1 and C2 are the coefficients of expansion and contraction and were taken to be 1.0 and .85 respectively from Reference 1. The losses due to the connection tubing were estimated from,

$$\Delta P\text{tube}_{hs} = -\frac{1}{2}\rho_{cp} \frac{V_{cp}^2}{g} \frac{L_{cp}}{D_{cp}} \left\{ .316 \left(\frac{\rho_{cp} V_{cp} D_{cp}}{\mu_{g,cp}}\right)^{-1/4} \right\}$$

$$- \frac{1}{2}\rho_c \frac{V_c^2}{g} \frac{L_c}{D_c} \left\{ .316 \left(\frac{\rho_c V_c D_c}{\mu_{g,c}}\right)^{-1/4} \right\} \tag{2}$$

where the term in braces represents the smooth tube turbulent friction factor. Since D_c and D_{cp} are not generally equal, as are D_e, D_a, D_c, the velocity contribution to static pressure gain must be computed. Therefore,

$$\Delta P\text{rise} = \frac{\rho_{cp}}{2g} (V^2_{cp} - V^2_c) \quad . \qquad (3)$$

Thus, the static pressure change from the compressor to the condenser connection is,

$$\Delta P\text{high side} = \Delta P\text{internal} + \Delta P\text{tube} + \Delta P\text{rise} \qquad (4)$$

B. Low Side Pressure Losses

The low side is geometrically simple but complicated by the presence of a two-phase mixture. A complete description of the method used to estimate the losses may be found in Reference 2. Essentially the loss model is based on the fact that,

$$\frac{\Delta P_{TP}}{\Delta P_f} = 1 + \frac{C}{X} + \frac{1}{X^2} \qquad (5)$$

where ΔP_f is the pressure loss computed for the liquid flowing alone. The subscript TP indicates the actual pressure drop for a two-phase flow. X is a parameter which indicates the degree to which a mixture behaves as a gas and is defined by,

$$X^2 = \frac{(dP/dZ)_f}{(dP/dZ)_g} \qquad (6)$$

where again the subscripts indicate that dP/dZ would be calculated for the gas or liquid flowing alone. It can be shown (Reference 2) that for a turbulent liquid film and gas core that,

$$X = \left(\frac{1-x}{x}\right)^{7/8} \left(\frac{\rho_g}{\rho_f}\right)^{1/2} \left(\frac{\mu_f}{\mu_g}\right)^{1/8} \quad . \qquad (7)$$

For C in (5), Collier[2] recommends,

$$C = \left[1 + \left(\frac{20D_e}{L_{eq}}\right)\left(\frac{v_g - v_f}{v_g}\right)^{1/2}\right]\left[\left(\frac{v_g}{v_f}\right)^{1/2} + \left(\frac{v_f}{v_g}\right)^{1/2}\right] . \qquad (8)$$

Note that the bend characteristic diameter was taken to be D_e. If ΔP_f is known, then by virtue of (7) and (8) the two-phase pressure drop can be calculated via (5). ΔP_f can be estimated from,

$$\Delta P_f = -\frac{G_e^2(1-x_e)^2}{2\rho_f\, g}\frac{L_{eq}}{D_e} f \qquad (9)$$

where G_e is the mass velocity and is the mass flowrate divided by the flow cross sectional area, i.e.,

$$G_e = \frac{\dot{m}}{\frac{\pi D_e^2}{4}} \qquad (10)$$

The equivalent length of the bend in (9) is an unknown which is probably constant to a first approximation. Since it is desirable to model valves of varying but similar geometry, the change in equivalent length with respect to the radius-to-diameter ratio is necessary. This can be estimated from,

$$L_{eq} = \frac{L_{eq,ref}}{Kb_{ref}D_{ref}}\left[\frac{(R/D_e)^{-.6349}}{3.127}\right] D_e \qquad (11)$$

where the subscript ref denotes a reference value, and Kb a loss coefficient. Equation (11) is more fully derived in Reference 3 but essentially estimates the effect of R/D changes for bends of a similar geometry. R, the bend centerline radius of curvature, was taken to be one-half the distance between the accumulator and evaporator connection centerlines. Since $L_{eq,ref}$, Kb_{ref}, and D_{ref} are unknown, they can be combined to form a single unknown parameter, thus,

$$C3 = \frac{L_{eq,ref}}{(3.127)(Kb_{ref})(D_{ref})} \quad . \tag{12}$$

C3 was determined by matching the low side calculated pressure drop with the loss data supplied by the manufacturer. It was determined that C3 = 100.9 allowed the Ranco V30 and V25 valves' pressure loss performance to be faithfully reproduced whereas the V26 valve's R/D ratio had to be increased 20% (for C3 = 100.9) over the measured value for adequate performance prediction.

The friction factor in (9) must be calculated for the liquid flowing alone, thus,

$$f = .316 \left\{ \frac{(1-x_e)\, G_e\, D_e}{\mu_f} \right\}^{-1/4} \tag{13}$$

which is the same expression used for straight tubes in (2). Substitution of (11), (12), and (13) into (9) yields,

$$\Delta P_f = \frac{G_e^2\,(1-x_e)^2}{2\rho_f g}\, C3\, (R/D_e)^{-.6349} \left[.316 \left\{ \frac{(1-x_e)\, G_e\, D_e}{\mu_f} \right\}^{-1/4} \right] \tag{14}$$

The two-phase pressure loss attributable to flow through the 180° bend ($\Delta Pbend_{TP}$) can now be calculated by evaluating (5) via the results of (7), (8), (11), and (14).

The pressure losses due to flow through the accumulator and evaporator stub connectors can be estimated in a manner analogous to that employed for the bend. Equation (9) is used as before but the actual tube length is required, i.e.,

$$\Delta Ptube_{f,\ell s} = - \frac{G_e^2 (1-x_e)^2}{2\rho_f g} \left(\frac{L_e+L_a}{D_e}\right) \left[.316 \left(\frac{(1-x_e)\, G_e\, D_e}{\mu_f}\right)^{-1/4}\right]. \quad (15)$$

From Reference 2 it is recommended that C = 20 be employed for turbulent flow through tubing. Once again (5) is evaluated for $\Delta Ptube_{TP,\ell s}$ by (7), (15), and the above recommendation for C. The total low side pressure drop ($\Delta Plow$) is then the sum of $\Delta Pbend_{TP}$ and $\Delta Ptube_{TP,\ell s}$.

C. Heat Transfer Between the High and Low Pressure Sides. The heat transfer between the high and low sides occurs primarily at the return bend surface. Figure 3 indicates that the hot, high pressure compressor gas will impinge upon the bend surface. Since only one chamber (in this instance, the left as seen in Figure 4) is open, the majority of the flow will be diverted to one side only. Figure 5 shows the cross-section of a V25 sliding return bend. Note that there is good thermal contact between the sides or flat sections of the bend whereas the curved surfaces are separated by an air gap. The curved surfaces were, therefore, treated as adiabatic while the flat sections were taken as being in perfect thermal contact. A circle of radius equal to the bend radius of curvature was the area over which the heat transfer was presumed to occur. One further notes, from Figure 5, that the bend sealing surface or flange (the metal surface which is parallel to the scale in the photograph) is insulated by the nylon sliding surface. Consequently, one side of the flange was considered adiabatic while the other was presumed to be exposed to the compressor gas.

Since the gas must impinge on and flow over the bend, the film coefficient for the flat conducting metal parallel to the flow was modeled as a turbulent flow over a flat plate. Due to the unknown nature of the actual flow field, the characteristic length was assumed

Dwg. 6260A89

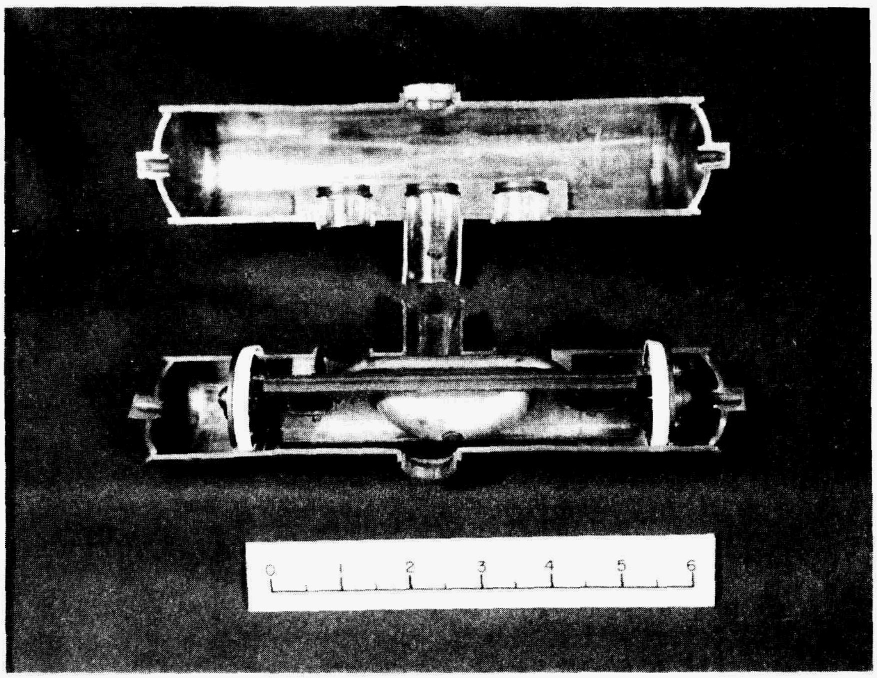


Figure 3. A Photograph of the Ranco V25 Reversing Valve. In the Foreground One Can See the High Pressure Side of the Sliding Return Bend. The Compressor Port Inlet is Directly Below the Bend.

Dwg. 6260A90

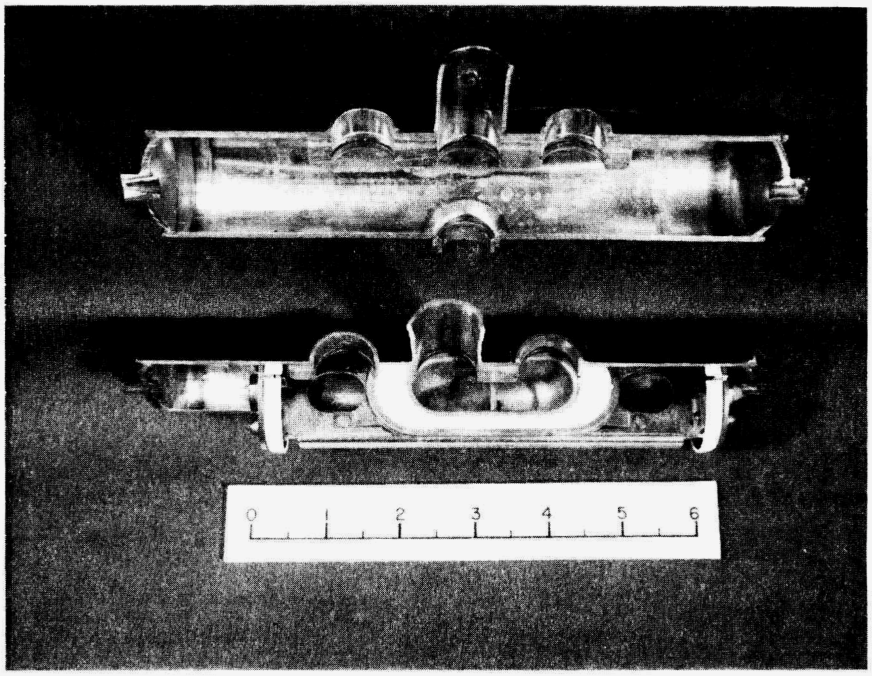


Figure 4. A Photograph of the Ranco V25 Reversing Valve. In the Foreground is a View of the Low Pressure Side of the Sliding Return Bend.

Dwg. 6260A91

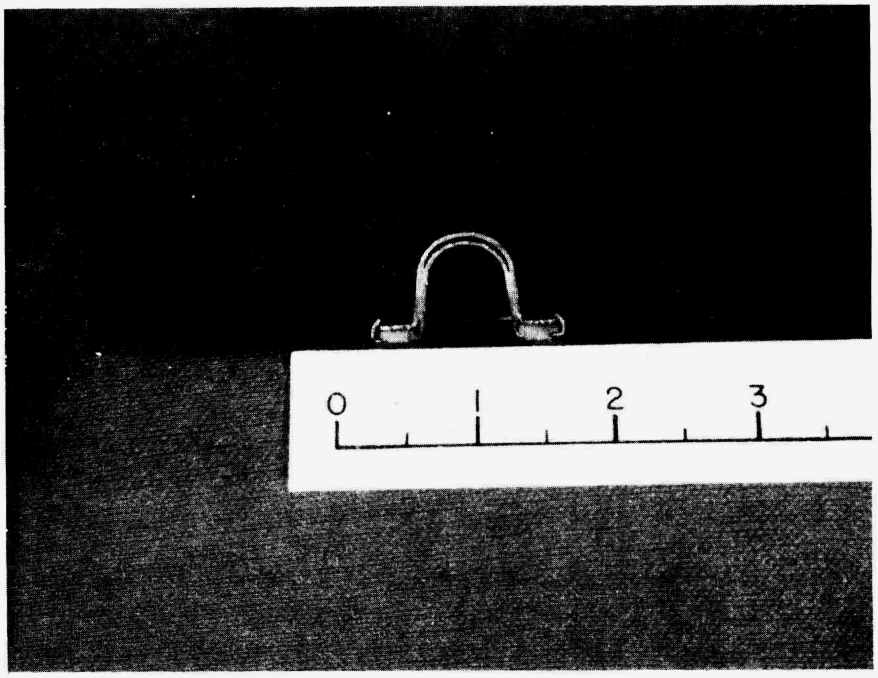


Figure 5. A Cross Sectional View of a V25 Sliding Return Bend. The White Nylon Pads are the Sliding and Sealing Surfaces of the Bend.

to be one-half the bend radius of curvature and the characteristic free stream velocity that of the compressor discharge port.

Because the bulk of the flow will be directed to one side, not all of the bend flat area will be characterized by the high flow velocity. One would anticipate portions of the surface to be in contact with stagnant gas. Because of the above arguments, the potential heat transfer area was reduced by one-third. This was a somewhat arbitrary choice, perhaps one-half is better, but is an attempt at analyzing a difficult flow field with a simple model. A more complete method, advanced by Stewart[7], to analyze the bend heat transfer would include the area treated as adiabatic (because of the air gap) as a fin conducting heat from the flat portion of the bend. An experimental multiplier could then be used on the flat surface film coefficient rather than an experimental area reduction fraction.

With respect to the above discussions, the heat transfer coefficient for the flat return bend area was estimated from,

$$h_{dry} = \left(\frac{2k_g}{R}\right) .036 \left(\frac{2R\dot{m}}{\pi D_{cp}^2 \mu_g}\right)^{.8} Pr_g^{1/3} \qquad (16)$$

where the subscript 'dry' indicates that the possibility of condensation was neglected.

If condensation were to occur, the film coefficient would be greatly enhanced. A discussion of various methods by which a single-phase heat transfer correlation can be corrected for condensation may be found in Reference 4. Although the discussion is for internal flows in straight tubes, the methods should give a fairly reliable means by which condensation may be taken into account for this more complex flow situation. The method chosen was that offered by Akers,

$$h_{TP} = h_{fo} \left[1 + x \left\{ \left(\frac{\rho_f}{\rho_g}\right)^{1/2} - 1 \right\}^{1/3}\right] \qquad (17)$$

where h_{fo} is the film coefficient calculated as if the liquid component were flowing at the same mass velocity as the mixture, and h_{TP} is the actual film coefficient. Since the flow will more than likely be only locally two-phase (the bulk thermodynamic state will be superheated), the mass quality is a somewhat ambiguous term. Nevertheless, one would expect (17) to give a good estimate of the heat transport augmentation by wall condensation for mass qualities approaching one. Therefore,

$$h_{TP} \approx h_{fo} \left(\frac{\rho_f}{\rho_g}\right)^{1/6} \quad . \tag{18}$$

Since,

$$h_{fo} = \left(\frac{2k_f}{R}\right) (.036) \left(\frac{2R\dot{m}}{\pi D_{cp}^2 \mu_f}\right)^{.8} (Pr_f)^{1/3} \quad , \tag{19}$$

then, from (16), and (19),

$$\frac{h_{fo}}{h_{dry}} = \left(\frac{k_f}{k_g}\right) \left(\frac{\mu_g}{\mu_f}\right)^{.8} \left(\frac{Pr_f}{Pr_g}\right)^{1/3} \tag{20}$$

and substitution of (20) into (18) yields,

$$h_{TP} = h_{dry} \left(\frac{k_f}{k_g}\right) \left(\frac{\mu_g}{\mu_f}\right)^{.8} \left(\frac{Pr_f}{Pr_g}\right)^{1/3} \left(\frac{\rho_f}{\rho_g}\right)^{1/6} \quad . \tag{21}$$

Equation (21) was used in lieu of (17) if the outer bend surface temperature was less than the saturation temperature corresponding to P_{cp}.

As noted earlier, one side of the bend sealing surface was considered adiabatic whereas the other was presumed to transfer heat.

With reference to Figures 5 and 6, one can see that the metal flange will act much like a fin, although one side is insulated. The heat transfer through this surface was estimated by modeling it as a one-dimensional fin with an insulated tip, and a perimeter equal to the flange perimeter (this is half the perimeter normally employed in fin calculations due to the insulated surface). Thus,

$$q_{flange} = \sqrt{h_{dry}\ \tilde{P}_b\ k_b'\ Ac_b}\ (T_{cp} - T_b)\ \tanh\ (mL) \tag{22}$$

where,

$$m = \sqrt{h_{dry}\ \tilde{P}_b\ /\ k_b'\ Ac_b}\ . \tag{23}$$

Since,

$$Ac_b = \tilde{P}_b t_b \tag{24}$$

then (22) becomes,

$$q_{flange} = \sqrt{h_{dry}\ \tilde{P}_b^2\ t_b\ k_b'}\ (T_{cp} - T_b)\ \tanh\ \{L_b\sqrt{\frac{h_{dry}}{k_b' t_b}}\} \tag{25}$$

Note that h_{dry} rather than h_{TP} has been used in (22). If condensate did form on the body of the bend, then downstream portions of the flange would realize the droplet scrubbing action whereas other portions, not in the condensate flow stream, would remain dry. In light of the uncertainties, it was felt best to just use h_{dry}. Note also that k_b' is the effective thermal conductivity of the flange material for conduction parallel to the flange surface. Thus, for a plated flange, such as in the V25 valve,

Dwg. 6260A92

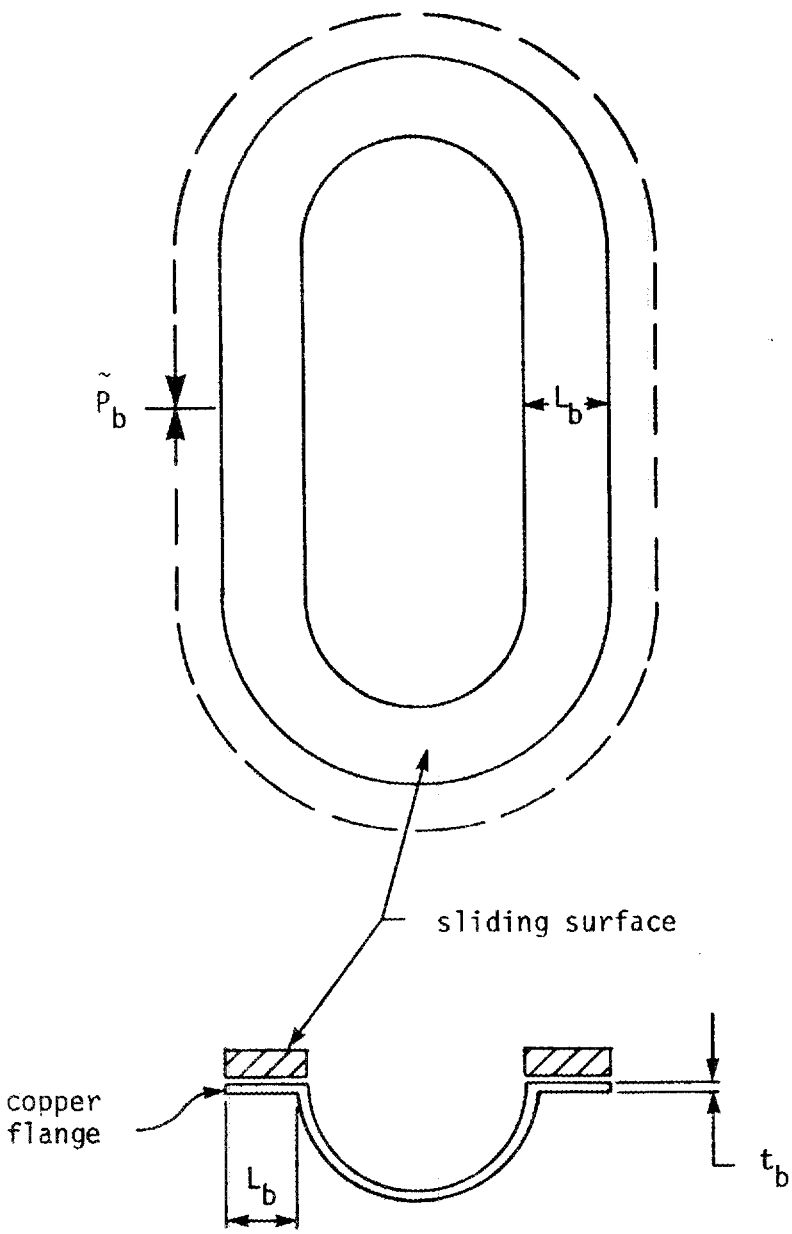


Figure 6. A Schematic of the Sliding Return Bend Used in the Ranco Reversing Valves. Note that $\tilde{P}_b$ is the Perimeter of the Metal Flange.

$$k_b' = \frac{k_{b,1}\, t_{b,1} + k_{b,2}\, t_{b,2}}{t_{b,1} + t_{b,2}}$$

The low side of the bend is exposed to the high quality, low pressure mixture from the evaporator. Because of the low density, the two-phase mixture, and the radial geometry, one would anticipate an extremely large heat transfer coefficient at this surface. As a result, the low side bend surface was assumed to be at the saturation temperature corresponding to P_e. The total heat transfer rate from the high to the low side of the bend can therefore be estimated from,

$$q_b = \frac{T_{cp} - T_e}{\dfrac{2t_b}{k_b'' As_b} + \dfrac{1}{h_{dry} As_b}} + q_{flange} \tag{26}$$

where h_{dry} is replaced by h_{TP} if a check of the high side bend temperature indicates condensation. The wall temperature was calculated from,

$$T_b = T_e + \frac{2t_b}{k_b'' As_b} \cdot \frac{T_{cp} - T_e}{\dfrac{2t_b}{k_b'' As_b} + \dfrac{1}{h_{dry} As_b}} \tag{27}$$

Note that the term $2t_b$ which appears in (26) and (27) is a result of the definition of t_b being one thickness as shown in Figure 6, whereas the thermal resistance of the wall must be calculated on the basis of two thicknesses as indicated by Figure 5. The double prime superscript on k_b indicates the effective thermal conductivity for heat conduction normal to the metal surface. Thus, for the V25 bend (the one instance in which the distinction is necessary),

$$k_b'' = \frac{t_{b,1} + t_{b,2}}{\dfrac{t_{b,1}}{k_{b,1}} + \dfrac{t_{b,2}}{k_{b,2}}}$$

Since the hot gas is routed through the valve body, there will always be natural convection losses to the local environment. These losses were estimated from simplified relations for air over heated cylinders from McAdams[5],

$$h_{nc} = .27 \left(\frac{T_{cp} - T_{amb}}{D_{vb}}\right)^{1/4} \tag{28}$$

or,

$$h_{nc} = .18 \left(T_{cp} - T_{amb}\right)^{1/3} \tag{29}$$

where (28) applies to laminar conditions and (29) to turbulent. Both equations were evaluated for a given set of conditions and the largest coefficient was assumed applicable, thus,

$$q_{nc} = h_{nc}\, As_{vb}\, (T_{cp} - T_{amb}) \quad . \tag{30}$$

D. <u>Calculation of Exit Thermodynamic States</u>

1. <u>High Side Exit State.</u> The high side exit pressure is calculated from (4),

$$P_c = P_{cp} + \Delta P\text{high side} \tag{31}$$

As a result of an energy balance about the high side refrigerant stream, the exit enthalpy can be computed via,

$$\tilde{h}_c = \tilde{h}_{cp} - \frac{q_{nc} + q_b}{\dot{m}} \frac{1}{3600} \tag{32}$$

Since two properties are known, the state of the fluid is fixed and the outlet temperature may be determined from the refrigerant thermodynamic state relations, thus,

$$T_c = f\ (P_c, h_c) \ . \tag{33}$$

2. <u>Low Side Exit State</u>. Since x_e, T_e, and P_e are known, the inlet enthalpy can be calculated from

$$h_e = h_f\ (P_e) + x_e\ (h_{fg}\ (P_e)) \quad , \tag{34}$$

then the outlet enthalpy from,

$$h_a = h_e + \frac{q_b}{\dot{m}} \frac{1}{3600} \quad . \tag{35}$$

The exit pressure can be computed from,

$$P_a = P_e + \Delta Plow \tag{36}$$

and thus,

$$x_a = \frac{h_a - h_f\ (P_a)}{h_{fg}\ (P_a)} \quad , \tag{37}$$

and,

$$T_a = f\ (P_a) \quad . \tag{38}$$

If, however, x_a from (37) was greater than one, then,

$$T_a = f\ (P_a, h_a) \tag{39}$$

and $x_a = 1.0$.

Curve 659599A

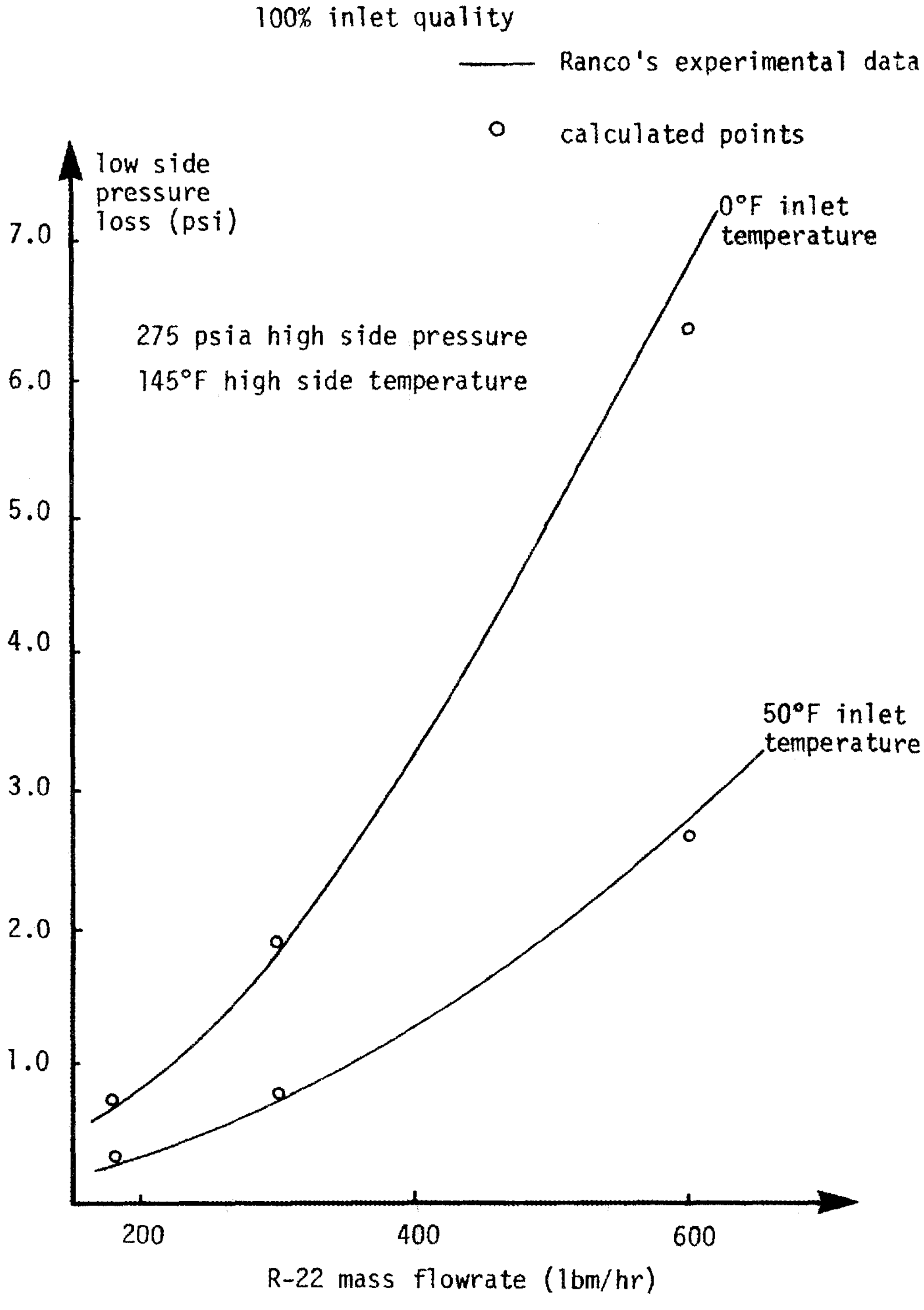


Figure 7. A Comparison of the Calculated and Experimental Low Side Pressure Loss as a Function of R-22 Mass Flowrate for the Ranco V26 Reversing Valve.

IV. RESULTS OF SAMPLE CALCULATIONS

The previously developed model was coded in Fortran IV and cast into a subroutine format. A listing of each subroutine may be found in Appendix B. Dimensions and material properties were determined from the data supplied by the manufacturer, found in Appendix A, and by inspection of the valve proper. Although the V26 valve's return bend geometry varied somewhat from the others—it was made of molded plastic (Zytel 103)—this perturbation was assumed to be of second order. The bend material thermal conductivity, however, varied from 1.0 Btu/(hr-ft-°F) for the Zytel V26, 12 Btu/(hr-ft-°F) for the stainless steel V30, to 30 Btu/(hr-ft-°F) for the steel (the major component) in the copper-plated V25.

The suction (low pressure side) pressure drop performance data supplied by the manufacturer enabled the empirical coefficient C3 [Equation (12)] to be established. Figures 7, 8, and 9 show the model predictions relative to the experimental data for inlet mass qualities of 100%. Though the model tends to depart from the experimental data at the higher mass flowrates, it is important to note that the valves are usually sized for a 2 psi suction pressure drop. One would therefore not anticipate them operating far from this design point. If this is the case, then the model will produce good results for all but the most severe off-design conditions.

Figure 10 indicates the calculated effect of various low side inlet qualities. It is apparent that, for this saturation temperature and R-22, there is little significant difference between the 100% and 75% inlet quality low side pressure drops. This is a result of the large void fraction, the ratio of the area occupied by the vapor relative to the total cross sectional area. For the 75% inlet quality case the void fraction is .98 which indicates only a 2% cross sectional area occupied by liquid. Based on these results, one would not anticipate the suction pressure drop to vary much from the 100% inlet quality

Curve 659600A

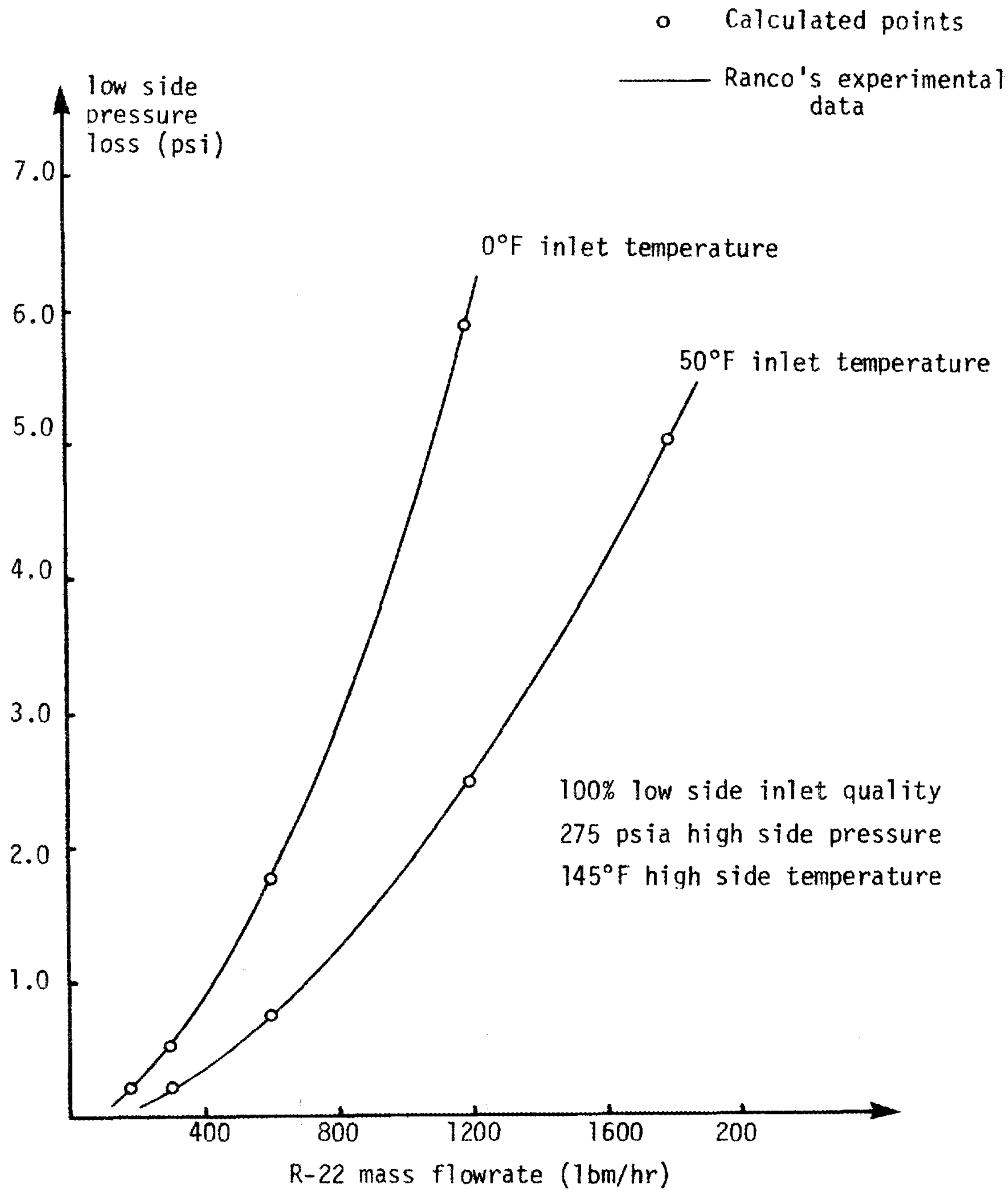


Figure 8. A Comparison of the Calculated and Experimental Low Side Pressure Loss as a Function of the R-22 Mass Flowrate for the Ranco V30 Reversing Valve.

Curve 659601A

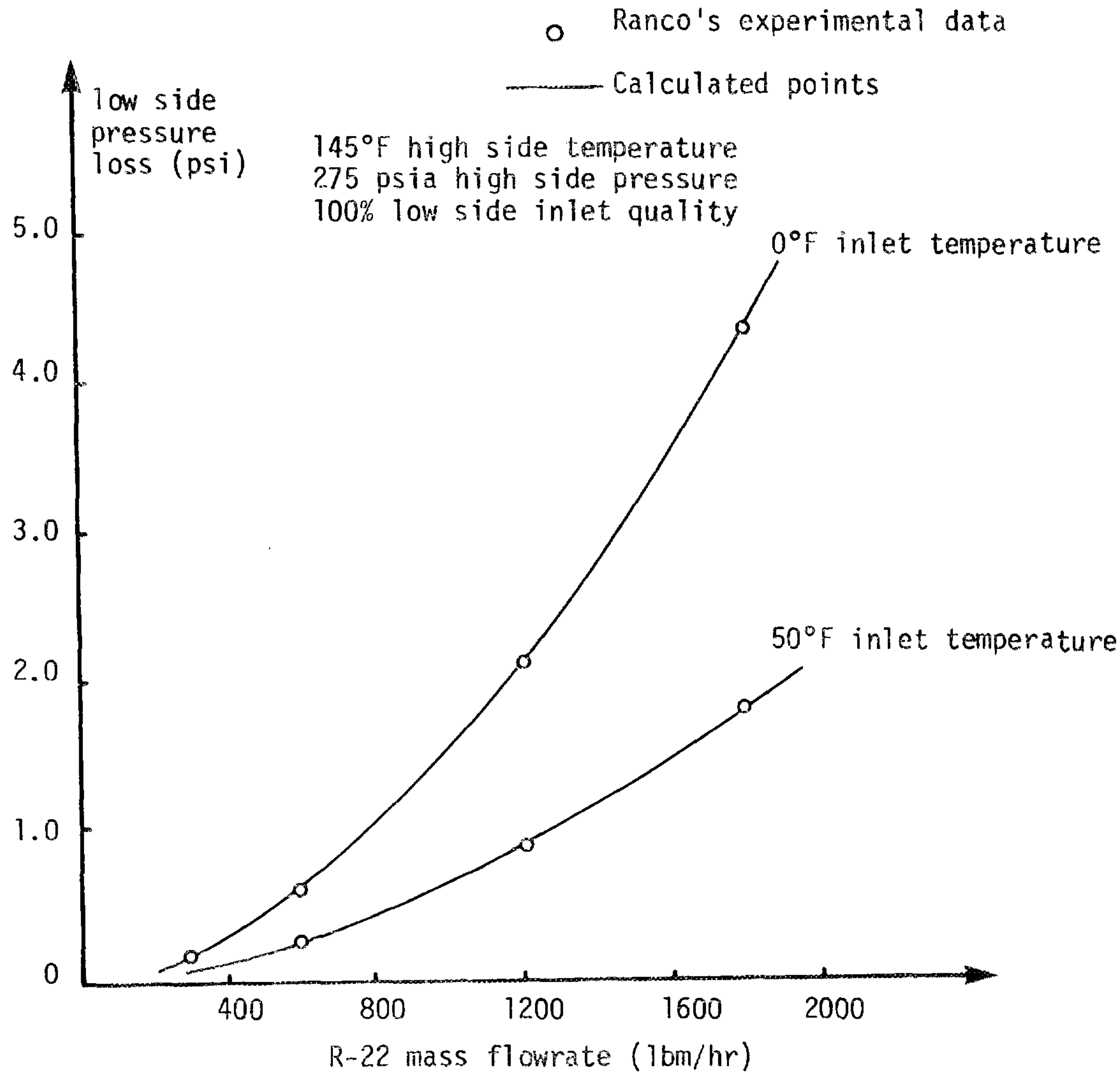


Figure 9. A Comparison of the Calculated and Experimental Low Side Pressure Loss as a Function of R-22 Mass Flowrate for the Ranco V25 Reversing Valve.

Curve 659602A

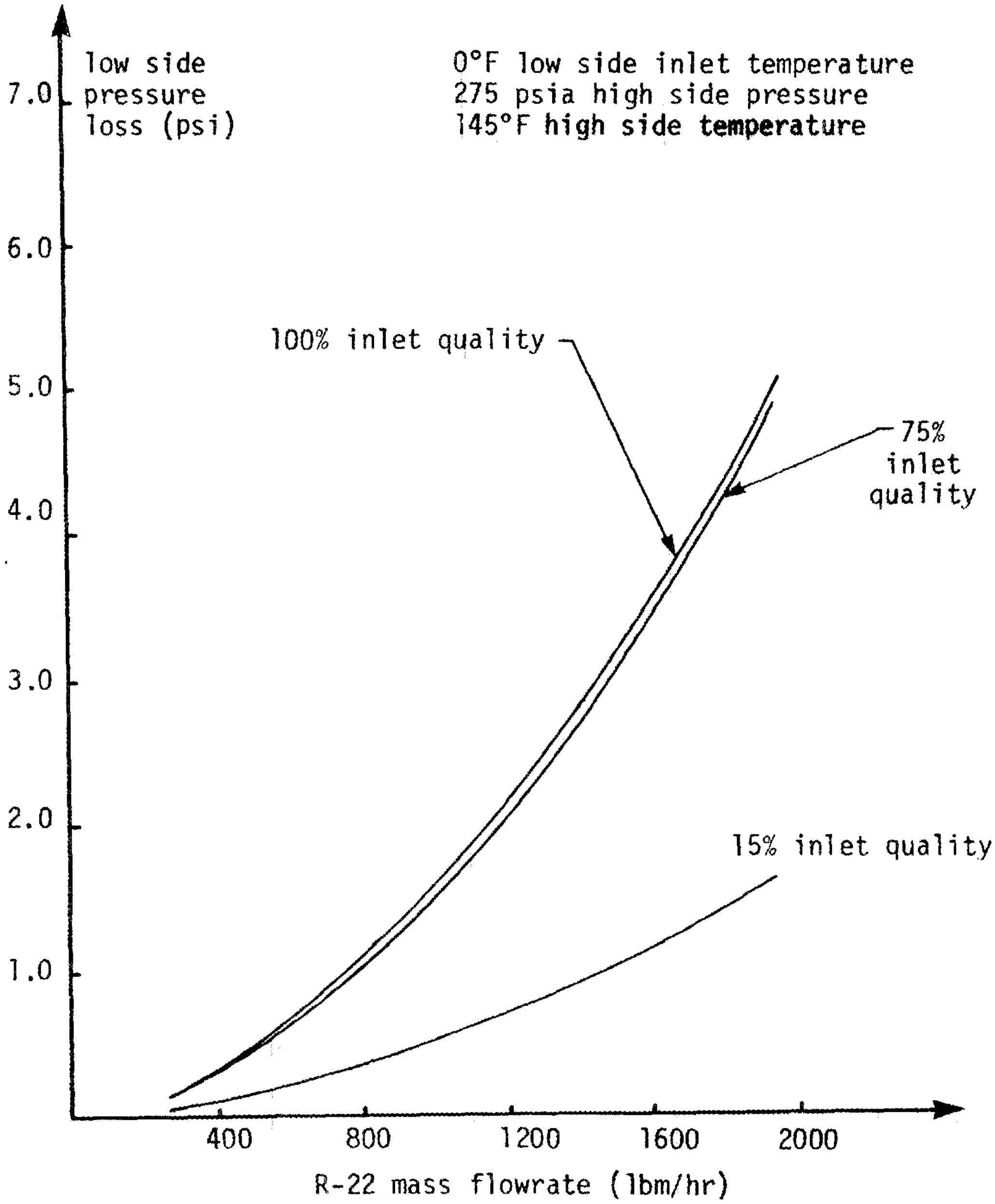


Figure 10. The Calculated Effect of Changes in the Low Side Inlet Mass Quality on the Low Side Pressure Loss as a Function of the R-22 Mass Flowrate for the Ranco V25 Reversing Valve.

solution as long as the qualities are relatively high. There may, however, be applications and/or circumstances where the two-phase flow refinement will make a significant contribution to the overall solution.

Shown in Figure 11 is the calculated performance of the V25 reversing valve as a function of mass flowrate. By examining the outlet relative to the inlet state curves (the dashed horizontal line), one can see graphically the deviation of the real valve from the ideal. Note that the low side pressure, and thus temperature, and the high side temperature are significantly affected by the reversing valve. Both are of importance since they tend to decrease the gas density, and thus increase the line pressure drop and decrease the compressor capacity. The high side outlet pressure decreases only slightly, which is not unexpected since the gas density is high relative to the low pressure side. Due to heat transfer from the high to the low pressure circuit, the mass quality exiting the valve is increased by approximately 2-2/3%.

At first glance it would appear that the exit mass quality curve exhibits an anomalous behavior with respect to mass flowrate. Since the heat transfer rate increases at a rate proportional to $\dot{m}^{.8}$, one would expect the exit mass quality to asymptotically approach the inlet value. This is not the case however due to the simultaneous decrease in the low side pressure. Because the saturated liquid enthalpy decreases faster than the latent heat of vaporization increases with respect to pressure, a second force on the mass quality comes into effect which tends to increase the mass quality (refer to Equation [37]). The two forces are just balanced at about 800 lbm/hr after which the second becomes dominant.

As stressed by Stewart[7], the significance of heat transfer from the high to low side and from the valve to the local environment is that it reduces the capacity of the heat pump as well as the coefficient of performance. Figure 12 illustrates this point for the same conditions as in the previous example. Both the heat transfer from

Curve 659603A

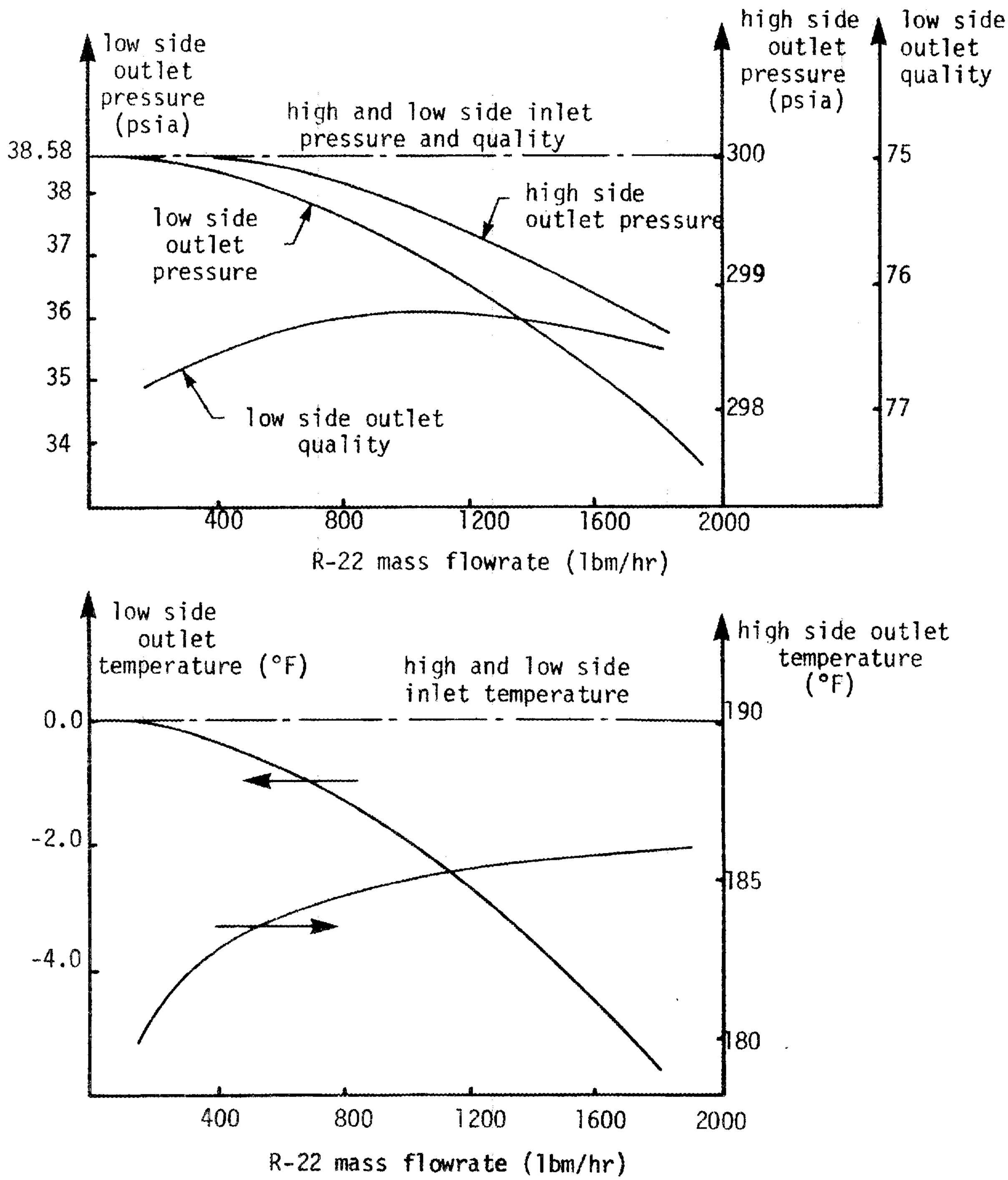


Figure 11. The Calculated Performance of the Ranco V25 Reversing Valve as a Function of Mass Flowrate.

the high to low side (q_b) and to the local environment (q_{nc}) are presented as a percent of the heat pump ideal capacity (constant pressure condensation at the compressor discharge pressure and with no subcooling). Both q_b and q_{nc} are small. They do, however, become more significant as the refrigerant mass flow rate (capacity) decreases. This can be important when the outside air temperature decreases. At these temperatures the heat pump capacity is usually greatly reduced, and, as can be seen in Figure 12, aggravated by q_b and q_{nc}.

One of the simplest means by which a valve's performance can be improved would be the replacement of the metal (for the V25) return bend with one exhibiting a lower thermal conductivity. The computed results of such a substitution are shown in Figure 13. Note that the thermal conductivity of the plated material has been reduced by 30. It can be seen that the high-to-low heat transfer losses are reduced by a factor of three whereas the natural convection heat transfer is unchanged, as expected. Whether such a modification is worthy of consideration is entirely dependent on the economics of the heat pump system.

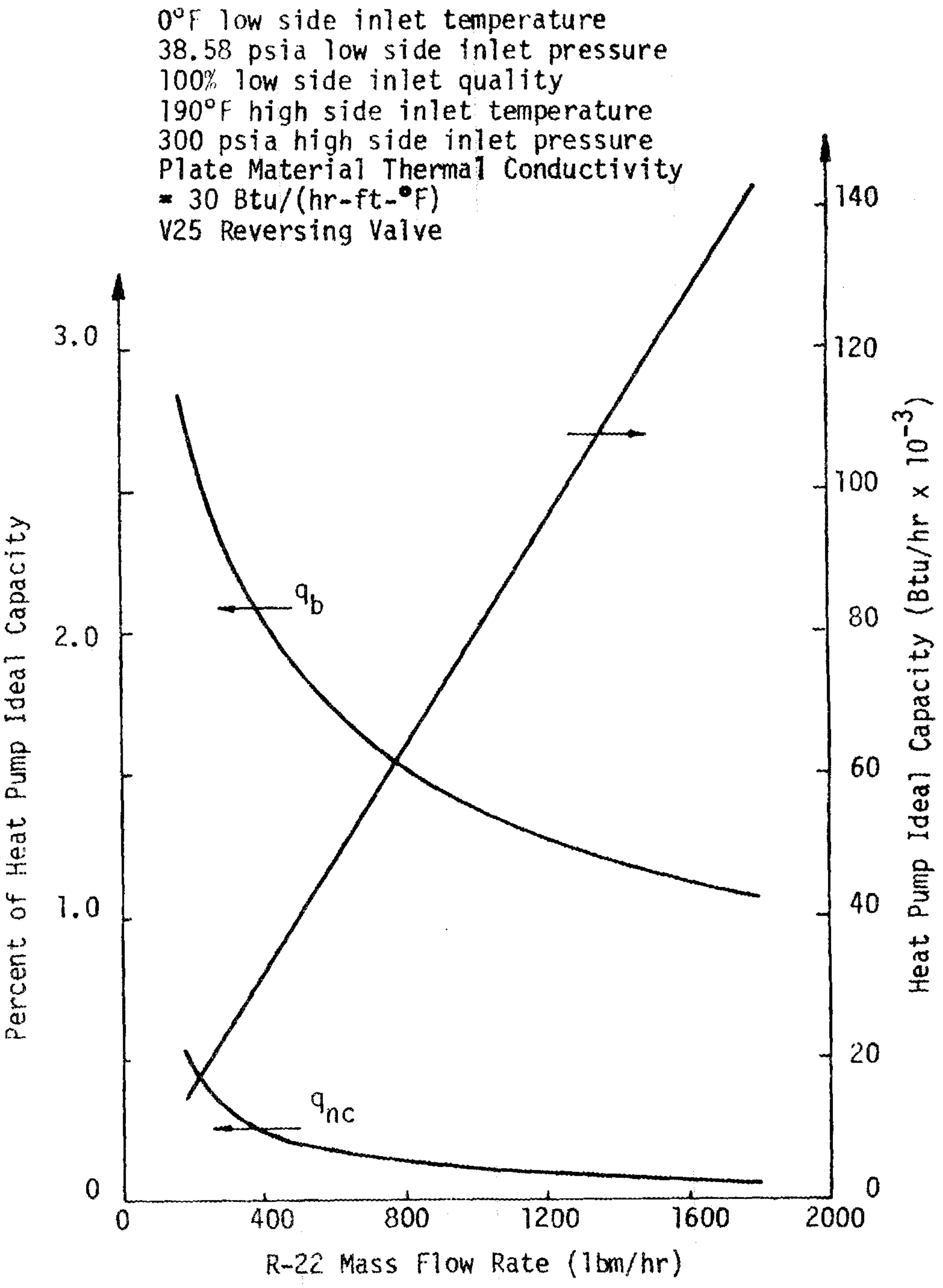


Figure 12. Heat Transfer from High to Low Side and Natural Convection from the Valve Body to a 0°F Ambient as a Percent of Heat Pump Ideal Capacity versus R-22 Mass Flow Rate.

Curve 692750A

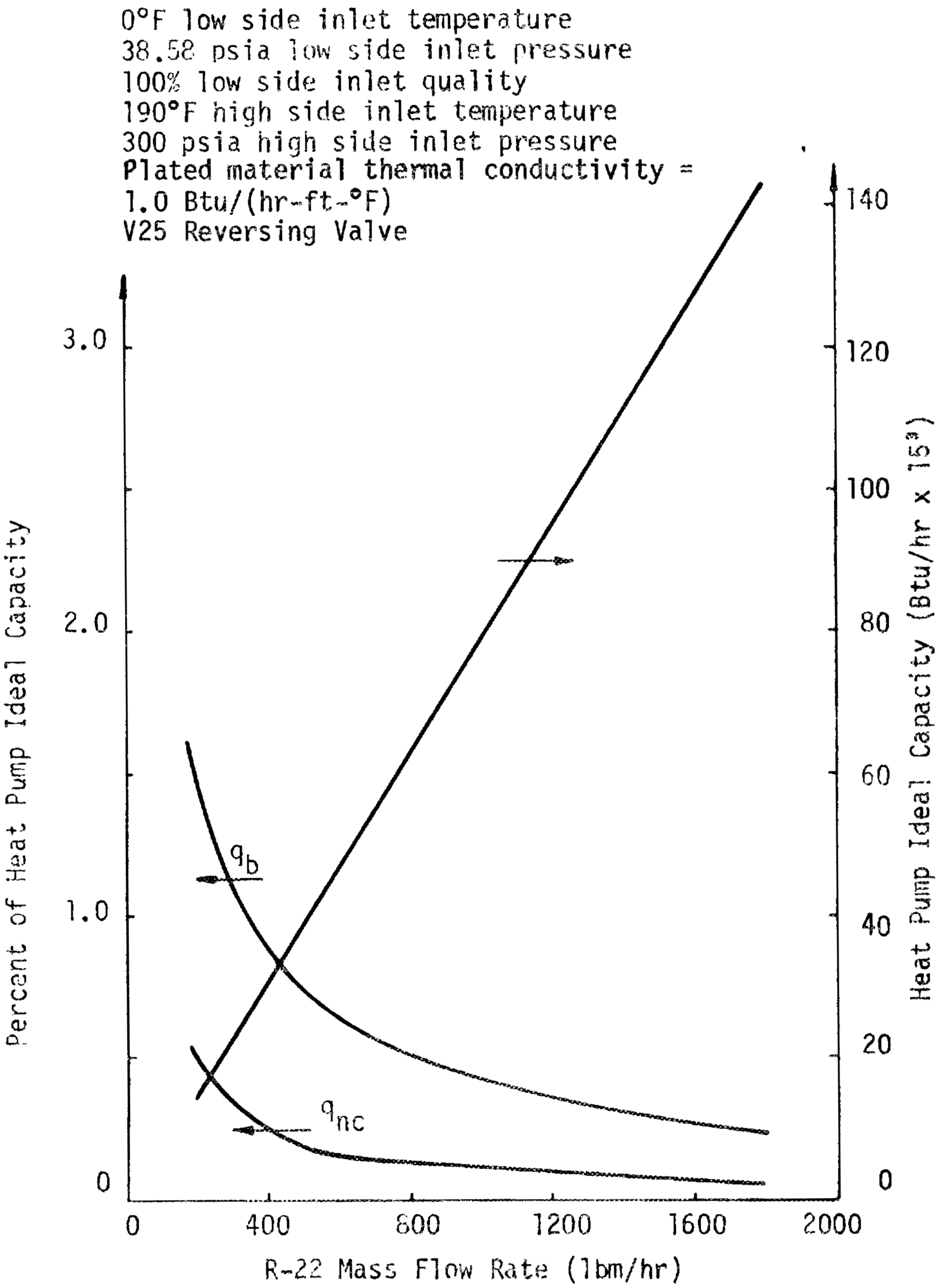


Figure 13. Heat Transfer from High to Low Side and Natural Convection from the Value Body to a 0°F Ambient as a Percent of Heat Pump Ideal Capacity versus R-22 Mass Flow Rate. The Plated Material Thermal Conductivity has been Reduced by 30.

REFERENCES

1. ASHRAE Handbook of Fundamentals, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., 1972.

2. Collier, J. G., Convective Boiling and Condensation, McGraw-Hill Book Company, 1972.

3. Hogan, M. R., "The Development of a Mathematical Model for the Study of Vapor-Compression Cycle Systems as Applied to Heated Air Forced Convection Drying Processes", Westinghouse Research Report 76-1E9-HPUMP-R2, Fluid Systems Laboratory, December 31, 1976.

4. Bae, S., Maulbetsch, J. S., and Rohsenow, W. M., "Refrigerant Forced Convection Condensation Inside Horizontal Tubes", distributed at the Industrial Liaison Symposium on Heat Transfer, April 22, 1971, sponsored by Technical Committee 1.3, ASHRAE, Contract Number ASHRAE RP63.

5. McAdams, W. H., Heat Transmission, Third Edition, McGraw-Hill Book Company, Inc., New York, 1954.

6. "RANCO Reversing Valves for Heat Pump Air Conditioning Systems", RANCO Bulletin 1919-1, RANCO, Columbis, Ohio.

7. Stewart, W. A., Private Communication, January 30, 1978.

PERMANENT RECORD BOOK ENTRIES

None

NOMENCLATURE

Ac	Cross sectional area (ft^2).
As	Surface area (ft^2).
C	An empirical constant employed in Equation (5) and found in Reference 2 (dimensionless).
C1	Expansion coefficient (dimensionless).
C2	Contraction coefficient (dimensionless).
C3	An empirical constant, defined in Equation (2) (dimensionless).
D	Diameter (ft).
$\tilde{D}$	Diameter (in).
f	Friction coefficient (dimensionless).
G	Mass velocity, defined in Equation (10) ($lbm/(ft^2\text{-}sec^2)$).
g	Gravitational constant ($lbm\text{-}ft/(lbf\text{-}sec^2)$).
h	Heat transfer coefficient ($Btu/(hr\text{-}ft^2\text{-}°F)$).
$\tilde{h}$	Specific enthalpy (Btu/lbm).
k	Thermal conductivity (Btu/(hr-ft-°F)).
Kb	A loss coefficient (dimensionless).
L	A length (ft).
m	Refrigerant mass (lbm).
P	Absolute pressure (lbf/ft^2).
$\tilde{P}$	Perimeter (ft).
Pr	Prandtl number (dimensionless).
q	Heat transfer rate (Btu/hr).
R	Sliding return bend radius of curvature (ft).
T	Temperature (°F).
t	Thickness (ft).
V	Velocity (ft/sec).
v	Specific volume (ft^3/lbm).

X	A two-phase mixture parameter defined in Equation (6) (dimensionless).
x	Mass quality (dimensionless).
Z	Distance (ft).

Greek Symbols

ΔP	Exit minus inlet pressure (lbf/ft^2)
ΔP_{bend}	Irreversible pressure loss through the sliding return bend (lbf/ft^2)
ΔP_{low}	Irreversible pressure loss through the low-side flow path (lbf/ft^2)
$\Delta P_{high\ side}$	Static pressure change through the high pressure flow circuit (lbf/ft^2).
$\Delta P_{internal}$	Irreversible pressure loss through high-side flow path (lbf/ft^2).
ΔP_{tube}	Irreversible pressure loss through valve body stub tube connections (lbf/ft^2).
ΔP_{rise}	Static pressure rise due to a decrease in flow velocity (lbf/ft^2)
μ	Dynamic viscosity (lbm/(ft-sec))
ρ	Density (lbm/ft^3).

Subscripts

a	Accumulator port.
amb	Ambient.
b	Sliding return bend.
bend	Sliding return bend.
c	Condenser port.
cp	Compressor port.
dry	No surface condensation.
e	Evaporator port.
eq	Equivalent.
f	Liquid component.
flange	Return bend flange.
fg	Difference between gas and liquid component value.

fo	A quantity calculated as if the liquid were flowing alone with the same mass velocity as the mixture.
g	Gas component.
hs	High side.
ℓs	Low side.
nc	Natural convection.
ref	Reference value.
TP	Two-phase.
vb	Valve body.
1	Material 1.
2	Material 2.

Superscripts

$\cdot$	Time derivative (1/sec).
$'$	Effective value for heat conduction parallel to the surface.
$''$	Effective value for heat conduction normal to the surface.

APPENDIX A

RANCO REVERSING VALVE PERFORMANCE AND CONSTRUCTION DATA

The performance data employed in determining the empirical constant C3 can be seen in Figure A-1. Figures A-2 and A-3 are the outline dimension drawings of the three Ranco reversing valves modeled.

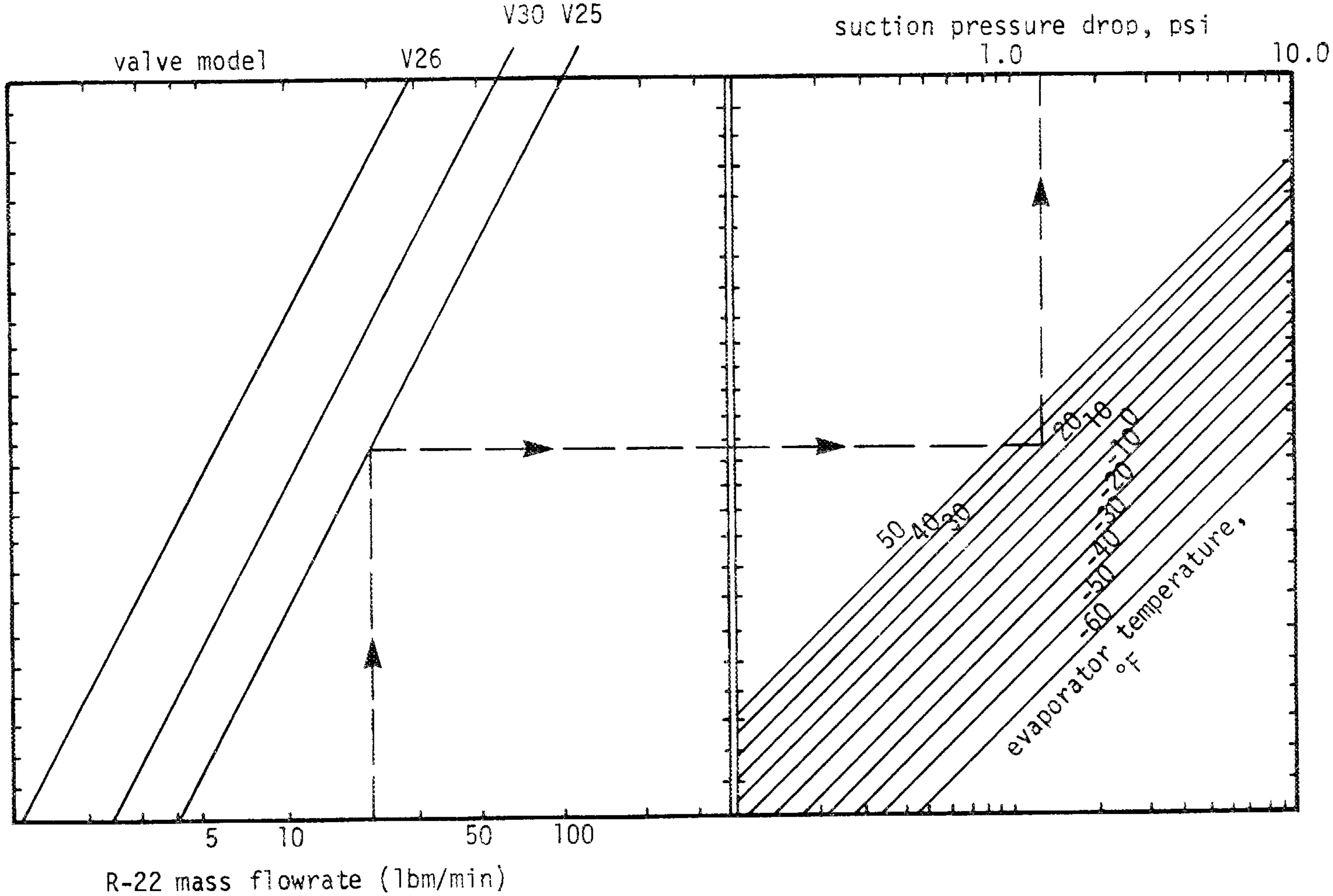
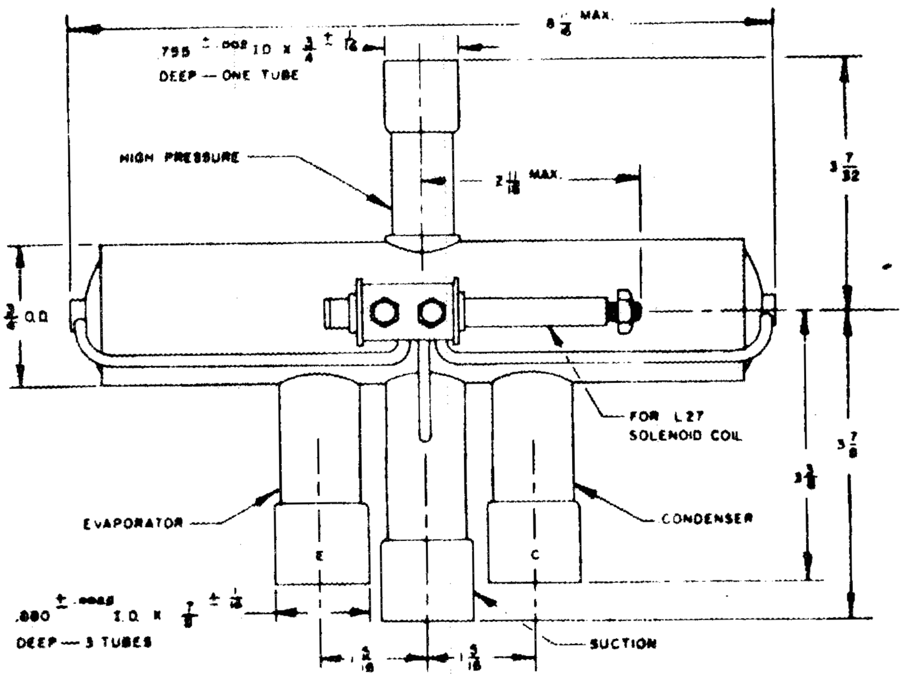


Figure A-1. The Suction Pressure Drop in Nomograph Form as a Function of Evaporator Temperature, R-22 Mass Flowrate, and Valve Model. The Above Plot was Reproduced from a Ranco Brochure (Reference 6).

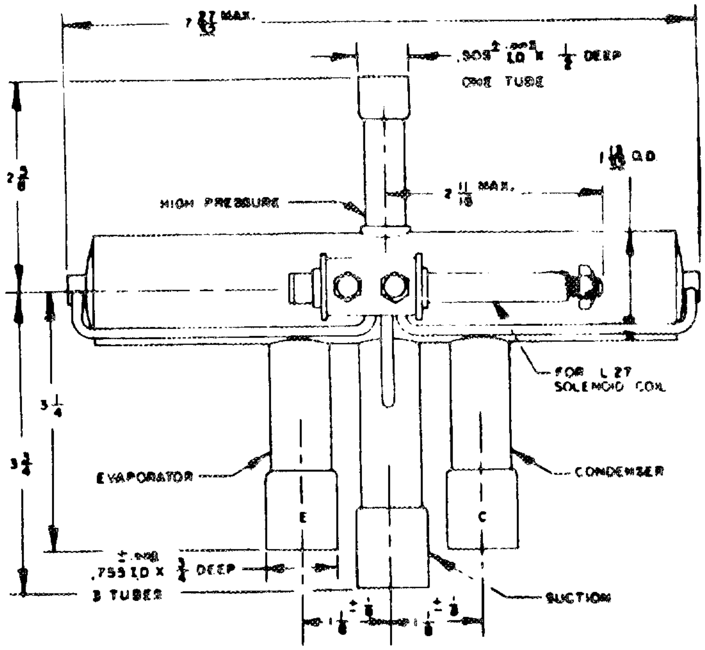
Dwg. 6260A93



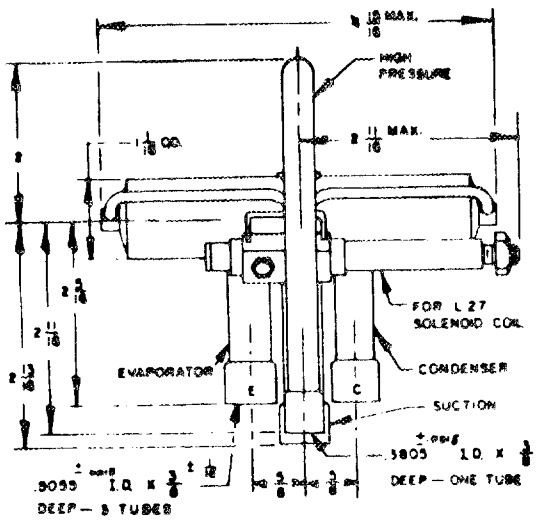
Type V25

Figure A-2. An Outline Dimension Drawing of the Ranco V25 Reversing Valve.

Dwg. 6260A94



Type V30



Type V26

Figure A-3. An Outline Dimension Drawing of the Ranco V26 and V30 Reversing Valves.

APPENDIX B

CODE LISTINGS FOR THE V25, V26, AND V30 REVERSING VALVES

Contained within this appendix are the program listings for the three reversing valves. The listing for the V25 valve may be found in Table B-I, the V26 in Table B-II, and the V30 in Table B-III. The subroutines are identical except for the physical input data and the R/D adjustment made in the V26 routine. Comment cards are employed to enable one to follow the program logic.

Table B-I - The V25 Reversing Valve Subroutine Listing

```
C      SUBROUTINE VALVE WILL CALCULATE THE CONDENSER AND
C      ACCUMULATOR PORT THERMODYNAMIC STATES GIVEN THE
C      COMPRESSOR AND EVAPORATOR INLET PORT STATES FOR THE
C      *RANCO* TYPE V25 REVERSING VALVE.  IT HAS BEEN ASSUMED
C      THAT THE HI-RE-LI SYSTEM IS BEING USED, I.E., THE
C      SUCTION PORT REFRIGERANT STATE IS SUCH THAT THE MASS
C      QUALITY IS LESS THAN 1.0.  IF THE PROGRAM CALCULATES
C      THE ACCUMULATOR PORT STATE TO BE SUPERHEATED, THE
C      ASSOCIATED MASS QUALITY WILL BE RETURNED AT 1.0
C      (THE CORRECT PRESSURE AND TEMPERATURE WILL BE
C      RETURNED ALSO).
C      TAMB-THE REVERSING VALVE*S AMBIENT TEMPERATURE (DEG.F)
C      PE,TE,XE-THE EVAPORATOR PORT PRESSURE (PSIA),
C      TEMPERATURE (DEG.F), AND MASS QUALITY (FRACTION LESS
C      THAN 1.0)
C      PCP,TCP-THE COMPRESSOR PORT PRESSURE (PSIA), AND
C      TEMPERATURE (DEG.F)
C      MDOT-REFRIGERANT MASS FLOWRATE (LBM/HR)
C      XA,PA,TA-THE ACCUMULATOR PORT OUTLET MASS QUALITY
C      (FRACTION WHICH SHOULD BE LESS THAN 1.0--SEE
C      INTRODUCTORY REMARKS), PRESSURE (PSIA), AND TEMPERATURE
C      (DEG.F).
C      PC,TC-THE CONDENSER PORT PRESSURE (PSIA), AND TEMP-
C      ERATURE (DEG.F)
C
```

Table B-I (cont'd)

```
      REAL MDOT,KF,KV,KTURN,LENG,MUFHI,MUFLO,MUVHI,MUVLO,KCOAT
      IPRINT=1
      PIE=3.141
      C1=1.0
C
C     WHERE C1 IS THE NUMBER OF VELOCITY HEADS LOST AS THE
C     GAS FROM THE COMPRESSOR ENTERS THE VALVE
C
      C2=.85
C
C     WHERE C2 IS THE NUMBER OF V.H.'S LOST AS THE GAS EXITS
C     THE VALVE IN ROUTE TO THE CONDENSER
C
      C3=100.9
C
C     WHERE C3 IS THE EQUIVALENT RETURN BEND LENGTH CONSTANT
C     WHERE DTI IS THE INSIDE DIAMETER (IN) OF THE 3 SUCTION
C     PORTS (LOCATED OPPOSITE THE COMPRESSOR PORT)
C
      DTI=.75
C
C     WHERE DTI IS THE INSIDE DIAMETER (IN) OF THE 3 SUCTION
C     PORTS (LOCATED OPPOSITE THE COMPRESSOR PORT)
C
      XLEVC=3.0+1.0/16.0
C
C     WHERE XLEVC IS THE LENGTH (IN) OF THE EVAPORATOR AND
C     CONDENSER TUBE CONNECTIONS
C
      XLACC=3.0+9.0/16.0
C
C     WHERE XLACC IS THE LENGTH (IN) OF THE ACCUMULATOR
C     TUBE CONNECTION
C
      XLCMP=3.0
C
C     WHERE XLCMP IS THE COMPRESSOR CONNECTOR LENGTH (IN)
C
      DTC=21.0/32.0
C
C     WHERE DTC IS THE INSIDE DIAMETER (IN) OF THE COMPRESSOR
C     PORT
C
      RADIUS=1.3125/2.0
C
C     WHERE RADIUS IS THE RADIUS (IN) OF CURVATURE FOR THE
C     SLIDING RETURN BEND AND IS APPROXIMATED AS HALF THE
C     DISTANCE BETWEEN THE EVAPORATOR AND ACCUMULATOR SUCTION
C     PORT CENTERLINES
```

Table B-I (cont'd)

```
C
      PERIM=8.75
C
C     WHERE PERIM IS TWICE THE PERIMETER (IN) OF THE SLIDING
C     RETURN BEND*S PERIPHERY
C
      THICK=.047
C
C     WHERE THICK IS THE THICKNESS (IN) OF THE SLIDING
C     RETURN BEND
C
      TCOAT=.002
C
C     WHERE TCOAT IS THE THICKNESS(IN) OF THE BEND PLATING MATERIAL
C
      LENG=.312
C
C     WHERE LENG IS THE LENGTH (IN) OF THE SLIDING SURFACE
C     OF THE BEND
C
      KTURN=30.0
C
C     WHERE KTURN IS THE THERMAL CONDUCTIVITY(BTU/(HR-FT-DEG F)
C     OF THE RETURN BEND MATERIAL
C
      KCOAT=223.0
C
C     WHERE KCOAT IS THE THERMAL CONDUCTIVITY(BTU/HR-FT-DEG. F)
C     OF THE RETURN BEND PLATING MATERIAL
C
      DTO=1.75
C
C     WHERE DTO IS THE OUTSIDE DIAMETER(IN) OF THE VALVE
C
      XLVAL=8.0+11.0/16.0
C
C     WHERE XLVAL IS THE VALVE LENGTH(IN)
C
      FRACT=.66
C
C     WHERE FRACT IS THE FRACTION OF THE AREA WHICH IS
C     EXPOSED TO THE HIGH VELOCITY COMPRESSOR PORT GAS
C
      BTHICK=THICK-TCOAT
C
C     CALCULATE REQUIRED HIGH AND LOW PRESSURE PROPERTIES
C
```

```
      TSATCP=SATEMP(PCP)
      HFGHI=VAPHFG(TSATCP)
      HFGLO=VAPHFG(TE)
      HFLC=HLOSAT(TE)
      HHITN=GASH(PCP,TCP)
      HLOTN=XF*HFGLC+HFLO
      MUFHI=XLOVIS(TSATCP)
      MUVHI=(GASVIS(TCP)/GVISAT(SATEMP(14.7)))*GVISAT(TSATCP)
      MUFLO=XLOVIS(TE)
      MUVLO=GVISAT(TE)
      RHOFHT=DENLIQ(TCP)
      RHOVHT=GASRHO(PCP,TCP)
      RHOFLO=DENLIQ(TE)
      RHOVLO=GASRHO(PE,TE)
      CPVHI=CPSATG(TSATCP)
      CPFHI=SPHTLQ(TSATCP)
      KF=XQCOND(TSATCP)
      KV=SATGSK(TSATCP)
C
C     CALCULATE BEND MATERIAL CONDUCTIVITY FOR CONDUCTION PERPENDICULAR
C     TO THE SURFACE
C
      XKPER=THICK/(TCOAT/KCOAT+PTHICK/KTUBN)
C
C     THE FOLLOWING HEAT TRANSFER CALCULATIONS ARE FOR THE HIGH
C     PRESSURE SIDE AND ASSUME THE LOW SIDE OF THE BEND TO BE
C     AT THE SATURATION TEMPERATURE
C
      HDRY=(2.0*KV/(RADIUS/12.0))*.036*(2.0*(RADIUS/12.0)*MDOT/(PIE*DTC*
     1*2*MUVHT/144.0))**.8*(CPVHI*MUVHI/KV)**.333
      RFILM=1.0/(HDRY*PIE*FRACT*(RADIUS/12.0)**2)
      RWALL=2.0*THICK/(12.0*XKPER*PIE*FRACT*(RADIUS/12.0)**2)
      Q=(TCP-TE)/(RFILM+RWALL)
      TWALL=TE+Q*RWALL
      IF (TWALL.GT.TSATCP) GO TO 100
C
C     SINCE CONDENSATION MUST OCCUR IF LOGIC DOES NOT GO
C     TO 6, THEN FILM COEFFICIENT MUST BE CORRECTED TO REFLECT
C     THE CHANGE IN PHYSICS
C
      HWET=HDRY*(KF/KV)*SQRT(GVISAT(TSATCP)/MUFHT)*(CPFHI*KV*MUFHI/(CPVH
     1T*KF*GVISAT(TSATCP)))**.333*(RHOFHI/GASRHO(PCP,TSATCP))**(1.0/6.0)
      RFILM=1.0/(HWET*FRACT*PIE*(RADIUS/12.0)**2)
      Q=(TSATCP-TE)/(RFILM+RWALL)
      TWALL=TE+Q*RWALL
      PRINT 130, HWET
  100 CONTINUE
```

```
C
C     CALCULATE BEND MATERIAL THERMAL CONDUCTIVITY FOR CONDUCTION
C     PARALLEL TO SEAL SURFACE
C
      XKPAR=(KCOAT*TCOAT+KTURN*PTHICK)/THICK
C
C     NOW CALCULATE THE HEAT TRANSFER THROUGH THE SEALING
C     FACE OF THE BEND.  IT WILL BE TREATED AS A
C     FIN WITH ONE FACE INSULATED.
C
      QSEAL=SQRT(HDRY*(PERIM/12.0)**2*XKPAR*(THICK/12.0))*((TCP-TWALL)*T
     1ANH((LENG/12.0)*SQRT(HDRY/(XKPAR*THICK/12.0))))
      DELT=TCP-TAMB
      IF (DELT.LT.0.0) DELT=0.0
      HLAM=.27*(DELT*12.0/DTO)**.25
      HTURB=.18*DELT**.3333333
      HUSE=HLAM
      IF (HTURB.GT.HLAM) HUSE=HTURB
      QNAT=HUSE*PIE*(DTO/12.0)*(XLVAL/12.0)*DELT
      QTOTAL=Q+QSEAL+QNAT
      HHIOUT=HHIIN-QTOTAL/MDOT
      IF (IPRINT.EQ.1) WRITE (6,140) HDRY,Q,QSEAL,QNAT
      HLOOUT=HLOIN+(Q+QSEAL)/MDOT
      XA=(HLOOUT-HFLO)/HFGLO
C
C     THE FOLLOWING PRESSURE DROP CALCULATIONS ARE FOR
C     THE HIGH PRESSURE SIDE
C
      VCP=MDOT/(3600.0*RHOVHI*PIE*DTC**2/(4.0*144.0))
      VC=VCP*(DTC/DTI)**2
      DPRISE=(.5*RHOVHI/32.2)*(VCP**2-VC**2)/144.0
      VIS=GASVIS(TCP)*GVISAT(TSATCP)/GVISAT(SATEMP(14.7))
      DPLOSS=(.5*RHOVHI/(32.2*144.0))*(C1*VCP**2+C2*VC**2+VCP**2*(XLCMP/
     1DTC)*.316*(RHOVHI*VCP*DTC/(12.0*VIS/3600.0))**(-.25)+VC**2*(XLACC/
     2DTI)*.316*(RHOVHI*VC*DTI/(12.0*VIS/3600.0))**(-.25))
      PC=PCP+DPRISE-DPLOSS
      DPHI=DPLOSS-DPRISE
      TC=TEMPH(PC,HHIOUT)
      IF (IPRINT.EQ.1) WRITE (6,150) XA,VCP,VC,DPRISE,DPLOSS,PC,TC
C
C     NEXT SET OF PRESSURE DROP CALCULATIONS ARE FOR THE
C     TWO PHASE LOW PRESSURE SIDE
C
      X=(XA+XE)/2.0
      IF (X.GT.1.0) X=1.0
      IF (X.LE.0.0) X=1.0E-13
      RX=(1.0-X)/X
```

Table B-I (cont'd)

```
      IF (RX.LT.0.0) RX=0.0
      IF (RX.GT.1.0E+8) RX=1.0E+8
      RRHO=RHOVLO/RHOFLO
      RMU=MUFLO/MUVLO
      RD=RADIUS/DTI
      C=(1.0+(20.0*(DTI/12.0)*RD**.6349/C3)*((RHOFLO-RHOVLO)/RHOFLO))*(S
     1QRT(1.0/RRHO)+SQRT(RRHO))
      GE=MDOT*4.0/(PIE*(DTI/12.0)**2)
      BETA=(.5*(GE/3600.0)**2*C3*RD**(-.6349)*.316*(GE*DTI/(12.0*MUFLO))
     1**(-.25))/(32.2*144.0*RHOFLO)
      DPTFI=BETA*(1.0-X)**(7.0/4.0)+BETA*X**(7.0/4.0)*(C*RX**.875/(SQRT(
     1RRHO)*RMU**.125)+1.0/(RRHO*RMU**.25))
      BETA=BETA/(C3*RD**(-.6349))
      C=20.0
      DPTUBE=BETA*(1.0-X)**(7.0/4.0)+BETA*X**(7.0/4.0)*(C*RX**.875/(SQRT
     1(RRHO)*RMU**.125)+1.0/(RRHO*RMU**.25))*(XLEVC+XLACC)/DTI
      PA=PE-DPTUBE-DPTFI
      DPLOW=PA-PE
      XA=(HLOOUT-HLGSAT(SATEMP(PA)))/VAPHFG(SATEMP(PA))
      IF (XA.LE.1.0) GO TO 110
      TA=TEMPH(PA,HLOOUT)
      XA=1.0
      GO TO 120
  110 TA=SATEMP(PA)
  120 CONTINUE
      IF (IPRINT.EQ.1) WRITE (6,160) DPHI,DPLOW,PCP,TCP,PC,TC,PE,TE,XE,P
     1A,TA,XA
      RETURN
C
  130 FORMAT (1X,29HCONDENSATION WILL OCCUR-HWET=,E10.3)
  140 FORMAT (1X,5HHDRY=,E10.3,2HQ=,E10.3,6HQSFAL=,E10.3,5HQNAT=,E10.3)
  150 FORMAT (1X,3HXA=,E10.3,4HVCP=,E10.3,3HVC=,E10.3,7HDPRISE=,E10.3,/,
     11X,7HDPLOSS=,E10.3,3HPC=,E10.3,3HTC=,E10.3)
  160 FORMAT (1X,5HDPHI=,E10.3,6HDPLOW=,E10.3,4HPCP=,E10.3,4HTCP=,E10.3,
     1/,1X,3HPC=,E10.3,3HTC=,E10.3,3HPE=,E10.3,3HTE=,E10.3,3HXE=,E10.3,/
     2,1X,3HPA=,E10.3,3HTA=,E10.3,3HXA=,E10.3)
C
      END
```

Table B-II - The V26 Reversing Valve Subroutine Listing

```
      SUBROUTINE VALVE (PE,TE,XE,PCP,TCP,MDOT,TAMB,XA,PA,TA,PC,TC)
C**********IMPORTANT**********
C     THE SUBROUTINE-DUMMY-MUST BE CALLED AT LEAST ONCE
C     PRIOR TO THE USE OF SUBROUTINE VALVE.  THIS IS TO
C     ALLOW THE R22 PROPERTY ROUTINE TO FILL THE DATA ARRAYS
C***********
C     SUBROUTINE VALVE WILL CALCULATE THE CONDENSER AND
C     ACCUMULATOR PORT THERMODYNAMIC STATES GIVEN THE
C     COMPRESSOR AND EVAPORATOR INLET PORT STATES FOR THE
C     *RANCO* TYPE V26 REVERSING VALVE.  IT HAS BEEN ASSUMED
C     THAT THE HI-PE-LI SYSTEM IS BEING USED, I.E., THE
C     SUCTION PORT REFRIGERANT STATE IS SUCH THAT THE MASS
C     QUALITY IS LESS THAN 1.0.  IF THE PROGRAM CALCULATES
C     THE ACCUMULATOR PORT STATE TO BE SUPERHEATED, THE
C     ASSOCIATED MASS QUALITY WILL BE RETURNED AT 1.0
C     (THE CORRECT PRESSURE AND TEMPERATURE WILL BE
C     RETURNED ALSO).
C     TAMB-THE REVERSING VALVE*S AMBIENT TEMPERATURE (DEG.F)
C     PE,TE,XE-THE EVAPORATOR PORT PRESSURE (PSIA),
C     TEMPERATURE (DEG.F), AND MASS QUALITY (FRACTION LESS
C     THAN 1.0)
C     PCP,TCP-THE COMPRESSOR PORT PRESSURE (PSIA), AND
C     TEMPERATURE (DEG.F)
C     MDOT-REFRIGERANT MASS FLOWRATE (LBM/HR)
C     XA,PA,TA-THE ACCUMULATOR PORT OUTLET MASS QUALITY
C     (FRACTION WHICH SHOULD BE LESS THAN 1.0--SEE
C     INTRODUCTORY REMARKS), PRESSURE (PSIA), AND TEMPERATURE
C     (DEG.F).
C     PC,TC-THE CONDENSER PORT PRESSURE (PSIA), AND TEMP-
C     ERATURE (DEG.F)
      REAL MDOT,KE,KV,KTURN,LENG,MUEHI,MUELO,MUVHI,MUVLO
      IPRINT=0
      PIE=3.141
      C1=1.0
C     WHERE C1 IS THE NUMBER OF VELOCITY HEADS LOST AS THE
C     GAS FROM THE COMPRESSOR ENTERS THE VALVE
      C2=.85
C     WHERE C2 IS THE NUMBER OF V.H.*S LOST AS THE GAS EXITS
C     THE VALVE IN ROUTE TO THE CONDENSER
      C3=100.0
C     WHERE C3 IS THE EQUIVALENT RETURN BEND LENGTH CONSTANT
      DTI=13.0/32.0
C     WHERE DTI IS THE INSIDE DIAMETER (IN) OF THE 3 SUCTION
C     PORTS (LOCATED OPPOSITE THE COMPRESSOR PORT)
      XLEVC=2.0+3.0/32.0+5.0/16.0
```

Table B-II (cont'd)

```
C       WHERE XLEVC IS THE LENGTH (IN) OF THE EVAPORATOR AND
C       CONDENSER TUBE CONNECTIONS
        XLACC=XLEVC+.5
C       WHERE XLACC IS THE LENGTH (IN) OF THE ACCUMULATOR
C       TUBE CONNECTION
        XLCMP=3.0+7.0/32.0+PIE*1.125
C       WHERE XLCMP IS THE COMPRESSOR CONNECTOR LENGTH (IN)
        DTC=.25
C       WHERE DTC IS THE INSIDE DIAMETER (IN) OF THE COMPRESSOR
C       PORT
        RADIUS=(5.0/8.0)/2.0
C       WHERE RADIUS IS THE RADIUS (IN) OF CURVATURE FOR THE
C       SLIDING RETURN BEND AND IS APPROXIMATED AS HALF THE
C       DISTANCE BETWEEN THE EVAPORATOR AND ACCUMULATOR SUCTION
C       PORT CENTERLINES
        PERIM=4.25
C       WHERE PERIM IS TWICE THE PERIMETER (IN) OF THE SLIDING
C       RETURN BEND#S PERIPHERY
        THICK=1.0/8.0
C       WHERE THICK IS THE THICKNESS (IN) OF THE SLIDING
C       RETURN BEND
        LENG=1.0/8.0
C       WHERE LENG IS THE LENGTH (IN) OF THE SLIDING SURFACE
C       OF THE BEND
        KTURN=1.0
C       WHERE KTURN IS THE THERMAL CONDUCTIVITY(BTU/(HR*FT*DEG F))
C       OF THE RETURN BEND MATERIAL
        DTO=1.0+1.0/16.0
C       WHERE DTO IS THE OUTSIDE DIAMETER(IN) OF THE VALVE
        XLVAL=4.0+15.0/16.0
C       WHERE XLVAL IS THE VALVE LENGTH(IN)
        FRACT=.66
C       WHERE FRACT IS THE FRACTION OF THE AREA WHICH IS
C       EXPOSED TO THE HIGH VELOCITY COMPRESSOR PORT GAS
C
C       CALCULATE REQUIRED HIGH AND LOW PRESSURE PROPERTIES
C
        TSATCP=SATEMP(PCP)
        HFGHI=VAPHFG(TSATCP)
        HFGLO=VAPHFG(TE)
        HFLO=HLOSAT(TE)
        HHIIN=GASH(PCP,TCP)
        HLOIN=XE*HFGLO+HFLO
        MUFHI=XLOVIS(TSATCP)
        MUVHI=(GASVIS(TCP)/GVISAT(SATEMP(14.7)))*GVISAT(TSATCP)
        MUFLO=XLOVIS(TE)
        MUVLO=GVISAT(TE)
```

Table B-II (cont'd)

```
      RHOFHI=DENLIQ(TCP)
      RHOVHI=GASRHO(PCP,TCP)
      RHOFLO=DENLIQ(TF)
      RHOVLO=GASRHO(PF,TF)
      CPVHI=CPSATG(TSATCP)
      CPFHI=SPHTLQ(TSATCP)
      KF=XLCOND(TSATCP)
      KV=SATGSK(TSATCP)
C
C     THE FOLLOWING HEAT TRANSFER CALCULATIONS ARE FOR THE HIGH
C     PRESSURE SIDE AND ASSUME THE LOW SIDE OF THE BEND TO BE
C     AT THE SATURATION TEMPERATURE
C
      HDRY=(2.0*KV/(RADIUS/12.0))*.036*(2.0*(RADIUS/12.0)*MDOT/(PIE*DTC*
     1*2*MUVHI/144.0))**.8*(CPVHI*MUVHI/KV)**.333
      RFILM=1.0/(HDRY*PIE*FRACT*(RADIUS/12.0)**2)
      RWALL=2.0*THICK/(12.0*KTURN*PIE*FRACT*(RADIUS/12.0)**2)
      Q=(TCP-TF)/(RFILM+RWALL)
      TWALL=TF+Q*RWALL
      IF (TWALL.GT.TSATCP) GO TO 100
C
C     SINCE CONDENSATION MUST OCCUR IF LOGIC DOES NOT GO
C     TO 4, THEN FILM COEFFICIENT MUST BE CORRECTED TO REFLECT
C     THE CHANGE IN PHYSICS
C
      HWET=HDRY*(KF/KV)*SQRT(GVISAT(TSATCP)/MUFHI)*(CPFHI*KV*MUFHI/(CPVH
     1I*KF*GVISAT(TSATCP)))**.333*(RHOFHI/GASRHO(PCP,TSATCP))**(1.0/6.0)
      RFILM=1.0/(HWET*FRACT*PIE*(RADIUS/12.0)**2)
      Q=(TSATCP-TF)/(RFILM+RWALL)
      TWALL=TF+Q*RWALL
      PRINT 130, HWET
  100 CONTINUE
C
C     NOW CALCULATE THE HEAT TRANSFER THROUGH THE SEALING
C     FACE OF THE BEND.  IT WILL BE TREATED AS A
C     FIN WITH ONE FACE INSULATED.
C
      QSEAL=SQRT(HDRY*(PERIM/12.0)**2*KTURN*(THICK/12.0))*((TCP-TWALL
     1)*TANH((LENG/12.0)*SQRT(HDRY/(KTURN*THICK/12.0))))
      DELT=TCP-TAMB
      IF (DELT.LT.0.0) DELT=0.0
      HLAM=.27*(DELT*12.0/DTU)**.25
      HTURB=.18*DELT**.3333333
      HUSE=HLAM
      IF (HTURB.GT.HLAM) HUSE=HTURB
      QNAT=HUSE*PIE*(DTO/12.0)*(XLVAL/12.0)*DELT
      QTOTAL=Q+QSEAL+QNAT
```

Table B-II (cont'd)

```
      HHIOUT=HHIIN-QTOTAL/MDOT
      IF (IPRINT.EQ.1) WRITE (6,140) HDRY,Q,QSEAL,QNAT
      HLOOUT=HLOIN+(Q+QSEAL)/MDOT
      XA=(HLOOUT-HFLO)/HFGLO
C
C     THE FOLLOWING PRESSURE DROP CALCULATIONS ARE FOR
C     THE HIGH PRESSURE SIDE
C
      VCP=MDOT/(3600.0*RHOVHI*PIE*DTC**2/(4.0*144.0))
      VC=VCP*(DTC/DTI)**2
      DPRISE=(.5*RHOVHI/32.2)*(VCP**2-VC**2)/144.0
      VIS=GASVIS(TCP)*GVISAT(TSATCP)/GVISAT(SATEMP(14.7))
      DPLOSS=(.5*RHOVHI/(32.2*144.0))*(C1*VCP**2+C2*VC**2+VCP**2*(XLCMP/
     1DTC)*.316*(RHOVHI*VCP*DTC/(12.0*VIS/3600.0))**(-.25)+VC**2*(XLACC/
     2DTI)*.316*(RHOVHI*VC*DTI/(12.0*VIS/3600.0))**(-.25))
      PC=PCP+DPRISE-DPLOSS
      DPHI=DPLOSS-DPRISE
      TC=TEMPH(PC,HHIOUT)
      IF (IPRINT.EQ.1) WRITE (6,150) XA,VCP,VC,DPRISE,DPLOSS,PC,TC
C
C     NEXT SET OF PRESSURE DROP CALCULATIONS ARE FOR THE
C     TWO PHASE LOW PRESSURE SIDE
C
      X=(XA+XE)/2.0
      IF (X.GT.1.0) X=1.0
      IF (X.LE.0.0) X=1.0E-13
      RX=(1.0-X)/X
      IF (RX.LT.0.0) RX=0.0
      IF (RX.GT.1.0E+8) RX=1.0E+8
      RRHO=RHOVLO/RHOFLO
      RMU=MUFLO/MUVLO
      RD=RADIUS/DTI
      RD=RD*1.2
C     NOTE THAT THE R/D RATIO WAS ADJUSTED TO FORCE THE
C     MODEL TO FIT EXPERIMENTAL DATA
      C=(1.0+(20.0*(DTI/12.0)*RD**.6349/C3)*((RHOFLO-RHOVLO)/RHOFLO))*(S
     1QRT(1.0/RRHO)+SQRT(RRHO))
      GE=MDOT*4.0/(PIE*(DTI/12.0)**2)
      BETA=(.5*(GE/3600.0)**2*C3*RD**(-.6349)*.316*(GE*DTI/(12.0*MUFLO))
     1**(-.25))/(32.2*144.0*RHOFLO)
      DPTP1=BETA*(1.0-X)**(7.0/4.0)+BETA*X**(7.0/4.0)*(C*RX**.875/(SQRT(
     1RRHO)*RMU**.125)+1.0/(RRHO*RMU**.25))
      BETA=BETA/(C3*RD**(-.6349))
      C=20.0
      DPTUBE=BETA*(1.0-X)**(7.0/4.0)+BETA*X**(7.0/4.0)*(C*RX**.875/(SQRT
     1(RRHO)*RMU**.125)+1.0/(RRHO*RMU**.25))*(ALEVC+XLACC)/DTI
      PA=PE-DPTUBE-DPTP1
      DPLOW=PA-PE
```

Table B-II (cont'd)

```
      XA=(HLOOUT-HLQSAT(SATEMP(PA)))/VAPHFG(SATEMP(PA))
      IF (XA.LE.1.0) GO TO 110
      TA=TEMPH(PA,HLOOUT)
      XA=1.0
      GO TO 120
 110  TA=SATEMP(PA)
 120  CONTINUE
      IF (IPRINT.EQ.1) WRITE (6,160) DPHI,DPLOW,PCP,TCP,PC,TC,PE,TE,XE,P
     1A,TA,XA
      RETURN
C
 130  FORMAT (1X, 29HCONDENSATION WILL OCCUR-HWET=,F10.3)
 140  FORMAT (1X,   5HHDRY=,E10.3,  2HQ=,E10.3,  6HQSEAL=,E10.3,  5HQNAT=
     1,F10.3)
 150  FORMAT (1X,   3HXA=,F10.3,  4HVCP=,E10.3,  3HVC=,F10.3,  7HDPRISE=,
     1F10.3,/,1X,   7HDPLOSS=,F10.3,  3HPC=,E10.3,  3HTC=,E10.3)
 160  FORMAT (1X,   5HDPHI=,E10.3,  6HDPLOW=,F10.3,  4HPCP=,E10.3,  4HTCP
     1=,F10.3,/,1X,  3HPC=,F10.3,  3HTC=,E10.3,  3HPE=,F10.3,  3HTE=,F10
     2.3,  3HXE=,F10.3,/,1X,  3HPA=,E10.3,  3HTA=,F10.3,  3HXA=,E10.3)
C
      END
```

Table B-III - The V30 Reversing Valve Subroutine Listing

```
      SUBROUTINE VALVE (PE,TE,XE,PCP,TCP,MDOT,TAMB,XA,PA,TA,PC,TC)
C**********IMPORTANT**********
C     THE SUBROUTINE-DUMMY-MUST BE CALLED AT LEAST ONCE
C     PRIOR TO THE USE OF SUBROUTINE VALVE.  THIS IS TO
C     ALLOW THE R22 PROPERTY ROUTINE TO FILL THE DATA ARRAYS
C************
C     SUBROUTINE VALVE WILL CALCULATE THE CONDENSER AND
C     ACCUMULATOR PORT THERMODYNAMIC STATES GIVEN THE
C     COMPRESSOR AND EVAPORATOR INLET PORT STATES FOR THE
C     *RANCO* TYPE V30 REVERSING VALVE.  IT HAS BEEN ASSUMED
C     THAT THE HI-RE-LI SYSTEM IS BEING USED, I.E., THE
C     SUCTION PORT REFRIGERANT STATE IS SUCH THAT THE MASS
C     QUALITY IS LESS THAN 1.0.  IF THE PROGRAM CALCULATES
C     THE ACCUMULATOR PORT STATE TO BE SUPERHEATED, THE
C     ASSOCIATED MASS QUALITY WILL BE RETURNED AT 1.0
C     (THE CORRECT PRESSURE AND TEMPERATURE WILL BE
C     RETURNED ALSO).
C     TAMB-THE REVERSING VALVE#S AMBIENT TEMPERATURE (DEG.F)
C     PE,TE,XE-THE EVAPORATOR PORT PRESSURE (PSIA),
C     TEMPERATURE (DEG.F), AND MASS QUALITY (FRACTION LESS
C     THAN 1.0)
C     PCP,TCP-THE COMPRESSOR PORT PRESSURE (PSIA), AND
C     TEMPERATURE (DEG.F)
C     MDOT-REFRIGERANT MASS FLOWRATE (LBM/HR)
C     XA,PA,TA-THE ACCUMULATOR PORT OUTLET MASS QUALITY
C     (FRACTION WHICH SHOULD BE LESS THAN 1.0--SEE
C     INTRODUCTORY REMARKS), PRESSURE (PSIA), AND TEMPERATURE
C     (DEG.F).
C     PC,TC-THE CONDENSER PORT PRESSURE (PSIA), AND TEMP-
C     ERATURE (DEG.F)
      REAL MDOT,KE,KV,KTURN,LENG,MUFHI,MUFLO,MUVHI,MUVLO
      IPRINT=0
      PIE=3.141
      C1=1.0
C     WHERE C1 IS THE NUMBER OF VELOCITY HEADS LOST AS THE
C     GAS FROM THE COMPRESSOR ENTERS THE VALVE
      C2=.85
C     WHERE C2 IS THE NUMBER OF V.H.#S LOST AS THE GAS EXITS
C     THE VALVE IN ROUTE TO THE CONDENSER
      C3=100.0
C     WHERE C3 IS THE EQUIVALENT RETURN BEND LENGTH CONSTANT
      DTT=9.0/16.0
C     WHERE DTT IS THE INSIDE DIAMETER (IN) OF THE 3 SUCTION
C     PORTS (LOCATED OPPOSITE THE COMPRESSOR PORT)
      XLEVC=2.0+35.0/64.0+7.0/16.0
```

Table B-III (cont'd)

```
C     WHERE XLEVC IS THE LENGTH (IN) OF THE EVAPORATOR AND
C     CONDENSER TUBE CONNECTIONS
      XLACC=2.0+35.0/64.0+.5+7.0/16.0
C     WHERE XLACC IS THE LENGTH (IN) OF THE ACCUMULATOR
C     TUBE CONNECTION
      XLCMP=1.0+59.0/64.0
C     WHERE XLCMP IS THE COMPRESSOR CONNECTOR LENGTH (IN)
      DTC=7.0/16.0
C     WHERE DTC IS THE INSIDE DIAMETER (IN) OF THE COMPRESSOR
C     PORT
      RADIUS=(1.0+1.0/8.0)/2.0
C     WHERE RADIUS IS THE RADIUS (IN) OF CURVATURE FOR THE
C     SLIDING RETURN BEND AND IS APPROXIMATED AS HALF THE
C     DISTANCE BETWEEN THE EVAPORATOR AND ACCUMULATOR SUCTION
C     PORT CENTERLINES
      PERIM=8.0
C     WHERE PERIM IS TWICE THE PERIMETER (IN) OF THE SLIDING
C     RETURN BEND≠S PERIPHERY
      THICK=.047
C     WHERE THICK IS THE THICKNESS (IN) OF THE SLIDING
C     RETURN BEND
      LENG=3.0/16.0
C     WHERE LENG IS THE LENGTH (IN) OF THE SLIDING SURFACE
C     OF THE BEND
      KTURN=12.0
C     WHERE KTURN IS THE THERMAL CONDUCTIVITY(BTU/(HR-FT-DEG F)
C     OF THE RETURN BEND MATERIAL
      DTO=1.0+13.0/32.0
C     WHERE DTO IS THE OUTSIDE DIAMETER(IN) OF THE VALVE
      XLVAL=7.0+27.0/32.0
C     WHERE XLVAL IS THE VALVE LENGTH(IN)
      FRACT=.66
C     WHERE FRACT IS THE FRACTION OF THE AREA WHICH IS
C     EXPOSED TO THE HIGH VELOCITY COMPRESSOR PORT GAS
C
C     CALCULATE REQUIRED HIGH AND LOW PRESSURE PROPERTIES
C
      TSATCP=SATEMP(PCP)
      HFGHI=VAPHFG(TSATCP)
      HFGLO=VAPHFG(TE)
      HFLO=HLQSAT(TE)
      HHIIN=GASH(PCP,TCP)
      HLOIN=XE*HFGLO+HFLO
      MUFHI=XLQVIS(TSATCP)
      MUVHI=(GASVIS(TCP)/GVISAT(SATEMP(14.7)))*GVISAT(TSATCP)
      MUFLO=XLQVIS(TE)
      MUVLO=GVISAT(TE)
      RHOFHI=DENLIQ(TCP)
```

Table B-III (cont'd)

```
      RHOVHT=GASRHO(PCP,TCP)
      RHOFLO=DENLIQ(TF)
      RHOVLO=GASRHO(PF,TF)
      CPVHI=CPSATG(TSATCP)
      CPFHI=SPHTLQ(TSATCP)
      KF=XCOND(TSATCP)
      KV=SATGSK(TSATCP)
C
C     THE FOLLOWING HEAT TRANSFER CALCULATIONS ARE FOR THE HIGH
C     PRESSURE SIDE AND ASSUME THE LOW SIDE OF THE BEND TO BE
C     AT THE SATURATION TEMPERATURE
C
      HDRY=(2.0*KV/(RADIUS/12.0))*.036*(2.0*(RADIUS/12.0)*MDOT/(PIE*DTC*
     1*2*MUVHT/144.0))**.8*(CPVHI*MUVHT/KV)**.333
      RFILM=1.0/(HDRY*PIE*FRACT*(RADIUS/12.0)**2)
      RWALL=2.0*THICK/(12.0*KTURN*PIE*FRACT*(RADIUS/12.0)**2)
      Q=(TCP-TF)/(RFILM+RWALL)
      TWALL=TF+Q*RWALL
      IF (TWALL.GT.TSATCP) GO TO 100
C
C     SINCE CONDENSATION MUST OCCUR IF LOGIC DOES NOT GO
C     TO 6, THEN FILM COEFFICIENT MUST BE CORRECTED TO REFLECT
C     THE CHANGE IN PHYSICS
C
      HWET=HDRY*(KF/KV)*SQRT(GVTSAT(TSATCP)/MUFHI)*(CPFHI*KV*MUFHI/(CPVH
     1I*KF*GVTSAT(TSATCP)))**.333*(RHOFHI/GASRHO(PCP,TSATCP))**(1.0/6.0)
      RFILM=1.0/(HWET*FRACT*PIE*(RADIUS/12.0)**2)
      Q=(TSATCP-TF)/(RFILM+RWALL)
      TWALL=TF+Q*RWALL
      PRINT 130, HWET
  100 CONTINUE
C
C     NOW CALCULATE THE HEAT TRANSFER THROUGH THE SEALING
C     FACE OF THE BEND.  IT WILL BE TREATED AS A
C     FIN WITH ONE FACE INSULATED.
C
      QSEAL=SQRT(HDRY*(PERIM/12.0)**2*KTURN*(THICK/12.0))*((TCP-TWALL
     1)*TANH((LENG/12.0)*SQRT(HDRY/(KTURN*THICK/12.0))))
      DELT=TCP-TAMB
      IF (DELT.LT.0.0) DELT=0.0
      HLAM=.27*(DELT*12.0/DTO)**.25
      HTURB=.18*DELT**.3333333
      HUSE=HLAM
      IF (HTURB.GT.HLAM) HUSE=HTURB
      QNAT=HUSE*PIE*(DTO/12.0)*(XLVAL/12.0)*DELT
      QTOTAL=Q+QSEAL+QNAT
      HHTOUT=HHTIN-QTOTAL/MDOT
```

Table B-III (cont'd)

```
      IF (IPRINT.EQ.1) WRITE (6,140) HDRY,Q,QSEAL,QNAT
      HLOOUT=HLOIN+(Q+QSEAL)/MDOT
      XA=(HLOOUT-HFLO)/HFGLO
C
C     THE FOLLOWING PRESSURE DROP CALCULATIONS ARE FOR
C     THE HIGH PRESSURE SIDE
C
      VCP=MDOT/(3600.0*RHOVHI*PIE*DTC**2/(4.0*144.0))
      VC=VCP*(DTC/DTI)**2
      DPRISE=(.5*RHOVHI/32.2)*(VCP**2-VC**2)/144.0
      VIS=GASVIS(TCP)*GVISAT(TSATCP)/GVISAT(SATEMP(14.7))
      DPLOSS=(.5*RHOVHI/(32.2*144.0))*(C1*VCP**2+C2*VC**2+VCP**2*(XLCMP/
     1DTC)*.316*(RHOVHI*VCP*DTC/(12.0*VIS/3600.0))**(-.25)+VC**2*(XLACC/
     2DTI)*.316*(RHOVHI*VC*DTI/(12.0*VIS/3600.0))**(-.25))
      PC=PCP+DPRISE-DPLOSS
      DPHI=DPLOSS-DPRISE
      TC=TEMPH(PC,HHIOUT)
      IF (IPRINT.EQ.1) WRITE (6,150) XA,VCP,VC,DPRISE,DPLOSS,PC,TC
C
C     NEXT SET OF PRESSURE DROP CALCULATIONS ARE FOR THE
C     TWO PHASE LOW PRESSURE SIDE
C
      X=(XA+XE)/2.0
      IF (X.GT.1.0) X=1.0
      IF (X.LE.0.0) X=1.0E-13
      RX=(1.0-X)/X
      IF (RX.LT.0.0) RX=0.0
      IF (RX.GT.1.0E+8) RX=1.0E+8
      RRHO=RHOVLO/RHOFLO
      RMU=MUFLO/MUVLO
      RD=RADIUS/DTI
      C=(1.0+(20.0*(DTI/12.0)*RD**.6349/C3)*((RHOFLO-RHOVLO)/RHOFLO))*(S
     1QRT(1.0/RRHO)+SQRT(RRHO))
      GE=MDOT*4.0/(PIE*(DTI/12.0)**2)
      BETA=(.5*(GE/3600.0)**2*C3*RD**(-.6349)*.316*(GE*DTI/(12.0*MUFLO))
     1**(-.25))/(32.2*144.0*RHOFLO)
      DPTP1=BETA*(1.0-X)**(7.0/4.0)+BETA*X**(7.0/4.0)*(C*RX**.875/(SQRT(
     1RRHO)*RMU**.125)+1.0/(RRHO*RMU**.25))
      BETA=BETA/(C3*RD**(-.6349))
      C=20.0
      DPTUBE=BETA*(1.0-X)**(7.0/4.0)+BETA*X**(7.0/4.0)*(C*RX**.875/(SQRT
     1(RRHO)*RMU**.125)+1.0/(RRHO*RMU**.25))*(XLEVC+XLACC)/DTI
      PA=PE-DPTUBE-DPTP1
      DPLOW=PA-PE
      XA=(HLOOUT-HLOSAT(SATEMP(PA)))/VAPHFG(SATEMP(PA))
      IF (XA.LE.1.0) GO TO 110
      TA=TEMPH(PA,HLOOUT)
      XA=1.0
      GO TO 120
  110 TA=SATEMP(PA)
```

Table B-III (cont'd)

```
  120 CONTINUE
      IF (IPRINT.EQ.1) WRITE (6,160) DPHI,DPLOW,PCP,TCP,PC,TC,PE,TE,XE,P
     1A,TA,XA
      RETURN
C
  130 FORMAT (1X, 29HCONDENSATION WILL OCCUR-MYE1=,F10.3)
  140 FORMAT (1X,  5HHDRY=,E10.3,  2HQ=,E10.3,  6HQSEAL=,F10.3,  5HQNAT=
     1,F10.3)
  150 FORMAT (1X,  3HXA=,F10.3,  4HVCP=,E10.3,  3HVC=,F10.3,  7HDPRISE=,
     1F10.3,/,1X,  7HDPLOSS=,F10.3,  3HPC=,F10.3,  3HTC=,F10.3)
  160 FORMAT (1X,  5HDPHI=,F10.3,  6HDPLOW=,F10.3,  4HPCP=,F10.3,  4HTCP
     1=,F10.3,/,1X,  3HPC=,E10.3,  3HTC=,E10.3,  3HPE=,F10.3,  3HTE=,E10
     2.3,  3HXE=,F10.3,/,1X,  3HPA=,F10.3,  3HTA=,F10.3,  3HXA=,E10.3)
C
      END
```

APPENDIX B

Listing of VALVER Subroutine

A listing of the VALVER subprogram follows.

```
      SUBROUTINE VALVER(IV,PA,TA,XA,PCP,TCP,MDOT,TAMB,PDLO,PDHI,QLEAK,
     1      QAMB,PRINT)
C
C    ************************************************
C     SUBROUTINE VALVER WILL CALCULATE THE LOW SIDE
C     REVERSING VALVE INLET THERMODYNAMIC STATES AND
C     THE HIGH SIDE REVERSING VALVE EXIT THERMODYNAMIC
C     STATES GIVEN THE COMPRESSOR SUCTION AND DISCHARGE
C     THERMODYNAMIC STATES, THE REFRIGERANT MASS FLOW RATE
C     AND THE REVERSING VALVE AMBIENT TEMPERATURE FOR THE
C     EMPIRICAL MODEL BASED ON BREADBOARD TEST RESULTS AND THE
C    *RANCO* TYPE V25,V26 AND V30 REVERSING VALVES.
C     THE CASE WHERE THE COMPRESSOR SUCTION CONDITIONS
C     ARE SLIGHTLY WET IS INCLUDED.
C
C     INPUTS:
C           IV   = VALVE IDENTIFIER (INTEGER)
C                   1 FOR EMPIRICAL MODEL
C                  25 FOR V25
C                  26 FOR V26
C                  30 FOR V30
C           PA   = COMPRESSOR SUCTION PRESSURE - PSIA
C           TA   = COMPRESSOR SUCTION TEMPERATURE - DEF F
C           XA   = COMPRESSOR SUCTION QUALITY (1.0 = 100%)
C           PCP  = COMPRESSOR DISCHARGE PRESSURE - PSIA
C           TCP  = COMPRESSOR DISCHARGE TEMPERATURE - DEG F
C           MDOT = REFRIGERANT MASS FLOW RATE - LBM/HR
```

```
C           TAMB = VALVE AMBIENT TEMPERATURE - DEG F
C
C       OUTPUTS:
C           PDLO = LOW SIDE PRESSURE DROP (PSI)
C           PDHI = HIGH SIDE PRESSURE DROP (PSI)
C           QLEAK= INTERNAL HEAT LEAK (BTU/HR)
C           QAMB = HEAT LEAK TO AMBIENT FROM
C                  HIGH SIDE (BTU/HR)
C
        REAL MDOT,KF,KV,KTURN,LENG,MUFHI,MUVHI,MUVLO,MUFLO,
     1         KCOAT
        REAL MUSAT,MGX,KGX,MFX,MVX,MUFHX,MUREF,KXF,KXG
        LOGICAL PRINT
        DIMENSION ADTI(3),ALEV(3),ALAC(3),ALCM(3),ADTC(3),ARAD(3),
     1         APER(3),ATHK(3),ATCO(3),ALEN(3),AKTU(3),AKCO(3),
     2         ADTO(3),ALVA(3)
        DATA ADTI/ 0.750,0.40625,0.5625/
        DATA ALEV/ 3.0625,2.40625,2.984375/
        DATA ALAC/ 3.5625,2.90625,1.921875/
        DATA ALCM/ 3.0,6.75304,1.921875/
        DATA ADTC/ 0.65625,0.25,0.4375/
        DATA ARAD/ 0.65625,0.3125,0.5625/
        DATA APER/ 8.75,4.25,8.0/
        DATA ATHK/ 0.047,0.125,0.047/
        DATA ATCO/ 0.002,0.0,0.0/
        DATA ALEN/ 0.312,0.125,0.1875/
        DATA AKTU/ 30.0,1.0,12.0/
        DATA AKCO/ 223.0,1.0,1.0/
        DATA ADTO/ 1.75,1.0625,1.40625/
        DATA ALVA/ 8.6875,4.9375,7.84375/
        IPRINT = 0
        IF(PRINT) IPRINT = 1
        IF(IPRINT .EQ. 1) WRITE(6,196) IV,PA,TA,XA,PCP,TCP,MDOT,TAMB
        IF(IV .EQ. 1) GO TO 2000
```

```
      JV = 1
      IF(IV .EQ. 26)  JV=2
      IF(IV .EQ. 30)  JV=3
      PIE = 3.1415927
      C1 = 1.0
C
C     WHERE C1 IS THE NUMBER OF VELOCITY HEADS LOST AS THE
C     GAS FROM THE COMPRESSOR ENTERS THE VALVE
C
C
      C2 = 0.85
C
C     WHERE C2 IS THE NUMBER OF V.H.'S LOST AS THE GAS EXITS
C     THE VALVE IN ROUTE TO THE CONDENSER
C
      C3 = 100.9
C
C     WHERE C3 IS THE EQUIVALENT RETURN BEND LENGTH CONSTANT
C
      DTI = ADTI(JV)
C
C     WHERE DTI IS THE INSIDE DIAMETER (IN) OF THE 3 SUCTION
C     PORTS (LOCATED OPPOSITE THE COMPRESSOR PORT)
C
      XLEVC = ALEV(JV)
C
C     WHERE XLEVC IS THE LENGTH (IN) OF THE EVAPORATOR AND
C     CONDENSER TUBE CONNECTIONS
C
      XLACC = ALAC(JV)
C
C     WHERE XLACC IS THE LENGTH (IN) OF THE ACCUMULATOR
C     TUBE CONNECTION
C
```

```
      XLCMP = ALCM(JV)
C
C     WHERE XLCMP IS THE COMPRESSOR CONNECTOR LENGTH (IN)
C
      DTC = ADTC(JV)
C
C     WHERE DTC IS THE INSIDE DIAMETER (IN) OF THE COMPRESSOR
C     PORT
C
      RADIUS = ARAD(JV)
C
C     WHERE RADIUS IS THE RADIUS (IN) OF CURVATURE FOR THE
C     SLIDING RETURN BEND AND IS APPROXIMATED AS HALF THE
C     DISTANCE BETWEEN THE EVAPORATOR AND ACCUMULATOR SUCTION
C     PORT CENTERLINES
C
      PERIM = APER(JV)
C
C     WHERE PERIM IS TWICE THE PERIMETER (IN) OF THE SLIDING
C     RETURN BEND PERIPHERY
C
      THICK = ATHK(JV)
C
C     WHERE THICK IS THE THICKNESS (IN) OF THE SLIDING
C     RETURN BEND
C
      LENG = ALEN(JV)
C
C     WHERE LENG IS THE LENGTH (IN) OF THE SLIDING SURFACE
C     OF THE BEND
C
      KTURN = AKTU(JV)
C
C     WHERE KTURN IS THE THERMAL CONDUCTIVITY (BTU/(HR-FT-DEG.F)
```

```
C     OF THE RETURN BEND MATERIAL
C
      DTO = ADTO(JV)
C
C     WHERE DTO IS THE OUTSIDE DIAMETER (IN) OF THE VALVE
C
      XLVAL = ALVA(JV)
C
C     WHERE XLVAL IS THE VALVE LENGTH (IN)
C
      FRACT = 0.66
C
C     WHERE FRACT IS THE FRACTION OF THE AREA WHICH IS
C     EXPOSED TO THE HIGH VELOCITY COMPRESSOR PORT GAS
C
      KCOAT = AKCO(JV)
C
C     WHERE KCOAT IS THE THERMAL CONDUCTIVITY (BTU/HR-FT-DEG.F)
C     OF THE RETURN BEND PLATING MATERIAL
C
      TCOAT = ATCO(JV)
C
C     WHERE TCOAT IS THE THICKNESS (IN) OF THE BEND PLATING MATERIAL
C
      PTHICK = THICK-TCOAT
C
C     CALCULATE REQUIRED HIGH AND LOW PRESSURE PROPERTIES
C
C
C     FIRST ESTIMATE OF HIGH AND LOW SIDE PRESSURE DROP AND
C     HEAT LEAK
      PDLO = 2.0
      QLEAK = 200.0
      NLEAK = 0
```

```
      NPDLO = 0
      TSATA = TSAT(PA,IFLAG)
      CALL SATPRP(TSATA,PXX,VFX,VGX,HLIQA,HFGRA,HGX,SFX,SGX,IFLAG)
      IF(XA .EQ. 1.0) GO TO 1010
      HLOOUT = HLIQA + XA*HFGRA
      GO TO 1111
1010  CONTINUE
      CALL VAPOR(TA,PA,VFX,HLOOUT,SVX,IERROR)
1111  CONTINUE
      IF(NPDLO .GT. 50) GO TO 1991
      IF(NLEAK .GT. 50) GO TO 1992
      PE = PA + PDLO
      HLOIN = HLOOUT + QLEAK/MDOT
      TSATE = TSAT(PE,IFLAG)
      CALL SATPRP(TSATE,PXX,VFX,VGX,HLIQE,HFGRE,HGX,SFX,SGX,IFLAG)
      HSATE = HLIQE + HFGRE
      IF(HSATE .LT. HLOIN) GO TO 1122
      TE = TSATE
      XE = (HLOIN-HLIQE)/HFGRE
      GO TO 1133
1122  CONTINUE
      XE = 1.0
      TTRY = TSATE + 5.0
      NLOC = 1
      CALL TRIAL2(TTRY,0.5,PE,3,HLOIN,0.001,VX,HX,SX,TE,IGOOF)
1133  CONTINUE
      TSATCP = TSAT(PCP,IFLAG)
      TREFR = TSAT(14.7,IFLAG)
      CALL SATPRP(TSATCP,PXX,VOLHF,VGX,HFX,HFGHI,HGX,SFX,SGX,IFLAG)
      CALL SATPRP(TE,PXX,VOLLO,VGC,HFLO,HFGLO,HGX,SFX,SGX,IFLAG)
      CALL MUKCP(TSATCP,MUFHI,MUSAT,MGX,KF,KV,KGX,CPFHI,CPVHI,
     1           CPX,IMUKCP)
      CALL MUKCP(TCP,MFX,MVX,MUFHX,KXF,KXG,KGX,CPXF,CPXG,
     1           CPX,IMUKCP)
```

```
      CALL MUKCP(TE,MUFLO,MUVLO,MGX,KXF,KXG,KGX,CPXF,CPXG,
     1       CPX,IMUKCP)
      CALL MUKCP(TREFR,MFX,MUREF,MGX,KXF,KXG,KGX,CPXF,CPXG,
     1       CPX,IMUKCP)
      CALL VAPOR(TCP,PCP,VOLHI,HHIIN,SVX,IERROR)
      HLOIN = XE*HFGLO+HFLO
      MUVHI = (MUFHX/MUREF)*MUSAT
      RHOFHI = 1.0/VOLHF
      RHOVHI = 1.0/SPVOL(TCP,PCP)
      RHOFLO = 1.0/VOLLO
      RHOVLO = 1.0/SPVOL(TE,PE)
      IF(IPRINT .EQ. 1) WRITE(6,197) TE,PE,XE,TSATE,HLOIN,
     1       HFLO,HFGLO,RHOFLO,RHOVLO,MUFLO,MUVLO
      IF(IPRINT .EQ. 1) WRITE(6,198) TA,PA,XA,TSATA,HLOOUT,
     1       HLIQA,HFGRA
      IF(IPRINT .EQ. 1) WRITE(6,199) TCP,PCP,TSATCP,HHIIN,
     1       HFGHI,RHOFHI,RHOVHI,MUFHI,MUVHI,TREFR,CPFHI,
     2       CPVHI,KF,KV
C
C     THE FOLLOWING HEAT TRANSFER CALCULATIONS ARE FOR THE HIGH
C     PRESSURE SIDE AND ASSUME THE LOW SIDE OF THE BEND TO BE
C     AT THE SATURATION TEMPERATURE
C
      XKPER = THICK/((TCOAT/KCOAT)+(PTHICK/KTURN))
      HDRY = (2.0*KV/(RADIUS/12.0))*0.036*(2.0*(RADIUS/12.0)*MDOT/
     1       (PIE*DTC**2*MUVHI/144.0))**0.8*(CPVHI*MUVHI/KV)**0.333
      RFILM = 1.0/(HDRY*PIE*FRACT*(RADIUS/12.0)**2)
      RWALL = 2.0*THICK/(12.0*XKPER*PIE*FRACT*(RADIUS/12.0)**2)
      Q = (TCP-TE)/(RFILM+RWALL)
      TWALL = TE+Q*RWALL
      IF(TWALL .GT. TSATCP)  GO TO 100
C
C     SINCE CONDENSATION MUST OCCUR IF LOGIC DOES NOT GO
C     TO 6.0 THEN FILM COEFFICIENT MUST BE CORRECTED TO REFLECT
```

```
C     THE CHANGE IN PHYSICS
C
      HWET = HDRY*(KF/KV)*SQRT(MUSAT/MUFHI)*(CPFHI*KV*MUFHI/
     1       (CPVHI*KF*MUSAT))**0.333*(RHOFHI*SPVOL(TSATCP,
     2       PCP))**(1.0/6.0)
      RFILM = 1.0/(HWET*FRACT*PIE*(RADIUS/12.0)**2)
      Q = (TSATCP-TE)/(RFILM+RWALL)
      TWALL = TE+Q*RWALL
      IF(IPRINT .EQ. 1) WRITE(6,130) HWET
100   CONTINUE
C
C     NOW CALCULATE THE HEAT TRANSFER THROUGH THE SEALING
C     FACE OF THE BEND. IT WILL BE TREATED AS A
C     FIN WITH ONE FACE INSULATED.
C
      XKPAR = (KCOAT*TCOAT+KTURN*PTHICK)/THICK
      QSEAL = SQRT(HDRY*(PERIM/12.0)**2*XKPAR*(THICK/12.0))*((TCP-
     1       TWALL)*TANH((LENG/12.0)*SQRT(HDRY/(XKPAR*THICK/
     2       12.0))))
      QHILO = Q + QSEAL
      ERR = ABS(QHILO-QLEAK)/QHILO
      IF(ERR .LE. 0.001) GO TO 1166
      QLEAK = QHILO
      NLEAK = NLEAK + 1
      GO TO 1111
1166  CONTINUE
      DELT = TCP-TAMB
      IF(DELT .LT. 0.0)  DELT = 0.0
      HLAM = 0.27*(DELT*12.0/DTO)**0.25
      HTURB = 0.12*DELT**0.3333333
      HUSE = HLAM
      IF(HTURB .GT. HLAM)  HUSE = HTURB
      QNAT = HUSE*PIE*(DTO/12.0)*(XLVAL/12.0)*DELT
      QTOTAL = Q+QSEAL+QNAT
```

```
      QAMB = QNAT
      HHIOUT = HHIIN-QTOTAL/MDOT
      IF(IPRINT .EQ. 1) WRITE(6,140) HDRY,Q,QSEAL,QNAT
C
C     THE FOLLOWING PRESSURE DROP CALCULATIONS ARE FOR THE
C     HIGH PRESSURE SIDE
C
      VCP = MDOT/(3600.0*RHOVHI*PIE*DTC**2/(4.0*144.0))
      VC = VCP*(DTC/DTI)**2
      DPRISE = (0.5*RHOVHI/32.2)*(VCP**2-VC**2)/144.0
     3      (12.0*VIS/3600.0))**(-0.25))
      PC = PCP+DPRISE-DPLOSS
      DPHI = DPLOSS-DPRISE
      PDHI = DPHI
      TTRY = TCP - 10.0
      NLOC = 2
      CALL TRIAL2(TTRY,5.0,PC,3,HHIOUT,0.001,VX,HX,SX,TC,IGOOF)
      IF(IPRINT .EQ. 1)  WRITE(6,150) XA,VCP,VC,DPRISE,DPLOSS,PC,TC
C
C     NEXT SET OF PRESSURE DROP CALCULATIONS ARE FOR THE
C     TWO PHASE LOW PRESSURE SIDE
C
      X = (XA+XE)/2.0
      IF(X .GT. 1.0)  X = 1.0
      IF(X .LE. 0.0)  X = 1.0E-13
      RX = (1.0-X)/X
      IF(RX .LT. 0.0)  RX = 0.0
      IF(RX .GT. 1.0E8)  RX = 1.0E8
      RRHO = RHOVLO/RHOFLO
      RMU = MUFLO/MUVLO
      RD = RADIUS/DTI
      IF(JV .EQ. 2)  RD = RD*1.2
C     NOTE THAT THE R/D RATIO WAS ADJUSTED TO FORCE THE MODEL
C     TO FIT EXPERIMENTAL DATA
```

```
C
      C = (1.0+(20.0*(DTI/12.0)*RD**0.6349/C3)*((RHOFLO-RHOVLO)/
     1       RHOFLO))*(SQRT(1.0/RRHO)+SQRT(RRHO))
      GE =
     1    MDOT*4.0/(PIE*(DTI/12.0)**2)
      BETA = (0.5*(GE/3600.0)**2*C3*RD**(-0.6349)*0.316*(GE*DTI/
     1       (12.0*MUFLO))**(-0.25))/(32.2*144.0*RHOFLO)
      DPTP1 = BETA*(1.0-X)**(7.0/4.0)+BETA*X**(7.0/4.0)*(C*
     1       RX**0.875/(SQRT(RRHO)*RMU**0.125)+1.0/(RRHO*
     2       RMU**0.25))
      IF(IPRINT .EQ. 1) WRITE(6,170) DPTP1,BETA,X,C,RX,RRHO,RMU
      BETA = BETA/(C3*RD**(-0.6349))
      C = 20.0
      DPTUBE = BETA*(1.0-X)**(7.0/4.0)+BETA*X**(7.0/4.0)*(C*
     1       RX**0.875/(SQRT(RRHO)*RMU**0.125)+1.0/(RRHO*
     2       RMU**0.25))*(XLEVC+XLACC)/DTI
      IF(IPRINT .EQ. 1) WRITE(6,180) DPTUBE,BETA,C,XLEVC,XLACC,DTI
      PE = PA+DPTUBE+DPTP1
      DPLOW = PE-PA
      ERRLO = ABS(DPLOW-PDLO)/DPLOW
      IF(ERRLO .LE. 0.001) GO TO 1144
      PDLO = DPLOW
      NPDLO = NPDLO + 1
      GO TO 1111
 1144 CONTINUE
      PDLO = DPLOW
      PDHI = DPHI
      QLEAK = Q+QSEAL
      QAMB = QNAT
      IF(IPRINT .EQ. 1)  WRITE(6,160) DPHI,DPLOW,PCP,TCP,PC,TC,PE,
     1       TE,XE,PA,TA,XA
      GO TO 1998
 1991 CONTINUE
      WRITE(6,195)
```

```
      GO TO 1998
1992  CONTINUE
      WRITE(6,190)
1998  CONTINUE
      RETURN
2000  CONTINUE
      PDHI = 0.0
      QAMB = 0.0
      PDLO = REVVPD(TA,PA,MDOT)
      QLEAK = REVVHT(TCP,TA)
      RETURN
C
C
130   FORMAT(1X,29HCONDENSATION WILL OCCUR-HWET=,F10.3)
140   FORMAT(1X,5HHDRY=,E10.3,3X,2HQ=,E10.3,3X,6HQSEAL=,E10.3,3X,
     1      5HQNAT=,E10.3)
150   FORMAT(1X,3HXA=,E10.3,3X,4HVCP=,E10.3,3X,3HVC=,E10.3,3X,7HDPRISE=,
     1      E10.3,/,1X,7HDPLOSS=,E10.3,3X,3HPC=,E10.3,3X,3HTC=,E10.3)
160   FORMAT(1X,5HDPHI=,E10.3,3X,6HDPLOW=,E10.3,3X,4HPCP=,E10.3,3X,
     1   4HTCP=,E10.3,/,1X,3HPC=,E10.3,3X,3HTC=,E10.3,3X,3HPE=,E10.3,
     2 3X,3HTE=,E10.3,3X,3HXE=,E10.3,/,1X,3HPA=,E10.3,3X,3HTA=,E10.3,
     3      3X,3HXA=,E10.3)
170   FORMAT(1X,'DPTP1=',E10.3,' BETA=',E10.3,' X=',E10.3,' C=',E10.3,
     1      ' RX=',E10.3,/,1X,'RRHO=',E10.3,' RMU=',E10.3,/)
180   FORMAT(1X,'DPTUBE=',E10.3,' BETA=',E10.3,' C=',E10.3,' XLEVC=',
     1      E10.3,' XLACC=',E10.3,/,1X,'DTI=',E10.3,/)
190   FORMAT(1X,'HEAT LEAK ITERATION DOES NOT CONVERGE IN VALVER',/)
195   FORMAT(1X,'LOW SIDE PRESSURE DROP ITERATION DOES NOT',
     1      ' CONVERGE IN VALVER',/)
196   FORMAT(1X,'IV=',4X,I6,' PA=',E10.3,' TA=',E10.4,' XA=',E10.3,
     1      /,1X,'PCP=',E10.3,' TCP=',E10.3,' MDOT=',E10.3,
     2      ' TAMB=',E10.3,/)
197   FORMAT(1X,'TE=',E10.3,' PE=',E10.3,' XE=',E10.3,' TSATE=',E10.4,
     1      /,1X,'HLOIN=',E10.3,' HFLO=',E10.3,' HFGLO=',E10.3,
```

```
  2       ' RHOFLO=',E10.3,/,1X,'RHOVLO=',E10.3,' MUFLO=',E10.3,
  3       ' MUVLO=',E10.3,/)
198   FORMAT(1X,'TA=',E10.3,' PA=',E10.3,' XA=',E10.3,' TSATA=',E10.3,
  1       /,1X,'HLLOUT=',E10.3,' HLIQA=',E10.3,' HFGRA=',E10.3,/)
199   FORMAT(1X,'TCP=',E10.3,' PCP=',E10.3,' TSATCP=',E10.3,' HHIIN=',
  1       E10.3,/,1X,'HFGHI=',E10.3,' RHOFHI=',E10.3,' RHOVHI=',
  2       E10.3,' MUFHI=',E10.3,/,1X,'MUVHI=',E10.3,' TREFR=',
  3       E10.3,' CPFHI=',E10.3,' CPVHI=',E10.3,/,1X,'KF=',
  4       E10.3,' KV=',E10.3,/)
C
C
      END
```

APPENDIX C

Listing of REVVHT subroutine

A listing of the subprogram REVVHT follows.

```
      REAL FUNCTION REVVHT(TCX,TVX)
C
C     ******************************************************************
C     CALCULATES THE HEAT TRANSFER IN A REVERSING VALVE REVVHT IN
C     BTU/HR AS A FUNCTION OF:
C           TCX = COMPRESSOR DISCHARGE TEMPERATURE - DEG F
C           TVX = EVAPORATOR EXIT TEMPERATURE      - DEG F
C     ******************************************************************
C
      TH = TCX
      TL = TVX
      REVVHT = -105.64513 + 8.7219286 * (TH-TL)
      RETURN
      END
```

Listing of REVVPD subroutine

A listing of the REVVPD subprogram follows.

```
      REAL FUNCTION REVVPD(TREF,PREF,REFM)
C
C     ******************************************************************
C     CALULATES THE REFRIGERANT PRESSURE DROP THROUGH A REVERSING
C     VALVE AND CONNECTING TUBING AND AEROQUIP FITTINGS REVVPD IN PSIA
C     AS A FUNCTION OF:
C           TREF = REFRIGERANT INLET TEMPERATURE - DEG F
C           PREF = REFRIGERANT INLET PRESSURE    - PSIA
C           REFM = REFRIGERANT MASS FLOW RATE    - LBM/HR
C
C     BASED ON COPELAND TWIN BREADBOARD TEST DATA
C
C     ******************************************************************
C
      COMMON / MPASS / CNDCON, AMBCON, EVPCON, CONMST, CMPCON, FLOCON,
     1   TOLS, TOLH, LPRINT, NCORH, MCMPOP, MFANIN,
     2   MFANOU, MFANFT
      COMMON / LINES / DLL, XLEQLL, DSL, XLEQSL, DDL, XLEQDL, DSLRV,
     1   XLEQLP, DDLRV, XLEQHP, DPDL, DPSL, DPLL,
     2   QDISLN, QSUCLN, QLIQLN, E
      IF(NCORH .EQ. 1) GO TO 50
      NN = 1
      IF(XLEQSL .GT. XLEQDL) NN = 0
      GO TO 60
50    CONTINUE
      NN = 0
      IF(XLEQDL .GT. XLEQSL) NN = 1
60    CONTINUE
      TT = TREF
```

```
      PP = PREF
      RMF = REFM
      TVAP = TT
      TRSAT = TSAT(PP,IFLAG)
      IF(TVAP .LT. TRSAT) TVAP = TRSAT
      RHOV = 1.0/SPVOL(TVAP,PP)
      RMS = RMF/3600.0
      RMSQ = RMS*RMS/RHOV
      IF(NN .EQ. 0) GO TO 100
      REVVPD = 5.9160745 * (RMSQ**0.0763784)
      GO TO 200
100   CONTINUE
      REVVPD = 16.997202 * (RMSQ**0.16876337)
200   CONTINUE
      RETURN
      END
```

APPENDIX D

Reversing Valve Leakage Model

The reversing valve leakage subprogram - FLKRV - is written in a form to support a large variety of models without the need to change any other subprogram in the heat pump model, other than BLOCK DATA RLADP in which the dimension of RVLKP is declared and the values of NLKTB are specified. Thus, all inputs are received via a formal parameter list, and the parameters chosen were felt to be a sufficiently comprehensive list to permit any reasonable model to be accomodated.

In order to avoid the inflexibility of passing parameters by COMMON and yet not have the overhead associated with the formal parameters of a subprogram call, the use of the formal parameters was minimized by equating the formal parameters to local parameters immediately on entering the program. The character of FORTRAN object code is such that this is equivalent to a "call by value", thereby achieving considerable economy in both code size and execution time.

Listing of FLKRV

A listing of FLKRV follows.

```
      FUNCTION FLKRV ( NRVLD, NLPMD, RVLKP,
     1       RFLHD, PRFHD, TRFHD, HRFHD, PRFLD, TRFLD, HRFLD )
C
C       ##################################################
C
C       A FUNCTION WHOSE VALUE IS THE LEAKAGE OF REFRIGERENT FROM
C       THE HIGH SIDE TO THE LOW SIDE OF THE REVERSING VALVE
C       IN LBM/SEC.
C       INPUT PARAMETERS ARE AS FOLLOWS:
C
C             NRVLK, NRVLD - THE NUMERICAL DESIGNATOR OF THE
```

```
C                 LEAKAGE MODEL TO BE USED.  AT PRESENT, POSSIBLE
C                 VALUES THEIR CORRESPONDING MODELS ARE
C                   0 - NO LEAKAGE;
C                   1 - LEAKAGE A FIXED PERCENTAGE OF THE MASS
C                       THROUGHPUT;
C           NLPMX, NLPMD - THE DIMENSION OF THE DATA ARRAY RVLKP;
C           RVLKP(NLPMD) - THE ARRAY OF PARAMETERS TO IMPLEMENT THE
C                   CHOSEN MODEL.  PRESENT PARAMETERS CORRESPONDING TO
C                   NRVLK ARE:
C                   NRVLK = 0; RVLKP HAS NO MEANING;
C                   NRVLK = 1; RVLKP(1) IS THE PERCENTAGE LEAKAGE FLOW;
C           RFLHI, RFLHD - REFRIGERENT MASS FLOW RATE ENTERING THE HIGH
C                 PRESSURE SIDE OF THE REVERSING VALVE (LBM/SEC)
C           PRFHI, PRFHD - REFRIGERENT PRESSURE ENTERING THE HIGH SIDE
C                 OF THE REVERSING VALVE (PSIA);
C           TRFHI, TRFHD - REFRIGERENT TEMPERATURE ENTERING THE HIGH
C                 SIDE OF THE REVERSING VALVE (DEG. F);
C           HRFHI, HRFHD - REFRIGERENT ENTHALPY ENTERING THE HIGH SIDE
C                 OF THE REVERSING VALVE (BTU/LBM);
C           PRFLO, PRFLD - REFRIGERENT PRESSURE EXITING THE LOW SIDE OF
C                 OF THE REVERSING VALVE (PSIA);
C           TRFLO, TRFLD - REFRIGERENT TEMPERATURE EXITING THE LOW
C                 OF THE REVERSING VALVE (DEG. F);
C           HRFLO, HRFLD - REFRIGERENT ENTHALPY EXITING THE LOW SIDE OF
C                 OF THE REVERSING VALVE (BTU/LBM).
C
C     ##################################################
C
      DIMENSION RVLKP(NLPMD)
C
C     ##################################################
C
      DATA NOTPR / 6 /
C
```

```
C       ##################################################
C
1001  CONTINUE
C        INITIALIZE VARIABLES
      NRVLK = NRVLD
      NLPMX = NLPMD
      RFLHI = RFLHD
      PRFHI = PRFHD
      TRFHI = TRFHD
      HRFHI = HRFHD
      PRFLO = PRFLD
      TRFLO = TRFLD
      HRFLO = HRFLD
      RVLO1 = RVLKP(1)
C
      FLKRD = 0.0
C
      GO TO (1011,1021) (NRVLD+1)
C AA
      WRITE (NOTPR,*) ' FLKRV:AA: INVALID NRVLD = ', NRVLD
      GO TO 1011
C
1011  CONTINUE
C        NO REVERSING VALVE LEAKAGE
      FLKRD = 0.0
      GO TO 8990
C
1021  CONTINUE
C        REVERSING VALVE LEAKAGE A FIXED PERCENTAGE OF REFRIGERENT
C        FLOW, REGARDLESS OF PRESSURE DROP.
      FLKRD = RVLO1 * RFLHI
      GO TO 8990
C
C       ##################################################
```

```
C
8990  CONTINUE
      FLKRV = FLKRD
      RETURN
C
C       ##################################################
C
      END
```

APPENDIX E

Suction Line Accumulator Model

The design of DHSUL followed the philosophy of FLKRV, in that a major objective was to eliminate the need to modify any subprograms other than DHSUL itself, and the supporting BLOCK DATA program RLADP if changes of the model were made. Again, the approach was the same - that is, to provide for a complete formal parameter list, minimize the use of COMMON, but avoid the overhead of using formal parameters by using local variables which were equated to the formal parameters immediately upon entering the program.

Listing of DHSUL

A listing of DHSUL follows.

```
      SUBROUTINE DHSUL ( NDADM, ACPAR, NACDM, NCORD,
     1      PLODM, TLODM, XLODM, XLIDM, PHIDM, THIDM, XHIDM, RFLAD,
     2      DHLOD, DHHID, DPLOD, DPHID )
C
C      ##################################################
C
C     A SUBPROGRAM WHICH CALCULATES THE ENTHALPY CHANGE AND
C     PRESSURE DROP ACROSS A SUCTION LINE ACCUMULATOR.
C      SULDR IS A MAIN PROGRAM WHICH CAN BE USED TO EXERCISE THIS
C      PROGRAM.
C
C     INPUT DATA ARE AS FOLLOWS:
C        NDADM, NDACL - THE IDENTIFIER OF THE TYPE OF ACCUMULATOR;
C        ACPAR(NACDM) - AN ARRAY CONTAINING GEOMETRIC OR OTHER FIXED
C                       DATA;
C        NACDM, NACPR - THE DIMENSION OF ACPAR (DECLARED IN THE
```

```
C                         CALLING PROGRAM);
C          NCORD, NCORH - THE INDICATOR FOR HEATING MODE (=2) OR
C                         COOLING MODE (=1);
C          PLODM, PLOAC - THE LOW (SUCTION) SIDE OUTLET PRESSURE (PSIA);
C          TLODM, TLOAC -  "   "     "     "     "   TEMPERATURE
C                          (DEG F.)
C          XLODM, XLOAC -  "   "     "     "     "   QUALITY;
C          XLIDM, XLIAC -  "   "     "     "   INLET  QUALITY;
C          PHIDM, PHIAC - THE HIGH (DISCHARGE) SIDE INLET PRESSURE
C                         (PSIA);
C          THIDM, THIAC -  "   "      "      "     "   TEMPERATURE
C                         (DEG F);
C          XHIDM, XHIAC -  "   "      "      "     "   QUALITY;
C          RFLAD, RFLAC - THE REFRIGERANT MASS FLOW RATE (LBM/SEC).
C
C       OUTPUT DATA ARE AS FOLLOWS:
C          DHLOD, DHLOP - THE ENTHALPY CHANGE PER UNIT MASS ON THE LOW
C                          SIDE (BTU/LBM);
C          DHHID, DHHIP -  "    "     "    "  "    "  "   "  HIGH
C                          SIDE (BTU/LBM);
C          DPLOD, DPLOP - THE PRESSURE CHANGE ON THE LOW SIDE (PSIA);
C          DPHID, DPHIP -  "    "     "    "   "  HIGH SIDE (PSIA).
C
C       OPTIONS WITH RESPECT TO THE CHOSEN INPUT DATA
C       ARE GIVEN BY THE PARAMETER NDACL.
C       UNITS OF MEASURE ARE FEET, POUNDS-MASS, AND SECONDS, EXCEPT
C       FOR PRESSURE WHICH IS LEFT IN PSI.
C       THE GEOMETRIC OR OTHER FIXED INPUT DATA
C       DESCRIBING THE ACCUMULATOR ARE PLACED IN A
C       VECTOR ACPAR, WHOSE CONTENTS ARE VARIOUSLY INTERPRETED DEPENDING
C       ON THE TYPE OF ACCUMULATOR CHOSEN.  THE PRESENT
C       POSSIBILITIES (5/1/85) ARE:
C             0 - NO ACCUMULATOR
C             1 - A VERY SIMPLIFIED HI-RE-LI ACCUMULATOR,
```

```
C                WHICH IS ACTIVE ONLY DURING HEATING MODE,
C                AND HAS OTHER SIMPLIFICATIONS AS WELL.
C                IN THIS CASE, THE INPUT PARAMETERS ARE:
C                ACPAR(1) - INSIDE DIAMETER OF CONDENSATE TUBING IN
C                FEET (READ AS INCHES, BUT IMMEDIATELY CONVERTED
C                TO FEET);
C                ACPAR(2) - OUTSIDE DIAMETER OF CONDENSATE TUBING IN
C                FEET (READ AS INCHES BUT IMMEDIATELY CONVERTED TO
C                FEET);
C                ACPAR(3) - LENGTH OF CONDENSATE TUBING IN FEET;
C                ACPAR(4) - I.D. OF VAPOR LINE IN FEET (READ AS
C                INCHES, ETC.;
C                ACPAR(5) - LENGTH OF VAPOR LINE IN FEET (READ AS
C                INCHES, ETC;
C               NOTE THAT THE VAPOR LINE IS ASSUMED TO HAVE ONE
C                BREAK, IN WHICH TWO VELOCITY HEADS ARE LOST.
C           2 - A GENERIC ACCUMULATOR, DEFINED ONLY IN TERMS OF
C               SPECIFIED PRESSURE DROPS, SUPERHEATS, AND SHELL HEAT
C               TRANSFERS.  VALUES OF ACPAR ARE
C                ACPAR(1) - SUPERHEAT OF REFRIGERENT LEAVING THE
C                ACCUMULATOR (DEG. F);
C                ACPAR(2) - PRESSURE DROP OF VAPOR ON LOW (SUCTION)
C                SIDE OF ACCUMULATOR (PSIA);
C                ACPAR(3) - PRESSURE DROP OF LIQUID ON HIGH (DISCHARGE)
C                SIDE OF ACCUMULATOR (PSIA);
C                ACPAR(4) - RATE OF HEAT GAIN/LOSS FROM THE OUTER
C                SHELL OF THE ACCUMULATOR - ASSUMED TO FLOW TO/FROM
C                THE VAPOR (BTU/SEC); POSITIVE VALUES
C                ARE HEAT FLOW FROM THE ACCUMULATOR TO THE ENVIROMENT,
C                AND ARE ASSUMED TO DECREASE THE ENTHALPY OF THE VAPOR
C                ON THE LOW PRESSURES SIDE;
C
C     ####################################################
C
```

```
      DIMENSION ACPAR(NACDM)
C
C      ##################################################
C
      DIMENSION RLINP(10), ININP(6), FLINP(10), FUINP(10)
C          RLINP - AN ARRAY EQUIVALENCED TO THE VARIOUS LOCALS OF
C                  FLOATING INPUT VARIABLES;
C          FLINP - THE CORRESPONDING LOWER LIMITS;
C          FUINP - THE CORRESPONDING UPPER LIMITS;
C          ININP - AN ARRAY OF INTEGER INPUTS (NOT PRESENTLY USED).
C
C           ## NOW CREATE AN EQUIVALENCE BETWEEN THE MNEMONIC NAMES OF
C              THE INPUT VARIABLES AND THE ARRAY (NOTE THAT THIS COSTS
C              NO EXTRA MEMORY IN THE OBJECT PROGRAM).
      EQUIVALENCE ( RLINP(1),PLOAC ), ( RLINP(2),TLOAC ),
     1      ( RLINP(3),XLOAC ), ( RLINP(4),XLIAC ),
     2      ( RLINP(5),PHIAC ), ( RLINP(6),THIAC ),
     3      ( RLINP(7),XHIAC ), ( RLINP(8),RFLAC )
C
C     BYTE CHFPN (6,10)
      CHARACTER*1 CHFPN (6,10)
C          CHFPN - AN ARRAY OF VARIABLE NAMES, TO FLAG DATA WHEN
C                  AN INPUT VARIABLE IS OUT OF RANGE.
C                  THE SECOND SUBSCRIPT OF CHFPN CORRESPONDS TO THE
C                  SUBSCRIPT OF THE CORRESPONDING ELEMENT OF RLINP,
C                  FLINP, AND FUINP.
C
C      ## FOLLOWING ADDED ON 8/5/85 BY R. LUCHETA AT THE
C         WESTINGHOUSE RESEARCH LABORATORY.
C
      LOGICAL PRINT
      COMMON / A1 / PRINT
C
C      ## END OF 8/5/85  ADDITION
```

```
C
C
C     DATA (FLINP(INDEX), INDEX = 1, 10, 1)  /
      DATA  FLINP /
     1      10.0, -40.0, 0.0, 0.0, 10.0, -40.0, 0.0, 0.05, 2*0.0 /
C     DATA  FUINP(INDEX), INDEX = 1, 10, 1  /
      DATA  FUINP /
     1      400.0, 300.0, 1.0, 1.0, 400.0, 300.0, 1.0, 1.0E+06, 2*0.0 /
C
C     DATA ( CHFPN(INDEX,1), INDEX = 1, 6, 1 ) /
C    1      'P', 'L', 'O', 'A', 'C', ' ' /,
C    2     ( CHFPN(INDEX,2), INDEX = 1, 6, 1 ) /
C    3      'T', 'L', 'O', 'A', 'C', ' ' /,
C    4     ( CHFPN(INDEX,3), INDEX = 1, 6, 1 ) /
C    5      'X', 'L', 'O', 'A', 'C', ' ' /,
C    6     ( CHFPN(INDEX,4), INDEX = 1, 6, 1 ) /
C    7      'X', 'L', 'I', 'A', 'C', ' ' /,
C    8     ( CHFPN(INDEX,5), INDEX = 1, 6, 1 ) /
C    9      'P', 'H', 'I', 'A', 'C', ' ' /,
C    +     ( CHFPN(INDEX,6), INDEX = 1, 6, 1 ) /
C    1      'T', 'H', 'I', 'A', 'C', ' ' /,
C    2     ( (CHFPN(INDEX,JNDEX), INDEX = 1, 6, 1), JNDEX = 7, 10, 1 ) /
C    3      6*' ', 6*' ', 6*' ', 6*' ' /
C
      DATA   CHFPN /
     1      'P', 'L', 'O', 'A', 'C', ' ' ,
     2      'T', 'L', 'O', 'A', 'C', ' ' ,
     3      'X', 'L', 'O', 'A', 'C', ' ' ,
     4      'X', 'L', 'I', 'A', 'C', ' ' ,
     5      'P', 'H', 'I', 'A', 'C', ' ' ,
     6      'T', 'H', 'I', 'A', 'C', ' ' ,
     7      6*' ', 6*' ', 6*' ', 6*' ' /
C
C         THE PRINT FILE NOTPR
```

```
      DATA NOTPR / 6 /
C
C         THE VALUE OF PI & ACCELERATION OF GRAVITY
      DATA PIVAL, GEEVL / 3.141592653589793, 32.16 /
C
C         AN ASSUMED OVERALL HEAT TRANSFER COEFFICIENT AND EQUIVALENT
C         ROUGHNESS
      DATA HTBEC, EQROU / 1.0, 0.0002 /
C
C            THE CONVERGANCE CRITERION FOR PRESSURE DROP CONVERGANCE,
C            AND THE NUMBER OF LOOPS
      DATA CNCRI, NLOPM / 0.001, 100 /
C
      DATA NFLIN / 8 /
C            ## NFLIN - THE NUMBER OF FLOATING INPUTS TO THE
C               SUBPROGRAM.
C
9001  FORMAT ( ' DHSUL:BA:INVALID INPUT: ', 5A1, ' = ',
     1       1X, 1PG12.4 )
C
C      ##################################################
C
1001  CONTINUE
C       CONVERT FORMAL PARAMETERS TO LOCAL PARAMETERS
      NDACL = NDADM
      NACPR = NACDM
      NCORH = NCORD
      PLOAC = PLODM
      TLOAC = TLODM
      XLOAC = XLODM
      XLIAC = XLIDM
      PHIAC = PHIDM
      THIAC = THIDM
      XHIAC = XHIDM
```

```
      RFLAC = RFLAD
C             AS MANY OF ACPAR() AS ARE USED
      ACPR1 = ACPAR(1)
      ACPR2 = ACPAR(2)
      ACPR3 = ACPAR(3)
      ACPR4 = ACPAR(4)
      ACPR5 = ACPAR(5)
      ACPR6 = ACPAR(6)
C
C             ## NOW SCREEN AND FILTER INPUT DATA
      DO 1011 INDEX = 1, NFLIN, 1
C BA
      IF (         PRINT
     1             .AND.
     2    ( (RLINP(INDEX) .LT. FLINP(INDEX) )
     3             .OR.
     4      (RLINP(INDEX) .GT. FUINP(INDEX)) )   )
     5      WRITE (NOTPR,9001)
     5      ( CHFPN(JNDEX,INDEX), JNDEX = 1, 5, 1 ), RLINP(INDEX)
      RLINP(INDEX) =
     1      AMAX1 ( FLINP(INDEX), AMIN1 ( FUINP(INDEX), RLINP(INDEX) ) )
1011  CONTINUE

C AA
C       WRITE (NOTPR,*) ' DHSUL:AA: INPUT FOLLOWS '
C       WRITE (NOTPR,*) ' NDACL, NACPR, NCORH = '
C       WRITE (NOTPR,*)   NDACL, NACPR, NCORH
C       WRITE (NOTPR,*) ' PLOAC, TLOAC, XLOAC, XLIAC = '
C       WRITE (NOTPR,*)   PLOAC, TLOAC, XLOAC, XLIAC
C       WRITE (NOTPR,*) ' PHIAC, THIAC, XHIAC, RFLAC = '
C       WRITE (NOTPR,*)   PHIAC, THIAC, XHIAC, RFLAC
C       REWIND NOTPR
C           INITIALIZE OUTPUTS
      DHLAC = 0.0
```

```
      DHHAC = 0.0
      DPLAC = 0.0
      DPHAC = 0.0
      DHLOD = 0.0
      DHHID = 0.0
      DPLOD = 0.0
      DPHID = 0.0
C
C         BRANCH TO APPROPRIATE TYPE OF HEAT EXCHANGER
      GO TO ( 1101, 1201, 1301 ) (NDACL+1)
C AA
      WRITE (NOTPR,*) ' DHSUL:AA: INVALID NDACL = ', NDACL
      GO TO 8990
C
C       %%%%%%%%%%%%%%%%%%%%
C
1101  CONTINUE
C        NO ACCUMULATOR AT ALL - DO NOTHING
      GO TO 8990
C
C       %%%%%%%%%%%%%%%%%%%%
C
1201  CONTINUE
C         SIMPLIFIED HI-RE-LI; ASSUMED TO BE ACTIVE ONLY DURING
C         HEATING, WITH SIMPLE HEAT ABSORPTION AND PRESSURE DROP
C         LAWS.
C         DELTH - ENTHALPY CHANGE
      DELTH = 0.0
C         DPLAC - PRESSURE CHANGE ON LOW PRESSURE SIDE (NOMINALLY PSI)
      DPLAC = 0.0
      DPACO = 0.0
C         SCREEN INPUT QUALITY VALUES
      IF ( (XLIAC .LE. 1.0) .AND. (XLIAC .GE. 0.0) ) GO TO 1211
C AC
```

```
      WRITE (NOTPR,*) ' DHSUL:AC: STRANGE XOE = ', XLIAC
      IF ( XLIAC .GE. 1.0 )  GO TO 1251
      IF ( XLIAC .LE. 0.0 )  GO TO 1256
      GO TO 8990
C
1251  CONTINUE
C         QUALITY GIVEN GREATER THAN 1.0
      XLIAC = 1.0
      GO TO 1211
C
1256  CONTINUE
C         QUALITY GIVEN AS NEGATIVE
      XLIAC = 0.0
      GO TO 1211
C
1211  CONTINUE
C         NOW CONTINUE WITH COMPUTATIONS; FIRST GET CONDENSATE
C         AREAS AND CONDUCTANCES.
C         OARTB - OUTSIDE AREA OF CONDENSATE TUBE - SQ. FT.
      OARTB = PIVAL * ACPR2 * ACPR3
C         UAOTB - OVEARALL UA FOR THE CONDENSATE TUBE - BTU/(SEC-DEG.F)
      UAOTB = OARTB * HTBEC / 3600.0
C         SCREEN RFLAC - MASS FLOW RATE
      IF ( RFLAC .GT. 0.0 )  GO TO 1261
C AD
      WRITE (NOTPR,*) ' CLUSA:AD: STRANGE RFLAC = ', RFLAC
1261  CONTINUE
C         GET AN ABSOLUTE FLOW RATE
      FLREF = ABS(RFLAC)
C
C         WE NOW GO TO AN ITERATION LOOP, IN WHICH THE ENTHALPY CHANGE
C         AND PRESSURE DROP IN THE ACCUMULATOR ARE SUCCESSIVELY
C         COMPUTED.  WE NOTE THAT THE PRESSURE DROP CHANGES THE
C         ASSOCIATED ENTHALPIES, AND THE CHANGE OF TEMPERATURE
```

```
C         AND PRESSURE CHANGES THE DENSITIES, THE DYNAMIC PRESSURE,
C         AND SO THE PRESSURE DROP.
C
C         %% 1221 IS THE POINT TO WHICH THE ITERATION LOOP RETURNS.
      NLOOP = 0
1221  CONTINUE
      DPACO = DPLAC
      PLIAC = PLOAC + DPLAC
      PHOAC = PHIAC - DPHAC
C
C         FIRST GET TRANSPORT PROPERTIES - NOT VERY CRITICAL OR
C         SENSITIVE.
C            EVAPORATOR PARAMETERS FOLLOW
      CALL MUKCP ( TLOAC, VSLEV, VSVEV,VSGEV, TCLEV, TCVEV,
     1      TCGEV, CPLEV, CPVEV, CPGEV, IEMUK )
C            LIQUID VISCOSITY
      VSLEV = VSLEV / 3600.0
C            SATURATED VAPOR VISCOSITY
      VSVEV = VSVEV / 3600.0
C            1 ATMOSPHERE GAS VISCOSITY
      VSGEV = VSGEV / 3600.0
C            LIQUID THERMAL CONDUCTIVITY
      TCLEV = TCLEV / 3600.0
C            SATURATED VAPOR THERMAL CONDUCTIVITY
      TCVEV = TCVEV / 3600.0
C            1 ATM GAS THERMAL CONDUCTIVITY
      TCGEV = TCGEV / 3600.0
C
C            CONDENSOR CALCULATIONS FOLLOW
      CALL MUKCP ( THIAC, VSLCN, VSVCN, VSGCN, TCLCN, TCVCN,
     1      TCGCN, CPLCN, CPVCN, CPGCN, ICMUK )
C            LIQUID VISCOSITY
      VSLCN = VSLCN / 3600.0
C            SATURATED VAPOR VISCOSITY
```

```
      VSVCN = VSVCN / 3600.0
C              1 ATM GAS VISCOSITY
      VSGCN = VSGCN / 3600.0
C              THERMAL CONDUCTIVITY - LIQUID
      TCLCN = TCLCN / 3600.0
C                 "          "         - SATURATED VAPOR
      TCVCN = TCVCN / 3600.0
C              THERMAL CONDUCTIVITY - 1 ATM. GAS
      TCGCN = TCGCN / 3600.0
C
C              SATURATION THERMODYNAMIC PROPERTIES FOLLOW - MORE
C              CRITICALLY SENSITIVE TO PRESSURE.
C              LOW PRESSURE SIDE - INLET
C AC
C     WRITE (NOTPR,*) ' DHSUL:AC: PLIAC, TSAT(PLIAC) = ',
C    1       PLIAC, TSAT(PLIAC,IEROR)
      TSLIA = TSAT(PLIAC,IEROR)
      CALL SATPRP ( TSLIA, PDUMY, VSLLI, VSGLI, HSLLI, HFGLI,
     1       HSGLI, SSLLI, SSGLI, ISTLI )
C              LOW PRESSURE SIDE - OUTLET
C AD
C     WRITE (NOTPR,*) ' DHSUL:AD: PLOAC, TSAT(PLOAC) = ',
C    1       PLOAC, TSAT(PLOAC,IEROR)
      TSLOA = TSAT(PLOAC,IEROR)
      CALL SATPRP ( TSLOA, PDUMY, VSLLO, VSGLO, HSLLO, HFGLO,
     1       HSGLO, SSLLO, SSGLO, ISTLO )
C              HIGH PRESSURE SIDE - INLET
      TSHIA = TSAT(PHIAC,IEROR)
      CALL SATPRP ( TSHIA, PDUMY, VSLHI, VSGHI, HSLHI, HFGHI,
     1       HSGHI, SSLHI, SSGHI, ISTHI )
C              HIGH PRESSURE SIDE - OUTLET
      TSHOA = TSAT(PHOAC,IEROR)
      CALL SATPRP ( TSHOA, PDUMY, VSLHO, VSGHO, HSLHO, HFGHO,
     1       HSGHO, SSLHO, SSGHO, ISTHO )
```

```
C
C            NOW GET ENTHALPY CHANGE TO EVAPORATE ALL LIQUID
      DHLIO = 1.0*HSGLO - XLIAC*HSGLI
C            BUT NOT IF COOLING DOWN
      IF ( DHLIO .LE. 0.0 ) DHLIO = 0.0
C AM
C     WRITE (NOTPR,*) ' DHSUL:AM: DHLIO, HSGLO, XLIAC, HSGLI, ',
C    1      'PLOAC, PLIAC = '
C     WRITE (NOTPR,*) '   ', DHLIO, HSGLO, XLIAC, HSGLI, PLOAC, PLIAC
C
      CPSLA = 0.5 * (CPLCN+CPVEV)
      QCOND = -(TLOAC-THIAC) /
     1      ( 1.0/(UAOTB) + 1.0/(CPSLA*FLREF) )
      QACCU = QCOND + DHLIO*FLREF
      DELTH = QACCU / FLREF
C
1270  CONTINUE
C      ## NOW (TEMPORARILY) OUTPUT THE ENTHALPY CHANGE PARAMETERS ##
C AL
C     WRITE (NOTPR,*) ' DHSUL:AL: DELTH, QACCU, FLREF, QCOND, ',
C    1      'DHLIO, TLOAC, THIAC, CPSLA = '
C     WRITE (NOTPR,*) ' ', DELTH, QACCU, FLREF, QCOND, DHLIO,
C    1      TLOAC, THIAC, CPSLA
C      WITH DELTH COMPUTED, THE SUCTION LINE
C      CONDITIONS ARE LEFT TO THE OUTER PROGRAM.  WE NOW
C      COMPUTE THE PRESSURE DROP ON THE EVAPORATOR SIDE
C            THE INLET LEG DYNAMIC HEAD AND HEAD LOSS.
C            MIXTURE SPECIFIC VOLUME
      VMXLI = XLIAC*VSGLI + (1.0-XLIAC)*VSLLI
C            MIXTURE VOLUME FLOW RATE
      VLFLI = FLREF * VMXLI
C            MIXTURE VELOCITY
      VELLI = VLFLI*4.0 / ( PIVAL*ACPR4*ACPR4 )
C            MIXTURE DYNAMIC HEAD (PSIA)
```

```
C
C      NOW COMPUTE SUCTION LINE DELTA-P
      DPLIN = 1.8 * (0.5*VMXLI/GEEVL) * VELLI*VELLI / 144.0
C             COMPUTE THE PRESSURE DROP IN THE EXIT LEG
C             VOLUME FLOW
      VLVLO = FLREF * VSGLO
C             VELOCITY
      VELLO = VLVLO * 4.0 / ( PIVAL*ACPR4*ACPR4 )
C             MORE VELOCITY HEADS
      DPLOT = 1.8 * ( 0.5 * VSGLO / GEEVL ) * VELLO*VELLO / 144.0
C
1281  CONTINUE
C             COMPUTE L/D (INCLUDING U-BEND)
      ELLOD = ACPR5 / ACPR4 + 40.0
C             USE STANDARD PRESSURE DROP PROGRAM
C AH
C      WRITE (NOTPR,*) ' DHSUL:AH:CALLING DPLINE; '
C      WRITE (NOTPR,*) '       ACPR4, ELLOD, EQROU, FLREF*3600.0,',
C     1       '1.0/VSGLO, 3600*VSVEV = '
C      WRITE (NOTPR,*) ACPR4, ELLOD, EQROU, FLREF*3600.0,
C     1       1.0/VSGLO, 3600.0*VSVEV
      CALL DPLINE ( ACPR4, ELLOD, EQROU, FLREF*3600.0,
     1       1.0/VSGLO, 3600.0*VSVEV, DPEVR )
C AI
C      WRITE (NOTPR,*) ' DHSUL:AI: LEAVING DPLINE; DPEVR = ', DPEVR
      DPLOT = DPLOT + DPEVR
      DPLAC = DPLOT + DPLIN
C
C           NOW GET THE PRESSURE DROP ON THE CONDENSOR SIDE
      ELLOD = ACPR3 / ACPR1
C AJ
C      WRITE (NOTPR,*) ' DHSUL:AJ: CALLING DPLINE'
C      WRITE (NOTPR,*) '      ACPR1, ELLOD, EQROU, FLREF*3600.0,',
C     1       '2.0/(VSLHI+VSLHO), 3600*VSLCN = '
```

```
C      WRITE (NOTPR,*) ACPR1, ELLOD, EQROU, FLREF*3600.0,
C     1      2.0 / (VSLHI+VSLHO), 3600.0*VSLCN
      CALL DPLINE ( ACPR1, ELLOD, EQROU, FLREF*3600.0,
     1      2.0 / (VSLHI+VSLHO), 3600.0*VSLCN,
     2      DPHAC )
C AK
C      WRITE (NOTPR,*) ' DHSUL:AK: LEAVING DPLINE; DPHAC = ', DPHAC
C
      NLOOP = NLOOP+1
      IF ( NLOOP .LE. NLOPM )  GO TO 1285
C AG
      WRITE (NOTPR,*) ' DHSUL:AG: NO CONVERGANCE'
      GO TO 1286
C
1285  CONTINUE
      IF ( ABS( (DPLAC-DPACO)/(DPLAC+DPACO) ) .GE. CNCRI )
     1      GO TO 1221
C            ELSE
C
      1286  CONTINUE
      DPLAC = 0.5 * ( DPLAC + DPACO )
C            NOW BRANCH TO APPROPRIATE TREATMENT FOR HEATING (NCORH=2)
C            OR COOLING (NCORH=1).
      GO TO ( 1287, 1288 ) NCORH
C AH
      WRITE ( NOTPR, * ) ' DHSUL:AH: STRANGE NCORH = ', NCORH
      GO TO 1288
C
1287  CONTINUE
C            NCORH = 1 INDICATES COOLING MODE
      DELTH = 0.0
C
1288  CONTINUE
C            NCORH = 2 INDICATES HEATING - ALSO THE GENERIC CASE
```

```
      DHLAC = DELTH
      DHHAC = -DELTH
      GO TO 8990
C
C      %%%%%%%%%%%%%%%%%%%%%%
C
1301  CONTINUE
C          SIMPLIFIED HI-RE-LI; ASSUMED TO BE ACTIVE ONLY DURING
C          HEATING; WITH SPECIFIED SUPERHEAT, PRESSURE DROPS, AND
C          ACCUMULATOR HEAT GAIN.
C          TSUPR - SUPERHEAT LEAVING ACCUMULATOR
      TSUPR = ACPR1
C          DELTH, DHLIO - ENTHALPY CHANGE
      DELTH = 0.0
      DHLIO = 0.0
C          DPLAC - PRESSURE CHANGE ON LOW PRESSURE SIDE (NOMINALLY PSI)
      DPLAC = ACPR2
C          DPHAC - PRESSURE CHANGE ON HIGH PRESSURE SIDE
      DPHAC = ACPR3
C          QACCU - HEAT FLOW OUT OF ACCUMULATOR AND SUCTION VAPOR
      QACCU = ACPR4
      DPACD = 0.0
C          PLIAC - THE INLET PRESSURE ON THE LOW PRESSURE SIDE;
C          PHOAC - THE EXIT PRESSURE ON THE HIGH PRESSURE SIDE;
      PLIAC = PLOAC + DPLAC
      PHOAC = PHIAC - DPHAC

C          SCREEN INPUT QUALITY VALUES
      IF ( (XLIAC .LE. 1.0) .AND. (XLIAC .GE. 0.0) ) GO TO 1311
C AC
      WRITE (NOTPR,*) ' DHSUL:AC: STRANGE XOE = ', XLIAC
      IF ( XLIAC .GE. 1.0 )  GO TO 1351
      IF ( XLIAC .LE. 0.0 )  GO TO 1356
      GO TO 8990
```

```
C
1351  CONTINUE
C          QUALITY GIVEN GREATER THAN 1.0
      XLIAC = 1.0
      GO TO 1311
C
1356  CONTINUE
C          QUALITY GIVEN AS NEGATIVE
      XLIAC = 0.0
      GO TO 1311
C
1311  CONTINUE
C          SCREEN RFLAC - MASS FLOW RATE
      IF ( RFLAC .GT. 0.0 )  GO TO 1361
C AD
      WRITE (NOTPR,*) ' CLUSA:AD: STRANGE RFLAC = ', RFLAC
1361  CONTINUE
C          GET AN ABSOLUTE FLOW RATE
      FLREF = ABS(RFLAC)
C
C          WE NOW GO TO AN ITERATION LOOP, IN WHICH THE ENTHALPY CHANGE
C          IN THE ACCUMULATOR IS REPETITIVELY
C          COMPUTED, TO PERMIT CONSISTENT HEAT FLOWS AND SUPERHEATS.
C
C          %% 1321 IS THE POINT TO WHICH THE ITERATION LOOP RETURNS.
      NLOOP = 0
1321  CONTINUE
      DELTH = DHLIO
C
C          FIRST GET TRANSPORT PROPERTIES - NOT VERY CRITICAL OR C
SENSITIVE.
C              EVAPORATOR PARAMETERS FOLLOW - SPECIFIC HEATS (CPXXX) ARE
C              OF INTEREST;
      CALL MUKCP ( TLOAC, VSLEV, VSVEV,VSGEV, TCLEV, TCVEV,
```

```
     1      TCGEV, CPLEV, CPVEV, CPGEV, IEMUK )
C           LIQUID VISCOSITY
C
C           CONDENSOR CALCULATIONS FOLLOW
      CALL MUKCP ( THIAC, VSLCN, VSVCN, VSGCN, TCLCN, TCVCN,
     1      TCGCN, CPLCN, CPVCN, CPGCN, ICMUK )
C
C           SATURATION THERMODYNAMIC PROPERTIES FOLLOW - MORE
C           CRITICALLY SENSITIVE TO PRESSURE.
C           LOW PRESSURE SIDE - INLET
C AC
C     WRITE (NOTPR,*) ' DHSUL:AC: PLIAC, TSAT(PLIAC) = ',
C    1      PLIAC, TSAT(PLIAC,IEROR)
      TSLIA = TSAT(PLIAC,IEROR)
      CALL SATPRP ( TSLIA, PDUMY, VSLLI, VSGLI, HSLLI, HFGLI,
     1      HSGLI, SSLLI, SSGLI, ISTLI )
C           LOW PRESSURE SIDE - OUTLET
C AD
C     WRITE (NOTPR,*) ' DHSUL:AD: PLOAC, TSAT(PLOAC) = ',
C    1      PLOAC, TSAT(PLOAC,IEROR)
      TSLOA = TSAT(PLOAC,IEROR)
      CALL SATPRP ( TSLOA, PDUMY, VSLLO, VSGLO, HSLLO, HFGLO,
     1      HSGLO, SSLLO, SSGLO, ISTLO )
C           HIGH PRESSURE SIDE - INLET
      TSHIA = TSAT(PHIAC,IEROR)
      CALL SATPRP ( TSHIA, PDUMY, VSLHI, VSGHI, HSLHI, HFGHI,
     1      HSGHI, SSLHI, SSGHI, ISTHI )
C           HIGH PRESSURE SIDE - OUTLET
      TSHOA = TSAT(PHOAC,IEROR)
      CALL SATPRP ( TSHOA, PDUMY, VSLHO, VSGHO, HSLHO, HFGHO,
     1      HSGHO, SSLHO, SSGHO, ISTHO )
C
C           NOW GET ENTHALPY CHANGE CORRESPONDING TO EVAPORATION AND
C           TSUPR SUPERHEAT;
```

```
      DHLIO = CPVEV*TSUPR + ( 1.0*HSGLO - XLIAC*HSGLI )
C              BUT NOT IF COOLING DOWN
      IF ( DHLIO .LE. 0.0 ) DHLIO = 0.0
      DHHIO = - ( DHLIO + QACCU / FLREF )
C AM
C     WRITE (NOTPR,*) ' DHSUL:AM: DHLIO, HSGLO, XLIAC, HSGLI, ',
C    1       'PLOAC, PLIAC = '
C     WRITE (NOTPR,*) '   ', DHLIO, HSGLO, XLIAC, HSGLI, PLOAC, PLIAC
C
1370  CONTINUE
C       ## NOW (TEMPORARILY) OUTPUT THE ENTHALPY CHANGE PARAMETERS ##
C AL
C     WRITE (NOTPR,*) ' DHSUL:AL: DELTH, QACCU, FLREF, QCOND, ',
C    1       'DHLIO, TLOAC, THIAC, CPSLA = '
C     WRITE (NOTPR,*) ' ', DELTH, QACCU, FLREF, QCOND, DHLIO,
C    1       TLOAC, THIAC, CPSLA
C       WITH DELTH COMPUTED, THE SUCTION LINE
C       CONDITIONS ARE LEFT TO THE OUTER PROGRAM.
C
1381  CONTINUE
C
      DPLAC = ACPR2
      DPHAC = ACPR3
      NLOOP = NLOOP+1
      IF ( NLOOP .LE. NLOPM )  GO TO 1385
C AG
      WRITE (NOTPR,*) ' DHSUL:AG: NO CONVERGANCE'
      GO TO 1386
C
1385  CONTINUE
      IF ( ABS( (DHLIO-DELTH)/(DHLIO+DELTH) ) .GE. CNCRI )
     1       GO TO 1321
C              ELSE
C
```

```
1386  CONTINUE
      DHLIO = 0.5 * ( DHLIO + DELTH )
      DELTH = DHLIO
C             NOW BRANCH TO APPROPRIATE TREATMENT FOR HEATING (NCORH=2)
C             OR COOLING (NCORH=1).
      GO TO ( 1387, 1388 ) NCORH
C AH
      WRITE ( NOTPR, * ) ' DHSUL:AH: STRANGE NCORH = ', NCORH
      GO TO 1388
C
1387  CONTINUE
C             NCORH = 1 INDICATES COOLING MODE
      DHLAC = 0.0
      DHHAC = 0.0
      GO TO 1391
C
1388  CONTINUE
C             NCORH = 2 INDICATES HEATING - ALSO THE GENERIC CASE
      DHLAC = DELTH
      DHHAC = - ( DELTH + QACCU / FLREF )
      GO TO 1391
C
1391  CONTINUE
      GO TO 8990
C
C      ##################################################
C
8990  CONTINUE
      DPLOD = DPLAC
      DPHID = DPHAC
      DHLOD = DHLAC
      DHHID = DHHAC
      RETURN
C
```

```
C        ####################################################
C
      END
```

Listing of RLADP

A listing of RLADP follows:

```
      BLOCK DATA RLADP
C
C       ##################################################
C
C           BLOCK DATA IN WHICH CERTAIN COMMONS ADDED BY R. LUCHETA,
C           AT THE WESTINGHOUSE R&D CENTER IN MAY, 1985, ARE
C           DECLARED AND SET TO INITIAL AND DEFAULT VALUES.
C
      COMMON / ACPAR / DHHAC, DHLAC, DPHAC, DPLAC,
     1        NACCU, NACPR, ACPAR(20)
C           COMPUTED
C           DHHAC - ENTHALPY CHANGE OF FLUID ON HIGH PRESSURE SIDE;
C           DHLAC - ENTHALPY CHANGE OF FLUID ON LOW PRESSURE SIDE;
C           DPHAC - PRESSURE CHANGE OF FLUID ON HIGH PRESSURE SIDE;
C           DPLAC - PRESSURE CHANGE OF FLUID ON LOW PRESSURE SIDE;
C           INPUTS
C           NACCU - DESCRIPTOR OF TYPE OF ACCUMULATOR - READ IN;
C           NACPR - DIMENSION OF VECTOR ACPAR - NEVER CHANGED;
C           ACPAR - GEOMETRIC PARAMETERS OF ACCUMULATOR - READ IN.
C           THE PROGRAM DHSUL WILL HAVE THE MEANINGS OF THE VALUES
C           OF ACPAR FOR DIFFERENT SUCTION LINE ACCUMULATOR
C           MODELS.
C
C
      COMMON / ACTBL / NTBDM, NACTB(12)
C           A COMMON IN WHICH IS PLACED A TABLE - NACTB - OF
C           DIMENSION NTBDM (THIS IS NECESSARY TO AVOID
C           THE USE OF A PARAMETER STATEMENT, AND SO MAINTAIN
C           COMPATIBILITY WITH EARLIER VERSIONS OF FORTRAN.)
```

```
C          EACH ELEMENT OF NACTB GIVES THE NUMBER OF
C          ELEMENTS OF ACPAR WHICH ARE SIGNIFICANT FOR THE
C          CORRESPONDING VALUE-1 OF NACCU.  THUS, THE FIRST
C          ELEMENT OF NACTB CONTAINS THE NUMBER OF ELEMENTS
C          OF ACPAR WHICH ARE SIGNIFICANT WHEN NACCU = 1-1 = 0.
C          THE SECOND ELEMENT OF NACTB CONTAINS THE NUMBER OF
C          ELEMENTS OF ACPAR WHICH ARE SIGNIFICANT WHEN NACCU = 2-1 = 1.
C          (IN THIS CASE NACTB(2)=5, WHICH MEANS THAN, WHEN NACCU = 1,
C          THE FIRST 5 ELEMENTS OF ACPAR ARE SIGNIFICANT).
C          THE SIGNIFICANCE OF EACH ELEMENT OF ACPAR, GIVEN THE VALUE
C          OF NACCU, IS DESCRIBED IN SUBPROGRAM DHSUL.
C          THE VARIABLES OF ACTBL ARE NEVER CHANGED.
C
      COMMON / RVLKP / NRVLK, NRDFL, NLPMX, NLKTB(5), RVLKP(5)
C          NRVLK - A DESIGNATOR WHICH IS READ IN TO SPECIFY THE
C                  LEAKAGE MODEL TO BE USED.
C          NRDFL - THE NUMBER OF DISTINCT VALUES OF NRVLK WHICH
C                  ARE PERMITTED, WHICH WILL ALSO EQUAL THE DIMENSION
C                  OF NLKTB - NEVER CHANGED.
C          NLPMX - THE MAXIMUM NUMBER OF LEAKAGE PARAMETERS PERMITTED
C                  FOR ANY REVERSING VALVE LEAKAGE MODEL - NEVER
C                  CHANGED;
C          NLKTB(NRDFL) - AN ARRAY CONTAINING THE ACTUAL NUMBER OF
C                  REVERSING VALVE LEAKAGE PARAMETERS TO BE USED FOR
C                  THE CURRENT NRVLK.  THE SUBSCRIPT TO BE USED
C                  IS (NRVLK+1) - NEVER CHANGED;
C          RVLKP(NLKTB) - AN ARRAY OF REVERSING VALVE LEAKAGE PARAMETERS
C                   - READ IN FROM INPUT FILE.
C            THE PROGRAM FLKRV HAS THE MEANINGS OF THE
C            VALUES OF RVLKP FOR DIFFERENT LEAKAGE MODELS.
C

      DATA DHHAC, DHLAC, DPHAC, DPLAC, NACCU, NACPR, ACPAR /
     1      4*0.0, 0, 20, 20*0.0 /
```

```
C
      DATA NTBDM / 12 /, NACTB / 0, 5, 4, 9*0 /
C
      DATA NRVLK / 0 /, NRDFL, NLPMX / 5, 5 /,
     1      NLKTB / 0, 1, 3*0 /, RVLKP / 5*0.0 /
C
C      ##################################################
C
      END
```

Appendix F
FIN PATTERNATION EFFECTS

INTRODUCTION

Computer analysis involving systems with continuous fin-tube heat exchangers as one or more of the system components such as air conditioners or heat pumps require a method of predicting the air side convection coefficient of a given heat exchanger and its pressure drop. Some data has been published for flat or unpatterned fin-tube heat exchangers (2-4). Much more data exists throughout the industry but is proprietary to the source company. An even larger body of data on the overall capacity and pressure drop of various continuous fin-tube heat exchangers exists but this data, if and as published, is not in a form where the air side convection coefficient can be adequately calculated from it. Necessary data includes not only the dimensions of the tubes and fins but also a knowledge of the tube side heat transfer coefficient and the contact impedance between the fin collars and the tubes. Using high velocity water in the tubes can minimize the effect of calculated tube-side convection coefficient error but if the heat exchanger manufacture is not done properly, the fin collars may be cracked or loose and make poor thermal contact with the tube outer surface. Westinghouse once bought one of a kind coils of each of several different geometries from an outside vendor to evaluate the effect of these geometries on air side convection coefficient. After several weeks of testing it was found that the air side convection coefficients calculated from the test data were inconsistent from one heat exchanger to the next. It became obvious that the contact conductance between the fin collars and the tubes were both low and non-uniform for the several heat exchangers purchased.

Fin collar to tube conductances of as little as 500 to 1500 Btu/h-ft^2-°F have been measured but it is felt that well made production heat exchangers should have contact conductances in excess of 2000 Btu/h-ft^2-°F. Because of the difficulty of measuring contact conductance accurately many experimenters lump the contact impedance with the air side convection impedance and thus calculated air side convection coefficients are low, especially at high air face velocities.

EXPERIMENTAL MODEL

In order to evaluate the relative thermal performance associated with changes in transverse tube pitch (P_t) longitudinal tube pitch (P_ℓ), tube diameter (D_t), fin spacing (W) and pattern type and depth it was decided to conduct single fin channel experiments.

The single fin model embodies two 0.25^+ in. thick brass plates as fins which can be spaced apart to form a single channel for air to flow through. The outer surface of the plates are heated electrically and thermally insulated. The inner surfaces of the plates which form the air channel can be machined to replicate any desired pattern on the fin surfaces. In this way new fin patterns can be investigated without the necessity of expensive modification of existing fin dies. By supplying electrical energy to maintain the plates at a constant temperature while heating air passing through the single fin channel, the fin efficiency will be 100 percent. There will be no uncertainty here either as to contact impedance or inside coefficient since all heat is transferred directly through the fin normal to its surface and from the primary surface.

Care was taken to thermally guard the test sections as well as the entry and exit plenums supplying air to the test section so that good energy balances could be obtained. The entry nozzle to the single fin channel and the diffuser leaving the single fin channel were made from micarta, a paper-phenylic laminate of low conductivity, but a small amount of energy would have been transferred to the air from these entry and exit sections so that the calculated air side convection coefficients would have been slightly high, especially at low air velocities.

Figure F-1 is a picture of the two sides of a pattern fin in common use in the air conditioning industry. Figures F-2 shows mating pairs of plates with different fin patterns, tube arrangements, etc. Fin bosses (unpatterned concentric areas around each fin collar) were represented by cylinders or disks the thickness of the fin pattern depth and around 0.25 in. larger than the fin collar diameter. Smaller disks were used to represent the fin collar. Collar diameters of 0.5, 0.375 and 0.3125 in. were considered. The majority of the work was based on a transverse tube pitch of 1.25 in. Seven of the thirty-seven configurations investigated had transverse pitches of 1.0 in. however. The test matrix is given in Table F-1. Only the flat-fin and triangular patterned fin results are discussed herein.

CORRELATING PARAMETER

In an attempt to correlate the data a hydraulic diameter was defined by Eq. F-1

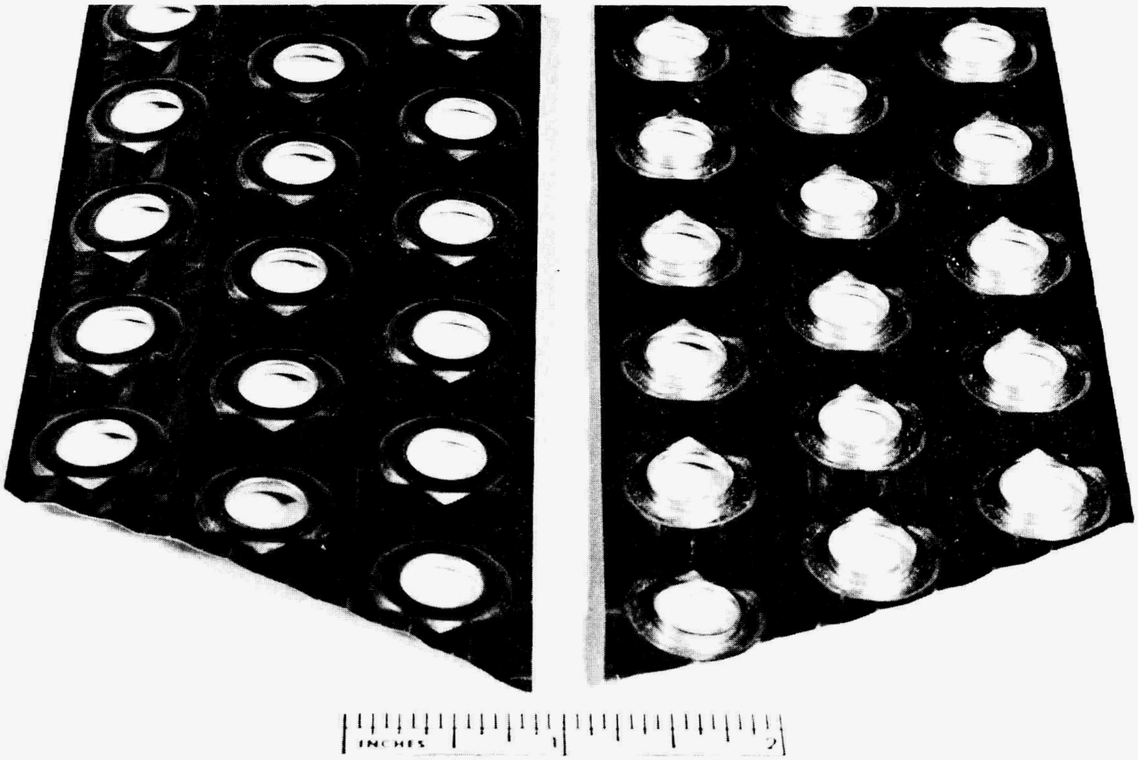


Figure F-1 - Commercial Residential Air Conditioning Fin (P_t = 1.0 in., P_ℓ = 0.866 in., 0.375 in., D_c = 0.401 in., P_D = 0.038 inch)

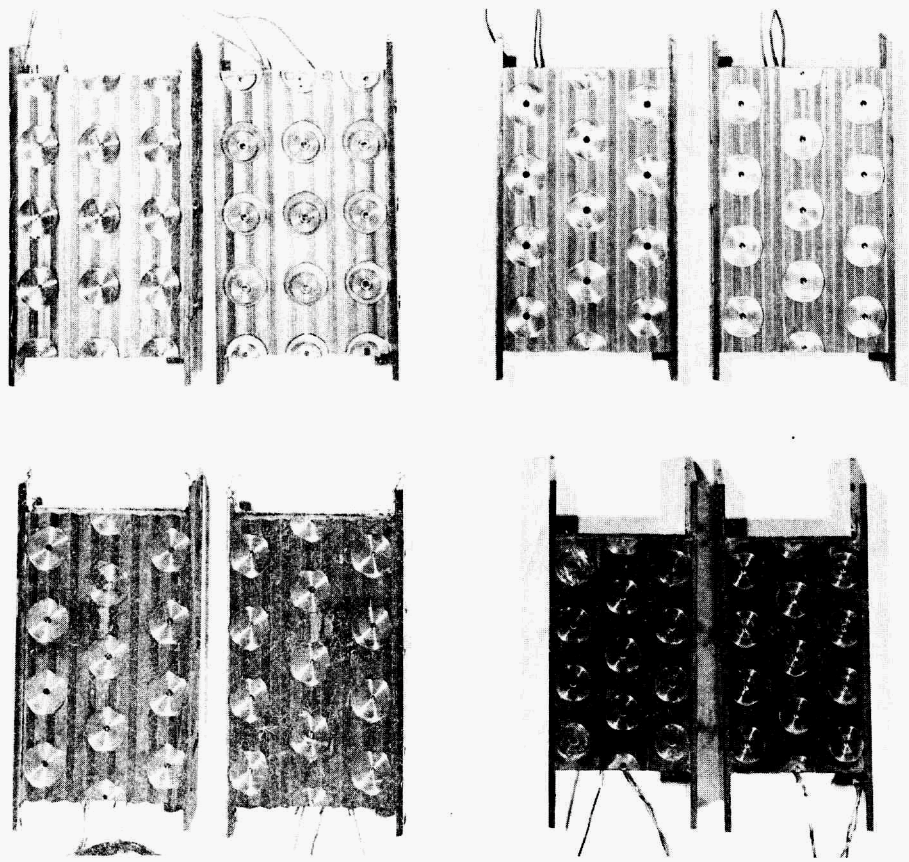


Figure F-2a - Sample Single Fin Model Heat Exchanger Plates, Top View

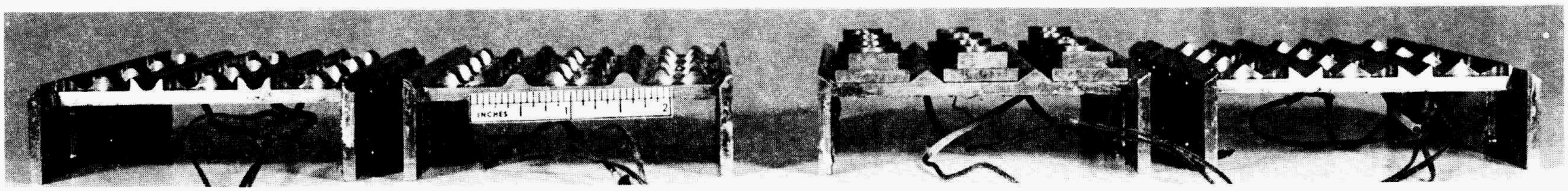


Figure F-2b - Sample of Single Fin Model Heat Exchanger Plates, Side View Showing 1/8 inch Deep Triangular and Sinuous Patterns

Table F-1

Single Fin Channel Geometries Tested

Table No.	Pattern*	P_t	P_L	D_t	W	No. of Rows	D_h	β	SO	P_D/W
1	None	1.25	1.083	0.5	0.077	6	.0121958	.14510	1.	0
2	None	1.25	1.083	0.5	0.077	4	.0121958	.14510	1.	0
3	None	1.25	1.083	0.5	0.077	3	.0121958	.14510	1.	0
4	None	1.25	1.083	0.5	0.077	2	.0121958	.14510	1.	0
5	None	1.25	0.875	0.5	0.077	3	.012023	.1795	1.	0
6	None	1.25	0.750	0.5	0.077	3	.011865	.2094	1.	0
7	T 3.038	1.00	0.866	0.375	0.077	3	.011729	.127533	1.03408	.494
8	T 3.038	1.25	1.083	0.375	0.077	3	.012125	.08162	1.02194	.494
9	T 3.038	1.25	1.083	0.500	0.077	3	.011946	.1451	1.02194	.494
10	T 3.038	1.00	0.866	0.500	0.077	3	.0114137	.226725	1.03408	.494
11	None	1.25	1.083	0.500	0.056	3	.00899148	.145104	1.	0
12	T 3.062	1.25	1.083	0.500	0.161	3	.0229814	.145104	1.0583	.389
13	T 3.062	1.25	1.083	0.500	0.110	3	.0161815	.145104	1.0583	.568
14	T 3.062	1.25	1.083	0.500	0.077	3	.0115555	.145104	1.0583	.812
15	T 3.038	1.25	1.083	0.500	0.161	3	.0237178	.145104	1.02194	.236
16	T 3.038	1.25	1.083	0.500	0.110	3	.016718	.145104	1.02194	.346
17	T 3.125	1.25	1.083	0.500	0.161	3	.0202384	.145104	1.21655	.779
18	T 3.125	1.25	1.083	0.500	0.077	3	.0101143	.145104	1.21655	1.624
19	T 2.038	1.00	0.866	0.375	0.094	3	.014392	.127533	1.01529	.404
20	T 2.038	1.00	0.866	0.375	0.077	3	.0119345	.127533	1.01529	.494
21	T 3.038	1.00	0.866	0.375	0.094	3	.014178	.127533	1.03408	.404
22	T 3.018	1.25	1.083	0.500	0.077	3	.0121385	.145104	1.00496	.234
23	T 3.028	1.25	1.083	0.500	0.077	3	.0120586	.145104	1.01197	.364
24	T 3.0485	1.25	1.083	0.500	0.077	3	.0118055	.145104	1.03478	.63
25	T 3.093	1.25	1.083	0.500	0.077	3	.0108829	.145104	1.12694	1.209
26	T 4.038	1.00	0.8663	0.375	0.077	3	.0114599	.127533	1.05982	.494
27	S 3.0625	1.25	1.083	0.500	0.077	3	.0114624	.145104	1.06733	.812
28	S 3.125	1.25	1.083	0.500	0.077	3	.00950732	.145104	1.29756	1.624
29	S 3.062	1.25	1.083	0.500	0.161	3	.0228052	.145104	1.06733	.388
30	S 3.125	1.25	1.083	0.500	0.161	3	.0190731	.145104	1.29756	.776
31	S 3.125**	1.25	1.083	0.500	0.161	3	.0190731	.145104	1.29756	.776
32	T 3.125**	1.25	1.083	0.500	0.161	3	.0202384	.145104	1.21655	.776
33	T 3.062**	1.25	1.083	0.500	0.077	3	.0115556	.145104	1.0583	.812
34	T 2.0625	1.00	0.8663	0.375	0.077	3	.0116575	.127533	1.04083	.812
35	T 3.038	1.00	0.8663	0.3125	0.077	3	.0118612	.0885645	1.03408	.494
36	T 3.038**	1.25	1.083	0.5	0.161	3	.023720	.145104	1.02194	.236
37	T 3.0625**	1.25	1.083	0.5	0.161	3	.0229814	.145104	1.0583	.388

T - 3.0625 Triangular pattern, 3 patterns/longitudinal pitch, pattern depth 0.0625 inch.

S - 3.125 Sinuous pattern, 3 patterns/longitudinal pitch, pattern depth 0.125 inch.

** - Inline Tube Geometry.

+ - Page in the Appendix showing air side convection coefficient and corrected pressure drop per row as a function of standard face velocity.

$$D_h = 4\left\{\frac{\text{Open Channel Volume}}{\text{Total Surface Area}}\right\} = \frac{4\left(P_tP_\ell - \frac{\pi D_c^2}{4}\right) W}{\left(2 \sec\theta \left(P_tP_\ell - \frac{\pi D_c^2}{4}\right) + \pi D_c W\right)12} \tag{F-1}$$

$$D_h = \frac{2\,W\,(1-\beta)}{12\,[\sec\theta\,(1-\beta) + \frac{2\,W\,\beta}{D_c}]}$$

$$\text{where } \beta = \frac{\pi D_c^2}{4\,P_tP_\ell}\ ,\quad \sec\theta = \frac{\text{Pattern Area}}{\text{Projected Area}} = \frac{\sqrt{x^2 + P_D^2}}{x},\ x = \frac{P_\ell}{2N_{ppp}}$$

The velocity used in the original correlation of the data was a mean velocity defined by Eq. F-2.

$$\overline{V} = V_{face}\left(\frac{\text{Total Channel Volume}}{\text{Open Channel Volume}}\right) = \frac{V_{fin}}{1-\beta}$$

$$\text{where } V_{fin} = V_{face}\left(\frac{\frac{1}{N_f}}{\frac{1}{N_f} - t_f}\right) = V_{face}\left(\frac{\frac{1}{N_f}}{W}\right) \tag{F-2}$$

and N_f is the number of fins of thickness t_f per inch. W is the width of the channel between fins. V_{fin} is the calculated air velocity in the channel between adjacent fins assuming no tubes or fin collars are present.

At low air velocities where heat exchanger effectiveness is high the leaving air temperature difference ($T_{wall}-T_{air\ out}$) is small. Smaller errors in exit air temperature measurement would lead to large errors in the calculated log mean temperature difference (LMTD). The data was therefore reduced using the average temperature difference (ATD), the temperature difference of the channel wall (which was held constant) and the average air temperature [0.5 ($T_{air\ in}$ - $T_{air\ out}$)].

It can be demonstrated that the Nusselt Number based on the LMTD (Nu_ℓ) is related to the Nusselt Number based on the ATD (Nu_a) by Eq. F-3.

$$Nu_\ell = \frac{Gz}{4}\ln\left(\frac{1 + 2\frac{Nu_a}{Gz}}{1 - 2\frac{Nu_a}{Gz}}\right)$$

$$\text{where } Gz = Re \cdot Pr \cdot \frac{D_h}{L} = \frac{2W^2\rho\, V_{fin}\, Pr\, (1-\beta)}{12\,\mu\, N_R P_\ell\, [\sec\theta\,(1-\beta) + \frac{2W\beta}{D_c}]^2} \tag{F-3}$$

From smoothed Nu_a vs Gz plots, Nu_ℓ vs Re or h_a vs V_{face} or V_{max} curves can be generated. It is also noted that for heat exchangers of high effectiveness that Nu_a will approach 0.5 Gz as an asymptote. This is very useful at low Gz where a small amount of heat loss would result in a large error in the calculated value of Nu_a.

Figure F-3 is the Nu_a vs Gz plot for 2, 3, 4 and 6 row flat fin models. Heat balance errors are greatest for the 2 row model, the entry and exit section heat transfer to the air being a greater percentage of the total. These curves are for a $P_t \times P_\ell$, D_C (1.25 x 1.083, 0.5 in.) surface. Figure F-4 and all following Nu_a vs Gz curves are for 3 row models only. Figure F-4 shows how a change in the longitudinal pitch (P_ℓ) affects Nu_a.

The remainder of the data presented is for patterned (wavy) fins with the T3.038 being used as a standard. The T3 indicates three transverse triangular ridges per longitudinal pitch with a double amplitude pattern depth, P_D, (peak to valley height) of 0.038 in. Figures F-5 and F-6 show that for either 0.5 or 0.375 in. collar diameters (D_C) data for 1.25 x 1.083 and 1.00 x 0.866 three row heat exchangers are represented by a single curve.

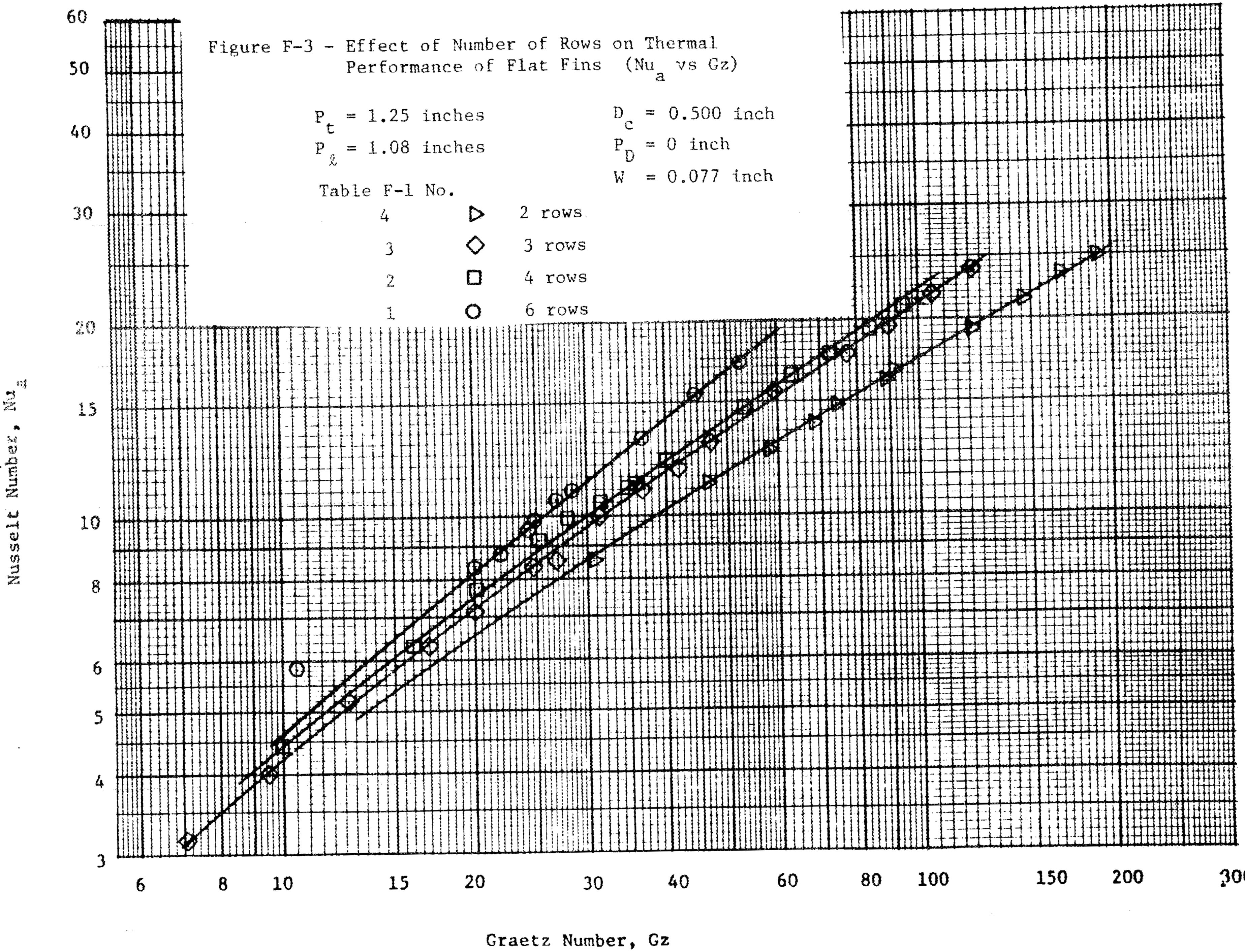
Figure F-7 shows data for a 1.00 x 0.866 in. model with 0.5, 0.375 and 0.312 in. diameter fin collars and Fig. F-8 presents data for a 1.25 x 1.083 in. model with 0.5 and 0.375 in. diameter fin collars. Figures F-5 through F-8 indicate that Nu_a vs Gz gives a good correlation for these equilateral staggered tube pitch variations and for tube collar diameter variations.

GRAPHICAL DATA PRESENTATION (HEAT TRANSFER SINGLE FIN TESTS)

The data to evaluate the effect of pattern depth on thermal performance was taken with 1.25 x 1.08, 0.5 in. models for the most part with a fin spacing (W) of 0.077 in. (N_f = 12 if t_f = 0.006 in.). Limited comparisons were also made with fin spacings of 0.110 and 0.161 in. (N_f of 8.62 and 6). Figure F-9 is a plot of the data for pattern depths (P_D) of 0, 0.018, 0.028, 0.038, 0.0485, 0.0625, 0.093 and 0.125 in. Each of these models used three patterns or transverse ridges per pitch. Figure F-9 clearly shows the value of this type of plot as the data for the high fin patterns closely approaches the asymptote $Nu_a \to 0.5$ Gz.

Figure F-10 plots data for the 6 fins/in. (W = 0.161 in.) model with pattern depths of 0.038, 0.0625 and 0.125 in. Figure F-11 plots data for an 8.62 fin/in. (W = 0.110 in.) model with pattern depths of 0.038 and 0.0625 in. It is apparent that high patterns are most effective with greater fin spacings. For example, the

Figure F-3 - Effect of Number of Rows on Thermal Performance of Flat Fins (Nu_a vs Gz)



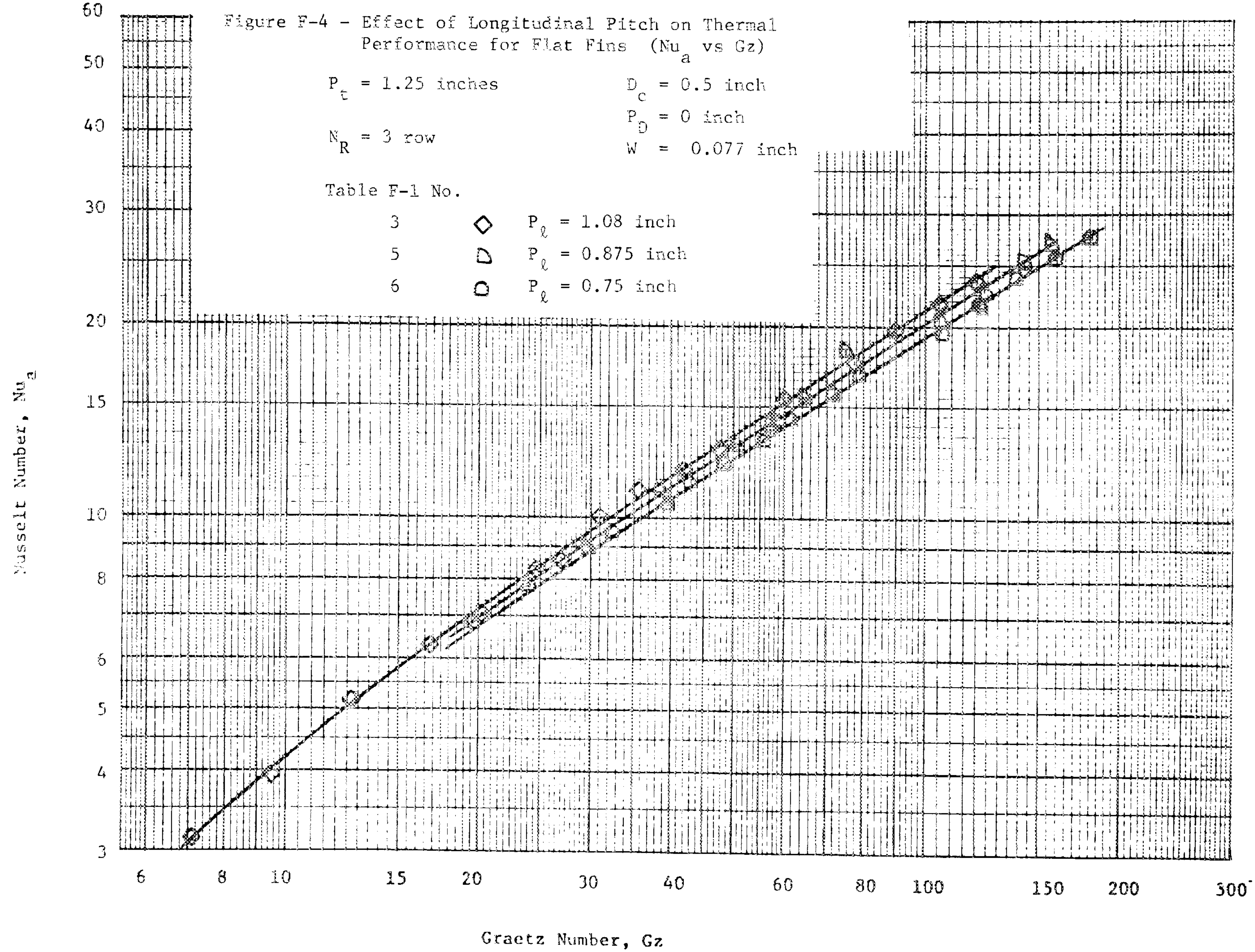
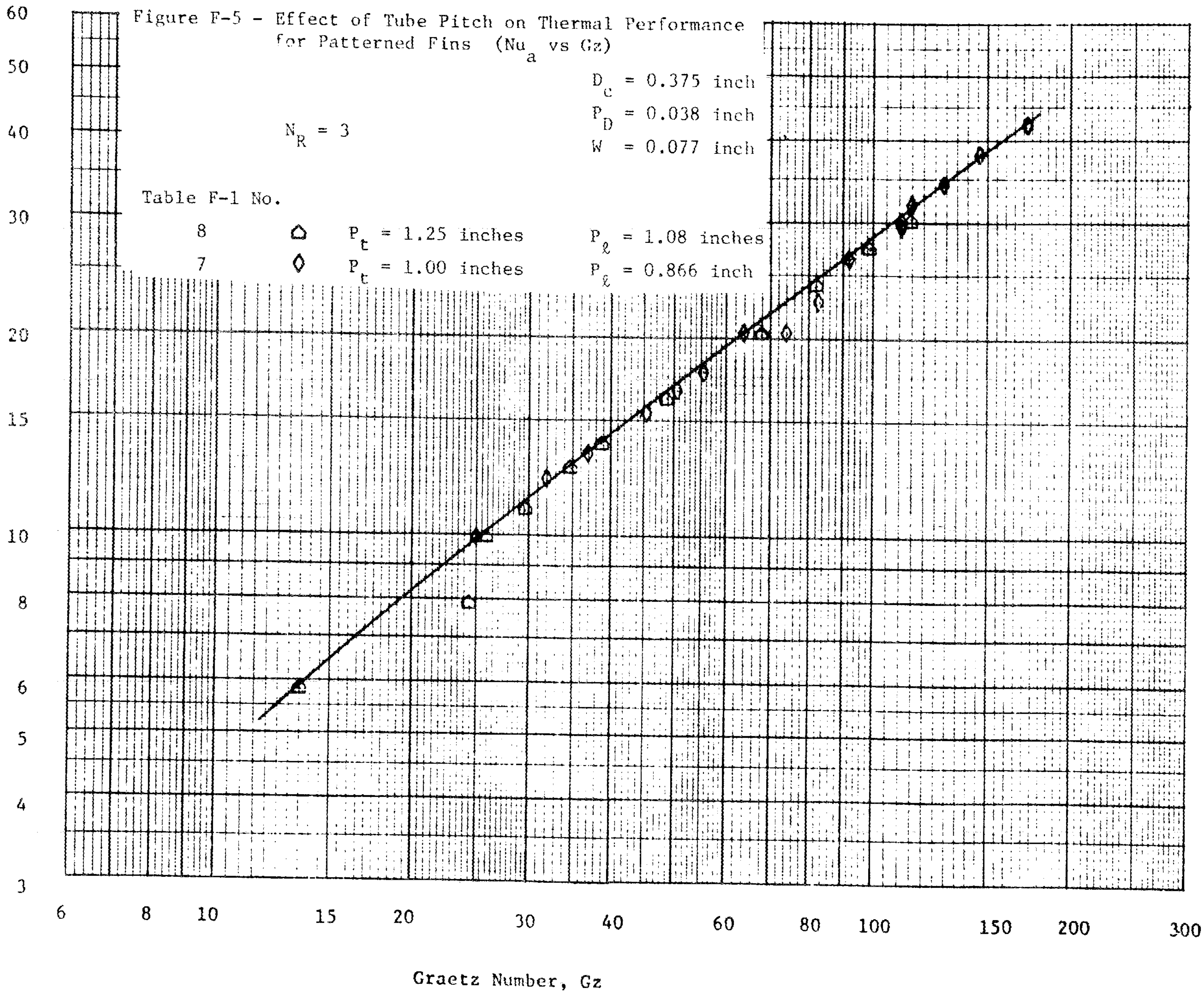


Figure F-4 - Effect of Longitudinal Pitch on Thermal Performance for Flat Fins (Nu_a vs Gz)
P_t = 1.25 inches
N_R = 3 row
D_c = 0.5 inch
P_D = 0 inch
W = 0.077 inch
Table F-1 No.
3 ◇ P_ℓ = 1.08 inch
5 P_ℓ = 0.875 inch
6 P_ℓ = 0.75 inch
Nusselt Number, Nu_a
60
50
40
30
20
15
10
8
6
5
4
3
6
8
10
15
20
30
40
60
80
100
150
200
300
Graetz Number, Gz

Figure F-5 - Effect of Tube Pitch on Thermal Performance for Patterned Fins (Nu_a vs Gz)



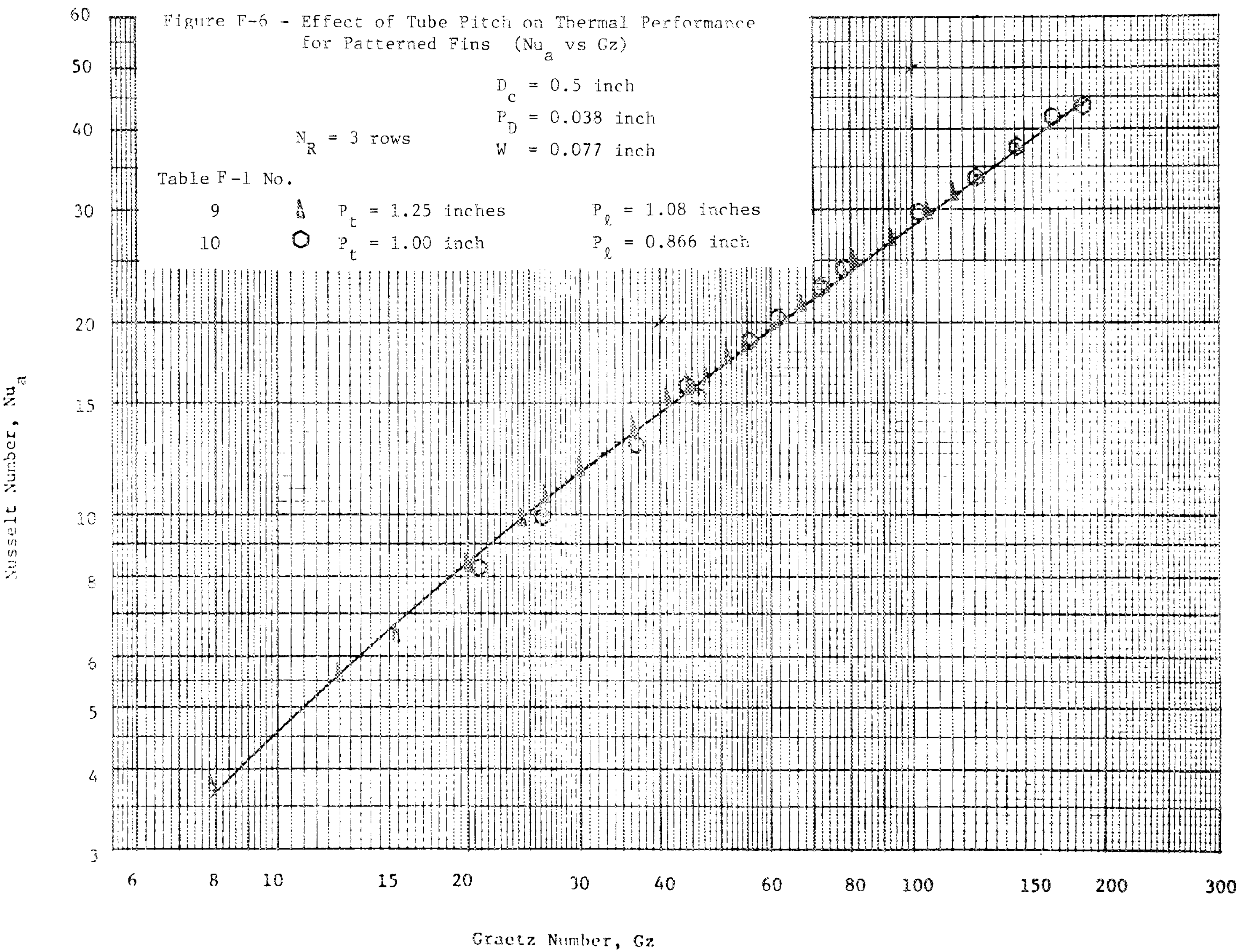


Figure F-6 - Effect of Tube Pitch on Thermal Performance for Patterned Fins (Nu_a vs Gz)
D_c = 0.5 inch
P_D = 0.038 inch
W = 0.077 inch
N_R = 3 rows
Table F-1 No.
9 P_t = 1.25 inches P_ℓ = 1.08 inches
10 P_t = 1.00 inch P_ℓ = 0.866 inch
Nusselt Number, Nu_a
Graetz Number, Gz
60
50
40
30
20
15
10
8
6
5
4
3
6
8
10
15
20
30
40
60
80
100
150
200
300

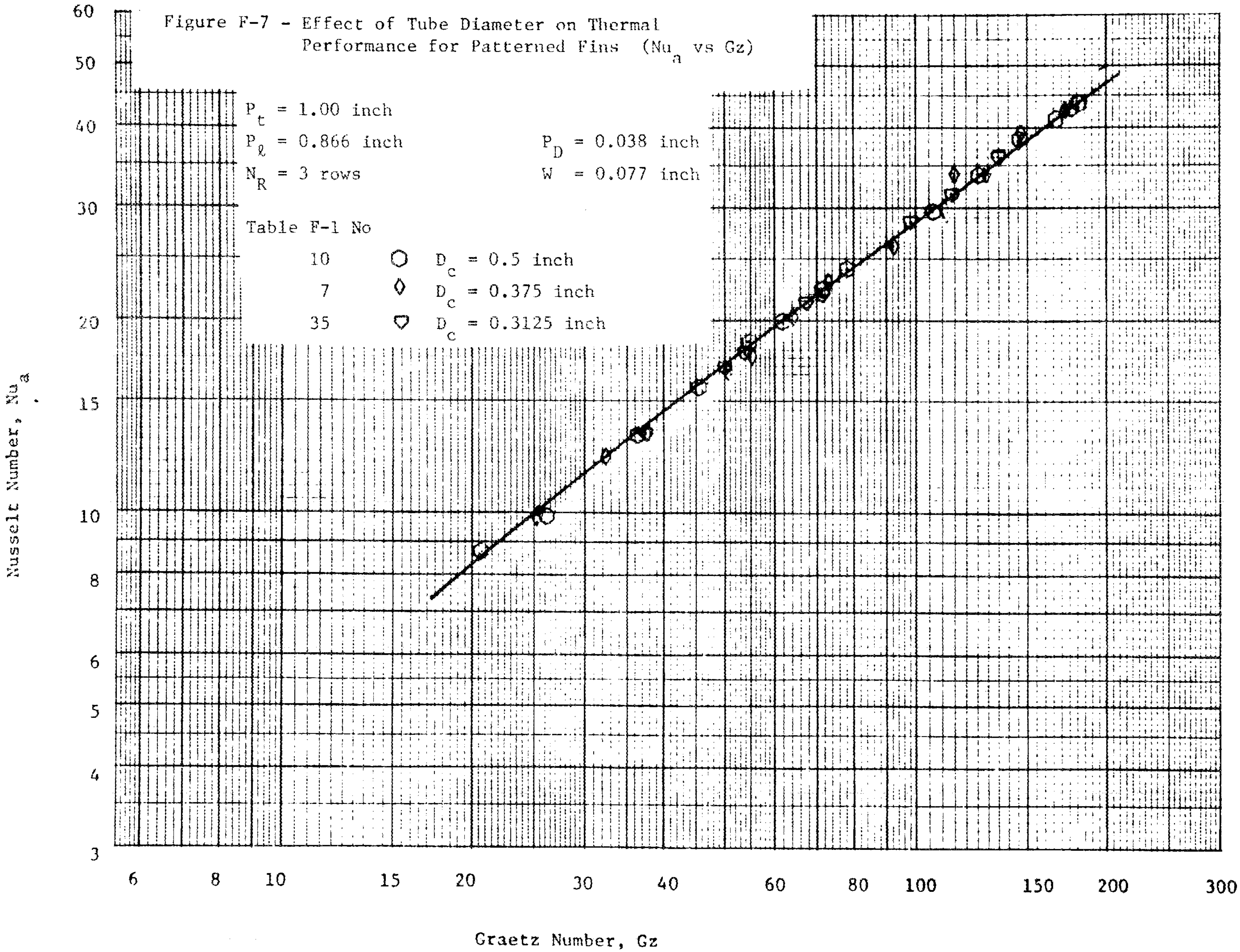


Figure F-7 - Effect of Tube Diameter on Thermal Performance for Patterned Fins (Nu_a vs Gz)

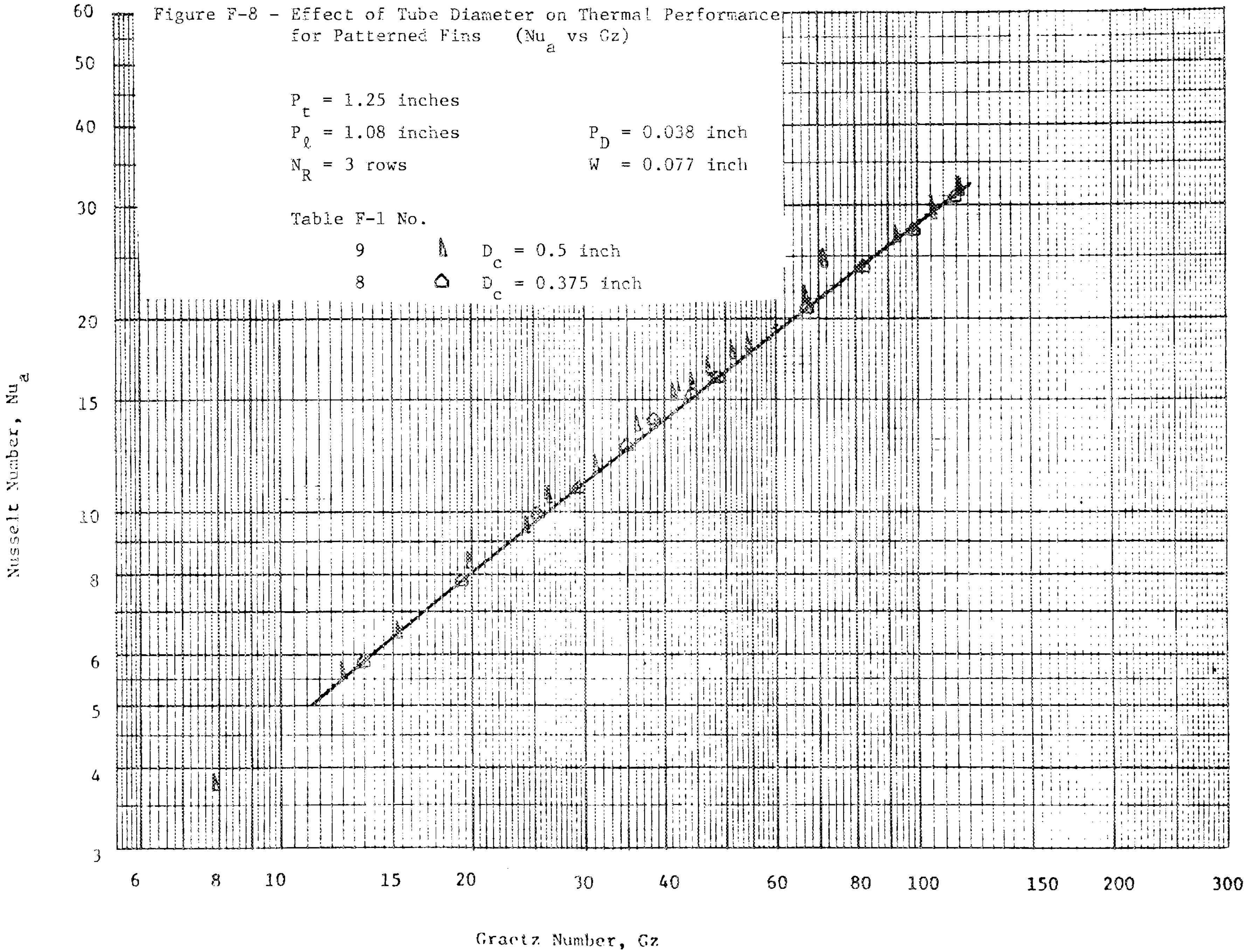


Figure F-8 - Effect of Tube Diameter on Thermal Performance for Patterned Fins (Nu_a vs Gz)

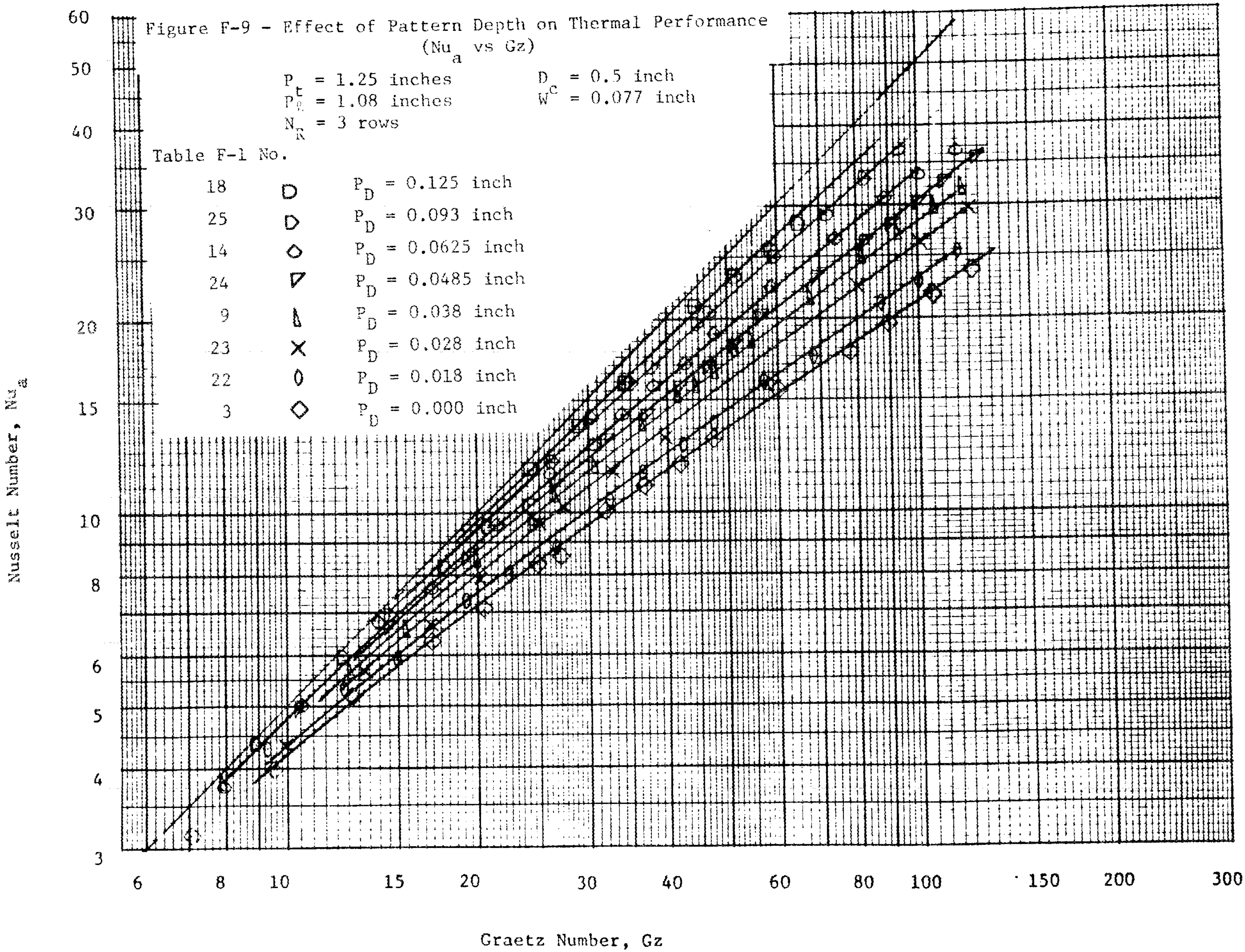


Figure F-9 - Effect of Pattern Depth on Thermal Performance (Nu_a vs Gz)

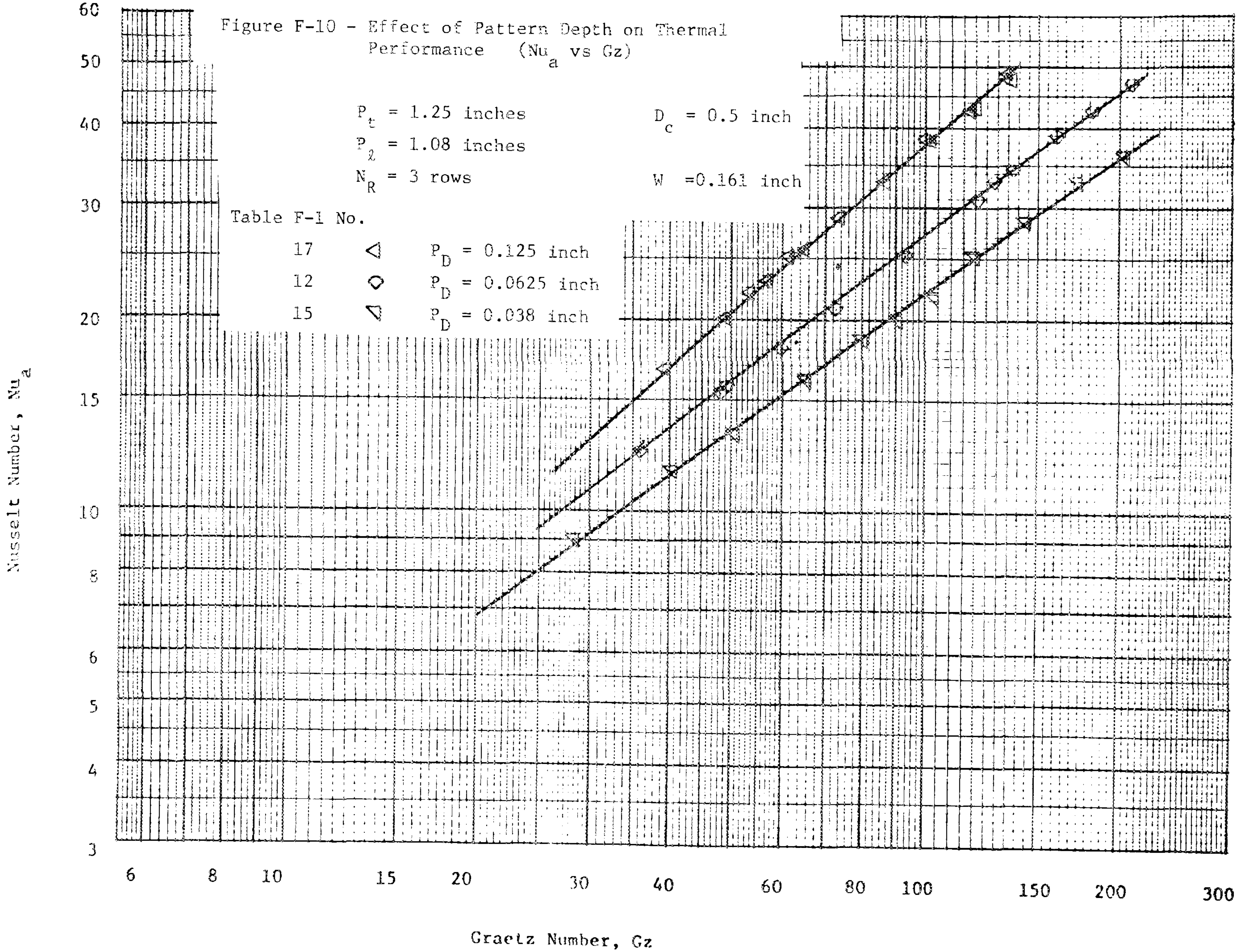


Figure F-10 - Effect of Pattern Depth on Thermal Performance (Nu_a vs Gz)

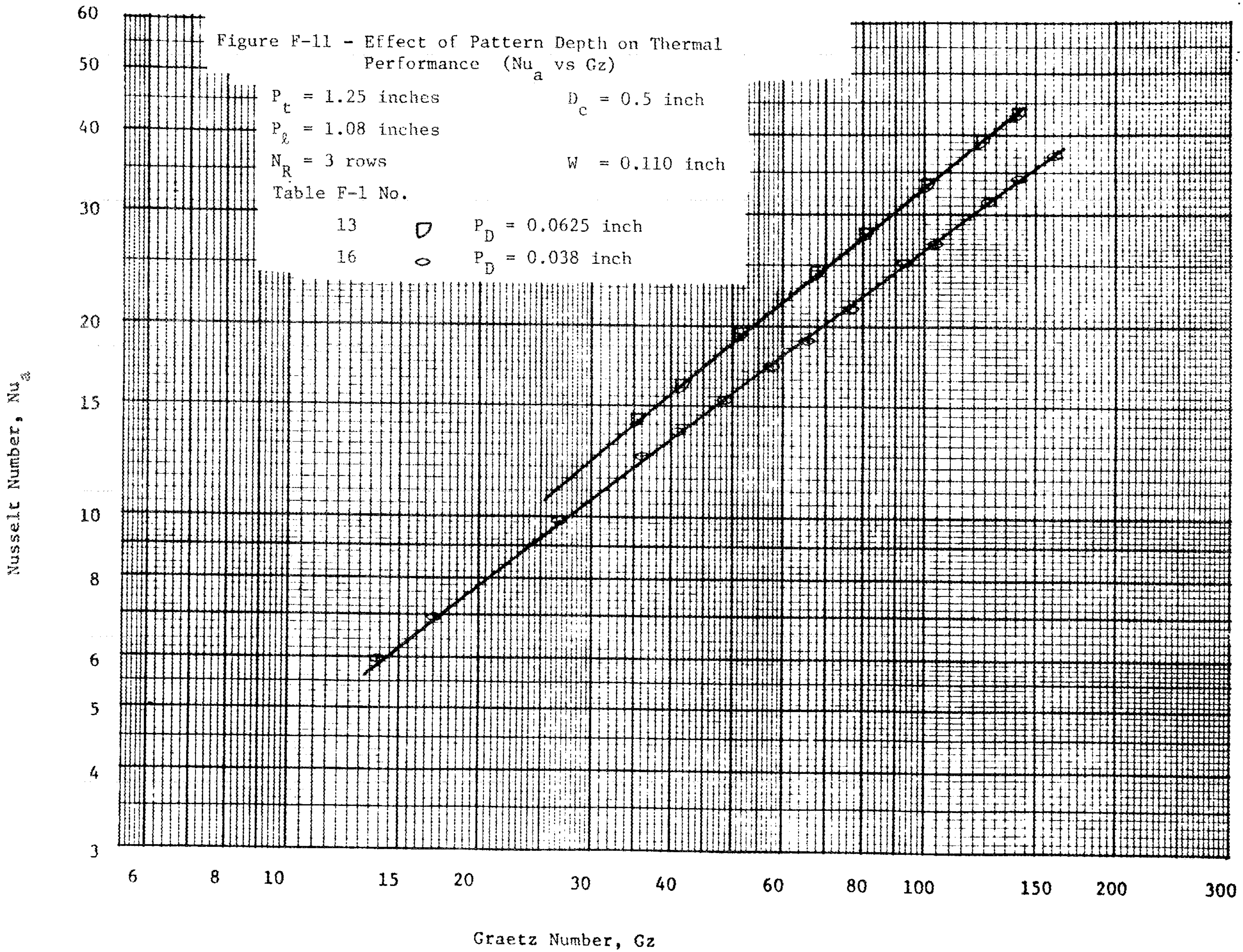


Figure F-11 - Effect of Pattern Depth on Thermal Performance (Nu_a vs Gz)

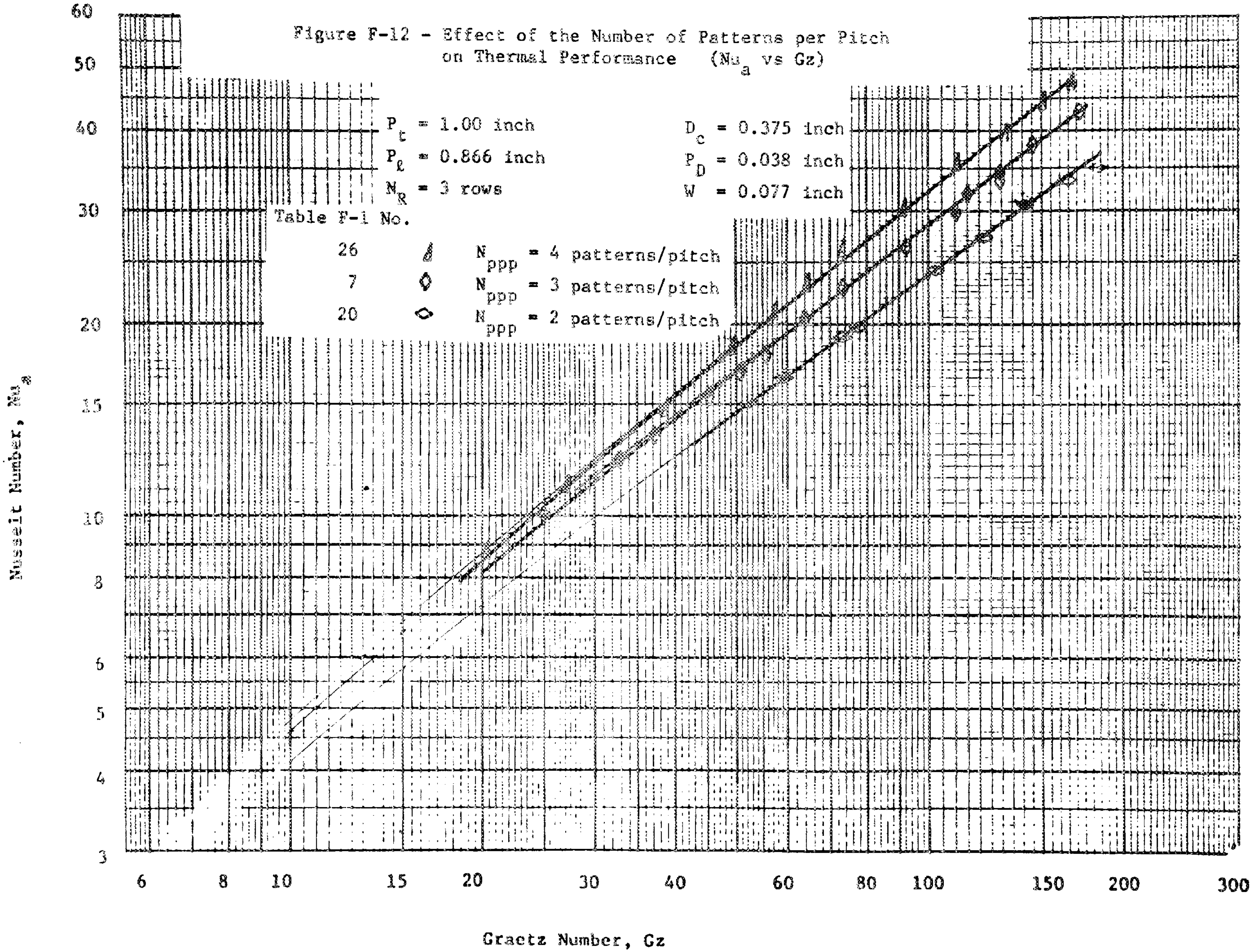


Figure F-12 - Effect of the Number of Patterns per Pitch on Thermal Performance (Nu_a vs Gz)
P_t = 1.00 inch
P_ℓ = 0.866 inch
N_R = 3 rows
D_c = 0.375 inch
P_D = 0.038 inch
W = 0.077 inch
Table F-1 No.
26 N_{ppp} = 4 patterns/pitch
7 N_{ppp} = 3 patterns/pitch
20 N_{ppp} = 2 patterns/pitch
Nusselt Number, Nu_a
Graetz Number, Gz

ratio $\frac{Nu_{a(P_D = 0.062)}}{Nu_{a(P_D = 0.038)}}$ is 1.099, 1.155 and 1.173 for fin spacings of 0.077, 0.110 and 0.161 in., respectively at a Gz of 30.

The effect of number of patterns/pitch (N_{ppp}) was investigated for a 1.00 x 0.866, 0.375 in. model only. T4.038 and T2.038 models were constructed and evaluated with a fin spacing 0.077 in. Figure F-12 shows the Nu_a vs Gz plot for the data. The faint lines below Gz = 30 were attempts to better extrapolate the data to low Gz using the 0.5 Gz asymptote.

Figure F-13 is a similar plot with a fin spacing of 0.094 in. (N_f 10 fins/in.) for T2.038 and T3.038 only. Remembering that Nu_a vs Gz appeared to correlate variations in equilateral tube pitch and collar diameter, Figure F-14 presents data from T2.062 and T3.062 models with fin spacings of 0.077 in. but at different pitches and collar diameters. Although the pitchs and collar diameters are different for the model data shown in Figure F-14, these effects have been shown to be amendable to the correlation used so the comparison is felt to be valid.

Only limited tests were run with other fin spacings. Figure F-15 is the only fin spacing data for flat fins (P_D = 0) and offers a comparison for the 1.25 x 1.083, 0.5 in. model at fin spacings of 0.077 and 0.056 in.(N_f ~ 12 and 16 fins/in.). The data are correlated well at high Gz but at lower Gz (lower velocities) the closer spaced fins with their higher heat transfer coefficient more closely approach the 0.5 Gz asymptote. Figures F-16, F-17 and F-18 are Nu_a vs Gz plots for T3.038 and T3.062 and T3.125 models respectively. The first two show results for fin spacings (W) of 0.077, 0.110 and 0.161 in. (12 to 6 fin/in). The T3.125 model was tested only at fin spacings of 0.077 and 0.161 in. The ratio of $Nu_{a(0.077}/Nu_{a(0.161)}$ is about 1.25, 1.18 and 1.11 for the 0.038, 0.062 and 0.125 in. deep patterns, respectively. These curves indicates the Nu_a vs Gz correlation over compensates for fin spacing ($Gz \propto W^2$) for large changes in fin spacing (ex. $\frac{W_1}{W_2} = 2$) but is a reasonable correlation for smaller changes (ex. $\frac{W_1}{W_2} < 1.5$) and that the correlation is better for high pattern than for low patterns (larger sec θ associated with higher patterns reduces Gz for a given fin spacing).

The j factor versus Reynolds number curves showing the effect of fin spacing of the data of Rich[2] for flat fins were reduced assuming (based on a phone conversation with the author) that the fin effectiveness was constant and calculated based on an air side convection coefficient of 10 Btu/h-ft^2-°F. The data was read from the curves in the published work and an air side convection coefficient calculated

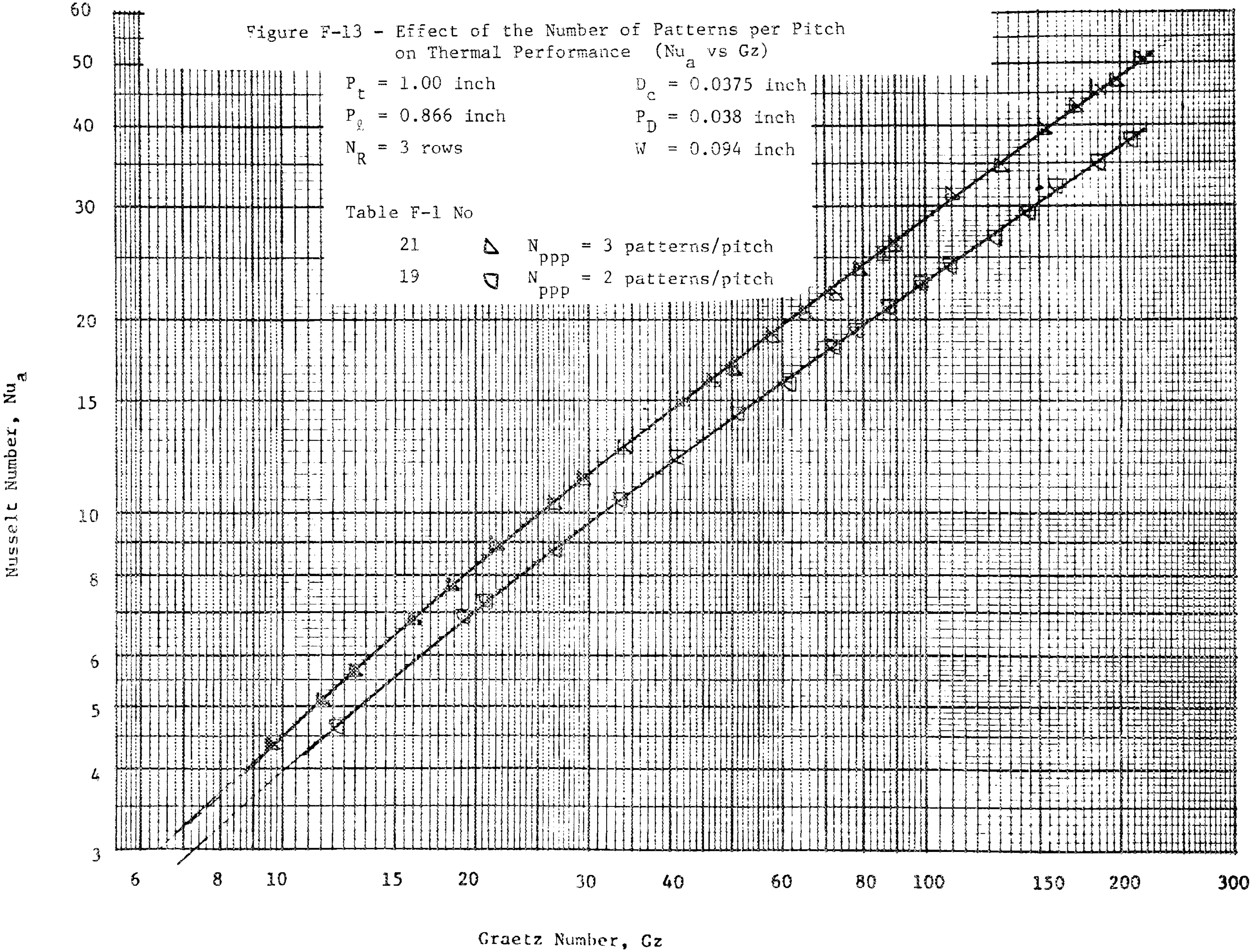


Figure F-13 - Effect of the Number of Patterns per Pitch on Thermal Performance (Nu_a vs Gz)

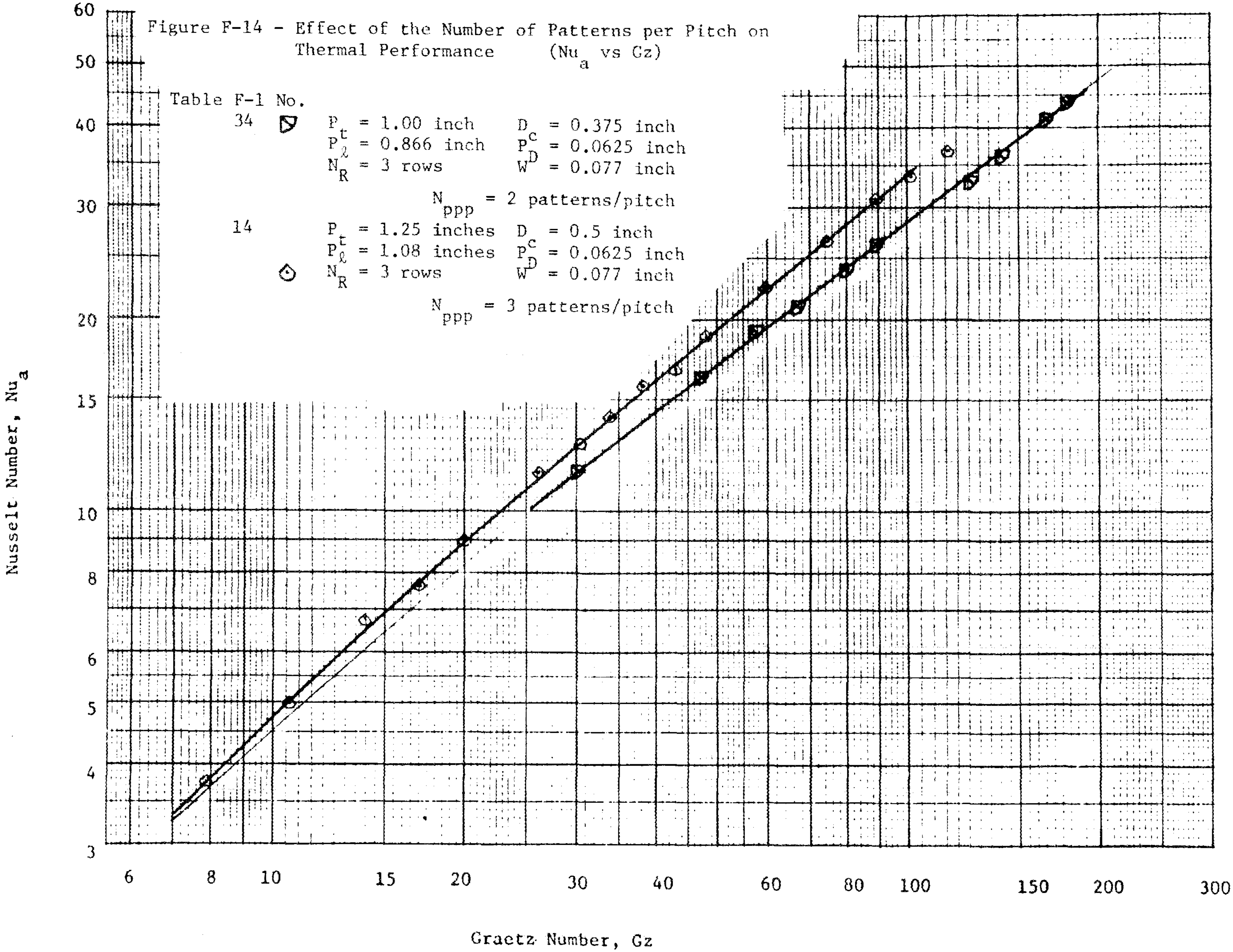


Figure F-14 - Effect of the Number of Patterns per Pitch on Thermal Performance (Nu_a vs Gz)

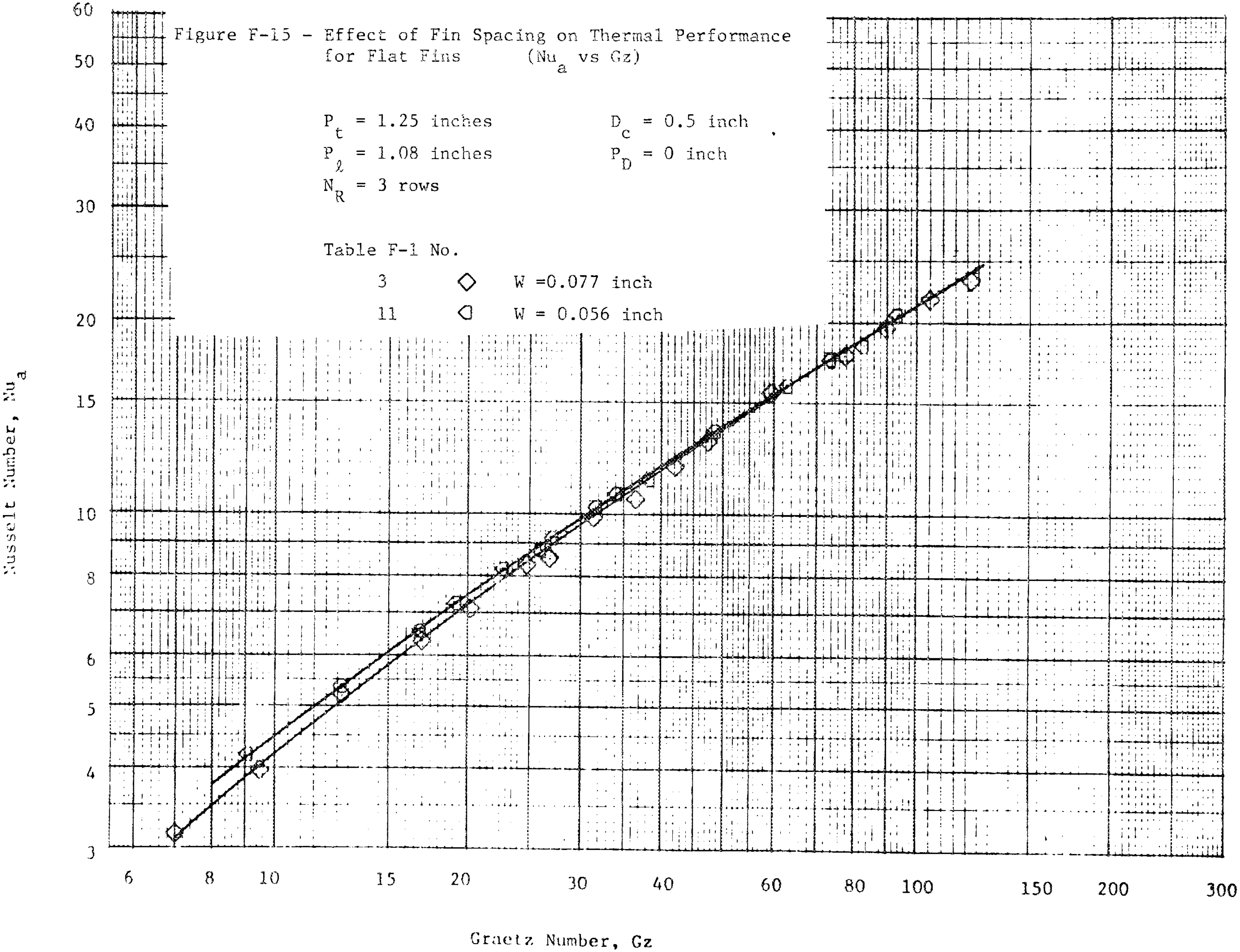


Figure F-15 - Effect of Fin Spacing on Thermal Performance for Flat Fins (Nu_a vs Gz)
P_t = 1.25 inches
P_ℓ = 1.08 inches
N_R = 3 rows
D_c = 0.5 inch
P_D = 0 inch
Table F-1 No.
3 W =0.077 inch
11 W = 0.056 inch
Nusselt Number, Nu_a
Graetz Number, Gz
60
50
40
30
20
15
10
8
6
5
4
3
6
8
10
15
20
30
40
60
80
100
150
200
300

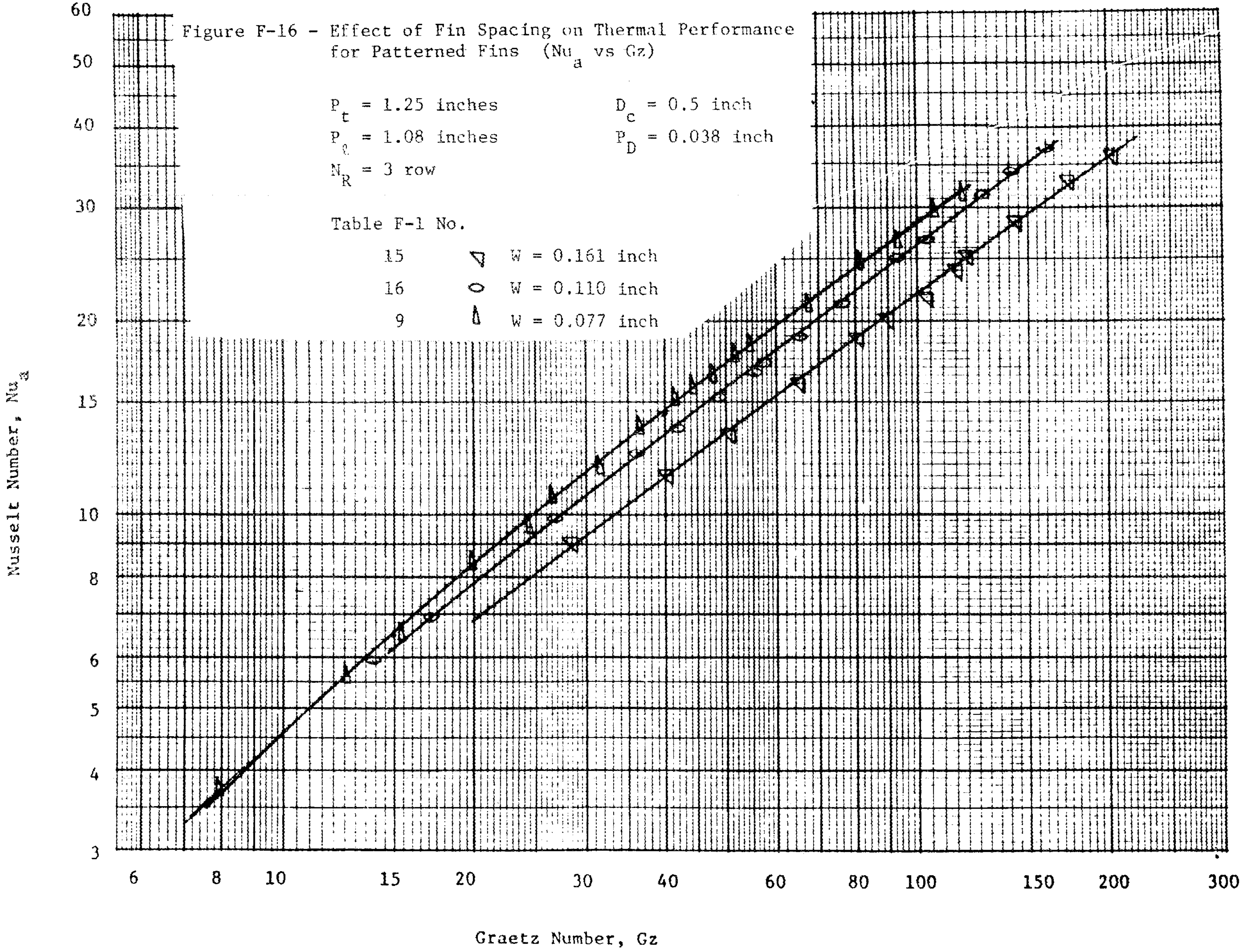


Figure F-16 - Effect of Fin Spacing on Thermal Performance for Patterned Fins (Nu_a vs Gz)

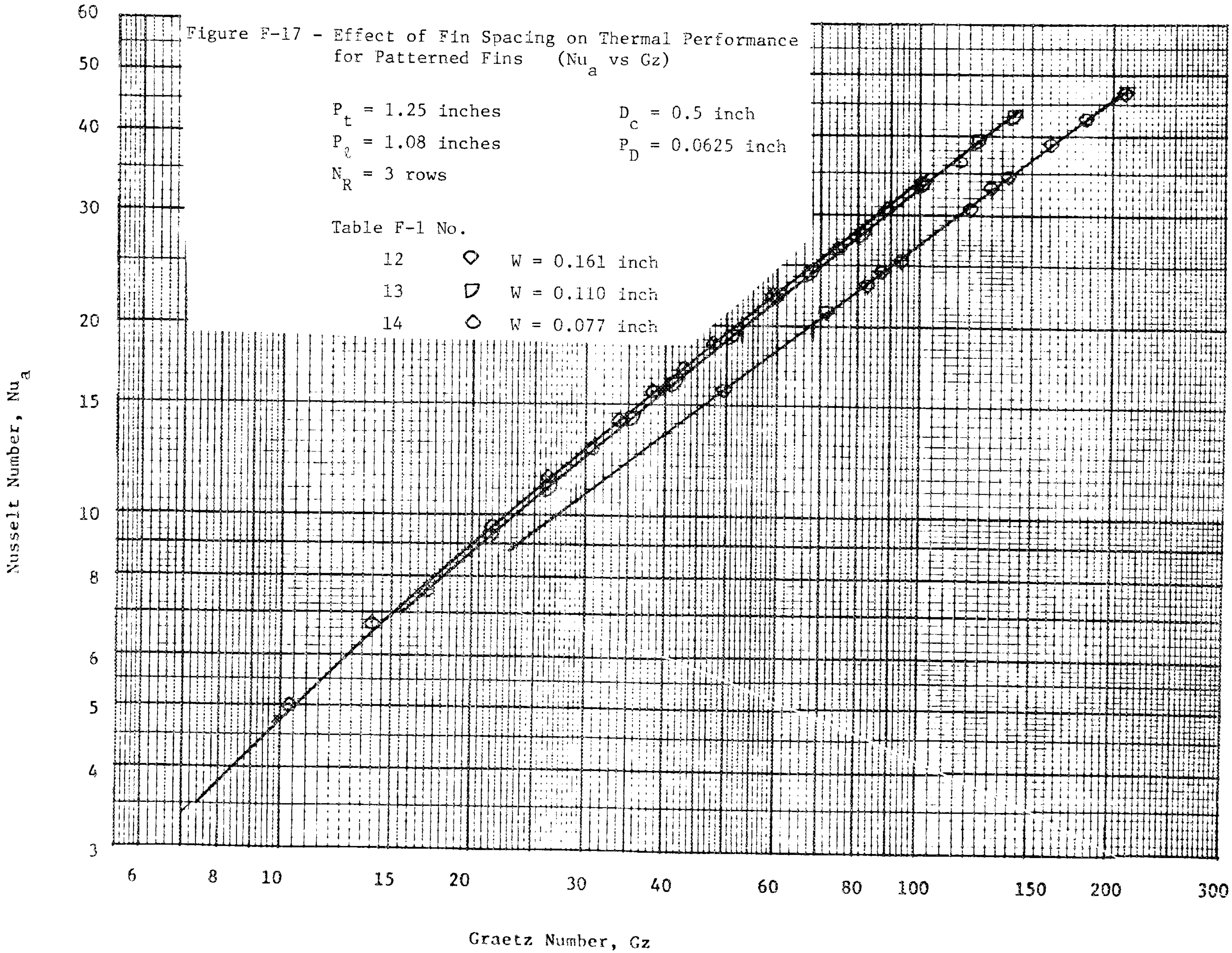


Figure F-17 - Effect of Fin Spacing on Thermal Performance for Patterned Fins (Nu_a vs Gz)

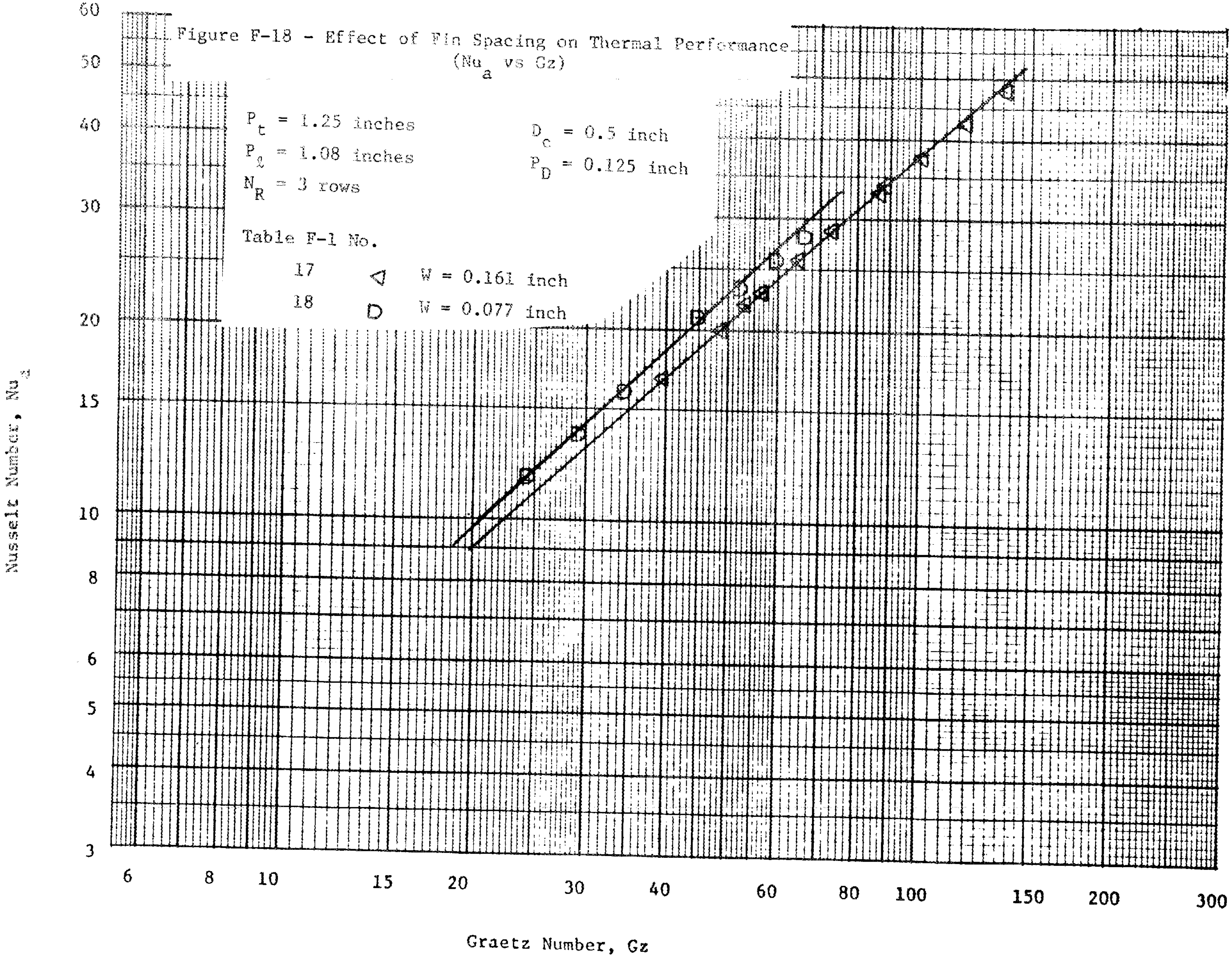


Figure F-18 - Effect of Fin Spacing on Thermal Performance (Nu_a vs Gz) assuming the 0.006 in. tin dipped copper fins had a conductivity of 220 Btu/h-ft^2-°F. Some alloying of the tin with the copper will have occurred substantially reducing the conductivity of the fin material. This factor being an unknown has been disregarded but could substantially increase the calculated value of h_a if included especially at higher air velocities. For example for an air side convection coefficient of 10 Btu/h-ft^2-°F, a conductivity of 220 Btu/h-ft^2-°F would result in a fin effectiveness of 90.6 percent whereas if the conductivity were only 80 Btu/hft°F the fin efficiency would be only 78.2 percent. If the data were reduced using too high a fin affectiveness the calculated values of the convection coefficient would be low.

The calculated results for 2.92, 4.42, 6.67, 7.67, 9.17, 11.75, 14.5 and 20.6 fins/in. are plotted as Figures F-19 through F-26 (log h_{a_ℓ} vs. log V_{max}). The data of McQuiston[3,4] for the five surfaces he investigated are included in the preceeding figures for comparison. McQuiston's data are reduced assuming no fin collar to tube thermal impedance but considering the effect of air side convection coefficient on fin efficiency and so will give an air side convection coefficient which is slightly low. The error in convection coefficient will be larger at higher air side velocities. In Figure F-24 the 12 fin per in. single fin model data is shown with that of Rich and McQuiston. The single fin data is higher than that reported by the others at low velocities but approaches that of Rich at higher velocities. This is the only single fin model directly comparable with the Rich and McQuiston data.

PRESSURE DROP DATA - SINGLE FIN TESTS

The pressure drop data corresponding to the single fin tests are given in Figures F-27 through F-41. The variation in slope and magnitude indicate that small variations in entry and exit block alignment can greatly affect the pressure drop results. These plots of pressure drop per row (in. of water/row) vs. V_{max}, the maximum velocity seen in the coil should be interpreted with care. The maximum velocity was used as the independent variable here in an attempt to correlate the flat fin pressure drop data. The face velocity is normally used in reporting the pressure drop data but face velocity is independent of fin spacing, tube pitch and collar diameter and is not a correlating factor. V_{max} as defined by Equation 4 includes the sec θ term which is a function of fin pattern.

$$V_{max} = V_{face} \frac{\left(\frac{1}{N_f - t_f}\right)}{\left(\frac{1}{N_f}\right)} \left(\frac{P_t}{P_t - D_c}\right) \sec \theta \qquad \text{(F-4)}$$

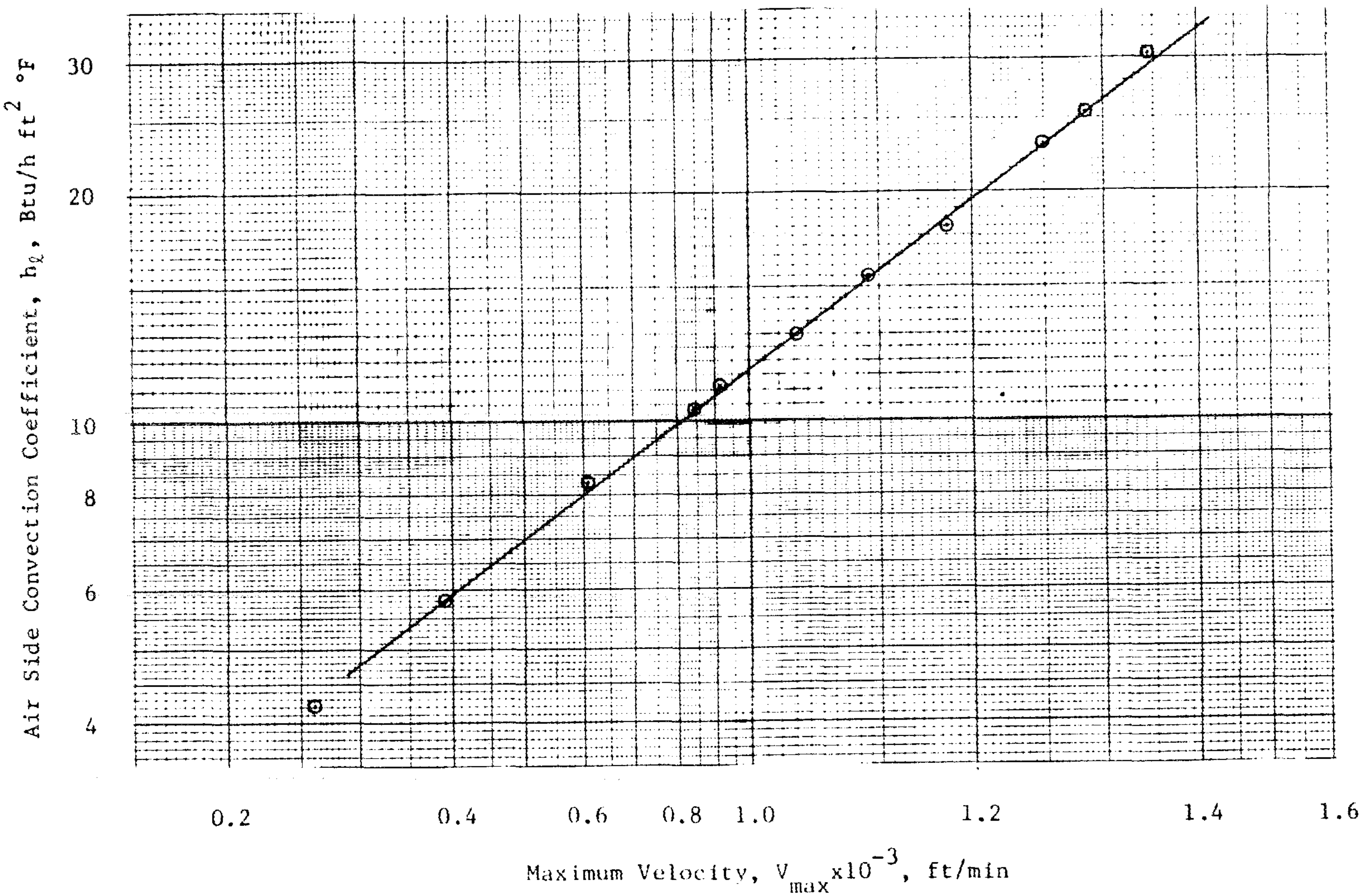


Figure F-19 - Air Side Convection Coefficient as a Function of Maximum Fin Channel Velocity for Flat Fins (~2.92 fins/inch)

○ P_t = 1.250 inch, P_ℓ = 1.086 inch, N_R = 4, D_c = 0.525 inch, W = 0.336 inch, Rich's Data[2]

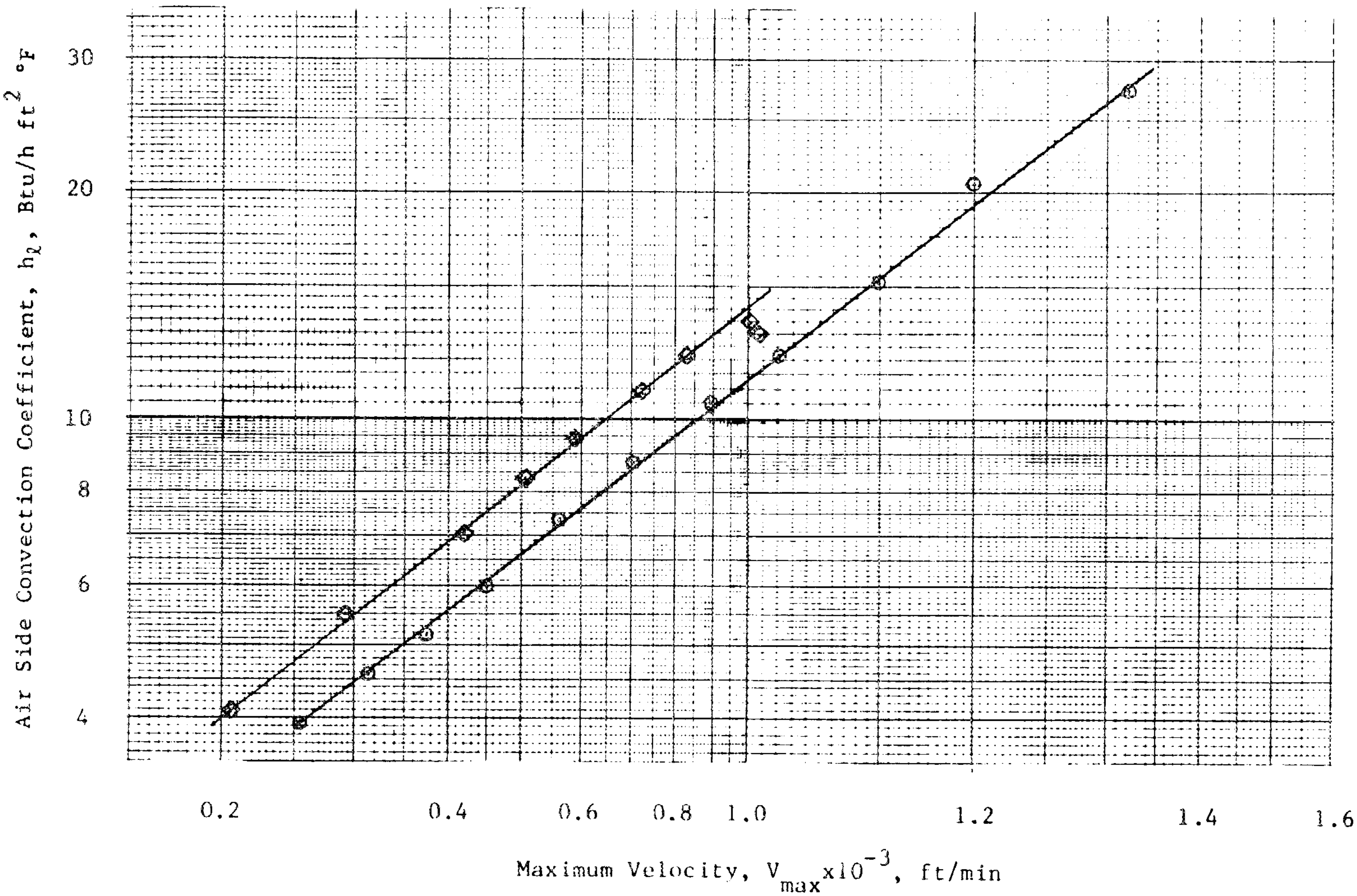


Figure F-20 - Air Side Convection Coefficient as a Function of Maximum Fin Channel Velocity for Flat Fins (∿4.42 fins/inch)

◇ P_t = 1.000 inch, P_ℓ = 0.866 inch, N_R = 4, D_c = 0.404 inch, W = 0.244 inch, (McQuiston's Data[4])

○ P_t = 1.250 inches, P_ℓ = 1.083 inches, N_R = 4, D_c = 0.525 inch, W = 0.220 inch (Rich's Data[2])

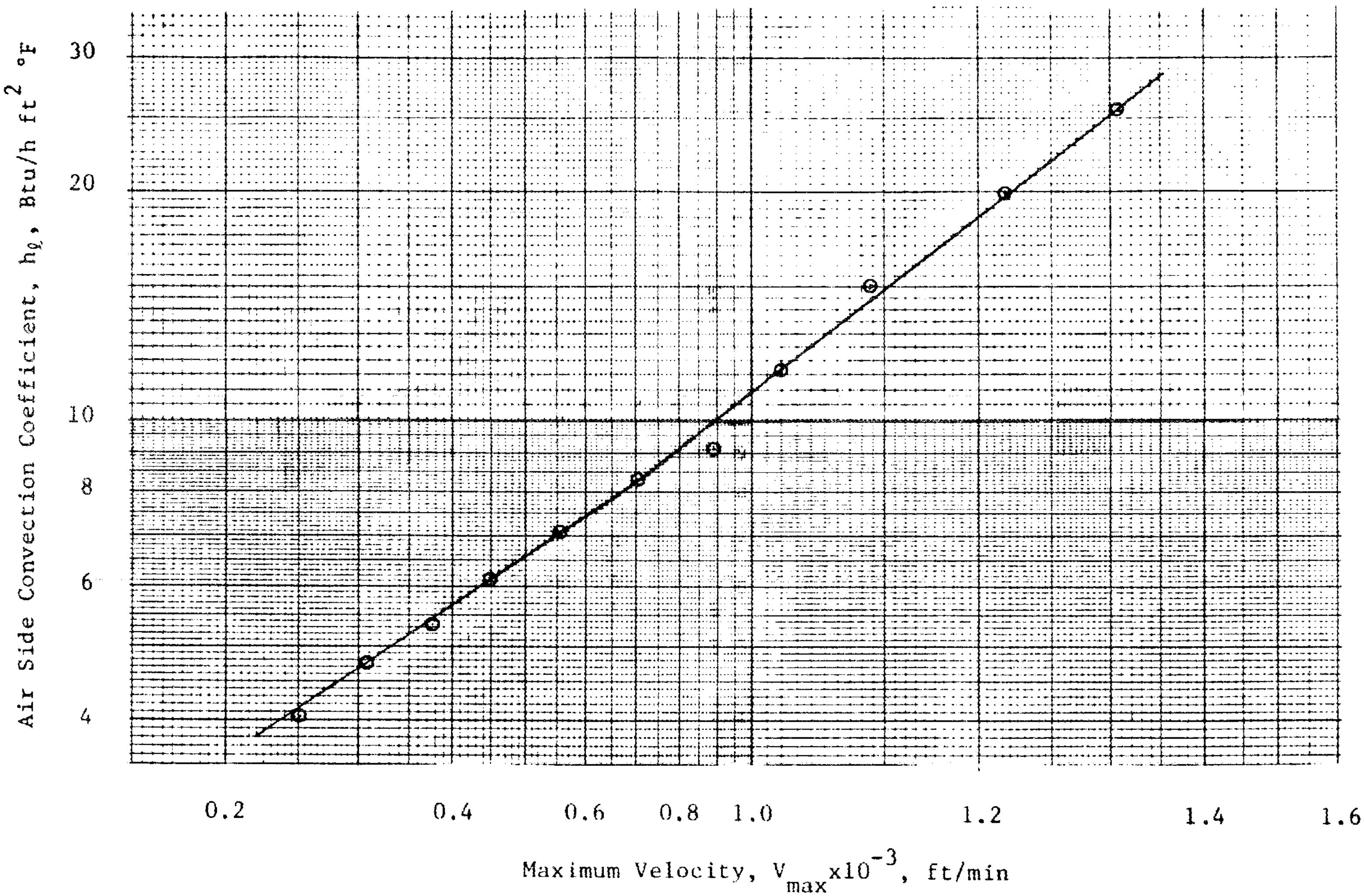


Figure F-21 - Air Side Convection Coefficient as a Function of Maximum Fin Channel Velocity for Flat Fins (∿6.67 fins/inch)

○ P_t = 1.250 inches, P_ℓ = 1.083 inches, N_R = 4, D_c = 0.525 inch, W = 0.144 inch (Rich's Data[2])

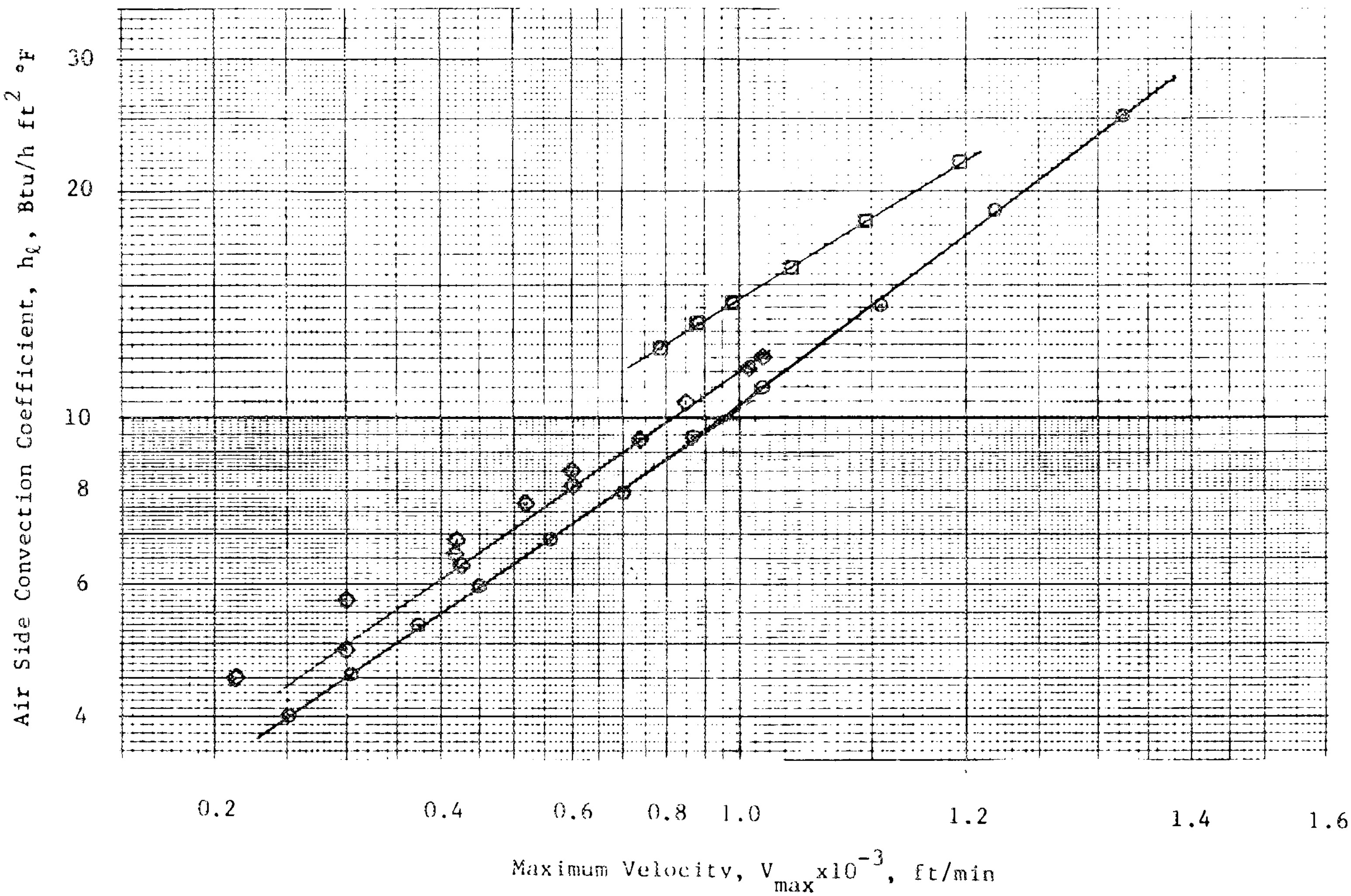


Figure F-22 - Air Side Convection Coefficient as a Function of Maximum Fin Channel Velocity for Flat Fins (∿7.67 fins/inch)

□ P_t = 0.800 inch, P_ℓ = 0.690 inch, N_R = 4, D_c = 0.407 inch, W_t = 0.119 inch (McQuiston's Data[3])

◇ P_t = 1.000 inch, P_ℓ = 0.866 inch, N_R = 4, D_c = 0.404 inch, W_t = 0.119 inch (McQuiston's Data[4])

○ P_t = 1.350 inches, P_ℓ = 1.083 inches, N_R = 4, D_c = 0.525 inch, W_t = 0.124 inch (Rich's Data[2])

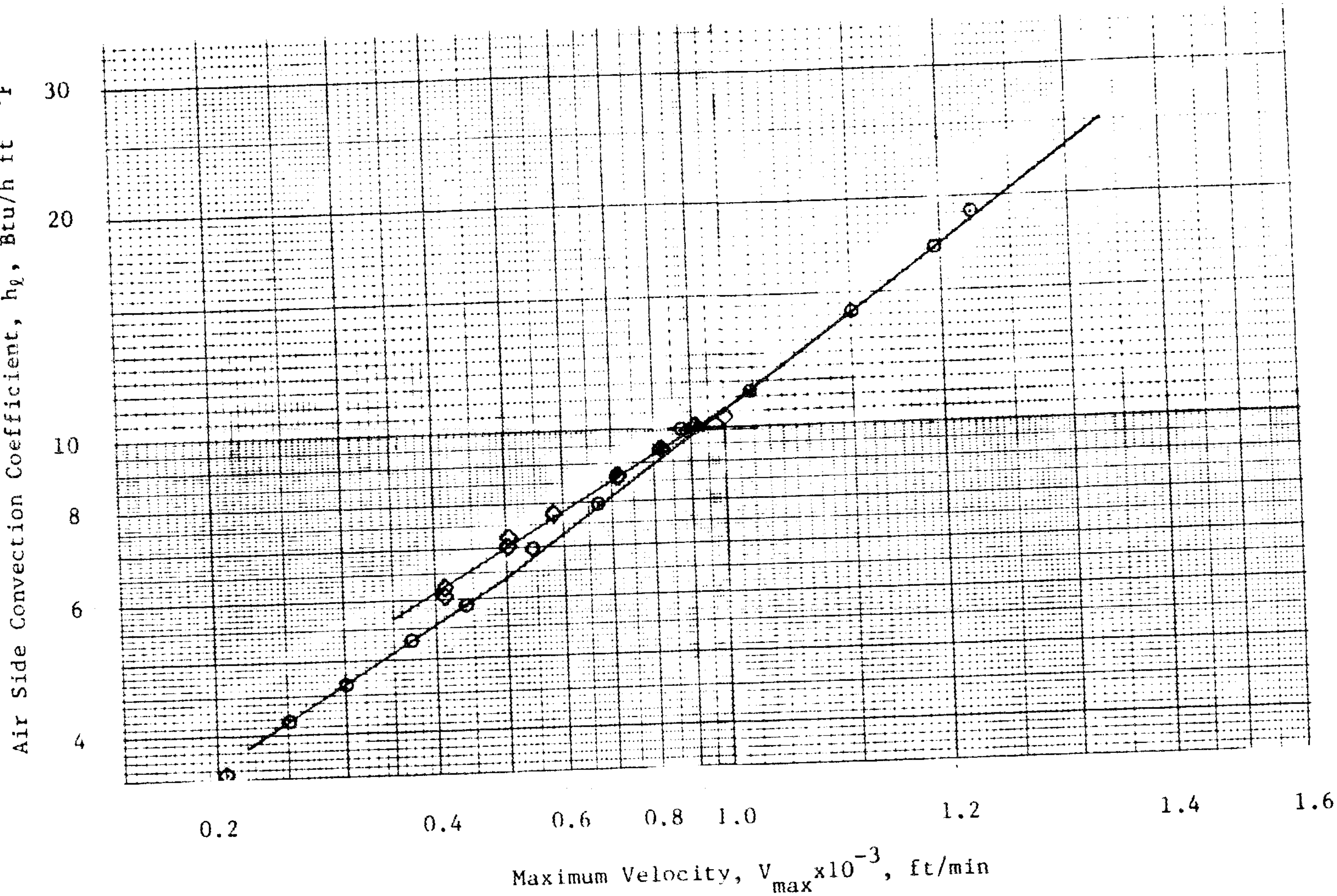


Figure F-23 - Air Side Convection Coefficient as a Function of Maximum Fin Channel Velocity for Flat Fins (∿9.17 fins/inch)

◇ P_t = 1.000 inch, P_ℓ = 0.866 inch, N_R = 4, D_c = 0.404 inch, W^t= 0.094 inch (McQuiston's Data[4])

○ P_t = 1.250 inches, P_ℓ = 1.083 inches, N_R = 4, D_c = 0.525 inch, W^t= 0.103 inch (Rich's Data[2])

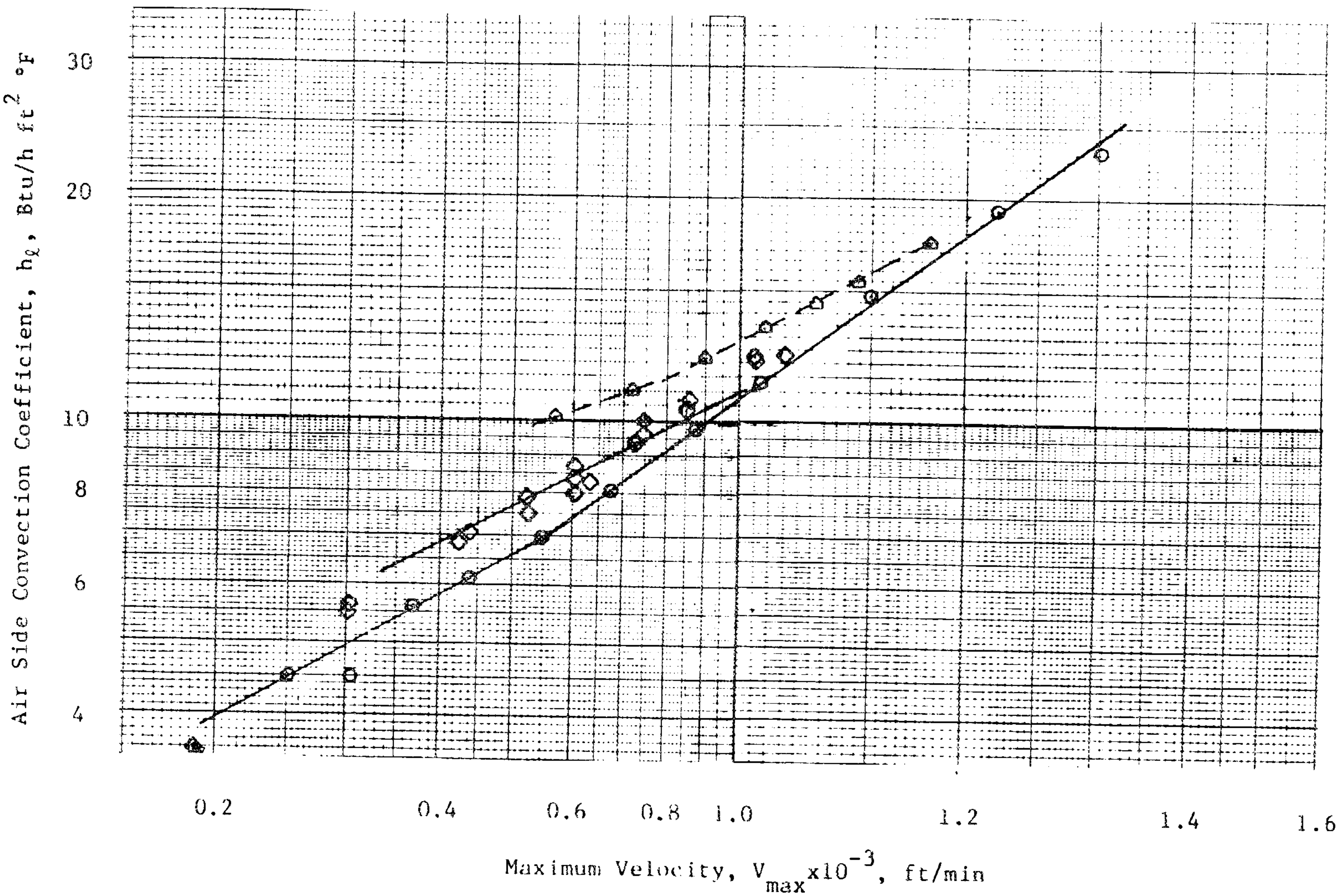


Figure F-24 - Air Side Convection Coefficient as a Function of Maximum Fin Channel Velocity for Flat Fins (∿11.7 fins/inch)

△ P_t = 1.250 inches, P_ℓ = 1.083 inches, N_R = 4, D_c = 0.500 inch, W = 0.077 (Beecher's Data[1])

◇ P_t = 1.000 inch, P_ℓ = 0.866 inch, N_R = 4, D_c = 0.404 inch, W = 0.077 (McQuiston's Data[4])

○ P_t = 1.250 inches, P_ℓ = 1.083 inch, N_R = 4, D_c = 0.525 inch, W = 0.0794 inch (Rich's Data[2])

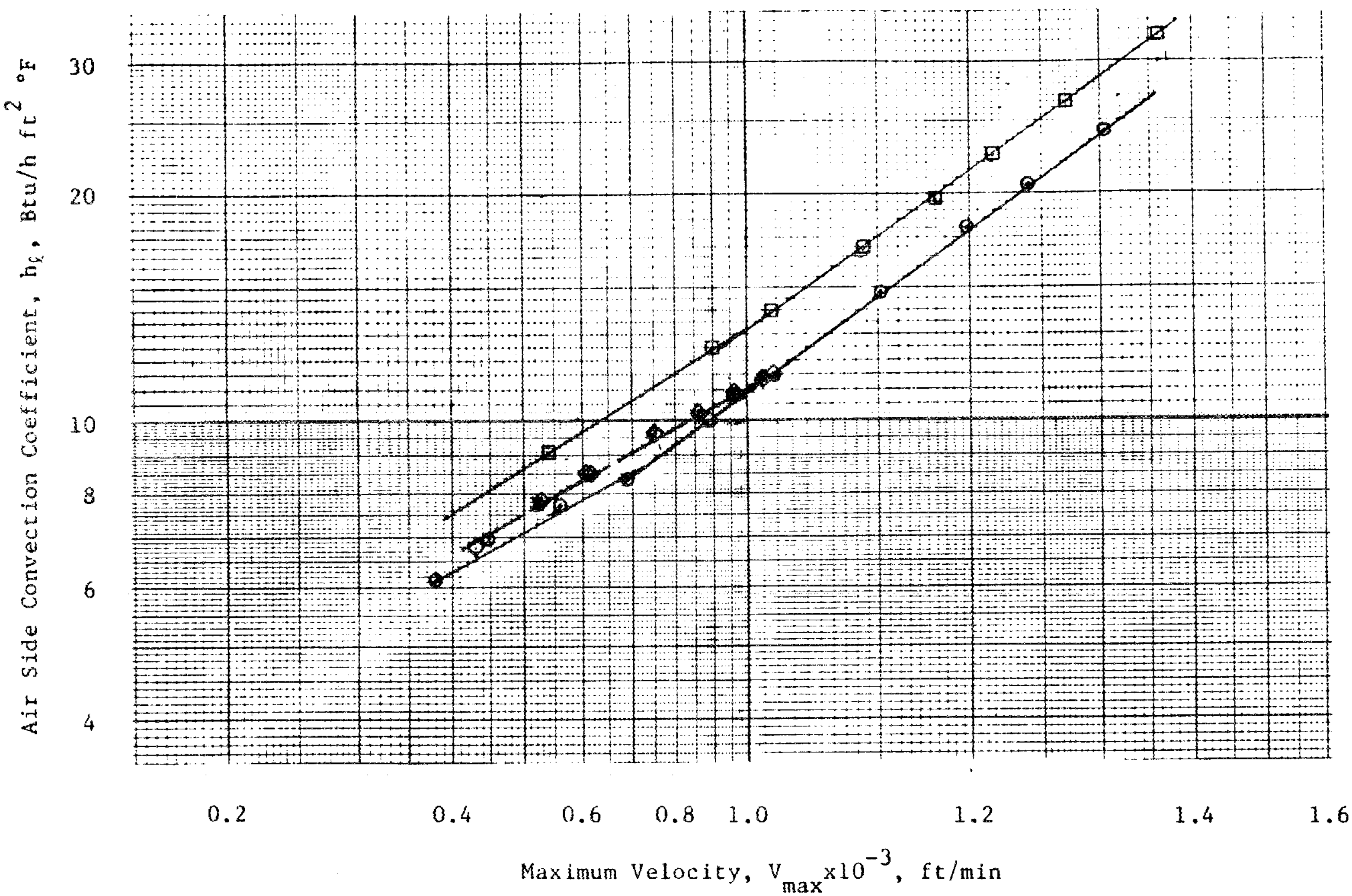


Figure F-25 - Air Side Convection Coefficient as a Function of Maximum Fin Channel Velocity for Flat Fins (∿14.5 fins/inch)

□ P_t = 0.800 inch, P_ℓ = 0.690 inch, N_R = 4, D_c = 0.407 inch, W_t = 0.064 inch (McQuiston's Data[3])

◇ P_t = 1.000 inch, P_ℓ = 0.866 inch, N_R = 4, D_c = 0.404 inch, W_t = 0.065 inch (McQuiston's Data[4])

○ P_t = 1.250 inches, P_ℓ = 1.083 inches, N_R = 4, D_c = 0.525 inch, W_t = 0.63 inch (Rich's Data[2])

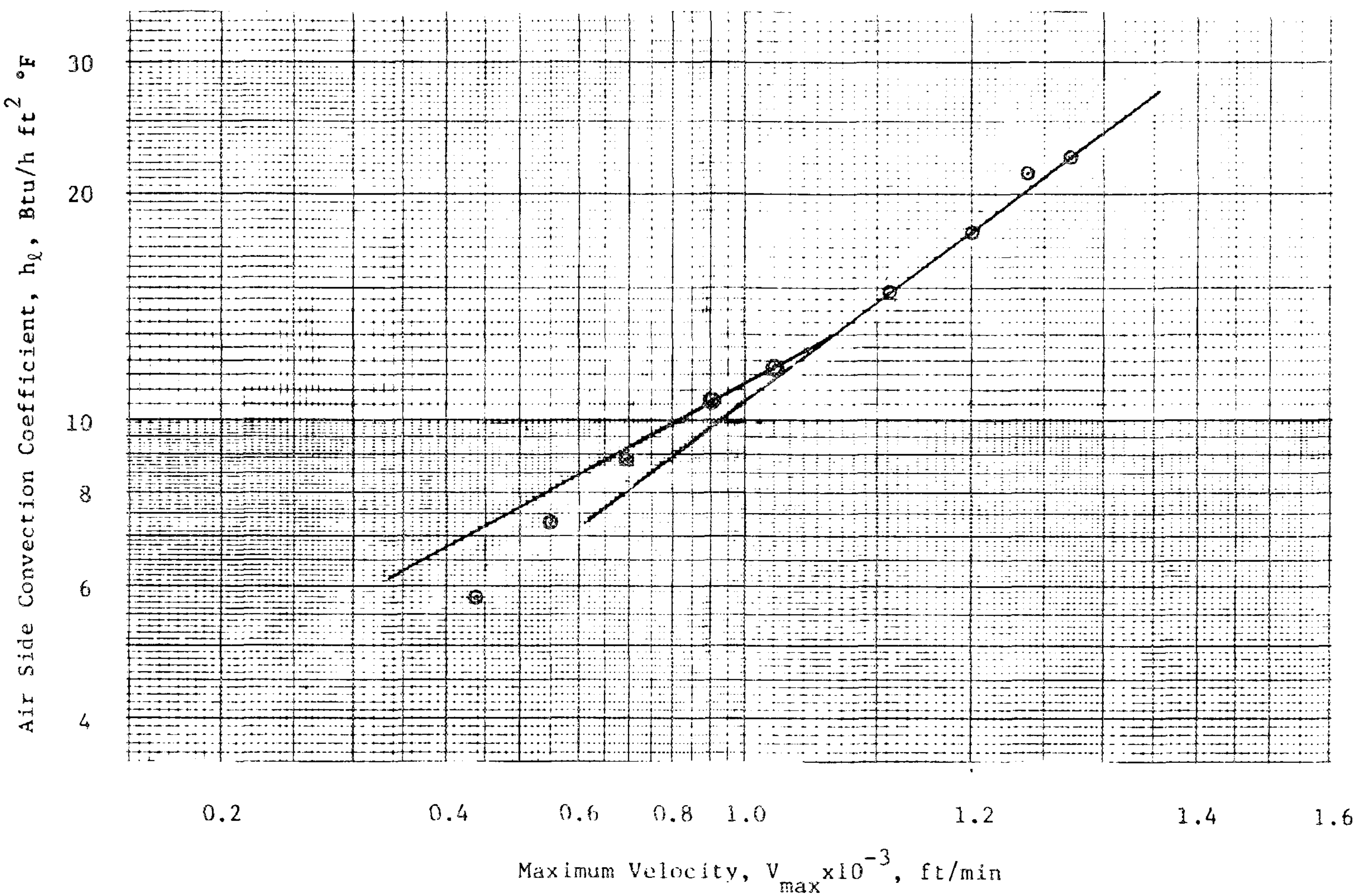


Figure F-26 - Air Side Convection Coefficient as a Function of Maximum Fin Channel Velocity for Flat Fins (20.6 fins/inch)

○ P_t = 1.250 inches, P_ℓ = 1.083 inches, N_R = 4, D_c = 0.525 inch, W = 0.0425 inch (Rich's Data[2])

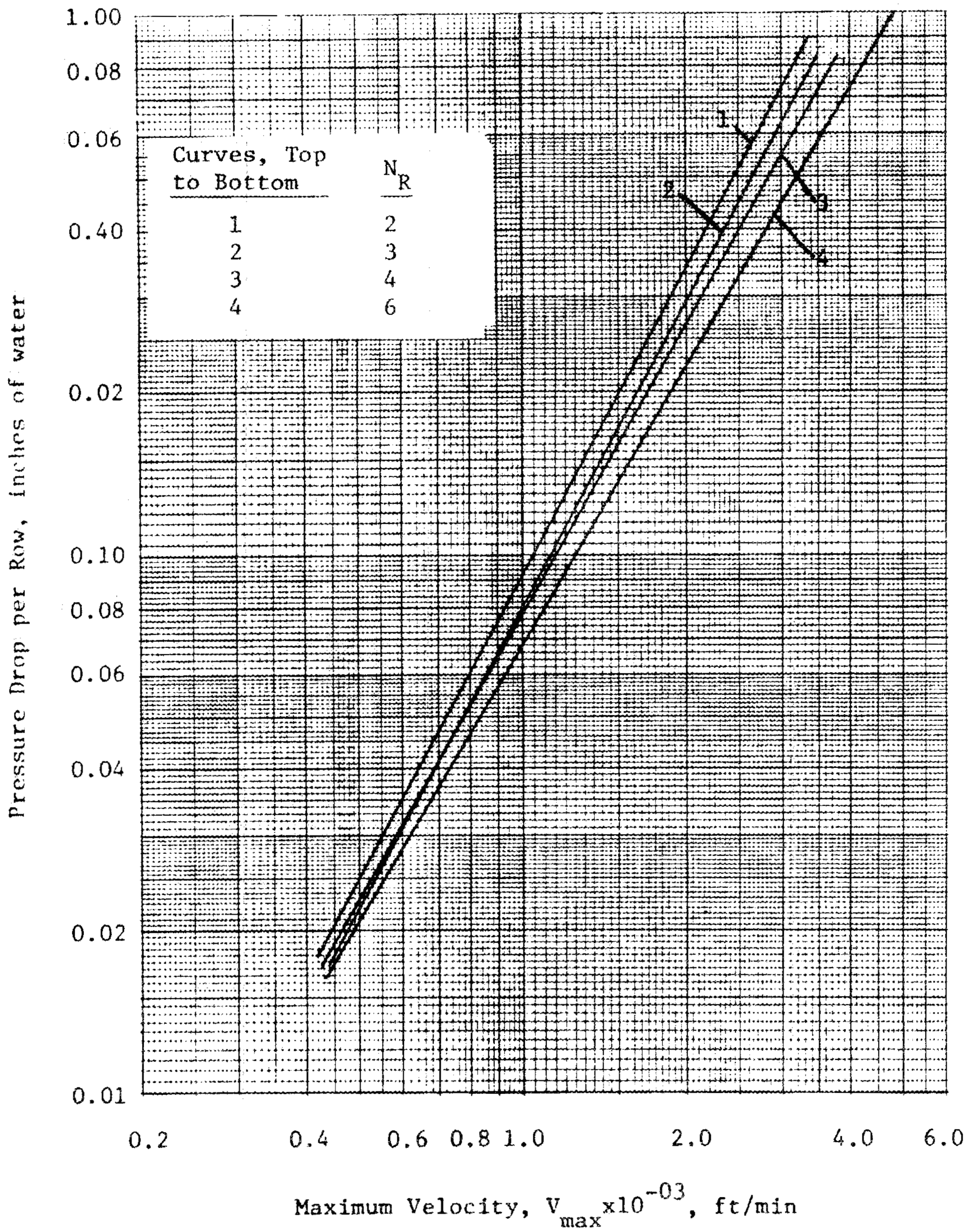


Figure F-27 - Effect of Number of Tube Rows, N_R, on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity for Flat Fins

P_t = 1.250 inches, P_ℓ = 1.083 inches, D_c = 0.500 inch, P_D = 0 inch, W = 0.077 inch

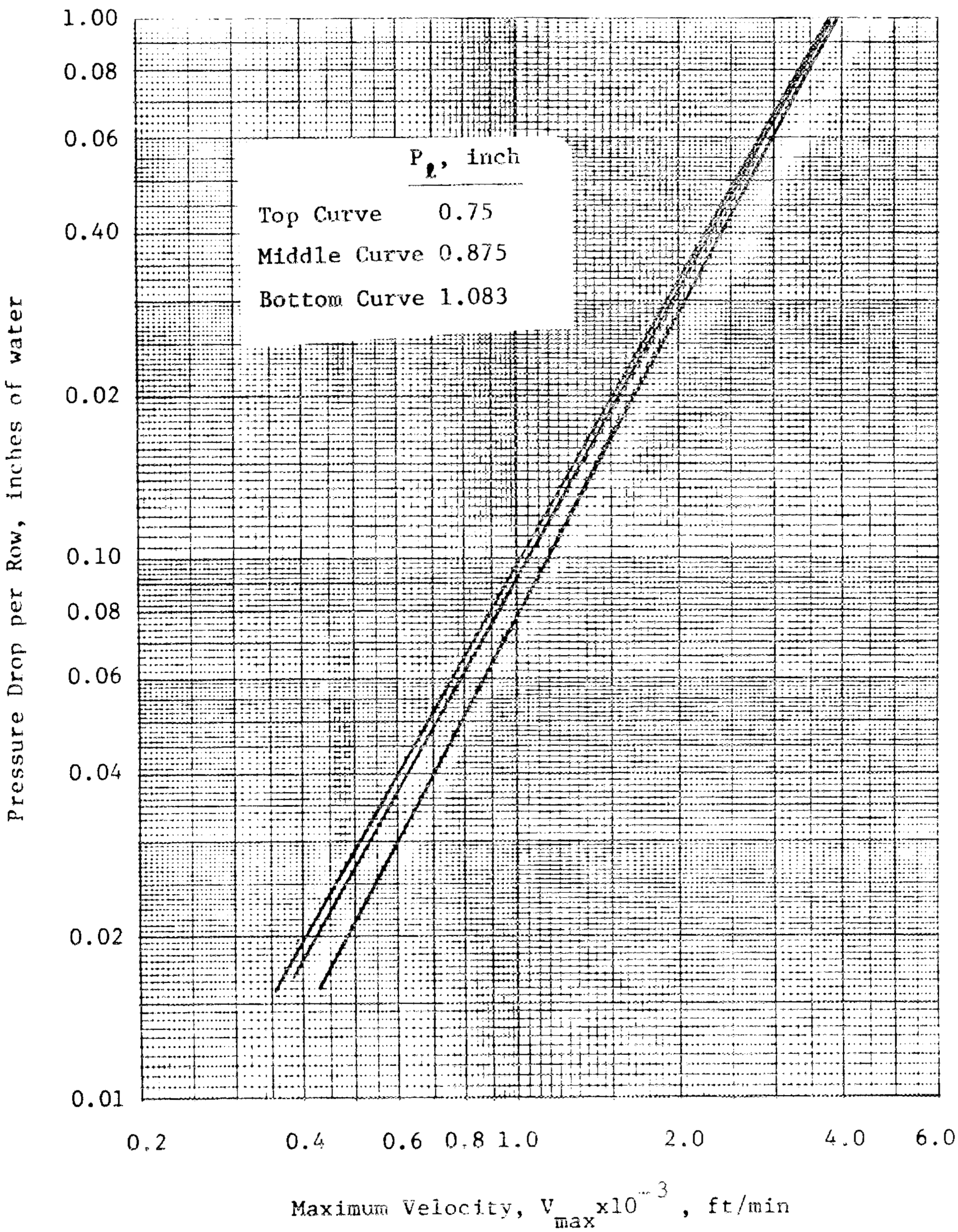


Figure F-28 - Effect of Longitudinal Pitch, P_ℓ, on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity for Flat Fins

P_t = 1.250 inches, D_c = 0.500 inch, W = 0.077 inch, N_R = 3, P_D = 0

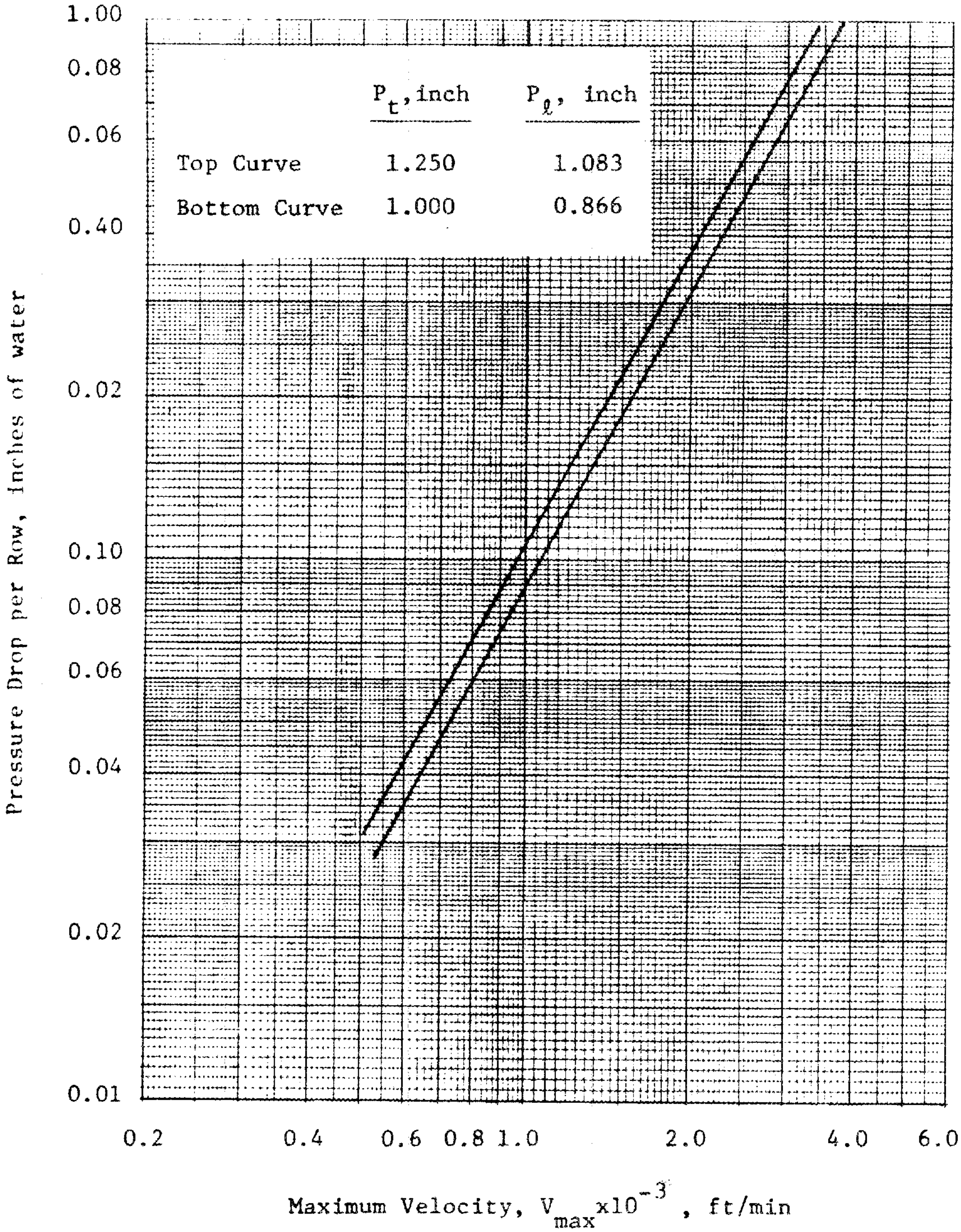


Figure F-29 - Effect of Tube Pitch (P_ℓ/P_t Constant) on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity

$N_R = 3$, $N_{ppp} = 3$, $W = 0.077$ inch, $D_c = 0.375$ inch, $P_D = 0.038$ inch

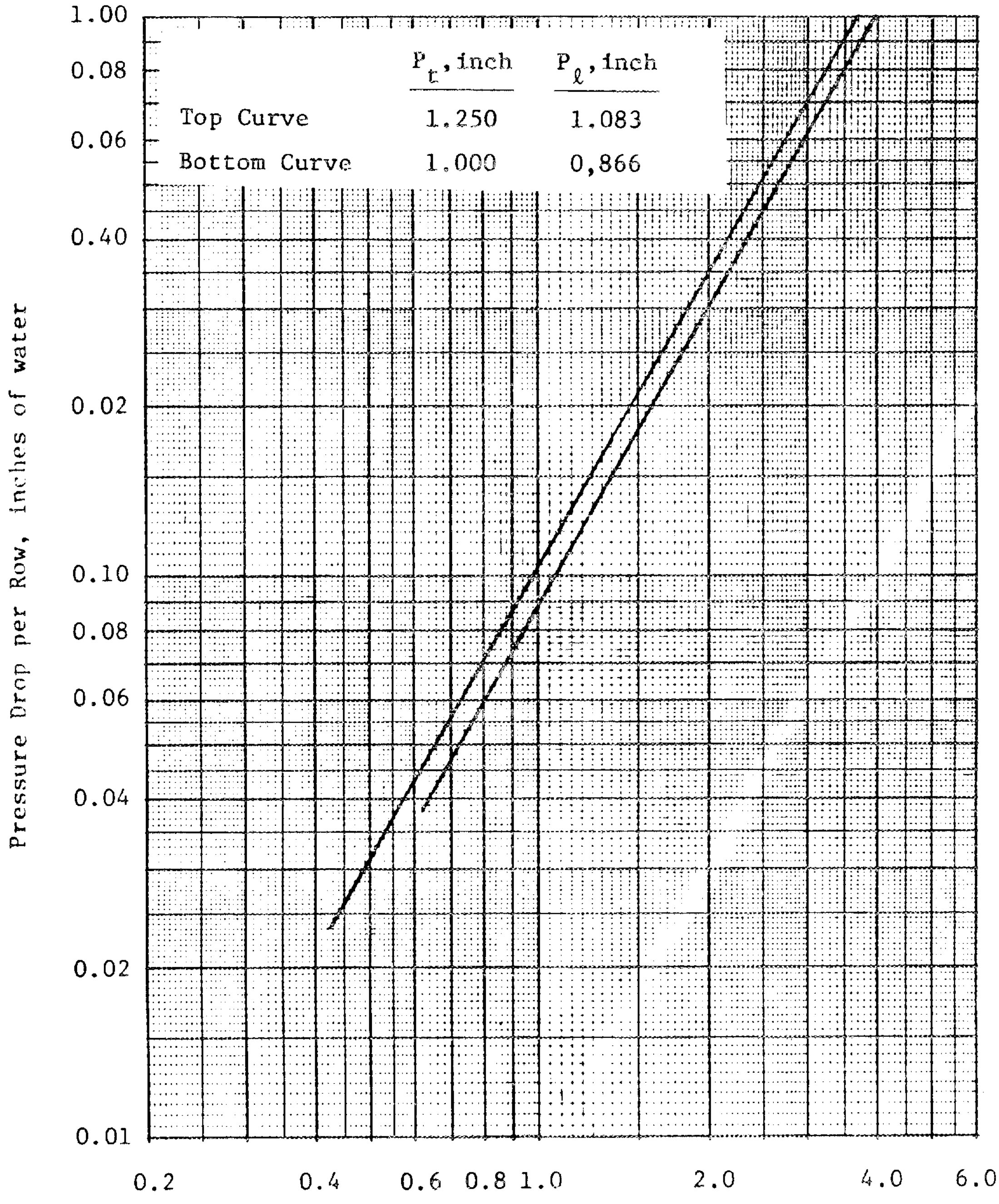


Figure F-30 - Effect of Tube Pitch (P_ℓ/P_t) on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity

N_R = 3, N_{ppp} = 3, W = 0.077 inch, D_c = 0.500 inch, P_D = 0.038 inch

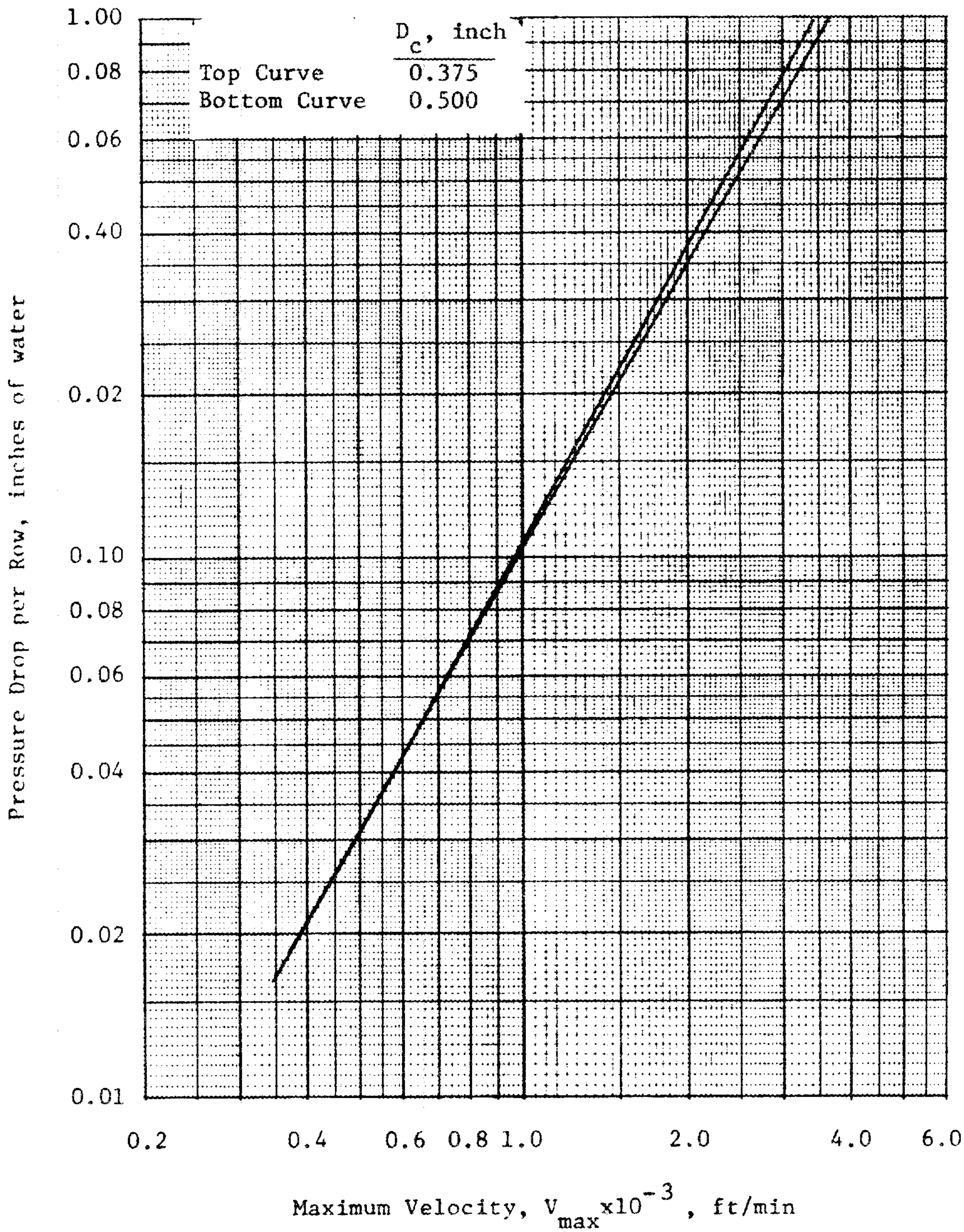


Figure F-31 - Effect of Fin Collar Diameter, D_c, on Pressure Drop per Row as a Function of Maximum Fin Channel Diameter

P_t = 1.250 inches, N_R = 3, P_D = 0.038 inch,
P_ℓ = 1.083 inches, N_{ppp} = 3, W = 0.077 inch

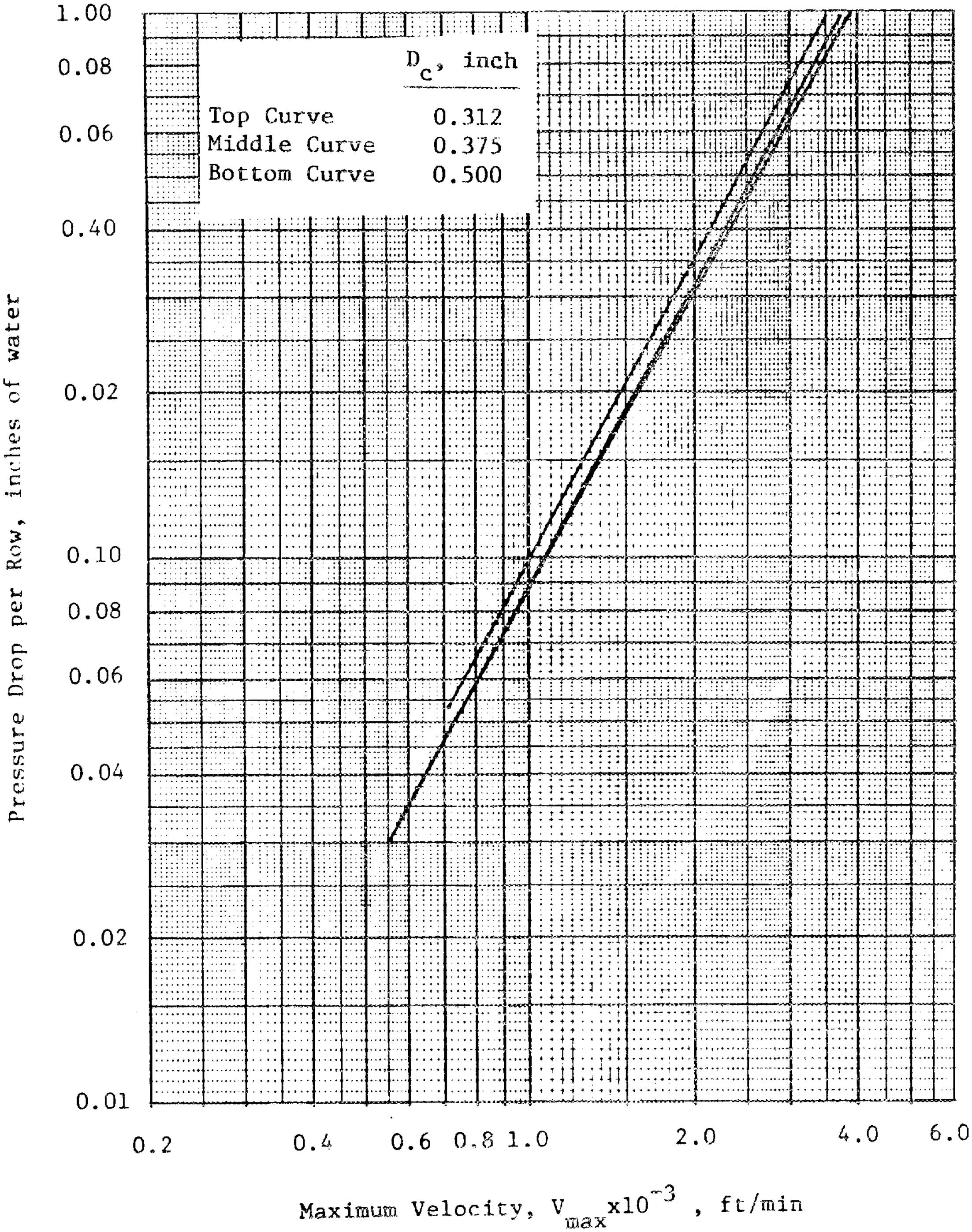


Figure F-32 - Effect of Fin Collar Diameter, D_c, on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity

P_t = 1.000 inch, N_R = 3, P_D = 0.038 inch,
P_ℓ = 0.866 inch, N_{ppp} = 3, W = 0.077 inch

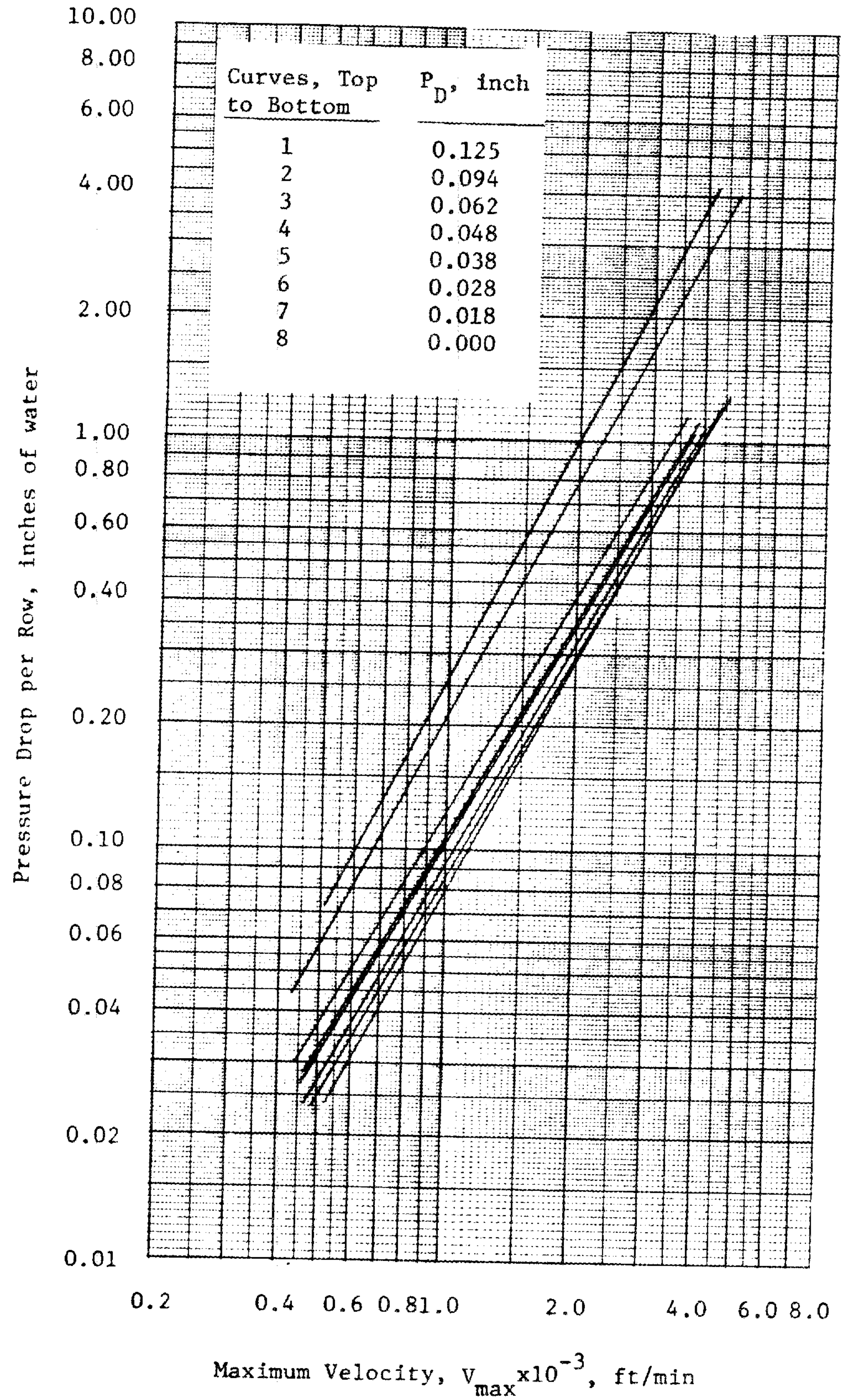


Figure F-33 - Effect of Pattern Depth, P_D, on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity
P_t = 1.250 inches, N_R = 3, N_{PPP} = 3,
P_ℓ = 1.083 inches, D_c = 0.500 inch,
W = 0.077 inch

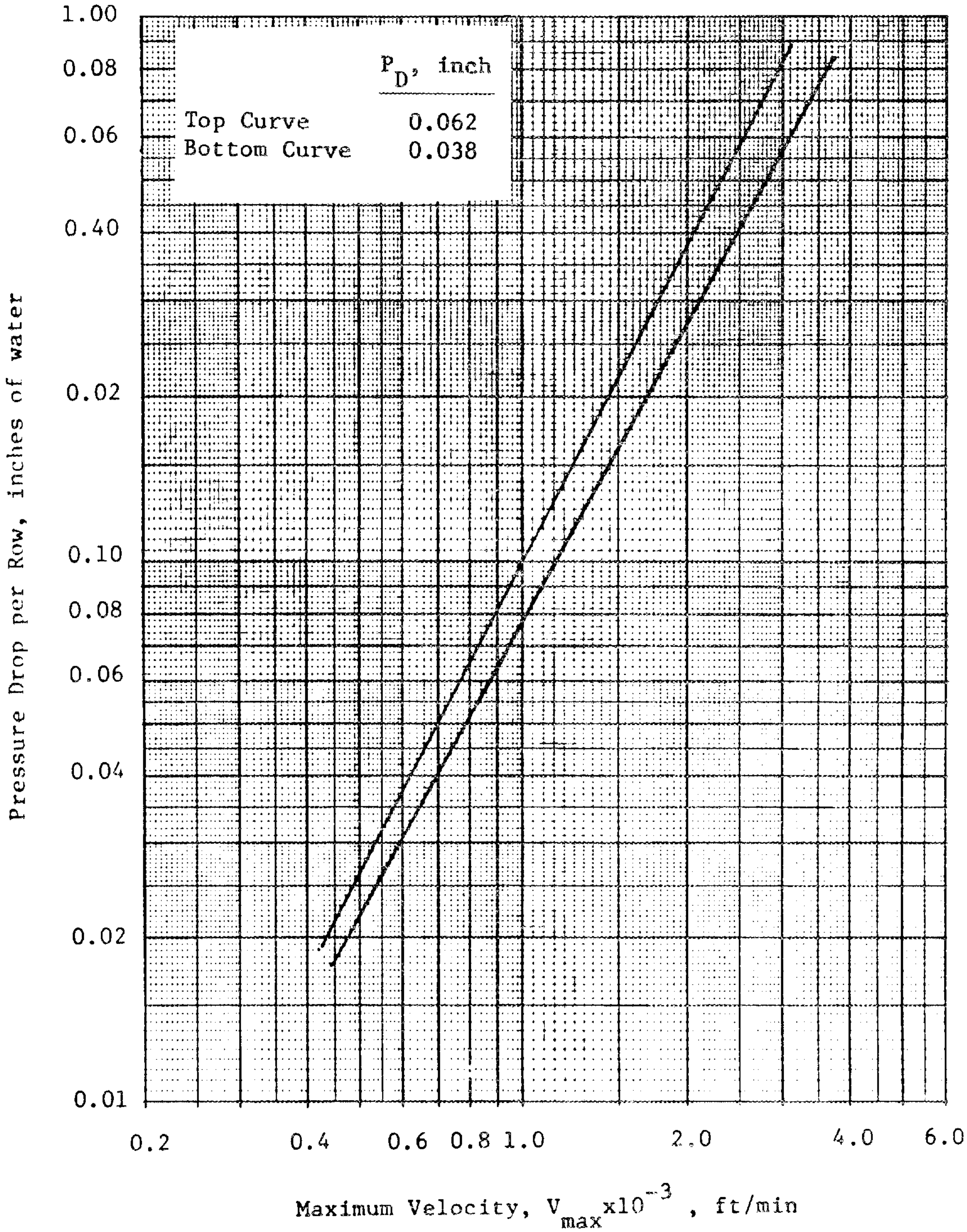


Figure F-34 - Effect of Pattern Depth, P_D, on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity

P_t = 1.250 inches, N_R = 3, N_{PPP} = 3
P_ℓ = 1.083 inches, D_c = 0.500 inch,
W = 0.110 inch

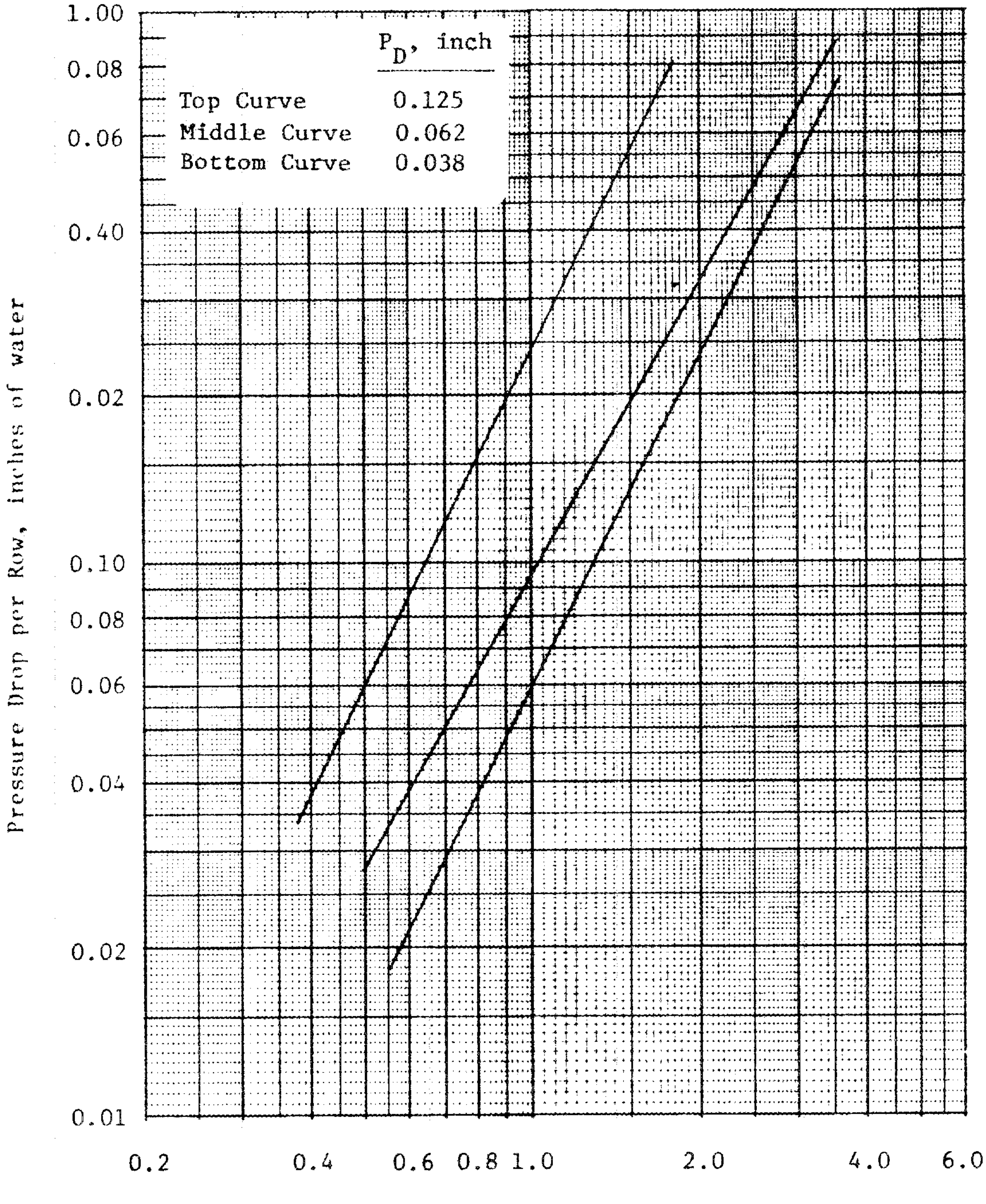


Figure F-35 - Effect of Pattern Depth, P_D, on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity

P_t = 1.250 inch, N_R = 3, N_{PPR} = 3
P_ℓ = 1.083 inches, D_c = 0.500 inch,
W = 0.161 inch

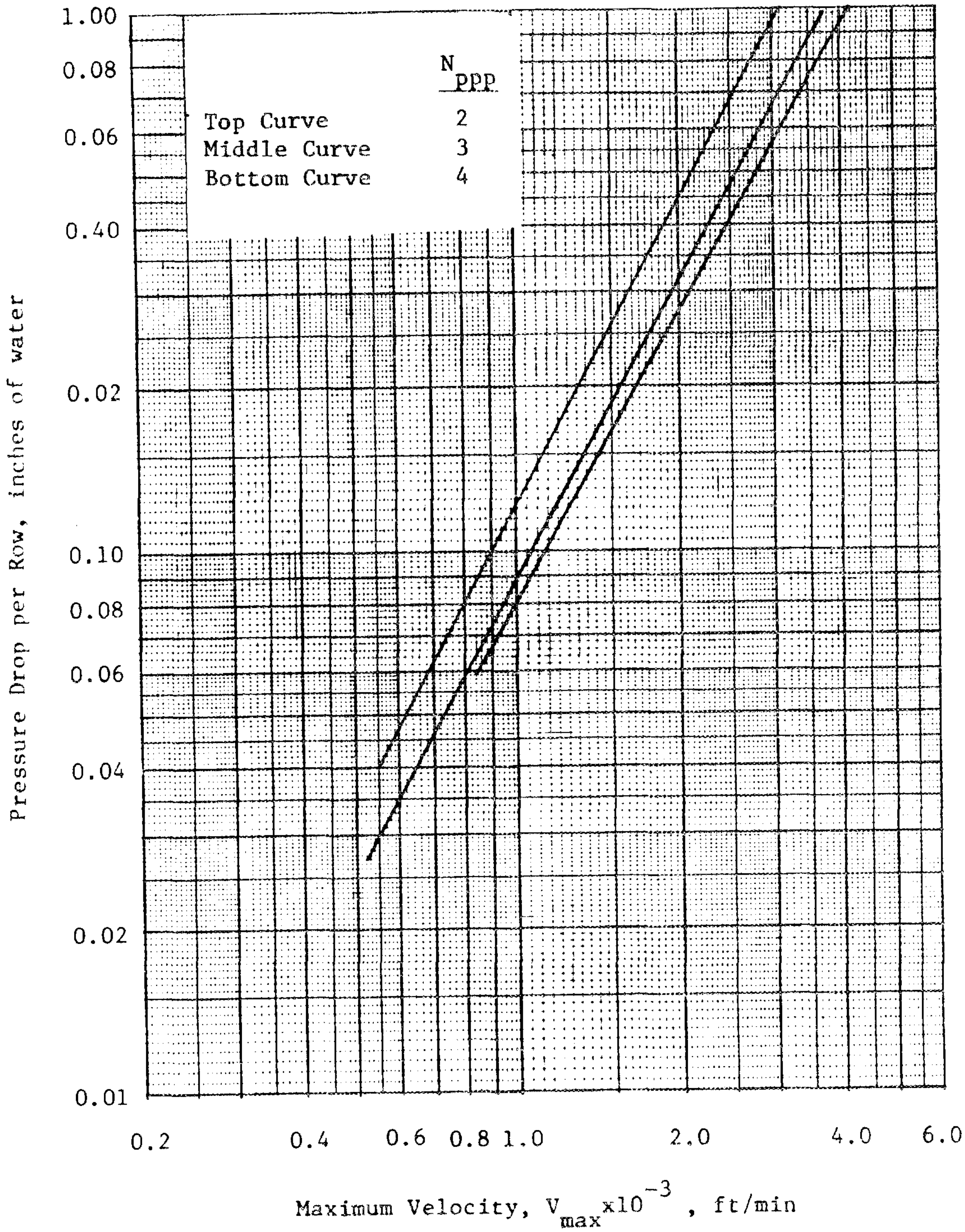
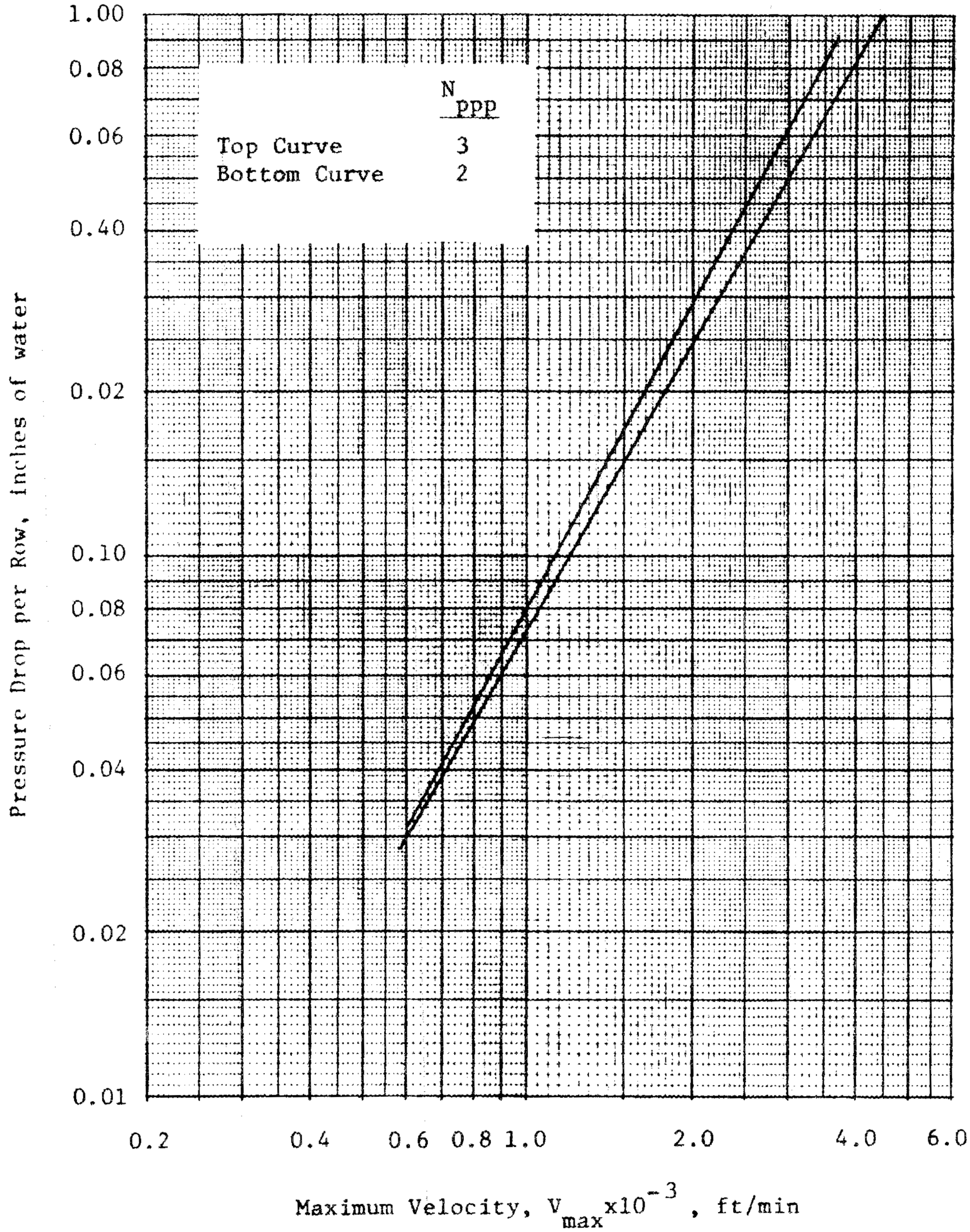


Figure F-36 - Effect of Number of Patterns per Pitch, N_{ppp}, on Pressure Dropper Row as a Function of Maximum Fin Channel Velocity

P_t = 1.000 inch, N_R = 3, P_D = 0.038 inch,
P^t = 0.866 inch, D_c = 0.375 inch,
W = 0.077 inch



Fifure F-37 - Effect of Number of Patterns per Pitch, N_{ppp}, on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity

P_t = 1.000 inch, N_R = 3, P_D = 0.038 inch, P^t = 0.866 inch, D_c = 0.375 inch, W = 0.094 inch

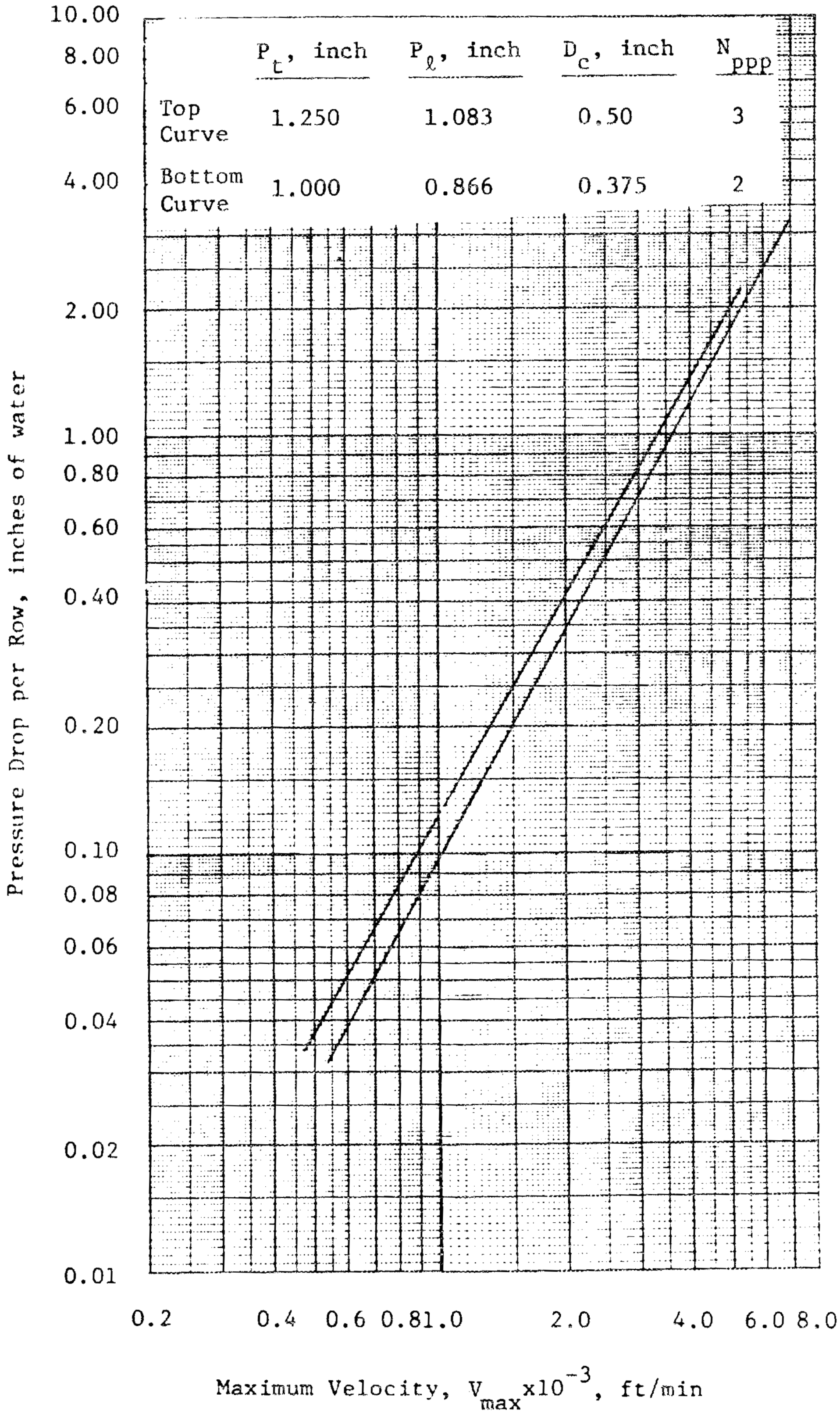


Figure F-38 - Effect of Number of Fin Patterns per Pitch, N_{ppp}, on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity

$N_R = 3$, $P_D = 0.062$ inch, $W = 0.077$ inch

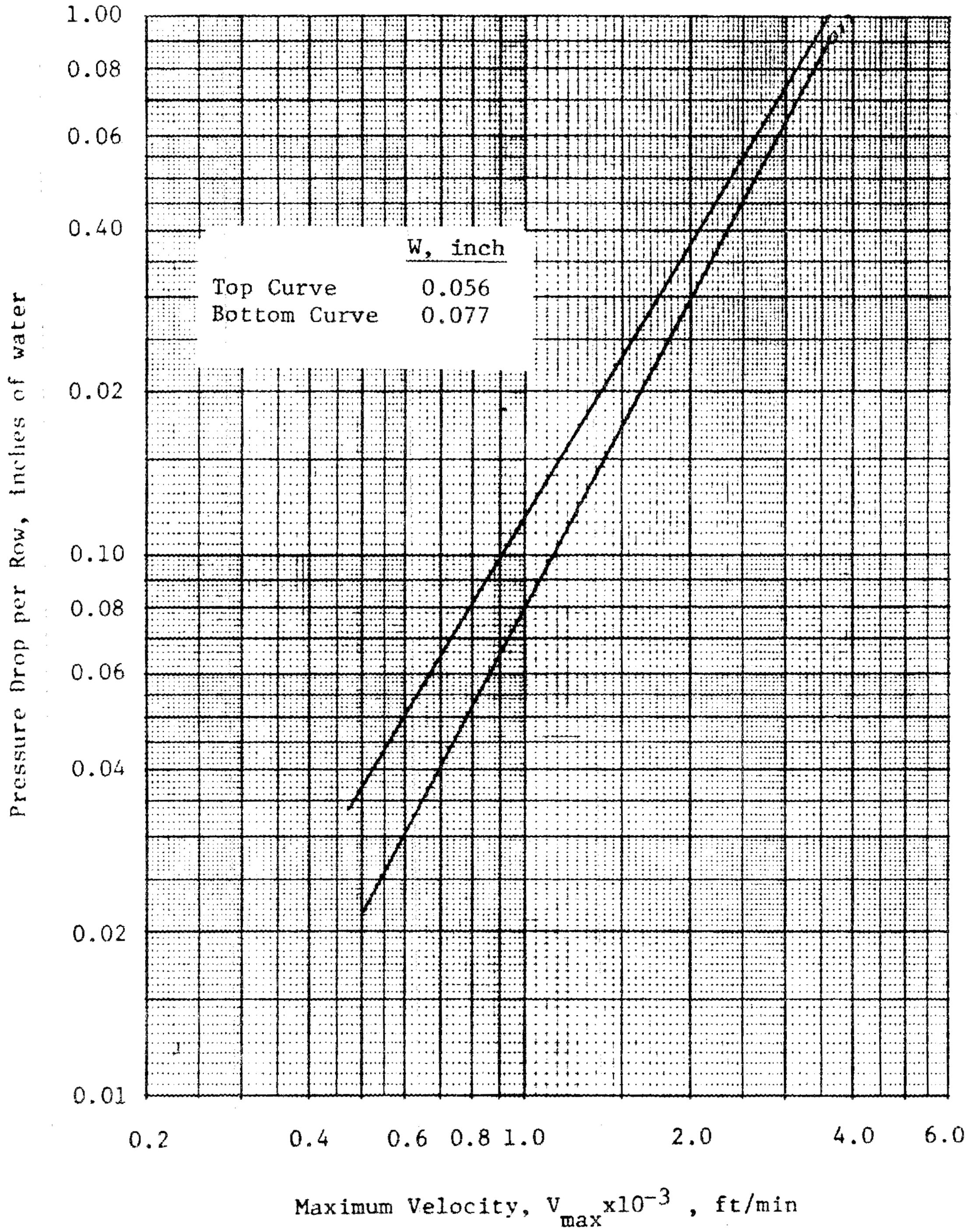


Figure F-39 - Effect of Fin Spacing, W, on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity for Flat Fins

P_t = 1.250 inch, N_R = 3, P_D = 0, P_ℓ = 1.083 inch, D_c = 0.500 inch

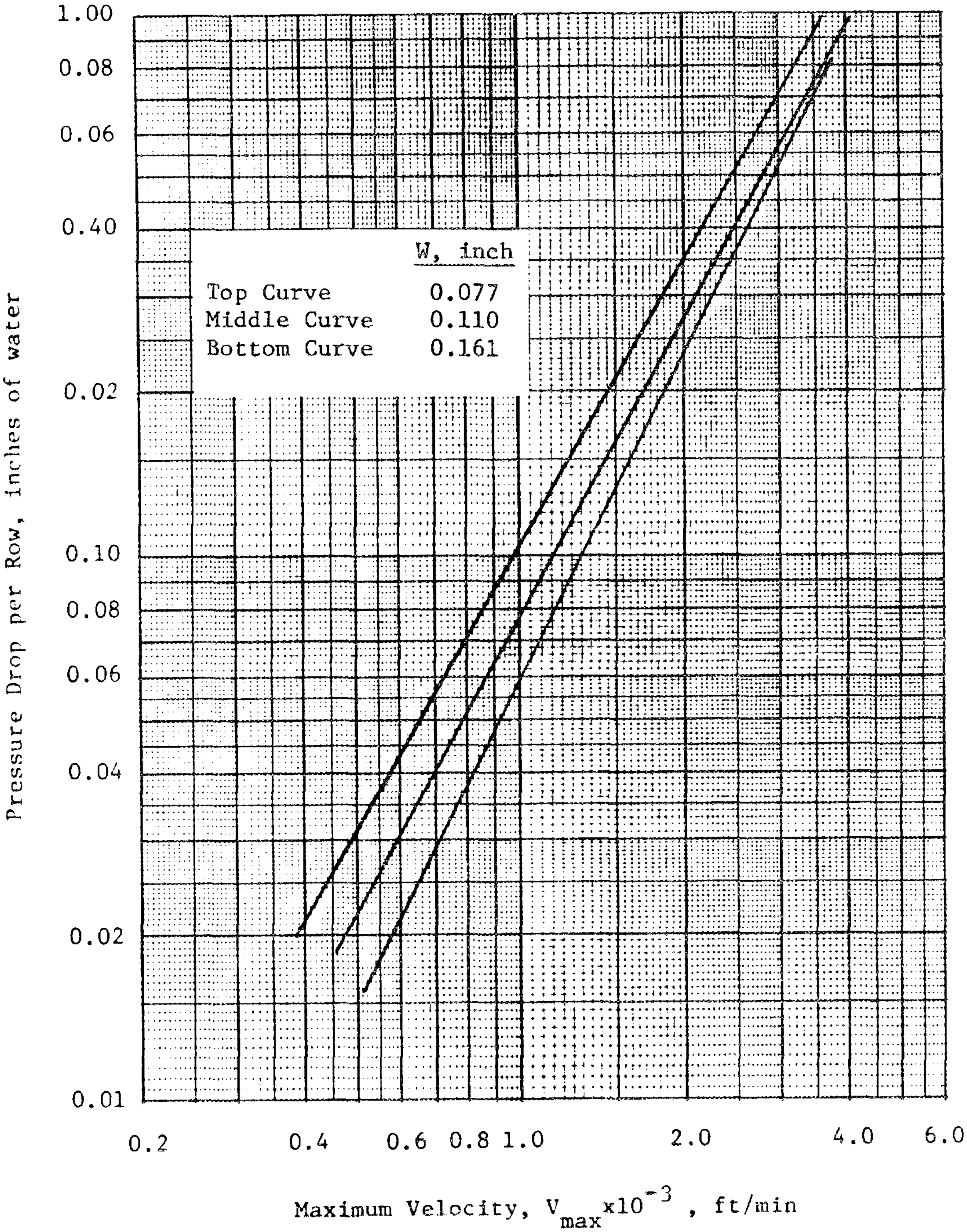


Figure F-40 - Effect of Fin Spacing, W, on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity

P_t = 1.250 inch, N_R = 3, N_{ppp} = 3,
P_ℓ = 1.083 inch, D_c = 0.500 inch, P_D = 0.038 inch

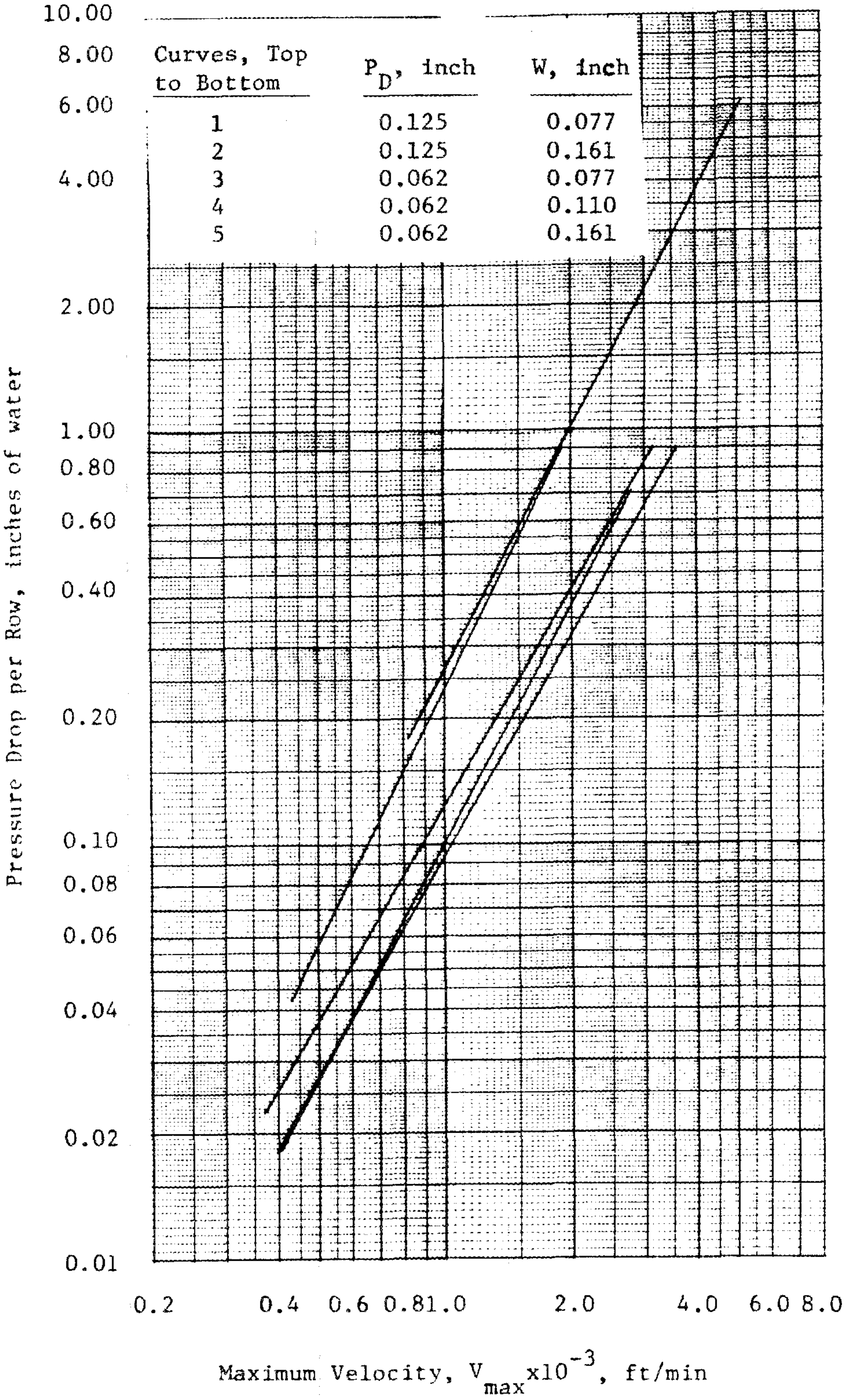


Figure F-41 - Effect of Fin Spacing, W, on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity for Highly Patterned Fins

P_t = 1.25 inch, N_R = 3, N_{PPP} = 3
P_ℓ = 1.083 inch, D_c = 0.500 inch

$$\text{where } \sec\theta = \frac{\sqrt{x^2 + P_D^2}}{x}$$

$$x = \frac{P_\ell}{2N_{ppp}} .$$

Only the high velocity portion of the data where the slope is nearly constant has been plotted. Extension of these lines to lower velocities will give a pressure drop per row which will be too low. The magnitude of the variation will increase as the number of fins per inch (N_f) increases (W decreases).

Figure F-27 indicates that the pressure drop per row decreases as the number of rows increases for a 1.25 x 1.083, 0.5 in. heat exchanger with flat fins spaced 0.077 in. apart. Most manufacturers data give $\Delta p/N_R$ as a constant. The slope of the lines increases from 1.71 to 1.87 as the number of rows decreases. The lines approach each other at low velocities justifying the assumption that Δp is independent of number of rows. Unfortunately no calibration run was made to measure the pressure difference between the inlet and outlet plenums with no heat exchanger present. Since the pressure difference was assumed to be zero, all pressure differences measured will be high by this amount. Dividing a fixed error at any given flow rate by a larger number of rows therefore gives a lower $\Delta p/N_R$. The data suggest an error of 0.004 in. of water at V_{max} = 500 ft/min. All of the remaining data was for a three row heat exchanger.

Figure F-28 indicates that decreasing the longitudinal pitch will only moderately increase the $\Delta p/N_R$ for flat fin heat exchangers and that at very high velocities the effect decreases. The pressure drop associated with flow across the tubes is mostly form drag and is proportional to velocity squared. The pressure drop associated with friction will vary as velocity to a power between 1 and 1.8. The lower slope for lines corresponding to lower values of P_ℓ in Figure F-28 indicate that the higher local velocities across a portion of the fin surface due to lower P_ℓ is resulting in a higher percentage loss due to friction.

Figures F-29 and F-30 each compare $\Delta p/N_R$ for two equilateral tube pitches 1.25 and 1.0 in for 0.375 and 0.5 in. diameter fin collars, respectively. In Figure F-29 the line representing the P_t ~ 1.25 in. data has an ordinate ~1.185 to 1.2 (low to high velocity) times the line representing the P_t ~ 1.0 in. data. A similar comparison in Figure F-30 is 1.19 to 1.13. The spacing of lines is nearly constant in Figure F-29 (D_c = 0.375 in) and slightly greater than in Figure F-30

(D_c = 0.5 in.). These comparisons are for the T3.038 fin pattern tested and a greater variation might be expected for flat fins.

Figures F-31 and F-32 demonstrate the value of V_{max} to correlate the pressure drop for a given transverse pitch for various fin collar diameters. The D_c = 0.5 and 0.375 in. data falls on top of the other at V_{max} less than 800. At higher velocities the correlation overcorrects and the pressure drops for smaller diameter fin collars (less blockage) are higher than those for larger collars at the same value of V_{max}.

Figure F-33, demonstrates the effect of pattern depth on $\Delta p/N_R$ for a 1.25 x 1.083, 0.5 in. three row heat exchanger with a fin spacing of 0.077 in. At this fin spacing the slope of the lines for P_D < 0.048 in. decrease with increasing P_D in a regular manner indicating a larger percentage friction as pattern depth increases. As the pattern depth becomes larger, however, this trend reverses and both the $\Delta p/N_R$ and the slope of the $\Delta p/N_R$ vs. V_{max} lines increase rapidly indicating larger regions of separation behind each pattern ridge. This separation is apparently minimized by the close fin spacings at the lower pattern depths.

Figures F-34 and F-35 show the pressure drop increases more rapidly with pattern depth for larger fin spacing. The slope of the data presented for the T3.062 model in Figure F-35 is less than expected.

The effect of number of patterns per pitch (N_{ppp}) is shown in Figures F-36, F-37 and 38. Figure F-36 has the only data taken for four patterns per pitch and refers to 1.0 x 0.866, 0.375 in. models at a fin spacing of 0.077 in. and 2, 3, and 4 patterns per longitudinal pitch. The slope of the lines increases with number of patterns per pitch. At a V_{max} of 1000 ft per min. the two pattern per pitch model has a $\Delta p/N_R$ of 0.92 times the $\Delta p/N_R$ for that of the three pattern per pitch model. The four pattern per pitch model has a $\Delta p/N_R$, 1.375 times the $\Delta p/N_R$ for three patterns/pitch model. Figure F-37 shows the data for 2 and 3 pattern per pitch models at a fin spacing of 0.094 in. The wider fin spacings reduces the pressure drop about 11 percent. Figure F-38 shows data for a 1.25 x 1.083, 0.5 in. and a 1.0 x 0.866, 0.375 in. model both with a 0.077 in. fin spacing and a 0.062 in. deep pattern. The two pattern per pitch pressure drop is 0.79 times the pressure drop for the three pattern per pitch model at a maximum velocity of 1000 ft per min. The correction for tube pitch with the T3.038 pattern data was 1.19 (see Figures F-29 and F-30) which times 0.79 is a ratio 0.92, the number obtained for this fin spacing in Figure F-36 for the 0.038 pattern depth.

Figures F-39, F-40 and F-41 show that the pressure drop is increased as the fin spacing is decreasing. Figure F-39 is the only fin spacing comparison for flat fins. Figures F-40 and F-41 show that as pattern depth increases from 0.038 to 0.125 inch, the pressure drop becomes less sensitive to the number of fins per inch. The slope of the P_D = 0.062 inch data in Figure F-41 is higher than that of either the 0.038 or 0.161 inch data indicating a probable entry plenum alignment problem during that test.

DATA FITTING

NUSSELT NUMBERS

Row Effect - Flat Fins $P_D = 0$

Data was read from the Nu_a vs. Gz curves in Figure F-3 at Gz numbers of 30, 40 and 60, and the ratio of $Nu_{a_{NR}}$ for a model of N_R rows to Nu_{a_3} for the three row model plotted against $N_R/3$. Lines drawn through the 6 row data and the 3 row data approximated the 2 row data with a maximum error of 4.5 percent at Gz = 60. The four row data show maximum error of 5.7 percent. The slopes of these lines at Gz of 30, 40 and 60 on a log-log plot against Gz gave the following relationship for flat fins

$$\frac{Nu_{a_{NR}}}{Nu_{a_3}} = \left(\frac{N_R}{3}\right)^{\left(0.22\left(\frac{Gz}{30}\right)^{0.626}\right)} \qquad 2 \le N_R \le 6 \tag{F-5}$$

A similar treatment of the data in Figure F-4 for flat fins atlongitudinal pitch to tangential pitch ratios (P_ℓ/P_t of 0.866, 0.7 and 0.6 result in Equation F6.

$$\frac{Nu_{a\left(\frac{P_\ell}{P_t}\right)}}{Nu_{a_{0.866}}} = 1.0332\left(\frac{P_\ell}{P_t}\right)^{\left(0.227\left(\frac{Gz}{30}\right)^{0.163}\right)} \tag{F-6}$$

Pattern depth effects for W = 0.077 inch ($N_f \sim 12$ fins/in.) can be approximated by fitting the data in Figure F-9. A log-log plot of Nu_a versus pattern depth (P_D) was made at Gz = 7, 15, 30 and 80. Straight lines approximated the data very closely at each Gz number. The fits are given as Equations F-7.

$$Nu_a = 3.85\, P_D^{0.046} \qquad Gz = 7 \tag{F-7a}$$

$$Nu_a = 9.30\ P_d^{0.107} \qquad Gz = 15 \qquad \text{(F-7b)}$$

$$Nu_a = 20.98\ P_d^{0.182} \qquad Gz = 30 \qquad \text{(F-7c)}$$

$$Nu_a = 64.6\ P_D^{0.293} \qquad Gz = 80 \qquad \text{(F-7d)}$$

The log-log curve of Nu_{a_o} vs. Gz for a flat fin has been approximated by three straight lines given by Equations F-8.

$$Nu_{a_o} = 4.17\ (Gz/10)^{0.822} \qquad 7 < Gz < 15 \qquad \text{(F-8a)}$$

$$Nu_{a_o} = 4.34\ (Gz/10)^{0.725} \qquad 15 \leq Gz < 30 \qquad \text{(F-8b)}$$

$$Nu_{a_o} = 4.64\ (Gz/10)^{0.664} \qquad 30 < Gz < 100 \qquad \text{(F-8c)}$$

To calculate the ratio for $Nu_{a_{(P_D)}}$ to Nu_{a_o} at a given Gz, the following procedure is suggested. Suppose the ratio for an T3.038 pattern to a flat plate is required at a Gz of 26. For $P_D = 0.038$ calculate $Nu_{a_{0.038}}$ at Gz = 15 and 30 using Eq. F-7b and F-7c, respectively.

$$Nu_{a_{0.038}} = 9.3\ (0.038)^{0.107} = 6.55 \text{ at } Gz = 15$$

$$Nu_{a_{0.038}} = 20.98\ (0.038)^{0.182} = 11.57 \text{ at } Gz = 30$$

The slope of a line through these data on a log-log plot of $Nu_{a_{0.038}}$ vs. Gz would be $\dfrac{\ln\left(\frac{11.57}{6.55}\right)}{\ln\left(\frac{30}{15}\right)} = 0.821$ with an intercept at Gz = 10 of 4.695.

Equation F-9 then represents that section of the curve $15 < Gz < 30$.

$$Nu_{a_{0.038}} = 4.695 \left(\frac{Gz}{10}\right)^{0.821} \tag{F-9}$$

and at Gz = 26, $Nu_{a_{0.038}} = 10.29$

From Equation F-8b, $Nu_{a_0} = 8.68$ at Gz = 26.

Using Equation F-3 the ratio of air side coefficients based on LMTD is calcualted to be 1.33.

$$\frac{h_{a_{\ell_{0.038}}}}{h_{a_{\ell_0}}} = \frac{\ln\left\{\dfrac{1+2\left(\frac{10.29}{26}\right)}{1-2\left(\frac{10.29}{26}\right)}\right\}}{\ln\left\{\dfrac{1+2\left(\frac{8.68}{26}\right)}{1-2\left(\frac{8.68}{26}\right)}\right\}} = \frac{2.151}{1.613} = 1.33$$

The limited data of log (Nu_a) for the T2.038, T3.038 and T4.038 models were plotted against log (N_{ppp}) at Gz of 20, 30, 40, 60 and 80. The curves were straight lines at Gz>40 and were approximated by straight lines at the lower Gz numbers. The slope of these lines were plotted against Gz and Equation F-10 resulted

$$\frac{Na_{a_{ppp}}}{Nu_{a_3}} = \left(\frac{N_{ppp}}{3}\right)^{\left(0.2134\left(\frac{Gz}{30}\right)^{0.596}\right)} \tag{F-10}$$

where the subscript ppp means patterns per pitch. Treating the T2.062 and T3.062 data (Figure F-14) in the same manner gives Equation F-11

$$\frac{Nu_{a_{ppp}}}{Nu_{a_3}} = \left(\frac{N_{ppp}}{3}\right)^{\left(0.241\left(\frac{Gz}{30}\right)^{0.461}\right)} \tag{F-11}$$

Combining Eq. F-10 and F-11 with the function of P_D which matches the data for the two pattern depths tested gives Equation F-12.

$$\frac{Nu_{a_{ppp}}}{Nu_{a_3}} = \left(\frac{N_{ppp}}{3}\right)^{\left\{0.2134\left(\frac{P_D}{0.038}\right)^{0.25}\left(\frac{Gz}{30}\right)^{\left[0.596\left(\frac{P_D}{0.038}\right)^{-0.516}\right]}\right\}} \quad \text{F-12)}$$

for W = 0.077 in.

This equation is specific to these two data sets and its generality is questionable.

The data from Figure F-13 for T2.038 and T3.038 at W = 0.093 in. (N_f ~ 10 fins/in) is represented by Equation F-13.

$$\frac{Nu_{a_{ppp}}}{Nu_{a_3}} = \left(\frac{N_{ppp}}{3}\right)^{\left(0.4354 \left(\frac{Gz}{30}\right)^{0.21}\right)} \qquad \text{for } W = 0.093 \text{ in.} \tag{F-13}$$

The large difference between the exponents in Equations F-10 and F-13 clearly demonstrates that the adding of additional patterns per pitch is more effective in increasing Nu_a for larger fin spacings. Since Gz is almost proportional to W^2 (see Equation F-3) for a given velocity a model with less fin per in. (larger W) would have a much larger Gz number. The Nu_a vs. Gz curve is further from the 0.5 Gz asymptote at the same mass flow rate for the 0.093 in. fin spacing than for the 0.077 in. fin spacing and the curves of the T2.038 and T3.038 models are further apart.

The limited data in Figures F-15 through F-17 is not adequate to correlate the effect of fin spacing. What has been done is to compare the Nu_{a_W} values for a fin spacing W with the $Nu_{a_{0.077}}$ for a fin spacing of 0.077 in. Equations F-14, F-15 and F-16 are suggested.

$$\frac{Nu_{a_{0.161}}}{Nu_{a_{0.077}}} = 1.2 \left(P_D^{0.113}\right) \left(\frac{Gz}{10}\right)^{\left(-0.0504\, P_D^{0.144}\right)} \tag{F-14}$$

$$\frac{Nu_{a_{0.110}}}{Nu_{a_{0.077}}} = 1.15 \left(P_D^{0.061}\right) \left(\frac{Gz}{10}\right)^{(-0.006)} \tag{F-15}$$

$$\frac{Nu_{a_{0.056}}}{Nu_{a_{0.077}}} = 1.059 \left(\frac{Gz}{10}\right)^{(-0.025)} \tag{F-16}$$

The ratio for the fin spacing of 0.056 in. is based upon flat fin model data. No effect of P_D is included. It is projected, however, that the ratio should never be less than 1.0 and that the rates should approach 1.0 as P_D increases. What has been done is for a given pattern depth and Gz number to construct a second order fit to the ratio $\frac{Nu_{a_W}}{Nu_{a_{0.077}}}$ and to use the second order fit to calculate the expected ratio for other fin spacings and pattern depths. The second order fit would have the form given in Equation F-17 for the data Figure F-15 through F-17.

Letting

$$\frac{Nu_{a_W}}{Nu_{a_{0.077}}} = R_w , \frac{Nu_{a_{0.110}}}{Nu_{a_{0.077}}} = R_{110}, \frac{Nu_{a_{0.161}}}{Nu_{a_{0.077}}} = R_{161}$$

then (F-17)

$$R_w = A + BW + CW^2$$

where

$$A = (9.0321 \times 10^{-4} - 0.001041348\ R_{110} + 2.7951 \times 10^{-4} R_{161})/D$$

$$B = (-0.013821 + 0.019992\ R_{110} - 0.006171\ R_{161})/D$$

$$C = (0.051 - 0.084\ R_{110} + 0.033\ R_{161})/D$$

$$D = \begin{vmatrix} 1 & 0.077 & 0.077^2 \\ 1 & 0.110 & 0.110^2 \\ 1 & 0.161 & 0.161^2 \end{vmatrix} = 1.42372 \times 10^{-4}$$

with the injunction that if $W > 0.077$, $R_W \leq 1$ and if $W < 0.077$, $R_W \geq 1$. The second order fit leads to values slightly greater than 1.0 for $0.077 < W < .11$ and less than 1.0 for $W < 0.077$ at large values of P_D

The pressure drop data presented in the figures is intended as a guide and may conservatively be used for design directly at high face velocities. These curves if extrapolated to low velocities will give too low a pressure drop. The approach taken here, however, was to assume that better pressure drop correlations for flat plate fin-tube heat exchangers exist and to suggest multipliers or ratios by which the calculated pressure drop for a flat plate fin-tube can be modified to estimate the pressure drop of a pattern fin using transverse ribs as a pattern.

The measured drops would include an entry and exit loss which is a larger percent of the total for the 2 row coil than the 6 row coil, therefore, the value of Δp/row would be higher than it should be for each case but the percent error would be larger with fewer rows. It is felt that the relative pressure drop per row data has an error associated with plenum to plenum loss which was not accounted for. Nonetheless a lower slope is indicated for a high number of rows (Figure F-27) and the variation with number of rows becomes larger at higher velocities for these flat fin models.

$$\left(\frac{\Delta p}{N_R}\right)_{N_R} = 0.0818 \left(\frac{N_R}{3}\right)^{(-0.276)} \left(\frac{V_{max}}{1000}\right)^{\left(1.814 \left(\frac{N_R}{3}\right)^{-0.0827}\right)} \qquad \text{(F-18)}$$

The ratio of pressure drop for a heat exchanger of N_R rows to one with 3 rows is given by Equation F-19

$$R_R = \frac{\Delta p_{N_R}}{\Delta p_3} = \left(\frac{N_R}{3}\right)^{(-0.276)} \left(\frac{V_{max}}{1000}\right)^{\left\{1.814 \left[\left(\frac{N_R}{3}\right)^{(-.0827)} -1\right]\right\}} \qquad \text{(F-19)}$$

It is noted that this correlation smooths the data and decreases the slope of the $N_R = 3$ line so that it does not cross the $N_R = 4$ line.

The slope of the lines in Figure F-28 are lower for lower values of P_ℓ and the intercept at V_{max} = 1000 ft/min is higher.

$$\left(\frac{\Delta p}{N_R}\right)_{P_\ell} = 0.0767 \left(\frac{P_\ell}{P_t}\right)^{-0.45} \left(\frac{V_{max}}{1000}\right)^{\left\{1.836 \left(\frac{P_\ell}{P_t}\right)^{0.082}\right\}} \qquad \text{(F-20)}$$

The data relative of an equilateral triangular pitch is approximated by Equation F-21.

$$R_{P_\ell} = \frac{\left(\frac{\Delta P}{N_R}\right)_{P_\ell}}{\left(\frac{\Delta P}{N_p}\right)_{0.866}} = 0.937 \left(\frac{P_\ell}{P_t}\right)^{-0.45} \left(\frac{V_{max}}{1000}\right)^{\left\{1.836 \left[\left(\frac{P_\ell}{P_t}\right)^{0.082} - 0.9883\right]\right\}} \qquad \text{(F-21)}$$

An approximation of the data shown in Figure F-29 through F-32 is given by Equation F-22. This relation approximates the effect of tube transverse pitch and collar diameter on pressure drop for a 3 row heat exchanger model with T3.038 fins 0.077 in. apart.

$$\left(\frac{\Delta p}{N_R}\right)_{P_t, D_c} = 0.0915\,(P_t)^{0.61}\left(\frac{D_c}{0.375}\right)^{-0.096}\left(\frac{V_{max}}{1000}\right)^{\left\{1.818\,\left(P_t^{-0.032}\right)\left(\frac{D_c}{0.375}\right)^{\left(0.093\,P_t^{1.51}\right)}\right\}} \tag{F-22}$$

The pattern depth data in Figure F-33 may be represented by Equation F-24 for pattern depths of 0.018 to 0.048 in.

$$\left(\frac{\Delta p}{N_R}\right)_{P_D} = \left[0.102\left(\frac{P_D}{0.038}\right)^{0.244}\right]\left(\frac{V_{max}}{1000}\right)^{\left[1.737\left(\frac{P_D}{0.038}\right)^{-0.033}\right]} \tag{F-23}$$

for $0.018 < P_D \leq 0.048$ in.

At larger pattern depths the slope and intercept of the lines increases rapidly. A fourth order fit (Equation F-24) gives the slope as a function of $\left(\frac{P_D}{0.038}\right)$ assuming an asymptote of 20 at a pattern depth of 1.14 in. $\left(\frac{P_D}{0.038} = 30\right)$.

$$\begin{aligned} \text{Slope} = S_{P_D} &= 2.76153 - 1.73507\left(\frac{P_D}{0.038}\right) \\ &+ 0.915199\left(\frac{P_D}{0.038}\right)^2 - 0.155689\left(\frac{P_D}{0.038}\right)^3 \\ &+ 0.00423605\left(\frac{P_D}{0.038}\right)^4 \end{aligned} \tag{F-24}$$

for $0.048 < P_D < 0.125$ in.

A similar polynomial fit was used for the intercept at V_{max} equal 1000 ft/min in this range of pattern depth and is given by Equation F-25.

$$\text{Intercept} = I_{P_D} = 0.709804 - 1.18701\left(\frac{P_D}{0.038}\right) + 0.792040\left(\frac{P_D}{0.038}\right)^2 - 0.205322\left(\frac{P_D}{0.038}\right)^3 + 0.0187123\left(\frac{P_D}{0.038}\right)^4 \tag{F-25}$$

$0.048 < P_D < 0.125$.

The pressure drop per row for $0.048 < P_D < 0.125$ is then given by Equation F-26.

$$\left(\frac{\Delta p}{N_R}\right)_{P_D} = I_{P_D}\left(\frac{V_{max}}{1000}\right)^{S_{P_D}} \tag{F-26}$$

The ratio of the pressure drop for patterened fins to flat fins (R_{P_D}) is given by Equations F-27 and F-28.

$$R_{P_D} = 1.247\left(\frac{P_D}{0.038}\right)^{0.244}\left(\frac{V_{max}}{1000}\right)^{\left[1.737\left(\frac{P_D}{0.038}\right)^{-0.033} - 1.814\right]}$$

for $P_D < 0.048$ in. $\quad W = 0.077$ in. (F-27)

$$R_{P_D} = \frac{I_{P_D}}{0.0818}\left(\frac{V_{max}}{1000}\right)^{(S_{P_D} - 1.814)}$$

for $0.048 < P_D = 0.125$ in. $\quad W = 0.077$ in. (F-28)

$P_t = 1.25$ in. , $P_\ell = 1.083$ in. , $D_c = 0.5$ in.

The limited data available does not allow the approximation of pattern depth effects for other fin spacings. Equations F-23 through F-28 should apply conservatively for smaller fin spacings at high velocities. The expected decrease in pressure drop with increased fin spacing will always occur, but the magnitude of the pressure reduction is much less for higher fin patterns than for low fin patterns.

Only one four pattern per longitudinal pitch model test was performed. There is a rather large increase in pressure drop per row from 3 to 4 patterns per pitch. The slope of the lines in Figure F-36 increases with number of patterns per pitch. Since only three different patterns per pitch were used, the V_{max} = 1000 ft/min pressure drop per row intercept has been represented as a second order polynominal (Equation F-29). The slope of the ln-ln plot is seen to be a weak function of the number of patterns per pitch.

$$\left(\frac{\Delta p_{ppp}}{N_R}\right) = \left(0.1155 - 0.03725\ N_{ppp} + 0.00975\ N_{ppp}^{\ 2}\right) \left(\frac{V_{max}}{1000}\right)^{\left\{1.819\ \left(\frac{N_{ppp}}{3}\right)^{0.007}\right\}} \qquad \text{(F-29)}$$

The effect of fin spacing cannot adequately be described from the data in Figures F-39 through F-41 nor can a closed form relationship be easily represented. The previous relations were based on the supposition that the log-log plot of pressure drop per row versus maximum velocity was a straight line. This was a good assumption at high velocities for large fin spacings. The slope of the lines decreases at low velocities. This change in slope begins at higher values of V_{max} as fin spacing decreases. To demonstrate this the Carrier data of Donald G. Rich[2] as calculated from the f vs Reynolds curve in Reference 2 have been plotted as Figure F-42. These data were for four row 1.25 x 1.083, 0.525 in. heat exchangers with 0.006 in. thick fins. The line for 20.6 fin per in. (W = 0.0425 in.) does not approach a constant slope until V_m > 1700 ft per min. whereas the line for 2.92 fins per in. (W.= 0.336 in.) is straight down to V_m = 500 ft per min. Rich's data has been approximated by third order polynominal curve fits for fin spacings of 0.0425, 0.063, 0.079, 0.103, 0.123, 0.144, 0.220 and 0.336 in. The polynominal form is given by Equation F-30, and Table F-2 gives the coefficients for the various fin spacings

$$\frac{\Delta p}{N_R} = a + b\left(\frac{V_m}{1000}\right) + c\left(\frac{V_m}{1000}\right)^2 + d\left(\frac{V_m}{1000}\right)^3 \qquad \text{(F-30)}$$

Interpolating within Rich's data an intercept at V_{max} = 1000 ft/min of 0.0642 in. of water is calculated for 12 fins per in. Equation 18 shows a value of 0.0818

Table F-2

COEFFICIENTS FOR PRESSURE DROP REPRESENTATION
OF THE CARRIER DATA OF DONALD G. RICH

N_f fins/in.	W fin spacing	$a \times 10^4$	$b \times 10^2$	$c \times 10^1$	$d \times 10^3$
2.92	0.336	0.232968	0.524435	0.304869	-0.628066
4.42	0.220	-4.64179	0.718127	0.330303	-0.890074
6.67	0.144	-0.660208	1.09655	0.373563	-1.39782
7.67	0.123	-5.65715	1.56970	0.369718	-1.08710
9.17	0.103	-2.38408	1.65262	0.391920	-1.24869
11.75	0.079	-7.39091	2.24649	0.420248	-1.18275
14.5	0.063	-1.56136	3.55754	0.437723	-1.02635
20.6	0.425	+6.76952	7.96838	0.449478	-0.889989

in. of water which when a correction for collar diameter is made (Rich used 4 row coils) gives 0.0755 in. of water, a difference of 17.5 percent. The slope of Rich's data is less than the 4 row data in Figure F-27 [1.78 to 1.75 (Rich's data slope].

The slope of a straight line through Rich's high velocity data is plotted versus fin spacing in Figure F-43. The slope can be represented by Equation 31 for W between 0.062 and 0.336 in. ($14.5 > N_f > 2.92$). Only the 20.6 fin/in. data does not fall on this line.

$$\text{Slope} = 1.7448 \left(\frac{W}{0.077}\right)^{0.04777} \tag{F-31}$$

Also plotted in Figure F-43 are the intercepts of Rich's pressure drop data at V_{max} = 1000 ft per min. These intercept data increase rapidly at small values of fin spacing. A portion of this curve at the larger fin spacings can be represented by a straight line for $0.079 \leq W \leq 0.22$ in. The equation of the line drawn in the figure is:

$$\text{Intercept} - 0.0633 \left(\frac{W}{0.077}\right)^{-0.459} \tag{F-32}$$

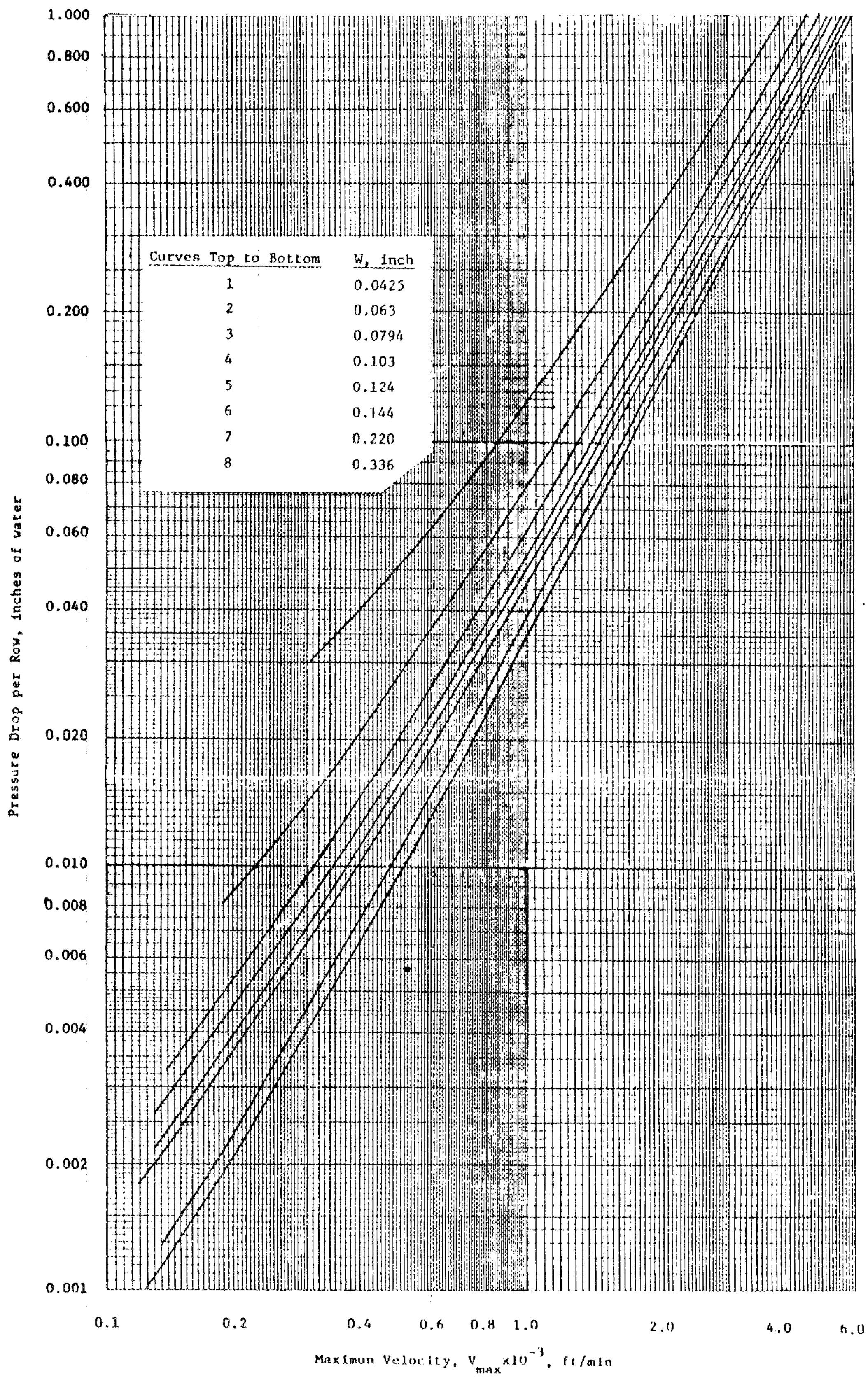


Figure F-42 - Effect of Fin Spacing, W, on Pressure Drop per Row as a Function of Maximum Fin Channel Velocity (Rich's Flat Fin Data)[2]
P_t = 1.250 inches, P_ℓ = 1.083 inches, D_c = 0.525 inch, P_D = 0, N_R = 4

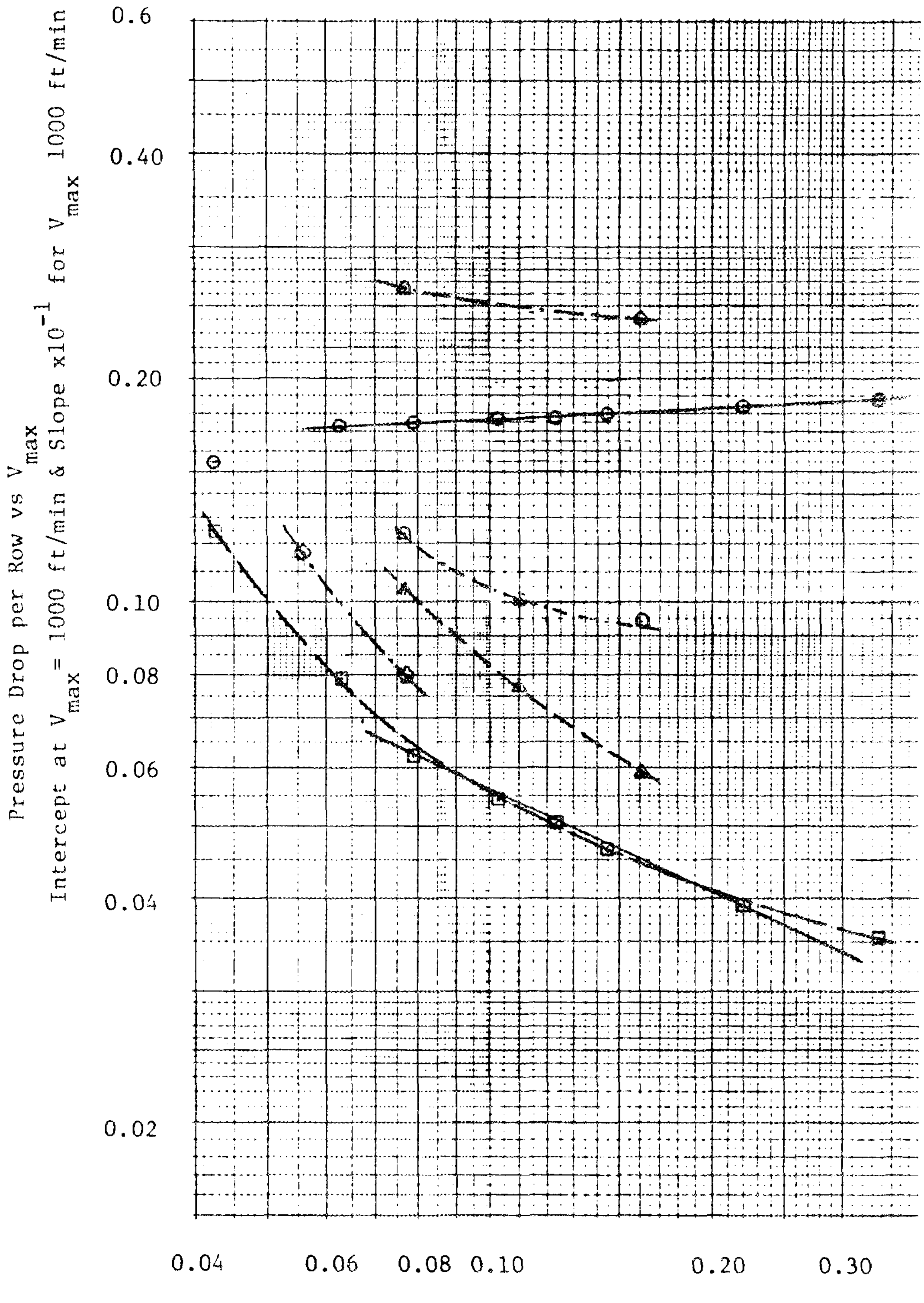


Figure F-43 - Effect of Fin Spacing, W, on the Intercept and Slope of the ℓn-ℓn Pressure Drop per Row vs Maximum Fin Channel Velocity Curves

○	Slope Rich's Data[2]	△	Intercept Single Channel T 3.038
	Intercept Rich's Data[2]		Intercept Single Channel T 3.062
◇	Intercept Single Channel Flat Plate	⌂	Intercept Sincle Channel T 3.125

The pressure drop per row ratio, (R_W), for a fin spacing W relative to a spacing of 0.077 inch would be given by Equation F-33 over a range of W from 0.077 to 0.22 inch.

$$R_W = \frac{\left(\frac{\Delta p}{N_R}\right)_W}{\left(\frac{\Delta p}{N_R}\right)_{0.077}} = \left(\frac{W}{0.077}\right)^{-0.459} \left(\frac{V_{max}}{1000}\right)^{\left\{1.7448\left[\left(\frac{W}{0.077}\right)^{0.04777} -1\right]\right\}} \qquad \text{(F-33)}$$

Also plotted in Figure F-43 are the intercepts from Figures F-39, F-40 and F-41. It is seen that the intercept is, as expected, larger with higher fin patterns and that the intercept for higher fin patterns is much less sensitive to fin spacing.

SUMMARY

The data presented and the relations developed in this paper are intended to be used in computational programs where patterned fin-tube heat exchanger performance must be calculated in the absence of adequate heat exchanger test data. Where actual data is available it should be used. The relations developed are specific to the model test data from which they are drawn, but the generality of the results can guide the inexperienced system designer in his choice of pattern fin surfaces for future heat transfer systems.

REFERENCE

1. Beecher, D. T., "Effect of Design Variable Changes on the Performance of Fin-Tube Heat Exchangers," Research Report 71-1E9-REAIR-R1, Westinghouse Research Laboratories, 1971.

2. Rich, D. G., "The Effect of Fin Spacing on the Heat Transfer and Friction Performance of Multi-Row, Smooth Plate Fin and Tube Heat Exchangers," ASHRAE Transactions, Volume 79, 1973, Part 2.

3. McQuiston, F. C and Tree, D. R., "Heat Transfer and Flow-Friction Data for Two Fin-Tube Surfaces," Journal of Heat Transfer, Vol. 77, May 1971, p. 249-250.

4. McQuiston, F. C., "Heat, Mass and Momentum Transfer Date for Five Plate-Fin-Tube Heat Transfer Surfaces," ASHRAE Transactions, Vol. 84, No. 1, 1978, p. 266-293.

NOMENCLATURE

c_p	–	Air specific heat	Btu/lbm °R
D_h	–	Hydraulic diameter of the fin passage defined by Equation 1	ft
D_c	–	Collar outside diameter	in.
D_t	–	Tube outside diameter	in.
Gz	–	Graetz number defined by Equation 3	dimensionless
h_a	–	Air side convection coefficient based on average air to surface temperature difference	Btu/hr ft^2 °R
h_ℓ	–	Air side convection coefficient based on logarithmic mean temperature difference	Btu/hr ft^2 °R
I_{P_D}	–	Intercept of $\Delta p/N_R$ vs V_{max} ℓn-ℓn curve at V_{max} = 1000 ft/min.	in. of water
k	–	Conductivity of the air	Btu/hr ft °R
L	–	Total axial length of the heat transfer surface	ft
N_{ppp}	–	Number of fin pattern cycles per longitudinal pitch	dimensionless
N_R	–	Number of tube rows in the air flow direction	dimensionless
N_f	–	Number of fins/inch	1/in.
$Nu_{a_{N_R}}$	–	Nusselt number based on h_a for a heat exchanger with N_R tube row	dimensionless
$Nu_{a(P_\ell/P_t)}$	–	Nusselt number based on h_a for heat exchangers as with a longitudinal to transverse pitch ratio of (P_ℓ/Pt)	dimensionless
$Nu_{a_{P_D}}$	–	Nusselt number based on h_a for a heat exchanger with patterned fin of pattern depth P_D	dimensionless
Nu_a	–	Nusselt number based on h_a $(h_a D_h/k)$	dimensionless
Nu_ℓ	–	Nusselt number based on h_ℓ $(h_\ell D_h/k)$	dimensionless
$Nu_{a_{ppp}}$	–	Nusselt number based on h_a for a heat exchanger with patterned fins with ppp patterns per longitudinal pitch	dimensionless
Nu_{a_W}	–	Nusselt number based on h_a for a heat exchanger with a fin spacing of W in longitudinal tube pitch	dimensionless

P_d	-	Pattern depth not including fin thickness	in.
P_ℓ	-	Longitudinal tube pitch	in.
Pr	-	Prandtl number $c_p\mu/k$	dimensionless
P_t	-	Transverse tube pitch	in.
Re	-	Reynolds number, $(D_h\rho\tilde{V}/\mu)$	dimensionless
R_{P_D}	-	Ratio of pressure drop per row for fins with a pattern depth of P_D to pressure drop per row for a flat fin	dimensionless
S_{P_D}	-	Slope of the $\Delta p/N_R$ vs V_{max} ℓn-ℓn curve at high velocities	dimensionless
$T_{air\ in}$		Air temperature at the test section entrance	°F
$T_{air\ out}$		Air temperature at the test section exit	°F
t_f	-	fin thickness	in.
T_m	-	Average air temperature, $0.5\ (T_{air\ in} + T_{air\ out})$	°F
T_{wall}	-	Fin temperature	°F
$\bar{V}$	-	Average velocity defined by Equation 2	ft/sec
V_{fin}	-	Air velocity between fins of a tubeless heat exchanger	ft/sec
V_{face}	-	Air velocity approaching heat exchanger face	in.
V_{max}	-	Maximum fin channel air velocity (Eq. 4)	ft/s or ft/m.
W	-	Spacing between fins	in.
X	-	Axial length of 1/2 cycle pattern, Eq. 14	in.
sec θ	-	Ratio of pattern fin area to plane area defined by Equation 11	dimensionless
β	-	Pitch to diameter ratio, $\frac{\pi}{4}\frac{P_tP_\ell}{D_c^2}$	dimensionless
ρ	-	Density of the air	lbm/ft^3
μ	-	Viscosity of the entering air	lbm/ft sec

Listing of CORFHA

A listing of the subprogram CORFHA follows.

```
      FUNCTION CORFHA(FP,DELTA,DEA,ST,WT,FPD,NT,NFP,UFA,KPA,PRA,
     1     XMUA,RHOA,IDIAG)
      REAL KPA,NT
C
C     ************************************************
C     AUGMENTATION OF AIR SIDE HEAT TRANSFER COEFFICIENT DUE TO FIN
C     PATTERNATION - CORRELATION DEVELOPED BY D T BEECHER FOR THE
C     COMMON 'ZIG-ZAG' PATTERN
C           FP    = FIN SPACING - FINS/FT
C           DELTA = FIN THICKNESS - FT
C           DEA   = TUBE OD - INCHES
C           ST    = TRANSVERSE TUBE PITCH - FT
C           WT    = LONGITUDINAL TUBE PITCH - FT
C           FPD   = CORRUGATION DEPTH PEAK TO VALLEY - INCHES
C                   (DOES NOT INCLUDE FIN THICKNESS)
C           NT    = NUMBER OF TUBE ROWS (REAL)
C           NFP   = NUMBER OF CORRUGATION PER
C                   LONGITUDINAL TUBE PITCH (INTEGER)
C           UFA   = COIL FACE VELOCITY - FT/MIN
C           KPA   = AIR THERMAL CONDUCTIVITY - BTU/HR-FT-DEG F
C           PRA   = AIR PRANDTL NUMBER - DIMENSIONLESS
C           XMUA  = AIR DYNAMIC VISCOSITY - LBM/HR-FT
C           RHOA  = AIR DENSITY - LBM/CU FT
C           GRTZ  = AIR GRAETZ NUMBER - DIMENSIONLESS (OUTPUT)
C           IDIAG = DIAGNOSTIC OUTPUT OPTION (INTEGER)
C                   (0 = NO OUTPUT 1 = FULL OUTPUT)
C     ************************************************
C
      REAL SEC,BETA,WFIN,UFIN,LFLO,NUAV,NULN,NUFP,NULF,NUAIR,
```

```
      1  NUFPR,NUPL,NUPH,NUREF,NUAFF,KAIR,NUA
       LOGICAL BZERO
       DATA PI,DDD/3.415926535,1.42372E-04/
       F = FP/12.0
       TF = 12.0*DELTA
       DTO = 12.0*DEA
       DCO = DTO + 2.0*DELTA
       PT = 12.0*ST
       PL = 12.0*WT
       PD = 12.0*FPD
       ROWS = NT
       NP = NFP
       UF = UFA
       KAIR = KPA
       PRAIR = PRA
       NUA = XMUA/RHOA
 1000  FORMAT(5X,'INPUT TO CORFHA',/,5X,'F     ',E11.4,' TF   ',E11.4,
      1  ' DCO  ',E11.4,' PT    ',E11.4,/,5X,'PL    ',E11.4,' PD   ',
      2  E11.4,' ROWS ',E11.4,' NP   ',7X,I5,/,5X,'UF    ',E11.4,
      3  ' KAIR ',E11.4,' PRAIR',E11.4,' NUA   ',E11.4,//)
 1100  FORMAT(5X,'BETA',E11.4,' WFIN',E11.4,' XPP ',E11.4,' SEC ',
      1  E11.4,/,5X,'UFIN',E11.4,' LFLO',E11.4,' GRTZ',E11.4,
      2  ' GRZF',E11.4,//)
 1110  FORMAT(5X,'NUFPR',E11.4,' XFINA',E11.4,' XFINB',E11.4,
      1  ' XFPIF',E11.4,//)
 1120  FORMAT(5X,'NUPH ',E11.4,' NUPL ',E11.4,' GRZH ',E11.4,
      1  ' GRZL ',E11.4,/,5X,'SLOPE',E11.4,' CONST',E11.4,
      2  ' NUREF',E11.4,//)
 1125  FORMAT(5X,'AAA  ',E11.3,' BBB  ',E11.4,' CCC  ',E11.4,
      1  ' XFPIP',E11.4,//)
 1130  FORMAT(5X,'XROWP',E11.4,' XPTLP',E11.4,' XFPIP',E11.4,
      1  ' XFPPP',E11.4,/,5X,'NUAV ',E11.4,//)
 1140  FORMAT(5X,'XROWF',E11.4,' XPTLF',E11.4,' XFPIF',E11.4,
      1  ' NUFP ',E11.4,//)
```

```
      IF(IDIAG .EQ. 1) WRITE(6,1000) F,TF,DCO,PT,PL,PD,ROWS,NP,UF,
     1  KAIR,PRAIR,NUA
      BZERO = .TRUE.
      IF(NP .EQ. 0) BZERO = .FALSE.
      IF(PD .LT. 0.0001) BZERO = .FALSE.
C
C      **************************************************
C      IF THERE IS NO FIN PATTERN (NP = 0) OR THE PATTERN
C      DEPTH PD IS LESS THAN 0.0001 INCHES THE EFFECT OF
C      FIN PATTERNATION IS ZERO OR NEGLIGIBLE AND BZERO IS
C      FALSE
C      **************************************************
C
C      *******************************************
C      CONVERT VISCOSITY FROM SQ FT/HR TO SQ FT/SEC
C      *******************************************
C
      NUAIR = NUA/3600.0
C
C      *****************************************************
C      CALCULATE THE RATIO OF THE COLLAR CROSS SECTIONAL AREA
C      TO THE FIN SURFACE AREA
C      *****************************************************
C
      BETA = (PI/4)*DCO*DCO/(PT*PL)
C
C      ************************
C     CALCULATE THE FIN SPACING
C      ************************
C
      WFIN = (1.0-F*TF)/F
C
C     .***********************************************************
C      CALCULATE THE PROJECTED LENGTH OF ONE HALF OF A CORRUGATION
```

```
C      **************************************************************
C
      XPP = PL/(2.0*NP)
C
C      *******************************************************
C      CALCULATE THE SECANT OF THE ANGLE OF THE CORRUGATION
C      *******************************************************
C
      SEC = SQRT(XPP*XPP+PD*PD)/XPP
      IF (.NOT. BZERO)  SEC = 1.0
C
C      *********************************************************
C      CALCULATE THE AIR MINIMUM AIR VELOCITY BETWEEN THE FINS
C      *********************************************************
C
      UFIN = UF/(60*(1-F*TF))
C
C      *********************************************************
C      CALCULATE THE FLOW LENGTH IN THE LONGITUDINAL DIRECTION
C      *********************************************************
C
      LFLO = ROWS*PL/12
C
C      *********************************************
C      CALCULATE THE GRAETZ NUMBER (DIMENSIONLESS)
C      FOR THE PATTERNED FIN GRTZ AND
C      FOR THE UNPATTERNED FIN GRZF
C      *********************************************
C
      GRTZ = (2*(WFIN/12)*UFIN/NUAIR)*PRAIR*(2*(WFIN/12)/LFLO)*(1-BETA)
     1  /((SEC-BETA*(SEC-2*WFIN/DCO))**2)
      GRZF = (2*(WFIN/12)*UFIN/NUAIR)*PRAIR*(2*(WFIN/12)/LFLO)*(1-BETA)
     1  /((1.0-BETA*(1.0-2*WFIN/DCO))**2)
      IF(IDIAG .EQ. 1) WRITE(6,1100) BETA,WFIN,XPP,SEC,UFIN,LFLO,
```

```
     1  GRTZ,GRZF
      IF (BZERO) GO TO 1200
      CORFHA = 1.0
      RETURN
1200  CONTINUE
C
C      ************************************************
C      CALCULATE THE NUSSULT NUMBER WITH NO PATTERN,
C      3 ROWS, A FIN SPACING OF 0.077 INCHES AND A
C      RATIO OF LONGITUDINAL TUBE PITCH TO TRANSVERSE
C      TUBE PITCH OF 0.866 - NUFPR
C      ************************************************
C
      IF(GRZF .GT. 30.0) GO TO 1210
      IF(GRZF .LT. 15.0) GO TO 1220
      NUFPR = 4.34*(GRZF/10.0)**0.725
      GO TO 1230
1210  CONTINUE
      NUFPR = 4.64*(GRZF/10.0)**0.664
      GO TO 1230
1220  CONTINUE
      NUFPR = 4.17*(GRZF/10.0)**0.822
1230  CONTINUE
      XFPIF = 1.0
      IF(WFIN .EQ. 0.077) GO TO 1240
C
C      ***************************************************
C      DETERMINE THE CORRECTION FACTOR FOR FIN SPACING
C      FOR THE FLAT FIN - FIRST CALCULATE THE CORRECTION
C      FACTOR FOR A FIN SPACING OF 0.077 INCHES (BEECHER'S
C      BASE VALUE) RELATIVE TO A 0.0794 INCH FIN SPACING
C      (RICH'S BASE VALUE)
C      ***************************************************
C
```

```
      XFINA = NUAFF( 0.077, GRZF)
C
C      ************************************************
C      NEXT CALCULATE THE RATIO OF THE NUSSULT NUMBER FOR
C      THE ACTUAL FIN SPACING TO THAT WITH A SPACING OF
C      0.0794 INCHES
C      ************************************************
      XFINB = NUAFF ( WFIN, GRZF)
      XFPIF = XFINB/XFINA
      IF(IDIAG .EQ. 1) WRITE(6,1110) NUFPR,XFINA,XFINB,XFPIF
1240  CONTINUE
C
C      *******************************************************
C      NEXT CALCULATE THE NUSSULT NUMBER FOR 3 PATTERNS PER
C      PITCH AND A FIN SPACING OF 0.077 INCHES WITH 3 TUBE
C      ROWS AND A RATIO OF LONGITUDINAL TUBE PITCH TO TRANSVERSE
C      TUBE PITCH OF 0.866 NUREF
C      *******************************************************
C
      IF(GRTZ .GT. 30.0) GO TO 1250
      IF(GRTZ .LT. 15.0) GO TO 1260
      NUPL =  9.30*PD**0.107
      NUPH = 20.98*PD**0.182
      GRZL = 15.0
      GRZH = 30.0
      GO TO 1270
1250  CONTINUE
      NUPL = 20.98*PD**0.182
      NUPH = 64.60*PD**0.293
      GRZL = 30.0
      GRZH = 80.0
      GO TO 1270
1260  CONTINUE
      NUPL = 3.85*PD**0.046
```

```
      NUPH = 9.30*PD**0.107
      GRZL = 7.0
      GRZH = 15.0
1270  CONTINUE
      SLOPE = ALOG(NUPH/NUPL)/ALOG(GRZH/GRZL)
      CONST = NUPH/(GRZH**SLOPE)
      NUREF = CONST*GRTZ**SLOPE
      IF(IDIAG .EQ. 1) WRITE(6,1120) NUPH,NUPL,GRZH,GRZL,SLOPE,
     1  CONST,NUREF
C
C      ****************************************************
C      CALCULATE THE CORRECTION FACTOR FOR OTHER NUMBERS OF
C      PATTERNS PER PITCH
C      ****************************************************
C
      XFPPP = 1.0
      IF(NP .EQ. 3) GO TO 1280
      POWGZ = 0.596*(PD/0.038)**(-0.516)
      POWPP = 0.2134*((PD/0.038)**0.25)*((GRTZ/30.0)**POWGZ)
      XFPPP = (1.0*NP/3.0)**POWPP
1280  CONTINUE
      XFPIP = 1.0
      IF(WFIN .EQ. 0.077) GO TO 1290
C
C      ****************************************************
C      CALCULATE THE RATIO OF THE NUSSULT NUMBER WITH A FIN
C      SPACING OF 0.161 INCHES TO THE NUSSULT NUMBER WITH A
C      FIN SPACING OF 0.077 INCHES
C      ****************************************************
C
      R161 = 1.2*(PD**0.113)*((GRTZ/10.0)**(-0.0504*PD**0.144))
C
C      ****************************************************
C      CALCULATE THE RATIO OF THE NUSSULT NUMBER WITH A FIN
```

```
C      SPACING OF 0.110 INCHES TO THE NUSSULT NUMBER WITH A
C      FIN SPACING OF 0.077 INCHES
C      *****************************************************
C
      R110 = 1.15*(PD**0.061)*((GRTZ/10.0)**(-0.006))
C
C      *******************************************************
C      CALCULATE THE COEFFICIENTS OF A SECOND ORDER CURVE FIT
C      IN TERMS OF FIN SPACING
C      *******************************************************
C
      AAA = (9.0321E-04 -0.001041348*R110 +2.7951E-04*R161)/DDD
      BBB = (-0.013821 +0.019992*R110 -0.006171*R161)/DDD
      CCC = (0.051 -0.084*R110 +0.033*R161)/DDD
      XFPIP = AAA +BBB*WFIN +CCC*WFIN*WFIN
      IF(IDIAG .EQ. 1)
     1      WRITE(6,1125) AAA,BBB,CCC,XFPIP
      IF(WFIN .LT. 0.077 .AND. XFPIP .LT. 1.0) XFPIP = 1.0
      IF(WFIN .GT. 0.077 .AND. XFPIP .GT. 1.0) XFPIP = 1.0
1290  CONTINUE
C      *****************************************************
C      CALCULATE THE EFFECT OF NUMBER OF TUBE ROWS IF OTHER
C      THAN 3
C      *****************************************************
C
      XROWP = 1.0
      XROWF = 1.0
      IF(ROWS .EQ. 3.0) GO TO 1300
      XROWP = (ROWS/3.0)**(0.22*(GRTZ/30.0)**0.626)
      XROWF = (ROWS/3.0)**(0.22*(GRZF/30.0)**0.626)
1300  CONTINUE
C
C      *******************************************************
C      CALCULATE THE EFFECT OF THE RATIO OF LONGITUDINAL TUBE
```

```
C      PITCH TO TRANSVERSE TUBE PITCH IF OTHER THAN 0.866
C      ******************************************************
C
      XPTLP = 1.0
      XPTLF = 1.0
      PRAT = PL/PT
      IF(PRAT .EQ. 0.866) GO TO 1310
      XPTLP = 1.0332*PRAT**(0.227*(GRTZ/30.0)**0.163)
      XPTLF = 1.0332*PRAT**(0.227*(GRZF/30.0)**0.163)
1310  CONTINUE
      NUAV = NUREF*XROWP*XPTLP*XFPIP*XFPPP
      IF(IDIAG .EQ. 1) WRITE(6,1130) XROWP,XPTLP,XFPIP,XFPPP,NUAV
      NUFP = NUFPR*XROWF*XPTLF*XFPIF
      IF(IDIAG .EQ. 1) WRITE(6,1140) XROWF,XPTLF,XFPIF,NUFP
C
C      ****************************************************
C      CALCULATE THE NUSSULT NUMBERS BASED ON LOGARITHMIC
C      MEAN TEMPERATURE DIFFERENCE FROM THE NUSSULT NUMBERS
C      BASED ON ARITHMETIC MEAN TEMPERATURE DIFFERENCE FOR
C      BOTH THE CORRUGATED AND FLAT FIN
C      ****************************************************
C
      NULN = (GRTZ/4)*ALOG((1+2*NUAV/GRTZ)/(1-2*NUAV/GRTZ))
      NULF = (GRZF/4)*ALOG((1+2*NUFP/GRZF)/(1-2*NUFP/GRZF))
      CORFHA = NULN/NULF
      IF(CORFHA .LT. 1.0) CORFHA = 1.0
      RETURN
      END
```

Listing of APDCOR

A listing of the subprogram APDCOR follows.

```
      FUNCTION APDCOR(PDF,NPP,VCF)
C
C       ****************************************
C       CALCULATES A MULTIPLYING FACTOR FOR AIR
C       PRESSURE DROP OF CORRUGATED PLATE FINNED
C       TUBE SURFACES RELATIVE TO AN UNPATTERNED
C       FIN. THE INPUT PARAMETERS INCLUDE:
C
C         PDF = FIN PATTERN DEPTH PEAK TO VALLEY - FT
C         NPP = THE NUMBER OF PATTERNS PER
C               LONGITUDINAL TUBE PITCH -
C               ONLY 2 OR THREE ARE VALID
C         CCF = AIR VELOCITY BASED ON COIL FACE
C               AREA - FT/MIN
C       ****************************************
C
1000  FORMAT(5X,'INVALID NUMBER OF FIN PATTERNS PER PITCH IN APDCOR',
     1      I5,/)
      PD = PDF*12.0
      NP = NPP
      VF = VCF
      IF(NP .LT. 2) GO TO 60
      IF(NP .GT. 3) GO TO 60
      IF(PD .LT. 0.017) GO TO 10
      IF(PD .LE. 0.038) GO TO 20
      IF(PD .GT. 0.038) GO TO 30
10    CONTINUE
C
C       ************************************************
```

```
C       IF THE PATTERN DEPTH IS LESS THAN 0.017 INCHES
C       THE CORRECTION TO AIR PRESSURE DROP IS NEGLIGIBLE
C       ************************************************
C
      APDCOR = 1.0
      RETURN
20    CONTINUE
C
C       **************************************************
C       CORRECTION FOR PATTERNS DEPTHS BETWEEN 0.017 INCHES
C       AND 0.038 INCHES - X IS IN PERCENT
C       **************************************************
C
      X = 1.1009376E+06*PD**3.0 - 8.1994418E+04*PD**2.0
     1    + 3.7196102E+03*PD - 4.4945482E+01
      GO TO 40
30    CONTINUE
C
C       ******************************************************
C       CORRECTION FOR PATTERN DEPTHS GREATER THAN 0.038 INCHES
C       X IS IN PERCENT
C       ******************************************************
C
      X = 9.13294E+05*PD**3.0 - 8.82083E+04*PD**2.0
     1    + 4.32138E+03*PD - 4.79662E+01
40    CONTINUE
      IF(NP .EQ. 2) GO TO 50
C
C       ********************************************************
C       REFERENCE VALUE FOR THREE FIN CORRUGATIONS PER TUBE PITCH
C       ********************************************************
C
      APDCOR = 1.0 + X/100.0
      RETURN
```

```
C
C      **************************************************
C      CORRECTION FOR TWO FIN CORRUGATIONS PER TUBE PITCH
C      **************************************************
C
50    CONTINUE
      XTWO = 1.01337E-06*VF**1.7778/(8.66881E-07*VF**1.82128)
      APDCOR = XTWO*(1.0 + X/100.0)
      IF(APDCOR .LT. 1.0) APDCOR = 1.0
      RETURN
60    CONTINUE
      APDCOR = 1.0
      WRITE(6,1000) NP
      RETURN
      END
```

APPENDIX G

Listings for Modified Subroutines

Listing of DATAIN Subroutine

A listing of the DATAIN subroutine follows:

```
      SUBROUTINE DATAIN
C
C
      REAL NSECTE,NTE,NSECTC,NTC
C
      COMMON / CMPRSR / TRICMP, TSICMP, HINCMP, PINCMP, XINCMP,
     1                  TROCMP, TSOCMP, HOUCMP, POUCMP, XOUCMP
C
      COMMON / COMPR / VR,     SYNC,   FLMOT,  EFFMMX, ETAISN, ETAMEC,
     1                 ETAVLA, ETAVLB, POW,    CANFAC, HILOFC, QCAN,
     2                 QHILO,  DISPL, MTRCLC
C
      COMMON / CONDEN /  DEAC,   DERC, DELTAC,   FPC, XKFC, XKTC, AAFC,
     1                   NTC, NSECTC, HCONTC,   STC,   WTC,  SIGAC,
     2                    PC,   ARFTC,  ARHTC, ALFARC,ALFAAC,  FARC,
     3                  CARC,     QAC, RTBCND,   DZC,FANEFC,  RHIC,
     4                FINTYC, MUNITC
C
      COMMON / CONDSR / TAIIC,  TIC, TSATCI, HIC, PIC, XIC,
     1                  TAOC,  TROC, TSATCO, HOC, POC, XOC
C
      COMMON /EVAPOR /  DEAE,  DERE,DELTAE,   FPE, XKFE, XKTE, AAFE,
     1                   NTE,NSECTE,HCONTE,   STE,   WTE, SIGAE,
     2                    PE, ARFTE, ARHTE,ALFARE,ALFAAE,  FARE,
     3                  CARE,   QAE,RTBEVP,   DZE,FANEFE,  RHIE,
```

```
     4                FINTYE,MUNITE
C
      COMMON / EVAPTR / TAIIE,  TIE, TSATEI, HIE, PIE, XIE,
     1                  TAOE,  TROE, TSATEO, HOE, POE, XOE
C
      COMMON / FANMOT / COFAN, C1FAN, C2FAN, EFFMOT, RPMFAN
C
      COMMON / FLOWBA / DTROC, SUPER, CAPFLO, ORIFD, XMR, NCAP,
     1                  IREFC, ICOMP, ITRPIE
C
      COMMON / LINES / DLL, XLEQLL, DSL,  XLEQSL, DDL, XLEQDL, DSLRV,
     1             XLEQLP, DDLRV,  XLEQHP,DPDL,   DPSL,DPLL,   QDISLN,
     2             QSUCLN, QLIQLN, E
C
C     ****** ADDED BY WESTINGHOUSE R&D JAN 10 1985 *******
C
      COMMON / RVALVE / TAMBRV, NRVALV, DPLOV, DPHIV, QINTV, QEXTV
C
C
      COMMON / MAPFIT / CPOW(6),CXMR(6),SUCFAC,VOLFAC,SUPERB,
     1                  DISPLB, POWCOR, XMRCOR
C
      COMMON / MPASS / CNDCON, AMBCON, EVPCON, CONMST, CMPCON, FLOCON,
     1                TOLS,   TOLH,   LPRINT, NCORH,  MCMPOP, MFANIN,
     2                MFANOU, MFANFT
C
      COMMON / TXVDAT / SUPRAT, TERAT,  TLQRAT, DPRAT,  BLEEDF,
     1                  DPTXV,  DPNOZ,  DPTUBE, CAP,    CAPNOZ,
     2                  CAPTUB, STATIC, TXVRAT, CTXV,   TLQCOR,
     3                  XLCORR, SUPERM, NZTBOP
C
C
C          ADDED AT WESTINGHOUSE R&D ON 5/10/85 - R. LUCHETA
C
```

```
      COMMON / ACPAR / DHHAC, DHLAC, DPHAC, DPLAC, NACCU,
     1                NACPR, ACPAR(20)
      COMMON / ACTBL / NTBDM, NACTB(12)
C        FOR THE MEANING OF THE CONTENTS OF COMMONS ACPAR AND
C        ACTBL, SEE THE LISTING FOR BLOCK DATA SUBPROGRAM RLADP.
C
C          END OF 5/10/85 WESTINGHOUSE R&D ADDITION.
C
C       ## ADDED AT WESTINGHOUSE R&D ON 8/2/85 - R. LUCHETA
C
      COMMON / RVLKP / NRVLK, NRDFL, NLPMX, NLKTB(5), RVLKP(5)
C       SEE BLOCK DATA PROGRAM RLADP FOR THE DEFINITION OF
C        THESE PARAMETERS.
C
C       ## END OF 8/2/85  WESTINGHOUSE R&D ADDITION ##
C
C    ## ADDED BY WESTINGHOUSE R&D ON 9/26/85 - T. J. FAGAN
C
      COMMON / FINPAT / NFPE, FPDE, XFPE, XAPE, NFPC, FPDC, XFPC, XAPC
C
C      ## END OF 9/26/85 ADDITION BY WESTINGHOUSE R & D
C
      COMMON  DDUCT, FIXCAP, ITITLE(20)
       DATA  NOTPR, NOTTT /  6, 6 /
C
C
C***********************************************************************
C
C        INPUT FOR MAIN
C
C***********************************************************************
C
C
C***********************************************************************
```

```
C
C       HEAT PUMP IDENTIFICATION
C
C************************************************************************
C
      READ(5,1001)  ITITLE
      WRITE(6,1004) ITITLE
C
C************************************************************************
C
C       PRINT OPTIONS
C
C************************************************************************
C
C     LPRINT - FLAG FOR PRINT CONTROL
C              =0 FOR MINIMUM OUTPUT
C              =1 FOR SUMMARY OUTPUT
C              =2 FOR OUTPUT AFTER EACH INTERMEDIATE ITERATION CONVERGES
C              =3 FOR CONTINUOUS OUTPUT DURING INTERMEDIATE ITERATIONS
C
      READ(5,1003) LPRINT
C
      GO TO (1,2,3), LPRINT
      WRITE(6,1020)
      GO TO 4
    1 WRITE(6,1021)
      GO TO 4
    2 WRITE(6,1022)
      GO TO 4
    3 WRITE(6,1023)
    4 CONTINUE
C
C************************************************************************
C
```

```
C       OPERATING MODE SELECTION
C
C***********************************************************************
C
C        NCORH - INDICATOR FOR COOLING OR HEATING MODE
C                IF 'NCORH' = 1 -- COOLING MODE
C                IF 'NCORH' = 2 -- HEATING MODE
C
      READ(5,1003) NCORH
C
      IF (NCORH .EQ. 1) WRITE(6,1005)
      IF (NCORH .EQ. 2) WRITE(6,1006)
C
C***********************************************************************
C
C       CHARGE INVENTORY OPTION SELECTION
C
C       ***NOTE: THE REFRIGERANT CHARGE INVENTORY BALANCE OPTION
C                (ICHRGE=1) HAS NOT BEEN COMPLETED NOR HAS THE
C                COMPUTATION OF REQUIRED REFRIGERANT CHARGE IN THE
C                'ICHRGE=0' OPTION.
C
C***********************************************************************
C
C        ICHRGE- INDICATOR FOR SPECIFYING COMPRESSOR INLET SUPERHEAT
C                                      OR SYSTEM REFRIGERANT CHARGE
C                =0 SPECIFY REFRIGERANT SUPERHEAT (OR QUALITY) AND
C                   COMPUTE REQUIRED SYSTEM REFRIGERANT CHARGE
C                =1 ESTIMATE COMPRESSOR INLET SUPERHEAT AND
C                   SPECIFY SYSTEM REFRIGERANT CHARGE
C
C        V1-     SPECIFIED REFRIGERANT SUPERHEAT (OR QUALITY)
C                OR
C                ESTIMATE OF REFRIGERANT SUPERHEAT (OR QUALITY)
```

```
C                 AT COMPRESSOR INLET (F DEG OR NEGATIVE FRACTION)
C
C        V2-      SPECIFIED SYSTEM REFRIGERANT CHARGE (LBM)
C
      READ(5,1101) ICHRGE,V1,V2
C
      SUPER = V1
      IF(ICHRGE .EQ. 1) GO TO 5
      WRITE(6,1024)
      IF(SUPER .GE. 0.0) WRITE(6,1025) SUPER
      XSUPER = -SUPER
      IF(SUPER .LT. 0.0) WRITE(6,1026) XSUPER
      GO TO 6
    5 CONTINUE
      REFCHG = V2
      WRITE(6,1027) REFCHG
      IF(SUPER .GE. 0.0) WRITE(6,1028) SUPER
      XSUPER = -SUPER
      IF(SUPER .LT. 0.0) WRITE(6,1029) XSUPER
    6 CONTINUE
C
C
C***********************************************************************
C
C       FLOW CONTROL INPUT SECTION
C
C***********************************************************************
C
C        IREFC-   INDICATOR FOR TYPE OF REFRIGERANT FLOW CONTROL DEVICE
C                 =0 FOR SPECIFIED CONDENSER SUBCOOLING (OR QUALITY)
C                    (F DEG OR NEGATIVE FRACTION)
C                 =1 FOR THERMAL EXPANSION VALVE (TXV)
C                 =2 FOR CAPILLARY TUBE(S)
C                 =3 FOR SHORT-TUBE ORIFICE
```

```
C       A1-     SPECIFIED CONDENSER SUBCOOLING (OR QUALITY),
C               OR
C               RATED CAPACITY OF TXV (TONS),
C               OR
C               CAPILLARY TUBE FLOW FACTOR,
C               OR
C               DIAMETER OF SHORT-TUBE ORIFICE (IN).
C       B1-     STATIC SUPERHEAT SETTING FOR TXV (F),
C               OR
C               NUMBER OF CAPILLARY TUBES
C       C1-     TXV SUPERHEAT AT RATING CONDITIONS (F)
C       D1-     TXV BYPASS OR BLEED FACTOR
C       E1-     SWITCH TO OMIT TXV NOZZLE AND TUBE PRESSURE DROP
C               CALCULATIONS
C               =0, OMIT TUBE AND NOZZLE PRESSURE DROPS,
C               =1, INCLUDE TUBE AND NOZZLE PRESSURE DROPS.
C
      READ(5,1101) IREFC,A1,B1,C1,D1,E1
      DTROC = A1
      IF (IREFC .EQ. 0) GO TO 40
      GO TO (10,20,30),IREFC
      WRITE(6,1102)  IREFC
      IREFC = 0
      GO TO 40
C
C     TXV INPUT
C
   10 TXVRAT = A1
      STATIC = B1
      SUPRAT = C1
      BLEEDF = D1
      NZTBOP = IFIX(E1)
      IF (NZTBOP .NE. 0) NZTBOP = 1
      WRITE(6,1103) TXVRAT,STATIC,SUPRAT,BLEEDF
```

```
      IF (NZTBOP .EQ. 0) WRITE(6,1104)
      IF (NZTBOP .EQ. 1) WRITE(6,1105)
      GO TO 45
C
C     CAPILLARY TUBE INPUT
C
   20 CAPFLO = A1
      NCAP = IFIX(B1)
      WRITE(6,1106) NCAP,CAPFLO
      GO TO 45
C
C     SHORT-TUBE ORIFICE INPUT
C
   30 WRITE(6,1107) A1
      ORIFD = A1/12.
      GO TO 45
C
   40 CONTINUE
C
C     FIXED CONDENSER SUBCOOLING INPUT
C
      IF(DTROC .GE. 0.0) WRITE(6,1108) DTROC
      XDTROC = -DTROC
      IF(DTROC .LT. 0.0) WRITE(6,1109) XDTROC
C
   45 CONTINUE
C
C***********************************************************************
C
C       ESTIMATES OF LOW AND HIGH SIDE REFRIGERANT SATURATION
C       TEMPERATURES
C
C***********************************************************************
C
```

```
C        TSICMP- ESTIMATE OF SATURATION TEMPERATURE AT COMPRESSOR INLET
C                 (F)
C        TSOCMP- ESTIMATE OF SATURATION TEMPERATURE AT COMPRESSOR OUTLET
C                 (F)
C
C
      READ(5,1002) TSICMP,TSOCMP
      WRITE(6,1204) TSICMP,TSOCMP
C
C**********************************************************************
C
C       COMPRESSOR INPUT SECTION
C
C**********************************************************************
C
C     INPUT DATA FOR THE COMPRESSOR SUBMODELS
C
C       ICOMP - SWITCH TO SPECIFY SUBMODEL
C               =1 FOR THE EFFICIENCY & LOSS SUBMODEL
C               =2 FOR THE MAP-BASED SUBMODEL
C       DISPL - TOTAL COMPRESSOR DISPLACEMENT (CU IN)
C       SYNC  - SYNCHRONOUS MOTOR SPEED
C               WHEN ICOMP=1 AND FLMOT IS SPECIFIED;
C               RATED MOTOR SPEED
C               WHEN ICOMP=1 AND FLMOT IS TO BE CALCULATED
C               OR WHEN ICOMP=2
C       QCAN  - COMPRESSOR SHELL HEAT LOSS (BTU/H)
C       CANFAC- SWITCH TO DETERMINE THE COMPRESSOR SHELL HEAT LOSS,
C               QCAN:
C               =0,  TO SPECIFY QCAN EXPLICITLY
C               <1,  TO CALCULATE QCAN = CANFAC * POW
C               >=1, TO CALCULATE QCAN AS 0.90 * (1 - MOTOR * MECHANICAL
C                    EFFICIENCY) * POW
C
```

```
      READ(5,1101) ICOMP,DISPL,SYNC,QCAN,CANFAC
      GO TO (90,100,100),ICOMP
      WRITE(6,1302) ICOMP
      ICOMP = 1
C
C     **INPUT DATA FOR THE EFFICIENCY & LOSS SUBMODEL**
C
C       VR     - COMPRESSOR ACTUAL CLEARANCE VOLUME RATIO
C       EFFMMX- MAXIMUM EFFICIENCY OF THE COMPRESSOR MOTOR
C       ETAISN- ISENTROPIC EFFICIENCY OF COMPRESSOR
C       ETAMEC- COMPRESSOR MECHANICAL EFFICIENCY
C       MTRCLC- SWITCH TO DETERMINE WHETHER TO
C               CALCULATE THE FULL LOAD MOTOR POWER (FLMOT)
C               OR TO USE THE INPUT VALUE OF 'FLMOT'
C               =0, CALCULATE 'FLMOT',
C               =1, USE THE INPUT VALUE OF 'FLMOT'.
C       FLMOT - COMPRESSOR MOTOR OUTPUT AT FULL LOAD (KW)
C               (NOT USED IF MTRCLC=1)
C       QHILO - HEAT TRANSFER RATE FROM COMPRESSOR DISCHARGE LINE
C               TO INLET GAS (BTU/H)
C       HILOFC- SWITCH TO DETERMINE INTERNAL HEAT TRANSFER FROM
C               THE HIGH SIDE TO THE LOW SIDE, 'QHILO':
C               =0,  'QHILO' IS SPECIFIED EXPLICITLY
C               <1,  'QHILO' IS CALCULATED AS 'HILOFC * POW'
C               >=1, 'QHILO' IS CALCULATED AS '0.03 * POW'
C
   90 READ(5,1002) VR,EFFMMX,ETAISN,ETAMEC
      READ(5,1101) MTRCLC,FLMOT,QHILO,HILOFC
C
      WRITE(6,1304) DISPL, ETAISN,
     1              SYNC,  EFFMMX,
     2              VR,    ETAMEC
      IF (CANFAC .LE. 0.0) GO TO 92
      IF (CANFAC .LT. 1.0) WRITE(6,1305) CANFAC
```

```
      IF (CANFAC .GE. 1.0) WRITE(6,1306)
      GO TO 94
   92 WRITE(6,1307) QCAN
   94 IF(HILOFC .LE. 0.0) GO TO 96
      IF (HILOFC .LT. 1.0) WRITE(6,1308) HILOFC
      IF (HILOFC .GE. 1.0) WRITE(6,1309)
      GO TO 98
   96 WRITE(6,1310) QHILO
   98 IF (MTRCLC .NE. 0) WRITE(6,1311)
      IF (MTRCLC .EQ. 0) WRITE(6,1312) FLMOT
      GO TO 110
C
C     **INPUT DATA FOR THE MAP-BASED COMPRESSOR MODEL**
C
C       CPOW  - COEFFICIENTS FOR BI-QUADRATIC FIT TO COMPRESSOR POWER
C               (KW)
C       CXMR  - COEFFICIENTS FOR BI-QUADRATIC FIT TO MASS FLOW RATE
C       DISPLB- BASE COMPRESSOR DISPLACEMENT FOR MAP (CUBIC INCHES)
C       SUPERB- BASE 'SUPERHEAT' VALUE FOR COMPRESSOR MAP,
C               IF POSITIVE,
C                 BASE SUPERHEAT ENTERING COMPRESSOR (F),
C               IF NEGATIVE,
C                 NEGATIVE OF RETURN GAS TEMPERATURE INTO COMPRESSOR
C                 (F).
C
  100 READ(5,1313) CPOW(1),CPOW(2),CPOW(3),CPOW(4),CPOW(5),CPOW(6),
     1              DISPLB, SUPERB
      READ(5,1313) CXMR(1),CXMR(2),CXMR(3),CXMR(4),CXMR(5),CXMR(6)
C
      WRITE(6,1314) DISPL,SYNC
      WRITE(6,1315) CPOW(1),CPOW(2),CPOW(3),CPOW(4),CPOW(5),CPOW(6)
      WRITE(6,1316) CXMR(1),CXMR(2),CXMR(3),CXMR(4),CXMR(5),CXMR(6)
      IF(SUPERB.GE.0.0) GO TO 105
      RTURNG = -SUPERB
```

```
      WRITE(6,1318) RTURNG,DISPLB
      GO TO 106
105   CONTINUE
      WRITE(6,1317) SUPERB,DISPLB
106   CONTINUE
      WRITE(6,1319) SUCFAC,VOLFAC
      IF (CANFAC .LE. 0.0) WRITE(6,1307) QCAN
      IF (CANFAC .LE. 1.0) WRITE(6,1305) CANFAC
C
C
C***********************************************************************
C
C        INDOOR UNIT INPUT SECTION
C
C***********************************************************************
C
C    INPUT DATA FOR EITHER HEAT EXCHANGER
C
C    HEAT EXCHANGER CHARACTERISTICS (INDOOR AND OUTDOOR)
C    THE SUFFIX 'C' IS USED FOR THE CONDENSER AND 'E' FOR
C    THE EVAPORATOR
C
C      TAII   -  AIR TEMPERATURE ENTERING HEAT EXCHANGER (F)
C      RHI    -  RELATIVE HUMIDITY OF AIR ENTERING THE HEAT EXCHANGER
C
C      QA     -  AIR FLOW RATE (CU FT/MIN)
C      FANEF     IF <= 1.0, SPECIFIED VALUE OF COMBINED FAN - FAN MOTOR
C                           EFFICIENCY
C                IF  > 1.0, SPECIFIED VALUE OF FAN POWER (WATTS)
C     **NOTE**   IF FOR OUTDOOR COIL, 'FANEF' USED ONLY IF 'MFANFT'=0
C
C    INDOOR COIL VARIABLES ONLY:
C      DDUCT  -  DIAMETER OF 6 CIRCULAR AIR DUCTS (IN) --
C                EACH WITH AN EQUIVALENT LENGTH OF 100 FEET
```

```
C      FIXCAP -  HOUSE HEATING LOAD (USED TO CALCULATE RESISTANCE HEAT)
C                (BTU/H)
C    OUTDOOR COIL VARIABLES ONLY:
C      MFANFT -  FLAG FOR USING STATIC EFFICIENCY VS. SPECIFIC SPEED
C                CURVE FIT FOR THE EFFICIENCY OF THE OUTDOOR FAN
C                =0, SPECIFIED VALUE OF 'FANEF' IS USED
C                =1, CURVE FIT FOR FAN STATIC EFFICIENCY IS USED WITH
C                    FIXED FAN MOTOR EFFICIENCY VALUE GIVEN IN BLOCK DATA
C
C      AAF    -  HEAT EXCHANGER FRONTAL AREA (SQ FT)
C      NT     -  NUMBER OF TUBES IN DIRECTION OF AIR FLOW
C      NSECT  -  NUMBER OF PARALLEL CIRCUITS IN HEAT EXCHANGER
C      ST     -  VERTICAL SPACING OF TUBE PASSES (IN)
C      WT     -  SPACING OF TUBE ROWS IN DIRECTION OF AIR FLOW (IN)
C      RTB    -  TOTAL # OF RETURN BENDS IN HEAT EXCHANGER (ALL CIRCUITS)
C
C      FINTY  -  TYPE OF FIN SURFACE
C                 1.0 -- SMOOTH FIN
C                 2.0 -- WAVY FIN
C                 3.0 -- LOUVERED FIN
C                 4.0 -- CORRUGATED FIN
C      FP     -  FIN PITCH (FINS/IN)
C      DELTA  -  FIN THICKNESS (IN)
C      DEA    -  OUTSIDE DIAMETER OF TUBES (IN)
C      DER    -  INSIDE DIAMETER OF TUBES (IN)
C      XKF    -  THERMAL CONDUCTIVITY OF FINS (BTU/H-FT-F)
C      XKT    -  THERMAL CONDUCTIVITY OF TUBES (BTU/H-FT-F)
C      HCONT  -  FRACTION OF COMPUTED CONTACT CONDUCTANCE BETWEEN
C                FINS AND TUBES
C
C    ####### ADDED BY WESTINGHOUSE R&D 9/26/85 - T. J. FAGAN ###
C
C    THE FOLLOWING DATA IS READ ONLY WHEN FINTY IS 4
C
```

```
C     NFP     - NUMBER OF FIN PATTERNS PER LONGITUDINAL TUBE PITCH
C               (INTEGER)
C     FPD     - FIN PATTERN DEPTH - PEAK TO VALLEY (IN)
C
C   ####### END OF ADDITION OF 9/30/85 #####
C
  110 IF (NCORH .EQ. 2) GO TO 120
C
C       NCORH = 1 FOR COOLING MODE
C       INDOOR UNIT IS THE EVAPORATOR
C
      READ(5,1002) TAIIE,RHIE
      READ(5,1002) QAE,FANEFE,DDUCT,FIXCAP
      READ(5,1002) AAFE,NTE,NSECTE,WTE,STE,RTBEVP
      READ(5,1002) FINTYE,FPE,DELTAE,DEAE,DERE,XKFE,XKTE,HCONTE
C   ## ADDED BY WESTINGHOUSE R & D ON 9/26/85 - T. J. FAGAN
C
      IF(FINTYE .EQ. 4) READ(5,1007) NFPE,FPDE
      GO TO 200
C
C     NCORH = 2 FOR HEATING MODE
C     INDOOR UNIT IS THE CONDENSER
C
  120 READ(5,1002) TAIIC,RHIC
      READ(5,1002) QAC,FANEFC,DDUCT,FIXCAP
      READ(5,1002) AAFC,NTC,NSECTC,WTC,STC,RTBCND
      READ(5,1002) FINTYC,FPC,DELTAC,DEAC,DERC,XKFC,XKTC,HCONTC
C
C     #### ADDED BY WESTINGHOUSE R&D 9/26/85 - T. J. FAGAN
      IF(FINTYC .EQ. 4) READ(5,1007) NFPC,FPDC
C
C     ##### END OF 9/26/85 ADDITION #####
C
  200 CONTINUE
```

```
C
C*********************************************************************
C
C         OUTDOOR UNIT INPUT SECTION
C
C*********************************************************************
C
      IF (NCORH .EQ. 2) GO TO 300
C
C     NCORH = 1 FOR COOLING MODE
C     OUTDOOR UNIT IS THE CONDENSER
C
      READ(5,1002) TAIIC,RHIC
      READ(5,1400) QAC,FANEFC,MFANFT
      READ(5,1002) AAFC,NTC,NSECTC,WTC,STC,RTBCND
      READ(5,1002) FINTYC,FPC,DELTAC,DEAC,DERC,XKFC,XKTC,HCONTC
C
C     ##### ADDED BY WESTINGHOUSE R&D 9/26/85 - T. J. FAGAN
      IF(FINTYC .EQ. 4) READ(5,1007) NFPC,FPDC
C
C     ##### END OF 9/26/85 ADDITION #####
      GO TO 400
C
C     NCORH = 2 FOR HEATING MODE
C     OUTDOOR UNIT IS THE EVAPORATOR
C
  300 READ(5,1002) TAIIE,RHIE
      READ(5,1400) QAE,FANEFE,MFANFT
      READ(5,1002) AAFE,NTE,NSECTE,WTE,STE,RTBEVP
      READ(5,1002) FINTYE,FPE,DELTAE,DEAE,DERE,XKFE,XKTE,HCONTE
C
C   #### ADDED BY WESTINGHOUSE R&D CENTER 9/26/85/ - T. J. FAGAN
      IF(FINTYE .EQ. 4) READ(5,1007) NFPE,FPDE
C
```

```
C   ##### END OF 9/26/85 ADDITION ######
C
  400 CONTINUE
C
C      ECHO INPUT
C
       IF (NCORH .EQ. 2) GO TO 500
C
C      NCORH = 1 FOR COOLING MODE
C      THE EVAPORATOR IS THE INDOOR UNIT
C      THE CONDENSER IS THE OUTDOOR UNIT
C
       WRITE(6,1403)
       WRITE(6,1407) TAIIE,RHIE
C
       IF(FANEFE .GT. 1.0) GO TO 50
       WRITE(6,1411) QAE,FANEFE
       GO TO 60
50     CONTINUE
       WRITE(6,1412) QAE,FANEFE
60     CONTINUE
C
       WRITE(6,1415) DDUCT,FIXCAP
C
       IF (FINTYE .EQ. 1.) WRITE(6,1408) AAFE
       IF (FINTYE .EQ. 2.) WRITE(6,1409) AAFE
       IF (FINTYE .EQ. 3.) WRITE(6,1410) AAFE
C
C   ###### ADDED BY WESTINGHOUSE R&D 9/27/95 - T. J. FAGAN
C
       IF (FINTYE .EQ. 4.) WRITE(6,1509) AAFE,NFPE,FPDE
C
C   ###### END OF 9/27/85 ADDITION
C
```

```
C
      WRITE(6,1414) NTE,          FPE,
     1              NSECTE,       DELTAE,
     2              DEAE,         XKFE,
     3              DERE,         XKTE,
     4              WTE,          HCONTE,
     5              STE,          RTBEVP
C
C
      WRITE(6,1404)
      WRITE(6,1407) TAIIC,RHIC
C
      IF(MFANFT .EQ. 1) GO TO 51
      IF(FANEFC .LE. 1.0) WRITE(6,1411) QAC,FANEFC
      IF(FANEFC .GT. 1.0) WRITE(6,1412) QAC,FANEFC
      GO TO 61
51    CONTINUE
      WRITE(6,1413) QAC,EFFMOT
61    CONTINUE
C
      IF (FINTYC .EQ. 1.) WRITE(6,1408) AAFC
      IF (FINTYC .EQ. 2.) WRITE(6,1409) AAFC
      IF (FINTYC .EQ. 3.) WRITE(6,1410) AAFC
C
C   ##### ADDED BY WESTINGHOUSE R&D 9/27/85 - T. J. FAGAN
C
      IF (FINTYC .EQ. 4.) WRITE(6,1509) AAFC,NFPC,FPDC
C
C   ###### END OF 9/27/85 ADDITION #####
C
C
      WRITE(6,1414) NTC,          FPC,
     1              NSECTC,       DELTAC,
     2              DEAC,         XKFC,
```

```
     3                DERC,        XKTC,
     4                WTC,         HCONTC,
     5                STC,         RTBCND
C
      IF(MFANFT .EQ. 1) WRITE(6,1416) COFAN, C1FAN, C2FAN, RPMFAN
C
      GO TO 600
C
C     NCORH = 2 FOR HEATING MODE
C     THE CONDENSER IS THE INDOOR UNIT
C     THE EVAPORATOR IS THE OUTDOOR UNIT
C
  500 WRITE(6,1405)
      WRITE(6,1407) TAIIC,RHIC
C
      IF(FANEFC .GT. 1.0) GO TO 52
      WRITE(6,1411) QAC,FANEFC
      GO TO 62
52    CONTINUE
      WRITE(6,1412) QAC,FANEFC
62    CONTINUE
C
      WRITE(6,1415) DDUCT,FIXCAP
C
      IF (FINTYC .EQ. 1.) WRITE(6,1408) AAFC
      IF (FINTYC .EQ. 2.) WRITE(6,1409) AAFC
      IF (FINTYC .EQ. 3.) WRITE(6,1410) AAFC
C
C   ###### ADDED BY WESTINGHOUSE R&D 9/27/85 - T. J. FAGAN
C
      IF (FINTYC .EQ. 4.) WRITE(6,1509) AAFC,NFPC,FPDC
C
C   ###### END OF 9/27/85 ADDITION #######
C
```

```
C
      WRITE(6,1414) NTC,         FPC,
     1              NSECTC,      DELTAC,
     2              DEAC,        XKFC,
     3              DERC,        XKTC,
     4              WTC,         HCONTC,
     5              STC,         RTBCND
C
C
      WRITE(6,1406)
      WRITE(6,1407) TAIIE,RHIE
C
      IF(MFANFT .EQ. 1) GO TO 53
      IF(FANEFE .LE. 1.0) WRITE(6,1411) QAE,FANEFE
      IF(FANEFE. GT. 1.0) WRITE(6,1412) QAE,FANEFE
      GO TO 63
53    CONTINUE
      WRITE(6,1413) QAE,EFFMOT
63    CONTINUE
C
      IF (FINTYE .EQ. 1.) WRITE(6,1408) AAFE
      IF (FINTYE .EQ. 2.) WRITE(6,1409) AAFE
      IF (FINTYE .EQ. 3.) WRITE(6,1410) AAFE
C
C    ##### ADDED BY WESTINGHOUSE R&D 9/27/85 - T. J. FAGAN
C
      IF (FINTYE .EQ. 4.) WRITE(6,1509) AAFE,NFPE,FPDE
C
C    ##### END OF 9/27/85 ADDITION #######
C
C
      WRITE(6,1414) NTE,         FPE,
     1              NSECTE,      DELTAE,
```

```
     2               DEAE,        XKFE,
     3               DERE,        XKTE,
     4               WTE,         HCONTE,
     5               STE,         RTBEVP
C
C
      IF(MFANFT .EQ. 1) WRITE(6,1416) COFAN, C1FAN, C2FAN, RPMFAN
C
  600 CONTINUE
C
C        CONVERT DIMENSIONS FROM INCHES TO FEET
C
      DEAC = DEAC/12.
      DEAE = DEAE/12.
C
      DELTAC = DELTAC/12.
      DELTAE = DELTAE/12.
C
      DERC = DERC/12.
      DERE = DERE/12.
C
      FPC = 12.*FPC
      FPE = 12.*FPE
C
      WTC = WTC/12.
      WTE = WTE/12.
C
      STC = STC/12.
      STE = STE/12.
C
C #######ADDED BY WESTINGHOUSE R&D 9/26/86/ - T. J. FAGAN#####
C
      FPDC = FPDC/12.0
      FPDE = FPDE/12.0
```

```
C
C  ###### END OF ADDITION OF 9/26/85 ####
C
      DDUCT = DDUCT/12.
C
C***********************************************************************
C        COMPRESSOR AND FAN CONFIGURATION OPTIONS
C***********************************************************************
C
C     MCMPOP - FLAG FOR ADDING
c                    COMPRESSOR CAN HEAT LOSS TO AIR FROM OUTDOOR COIL
C              =0, COMPRESSOR CAN LOSS NOT ADDED TO OUTDOOR AIR
C              =1, COMPRESSOR CAN LOSS ADDED
c                    TO AIR BEFORE CROSSING OUTDOOR COIL
C              =2, COMPRESSOR CAN LOSS
c                    ADDED TO AIR AFTER CROSSING OUTDOOR COIL
C     MFANIN - FLAG FOR ADDING HEAT LOSS FROM INDOOR FAN TO AIR STREAM
C              SETTINGS SIMILAR TO THOSE FOR MCMPOP
C     MFANOU - FLAG FOR ADDING HEAT LOSS FROM OUTDOOR FAN TO AIR STREAM
C              SETTINGS SIMILAR TO THOSE FOR MCMPOP
C
      READ(5,1003) MCMPOP, MFANIN, MFANOU
C
      IF (MCMPOP .EQ. 1) WRITE(6,1010)
      IF (MCMPOP .EQ. 2) WRITE(6,1011)
      IF (MFANIN .EQ. 1) WRITE(6,1012)
      IF (MFANIN .EQ. 2) WRITE(6,1013)
      IF (MFANOU .EQ. 1) WRITE(6,1014)
      IF (MFANOU .EQ. 2) WRITE(6,1015)
C
C***********************************************************************
C
C        COMPONENT PIPE CONNECTION INPUT SECTION
C
```

```
C***********************************************************************
C
C     HEAT LOSSES IN CONNECTING LINES
C
C        QSUCLN- RATE OF HEAT GAIN IN COMPRESSOR SUCTION LINE (BTU/H)
C        QDISLN- RATE OF HEAT LOSS IN COMPRESSOR DISCHARGE LINE (BTU/H)
C        QLIQLN- RATE OF HEAT LOSS IN THE LIQUID LINE (BTU/H)
C
      READ(5,1002) QSUCLN,QDISLN,QLIQLN
C
C
C     LINES BETWEEN COILS AND FROM REVERSING VALVE TO COILS
C
C        DLL   - INSIDE DIAMETER OF   LIQUID LINE FROM INDOOR COIL
C                                                 TO   OUTDOOR COIL (IN)
C        XLEQLL- EQUIVALENT LENGTH OF LIQUID LINE FROM INDOOR COIL
C                                                 TO   OUTDOOR COIL (FT)
C        DLRVIC- INSIDE DIAMETER OF   VAPOR LINE  FROM REVERSING VALVE
C                                                 TO   INDOOR COIL  (IN)
C        XLRVIC- EQUIVALENT LENGTH OF VAPOR LINE  FROM REVERSING VALVE
C                                                 TO   INDOOR COIL  (FT)
C        DLRVOC- INSIDE DIAMETER OF   VAPOR LINE  FROM REVERSING VALVE
C                                                 TO   OUTDOOR COIL (IN)
C        XLRVOC- EQUIVALENT LENGTH OF VAPOR LINE  FROM REVERSING VALVE
C                                                 TO   OUTDOOR COIL (FT)
C
      READ(5,1002) DLL,XLEQLL,DLRVIC,XLRVIC,DLRVOC,XLRVOC
C
C
C     LINES FROM REVERSING VALVE TO COMPRESSOR
C
C        DSLRV -  INSIDE DIAMETER OF   LINE FROM REVERSING VALVE
C                                            TO   COMPRESSOR INLET  (IN)
C        XLEQLP-  EQUIVALENT LENGTH OF LINE FROM REVERSING VALVE
```

```
C                                          TO   COMPRESSOR INLET  (FT)
C       DDLRV -  INSIDE DIAMETER OF   LINE FROM REVERSING VALVE TO
C                                          TO   COMPRESSOR OUTLET (IN)
C       XLEQHP-  EQUIVALENT LENGTH OF LINE FROM REVERSING VALVE
C                                          TO   COMPRESSOR OUTLET (FT)
C
      READ(5,1002) DSLRV,XLEQLP,DDLRV,XLEQHP
C
C
C     IDENTIFY SUCTION AND DISCHARGE LINES BASED ON MODE OF OPERATION
C
C       DSL     - INSIDE DIAMETER OF   SUCTION LINE   FROM EVAPORATOR
C                                                      TO   REVERSING
C                                                      VALVE (IN)
C       XLEQSL  - EQUIVALENT LENGTH OF SUCTION LINE   FROM EVAPORATOR
C                                                      TO   REVERSING
C                                                      VALVE (FT)
C       DDL     - INSIDE DIAMETER OF   DISCHARGE LINE FROM REVERSING
C                                                      VALVE
C                                                      TO   CONDENSER
C                                                      (IN)
C       XLEQDL  - EQUIVALENT LENGTH OF DISCHARGE LINE FROM REVERSING
C                                                      VALVE
C                                                      TO   THE CONDENSER
C                                                      (FT)
C
C   ********** ADDED BY WESTINGHOUSE R&D JAN 10 1985 ******
C
      READ(5,1007) NRVALV, TAMBRV
C
C     IDENTIFY REVERSING VALVE MODEL NUMBER AND AMBIENT TEMPERATURE
C     IF NRVALV IS 0 THE REVERSING VALVE MODEL IS BYPASSED
C        NRVALV- RANCO REVERSING VALVE MODEL NUMBER
C                0, 25, 26 OR 30
```

```
C        TAMBRV- REVERSING VALVE AMBIENT TEMPERATURE (DEG F)
C
C   ***** END OF WESTINGHOUSE R&D ADDITION OF JAN 10 1985 ***
C
      IF (NCORH .NE. 1) GO TO 70
C
C-- COOLING MODE
C
      MUNITC = 1
      MUNITE = 2
      DSL    = DLRVIC
      XLEQSL = XLRVIC
      DDL    = DLRVOC
      XLEQDL = XLRVOC
      XLEQIC = XLEQSL
      XLEQOC = XLEQDL
      GO TO 80
C
C-- HEATING MODE
C
   70 NCORH = 2
      MUNITE = 1
      MUNITC = 2
      DSL    = DLRVOC
      XLEQSL = XLRVOC
      DDL    = DLRVIC
      XLEQDL = XLRVIC
      XLEQIC = XLEQDL
      XLEQOC = XLEQSL
   80 CONTINUE
C
      WRITE(6,1203) QSUCLN,QDISLN,QLIQLN
      WRITE(6,1201) DLL,XLEQLL
      WRITE(6,1202) DLRVIC, DLRVOC, XLEQIC, XLEQOC,
```

```
     1               DSLRV,  DDLRV,  XLEQLP, XLEQHP
C
C     CONVERT DIAMETERS FROM INCHES TO FEET
C
      DLL = DLL/12.
      DSL = DSL/12.
      DDL = DDL/12.
      DLRVIC = DLRVIC/12.
      DLRVOC = DLRVOC/12.
      DDLRV = DDLRV/12.
      DSLRV = DSLRV/12.
C
C     CONVERT EQUIVALENT LENGTHS TO NON-DIMENSIONAL RATIOS
C
      XLEQLL = XLEQLL/DLL
      XLEQLP = XLEQLP/DSLRV
      XLEQHP = XLEQHP/DDLRV
      XLEQSL = XLEQSL/DSL
      XLEQDL = XLEQDL/DDL
C
C ****** ADDED BY WESTINGHOUSE R&D JAN 10 1985 *****
C        WRITE REVERSING VALVE MODEL NUMBER AND AMBIENT TEMPERATURE
C
      IF(NRVALV .GT. 1) WRITE(6,1205) NRVALV, TAMBRV
      IF(NRVALV .EQ. 1) WRITE(6,1206)
C
C ****** END OF WESTINGHOUSE ADDITION OF JAN 10 1985 *****
C
C      ** ADDED AT WESTINGHOUSE R&D ON 2 AUGUST, 1985 - R. LUCHETA **
C
      READ  (5,1016) NRVLK,
     1      (RVLKP(INDEX), INDEX = 1, NLKTB(NRVLK+1), 1)
      GO TO (461, 462, 462, 462) (NRVLK+1)
C      ## INVALID NRVLK ##
```

```
      WRITE (NOTTT,1033) NRVLK
      NRVLK = 0
      GO TO 461
C
461   CONTINUE
C       NO VALVE LEAKAGE
      WRITE (NOTPR,1031)
      GO TO 469
C
462   CONTINUE
C       ** SOME SORT OF VALVE LEAKAGE LAW ASSUMED (SEE PROGRAM
C          FLKRV FOR DETAILS).
      WRITE (NOTPR,1032) NRVLK,
     1       (INDEX, RVLKP(INDEX), INDEX = 1, NLKTB(NRVLK+1), 1)
      GO TO 469
C
469   CONTINUE
C
C       ## END OF 8/2/85 ADDITIONS ##
C
C
C  ****** ADDED AT WESTINGHOUSE R&D ON 15 MAY, 1985 BY R. LUCHETA *****
C
      READ (5,1016) NACCU,
     1       (ACPAR(INDEX), INDEX = 1, NACTB(NACCU+1), 1)
      GO TO (451,452,452,452,452,452,452,452,452,452,452,452) (NACCU+1)
      WRITE (6,1019) NACCU
      NACCU = 0
      GO TO 451
C
451   CONTINUE
C           ** NO SUCTION LINE ACCUMULATOR - NACCU = 0 **
      WRITE (6,1017)
      GO TO 459
```

```
C
452   CONTINUE
C           ** NACCU BETWEEN 1 AND 11 **
      WRITE (6,1018) NACCU,
     1      (INDEX, ACPAR(INDEX), INDEX = 1, NACTB(NACCU+1),1 )
      GO TO 459
C
459   CONTINUE

C
C*************************************************************************
C
C       WRITE ITERATION LOOP CONVERGENCE TOLERANCES SPECIFIED IN BLOCK
C       DATA
C
C*************************************************************************
C
      WRITE(6,1501) AMBCON, CMPCON, TOLH,
     1              CNDCON, FLOCON, TOLS,
     2              EVPCON, CONMST
C
      RETURN
C
 1001 FORMAT(20A4)
 1002 FORMAT(8F10.4)
 1003 FORMAT(8I10)
 1004 FORMAT(' ***** INPUT DATA *****',/,9X,20A4/)
 1005 FORMAT('         COOLING MODE OF OPERATION')
 1006 FORMAT('         HEATING MODE OF OPERATION')
C
C    ***** FORMAT 1007 ADDED BY WESTINGHOUSE R&D JAN 10 1985 ***
C
 1007 FORMAT(I10,F10.3)
C
```

```
C     ***** FORMATS 1016-1018 ADDED AT WESTINGHOUSE R&D 15 MAY 1985
C           BY R. LUCHETA *****
C
1016  FORMAT ( I4: 5F12.4 :/ 3( 6F12.4 :/ ) )
 1017 FORMAT ( / '     >>  NO SUCTION LINE ACCUMULATOR << ' / )
 1018 FORMAT ( / '   SUCTION LINE ACCUMULATOR DESIGNATOR = ', I4:
     1      '; PARAMETERS ARE: ' /
     2      5( 1X, T4, 4('ACPAR(', I2, ') = ', 1PG12.4: ';  ') / ) )
 1019 FORMAT ( 1X, T4, '>> INVALID NACCU = ', I4, '; ZERO ASSUMED <<' )
C     ***** END OF 15 MAY 1985 ADDITION *****
C
C       ## CHANGES MADE ON 8/2/85 AT WESTINGHOUSE R&D - R. LUCHETA ##
C
1031  FORMAT ( / '     >>  NO REVERSING VALVE LEAKAGE  << ' / )
1032  FORMAT ( / '   REVERSING VALVE LEAKAGE DESIGNATOR = ', I4:
     1      '; PARAMETERS ARE: ' /
     2      5( 1X, T4, 4('RVLKP(', I2, ') = ', 1PG12.4: ';  ') / ) )
1033  FORMAT ( 1X, T4, '>> INVALID RVLKP = ', I4, '; ZERO ASSUMED <<' )
C
C       ## END OF 8/2/85 ADDITION
C
 1020 FORMAT('         MINIMUM OUTPUT')
 1021 FORMAT('         SUMMARY OUTPUT')
 1022 FORMAT('         OUTPUT AFTER EACH INTERMEDIATE ITERATION',
     1              ' CONVERGES')
 1023 FORMAT('         CONTINUOUS OUTPUT DURING',
     1              ' INTERMEDIATE ITERATIONS')
 1024 FORMAT('         REFRIGERANT CHARGE IS NOT SPECIFIED'/)
 1025 FORMAT('         COMPRESSOR INLET SUPERHEAT IS SPECIFIED AT  ',
     1               F6.2,' F')
 1026 FORMAT('         COMPRESSOR INLET QUALITY   IS SPECIFIED AT  ',
     1               F6.4)
 1027 FORMAT('         SPECIFIED REFRIGERANT CHARGE IS             ',
     1       F6.2,' LBM',/)
```

```
1028 FORMAT('          ESTIMATE OF COMPRESSOR INLET SUPERHEAT  IS  ',
   1                   F6.2,' F')
1029 FORMAT('          ESTIMATE OF COMPRESSOR INLET QUALITY    IS  ',
   1                   F6.4)
1010 FORMAT(9X,'COMPRESSOR CAN HEAT LOSS ADDED TO AIR BEFORE CROSSING',
   1           ' THE OUTDOOR COIL.')
1011 FORMAT(9X,'COMPRESSOR CAN HEAT LOSS ADDED TO AIR AFTER  CROSSING',
   1           ' THE OUTDOOR COIL.')
1012 FORMAT(9X,'POWER TO THE INDOOR  FAN ADDED TO AIR BEFORE CROSSING',
   1           ' THE INDOOR  COIL.')
1013 FORMAT(9X,'POWER TO THE INDOOR  FAN ADDED TO AIR AFTER  CROSSING',
   1           ' THE INDOOR  COIL.')
1014 FORMAT(9X,'POWER TO THE OUTDOOR FAN ADDED TO AIR BEFORE CROSSING',
   1           ' THE OUTDOOR COIL.')
1015 FORMAT(9X,'POWER TO THE OUTDOOR FAN ADDED TO AIR AFTER  CROSSING',
   1           ' THE OUTDOOR COIL.')
C
1101 FORMAT(I10,7F10.4)
1102 FORMAT('ODATAIN: ***** IREFC = ',I2,' IS NOT ALLOWED. ',
   1                 'IREFC IS SET TO 0 TO FIX CONDENSER EXIT ',
   2                 'SUBCOOLING *****',/)
1103 FORMAT(1H0,8X,'THERMAL EXPANSION VALVE USED TO REGULATE FLOW:',/,
   1           11X,'RATED TXV CAPACITY          ',F8.3,' TONS',/,
   2           11X,'STATIC SUPERHEAT RATING     ',F8.3,' F',/,
   3           11X,'RATED OPERATING SUPERHEAT   ',F8.3,' F',/,
   4           11X,'BLEED FACTOR                ',F8.3)
1104 FORMAT(11X,'NOZZLE AND TUBE PRESSURE DROPS ARE BYPASSED')
1105 FORMAT(11X,'NOZZLE AND TUBE PRESSURE DROPS ARE INCLUDED')
1106 FORMAT(1H0,8X,I1,' CAPILLARY TUBE(S) USED TO REGULATE FLOW',/,
   1            9X,'FLOW FACTOR FOR CAPILLARY TUBE(S)    ',F10.4)
1107 FORMAT(1H0,8X,'ORIFICE USED TO REGULATE FLOW',/,9X,
   1        'ORIFICE DIAMETER    ',F10.4,' IN')
1108 FORMAT('0         CONDENSER  EXIT SUBCOOLING IS SPECIFIED AT  ',
   1                   F6.2,' F')
```

```
 1109 FORMAT('0        CONDENSER  EXIT QUALITY    IS SPECIFIED AT  ',
     1        F6.4)
C
 1201 FORMAT(1H0,8X,'DESCRIPTION OF CONNECTING TUBING:',/,
     1      9X,'  LIQUID LINE FROM INDOOR TO OUTDOOR HEAT EXCHANGER',/,
     2      9X,'      ID                       ',F8.5,' IN',/,
     3      9X,'      EQUIVALENT LENGTH       ',F8.2,' FT')
 1202 FORMAT(9X,'  FROM INDOOR COIL TO REVERSING VALVE',13X,
     1         'FROM OUTDOOR COIL TO REVERSING VALVE',/,
     2     2(9X,'      ID                       ',F8.5,' IN '),/,
     3     2(9X,'      EQUIVALENT LENGTH       ',F8.2,' FT '),/,
     4       9X,'  FROM REVERSING VALVE TO COMPRESSOR INLET',8X,
     5          'FROM REVERSING VALVE TO COMPRESSOR OUTLET',/,
     6     2(9X,'      ID                       ',F8.5,' IN '),/,
     7     2(9X,'      EQUIVALENT LENGTH       ',F8.2,' FT '))
 1203 FORMAT(1H0,8X,'HEAT GAIN IN SUCTION LINE    ',F19.1,' BTU/H',
     1     /,9X,'HEAT LOSS IN DISCHARGE LINE  ',F19.1,' BTU/H',
     2     /,9X,'HEAT LOSS IN LIQUID LINE     ',F19.1,' BTU/H')
 1204 FORMAT(1H0,8X,'ESTIMATE OF:',/,
     1 11X,'SATURATION TEMPERATURE INTO   COMPRESSOR  ',F6.2,' F',/,
     2 11X,'SATURATION TEMPERATURE OUT OF COMPRESSOR  ',F6.2,' F')
C
C  **** FORMAT 1205 ADDED BY WESTINGHOUSE R&D JAN 10 1985 ****
 1205 FORMAT(1H0,8X,'REVERSING VALVE CHARACTERISTICS:',/,
     1        11X,'RANCO MODEL NUMBER          ',I7,
     2        11X,'AMBIENT TEMPERATURE         ',F7.3,' F')
C  ***** FORMAT 1206 ADDED BY WESTINGHOUSE R&D JAN 23, 1985****
 1206 FORMAT(1H0,8X,'EMPIRICAL REVERSING VALVE MODEL',//)
C
 1302 FORMAT('ODATAIN: ***** ICOMP = ',I2,' IS NOT ALLOWED, ',
     1       'ICOMP SET TO 1 TO USE EFFICIENCY & LOSS MODEL *****',/)
 1304 FORMAT(1H0,8X,'COMPRESSOR CHARACTERISTICS:',/,
     1         11X,'TOTAL PISTON DISPLACEMENT  ',F7.3,' CU IN',
     2          5X,'ISENTROPIC EFFICIENCY      ',F5.3,/,
```

```
     3          11X,'SYNCHRONOUS MOTOR SPEED    ',F7.2,' RPM',
     4           7X,'MAXIMUM MOTOR EFFICIENCY   ',F5.3,/,
     5          11X,'CLEARANCE VOLUME RATIO     ',F7.4,
     6          11X,'MECHANICAL EFFICIENCY      ',F5.3)
 1305 FORMAT('0          HEAT REJECTED FROM COMPRESSOR SHELL IS ',F5.3,
     1              ' TIMES THE COMPRESSOR POWER')
 1306 FORMAT(1H0,10X,'HEAT REJECTION RATE FROM THE SHELL IS SET TO ',
     1           '0.90*(1.0 - MOTOR*MECHANICAL EFFICIENCY)*',
     2           'COMPRESSOR POWER')
 1307 FORMAT(1H0,10X,'HEAT REJECTION RATE FROM SHELL  ',F8.2,' BTU/H')
 1308 FORMAT(11X,'HEAT TRANSFER RATE FROM DISCHARGE LINE TO THE ',
     1           'INLET GAS IS ',F4.2,' TIMES THE COMPRESSOR POWER')
 1309 FORMAT(11X,'HEAT TRANSFER RATE FROM DISCHARGE LINE TO THE ',
     1           'INLET GAS IS SET TO 0.03*COMPRESSOR POWER')
 1310 FORMAT(11X,'HEAT TRANSFER RATE FROM DISCHARGE LINE TO THE ',
     1           'INLET GAS IS ',F8.2,' BTU/H')
 1311 FORMAT(11X,'MOTOR OUTPUT AT FULL LOAD IS CALCULATED')
 1312 FORMAT(11X,'MOTOR OUTPUT AT FULL LOAD IS FIXED AT ',F8.3,' KW')
 1313 FORMAT(8E10.3)
 1314 FORMAT(1H0,8X,'COMPRESSOR CHARACTERISTICS:',/,
     1          11X,'TOTAL DISPLACEMENT     ',16X,F9.3,' CUBIC INCHES',
     2        /,11X,'GIVEN MOTOR SPEED      ',16X,F9.3,' RPM')
 1315 FORMAT(1H0,8X,'MAP-BASED COMPRESSOR INPUT:',/,
     &11X,'POWER CONSUMPTION=    ',1PE10.3,'*CONDENSING  TEMPERATURE**2'
     &                      ,' + ',1PE10.3,'*CONDENSING  TEMPERATURE',
     &                 /,30X,' + ',1PE10.3,'*EVAPORATING TEMPERATURE**2'
     &                      ,' + ',1PE10.3,'*EVAPORATING TEMPERATURE',
     &                 /,30X,' + ',1PE10.3,'*CONDENSING  TEMPERATURE*',
     &                                     'EVAPORATING TEMPERATURE',
     &                       ' + ',1PE10.3)
 1316 FORMAT(1H0,10X,
     &     'MASS FLOW RATE=        ',1PE10.3,'*CONDENSING  TEMPERATURE**2'
     &                      ,' + ',1PE10.3,'*CONDENSING  TEMPERATURE',
     &                 /,30X,' + ',1PE10.3,'*EVAPORATING TEMPERATURE**2'
```

```
     &                            ,' + ',1PE10.3,'*EVAPORATING TEMPERATURE',
     &                      /,30X,' + ',1PE10.3,'*CONDENSING  TEMPERATURE*',
     &                                             'EVAPORATING TEMPERATURE',
     &                             ' + ',1PE10.3)
 1317 FORMAT(1H0,10X,'BASE SUPERHEAT FOR COMPRESSOR MAP          ',
     &             F6.3,' F',/,
     &             11X,'BASE DISPLACEMENT FOR COMPRESSOR MAP       ',
     &             F6.3,' CU IN')
 1318 FORMAT(1H0,10X,'RETURN GAS TEMPERATURE FOR COMPRESSOR MAP ',
     &             F6.3,' F',/,
     &             11X,'BASE DISPLACEMENT FOR COMPRESSOR MAP       ',
     &             F6.3,' CU IN')
 1319 FORMAT(1H0,10X,'SUPERHEAT CORRECTION TERMS (SET IN BLOCK DATA):',
     &           /,11X,'  SUCTION GAS HEATING FACTOR               ',
     &          1X,F6.3,/,
     &             11X,'  VOLUMETRIC EFFICIENCY CORRECTION FACTOR',
     &          1X,F6.3)
C
 1400 FORMAT(2F10.4,I10)
 1403 FORMAT('0        INDOOR UNIT: EVAPORATOR')
 1404 FORMAT('0        OUTDOOR UNIT: CONDENSER')
 1405 FORMAT('0        INDOOR UNIT: CONDENSER')
 1406 FORMAT('0        OUTDOOR UNIT: EVAPORATOR')
 1407 FORMAT(11X,'INLET AIR TEMPERATURE                       ',F8.3,' F',
     &    9X,'RELATIVE HUMIDITY                          ',F8.5)
C
 1408 FORMAT(11X,'FRONTAL AREA OF HX                          ',F8.3,
     &' SQ FT',5X,'SMOOTH FINS                               ')
 1409 FORMAT(11X,'FRONTAL AREA OF HX                          ',F8.3,
     &' SQ FT',5X,'WAVY FINS                                 ')
 1509 FORMAT(11X,'FRONTAL AREA OF HX                          ',F8.3,
     &' SQ FT',5X,'CORRUGATED FINS                           ',/,
     &         11X,'FIN PATTERNS PER TUBE PITCH               ',I8,
     &         11X,'FIN PATTERN DEPTH (PEAK TO VALLEY)        ',F8.4,
```

```
     & ' IN')
 1410 FORMAT(11X,'FRONTAL AREA OF HX                              ',F8.3,
     &' SQ FT',5X,'LOUVERED FINS                                  ')
C
 1411 FORMAT(11X,'AIR FLOW RATE                                   ',F8.2,' CFM',
     &   7X,'COMBINED FAN - FAN MOTOR EFFICIENCY     ',F8.5)
 1412 FORMAT(11X,'AIR FLOW RATE                                   ',F8.2,' CFM',
     &   7X,'SPECIFIED FAN POWER                     ',F8.2,' WATTS')
 1413 FORMAT(11X,'AIR FLOW RATE                                   ',F8.2,' CFM',
     &   7X,'FAN MOTOR EFFICIENCY (SET IN BLOCK DATA)',F8.5)
 1414 FORMAT(11X,'NUMBER OF TUBES IN DIRECTION OF AIR FLOW',F8.2,
     &  11X,'FIN PITCH                               ',F8.2,' FINS/IN',/
     &  11X,'NUMBER OF PARALLEL CIRCUITS             ',F8.2,
     &  11X,'FIN THICKNESS                           ',F8.5,' IN',/,
     &  11X,'OD OF TUBES IN HX                       ',F8.5,' IN',
     &   8X,'THERMAL CONDUCTIVITY: FINS              ',F8.2,
     &      ' BTU/H-FT-F',/,
     &  11X,'ID OF TUBES IN HX                       ',F8.5,' IN',
     &   8X,'THERMAL CONDUCTIVITY: TUBES             ',F8.2,
     &      ' BTU/H-FT-F',/,
     &  11X,'HORIZONTAL TUBE SPACING                 ',F8.3,' IN',
     &   8X,'FRACTION OF COMPUTED CONTACT CONDUCTANCE',F8.3,/,
     &  11X,'VERTICAL TUBE SPACING                   ',F8.3,' IN',
     &   8X,'NUMBER OF RETURN BENDS                  ',F8.2,/)
 1415 FORMAT(11X,'ID OF EACH OF 6 EQUIVALENT DUCTS          ',F8.2,' IN',
     &   8X,'HOUSE LOAD TO BE MET BY INDOOR UNIT     ',F8.1,' BTU/H')
 1416 FORMAT('0          OUTDOOR UNIT: FAN',/,
     &11X,'FAN STATIC EFFICIENCY =     ',1PE10.3,
     &                      ' + ',1PE10.3,' *  LOG10(S/1000.)',
     &                      ' + ',1PE10.3,' * (LOG10(S/1000.))**2',
     &/,11X,'WHERE',
     &/,13X,'SPECIFIC SPEED (S)  =    FAN RPM * (AIR FLOW CFM**0.5) /',
     &' (COIL STATIC PRESSURE DROP**0.75)',
     &/,11X,'AND',
```

```
     &/,13X,'A FAN SPEED OF      ',OPF8.1,' RPM IS ASSUMED',/)
C
 1501 FORMAT('OITERATION TOLERANCES:',/,
     &        9X,'AMBCON  ',F6.3,' F',5X,'CMPCON  ',F6.3,' BTU/LBM',5X,
     &           'TOLH    ',F7.5,' BTU/LBM',/,
     &        9X,'CNDCON  ',F6.3,' F',5X,'FLOCON  ',F6.3,' LBM/H',7X,
     &           'TOLS    ',F7.5,' BTU/LBM-R'/,
     &        9X,'EVPCON  ',F6.3,' F',5X,'CONMST  ',F6.3,' F'///)
      END
```

Listing of the CNDNSR Subroutine

A listing of the CNDNSR subroutine follows:

```
      FUNCTION CNDNSR(TINPUT,IERR)
C
C       CNDNSR(TEMPERATURE) = CDTROC(TEMPERATURE) - DTROC OR
C                           = XMRFLD(TEMPERATURE) - XMR(TEMPERATURE)
C
C       CNDNSR IS USED WITH THE ROOT SOLVER 'ZERO' TO FIND THE
C              SATURATION TEMPERATURE OUT OF THE COMPRESSOR, TSOCMP,
C              SO THAT EITHER
C              THE CALCULATED SUBCOOLING OUT OF THE CONDENSER IS NEARLY
C              EQUAL TO THE SPECIFIED SUBCOOLING:
C              I.E. ABS(CDTROC(TSATCI) - DTROC) < CNDCON.
C              OR
C              THE FLOW CONTROL  MASS FLOW RATE IS NEARLY
C              EQUAL TO THAT OF THE COMPRESSOR
C              I.E. ABS[XMRFLD(TSATCI) - XMR(TSATCI)] < FLOCON.
C
C       'ZERO' CONTAINS ALL OF THE LOGIC NECESSARY TO ITERATE
C              TO A ROOT, TSATCI.
C
      LOGICAL PRINT
      REAL NTC,NTE,NSECTC,NSECTE
C
      COMMON / A1 / PRINT
C
      COMMON / CMPRSR / TRICMP, TSICMP, HINCMP, PINCMP, XINCMP,
     &                  TROCMP, TSOCMP, HOUCMP, POUCMP, XOUCMP
C
      COMMON / COMPR / VR,     SYNC,   FLMOT,  EFFMMX, ETAISN, ETAMEC,
```

```
     &                   ETAVLA, ETAVLB, POW,    CANFAC, HILOFC, QCAN,
     &                   QHILO,  DISPL, MTRCLC
C
      COMMON / CONDSR / TAIIC,  TIC, TSATCI, HIC, PIC, XIC,
     &                  TAOC,  TROC, TSATCO, HOC, POC, XOC
C
      COMMON / EVAPOR / DEAE,   DERE, DELTAE,    FPE, XKFE, XKTE, AAFE,
     &                   NTE, NSECTE, HCONTE,    STE,    WTE,  SIGAE,
     &                    PE,  ARFTE,  ARHTE, ALFARE, ALFAAE,   FARE,
     &                  CARE,    QAE, RTBEVP,    DZE, FANEFE,   RHIE,
     &                  FINTYE, MUNITE
C
      COMMON / FLOWBA / DTROC, SUPER, CAPFLO, ORIFD, XMR, NCAP,
     &                  IREFC, ICOMP, ITRPIE
C
      COMMON / MPASS / CNDCON, AMBCON, EVPCON, CONMST, CMPCON, FLOCON,
     &                 TOLS,   TOLH,   LPRINT, NCORH,  MCMPOP, MFANIN,
     &                 MFANOU, MFANFT
C
      COMMON / TIME / CONTIM, EVATIM, APFTIM, SSTIM,  OPTTIM, DATTIM,
     &                HITIM,  CMPTIM, EVPTIM, TIMLO,  EXCTIM, AMBTIM,
     &                IOV1HD, IOVOHD, ISTART
C
      TSOCMP = TINPUT
      CNDNSR = 1.0E+10
      IERR = 0
C
C     CALL SUBROUTINE COMP TO DETERMINE THE COMPRESSOR
C     PERFORMANCE AND REFRIGERANT FLOW RATE 'XMR'
C
      IO = ICLOCK(0)
      IOHI = IO
      IF(ICOMP.EQ.1) CALL COMPV(TOLS,TOLH,CMPCON)
      IF(ICOMP.EQ.2) CALL COMPMP(TOLS,TOLH,CMPCON)
```

```
      I1 = ICLOCK(0)
      CMPTIM = CMPTIM + (I1 - I0)/100.
C
      IF(PRINT) WRITE(6,1001) XMR,TSATCI, TIC
C
      I0 = ICLOCK(0)
      CALL COND(XMR,QCAN,IERR)
      I1 = ICLOCK(0)
      CONTIM = CONTIM + (I1 - I0)/100.
      IF (IERR .EQ. 1) GO TO 200
C
      IF(IREFC.NE.0) GO TO 100
C
      CDTROC = TSATCO - TROC
      CDTRC = CDTROC
      IF (XOC .GT. 0.) CDTRC = -200.0*XOC
      DTRC = DTROC
      IF (DTROC .LT. 0.) DTRC = 200.*DTROC
C
      CNDNSR = CDTRC - DTRC
C
      IF (.NOT. PRINT) GO TO 200
      WRITE(6,1002) TSATCI,TSATCO,TROC
C
      IF(DTROC.LT.0.0) GO TO 1
      WRITE(6,1009) DTROC
      GO TO 2
    1 CONTINUE
      SXOC = -DTROC
      WRITE(6,1010) SXOC
    2 CONTINUE
      IF(XOC.GT.0.0) GO TO 3
      WRITE(6,1011) CDTROC
      GO TO 4
```

```
    3 CONTINUE
      WRITE(6,1012) XOC
    4 CONTINUE
C
      IF (ABS(CNDNSR) .GT. CNDCON) WRITE(6,1003) TSATCI
      GO TO 200
C
  100 CONTINUE
      CALL FLOBAL(NSECTE,XMRFLD)
C
      CNDNSR = ( XMRFLD - XMR ) / 20.
C
      IF(.NOT. PRINT) GO TO 200
      WRITE(6,1004) TSATCI,TSATCO,TROC,XMRFLD,XMR
      IF(ABS(CNDNSR).GT.(FLOCON/20.)) WRITE(6,1005) TSATCI
  200 CONTINUE
      I1 = ICLOCK(0)
      HITIM = HITIM + (I1 - IOHI)/100.
      RETURN
 1001 FORMAT('OCNDNSR: DATA SENT TO COND',/,
     &       10X,'REFRIGERANT MASS FLOW RATE                   ',F10.3,
     &' LBM/H',/,10X,'SATURATION  TEMPERATURE ENTERING CONDENSER  ',
     &F10.3,' F',/,10X,'REFRIGERANT TEMPERATURE ENTERING CONDENSER  ',
     &F10.3,' F')
 1002 FORMAT('OCNDNSR: TEST FOR CONVERGENCE ON CONDENSER EXIT ',
     &       'SUBCOOLING',/,
     &       10X,'SATURATION  TEMPERATURE ENTERING CONDENSER  ',F10.3,
     &' F',/,10X,'SATURATION  TEMPERATURE LEAVING  CONDENSER  ',F10.3,
     &' F',/,10X,'REFRIGERANT TEMPERATURE LEAVING  CONDENSER  ',F10.3,
     &' F')
 1009 FORMAT(10X,'SPECIFIED   SUBCOOLING                      ',F10.3,
     &' F')
 1010 FORMAT(10X,'SPECIFIED   QUALITY                         ',F10.4)
 1011 FORMAT(10X,'CALCULATED  SUBCOOLING                      ',F10.3,
```

```
   &' F')
1012 FORMAT(10X,'CALCULATED  QUALITY                             ',F10.4)
1003 FORMAT('OCNDNSR: DID NOT CONVERGE ON SPECIFIED "SUBCOOLING" ',
   &                 'FOR TSATCI OF ',F10.3,' F')
1004 FORMAT('OCNDNSR: TEST FOR CONVERGENCE ON MASS FLOW RATE',/,
   &       10X,'SATURATION  TEMPERATURE ENTERING CONDENSER  ',F10.3,
   &' F',/,10X,'SATURATION  TEMPERATURE LEAVING  CONDENSER  ',F10.3,
   &' F',/,10X,'REFRIGERANT TEMPERATURE LEAVING  CONDENSER  ',F10.3,
   &' F',/,10X,'FLOW CONTROL MASS FLOW RATE                 ',F10.3,
   &' LBM/H',/,10X,'COMPRESSOR   MASS FLOW RATE              ',
   &F10.3,' LBM/H')
1005 FORMAT('OCNDNSR: DID NOT CONVERGE ON MASS FLOW RATE ',
   &                 'FOR TSATCI OF ',F10.3,' F')
    END
```

Listing of the COND Subroutine

A listing of the COND subroutine follows:

```
      SUBROUTINE COND(XMR,QCAN,IERR)
C
C**** CONDENSER SIMULATION ROUTINE ****
C
C     PURPOSE
C
C       TO COMPUTE CONDENSER PERFORMANCE FOR A CROSS-FLOW,
C       REFRIGERANT-TO-AIR, TUBE AND PLATE-FIN TYPE HEAT EXCHANGER
C
C     ADAPTED FROM SUBROUTINE WRITTEN BY
C       C. C. HILLER AND L. R. GLICKSMAN,
C       REPORT NO. 24525-96,
C       HEAT TRANSFER LABORATORY,
C       MASSACHUSETTS INSTITUTE OF TECHNOLOGY.
C
C     OUTPUT
C        QC    -  TOTAL HEAT TRANSFER RATE (BTU/H)
C        QFANC-  CONDENSER FAN-MOTOR ENERGY CONSUMPTION (BTU/H)
C        TAIC -  AVERAGE AIR TEMPERATURE ENTERING CONDENSER COIL (F)
C        TAOC -  AVERAGE AIR TEMPERATURE LEAVING CONDENSER COIL (F)
C        TAOOC-  AVERAGE AIR TEMPERATURE LEAVING CONDENSER UNIT (F)
C        TROC -  TEMPERATURE OF REFRIGERANT LEAVING CONDENSER (F)
C        HOC   -  ENTHALPY OF REFRIGERANT LEAVING CONDENSER (BTU/LBM)
C
C     REMARKS
C     THIS SUBROUTINE CALLS:
C        EXCH   TO DETERMINE THE OVERALL HEAT EXCHANGER PERFORMANCE,
C               HEAT TRANSFER RATES, AIR TEMPERATURES, ETC.
C        CHTC   TO DETERMINE THE CONDENSATION TWO-PHASE HEAT TRANSFER
```

```
C             COEFFICIENT FOR FORCED CONVECTION CONDENSATION INSIDE
C             TUBES
C       SPHTC  TO DETERMINE REFRIGERANT GAS HEAT TRANSFER COEFFICIENTS
C       SPHTC2 TO DETERMINE REFRIGERANT LIQUID HEAT TRANSFER
C             COEFFICIENTS
C   ## HAIR CHANGED TO HAIR2 BY WESTINGHOUSE R&D 9/27/85 - TJ FAGAN ##
C       HAIR2  TO DETERMINE THE AIR-SIDE HEAT TRANSFER COEFFICIENT
C       SEFF   TO DETERMINE OVERALL SURFACE EFFECTIVENESS OF A
C             FIN-AND-TUBE SURFACE
C       PDROP  TO DETERMINE REFRIGERANT-SIDE PRESSURE DROPS
C       PDAIR  TO DETERMINE AIR-SIDE PRESSURE DROPS
C       FANFIT TO DETERMINE OUTDOOR FAN EFFICIENCY
C       XMOIST TO DETERMINE THE PSYCHROMETRIC PROPERTIES OF MOIST AIR
C       MUKCPA TO DETERMINE THE THERMOPHYSICAL PROPERTIES OF AIR
C       MUKCP  TO DETERMINE THERMOPHYSICAL PROPERTIES OF
C             REFRIGERANT LIQUID AND SATURATED VAPOR
C       SPFHT  TO DETERMINE SPECIFIC HEAT OF SUPERHEATED
C             REFRIGERANT VAPOR
C       SATPRP TO DETERMINE SATURATION THERMODYNAMIC PROPERTIES
C       TSAT   TO DETERMINE SATURATION TEMPERATURES CORRESPONDING
C             TO GIVEN PRESSURES
C       VAPOR  TO DETERMINE THERMODYNAMIC PROPERTIES OF SUPERHEATED
C             REFRIGERANT VAPOR
C
C      ------------------------------------------------------------
C
      LOGICAL PRINT
      INTEGER FLAG
      REAL NTC,NSECTC
C
      COMMON / AIR / PA, CPA, CPM, RAU, AFILTR, AHEATR, RACKS
C
      COMMON / A1 / PRINT
C
```

```
      COMMON / CNDMSG / IERCND(2)
C
      COMMON / CONDEN /  DEAC,   DERC, DELTAC,    FPC, XKFC, XKTC, AAFC,
     &                    NTC, NSECTC, HCONTC,    STC,    WTC,  SIGAC,
     &                     PC,   ARFTC,  ARHTC, ALFARC,ALFAAC,   FARC,
     &                   CARC,     QAC, RTBCND,    DZC,FANEFC,   RHIC,
     &                 FINTYC, MUNITC
C
      COMMON / CONDS / HAC, SEFFXC, XMAC,    QC, PDAIRC, PDC,
     &                HSPC,   QSPC, FSPC, CPSPC, CPSP,
     &                HTPC,   QTPC, FTPC,
     &                HSCC,   QSCC, FSCC, CPSCC,
     &               TRVDS
C
      COMMON / CONDSR / TAIIC,  TIC, TSATCI, HIC, PIC, XIC,
     &                  TAOC,  TROC, TSATCO, HOC, POC, XOC
C
      COMMON / LINES / DLL, XLEQLL, DSL,   XLEQSL, DDL, XLEQDL,
     &               DSLRV, XLEQLP, DDLRV, XLEQHP,DPDL,   DPSL,
     &                DPLL, QDISLN,QSUCLN, QLIQLN,E
C
      COMMON / MPASS / CNDCON, AMBCON, EVPCON, CONMST, CMPCON,
     &                 FLOCON, TOLS,   TOLH,   LPRINT, NCORH,
     &                 MCMPOP, MFANIN, MFANOU, MFANFT
C
      COMMON / PRNT3 / TAIC, TAOOC, QFANC
C
      COMMON / PRNT8 / EINDF, EOUTF, ECOMP, RESIST, COP, DP,SS,
     &                 COPHP, QAIR,  FANOUT
C
      COMMON / UAS / CUAVR , CUATPR, CUASCR, CUAVA,  CUATPA, CUASCA,
     &              CUAVC,  CUATPC, CUASCC, CUAV,   CUATP,  CUASC,
     &              EUAVR,  EUATPR, EUAVA , EUATPA, EUAVC,  EUATPC,
     &              EUAV,    EUATP, UATPAW, UATPCW, UATPW
```

```
C
C  #### ADDED BY WESTINGHOUSE R&D 9/27/85 - T. J. FAGAN
C
      COMMON / FINPAT / NFPE, FPDE, XFPE, XAPE, NFPC, FPDC, XFPC, XAPC
C
C    ####### END OF 9/27/85 ADDITION ######
C
C
      COMMON  DDUCT, FIXCAP, ITITLE(20)
C
      DO 10 I=1,2
   10 IERCND(I) = 0
C
C        SET PRESSURE DROP IN CONDENSER, PDC, TO 0.0 TO INDICATE THAT A
C        PRESSURE DROP HAS NOT BEEN CALCULATED.
C
      PDC = 0.0
C
C        INITIALIZE VALUES
C
      XOC = 0.0
      ITEREX = 0
      SBCOOL = 0.0
      QFANC = 0.
      TAIC = TAIIC
      TAOC = TAIIC
      TSTTPO = TSATCI
      TSTTPI = TSATCI
      TSATCO = TSATCI
      TSAVG = TSATCI
      TROC = TSATCI
      PTPO=PIC
      PTPI=PIC
      TRVDS = TIC
```

```
      XNTU = 1.0
C
      IF(PRINT) WRITE(6,1001) TAIIC,QAC,TIC,TSATCI
C
      RHOA = 144.0*PA/(RAU*(TAIIC + 459.7))
      VA = QAC*60.0/(SIGAC*AAFC)
C
C   ####### ADDED BY WESTINGHOUSE R&D 9/27/85 - T. J. FAGAN
C
      UAF = QAC/AAFC
C
C   ###### END OF 9/27/85 ADDITION #######
C
C        SUBDIVIDE FLOW INTO PARALLEL CIRCUITS AND TREAT EACH
C        LIKE A SEPARATE HEAT EXCHANGER - CONVERT BACK TO TOTAL
C        FLOW AT THE END
C
      XMA = 60.0*QAC*RHOA
      XMAC = XMA/NSECTC
      XMR = XMR/NSECTC
      RTBPCR = RTBCND / NSECTC
      DZRB = 1.5708 * STC * RTBPCR
C
C        MASS FLOW OF AIR IS BASED ON AMBIENT TEMPERATURE RATHER THAN
C        WHAT TEMPERATURE OF AIR IS ACTUALLY CROSSING THE FAN
C
      GA = VA*RHOA
      GR = XMR/ARFTC
      AOM = NTC*PC/(STC*GA*SIGAC)
      ATAMIN = ALFAAC*ARHTC/(ALFARC*AAFC*SIGAC/NSECTC)
      CALL MUKCPA(TAIIC,X,Y,Z,XX,YY,ZZ,XXX,YYY,CPA,IMUCKP)
      CALL XMOIST(TAIIC,1,RHIC,WAIRI,XX,YY,ZZ,CONMST)
C
C        ITERATION LOOP ON SUBCOOLING
```

```
C
  100 CONTINUE
C
C        CALCULATE SUBCOOLED REFRIGERANT PROPERTIES
C
      TSBAVE=(TSTTPO+TROC)/2.
      CALL MUKCP(TSBAVE,XMULSB,Y,Z,XKSB,YY,ZZ,CPSCC,YYY,ZZZ,IMUKCP)
      PRSB = XMULSB*CPSCC/XKSB
C
C        DETERMINE SINGLE PHASE LIQUID HEAT TRANSFER COEFFICIENT
C        'HSCC' (BTU/H-SQ FT-F)
C
      CALL SPHTC2(1,DERC,GR,XMULSB,CPSCC,PRSB,RERL,HSCC)
C
C        CALCULATE'CONDENSING SUPERHEATED' REFRIGERANT PROPERTIES
C
      TRVAL=TRVDS
      IF(XIC.NE.1.0) TRVAL=TSTTPI+0.05
      TSUPAV=(TRVAL+TSTTPI)/2.
      CALL SPFHT(TSUPAV,PTPI,X,CPSPC,Y)
C
C        CALCULATE SUPERHEATED REFRIGERANT PROPERTIES
C
      TRVAL=TRVDS
      IF(XIC.NE.1.0.OR.TRVDS.EQ.TIC) TRVAL=TIC+0.05
      TSUPAV=(TRVAL+TIC)/2.
      CALL SPFHT(TSUPAV,PIC,X,CPSP,Y)
      CALL MUKCP(TSUPAV,X,Y,XMUVSP,XX,YY,XKSP,XXX,YYY,ZZZ,IMUKCP)
      PRSP=XMUVSP*CPSP/XKSP
C
C        DETERMINE SINGLE PHASE VAPOR HEAT TRANSFER COEFFICIENT
C        'HSPC' (BTU/H-SQ FT-F)
C
      CALL SPHTC(DERC,GR,XMUVSP,CPSP,PRSP,RERV,HSPC)
```

```
C
C       CALCULATE TWO-PHASE REFRIGERANT PROPERTIES
C
      CALL SATPRP(TSTTPI,Y,XX,VG,XXX,YYY,HVTPI,AAA,BBB,FLAG)
      IF(XIC.LT.1.0) HVTPI=HIC
      CALL SATPRP(TSTTPO,X,VF,Y,HLTPO,HFGO,AAA,BBB,CCC,FLAG)
      HFG = HVTPI-HLTPO
      RHOV = 1.0/VG
      RHOL = 1.0/VF
      CALL MUKCP(TSAVG,XMURL,XMURV,XXX,XKRL,YYY,ZZZ,
     &           CPRL,AAA,BBB,IMUKCP)
      PRRL = XMURL*CPRL/XKRL
C
C       DETERMINE CONDENSATION TWO-PHASE HEAT TRANSFER COEFFICIENT 'HTP'
C       (BTU/H-SQ FT-F)
C
      CALL CHTC(DERC,GR,XIC,XOC,PRRL,XKRL,XMURV,XMURL,RHOL,
     &          RHOV,HSCC,HTPC)
C
C       ADJUST THE TWO-PHASE HEAT TRANSFER COEFFICIENT FOR THE REGION
C       WHERE THE BULK TEMPERATURE EXCEEDS THE SATURATION TEMPERATURE
C
      IF (XIC .EQ. 1.) HTPC = HTPC*(1.+CPSPC*(TRVDS-TSTTPI)/HFG)**0.25
C
C       CALCULATE AIR PROPERTIES
C
      TAIRAV = TSAVG - (TAOC - TAIC)/XNTU
      CALL MUKCPA(TAIRAV,X,Y,XMUA,XX,YY,XKA,XXX,YYY,CPA,IMUKCP)
      CPM = CPA + 0.444*WAIRI
      PRA = XMUA * CPM / XKA
      RHOM = 144. * PA / (RAU * (TAIRAV+459.7))
C
C       USE SUBROUTINE PDAIR TO DETERMINE PRESSURE DROP OF
C       AIR THROUGH CONDENSER 'PDAIR' (PSI)
```

```
C
      REAIR = GA * DEAC / XMUA
      PDAIRC = PDAIR(MUNITC,FINTYC,DDUCT,AFILTR,AHEATR,RACKS,QAC,GA,
     &               REAIR,AAFC,NTC,STC,DEAC,FPC,FARC,DELTAC,ATAMIN,
     &               RHOM,0.0,NFPC,FPDC,XAPC)
C
C       IF CONDENSER IS THE OUTDOOR COIL AND 'MFANFT' = 1,
C       CALCULATE 'FANEFC' USING THE BUILT-IN 'FANFIT' CURVE.
C       OTHERWISE USE INPUT VALUE OF 'FANEFC'.
C
      IF (MFANFT.EQ.1 .AND. NCORH.EQ.1) CALL FANFIT(PDAIRC,QAC,FANEFC)
C
C       IF FAN POWER IS INPUT, BYPASS FAN POWER CALCULATIONS
C
      IF(FANEFC .LE. 1.0) GO TO 110
      QFANC = FANEFC * 3.413
      GO TO 120
C
C       CALCULATE FAN POWER
C
110   CONTINUE
      QFANC = QAC*PDAIRC*11.1/FANEFC
120   CONTINUE
C
C       OPTIONS FOR ADDING FAN AND/OR COMPRESSOR HEAT TO INCOMING AIR
C
      TAIC = TAIIC
      IF (NCORH .EQ. 2) GO TO 125
      IF (MCMPOP .EQ. 1) TAIC = TAIC + QCAN/(CPM*XMA)
      IF (MFANOU .EQ. 1) TAIC = TAIC + QFANC/(CPM*XMA)
      GO TO 150
  125 IF (MFANIN .EQ. 1) TAIC = TAIC + QFANC/(CPM*XMA)
  150 CONTINUE
      IF (.NOT. PRINT) GO TO 200
```

```
      IF (NCORH .EQ. 2) GO TO 175
      IF (MCMPOP .EQ. 1 .OR. MFANOU .EQ. 1) WRITE(6,1002) TAIC,
     &                                                   XMA
      IF (MCMPOP .EQ. 1) WRITE(6,1003) QCAN
      IF (MFANOU .EQ. 1) WRITE(6,1004) QFANC
      GO TO 200
  175 IF (MFANIN .EQ. 1) WRITE(6,1005) TAIC,XMA,QFANC
  200 CONTINUE
C
C       DETERMINE AIR SIDE HEAT TRANSFER COEFFICIENT 'HAC'(BTU/H-SQ
C                 FT-F)
C       CALL HAIR(FINTYC,CPM,PRA,XMUA,FARC,DEAC,WTC,NTC,GA,HAC)
C
C   ####### ADDED BY WESTINGHOSE R&D 9/27/85 - T. J. FAGAN
C
      CALL HAIR2(FINTYC,CPM,PRA,XMUA,FARC,DEAC,WTC,NTC,GA,
     &       FPC,DELTAC,STC,FPDC,NFPC,UAF,XKA,RHOA,HAC,XFPC)
C
C   ###### END OF 9/27/85 ADDITION #####
C
C
C
C       DETERMINE OVERALL SURFACE EFFICIENCY 'SEFFXC'
C
      CALL SEFF(XKFC,DELTAC,STC,WTC,DEAC,FARC,HAC,SEFFXC)
C
C       CHECK FOR INCONSISTENT ENTERING AIR TEMPERATURE
C
      IERR = 0
      IF(TSAVG.GT.TAIC) GO TO 325
      IERR = 1
      GO TO 900
  325 CONTINUE
C
```

```
C        USE SUBROUTINE EXCH TO DETERMINE CONDENSER HEAT
C        TRANSFER PERFORMANCE AND RETURN ALL RESULTS THROUGH
C        COMMON
C
      CALL EXCH(AOM,TSTTPI,TSTTPO,HFG,HFGO,TAIC,XMR,
     &          RTUBES,RCNCON,XNTU)
C
C        COMPUTE ENTHALPY AT CONDENSER EXIT
C
      CALL SATPRP(TROC,XXX,VOC,YYY,HOC,AAA,BBB,CCC,DDD,FLAG)
      IF (XOC .GT. 0.) HOC = HLTPO + XOC*HFGO
C
C        COMPUTE CIRCUIT CAPACITY AND HX LENGTH AND NUMBER OF RETURN
C        BENDS PER REGION
C
      QC1 = QSPC + QTPC + QSCC
      DZTP = FTPC * (DZC + DZRB)
      DZV = FSPC*DZC
      DZL = FSCC*DZC
      RTBV = RTBPCR * FSPC
      RTBTP = RTBPCR * FTPC
      RTBL = RTBPCR * FSCC
C
C        TEST FOR CONVERGENCE ON CONDENSER SUBCOOLING
C
      SBPREV = SBCOOL
      SBCOOL = TSATCO - TROC
      IF (XOC .GT. 0.) SBCOOL = -200.*XOC
      IF (ITEREX .EQ. 0) GO TO 600
      IF (ABS(SBPREV-SBCOOL) .LT. CNDCON/2.) GO TO 700
  600 CONTINUE
C
C        USE SUBROUTINE PDROP TO DETERMINE PRESSURE DROP OF
C        REFRIGERANT THROUGH CONDENSER 'PDC' (PSI)
```

```
C
      XOCT = XOC
      IF (XOC.GT.0.0 .AND. XOC.LT.0.0001) XOCT = 0.0
      IF(XIC.EQ.1.0) GO TO 525
      VIC=1./RHOV
      GO TO 550
  525 CONTINUE
      CALL VAPOR(TIC,PIC,VIC,X,Y,IERCND(1))
      IF(PRINT .AND. (IERCND(1).NE.0)) WRITE(6,1007)
  550 CONTINUE
      CALL PDROP(1,DERC,STC,E,GR,RTBV,RTBTP,RTBL,XMURV,XMURL,RHOV,
     &           RHOL,RERV,RERL,DZTP,XOCT,XIC,VIC,VOC,DZV,DZL,
     &           DPTP,DPV,DPL,PDC)
C
C       CALCULATE REFRIGERANT PRESSURES AND SATURATION TEMPERATURES
C
      POC = PIC - PDC
      PTPI=PIC-DPV
      PTPO=PTPI-DPTP
      TSTTPI=TSAT(PTPI,FLAG)
      TSTTPO=TSAT(PTPO,FLAG)
      TSAVG = (TSTTPI+TSTTPO)/2.
      TSATCO = TSAT(POC,FLAG)
      ITEREX = ITEREX+1
      IF (PRINT) WRITE(6,1009) ITEREX,TSAVG
C
      IF (ITEREX.LE.20) GO TO 100
C
      IERCND(2) = 1
      IF (PRINT) WRITE(6,1010)
      GO TO 725
  700 CONTINUE
      IF (PRINT) WRITE(6,1011)
  725 CONTINUE
```

```
C
C       OPTIONS FOR ADDING FAN AND/OR COMPRESSOR HEAT TO EXITING AIR
C
      TAOOC = TAOC
      IF (NCORH .EQ. 2) GO TO 735
      EOUTF = QFANC
      IF (MCMPOP .EQ. 2) TAOOC = TAOOC + QCAN/(CPM*XMA)
      IF (MFANOU .EQ. 2) TAOOC = TAOOC + QFANC/(CPM*XMA)
      GO TO 750
  735 IF (MFANIN .EQ. 2) TAOOC = TAOOC + QFANC/(CPM*XMA)
      EINDF = QFANC
  750 CONTINUE
      IF (.NOT. PRINT) GO TO 800
      IF (NCORH .EQ. 2) GO TO 775
      IF (MCMPOP .EQ. 2 .OR. MFANOU .EQ. 2) WRITE(6,1102) TAOOC,
     &                                                     XMA
      IF (MCMPOP .EQ. 2) WRITE(6,1003) QCAN
      IF (MFANOU .EQ. 2) WRITE(6,1004) QFANC
      GO TO 800
  775 IF (MFANIN .EQ. 2) WRITE(6,1105) TAOOC,XMA,QFANC
  800 CONTINUE
C
C       CALCULATE REFRIGERANT SIDE CHECK ON HEAT TRANSFER
C
      QC1K = XMR*(HIC-HOC)
      AAHTC = ARHTC*ALFAAC/ALFARC
C
C       CALCULATE UA VALUES TO BE PRINTED BY MAIN PROGRAM
C
      CUAVR = HSPC*FSPC*ARHTC
      CUATPR = HTPC*FTPC*ARHTC
      CUASCR = HSCC*FSCC*ARHTC
      CUATA = HAC*AAHTC*SEFFXC
      CUAVA = CUATA*FSPC
```

```
      CUATPA = CUATA*FTPC
      CUASCA = CUATA*FSCC
        CUATC = ARHTC/(RTUBES+RCNCON)
        CUAVC = CUATC*FSPC
        CUATPC = CUATC*FTPC
        CUASCC = CUATC*FSCC
      CUAV = 0.
      CUASC = 0.
      IF (CUAVA .GT. 0.10) CUAV = 1./(1./CUAVA + 1./CUAVC + 1./CUAVR)
      CUATP = 1./(1./CUATPA + 1./CUATPC + 1./CUATPR)
      IF (CUASCA .GT. 0.10) CUASC = 1./(1./CUASCA+1./CUASCC+1./CUASCR)
      IF (.NOT. PRINT) GO TO 900
      PDAH20 = PDAIRC/3.613E-02
      WRITE(6,1012) HAC,PDAH20,PDC,HSPC,HTPC,HSCC
      WRITE(6,1112) HCONTC
      WRITE(6,1013) QC1,QC1K
      WRITE(6,1014) HOC
      WRITE(6,1015) CUAVR, CUATPR, CUASCR,
     &              CUAVA, CUATPA, CUASCA,
     &              CUAVC, CUATPC, CUASCC,
     &              CUAV , CUATP , CUASC
  900 CONTINUE
C
C       RETURN TO TOTAL FLOW RATE REPRESENTATION
C
      XMR   = NSECTC*XMR
      QC    = NSECTC*QC1
C
      RETURN
C
 1001 FORMAT('0  COND: INLET CONDITIONS',/,
     &    1X,'         AIR TEMPERATURE           ',F8.3,' F',/,
     &    1X,'         AIR FLOW RATE (TOTAL)     ',F8.2,' CFM',/,
     &    1X,'         REFRIGERANT TEMPERATURE   ',F8.3,' F',/,
```

```
&    1X,'          SATURATION TEMPERATURE   ',F8.3,' F')
1002 FORMAT('0  COND: ELEVATED INLET AIR TEMPERATURE',F8.3,' F',/,
   &       '          AIR MASS FLOW RATE (TOTAL)    ',F8.1,' LBM/H')
1003 FORMAT('          COMPRESSOR SHELL HEAT LOSS    ',F8.1,' BTU/H')
1004 FORMAT('          FAN MOTOR HEAT LOSS           ',F8.1,' BTU/H')
1005 FORMAT('0  COND: ELEVATED INLET AIR TEMPERATURE',F8.3,' F',/,
   &       '          AIR MASS FLOW RATE (TOTAL)    ',F8.1,' LBM/H',/,
   &       '          FAN MOTOR HEAT LOSS           ',F8.1,' BTU/H')
1007 FORMAT('0  COND: ***** ERROR IN "VAPOR" IN CALCULATING ',
   &       'SUPERHEATED REGION PROPERTIES *****',/)
1009 FORMAT('0  COND: ITERATION #',I2,' ON EXIT STATE',/,
   &    1X,'          NEW AVERAGE SATURATION TEMPERATURE   ',F8.3,' F')
1010 FORMAT('0  COND: ***** NO CONVERGENCE ON EXIT STATE *****',/)
1011 FORMAT('0  COND: ITERATION CONVERGED ON CONDENSER SUBCOOLING')
1012 FORMAT('0  COND: RESULTS FOR EACH CIRCUIT',/,
   &       '0         HEAT TRANSFER COEFFICIENTS (BTU/H-SQ FT-F) ',
   &             2X,'PRESSURE DROPS',/,
   &       '            AIR SIDE           ',F8.3,20X,
   &                 'AIR SIDE           ',F8.3,' IN H2O',/,
   &       '            REFRIGERANT SIDE',29X,
   &                 'REFRIGERANT SIDE   ',F8.3,' PSI',/,
   &       '              SUPERHEATED    ',F8.3,/,
   &       '              TWO PHASE      ',F8.3,/,
   &       '              SUBCOOLED      ',F8.3)
1112 FORMAT('           CONTACT INTERFACE',/,
   &       '              CONTACT CONDUCTANCE  ',F10.3)
1013 FORMAT('0         CONDENSER CAPACITY PER CIRCUIT',/,
   &       '            DELTA-T METHOD  ',F10.2,' BTU/H',/,
   &       '            DELTA-H METHOD  ',F10.2,' BTU/H')
1014 FORMAT('0         REFRIGERANT ENTHALPY OUT OF CONDENSER  ',F8.3,
   &       ' BTU/LBM')
1015 FORMAT('0         UA VALUES PER CIRCUIT (BTU/HR-F) :',/,
   &    2X,'          VAPOR REGION',21X,'TWO PHASE REGION',17X,
   &                 'SUBCOOLED REGION',/,
```

```
     &    4X,'        REFRIGERANT SIDE  ',F10.3,5X,
     &               'REFRIGERANT SIDE  ',F10.3,5X,
     &               'REFRIGERANT SIDE  ',F10.3,/,
     &    4X,'        AIR SIDE          ',F10.3,5X,
     &               'AIR SIDE          ',F10.3,5X,
     &               'AIR SIDE          ',F10.3,/,
     &    4X,'        CONTACT INTERFACE ',F10.3,5X,
     &               'CONTACT INTERFACE ',F10.3,5X,
     &               'CONTACT INTERFACE ',F10.3,/,
     &    4X,'        COMBINED          ',F10.3,5X,
     &               'COMBINED          ',F10.3,5X,
     &               'COMBINED          ',F10.3)
 1102 FORMAT('0  COND: ELEVATED EXIT AIR TEMPERATURE ',F8.3,' F',/,
     &      '        AIR MASS FLOW RATE (TOTAL)    ',F8.1,' LBM/H')
 1105 FORMAT('0  COND: ELEVATED EXIT AIR TEMPERATURE ',F8.3,' F',/,
     &      '        AIR MASS FLOW RATE (TOTAL)    ',F8.1,' LBM/H',/,
     &      '        FAN MOTOR HEAT LOSS           ',F8.1,' BTU/H')
      END
```

Listing of the EVAPR Subroutine

A listing of the EVAPR subroutine follows:

```
      SUBROUTINE EVAPR(XMR,QCAN,IERR)
C
C**** EVAPORATOR SIMULATION ROUTINE ****
C
C     PURPOSE
C
C       TO COMPUTE EVAPORATOR PERFORMANCE, INCLUDING DEHUMIDIFICATION,
C       FOR A CROSS-FLOW, REFRIGERANT-TO-AIR, TUBE AND PLATE-FIN TYPE
C       HEAT EXCHANGER
C
C     ADAPTED FROM SUBROUTINE WRITTEN BY
C       C. C. HILLER AND L. R. GLICKSMAN,
C       REPORT NO. 24525-96,
C       HEAT TRANSFER LABORATORY,
C       MASSACHUSETTS INSTITUTE OF TECHNOLOGY.
C
C     OUTPUT
C        QE    -  TOTAL HEAT TRANSFER RATE (BTU/H)
C        QFANE-  EVAPORATOR FAN-MOTOR ENERGY CONSUMPTION (BTU/H)
C        TAIE -  AVERAGE AIR TEMPERATURE ENTERING EVAPORATOR COIL (F)
C        TAOE -  AVERAGE AIR TEMPERATURE LEAVING EVAPORATOR COIL (F)
C        TAOOE-  AVERAGE AIR TEMPERATURE LEAVING EVAPORATOR UNIT (F)
C        TROE -  TEMPERATURE OF REFRIGERANT LEAVING EVAPORATOR (F)
C
C   REMARKS
C    THIS SUBROUTINE CALLS:
C        EVAP   TO DETERMINE THE OVERALL HEAT EXCHANGER PERFORMANCE,
C               HEAT TRANSFER RATES, AIR TEMPERATURES, ETC.
C        EHTC   TO DETERMINE EVAPORATION TWO-PHASE HEAT TRANSFER
```

```
C              COEFFICIENT FOR FORCED CONVECTION EVAPORATION INSIDE
C              TUBES
C       SPHTC2 TO DETERMINE REFRIGERANT LIQUID AND VAPOR
C              HEAT TRANSFER COEFFICIENTS
C
C   ## HAIR CHANGED TO HAIR2 AT
C        WESTINGHOUSE R&D 9/26/85 - T. J. FAGAN ##
C
C       HAIR2  TO DETERMINE THE AIR-SIDE HEAT TRANSFER COEFFICIENT
C       PDROP  TO DETERMINE REFRIGERANT-SIDE PRESSURE DROPS
C       PDAIR  TO DETERMINE AIR-SIDE PRESSURE DROPS
C       FANFIT TO DETERMINE COMBINED OUTDOOR FAN - FAN
C              MOTOR EFFICIENCY
C       XMOIST TO DETERMINE THE PSYCHROMETRIC PROPERTIES OF MOIST AIR
C       MUKCPA TO DETERMINE THE THERMOPHYSICAL PROPERTIES OF AIR
C       MUKCP  TO DETERMINE THERMOPHYSICAL PROPERTIES OF
C              REFRIGERANT LIQUID AND SATURATED VAPOR
C       SPFHT  TO DETERMINE SPECIFIC HEAT OF SUPERHEATED
C              REFRIGERANT VAPOR
C       SATPRP TO DETERMINE SATURATION THERMODYNAMIC PROPERTIES
C       TSAT   TO DETERMINE SATURATION TEMPERATURES CORRESPONDING
C              TO GIVEN PRESSURES
C       VAPOR  TO DETERMINE THERMODYNAMIC PROPERTIES OF SUPERHEATED
C              REFRIGERANT VAPOR
C
C      -------------------------------------------------------------
C
      LOGICAL PRINT
      INTEGER FLAG
      REAL NTE,NSECTE
C
      COMMON / AIR / PA, CPA, CPM, RAU, AFILTR, AHEATR, RACKS
C
      COMMON / A1 / PRINT
```

```
C
      COMMON / EVAPOR / DEAE,   DERE, DELTAE,    FPE, XKFE, XKTE, AAFE,
     &                  NTE, NSECTE, HCONTE,    STE,    WTE,  SIGAE,
     &                   PE,  ARFTE,  ARHTE, ALFARE, ALFAAE,   FARE,
     &                 CARE,    QAE, RTBEVP,    DZE, FANEFE,   RHTE,
     &                 FINTYE, MUNITE
C
      COMMON / EVAPRS / SEFFD, HAW, HAWAV, FMOIST, SEFFWA, WETMAV
C
      COMMON / EVAPS / HAE, XMAE, QE,   PDAIRE, PDE,
     &                HSPE, QSPE, FSPE, CPSPE,
     &                HTPE, QTPE, FTPE
C
      COMMON / EVAPTR / TAIIE,  TIE, TSATEI, HIE, PIE, XIE,
     &                  TAOE,  TROE, TSATEO, HOE, POE, XOE
C
      COMMON / EVRMSG / IEREVR(3)
C
      COMMON / EXPAND / IPRINT, JPRINT
C
      COMMON / LINES / DLL, XLEQLL, DSL,   XLEQSL, DDL, XLEQDL,
     &               DSLRV, XLEQLP, DDLRV, XLEQHP,DPDL,   DPSL,
     &                DPLL, QDISLN,QSUCLN, QLIQLN,E
C
      COMMON / MPASS / CNDCON, AMBCON, EVPCON, CONMST, CMPCON,
     &                 FLOCON, TOLS,   TOLH,   LPRINT, NCORH,
     &                 MCMPOP, MFANIN, MFANOU, MFANFT
C
      COMMON / PRNT7 / TAIE, TAOOE, QFANE
C
      COMMON / PRNT8 / EINDF, EOUTF, ECOMP, RESIST, COP, DP,SS,
     &                 COPHP, QAIR,  FANOUT
C
      COMMON / TIME / CONTIM, EVATIM, APFTIM, SSTIM,  OPTTIM, DATTIM,
```

```
     &                 HITIM,  CMPTIM, EVPTIM, TIMLO,  EXCTIM, AMBTIM,
     &                 IOV1HD, IOVOHD, ISTART
C
      COMMON / UAS / CUAVR , CUATPR, CUASCR, CUAVA,  CUATPA, CUASCA,
     &               CUAVC,  CUATPC, CUASCC, CUAV,   CUATP,  CUASC,
     &               EUAVR,  EUATPR, EUAVA , EUATPA, EUAVC,  EUATPC,
     &               EUAV,    EUATP, UATPAW, UATPCW, UATPW
C
C    #######ADDED BY WESTINGHOUSE R&D 9/26/84 - T. J. FAGAN ##
C
      COMMON / FINPAT / NFPE, FPDE, XFPE, XAPE, NFPC, FPDC, XFPC, XAPC
C
C     ####### END OF ADDITION OF 9/26/85 ######
C
C
      COMMON DDUCT, FIXCAP, ITITLE(20)
C
C       INITIALIZE VALUES
C
      DO 10 I=1,3
   10 IEREVR(I) = 0
C
      XOE = 1.0
      ITEREX = 0
      SUPHT = 0.0
        PDE = 0.0
        DPV = 0.0
      QFANE = 0.0
      TAOE = TAIIE
      TAIE = TAIIE
      TSAVG = TSATEO
      TSATEI = TSATEO
      TIE = TSATEI
      TSTTPO = TSATEO
```

```
      FCOILW = 0.0
      XNTU = 1.0
      IF(PRINT) WRITE(6,1001) TAIIE,RHIE,QAE,HIE,TSATEO
C
C
      RHOA = 144.0*PA/(RAU*(TAIIE + 459.7))
      VA = QAE*60.0/(SIGAE*AAFE)
C
C  ####ADDED BY WESTINGHOUSE R&D 9/26/85 - T. J. FAGAN
C
      UAF = QAE/AAFE
C
C ####### END OF 9/26/85 ADDITION #####
C
C
C        SUBDIVIDE FLOW INTO PARALLEL CIRCUITS AND TREAT EACH
C        LIKE A SEPARATE HEAT EXCHANGER - CONVERT BACK TO TOTAL
C        FLOW AT THE END
C
      XMA = 60.0*QAE*RHOA
      XMAE = XMA/NSECTE
      XMR = XMR/NSECTE
      RTBPCR = RTBEVP / NSECTE
      DZRB = 1.5708 * STE * RTBPCR
C
C        MASS FLOW OF AIR IS BASED ON AMBIENT TEMPERATURE RATHER THAN
C        WHAT TEMPERATURE OF AIR IS ACTUALLY CROSSING THE FAN
C
      GA = VA*RHOA
      GR = XMR/ARFTE
      AOM = NTE*PE/(STE*GA*SIGAE)
      ATAMIN = ALFAAE*ARHTE/(ALFARE*AAFE*SIGAE/NSECTE)
      CALL MUKCPA(TAIIE,X,Y,Z,XX,YY,ZZ,XXX,YYY,CPA,IMUKCP)
      CALL XMOIST(TAIIE,1,RHIE,WAIRI,XX,YY,ZZ,CONMST)
```

```
C
C       ITERATION LOOP ON SUPERHEAT
C
        IPDCN2 = 0
  100 CONTINUE
C
C       CALCULATE SUPERHEATED REFRIGERANT VAPOR PROPERTIES
C
      SUPVAL=SUPHT
      IF(SUPHT.LE.0.02) SUPVAL=0.05
      TSUPAV=(TSTTPO+TSATEO+SUPVAL)/2.
      CALL SPFHT(TSUPAV,POE,X,CPSPE,Y)
      CALL MUKCP(TSUPAV,X,XMUSP,Y,XX,XKRSP,YY,XXX,YYY,ZZZ,IMUKCP)
      PRRSP=XMUSP*CPSPE/XKRSP
C
C       DETERMINE SINGLE PHASE VAPOR HEAT TRANSFER COEFFICIENT
C       'HRV' (BTU/H-SQ FT-F)
C
      CALL SPHTC2(2,DERE,GR,XMUSP,CPSPE,PRRSP,RERSP,HSPE)
C
C       CALCULATE ENTERING REFRIGERANT QUALITY 'XIE' AND
C       AVAILABLE TWO-PHASE HEAT TRANSFER RATE 'QTPI'
C
      CALL SATPRP(TSATEI,XXX,VF,YYY,HLI,HFGI,AAA,BBB,CCC,FLAG)
      CALL SATPRP(TSTTPO,X,Y,VG,HLO,HFGO,HVO,XX,YY,FLAG)
      XIE = (HIE-HLI)/HFGI
      QTPI=XMR*(HVO-HIE)
      IF (XIE.LT.0.0) IEREVR(1) = 1000*XIE
      IF (XIE.LT.0.0) XIE = 0.0
C
C       CALCULATE TWO-PHASE REFRIGERANT PROPERTIES
C
      RHOV = 1.0/VG
      RHOL = 1.0/VF
```

```
      CALL MUKCP(TSAVG,XMURL,XMURV,Z,XKRL,XKRV,ZZ,CPRL,
     &           CPRV,ZZZ,IMUKCP)
      PRRL = XMURL*CPRL/XKRL
      PRRV = XMURV*CPRV/XKRV
      CALL SPHTC2(2,DERE,GR,XMURL,CPRL,PRRL,RERL,HTLIQ)
      CALL SPHTC2(2,DERE,GR,XMURV,CPRV,PRRV,YY,HTVAP)
      COEF = 3.0 * HTLIQ * (RHOL/RHOV)**0.33 * (XMURV/XMURL)**0.0667
C
C     IF (ITEREX .EQ. 0) GO TO 300
C
C       DETERMINE EVAPORATION TWO-PHASE HEAT TRANSFER COEFFICIENT 'HTP'
C       (BTU/H-SQ FT-F)
C
      CALL EHTC(COEF,XIE,XOE,HTVAP,HTPE)
C
C       CALCULATE AIR PROPERTIES
C
      TAIRAV = TSAVG + (TAIE - TAOE)/XNTU
      CALL MUKCPA(TAIRAV,X,Y,XMUA,XX,YY,XKA,XXX,YYY,CPA,IMUKCP)
      CPM = CPA + 0.444*WAIRI
      PRA = XMUA * CPM / XKA
      RHOM = 144. * PA / (RAU * (TAIRAV+459.7))
C
C       USE SUBROUTINE PDAIR TO DETERMINE PRESSURE DROP OF
C       AIR THROUGH EVAPORATOR 'PDAIR' (PSI)
C
      REAIR = GA * DEAE / XMUA
      PDAIRE = PDAIR(MUNITE,FINTYE,DDUCT,AFILTR,AHEATR,RACKS,QAE,GA,
     &               REAIR,AAFE,NTE,STE,DEAE,FPE,FARE,DELTAE,ATAMIN,
     &               RHOM,FCOILW,NFPE,FPDE,XAPE)
C
C       IF THE EVAPORATOR IS THE OUTDOOR COIL AND 'MFANFT' = 1,
C       CALCULATE 'FANEFE' USING THE BUILT-IN 'FANFIT' CURVE.
C       OTHERWISE USE INPUT VALUE OF 'FANEFE'.
```

```
C
      IF (MFANFT.EQ.1 .AND. NCORH.EQ.2) CALL FANFIT(PDAIRE,QAE,FANEFE)
C
C       IF FAN POWER IS INPUT, BYPASS FAN POWER CALCULATIONS
C
      IF(FANEFE .LE. 1.0) GO TO 110
      QFANE = FANEFE * 3.413
      GO TO 120
C
C       CALCULATE FAN POWER
C
110   CONTINUE
      QFANE = QAE*PDAIRE*11.1/FANEFE
120   CONTINUE
C
C       OPTIONS TO ADD FAN AND/OR COMPRESSOR HEAT TO INCOMING AIR
C
      TAIE = TAIIE
      IF (NCORH .EQ. 2) GO TO 125
      IF (MFANIN .EQ. 1) TAIE = TAIE + QFANE/(CPM*XMA)
      GO TO 150
  125 IF (MCMPOP .EQ. 1) TAIE = TAIE + QCAN/(CPM*XMA)
      IF (MFANOU .EQ. 1) TAIE = TAIE + QFANE/(CPM*XMA)
  150 CONTINUE
      IF (.NOT. PRINT) GO TO 200
      IF (NCORH .EQ. 1) GO TO 175
      IF (MCMPOP .EQ. 1 .OR. MFANOU .EQ. 1) WRITE(6,1002) TAIE,
     &                                                   XMA
      IF (MCMPOP .EQ. 1) WRITE(6,1003) QCAN
      IF (MFANOU .EQ. 1) WRITE(6,1004) QFANE
      GO TO 200
  175 IF (MFANIN .EQ. 1) WRITE(6,1005) TAIE,XMA,QFANE
  200 CONTINUE
C
```

```
C       DETERMINE AIR SIDE HEAT TRANSFER COEFFICIENT 'HAE'(BTU/H-SQ
C             FT-F)
C
C     THE ORIGINAL CALL WAS:
C     CALL HAIR(FINTYE,CPM,PRA,XMUA,FARE,DEAE,WTE,NTE,GA,HAE)
C
C   ######ADDED BY WESTINGHOUSE R&D 9/26/85 - T. J. FAGAN ###
C
      CALL HAIR2(FINTYE,CPM,PRA,XMUA,FARE,DEAE,WTE,NTE,GA,
     1      FPE,DELTAE,STE,FPDE,NFPE,UAF,XKA,RHOA,HAE,XFPE)
C
C   ###### END OF 9/26/85 ADDITION #####
C
      HAW = HAE*0.626*(QAE/AAFE)**0.101
C
C       CHECK FOR INCONSISTENT ENTERING AIR TEMPERATURE
C
      IERR = 0
      IF(TSAVG.LT.TAIE) GO TO 300
      IERR = 1
      GO TO 900
  300 CONTINUE
C
C       CALCULATE PRESSURE DROP ASSUMING XOE=1 AND FTPE=1
C
        IF(ITEREX.GT.1) GO TO 297
        FTPE=1.0
        FSPE=0.0
        VOE=1./RHOV
        XOE=1.0
        IF(ITEREX.EQ.1) GO TO 297
        GO TO 330
297     CONTINUE
C
```

```
C       USE SUBROUTINE EVAP TO DETERMINE EVAPORATOR HEAT
C       TRANSFER PERFORMANCE AND RETURN ALL RESULTS THROUGH COMMON
C
      I0 = ICLOCK(0)
      CALL EVAP(AOM,TSAVG,TSTTPO,QTPI,TAIE,XMR,
     &          RTUBES,RCNCON,XNTU)
      I1 = ICLOCK(0)
      EVPTIM = EVPTIM + (I1 - I0)/100.
C
C       SET UP ROOT FINDING PROBLEM
C
        IF(ITEREX.GT.1) GO TO 305
        IF(FTPE.NE.1.0) GO TO 310
        XHI=1.0
        XLOW=XOE
        FHI=XHI-XLOW
        ITYPE=1
        GO TO 304
310     CONTINUE
        XHI=1.0
        XLOW=FTPE
        FHI=XHI-XLOW
        ITYPE=2
304     CONTINUE
        XNEW=XLOW
        GO TO 330
305     CONTINUE
        IF(ITYPE.EQ.2) GO TO 315
        XNEX=XOE
        IF(FTPE.NE.1.0) XNEX=2.0-FTPE
        GO TO 320
315     CONTINUE
        XNEX=FTPE
        IF(XOE.NE.1.0) XNEX=2.0-XOE
```

```
320     CONTINUE
        FUNC=XNEW-XNEX
        IF(ITEREX.EQ.2) FLOW=FUNC
        IF(ITEREX.EQ.2) GO TO 325
        FFAC=FHI*FUNC
        IF(FFAC) 321,600,322
321     CONTINUE
        XLOW=XNEW
        FLOW=FUNC
        GO TO 325
322     CONTINUE
        XHI=XNEW
        FHI=FUNC
325     CONTINUE
        CALL FALSEP(XHI,XLOW,FHI,FLOW,XNEW,EVAL)
        IF(EVAL.LT.EVPCON/1200.) GO TO 600
C
C       CONVERT XNEW VALUE TO XOE OR FTPE VALUE AND COMPUTE RELATED
C       VALUES
C
        XOE=XNEW
        FTPE=1.0
        FSPE=0.0
        IF(ITYPE.EQ.1) GO TO 330
        XOE=1.0
        FTPE=XNEW
        FSPE=1.0-FTPE
330     CONTINUE
        VOE = 1.0/RHOV
      IF(FTPE.NE.1.0) CALL VAPOR(TROE,POE,VOE,X,Y,IEREVR(2))
      DZTP = FTPE * (DZE + DZRB)
      DZV = FSPE * DZE
      DZL = 0.0
      RTBV = RTBPCR * FSPE
```

```
      RTBTP = RTBPCR * FTPE
      FCOILW = FMOIST * FTPE
C
C       USE SUBROUTINE PDROP TO DETERMINE PRESSURE DROP OF
C       REFRIGERANT THROUGH EVAPORATOR 'PDE' (PSI)
C
      CALL PDROP(2,DERE,STE,E,GR,RTBV,RTBTP,X,XMURV,XMURL,RHOV,RHOL,
     &           RERSP,RERL,DZTP,XOE,XIE,VOE,1.0,DZV,DZL,DPTP,DPV,
     &           ZZZ,PDE)
      PIE = POE + PDE
      TSATEI = TSAT(PIE,FLAG)
      TIE = TSATEI
      PTPO=POE+DPV
      TSTTPO=TSAT(PTPO,FLAG)
      TSAVG = (TSTTPO + TSATEI)/2.0
      ITEREX = ITEREX + 1
      IF(.NOT.PRINT) GO TO 550
      WRITE(6,1008) ITEREX,TSAVG
      IF(XOE.EQ.1.0)  WRITE(6,1108) FTPE
      IF(FTPE.EQ.1.0) WRITE(6,1109) XOE
  550 CONTINUE
      IF(ITEREX.LE.10) GO TO 100
      IEREVR(3) = 1
      IF (JPRINT .NE. 0) WRITE(6,1009)
      GO TO 700
  600 CONTINUE
      QE1 = QSPE+QTPE
      SUPHT = TROE - TSATEO
      IF (XOE .LT. 1.0) SUPHT = -500.*(1.0 - XOE)
      IF (PRINT) WRITE(6,1010)
  700 CONTINUE
C
C       OPTIONS FOR ADDING FAN AND/OR COMPRESSOR HEAT TO EXITING AIR
C
```

```
C
      TAOOE = TAOE
      IF (NCORH .EQ. 2) GO TO 725
      EINDF = QFANE
      IF (MFANIN .EQ. 2) TAOOE = TAOOE + QFANE/(CPM*XMA)
      GO TO 750
  725 EOUTF = QFANE
      IF (MCMPOP .EQ. 2) TAOOE = TAOOE + QCAN/(CPM*XMA)
      IF (MFANOU .EQ. 2) TAOOE = TAOOE + QFANE/(CPM*XMA)
  750 CONTINUE
      IF (.NOT. PRINT) GO TO 800
      IF (NCORH .EQ. 1) GO TO 775
      IF (MCMPOP .EQ. 2 .OR. MFANOU .EQ. 2) WRITE(6,1102) TAOOE,
     &                                                    XMA
      IF (MCMPOP .EQ. 2) WRITE(6,1003) QCAN
      IF (MFANOU .EQ. 2) WRITE(6,1004) QFANE
      GO TO 800
  775 IF (MFANIN .EQ. 2) WRITE(6,1105) TAOOE,XMA,QFANE
  800 CONTINUE
C
C
      AAHTE = ARHTE*ALFAAE/ALFARE
C
C        CALCULATE UA VALUES TO BE PRINTED BY MAIN PROGRAM
C
      EUAVR = HSPE*FSPE*ARHTE
      EUATPR = HTPE*FTPE*ARHTE
      EUATA = HAE*AAHTE*SEFFD
      EUAVA = EUATA*FSPE
      UATPAD = EUATA*FTPE*(1.0-FMOIST)
      UATPAW = HAWAV*AAHTE*SEFFWA*FTPE*FMOIST
      EUATCD = ARHTE / (RTUBES + RCNCON)
      EUATCW = ARHTE / (RTUBES + RCNCON/WETMAV)
      EUAVC  = EUATCD*FSPE
```

```
      UATPCD = EUATCD*FTPE*(1.0-FMOIST)
      UATPCW = EUATCW*FTPE*FMOIST
      EUAV = 0.0
      UATPD = 0.0
      UATPW = 0.0
      IF (EUAVA .GT. 0.10) EUAV = 1.0/ (1.0/EUAVA + 1.0/EUAVC +
     &                                1.0/EUAVR)
      IF (FMOIST.LT.1.0) UATPD = 1.0/ (1.0/UATPAD + 1.0/UATPCD +
     &                              1.0/EUATPR/(1.-FMOIST))
      IF (FMOIST.GT.0.0) UATPW = 1.0/ (1.0/UATPAW + 1.0/UATPCW +
     &                              1.0/EUATPR/FMOIST)
      EUATPA = UATPAD
      EUATPC = UATPCD
      EUATP  = UATPD
      IF(.NOT.PRINT) GO TO 900
C
      PDAH20 = PDAIRE/3.613E-02
      WRITE(6,1011)   XMAE,   XMR,
     &                  GA,    GR,
     &              PDAH20,   PDE,
     &                 HAE,  HSPE,
     &                 HAW,  HTPE
      WRITE(6,1111) HCONTE
      WRITE(6,1012) SEFFD, SEFFWA, WETMAV
      WRITE(6,1013) QE1
      WRITE(6,1014) EUAVR,EUATPR,EUAVA,UATPAD,UATPAW,EUAVC,UATPCD,
     &              UATPCW,EUAV,UATPD,UATPW
  900 CONTINUE
      XMR = NSECTE*XMR
      QE  = NSECTE*QE1
      IF(FSPE.GT.0.0) XOE = 1.0
      RETURN
C
C
```

```
1001 FORMAT('0 EVAPR: GIVEN CONDITIONS',/,
   &     1X,'          INLET AIR TEMPERATURE           ',F8.3,' F',/,
   &     1X,'          INLET AIR RELATIVE HUMIDITY     ',F8.3,/,
   &     1X,'          AIR FLOW  RATE (TOTAL)          ',F8.1,' CFM',/,
   &     1X,'          INLET REFRIGERANT ENTHALPY      ',F8.3,' BTU/LBM',
   &   /,1X,'          OUTLET SATURATION TEMPERATURE ',F8.3,' F')
1002 FORMAT('0 EVAPR: ELEVATED INLET AIR TEMPERATURE ',4X,F8.3,' F',/,
   &       9X,'AIR MASS FLOW RATE (TOTAL)             ',F8.1,' LBM/H')
1003 FORMAT(9X,'COMPRESSOR CAN HEAT LOSS               ',F8.1,' BTU/H')
1004 FORMAT(9X,'FAN HEAT LOSS                          ',F8.1,' BTU/H ')
1005 FORMAT('0 EVAPR: ELEVATED INLET AIR TEMPERATURE ',4X,F8.3,' F',/,
   &         9X,'AIR MASS FLOW RATE (TOTAL)           ',F8.1,' LBM/H',/,
   &         9X,'FAN HEAT LOSS                        ',F8.1,' BTU/H')
1008 FORMAT('0 EVAPR: ITERATION #',I2,' ON EXIT STATE',/,
   &     1X,'          NEW AVERAGE SATURATION TEMPERATURE ',F8.3,' F')
1108 FORMAT(1X,'         NEW TWO-PHASE FRACTION               ',F8.4)
1109 FORMAT(1X,'         NEW OUTLET QUALITY                   ',F8.4)
1009 FORMAT('  EVAPR: **** ITERATION DID NOT CONVERGE ON SUPERHEAT',
   &       ' IN 10 TRIES ****')
1010 FORMAT('0 EVAPR: ITERATION CONVERGED ON EVAPORATOR SUPERHEAT')
1011 FORMAT('0 EVAPR: RESULTS FOR EACH CIRCUIT',/,
   &       '0          AIR SIDE:',30X,'REFRIGERANT SIDE:',/,
   &     2X,'          MASS FLOW RATE  ',F10.1,' LBM/H',7X,
   &               'MASS FLOW RATE  ',F10.1,' LBM/H',/,
   &     2X,'          MASS FLUX       ',F10.1,' LBM/H-FT2',3X,
   &               'MASS FLUX       ',F10.1,' LBM/H-FT2',/,
   &     2X,'          PRESSURE DROP   ',F10.3,' IN H2O',6X,
   &               'PRESSURE DROP   ',F10.3,' PSI',//,
   &     2X,'          HEAT TRANSFER COEFFICIENT (BTU/HR-FT2-F)',/,
   &     2X,'            DRY COIL      ',F10.3,13X,
   &               'VAPOR REGION      ',F10.3,/,
   &     2X,'            WET COIL      ',F10.3,13X,
   &               'TWO PHASE REGION  ',F10.3,/)
1111 FORMAT('          CONTACT INTERFACE:',/,2X,
```

```
     &        '          CONTACT CONDUCTANCE',F8.2,' BTU/HR-FT2-F',/)
1012 FORMAT(1X,'           DRY FIN EFFICIENCY              ',F6.3,/,
     &        '          WET FIN EFFICIENCY (AVERAGE)    ',F6.3,/,
     &        '          WET CONTACT FACTOR (AVERAGE)    ',F6.3)
1013 FORMAT('0           EVAPORATOR CAPACITY PER CIRCUIT   ',F8.1,' BTU/H')
1014 FORMAT('0           UA VALUES PER    ',7X,'VAPOR',7X,'TWO PHASE',
     &      /,'          CIRCUIT (BTU/H-F) :     ','REGION',7X,
     &        'REGION',/,
     &       2X,'           REFRIGERANT SIDE',2X,F8.3,5X,F10.2,/,
     &       2X,'           AIR SIDE',/,
     &       2X,'             DRY COIL       ',2X,F8.3,5X,F10.2,/,
     &       2X,'             WET COIL       ', 15X,F10.2 ,/,
     &       2X,'           CONTACT INTERFACE',/,
     &       2X,'             DRY COIL       ',2X,F8.3,5X,F10.2,/,
     &       2X,'             WET COIL       ', 15X,F10.2 ,/,
     &       2X,'           COMBINED         ',/,
     &       2X,'             DRY COIL       ',2X,F8.3,5X,F10.2,/,
     &       2X,'             WET COIL       ', 15X,F10.2)
1102 FORMAT('0 EVAPR: ELEVATED EXIT AIR TEMPERATURE  ',4X,F8.3,' F',/,
     &        9X,'AIR MASS FLOW RATE (TOTAL)           ',F8.1,' LBM/H')
1105 FORMAT('0 EVAPR: ELEVATED EXIT AIR TEMPERATURE  ',4X,F8.3,' F',/,
     &          9X,'AIR MASS FLOW RATE (TOTAL)         ',F8.1,' LBM/H',/,
     &          9X,'FAN HEAT LOSS                      ',F8.1,' BTU/H')
      END
```

Listing of the COMPV Subroutine

A listing for the COMPV subroutine follows:

```
      SUBROUTINE COMPV (TOLSD, TOLHD, CMPCD)
C
C***          LOSS AND EFFICIENCY BASED COMPRESSOR MODEL
C
C        ##################################################
C
C     $$ THE PROGRAM WAS MODIFIED ON 5/6/85 AT THE WESTINGHOUSE
C        RESEARCH LABORATORY TO PERMIT MODELING A SUCTION LINE
C        ACCUMULATOR.
C     $$ THE PROGRAM WAS MODIFIED ON 8/2/85 AT THE WESTINGHOUSE
C        RESEARCH LABORATORY TO ACCOMMODATE LEAKAGE IN THE
C        REVERSING VALVE BETWEEN THE HIGH PRESSURE OUT OF THE
C        COMPRESSOR, AND THE INLET TO THE COMPRESSOR.
C
C        ##################################################
C
      LOGICAL PRINT
C
      COMMON / A1 / PRINT
C
      COMMON / CMPMOT / CETAM(6), RPMSLP
C
      COMMON / CMPRSR / TRICMP, TSICMP, HINCMP, PINCMP, XINCMP,
     1                  TROCMP, TSOCMP, HOUCMP, POUCMP, XOUCMP
C
      COMMON / COMPR / VR,     SYNC,   FLMOT,  EFFMMX, ETAISN, ETAMEC,
     1                 ETAVLA, ETAVLB, POW,    CANFAC, HILOFC, QCAN,
     2                 QHILO,  DISPL,  MTRCLC
```

```
C
      COMMON / CONDSR / TAIIC,  TIC, TSATCI, HIC, PIC, XIC,
     1                  TAOC,  TROC, TSATCO, HOC, POC, XOC
C
      COMMON / EVAPTR / TAIIE,  TIE, TSATEI, HIE, PIE, XIE,
     1                  TAOE,  TROE, TSATEO, HOE, POE, XOE
C
      COMMON / FLOWBA / DTROC,  SUPER, CAPFLO, ORIFD, XMR, NCAP,
     1                  IREFC, ICOMP, ITRPIE
C
      COMMON / LINES / DLL, XLEQLL, DSL,   XLEQSL, DDL, XLEQDL,
     1               DSLRV, XLEQLP, DDLRV, XLEQHP,DPDL,   DPSL,
     2                DPLL, QDISLN,QSUCLN, QLIQLN,E
C
      COMMON / RVALVE / TAMBRV, NRVALV, DPLOV, DPHIV, QINTV, QEXTV
C
      COMMON / PRNT1 / RPM, PCTFL, ETAMOT, ETAVOL, ETASUP, ETATOT,
     1                 PRATIO
C
      COMMON / SUPERE / SUPERE
C
C       ** CHANGES ADDED 5/4/85 AT WESTINGHOUSE R&D (ROGER LUCHETA) **
C
      COMMON / ACPAR / DHHAC, DHLAC, DPHAC, DPLAC, NACCU,
     1      NACPR, ACPAR(20)
C      ## SEE BLOCK DATA PROGRAM RLADP FOR DEFINITIONS OF THE
C         VARIABLES ABOVE.
      COMMON / MPASS / CNDCON, AMBCON, EVPCON, CONMST, CMPCON, FLOCON,
     1                TOLS,   TOLH,   LPRINT, NCORH,  MCMPOP, MFANIN,
     2                MFANOU, MFANFT
C
      COMMON / PRNT2 / TFLODV, TSATFL, PFLODV, XFLODV
C
C       ** END OF 5/4/85 CHANGES **
```

```
C      ## CHANGES ADDED 8/2/85 AT WESTINGHOUSE R&D (R. LUCHETA) ##
C
      COMMON / RVLKP / NRVLK, NRDFL, NLPMX, NLKTB(5), RVLKP(5)
C       SEE BLOCK DATA PROGRAM RLADP FOR THE DEFINITION OF
C        THESE PARAMETERS.
C
      DATA NOTTT / 6 /
C
C     ## END OF 8/2/85 CHANGES ##
C
C         ##################################################
C
1001  CONTINUE
      DPHIV = 0.0
      DPLOV = 0.0
      QINTV = 0.0
      QEXTV = 0.0
C
C       ** CHANGES ADDED 5/4/85 **
      CMPCN = CMPCD
      TOLSL = TOLSD
      TOLHL = TOLHD
      DHHAC = 0.0
      DHLAC = 0.0
      DPHAC = 0.0
      DPLAC = 0.0
C     ## AND A NOTATIONAL CHANGE
      TOC   = TROC
C       ** END 5/4/85 CHANGES **
C
C***            CALCULATE SUCTION LINE CONDITIONS
C
      CALL SATPRP
     1       (TSICMP,PINCMP,VFI,VGI,HFI,HFGI,HGI,SFI,SGI,IERROR)
```

```
      IF (SUPER.GE.0.0) GO TO 10
      XINCMP = -SUPER
      TRICMP = TSICMP
      HINCMP = HFI + XINCMP*HFGI
      VINCMP = VFI + XINCMP*(VGI-VFI)
      SINCMP = SFI + XINCMP*(SGI-SFI)
      GO TO 20
   10 CONTINUE
      XINCMP = 1.0
      TRICMP = TSICMP+SUPER
      CALL VAPOR
     1        (TRICMP,PINCMP,VINCMP,HINCMP,SINCMP,IERROR)
   20 CONTINUE
      HINSP = HINCMP
      XINSP = XINCMP
      VINSP = VINCMP
      SINSP = SINCMP
      TRISP = TRICMP
      CALL SATPRP(TSOCMP,POUCMP,VFO,VGO,HFO,HFGO,HGO,SFO,SGO,IERROR)
C
C***            ITERATION ON ENTHALPY AT SUCTION PORT
C
      DO 1200 I = 1,20
C
C***            CALCULATE SUCTION PORT CONDITIONS
C
      IF (I.EQ.1) GO TO 55
      XINSP = 1.0
      IF (HINSP.GT.HGI) GO TO 45
      XINSP = (HINSP-HFI) / HFGI
      VINSP = VFI + XINSP*(VGI-VFI)
      SINSP = SFI + XINSP*(SGI-SFI)
      TRISP = TSICMP
      GO TO 55
```

```
   45 CONTINUE
      CALL TRIAL(TSICMP,10.0,PINCMP,3,HINSP,TOLHL,VINSP,X,SINSP,
     1       TRISP,IERROR)
      IF (IERROR .NE. 0) WRITE(6,1002)
   55 CONTINUE
C
C***            CALCULATE ENTHALPY OUT OF COMPRESSOR  'HISENS'
C***            ASSUMING ISENTROPIC COMPRESSION FROM SUCTION PORT
C
      IF (SINSP.GT.SG0) GO TO 60
      XOUT = (SINSP-SF0)/(SG0-SF0)
      HISENS = HF0+XOUT*HFG0
      TGUESS = TSOCMP
      GO TO 70
   60 CONTINUE
      CALL TRIAL(TSOCMP,50.0,POUCMP,4,SINSP,TOLSL,XXX,HISENS,
     1           YYY,TGUESS,IERROR)
      IF (IERROR .NE. 0) WRITE(6,1003)
   70 CONTINUE
C
C***            CALCULATE ENTHALPY OUT OF COMPRESSOR DISCHARGE PORT
C***            'HISENS' BASED ON ASSUMED 'ETAISN'
C
      HOUDP = HINSP+(HISENS-HINSP)/ETAISN
C
C***            CALCULATE DISCHARGE PORT CONDITIONS
C
      XOUDP = 1.0
      IF (HOUDP.GT.HG0) GO TO 80
      XOUDP = (HOUDP-HF0)/HFG0
      VOUDP = VF0+XOUDP*(VG0-VF0)
      TOUDP = TSOCMP
      GO TO 85
   80 CONTINUE
```

```
      CALL TRIAL(TGUESS,50.0,POUCMP,3,HOUDP,TOLHL,VOUDP,X,Y,TOUDP,
     1             IERROR)
      IF (IERROR.NE.0) WRITE(6,1004)
   85 CONTINUE
C
C***              CALCULATE VOLUMETRIC EFFICIENCY 'ETAVOL' AS A
C***              FUNCTION OF SPECIFIC VOLUME RATIO
C
      CALL SPFHT(TRISP,PINCMP,XX,YY,GAMMA)
      ETATHE = 1.0 - VR*( (POUCMP/PINCMP)**(1.0/GAMMA) - 1.0)
      ETAVOL = ETATHE-ETAVLB*((GAMMA-1.0)/GAMMA)*(POUCMP/PINCMP)-ETAVLA
C
C***              CHOOSE EITHER MOTOR EFF & SPEED CURVES OR
C***              SPECIFIED MOTOR EFF & SPEED
C
      RPM = SYNC
      ETAMOT = EFFMMX
      PCTFL = 1.00
      IF (MTRCLC.NE.0) GO TO 130
C
C***              SECTION FOR CALCULATING COMPRESSOR RPM ASSUMING
C***              THAT RPM IS A LINEAR FUNCTION OF FRACTIONAL LOAD
C
        XMPRPM = ETAVOL*60.0*DISPL/1728.0/VINSP
        POPXM = HOUDP-HINSP
        PCTPPO = 1.0/ETAMEC/3413.0/FLMOT
C
C***              RPM EQUATION ASSUMES THAT
C***              'RPM = (1.0 + RPMSLP * PCTFL) * SYNC'
C
      RPM    = 1.0/(RPMSLP*XMPRPM*POPXM*PCTPPO + 1.0/SYNC)
C      ## GET THE REFRIGERENT MASS FLOW THROUGH THE COMPRESSOR
      XMRCM = XMPRPM*RPM
      POWCS = POPXM*XMRCM
```

```
      PCTFL = PCTPPO*POWCS
C      ## GET THE REVERSING VALVE LEAKAGE
      RFVLK = FLKRV ( NRVLK, NLPMX, RVLKP, XMRCM/3600.0,
     1      POUCMP, TOUCMP, HOUCMP, PINCMP, TRICMP, HINCMP )
C         AND CONVERT TO LBM/HR
      RFVLK = RFVLK * 3600.0
C      ## GET THE REFRIGERENT FLOW THROUGH THE REST OF THE SYSTEM
      XMR    = XMRCM - RFVLK
C
C***              CALCULATE MOTOR EFFICIENCY 'ETAMOT' AS A FUNCTION
C***              OF FRACTIONAL LOAD 'PCTFL' AND MAXIMUM EFFICIENCY
C***              'EFFMMX'
C
      IF (PCTFL .LT. 0.20) GO TO 110
      PCTMOT = CETAM(1)
      Y = 1.0
      DO 1100 J = 2,6
      Y = Y*PCTFL
1100   PCTMOT = PCTMOT+CETAM(J)*Y
      GO TO 120
  110 PCTMOT = 3.80*PCTFL
  120 ETAMOT = EFFMMX*PCTMOT
      GO TO 140
  130 CONTINUE
      XMRCM = ETAVOL*60.0*RPM*DISPL/1728.0/VINSP
      POWCS = XMRCM*(HOUDP-HINSP)
      FLMOT = POWCS / ETAMEC / 3413.0
C      ## GET THE REVERSING VALVE LEAKAGE
      RFVLK = FLKRV ( NRVLK, NLPMX, RVLKP, XMRCM/3600.0,
     1      POUCMP, TOUCMP, HOUCMP, PINCMP, TRICMP, HINCMP )
C         AND CONVERT TO LBM/HR
      RFVLK = RFVLK * 3600.0
C      ## GET THE REFRIGERENT FLOW THROUGH THE REST OF THE SYSTEM
      XMR    = XMRCM - RFVLK
```

```
  140 CONTINUE
C
C***            CALCULATE COMPRESSOR POWER
C
      POW = POWCS / ETAMEC / ETAMOT
C
C***            CALCULATE COMPRESSOR SHELL HEAT LOSS 'QCAN' AND
C***            INTERNAL HEAT TRANSFER RATE 'QHILO'
C
      IF (CANFAC .LE. 0.0) GO TO 150
      IF (CANFAC .LT. 1.0) QCAN = CANFAC*POW
      IF (CANFAC .GE. 1.0) QCAN = POW*(1.0-ETAMOT*ETAMEC)*0.90
  150 IF (HILOFC .LE. 0.0) GO TO 160
      IF (HILOFC .LT. 1.0) QHILO = HILOFC*POW
      IF (HILOFC .GE. 1.0) QHILO = 0.03*POW
C
C***             RECALCULATE 'HINSP' TAKING ACCOUNT OF MOTOR & MECH.
C***             LOSSES TO SUCTION GAS AND INTERNAL HEAT TRANSFER
C
  160 HSPOLD = HINSP
      DELHSP = (QHILO + POW*(1.0-ETAMOT*ETAMEC)-QCAN)/XMRCM
      IF (DELHSP.GE.0.0) GO TO 1170
      CMHEAT = POW*(1.0-ETAMOT*ETAMEC)
      WRITE(6,1005) QHILO,CMHEAT,QCAN,DELHSP,I
      STOP
1170   CONTINUE
      HINSP = HINCMP+DELHSP
      IF (ABS(HINSP-HSPOLD).LE.CMPCN) GO TO 1300
1200   CONTINUE
C
C***             FAILED TO CONVERGE
C
      WRITE(6,1006) HINSP, HSPOLD
C
```

```
1300  CONTINUE
C
C***            CALCULATE SHELL EXIT CONDITIONS
C
      HOUCMP = (POW-QCAN)/XMRCM + HINCMP
      XOUCMP = 1.0
      IF (HOUCMP.GT.HGO) GO TO 1090
      XOUCMP = (HOUCMP-HFO)/HFGO
      VOUCMP = VFO+XOUCMP*(VGO-VFO)
      TROCMP = TSOCMP
      GO TO 1095
1090  CONTINUE
      CALL TRIAL(TSOCMP,50.0,POUCMP,3,HOUCMP,TOLHL,VOUCMP,XXX,
     1           YYY,TROCMP,IERROR)
      IF (IERROR .NE. 0) WRITE(6,1030)
1095  CONTINUE
C
C***            CALCULATE THE REFRIGERANT CONDITIONS ENTERING THE
C***            CONDENSER
C
C
C               CALCULATE THE PRESSURE DROP 'DPDL' (PSI) OF VAPOR
C               IN THE DISCHARGE LINE AND 'POUCMP' (PSIA) THE
C               PRESSURE AT THE COMPRESSOR EXIT
C
C***    DPDL1 IS THE PRESSURE DROP FROM THE REVERSING VALVE TO THE COIL
C***    DPDL2 IS THE PRESSURE DROP FROM THE COMPRESSOR DISCHARGE
C             TO THE REVERSING VALVE
C***    DPDL  IS THE TOTAL PRESSURE DROP IN THE HIGH SIDE
C***    DPLAC IS THE PRESSURE DROP ON THE LOW PRESSURE
C                SIDE OF THE ACCUMULATOR
C***    DPHAC IS THE PRESSURE DROP ON THE HIGH PRESSURE
C                SIDE OF THE ACCUMULATOR
C***    DHLAC IS THE ENTHALPY CHANGE OF THE FLUID ON THE LOW PRESSURE
```

```
C                SIDE OF THE ACCUMULATOR
C***    DHHAC IS THE ENTHALPY CHANGE OF THE FLUID ON THE HIGH PRESSURE
C                SIDE OF THE ACCUMULATOR
C
      CALL MUKCP(TROCMP,XXX,XMUVH,YYY,ZZZ,AAA,BBB,CCC,DDD,EEE,IMUKCP)
      RHOVH = 1.0/VOUCMP
      CALL DPLINE (DDL,   XLEQDL, E, XMR,   RHOVH, XMUVH, DPDL1)
      CALL DPLINE (DDLRV, XLEQHP, E, XMRCM, RHOVH, XMUVH, DPDL2)
C
C        NOW ITERATE ON SUCTION LINE REVERSING VALVE AND ACCUMULATOR
C        PRESSURE DROPS TO GET THE CONSISTENT EVAPORATOR OUTLET
C        CONDITIONS
C        (THIS ITERATION ADDED 5/9/85 AT WESTINGHOUSE R&D BY R. LUCHETA)
C          PORVL - REVERSING VALVE OUTLET PRESSURE, LOW SIDE
C          TORVL -    "         "     "    TEMPERATURE, LOW SIDE
C          XORVL -    "         "     "    QUALITY, LOW SIDE
C          PIRVH -    "         "   INLET PRESSURE, HIGH SIDE
C          TIRVH -    "         "     "   TEMPERATURE, HIGH SIDE
C          RFRVL - REFRIGERENT FLOW - REVERSING VALVE
C          TARVL - AMBIENT TEMPERATURE - REVERSING VALVE
C
C       THE FOLLOWING ARE APPROXIMATIONS TO THE CONDITIONS OUT
C       OF THE THROTTLING DEVICE ( _OFLD ) SUFFICIENT FOR A
C       SIMPLIFIED HI-RE-LI MODEL, IN COOLING MODE.
      TOFLD = TIE
      XOFLD = XIE
      HOFLD = HIE
C
      DO 1221 INDEX = 1, 10, 1
C          UPDATE CONDITIONS OUT OF REVERSING VALVE
      PORVL = PINCMP + DPLAC
      HORVL = HINCMP - DHLAC
      TSORV = TSAT ( PORVL, IEROR )
      CALL SATPRP ( TSORV, PDUMY, VFORV, VGORV, HFORV, HLORV,
```

```
     1      HVORV, SFORG, SGORG, IEROR )
      CALL MUKCP  ( TSORV, VSORF, VSORV, VSORG, TKORF, TKORV,
     1      TKORG, CPORF, CPORV, CPORG, IEROR )
C
      IF ( HORVL .GE. HVORV )  GO TO 1211
      IF ( (HORVL .LE. HVORV) .AND. (HORVL .GE. HFORV) )
     1      GO TO 1212
      IF ( HORVL .LE. HFORV )  GO TO 1213
C AA
      WRITE (NOTTT,*) ' COMPV:AA: INVALID HORVL, HVORV, HFORV = ',
     1      HORVL, HVORV, HFORV
      XORVL = XINCMP
      TORVL = TRICMP
      GO TO 1214
C
1211  CONTINUE
C           SUPERHEATED VAPOR LEAVING REVERSING VALVE
      XORVL = 1.0
      TORVL = (HORVL - HVORV) / CPORV + TSORV
      GO TO 1214
C
1212  CONTINUE
C           TWO-PHASE LIQUID LEAVING REVERSING VALVE
      XORVL = (HORVL - HFORV) / (HVORV - HFORV)
      TORVL = TSORV
      GO TO 1214
C
1213  CONTINUE
C           LIQUID LEAVING REVERSING VALVE (THIS SHOULD NEVER HAPPEN!)
      XORVL = 0.0
      TORVL = TSORV - (HFORV-HORVL)/CPORF
      GO TO 1214
C
1214  CONTINUE
```

```
      PIRVH = POUCMP
      TIRVH = TROCMP
      HIRVH = HOUCMP
      RFRVM = XMRCM - RFVLK/2.0
      RFRVL = XMRCM
      TARVL = TAMBRV
      IF (NRVALV .NE. 0)
     1      CALL VALVER ( NRVALV, PORVL, TORVL, XORVL, PIRVH, TIRVH,
     2           RFRVM, TARVL,
     3           DPLOV, DPHIV, QINTV, QEXTV, PRINT )
      DPDL   = DPDL1 + DPDL2
      PIC    = POUCMP - DPDL - DPHIV
      TSATCI = TSAT(PIC,IERROR)
      CALL SATPRP(TSATCI,Y,VFC,VGC,HFC,HFGC,HGC,BBB,CCC,IERROR)
      HIC   = HOUCMP - ( (QDISLN)/XMRCM + (QINTV+QEXTV)/RFRVM )
      XIC = 1.0
C
      IF (HIC.GT.HGC) GO TO 1115
C          TWO-PHASE FLOW INTO CONDENSOR
      XIC = (HIC-HFC)/HFGC
      VIC = VFC+XIC*(VGC-VFC)
      TIC = TSATCI
      GO TO 1125
C
1115  CONTINUE
C          SUPERHEATED VAPOR INTO CONDENSOR
      CALL TRIAL(TSATCI,50.0,PIC,3,HIC,TOLHL,VIC,YYY,ZZZ,TIC,IERROR)
      IF (IERROR .NE. 0) WRITE(6,1040)
C
1125  CONTINUE
C
C***           CALCULATE THE REFRIGERANT CONDITIONS AT EVAPORATOR EXIT
C
C
```

```
      PIRVH = POUCMP
      TIRVH = TROCMP
      HIRVH = HOUCMP
      RFRVM = XMRCM - RFVLK/2.0
      RFRVL = XMRCM
      TARVL = TAMBRV
      IF (NRVALV .NE. 0)
     1     CALL VALVER ( NRVALV, PORVL, TORVL, XORVL, PIRVH, TIRVH,
     2          RFRVM, TARVL,
     3          DPLOV, DPHIV, QINTV, QEXTV, PRINT )
      DPDL   = DPDL1 + DPDL2
      PIC    = POUCMP - DPDL - DPHIV
      TSATCI = TSAT(PIC,IERROR)
      CALL SATPRP(TSATCI,Y,VFC,VGC,HFC,HFGC,HGC,BBB,CCC,IERROR)
      HIC   = HOUCMP - ( (QDISLN)/XMRCM + (QINTV+QEXTV)/RFRVM )
      XIC = 1.0
C
      IF (HIC.GT.HGC) GO TO 1115
C          TWO-PHASE FLOW INTO CONDENSOR
      XIC = (HIC-HFC)/HFGC
      VIC = VFC+XIC*(VGC-VFC)
      TIC = TSATCI
      GO TO 1125
C
1115  CONTINUE
C          SUPERHEATED VAPOR INTO CONDENSOR
      CALL TRIAL(TSATCI,50.0,PIC,3,HIC,TOLHL,VIC,YYY,ZZZ,TIC,IERROR)
      IF (IERROR .NE. 0) WRITE(6,1040)
C
1125  CONTINUE
C
C***            CALCULATE THE REFRIGERANT CONDITIONS AT EVAPORATOR EXIT
C
C
```

```
C***            CALCULATE PRESSURE DROP IN SUCTION LINE, DPSL
C***    DPSL1 IS THE PRESSURE DROP FROM THE COIL TO THE REVERSING VALVE
C***    DPSL2 IS THE PRESSURE DROP FROM THE REVERSING VALVE
C***          TO THE COMPRESSOR INLET
C***    DPSL  IS THE TOTAL PRESSURE DROP IN THE LOW SIDE LINES
C
      RHOVL = 1.0/VINCMP
      CALL MUKCP(TRICMP,XXX,XMUVL,YYY,ZZZ,AAA,BBB,CCC,DDD,EEE,IMUKCP)
C
C          CHANGES ADDED 5/6/85 AT WESTINGHOUSE R&D
C
      GO TO ( 1231, 1233 ) NCORH
C AB
      WRITE (NOTTT,*) ' COMPV:AB: INVALID NCORH = ', NCORH
      GO TO 1233
C
1233  CONTINUE
C          ## HEATING CASE
      CALL DHSUL ( NACCU, ACPAR, NACPR, NCORH,
     1           PINCMP, TRICMP, XINCMP, XORVL, POC, TOC, XOC,
     2           XMRCM/3600.0,
     3           DHLAC, DHHAC, DPLAC, DPHAC )
      GO TO 1239
C
1231  CONTINUE
C          ## COOLING CASE
      CALL DHSUL ( NACCU, ACPAR, NACPR, NCORH,
     1           PINCMP, TRICMP, XINCMP, XORVL, PIE+DPHAC, TOFLD, XOFLD,
     2           XMRCM/3600.0,
     3           DHLAC, DHHAC, DPLAC, DPHAC )
C
C          AT THIS POINT, IMPROVED CALCULATIONS OF _OFLD SHOULD BE
C          INSERTED, IF A GOOD WESTINGHOUSE HI-RE-LI MODEL IS DESIRED.
      GO TO 1239
```

```
C
1239  CONTINUE
C CA
      WRITE (NOTTT,*) ' COMPV:CA: RETURN FROM DHSUL; PINCMP,TRICMP,',
     1       'XINCMP, XORVL, POC, TOC, XOC, XMRCM , ',
     2       'DHLAC, DHHAC, DPLAC, DPHAC = '
      WRITE (NOTTT,*) PINCMP, TRICMP, XINCMP, XORVL, POC, TOC, XOC,
     1       XMRCM, DHLAC, DHHAC, DPLAC, DPHAC
C
C          END OF 5/6/85 WESTINGHOUSE R&D CHANGES
C
      CALL DPLINE (DSL,   XLEQSL, E, XMR,   RHOVL, XMUVL, DPSL1)
      CALL DPLINE (DSLRV, XLEQLP, E, XMRCM, RHOVL, XMUVL, DPSL2)
      DPSL = DPSL1 + DPSL2
      IF (SUPER.LE.0.0) DPSL = DPSL*1.9
      POE = PINCMP + DPSL + DPLOV + DPLAC
      TSATEO = TSAT(POE,IERROR)
      CALL SATPRP (TSATEO,XXX,VFE,VGE,HFE,HFGE,HGE,AAA,BBB,IERROR)
C       ## GET THE REVERSING VALVE LEAKAGE
      RFVLK = FLKRV ( NRVLK, NLPMX, RVLKP, XMRCM/3600.0,
     1       POUCMP, TOUCMP, HOUCMP, PINCMP, TRICMP, HINCMP )
C          AND CONVERT TO LBM/HR
      RFVLK = RFVLK * 3600.0
C       ## GET THE REFRIGERENT FLOW THROUGH THE REST OF THE SYSTEM
      XMR    = XMRCM - RFVLK
C
      HOE    = (HINCMP-DHLAC)*(XMRCM/XMR) - HIRVH*(RFVLK/XMR)
     1       - (QSUCLN+QINTV)/(RFRVM)
      XOE = 1.0
      IF (HOE.GE.HGE) GO TO 40
      XOE = (HOE-HFE) / HFGE
      VOE = VFE + XOE*(VGE-VFE)
      TROE = TSATEO
      SUPERE = -XOE
```

```
      GO TO 50
C
   40 CONTINUE
      CALL TRIAL(TSATEO,10.0,POE,3,HOE,TOLHL,VOE,X,Y,
     1       TROE,IERROR)
      IF (IERROR .NE. 0) WRITE(6,1010)
      SUPERE = TROE-TSATEO
   50 CONTINUE
C
C        STATEMENT 1221 IS END OF ITERATION ON SUCTION LINE DEVICES
C        (THIS ITERATION ADDED 5/9/85 AT WESTINGHOUSE R&D BY R. LUCHETA)
C        IF THERE IS NO SUCTION LINE ACCUMULATOR, NO ITERATION IS
C        ATTEMPTED.
      IF ( NACCU .EQ. 0 )  GO TO 1222
1221  CONTINUE
C
1222  CONTINUE
C
C
C***             CALCULATE ENTHALPY AFTER ISENTROPIC COMPRESSION
C***             FROM SHELL INLET CONDITIONS
C
      IF (SINCMP.GT.SGO) GO TO 1065
      XISEN = (SINCMP-SFO)/(SGO-SFO)
      HISEN = HFO+XISEN*HFGO
      GO TO 1075
1065  CONTINUE
      CALL TRIAL(TSOCMP,50.0,POUCMP,4,SINCMP,TOLSL,XXX,HISEN,
     1           YYY,ZZZ,IERROR)
      IF (IERROR .NE. 0) WRITE(6,1020)
1075  CONTINUE
C
C***             CALCULATE 'ETATOT' AND 'ETAVOL' BASED ON
C***             SHELL INLET CONDITIONS
```

```
C
      ETATOT = XMRCM*(HISEN-HINCMP)/POW
      ETASUP = (HISEN-HINCMP)/(HISENS-HINSP)
      ETAVOL = XMRCM*VINCMP/(60.0*SYNC*DISPL/1728.0)
      PRATIO = POUCMP/PINCMP
C
      POW   = POW / 3413.0
      IF (.NOT. PRINT) GO TO 400
      WRITE(6,1050) TSATEO, DPSL,
     1              TSICMP, PINCMP,
     2              TSOCMP, POUCMP,
     3              TSATCI, DPDL
      IF (NRVALV .NE. 0) WRITE(6,1058) DPHIV,DPLOV,QINTV,QEXTV
      IF (SUPERE.GE.0.0) WRITE(6,1051) SUPERE
      IF (SUPERE.LT.0.0) WRITE(6,1052) XOE
      IF (SUPER.GE.0.0)  WRITE(6,1053) SUPER , QSUCLN
      IF (SUPER.LT.0.0)  WRITE(6,1054) XINCMP, QSUCLN
      WRITE(6,1055) TROCMP, QCAN,
     1              TIC, QDISLN
      WRITE(6,1011) POW, ETAISN, ETATOT,
     1              RPM, ETAMOT, ETAVOL,
     2            PCTFL, ETAMEC, PRATIO,
     3            FLMOT, ETASUP
      WRITE(6,1012) QHILO
      IF (XINSP .EQ. 1.0) WRITE(6,1013) TRISP
      IF (XINSP.NE. 1.0)  WRITE(6,1014) XINSP
      WRITE(6,1015) XMRCM, (XMRCM-XMR)
  400 CONTINUE
      RETURN
C
C      ##################################################
C
 1002 FORMAT('0  COMP: ***** ERROR WAS IN THE CALL TO "TRIAL"',
     1       ' AFTER STATEMENT 45 *****',/)
```

```
1003 FORMAT('0  COMP: ***** ERROR WAS IN CALL TO "TRIAL" AFTER',
   1       ' STATEMENT 60 *****',/)
1004 FORMAT('0  COMP: ***** ERROR WAS IN CALL TO "TRIAL" AFTER',
   1       ' STATEMENT 80 *****',/)
1005 FORMAT('0  COMP: INCONSISTENT DATA SET',/,
   1       '    HEAT TRANSFER FROM DISCH. TO SUCT. GAS  = ',G12.3,/,
   2       '    MOTOR & MECH HEAT GENERATION   = ',G12.3,/,
   3       '    QCAN  = ',G12.3,3X,'DELTA H SHELL TO SUCTION',
   4       ' PORT  = ',G12.3,/,'       NITR  = ',I2)
1006 FORMAT('0  COMP: *****   FAILED TO CONVERGE ON HINSP  = ',F8.2,
   1       5X,'H1OLD  = ',F8.2,5X,'*****')
1010 FORMAT('0  COMP: ***** ERROR IN "TRIAL" WHEN   COMPUTING THE ',
   1                 'SHELL INLET CONDITIONS *****',/)
1020 FORMAT('0  COMP: ***** ERROR IN "TRIAL" WHEN   COMPUTING THE ',
   1                 'ENTHALPY AFTER ISENTROPIC COMPRESSION *****',/)
1030 FORMAT('0  COMP: ***** ERROR IN "TRIAL" WHEN   COMPUTING THE ',
   1                 'SHELL OUTLET CONDITIONS *****',/)
1040 FORMAT('0 COMP: ***** ERROR IN "TRIAL" WHEN COMPUTING THE ',
   1                 'STATE ENTERING THE CONDENSER *****',/)
1050 FORMAT('0  COMP: SATURATION TEMPERATURE LEAVING  EVAPORATOR  ',
   1   F8.3,' F',4X,'PRESSURE DROP IN SUCTION LINE    ',F8.3,' PSI'
   2   /,'          SATURATION TEMPERATURE ENTERING COMPRESSOR  ',
   3   F8.3,' F',4X,'PRESSURE ENTERING COMPRESSOR     ',F8.3,' PSIA'
   4   /,'          SATURATION TEMPERATURE LEAVING  COMPRESSOR  ',
   5   F8.3,' F',4X,'PRESSURE LEAVING  COMPRESSOR     ',F8.3,' PSIA'
   6   /,'          SATURATION TEMPERATURE ENTERING CONDENSER   ',
   7   F8.3,' F',4X,'PRESSURE DROP IN DISCHARGE LINE  ',F8.3,' PSI')
1051 FORMAT('0         SUPERHEAT   LEAVING  EVAPORATOR  ',F8.3,' F')
1052 FORMAT('0         QUALITY     LEAVING  EVAPORATOR  ',F8.4)
1053 FORMAT('          SUPERHEAT   ENTERING COMPRESSOR  ',F8.3,' F',4X,
   1   'HEAT GAIN IN SUCTION LINE        ',F8.2,' BTU/H')
1054 FORMAT('          QUALITY     ENTERING COMPRESSOR  ',F8.4,' F',4X,
   1   'HEAT GAIN   IN SUCTION LINE      ',F8.2,' BTU/H')
1055 FORMAT('          TEMPERATURE LEAVING  COMPRESSOR  ',F8.3,' F',4X,
```

```
     1    'HEAT LOSS FROM COMPRESSOR SHELL  ',F8.2,' BTU/H',
     2    /,'          TEMPERATURE ENTERING CONDENSER   ',F8.3,' F',4X,
     3    'HEAT LOSS IN   DISCHARGE LINE    ',F8.2,' BTU/H')
 1058 FORMAT(' COMPV: REVERSING VALVE PRESSURE DROP AND ',
     1       'HEAT TRANSFER',/,
     2        '          HIGH SIDE PRESSURE DROP ',F8.3,' PSI'
     3        '          LOW SIDE PRESSURE DROP  ',F8.3,' PSI',/,
     4        '         INTERNAL HEAT TRANSFER  ',F8.3,' BTU/HR',
     5        '       EXTERNAL HEAT TRANSFER  ',F8.3,' BTU/HR',/)
 1011 FORMAT('OCOMPRESSOR PERFORMANCE',/,
     1       48X,'INTERNAL EFFICIENCIES',11X,'OVERALL EFFICIENCIES',/,
     2        '          COMPRESSOR POWER   ',F10.3,' KW ',7X,
     3                   'ISENTROPIC           ',F6.4,5X,
     4                   'ISENTROPIC            ',F6.4,/,
     5        '          MOTOR SPEED        ',F10.3,' RPM',7X,
     6                   'MOTOR                ',F6.4,5X,
     7                   'VOLUMETRIC            ',F6.4,/,
     8        '          FRACTION OF FULL LOAD',F8.3,11X,
     9                   'MECHANICAL           ',F6.4,
     +                 4X,'AT A PRESSURE RATIO OF ',F5.3,/,
     1        '          FULL LOAD MOTOR OUTPUT ',F6.3,' KW ',7X,
     2                'SUCTION GAS HEATING  ',F6.4)
 1012 FORMAT('0          DISCHARGE  TUBE   HEAT LOSS',F10.3,' BTU/H')
 1013 FORMAT('           TEMPERATURE AT SUCTION PORT',F10.3,' F')
 1014 FORMAT('        ***** QUALITY  AT SUCTION PORT',F10.4,
     1                  ' *****',/)
 1015 FORMAT ( 1HO, 8X, 'COMPRESSOR REFRIGERANT MASS FLOW ',
     1       'RATE = ', F10.3, ' LBM/HR' /
     2       1X, 8X, 'REVERSING VALVE LEAKAGE = ', F10.5,
     3       ' LBM/HR' )
C
C     ##################################################
C
      END
```

Listing of the COMPMP Subroutine

A listing of the COMPMP subroutine follows:

```
      SUBROUTINE COMPMP ( TOLSD, TOLHD, CMPND )
C
C***              MAP-BASED MODEL OF COMPRESSOR
C
C          ##################################################
C
      LOGICAL PRINT
C
      COMMON / A1 / PRINT
C
      COMMON / CMPRSR / TRICMP, TSICMP, HINCMP, PINCMP, XINCMP,
     1                  TROCMP, TSOCMP, HOUCMP, POUCMP, XOUCMP
C
      COMMON / COMPR / VR,     SYNC,   FLMOT, EFFMMX, ETAISN, ETAMEC,
     1                 ETAVLA, ETAVLB, POW,   CANFAC, HILOFC, QCAN,
     2                 QHILO,  DISPL,  MTRCLC
C
      COMMON / CONDSR / TAIIC,  TIC, TSATCI, HIC, PIC, XIC,
     1                  TAOC,  TROC, TSATCO, HOC, POC, XOC
C
      COMMON / EVAPTR / TAIIE,  TIE, TSATEI, HIE, PIE, XIE,
     1                  TAOE,  TROE, TSATEO, HOE, POE, XOE
C
      COMMON / FLOWBA / DTROC, SUPER, CAPFLO, ORIFD, XMR, NCAP,
     1                  IREFC, ICOMP, ITRPIE
C
      COMMON / LINES / DLL, XLEQLL, DSL,   XLEQSL, DDL, XLEQDL,
     1               DSLRV, XLEQLP, DDLRV, XLEQHP,DPDL,   DPSL,
```

```
     2                    DPLL, QDISLN, QSUCLN,QLIQLN,E
C
      COMMON / RVALVE / TAMBRV, NRVALV, DPLOV, DPHIV, QINTV, QEXTV
C
      COMMON / MAPFIT / CPOW(6),CXMR(6),SUCFAC,VOLFAC,SUPERB,
     1                     DISPLB, POWCOR, XMRCOR
C
      COMMON / PRNT1 / RPM, PCTFL, ETAMOT, ETAVOL, ETASUP, ETATOT,
     1                   PRATIO
C
      COMMON / SUPERE / SUPERE
C
C          ** ADDED AT WESTINGHOUSE R&D ON 5/9/85 BY R. LUCHETA.
C
      COMMON / ACPAR / DHHAC, DHLAC, DPHAC, DPLAC, NACCU,
     1       NACPR, ACPAR(20)
C       ## SEE BLOCK DATA PROGRAM RLADP FOR THE DEFINITION OF
C          THE ABOVE VARIABLES. ##
C
      COMMON / MPASS / CNDCON, AMBCON, EVPCON, CONMST, CMPCON, FLOCON,
     1       TOLS, TOLH, LPRINT, NCORH, MCMPOP, MFANIN,
     2       MFANOU, MFANFT
C          THE VARIABLES TOLS, TOLH, AND CMPCON IN THE ORIGIONAL
C          VERSION OF THE PROGRAM WERE RENAMED TO TOLSL, TOLHL,
C          AND CMPNL TO AVOID CONFLICT WITH THE COMMON VARIABLES
C          ABOVE
C
      COMMON / PRNT2 / TFLODV, TSATFL, PFLODV, XFLODV
C
      DATA NOTTT / 6 /
C
C          ** END OF 5/9/85 ADDITIONS.
C
C       ## ADDED ON 8/2/85 AT WESTINGHOUSE R&D - R. LUCHETA
```

```
C
      COMMON / RVLKP / NRVLK, NRDFL, NLPMX, NLKTB(5), RVLKP(5)
C      SEE BLOCK DATA PROGRAM RLADP FOR THE DEFINITION OF
C       THESE PARAMETERS.
C
C      ## END OF 8/2/85 ADDITION ##
C
C          ##################################################
C
1001  CONTINUE
      DPHIV = 0.0
      DPLOV = 0.0
      QINTV = 0.0
      QEXTV = 0.0
      SIZFAC = DISPL/DISPLB
C
C          FOLLOWING ADDED AT WESTINGHOUSE R&D ON 5/9/85 BY R. LUCHETA
C          ## PERFORM "CALL-BY-VALUE" BY CONVERTING FORMAL PARAMETERS
C             TO LOCAL PARAMETERS.
      TOLSL = TOLSD
      TOLHL = TOLHD
      CMPNL = CMPND
C          ## SET ACCUMULATOR LOW AND HIGH SIDE ENTHALPY AND PRESSURE
C             CHANGES TO 0.0 .
      DHHAC = 0.0
      DHLAC = 0.0
      DPHAC = 0.0
      DPLAC = 0.0
C     ## AND A NOTATIONAL CHANGE
      TOC    = TROC
C          END OF 5/9/85 ADDITIONS.
C
C***            CALCULATE SUCTION LINE CONDITIONS
C
```

```
      CALL SATPRP(TSICMP,PINCMP,VFI,VGI,HFI,HFGI,XX,SFI,SGI,IERROR)
      IF(SUPER.GE.0.0) GO TO 10
C          ELSE        COMPRESSOR INLET FLOW IS TWO-PHASE
      XINCMP = -SUPER
      TRICMP = TSICMP
      HINCMP = HFI + XINCMP*HFGI
      VINCMP = VFI + XINCMP*(VGI-VFI)
      SINCMP = SFI + XINCMP*(SGI-SFI)
      GO TO 20
C
   10 CONTINUE
C          SUPERHEATED VAPOR ENTERING COMPRESSOR
      XINCMP = 1.0
      TRICMP = TSICMP+SUPER
      CALL VAPOR(TRICMP,PINCMP,VINCMP,HINCMP,SINCMP,IERROR)
C
   20 CONTINUE
C
C***             CALCULATE COMPRESSOR POWER AND MASS FLOW RATE
C***             AT BASE SUPERHEAT
C
      POWB  = (CPOW(1)*TSOCMP*TSOCMP + CPOW(2)*TSOCMP +
     1          CPOW(3)*TSICMP*TSICMP + CPOW(4)*TSICMP +
     2          CPOW(5)*TSOCMP*TSICMP + CPOW(6))*3413.
C
      XMRB  =   CXMR(1)*TSOCMP*TSOCMP + CXMR(2)*TSOCMP +
     1          CXMR(3)*TSICMP*TSICMP + CXMR(4)*TSICMP +
     2          CXMR(5)*TSOCMP*TSICMP + CXMR(6)
C
C***             CORRECT FOR ACTUAL SUPERHEAT LEVEL
C
      POWPXM = POWB/XMRB
      POWCOR = 1.0
      XMRCOR = 1.0
```

```
      IF(SUPERB.EQ.SUPER) GO TO 75
      CALL SUPCOR(SUCFAC,VOLFAC,TOLHL,TOLSL,
     1            TSICMP,TSOCMP,HINCMP,SUPERB,
     2            POWPXM,POWCOR,XMRCOR)
   75 CONTINUE
C       GET COMPRESSOR POWER
      POW   = POWB*POWCOR*SIZFAC
C       AND MASS FLOW RATE THROUGH COMPRESSOR
      XMRCM = XMRB*XMRCOR*SIZFAC
C       AND FIRST APPROXIMATION TO REVERSING VALVE LEAKAGE
      RFVLK = FLKRV (NRVLK, NLPMX, RVLKP, XMRCM/3600.0,
     1      POUCMP, TOUCMP, HOUCMP, PINCMP, TRICMP, HINCMP )
C       ## AND CONVERT UNITS BACK
      RFVLK = RFVLK * 3600.0
C       AND THE MASS FLOW THROUGH THE REMAINDER OF THE
C       SYSTEM
      XMR   = XMRCM - RFVLK
C
      IF (CANFAC .LE. 0.0) GO TO 80
      IF (CANFAC .LE. 1.0) QCAN = CANFAC*POW
   80 CONTINUE
C
C***           CALCULATE SHELL EXIT CONDITIONS
C
      CALL SATPRP(TSOCMP,POUCMP,VFO,VGO,HFO,HFGO,HGO,SFO,SGO,IERROR)
      HOUCMP = (POW-QCAN)/XMRCM + HINCMP
      XOUCMP = 1.0
      IF(HOUCMP.GT.HGO) GO TO 90
      XOUCMP = (HOUCMP-HFO)/HFGO
      VOUCMP = VFO+XOUCMP*(VGO-VFO)
      TROCMP = TSOCMP
      GO TO 100
   90 CONTINUE
      CALL TRIAL(TSOCMP,50.,POUCMP,3,HOUCMP,TOLHL,VOUCMP,XXX,
```

```
     1              YYY,TROCMP,IERROR)
      IF (IERROR .NE. 0) WRITE(6,1030)
  100 CONTINUE
C
C***            CALCULATE THE REFRIGERANT CONDITIONS ENTERING THE
C***            CONDENSER
C
C
C               CALCULATE THE PRESSURE DROP 'DPDL' (PSI) OF VAPOR
C               IN THE DISCHARGE LINE AND 'POUCMP' (PSIA) THE
C               PRESSURE AT THE COMPRESSOR EXIT
C
C***    DPDL1 IS THE PRESSURE DROP FROM THE REVERSING VALVE TO THE COIL
C***    DPDL2 IS THE PRESSURE DROP FROM THE COMPRESSOR DISCHARGE
C***          TO THE REVERSING VALVE
C***    DPDL  IS THE TOTAL PRESSURE DROP IN THE HIGH SIDE
C
      CALL MUKCP(TROCMP,XXX,XMUVH,YYY,ZZZ,AAA,BBB,CCC,DDD,EEE,IMUKCP)
      RHOVH = 1./VOUCMP
      CALL DPLINE (DDL, XLEQDL, E, XMR, RHOVH, XMUVH, DPDL1)
      CALL DPLINE (DDLRV, XLEQHP, E, XMRCM, RHOVH, XMUVH, DPDL2)
C
C          ## SECTION ADDED 5/10/85 AT WESTINGHOUSE R&D BY R. LUCHETA
C          NOW ITERATE ON THE SUCTION LINE REVERSING VALVE AND
C          ACCUMULATOR PRESSURE DROPS TO GET THE CONSISTENT EVAPORATOR
C          OUTLET CONDITIONS.
C           PORVL - REVERSING VALVE OUTLET PRESSURE, LOW SIDE
C           TORVL -    "        "      "    TEMPERATURE, LOW SIDE
C           XORVL -    "        "      "    QUALITY, LOW SIDE
C           PIRVH -    "        "    INLET  PRESSURE, HIGH SIDE
C           TIRVH -    "        "      "    TEMPERATURE, HIGH SIDE
C           HIRVH -    "        "      "    ENTHALPY, HIGH SIDE
C           RFRVL - REFRIGERENT FLOW - REVERSING VALVE
C           TARVL - AMBIENT TEMPERATURE - REVERSING VALVE
```

```
C
C          THE FOLLOWING ARE APPROXIMATIONS TO THE CONDITIONS OUT
C          OF THE THROTTLING DEVICE ( _OFLD ) SUFFICIENT FOR A
C          SIMPLIFIED HI-RE-LI MODEL, IN COOLING MODE.
      TOFLD = TIE
      XOFLD = XIE
      HOFLD = HIE
C
      DO 1221 INDEX = 1, 10, 1
C          UPDATE CONDITIONS OUT OF REVERSING VALVE
      PORVL = PINCMP + DPLAC
      HORVL = HINCMP - DHLAC
      TSORV = TSAT ( PORVL, IEROR )
      CALL SATPRP ( TSORV, PDUMY, VFORV, VGORV, HFORV, HLORV,
     1       HVORV, SFORG, SGORG, IEROR )
      CALL MUKCP  ( TSORV, VSORF, VSORV, VSORG, TKORF, TKORV,
     1       TKORG, CPORF, CPORV, CPORG, IEROR )
C
      IF ( HORVL .GE. HVORV ) GO TO 1211
      IF ( (HORVL .LE. HVORV) .AND. (HORVL .GE. HFORV) )
     1       GO TO 1212
      IF ( HORVL .LE. HFORV ) GO TO 1213
C AA
      WRITE ( NOTTT,* ) ' COMPMP:AA: INVALID HORVL = ', HORVL
      XORVL = XINCMP
      TORVL = TRICMP
      GO TO 1214
C
1211  CONTINUE
C        SUPERHEATED VAPOR LEAVING REVERSING VALVE
      XORVL = 1.0
      TORVL = (HORVL-HVORV) / CPORV + TSORV
      GO TO 1214
C
```

```
1212  CONTINUE
C        TWO-PHASE LIQUID LEAVING REVERSING VALVE
      XORVL = (HORVL-HFORV) / (HVORV-HFORV)
      TORVL = TSORV
      GO TO 1214
C
1213  CONTINUE
C        LIQUID LEAVING REVERSING VALVE (THIS SHOULD NEVER HAPPEN!!)
      XORVL = 0.0
      TORVL = TSORV - (HFORV-HORVL) / CPORF
      GO TO 1214
C
1214  CONTINUE
      PIRVH = POUCMP
      TIRVH = TROCMP
      HIRVH = HOUCMP
      RFRVM = XMRCM - RFVLK/2.0
      RFRVL = XMRCM
      TARVL = TAMBRV
      IF (NRVALV .NE. 0)
     1      CALL VALVER ( NRVALV, PORVL, TORVL, XORVL, PIRVH, TIRVH,
     2            RFRVM, TARVL,
     3            DPLOV, DPHIV, QINTV, QEXTV, PRINT )
C
C         NOW UPDATE CONDITIONS INTO THE CONDENSOR
      IF(PRINT) WRITE(6,1059)
      DPDL  = DPDL1 + DPDL2
      PIC   = POUCMP - DPDL - DPHIV
      TSATCI = TSAT(PIC,IERROR)
      CALL SATPRP(TSATCI,Y,VFC,VGC,HFC,HFGC,HGC,BBB,CCC,IERROR)
      HIC = HOUCMP - ( QDISLN/XMRCM + QINTV/RFRVM + QEXTV/XMR )
      XIC = 1.0
C
      IF(HIC.GT.HGC) GO TO 120
```

```
C         ELSE        CONDENSOR ENTRANCE CONDITIONS ARE TWO-PHASE
      XIC = (HIC-HFC)/HFGC
      VIC = VFC+XIC*(VGC-VFC)
      TIC = TSATCI
      GO TO 130
C
  120 CONTINUE
C         SUPERHEATED VAPOR INTO CONDENSOR
      CALL TRIAL(TSATCI,50.,PIC,3,HIC,TOLHL,VIC,YYY,ZZZ,TIC,IERROR)
      IF (IERROR .NE. 0) WRITE(6,1040)
C
  130 CONTINUE
C
C***           CALCULATE THE REFRIGERANT CONDITIONS AT EVAPORATOR EXIT
C
C
C***           CALCULATE PRESSURE DROP IN SUCTION LINE, DPSL
C***    DPSL1 IS THE PRESSURE DROP FROM THE COIL TO THE REVERSING VALVE
C***    DPSL2 IS THE PRESSURE DROP FROM THE REVERSING VALVE
C***          TO THE COMPRESSOR INLET
C***    DPSL  IS THE TOTAL PRESSURE DROP IN THE LOW SIDE LINES
C
      RHOVL = 1.0/VINCMP
      CALL MUKCP(TRICMP,XXX,XMUVL,YYY,ZZZ,AAA,BBB,CCC,DDD,EEE,IMUKCP)
C
C         CHANGES ADDED 5/10/85 AT WESTINGHOUSE R&D
C
      GO TO ( 1231, 1233 ) NCORH
C AB
      WRITE (NOTTT,*) ' COMPMP:AB: INVALID NCORH = ', NCORH
      GO TO 1233
C
1233  CONTINUE
C         ## HEATING MODE
```

```
      CALL DHSUL ( NACCU, ACPAR, NACPR, NCORH,
     1            PINCMP, TRICMP, XINCMP, XORVL, POC, TOC, XOC,
     2            XMRCM/3600.0,
     3            DHLAC, DHHAC, DPLAC, DPHAC )
      GO TO 1239
C
1231  CONTINUE
C          ## COOLING MODE
      CALL DHSUL ( NACCU, ACPAR, NACPR, NCORH,
     1            PINCMP, TRICMP, XINCMP, XORVL, PIE*DPHAC, TOFLD, XOFLD,
     2            XMRCM/3600.0,
     3            DHLAC, DHHAC, DPLAC, DPHAC )
      GO TO 1239
C
1239  CONTINUE
C CA
C     WRITE (6,*) ' COMPMP:CA: DHSUL RETURN; '
C     WRITE (6,*) '   PINCMP, TRICMP, XINCMP, XORVL, POC, TOC, XOC, ',
C    1       ' XMRCM, DHLAC, DHHAC, DPLAC, DPHAC  = '
C     WRITE (6,*) PINCMP, TRICMP, XINCMP, XORVL, POC, TOC, XOC, XMRCM
C     WRITE (6,*) '  ', DHLAC, DHHAC, DPLAC, DPHAC
C CB
C     WRITE (6,*) ' COMPMP:CB: XORVL, HORVL, HFORV, HVORV, ',
C    1       'HLORV, TSORV = '
C     WRITE (6,*) '  ', XORVL, HORVL, HFORV, HVORV, HLORV, TSORV
C
C         END OF 5/10/85 WESTINGHOUSE R&D CHANGES
C
      CALL DPLINE (DSL, XLEQSL, E, XMR, RHOVL, XMUVL, DPSL1)
      CALL DPLINE (DSLRV, XLEQLP, E, XMRCM, RHOVL, XMUVL, DPSL2)
      DPSL = DPSL1 + DPSL2
      IF(SUPER.LE.0.0) DPSL = DPSL*1.9
      POE = PINCMP + DPSL + DPLOV + DPLAC
      TSATEO = TSAT(POE,IERROR)
```

```
      CALL SATPRP(TSATEO,XXX,VFE,VGE,HFE,HFGE,HGE,SFE,SGE,IERROR)
C       ## NOW UPDATE LEAKAGES AND EVAPORATOR OUTLET CONDITIONS ##
      RFVLK = FLKRV ( NRVLK, NLPMX, RVLKP,
     1       XMRCM/3600.0, PIRVH, TIRVH, HIRVH,
     2       PORVL, TORVL, HORVL )
C       ## AND CONVERT UNITS BACK
      RFVLK = RFVLK * 3600.0
      XMR   = XMRCM - RFVLK
      RFRVL = XMRCM
      RFRVM = XMRCM - RFVLK/2.0
      HOE   = (HINCMP-DHLAC)*(XMRCM/XMR) - HIRVH*(RFVLK/XMR)
     1       - (QSUCLN+QINTV)/(RFRVM)
C AC
C     WRITE (NOTTT,*) ' COMPMP:AC: REVERSING VALVE LEAKAGE EFFECTS'
C     WRITE (NOTTT,*) ' COMPMP:AC: HINCMP, DHLAC, HIRVH, PIRVH,',
C    1       ' TIRVH, XMR  = '
C     WRITE (NOTTT,*) '  ', HINCMP, DHLAC, HIRVH, PIRVH, TIRVH, XMR
C     WRITE (NOTTT,*) ' COMPMP:AC: PORVL, XORVL, TORVL, XMRCM, ',
C    1       'RFRVM, RFVLK = '
C     WRITE (NOTTT,*) '  ', PORVL, XORVL, TORVL, XMRCM, RFRVM, RFVLK
C     WRITE (NOTTT,*) ' COMPMP:AC: HOE, HIC = '
C     WRITE (NOTTT,*) '  ', HOE, HIC
C
      XOE = 1.0
C
      IF(HOE.GE.HGE) GO TO 40
C           TWO-PHASE FLOW OUT OF THE EVAPORATOR
      XOE = (HOE-HFE) / HFGE
      VOE = VFE + XOE*(VGE-VFE)
      TROE = TSATEO
      SUPERE = -XOE
      GO TO 50
C
   40 CONTINUE
```

```
C          SUPERHEATED VAPOR OUT OF THE EVAPORATOR
      CALL TRIAL(TSATEO,10.0,POE,3,HOE,TOLHL,VOE,X,Y,
     1        TROE,IERROR)
      IF (IERROR .NE. 0) WRITE(6,1010)
      SUPERE = TROE-TSATEO
C
   50 CONTINUE
C          STATEMENT 1221 IS THE END OF THE ITERATION ON SUCTION LINE
C          DEVICES
C          (AS ADDED ON 5/10/85 BY R. LUCHETA).  IF THERE IS NO SUCTION
C          LINE ACCUMULATOR, NO ITERATION IS ATTEMPTED
 1221 CONTINUE
C
 1222 CONTINUE
C
C***            CALCULATE ENTHALPY AFTER ISENTROPIC COMPRESSION
C***            FROM SHELL INLET CONDITIONS
C
      IF(SINCMP.GT.SGO) GO TO 60
      XISEN = (SINCMP-SFO)/(SGO-SFO)
      HISEN = HFO+XISEN*HFGO
      GO TO 70
   60 CONTINUE
      CALL TRIAL(TSOCMP,50.,POUCMP,4,SINCMP,TOLSL,XXX,HISEN,
     1            YYY,ZZZ,IERROR)
      IF (IERROR .NE. 0) WRITE(6,1020)
   70 CONTINUE
C
C***            CALCULATE 'ETATOT' AND 'ETAVOL' BASED ON
C***            SHELL INLET CONDITIONS
C
      ETATOT = XMRCM*(HISEN-HINCMP)/POW
      ETAVOL = XMRCM*VINCMP/(60.*SYNC*DISPL/1728.)
      PRATIO = POUCMP/PINCMP
```

```
C
      POW = POW/3413.
      IF (.NOT. PRINT) GO TO 150
      WRITE(6,1050) TSATEO, DPSL,
     1              TSICMP, PINCMP,
     2              TSOCMP, POUCMP,
     3              TSATCI, DPDL
      IF(NRVALV .NE. 0) WRITE(6,1058) DPHIV,DPLOV,QINTV,QEXTV
      IF(SUPERE.GE.0.0) WRITE(6,1051) SUPERE
      IF(SUPERE.LT.0.0) WRITE(6,1052) XOE
      IF(SUPER.GE.0.0)  WRITE(6,1053) SUPER , QSUCLN
      IF(SUPER.LT.0.0)  WRITE(6,1054) XINCMP, QSUCLN
      WRITE(6,1055) TROCMP, QCAN,
     1              TIC, QDISLN
      WRITE(6,1056) POWPXM,POWCOR,XMRCOR
      WRITE(6,1057) POW, ETATOT,
     1              XMRCM, ETAVOL,
     2              SYNC, PRATIO
  150 CONTINUE
      RETURN
C
C        ##################################################
C
 1010 FORMAT(' COMPMP: ***** ERROR IN "TRIAL" WHEN COMPUTING THE ',
     1                 'SHELL INLET CONDITIONS *****',/)
 1020 FORMAT(' COMPMP: ***** ERROR IN "TRIAL" WHEN COMPUTING THE ',
     1                 'ENTHALPY AFTER ISENTROPIC COMPRESSION *****',/)
 1030 FORMAT(' COMPMP: ***** ERROR IN "TRIAL" WHEN COMPUTING THE ',
     1                 'SHELL OUTLET CONDITIONS *****',/)
 1040 FORMAT(/,' COMPMP: ***** ERROR IN "TRIAL" WHEN COMPUTING THE ',
     1                 'STATE ENTERING THE CONDENSER *****',/)
 1050 FORMAT(' COMPMP: SATURATION TEMPERATURE LEAVING  EVAPORATOR  ',
     1F8.3,' F',4X,'PRESSURE DROP IN SUCTION LINE    ',F8.3,' PSI',
     2/,'         SATURATION TEMPERATURE ENTERING COMPRESSOR  ',
```

```
     3F8.3,' F',4X,'PRESSURE ENTERING COMPRESSOR      ',F8.3,' PSIA',
     4/,'          SATURATION TEMPERATURE LEAVING  COMPRESSOR  ',
     5F8.3,' F',4X,'PRESSURE LEAVING  COMPRESSOR      ',F8.3,' PSIA',
     6/,'          SATURATION TEMPERATURE ENTERING CONDENSER   ',
     7F8.3,' F',4X,'PRESSURE DROP IN DISCHARGE LINE  ',F8.3,' PSI',/)
 1051 FORMAT('          SUPERHEAT   LEAVING  EVAPORATOR  ',F8.3,' F')
 1052 FORMAT('          QUALITY     LEAVING  EVAPORATOR  ',F8.4)
 1053 FORMAT('          SUPERHEAT   ENTERING COMPRESSOR  ',F8.3,' F',4X,
     1   'HEAT GAIN IN SUCTION LINE        ',F8.2,' BTU/H',/)
 1054 FORMAT('          QUALITY     ENTERING COMPRESSOR  ',F8.4,' F',4X,
     1'HEAT GAIN   IN SUCTION LINE      ',F8.2,' BTU/H',/)
 1055 FORMAT('          TEMPERATURE LEAVING  COMPRESSOR  ',F8.3,' F',4X,
     1'HEAT LOSS FROM COMPRESSOR SHELL  ',F8.2,' BTU/H',
     2/,'          TEMPERATURE ENTERING CONDENSER   ',F8.3,' F',4X,
     3'HEAT LOSS IN   DISCHARGE LINE    ',F8.2,' BTU/H',/)
 1056 FORMAT(1H0,8X,'SUPERHEAT CORRECTION TERMS',/,
     1          11X,'BASE POWER PER UNIT MASS FLOW  ',F8.3,
     2              ' BTU/LBM',/,
     3          11X,'POWER      CORRECTION FACTOR    ',F8.3,/,
     4          11X,'MASS FLOW CORRECTION FACTOR    ',F8.3)
 1057 FORMAT('          COMPRESSOR INPUT POWER    ',F8.3,' KW',10X,
     1'OVERALL ISENTROPIC EFFICIENCY    ',F8.4,
     2/,'          REFRIGERANT FLOW RATE     ',F8.3,' LBM/H',7X,
     3'VOLUMETRIC           EFFICIENCY    ',F8.4,
     4/,'          RATED MOTOR SPEED         ',F8.3,' RPM',9X,
     5'COMPRESSOR PRESSURE RATIO        ',F8.4,/)
1058  FORMAT(' COMPMP: REVERSING VALVE PRESSURE DROP AND HEAT TRANSFER',
     1     /,'          HIGH SIDE PRESSURE DROP ',F8.3,' PSI',
     2      '          LOW SIDE PRESSURE DROP  ',F8.3,' PSI',/,
     3      '         INTERNAL HEAT TRANSFER  ',F8.3,' BTU/HR',
     4      '       EXTERNAL HEAT TRANSFER  ',F8.3,' BTU/HR',/)
1059  FORMAT(//)
C
C          ####################################################
```

```
C
      END
```

Listing of the HAIR2 Subroutine

A listing for the HAIR2 Subroutine follows:

```
      SUBROUTINE HAIR2(FINTYP,CPA,PRA,XMUA,FAR,DEA,WT,NT,GA,
     1      FP,DELTA,ST,FPD,NFP,UAF,KPA,RHOA,HA,XHA)
      REAL KPA
C
C     PURPOSE
C       TO COMPUTE AIR-SIDE HEAT TRANSFER COEFFICIENTS FOR DRY
C       COILS WITH SMOOTH, WAVY, OR LOUVERED FINS
C
C     INPUT
C       FINTYP-  TYPE OF PLATE FIN SURFACE
C                = 1 -- SMOOTH FIN
C                = 2 -- WAVY FIN
C                = 3 -- LOUVERED FIN
C                = 4 -- CORRUGATED FIN (BEECHER'S CORRELATION)
C       CPA    - AIR SPECIFIC HEAT AT CONSTANT PRESSURE (BTU/LBM/DEG F)
C       PRA    - PRANDTL NUMBER OF AIR
C       XMUA   - DYNAMIC VISCOSITY OF AIR (LBM/HR-FT)
C       FAR    - RATIO OF FIN HEAT TRANSFER AREA TO TOTAL HEAT TRANSFER
C                AREA
C       DEA    - REFRIGERANT TUBE OUTSIDE DIAMETER (FT)
C       WT     - TUBE HORIZONTAL SPACING (FT)
C       NT     - NUMBER OF TUBE ROWS IN THE FLOW DIRECTION
C       GA     - AIR MASS FLUX BASED ON MINIMUM FREE-FLOW
C                AREA(LBM/HR/FT**2)
C       FP     - FINS PER FOOT OF TUBE LENGTH
C       DELTA  - FIN THICKNESS (FT)
C       ST     - VERTICAL TUBE SPACING (FT)
C       FPD    - FIN PATTERN DEPTH (FT)
C       NFP    - NUMBER OF FIN PATTERNS PER TUBE ROW IN DIRECTION OF AIR
```

```
C                FLOW
C       UAF   -  COIL AIR FACE VELOCITY (FT/MIN)
C       KPA   -  AIR THERMAL CONDUCTIVITY (BTU/HR/FT/DEG F)
C       RHOA  -  AIR DENSITY (LBM/FT**3)
C     OUTPUT
C       HA    -  AIR-SIDE DRY HEAT TRANSFER COEFFICIENT
C                (BTU/HR/FT**2/DEG F)
C       XHA   -  MULTIPLYING FACTOR FOR PATTERN AUGMENTATION
C
C     REFERENCES
C       F.C. MCQUISTON, "CORRELATION OF HEAT, MASS, AND MOMENTUM
C       TRANSPORT COEFFICIENTS FOR PLATE-FIN-TUBE TRANSFER
C       SURFACES WITH STAGGERED TUBE", ASHRAE TRANSACTIONS,
C       VOL.84, PART 1, 1978.
C
C       T. YOSHII, "TRANSIENT TESTING TECHNIQUE FOR HEAT
C       EXCHANGER FIN", REITO, 47:531, 1972, PP.23-29.
C
C       T. SENSHU, ET. AL., "SURFACE HEAT TRANSFER COEFFICIENT OF FINS
C       UTILIZED IN AIR-COOLED HEAT EXCHANGERS", REITO, 54:615,
C       1979, PP.11-17.
C
      REAL NT
      DATA IDIAG /0/
C     WRITE(6,1000) FINTYP,CPA,PRA,XMUA,FAR,DEA,WT,NT,GA
C1000  FORMAT(5X,'FINTYP=',E11.4,'    CPA=',E11.4,'    PRA=',
C    1  E11.4,'   XMUA=',E11.4,/,5X,'   FAR=',E11.4,'    DEA=',
C    2  E11.4,'     WT=',E11.4,'     NT=',E11.4,/,5X,'     GA=',
C    3  E11.4,//)
      ARATIO = 1.0/(1.0 - FAR)
      GXM = GA/XMUA
      RED = GXM*DEA
      XJP = ARATIO**(-0.15) * RED**(-0.4)
      XJ = 0.0014 + 0.2618*XJP
```

```
      RFAC = 1.0
C     WRITE(6,1100) ARATIO,GXM,RED,XJP,XJ
C1100  FORMAT(5X,'ARATIO=',E11.4,'    GXM=',E11.4,'    RED=',
C    1     E11.4,'    XJP=',E11.4,/,5X,'    XJ=',E11.4,//)
      IF (NT .EQ. 4.0) GO TO 4
      REW = GXM*WT
      IF (3000. .LE. REW .AND. REW .LE. 15000.) GO TO 3
C     WRITE(6,2) REW
C2     FORMAT('   HAIR: REYNOLDS NO. = ',G10.4,' IS OUT OF RANGE.')
C
C     IF(REW.LT.2500.0) REW=2500.0
C     IF(REW.GT.15000.0) REW=15000.0
    3 RFAC = (1.0 - 1280.*NT * REW**(-1.2) )/(1. -5120.* REW**(-1.2))
    4 CONTINUE
      FINFAC = 1.0
      IF(FINTYP.EQ.2.0) FINFAC=1.45
      IF(FINTYP.EQ.3.0) FINFAC=1.75
      IF(FINTYP .EQ. 4.0) FINFAC = CORFHA(FP,DELTA,DEA,ST,WT,FPD,
     1    NT,NFP,UAF,KPA,PRA,XMUA,RHOA,IDIAG)
C     WRITE(6,1200) FINFAC,RFAC
C1200  FORMAT(5X,'FINFAC=',E11.4,'    RFAC=',E11.4,//)
      HA = FINFAC * GA * CPA * PRA**(-0.667) * XJ * RFAC
      XHA = FINFAC
      RETURN
      END
```

Listing for the PDAIR Subroutine

A listing for the PDAIR subroutine follows:

```
      FUNCTION PDAIR(MUNIT,FINTYP,DUCT,AFILTR,AHEATR,RACKS,QA,GA,REAIR,
     &               AAF,NT,ST,DEA,FP,FAR,DELTA,ATAMIN,RHOM,FCOILW,
     &               NFP,FPD,XAPX)
C
C     THIS SUBROUTINE CALCULATES THE AIR PRESSURE DROP THROUGH A
C          FIN AND TUBE HEAT EXCHANGER WITH ADDITIONAL LOSSES DUE TO
C          CABINET, FILTER, SUPPLEMENTAL HEATERS, AND DUCTWORK
C
C     REFERENCES:
C
C          F.C. MCQUISTON, "CORRELATION OF HEAT,MASS, AND MOMENTUM
C          TRANSPORT COEFFICIENTS FOR PLATE-FIN-TUBE TRANSFER
C          SURFACES WITH STAGGERED TUBE", ASHRAE TRANSACTIONS,
C          VOL. 84, PART 1, 1978.
C
C          F.C. MCQUISTON, FINNED TUBE HEAT EXCHANGERS:
C          "STATE OF THE ART FOR THE AIR SIDE", ASHRAE TRANSACTIONS,
C          VOL. 87, PART 1, 1981.
C
C          T. HOSODA, H. UZUHASHI, N. KOBAYASHI,
C          "LOUVER FIN HEAT EXCHANGERS", HEAT TRANSFER JAPANESE RESEARCH,
C          VOL. 6, NO. 2, 1977, PP. 67-77.
C
C          SCIENCE APPLICATIONS, INC., ENERGY EFFICIENCY PROGRAM
C          FOR ROOM AIR CONDITIONERS, CENTRAL AIR CONDITIONERS,
C          DEHUMIDIFIERS, AND HEAT PUMPS, SAI-77-858-LJ, P. B-224,
C          MARCH 1978.
C
```

```
C        H. S. KIRSCHBAUM AND  S. E. VEYO, EPRI EM-319,
C        "AN INVESTIGATION OF METHODS TO IMPROVE
C        HEAT PUMP PERFORMANCE AND RELIABILITY IN A NORTHERN
C        CLIMATE", APPENDIX B, PP. B1-8, B1-21, JANUARY 1977.
C
C        GENERAL ELECTRIC PRODUCT DATA, PUB. NO. 22-1009-6,
C        "SPLIT SYSTEM WEATHERTRON HEAT PUMPS", P. 7, APRIL 1977.
C
C   INPUT:
C
C     MUNIT - INTEGER FLAG TO INDICATE IF UNIT IS INDOOR
C              = 1 TO INDICATE THE OUTDOOR UNIT AND
C              = 2 FOR THE INDOOR UNIT
C     FINTYP- TYPE OF PLATE FIN SURFACE
C              = 1 -- SMOOTH FIN
C              = 2 -- WAVY FIN
C              = 3 -- LOUVERED FIN
C     DUCT  = DIAMETER OF 1 OF 6 DUCTS EACH HAVING AN
C              EQUIVALENT LENGTH OF 100 FT. (FT)
C     AFILTR- FILTER AREA (FT**2)
C     AHEATR- CROSS-SECTIONAL AREA OF RESISTANCE HEATER
C              SECTION (USUALLY EQUAL TO BLOWER EXIT AREA) (FT**2)
C     RACKS - NUMBER OF RESISTANCE HEATER RACKS
C     QA    - AIR FLOW RATE (FT**3/MIN)
C     GA    - MASS FLUX OF AIR BASED ON MINIMUM FREE-FLOW AREA
C              (LBM/HR/FT**2)
C     REAIR - AIR-SIDE REYNOLDS NUMBER BASED ON OUTSIDE TUBE DIAMETER
C              AND MINIMUM FREE-FLOW AREA
C     AAF   - FRONTAL AREA OF HEAT EXCHANGER (FT**2)
C     NT    - NUMBER OF TUBE ROWS IN THE FLOW DIRECTION
C     ST    - TUBE VERTICAL SPACING (FT)
C     DEA   - REFRIGERANT TUBE OUTSIDE DIAMETER (FT)
C     FP    - FIN PITCH (1/FT)
C     FAR   - RATIO OF FIN HEAT TRANSFER AREA TO TOTAL HEAT TRANSFER
```

```
C              AREA
C      DELTA - FIN THICKNESS (FT)
C      ATAMIN- RATIO OF TOTAL AIR-SIDE AREA TO MINIMUM FREE-FLOW AREA
C      RHOM  - MEAN AIR DENSITY (LBM/FT**3)
C      FCOILW- FRACTION OF HEAT EXCHANGER OUTSIDE AREA THAT IS WET
C       -- THE FOLLOWING PARAMETERS ARE MEANINGFUL ONLY IF FINTYP = 4 --
C      NFP   - THE NUMBER OF FIN PATTERNS PER TUBE ROW
C      FPD   - THE FIN PATTERN DEPTH PEAK TO VALLEY (IN)
C
C    OUTPUT:
C
C      PDAIR - TOTAL AIR PRESSURE DROP THROUGH HEAT EXCHANGER
C              WITH ADDITIONAL LOSSES DUE TO CABINET, FILTER,
C              SUPPLEMENTAL HEATERS, AND DUCTWORK (PSIA)
C      -- THE FOLLOWING PARAMETER IS MEANINGFULL ONLY IF FINTYP = 4 --
C      XAPX  - PRESSURE DROP MULTIPLIER FOR FIN CORRUGATIONS
C
C
       REAL NT
       DPDUCT = 0.0
       DPFLHT = 0.0
       XAPX = 1.0
       VCF = QA/AAF
C
C      CABINET LOSS
C
       CMULT = 1.1
       IF(MUNIT.EQ.1) GO TO 1
C
C      INDOOR DUCTWORK PRESSURE LOSSES
C
       DPDUCT = 7.352E-10 * (QA)**1.84 / (DUCT)**5
C
C      FILTER AND SUPPLEMENTAL HEATER PRESSURE LOSSES
```

```
C
      DPFLTR = 6.75E-07 * .03613 * (QA/AFILTR)**2
      RACFAC = 1.0
      IF(RACKS.EQ.3.0) RACFAC =2.0
      IF(RACKS.EQ.4.0) RACFAC =2.4
      DPHTRS = 1.027E-07 * .03613 * (QA/AHEATR)**2 * RACFAC
      DPFLHT = DPFLTR + DPHTRS
    1 CONTINUE
C
C     HEAT EXCHANGER PRESSURE DROP (DRY AND/OR WET COILS)
C
      IF(FINTYP .GT. 1.0 .AND. FINTYP .LT. 4.0) GO TO 4
C
C     COIL PRESSURE DROP FOR SMOOTH FINS
C
C      CORRECTION FACTOR FOR FIN CORRUGATIONS
C
      IF(FINTYP .EQ. 4.0) XAPX = APDCOR(FPD,NFP,VCF)
C
      R = DEA / 2.
      FS = 1./(1.-FP*DELTA)
      ARATIO = 1./(1.-FAR)
      RSTAR = ARATIO * R / ((ST-DEA)*FP + 1.)
      RRSTAR = R / RSTAR
      F1 = (ST-DEA) * FP * FS / 4.
      F2 = ST / (2.*RSTAR)  -  1.
      FPF = (RRSTAR/REAIR)**.25 * F1**(-.4) / SQRT(F2)
      IF(.08 .LE. FPF .AND. FPF .LE. .24) GO TO 2
      WRITE(6,1000) FPF
 1000 FORMAT('   PDAIR: FRICTION NO. FPF =',G10.4,' IS OUT OF RANGE.')
    2 CONTINUE
C
C     FOR WET COIL, ASSUME FILM TYPE CONDENSATION
C
```

```
      FWET = 0.0
      IF(FCOILW.EQ.0.0) GO TO 3
      FSF = ( 1. + REAIR**(-.4) ) * FS**1.5
      FWET = 4.904E-03 + 1.382*(FPF*FSF)**2
    3 CONTINUE
      FDRY = 4.904E-03 + 1.382*FPF**2
      F = ( FCOILW*FWET + (1.-FCOILW)*FDRY )
      PDAIR = CMULT * GA**2 * F*ATAMIN/RHOM  /(2.*32.174*144.*3600.**2)
      PDAIR = XAPX * PDAIR
      GO TO 5
C
C     COIL PRESSURE DROP FOR WAVY AND LOUVERED FINS
C
    4 CONTINUE
      FAC1 = 3.84E-06 * (NT/2.)**0.7
      FAC2 = .235 + .0638*FP/12.
      FAC3 = (QA/AAF)**1.7
      PDAIR = CMULT*FAC1*FAC2*FAC3*3.613E-02
      IF (FINTYP.EQ.3.0) PDAIR = PDAIR * 1.1
      PDAIR = (RHOM/0.075) * PDAIR
      IF (FCOILW.EQ.0.0) GO TO 5
      FGAPMM = (1/FP - DELTA) * 12. * 25.4
      WFAC = 1.2 + 1.359*(FGAPMM)**(-.5786)
      PDAIR = (FCOILW*WFAC + (1.-FCOILW)) * PDAIR
    5 CONTINUE
      PDAIR = PDAIR + DPDUCT + DPFLHT
      RETURN
      END
```

Listing of TRIAL2 Subroutine

A listing of the TRIAL2 subroutine follows:

```
      SUBROUTINE TRIAL2(TI,DTI,P,N,ARG,TOL,V,H,S,T,IERROR)
C
C      PURPOSE
C         TO DETERMINE REMAINING SUPERHEATED VAPOR PROPERTIES,
C         GIVEN THE PRESSURE AND ONE OTHER PROPERTY OF
C         A SPECIFIED REFRIGERANT
C
C***   AUTHORS  G.T. KARTSOUNES AND R.A. ERTH              ***
C
C     ##################################
C     ITERATION TECHNIQUE MODIFIED BY
C     WESTINGHOUSE RESEARCH LABORATORIES
C     ON APRIL 19, 1985 BY T. J. FAGAN
C     ##################################
C
C      DESCRIPTION OF PARAMETERS
C        INPUT
C          TI    -  INITIAL TEMPERATURE GUESS   (F)
C          DTI   -  INITIAL STEP SIZE FOR TEMPERATURE ITERATION (F)
C          P     -  PRESSURE (PSIA)
C          N     -  ARGUMENT INDICATOR
C                   IF N = 2, THE SECOND KNOWN PROPERTY IS SPECIFIC VOLUME
C                   IF N = 3, THE SECOND KNOWN PROPERTY IS ENTHALPY
C                   IF N = 4, THE SECOND KNOWN PROPERTY IS ENTROPY
C          ARG   -  THE SECOND KNOWN PROPERTY
C          TOL   -  CONVERGENCE TOLERANCE
C        OUTPUT
C          V     -  SPECIFIC VOLUME OF VAPOR (CU FT/LBM)
```

```
C        H     -  ENTHALPY OF VAPOR (BTU/LBM)
C        S     -  ENTROPY OF VAPOR (BTU/LBM-R)
C        T     -  TEMPERATURE OF VAPOR (F)
C        IERROR- ERROR FLAG
C     REMARKS - SUBROUTINES CALLED
C       VAPOR   TO DETERMINE THE DESIRED REFRIGERANT PROPERTIES
C
      T = TI
      DT = DTI
      TLOW = TSAT(P,IERROR)
      IF (IERROR .NE. 0) WRITE(6,102) P,IERROR
      DO 20 I=1,200
      IF (T .LT. TLOW) T = TLOW
      CALL VAPOR(T,P,VVAP,HVAP,SVAP,IERROR)
      IF (IERROR .NE. 0) WRITE(6,101)
      IF(N.EQ.2)ARGN = VVAP
      IF(N.EQ.3) ARGN = HVAP
      IF(N.EQ.4) ARGN = SVAP
      IF((N.NE.2).AND.(N.NE.3).AND.(N.NE.4)) GO TO  25
      DIFF = ARG - ARGN
C
C     NEW SECTION ADDED BY WESTINGHOUSE APRIL 19, 1985
C
      IF(DT.GT.0.0) DIFF = ARGN - ARG
      IF(ABS(DIFF).LE.TOL) GO TO 30
      IF(DIFF .LT. 0.0) ITEST = -1
      IF(DIFF .GT. 0.0) ITEST =  1
      IF(ITEST .NE. IREFX .AND. I .GT. 1) DT = DT/4.0
      IF(ITEST .EQ. -1) T = T + DT
      IF(ITEST .EQ.  1) T = T - DT
      IREFX = ITEST
C
C      END OF APRIL 19,1985 SECTION
C
```

```
 20 CONTINUE
 25 WRITE(6,100) N
    IERROR = 1
 30 V = VVAP
    H = HVAP
    S = SVAP
    RETURN
100 FORMAT(' *****TRIAL2 DOES NOT CONVERGE N=',I2,' ******')
101 FORMAT('0 TRIAL2: **** ERROR WAS MADE IN CALL TO "VAPOR" ***'/)
102 FORMAT('0 TRIAL2: ***** ERROR IN CALL TO "TSAT" - PRESSURE = ',
   &       1PE10.3,' PSIA,     ERROR FLAG = ',I2,' *****',/)
    END
```

APPENDIX H
Sample Data Files

Sample Heating Data

INPUT DATA FILES FOR HIGH CAPACITY HEATING RUNS - RANCO #26 VALVE

HIGH CAPACITY HEAT AT 47 DEG F TEST 103

```
        1
        2
        0    0.6000    0.0000
        0   13.1000    0.0000    0.0000    0.0000    0.0000
  30.0000  115.0000
        2    5.0500 3500.0000    0.0000    0.1700
-6.090E-5  1.814E-2 -1.840E-4 -4.632E-2  7.530E-4  1.416E+0  5.050E+0
 2.000E+1
-9.546E-3  2.479E-1  1.129E-1  7.820E+0 -1.214E-2  3.306E+2
  70.0000    0.5500
1460.0000    0.4601    8.0000   42000.0
   5.5000    4.0000    4.5000    0.8660    1.0000  104.0000
   2.0000   13.0000    0.0045    0.3960    0.3620  128.3000  225.0000
   4.0000
  47.0000    0.6000
3070.8000    0.3250         0
  14.7800    3.0000    8.0000    0.8660    1.0000  168.0000
   2.0000   13.0000    0.0045    0.3960    0.3620  128.3000  125.0000
   4.0000
        0         2         2
   0.0000    0.0000    0.0000
   0.4300   40.0000    0.6800    3.0000    0.8050   25.0000
   0.6800    7.0000    0.4300   10.0000
       26   47.0000
  1      0.0200
  1      0.0358      0.0417      37.5000      0.0671      3.0000
```

HIGH CAPACITY HEAT AT 35 DEG F TEST 109A

```
        1
        2
        0    0.0000    0.0000
        0    4.8000    0.0000    0.0000    0.0000    0.0000
  20.0000  103.0000
        2    5.0500 3500.0000    0.0000    0.2000
-6.090E-5  1.814E-2 -1.840E-4 -4.632E-2  7.530E-4  1.416E+0  5.050E+0
 2.000E+1
-9.546E-3  2.479E-1  1.129E-1  7.820E+0 -1.214E-2  3.306E+2
  69.8000    0.5500
1465.3000    0.4601    8.0000   42000.0
   5.5000    4.0000    4.5000    0.8660    1.0000  104.0000
```

```
   2.0000   13.0000     0.0045     0.3960     0.3620  128.3000  225.0000
   4.0000
  35.0000    0.7000
3042.6000    0.3250          0
  14.7800    3.0000     8.0000     0.8660     1.0000  168.0000
   2.0000   13.0000     0.0045     0.3960     0.3620  128.3000  125.0000
   4.0000
        0         2          2
   0.0000    0.0000     0.0000
   0.4300   40.0000     0.6800     3.0000     0.8050   25.0000
   0.6800    7.0000     0.4300    10.0000
       26   35.0000
  1      0.0200
  1      0.0358      0.0417     32.5000      0.0671      3.0000
```

HIGH CAPACITY HEAT AT 17 DEG F TEST 106

```
        1
        2
        0    3.1200     0.0000
        0    6.8900     0.0000     0.0000     0.0000     0.0000
   5.0000   95.0000
        2    5.0500  3500.0000     0.0000     0.2900
-6.090E-5  1.814E-2 -1.840E-4 -4.632E-2  7.530E-4  1.416E+0  5.050E+0
 2.000E+1
-9.546E-3  2.479E-1  1.129E-1  7.820E+0 -1.214E-2  3.306E+2
  69.6000    0.5500
1472.6000    0.4601     8.0000    42000.0
   5.5000    4.0000     4.5000     0.8660     1.0000  104.0000
   2.0000   13.0000     0.0045     0.3960     0.3620  128.3000  225.0000
   4.0000
  16.7000    0.6700
3032.2000    0.3250          0
  14.7800    3.0000     8.0000     0.8660     1.0000  168.0000
   2.0000   13.0000     0.0045     0.3960     0.3620  128.3000  125.0000
   4.0000
        0         2          2
   0.0000    0.0000     0.0000
   0.4300   40.0000     0.6800     3.0000     0.8050   25.0000
   0.6800    7.0000     0.4300    10.0000
       26   16.0000
  1      0.0200
  1      0.0358      0.0417     45.0000      0.0671      3.0000
```

HIGH CAPACITY HEAT AT 0 DEG F TEST 110 FIXED REFRIGERANT FLOW

```
        1
        2
        0    3.0800     0.0000
```

0	3.9400	0.0000	0.0000	0.0000	0.0000	
-7.0000	88.0000					
2	5.0500	3500.0000	0.0000	0.4900		
0.000E+0	0.000E+0	0.000E+0	0.000E+0	0.000E+0	2.284E+0	5.050E+0
3.080E+0						
0.000E+0	0.000E+0	0.000E+0	0.000E+0	0.000E+0	1.578E+2	
69.7000	0.5400					
1481.7000	0.4601	8.0000	42000.0			
5.5000	4.0000	4.5000	0.8660	1.0000	104.0000	
2.0000	13.0000	0.0045	0.3960	0.3620	128.3000	225.0000
4.0000						
-0.6000	0.9900					
2862.0000	0.3250	0				
14.7800	3.0000	8.0000	0.8660	1.0000	168.0000	
2.0000	13.0000	0.0045	0.3960	0.3620	128.3000	125.0000
4.0000						
0	2	2				
0.0000	0.0000	0.0000				
0.4300	40.0000	0.6800	3.0000	0.8050	25.0000	
0.6800	7.0000	0.4300	10.0000			
26	-0.6000					
1	0.0200					
1	0.0358	0.0417	30.0000	0.0671	3.0000	

INPUT DATA FILES FOR LOW CAPACITY HEATING RUNS - RANCO #26 VALVE

LOW CAPACITY HEAT AT 47 DEG F TEST 107

```
        1
        2
        0    1.4900    0.0000
        0    2.0000    0.0000    0.0000    0.0000    0.0000
  37.0000  100.0000
        2    3.8800 3500.0000    0.0000    0.1300
-2.225E-4  8.501E-3 -2.349E-4 -6.026E-2  1.008E-3  2.136E+0  3.880E+0
 2.000E+1
-1.991E-2  1.490E+0  1.181E-1  8.221E+0 -4.998E-2  1.130E+2
  69.6000    0.5600
1025.2000    0.4601    8.0000   42000.0
   5.5000    4.0000    4.5000    0.8660    1.0000  104.0000
   2.0000   13.0000    0.0045    0.3960    0.3620  128.3000  225.0000
   4.0000
  47.1000    0.5800
3058.2000    0.3250         0
  14.7800    3.0000    8.0000    0.8660    1.0000  168.0000
   2.0000   13.0000    0.0045    0.3960    0.3620  128.3000  125.0000
   4.0000
        0         2         2
   0.0000    0.0000    0.0000
   0.4300   40.0000    0.6800    3.0000    0.8050   25.0000
   0.6800    7.0000    0.4300   10.0000
       26   47.0000
  1      0.0200
  1      0.0358      0.0417     30.0000      0.0671      3.0000
```

LOW CAPACITY HEAT AT 35 DEG F TEST 109A

```
        1
        2
        0    2.6900    0.0000
        0    4.8600    0.0000    0.0000    0.0000    0.0000
  26.0000   95.0000
        2    3.8800 3500.0000    0.0000    0.3750
-2.225E-4  8.501E-3 -2.349E-4 -6.026E-2  1.008E-3  2.136E+0  3.880E+0
 2.000E+1
-1.991E-2  1.490E+0  1.181E-1  8.221E+0 -4.998E-2  1.130E+2
  69.6000    0.5500
1022.6000    0.4601    8.0000   42000.0
   5.5000    4.0000    4.5000    0.8660    1.0000  104.0000
   2.0000   13.0000    0.0045    0.3960    0.3620  128.3000  225.0000
   4.0000
  34.9000    0.7000
3058.2000    0.3250         0
```

14.7800	3.0000	8.0000	0.8660	1.0000	168.0000	
2.0000	13.0000	0.0045	0.3960	0.3620	128.3000	125.0000
4.0000						
0	2	2				
0.0000	0.0000	0.0000				
0.4300	40.0000	0.6800	3.0000	0.8050	25.0000	
0.6800	7.0000	0.4300	10.0000			
26	34.9000					

2	0.0200				
1	0.0358	0.0417	32.5000	0.0671	3.0000

Sample Cooling Input Data

INPUT DATA FILES FOR HIGH CAPACITY COOLING - RANCO #26 VALVE

```
HIGH CAPACITY COOL AT 82 DEG F TEST 114
        1
        1
        0     0.5300     0.0000
        0     5.3000     0.0000     0.0000     0.0000     0.0000
  44.0000   110.0000
        2     5.0500  3500.0000     0.0000     0.3000
-6.090E-5  1.814E-2  -1.840E-4  -4.632E-2   7.530E-4   1.416E+0   5.050E+0
 2.000E+1
-9.546E-3  2.479E-1   1.129E-1   7.820E+0  -1.214E-2   3.306E+2
  79.1000     0.5500
1450.4000     0.4601     8.0000    42000.0
   5.5000     4.0000     4.0000     0.8660     1.0000   104.0000
   2.0000    13.0000     0.0045     0.3960     0.3620   128.3000   225.0000
   4.0000
  82.5000     0.4900
3029.0000     0.3250          0
  14.7800     3.0000     5.8000     0.8660     1.0000   168.0000
   2.0000    13.0000     0.0045     0.3960     0.3620   128.3000   225.0000
   4.0000
        0          2          2
   0.0000     0.0000     0.0000
   0.4300    40.0000     0.6800     3.0000     0.8050    25.0000
   0.6800     7.0000     0.4300    10.0000
       26    62.0000
  1      0.0200
  0      0.0000


HIGH CAPACITY COOL AT 95 DEG F TEST 111
        1
        1
        0     1.0600     0.0000
        0     4.3900     0.0000     0.0000     0.0000     0.0000
  44.0000   125.0000
        2     5.0500  3500.0000     0.0000     0.2700
-6.090E-5  1.814E-2  -1.840E-4  -4.632E-2   7.530E-4   1.416E+0   5.050E+0
 2.000E+1
-9.546E-3  2.479E-1   1.129E-1   7.820E+0  -1.214E-2   3.306E+2
  79.0000     0.5500
1431.2000     0.4601     8.0000    42000.0
   5.5000     4.0000     4.0000     0.8660     1.0000   104.0000
   2.0000    13.0000     0.0045     0.3960     0.3620   128.3000   225.0000
   4.0000
```

```
   95.0000    0.4800
 2954.0000    0.3250          0
   14.7800    3.0000     5.8000     0.8660     1.0000   168.0000
    2.0000   13.0000     0.0045     0.3960     0.3620   128.3000   225.0000
    4.0000
         0         2          2
    0.0000    0.0000     0.0000
    0.4300   40.0000     0.6800     3.0000     0.8050    25.0000
    0.6800    7.0000     0.4300    10.0000
        26   62.0000
  1       0.0200
  0       0.0000


HIGH CAPACITY COOL AT 106 DEG F TEST 117
         1
         1
         0    6.6200     0.0000
         0   17.1000     0.0000     0.0000     0.0000     0.0000
   50.0000  135.0000
         2    5.0500  3500.0000     0.0000     0.0580
 -6.090E-5  1.814E-2  -1.840E-4  -4.632E-2   7.530E-4   1.416E+0   5.050E+0
  2.000E+1
 -9.546E-3  2.479E-1   1.129E-1   7.820E+0  -1.214E-2   3.306E+2
   79.1000    0.5400
 1451.0000    0.4601     8.0000    42000.0
    5.5000    4.0000     4.0000     0.8660     1.0000   104.0000
    2.0000   13.0000     0.0045     0.3960     0.3620   128.3000   225.0000
    4.0000
  105.8000    0.4800
 2541.6000    0.3250          0
   14.7800    3.0000     5.8000     0.8660     1.0000   168.0000
    2.0000   13.0000     0.0045     0.3960     0.3620   128.3000   225.0000
    4.0000
         0         2          2
    0.0000    0.0000     0.0000
    0.4300   40.0000     0.6800     3.0000     0.8050    25.0000
    0.6800    7.0000     0.4300    10.0000
        26   64.0000
  1       0.0200
  0       0.0000
```

INPUT DATA FILES FOR LOW CAPACITY COOLING RUNS - RANCO #26 VALVE

```
LOW CAPACITY COOL AT 76 DEG F TEST 116
        1
        1
        0    1.2200    0.0000
        0    4.0700    0.0000    0.0000    0.0000    0.0000
  45.0000   91.0000
        2    3.8800 3500.0000    0.0000    0.4500
-2.225E-4  8.501E-3 -2.349E-4 -6.026E-2  1.008E-3  2.136E+0  3.880E+0
 2.000E+1
-1.991E-2  1.490E+0  1.181E-1  8.221E+0 -4.998E-2  1.130E+2
  79.1000    0.5500
 974.4000    0.4601    8.0000   42000.0
   3.6670    4.0000    2.9000    0.8660    1.0000   69.0000
   2.0000   13.0000    0.0045    0.3960    0.3620  128.3000  225.0000
   4.0000
  75.5000    0.3200
3023.3000    0.3250         0
  14.7800    3.0000    5.8000    0.8660    1.0000  168.0000
   2.0000   13.0000    0.0045    0.3960    0.3620  128.3000  225.0000
   4.0000
        0         2         2
   0.0000    0.0000    0.0000
   0.4300   40.0000    0.6800    3.0000    0.8050   25.0000
   0.6800    7.0000    0.4300   10.0000
       26   68.0000
  1      0.0200
  0      0.0000
```

```
LOW CAPACITY COOL AT 82 DEG F TEST 115
        1
        1
        0    0.8100    0.0000
        0    1.9000    0.0000    0.0000    0.0000    0.0000
  47.0000   97.0000
        2    3.8800 3500.0000    0.0000    0.4500
-2.225E-4  8.501E-3 -2.349E-4 -6.026E-2  1.008E-3  2.136E+0  3.880E+0
 2.000E+1
-1.991E-2  1.490E+0  1.181E-1  8.221E+0 -4.998E-2  1.130E+2
  79.0000    0.5500
 972.3000    0.4601    8.0000   42000.0
   3.6670    4.0000    2.9000    0.8660    1.0000   69.0000
   2.0000   13.0000    0.0045    0.3960    0.3620  128.3000  225.0000
   4.0000
  82.5000    0.4900
3028.2000    0.3250         0
  14.7800    3.0000    5.8000    0.8660    1.0000  168.0000
```

```
   2.0000   13.0000    0.0045    0.3960    0.3620  128.3000  225.0000
   4.0000
        0         2         2
   0.0000    0.0000    0.0000
   0.4300   40.0000    0.6800    3.0000    0.8050   25.0000
   0.6800    7.0000    0.4300   10.0000
       26   69.0000
  1       0.0200
  0       0.0000


LOW CAPACITY COOL AT 95 DEG F TEST 113
        1
        1
        0    1.5200    0.0000
        0    3.6800    0.0000    0.0000    0.0000    0.0000
  46.0000  110.0000
        2    3.8800 3500.0000    0.0000    0.4000
-2.225E-4  8.501E-3 -2.349E-4 -6.026E-2  1.008E-3  2.136E+0  3.880E+0
 2.000E+1
-1.991E-2  1.490E+0  1.181E-1  8.221E+0 -4.998E-2  1.130E+2
  79.0000    0.5400
 919.2000    0.4601    8.0000   42000.0
   3.6670    4.0000    2.9000    0.8660    1.0000   69.0000
   2.0000   13.0000    0.0045    0.3960    0.3620  128.3000  225.0000
   4.0000
  95.7000    0.4800
2962.0000    0.3250         0
  14.7800    3.0000    5.8000    0.8660    1.0000  168.0000
   2.0000   13.0000    0.0045    0.3960    0.3620  128.3000  225.0000
   4.0000
        0         2         2
   0.0000    0.0000    0.0000
   0.4300   40.0000    0.6800    3.0000    0.8050   25.0000
   0.6800    7.0000    0.4300   10.0000
       26   69.0000
  1       0.0200
  0       0.0000
```