Energy Laboratory
in association with
Heat Transfer Laboratory,
Department of Mechanical Engineering

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

IMPROVING HEAT PUMP PERFORMANCE VIA COMPRESSOR CAPACITY CONTROL - ANALYSIS AND TEST,

Volume II: Appendices

by
Carl C. Hiller and Leon R. Glicksman

Energy Laboratory Report No. MIT-EL 76-002
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APPENDIX A

THERMOPHYSICAL PROPERTIES OF REFRIGERANTS

Presented here are curve fits and computer programs for producing thermophysical properties of refrigerants 12, 22, and 502 from basic equations. Program listings are given at the end of this section.

Plots of viscosity, thermal conductivity, and specific heat for refrigerants 12 and 22 are given in Figures A-1 through A-6 [1,2]. Curve fits for the above properties in both the liquid and vapor states are indicated on the figures.

The following thermodynamic properties subroutines have been modified from Kartsounes & Erth, 'Computer Calculation of the Thermodynamic Properties of Refrigerants 12, 22 and 502'[3]. The programs produce values of enthalpy, entropy, specific volume, specific heat, sonic velocity, pressure, and temperature. The Kartsounes & Erth programs have been checked and were found to be highly accurate, with the exception of a possible convergence problem near the critical point.

Subroutine TABLES

TABLES is a simple program which, when called upon by the other thermodynamic properties programs, delivers (into common) the constants necessary to calculate thermodynamic properties from basic equations. See comments in the listing for details.

Subroutine TSAT

TSAT is a program which produces the saturation temperature of a refrigerant corresponding to a given saturation pressure. See comments in the listing for details.

Subroutine SPVOL

SPVOL is a program which calculates specific volume of the vapor phase, given temperature and pressure. See comments in the listing for details.

Subroutine SATPRP

SATPRP is a program which, given saturation temperature, determines the corresponding saturation properties:

PSAT - saturation pressure

VF - specific volume of saturated liquid

VG - specific volume of saturated vapor

HF - enthalpy of saturated liquid

HFG - latent enthalpy of vaporization

HG - enthalpy of saturated vapor

SF - entropy of saturated lquid

SG - entropy of saturated vapor

See comments in the listing for more details.

Subroutine VAPOR

VAPOR is a program which determines specific volume, enthalpy, and entropy of superheated vapor, given the temperature and pressure. See comments in the listing for details.

Subroutine SPFHT

SPFHT is a program which, given temperature and pressure, determines the following:

CV - specific heat at constant volume

CP - specific heat at constant pressure

GAMMA - specific heat ratio CP/CV

SONIC - sonic velocity

Subroutine TRIAL

TRIAL is a program which, given pressure and one other property (temperature, specific volume, enthalpy or entropy), determines the remaining superheated vapor properties. See comments in the listing for more details.

REFERENCES

1. ASHRAE HANDBOOK OF FUNDAMENTALS (New York: American Soc. of Heat., Refg., & Air-Cond.Eng., Inc., 1972).

2. THERMOPHYSICAL PROPERTIES OF REFRIGERANTS (New York: American Soc. of Heat, Refg. & Air-Cond. Eng., Inc., 1973).

3. Kartsounes, G. T., and Erth, R. A., "Computer Calculation of the Thermodynamic Properties of Refrigerants 12, 22, and 502", ASHRAE Transactions, Vol. 77, Part II, 1971.

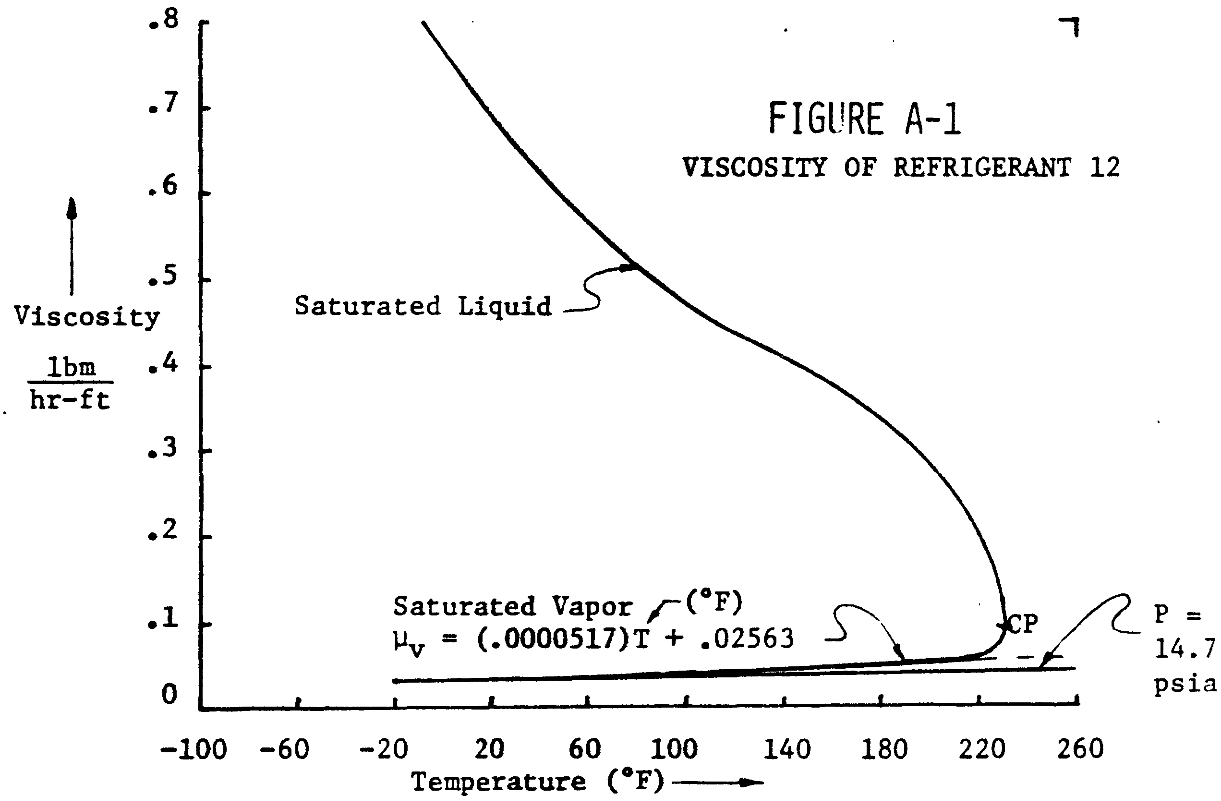


FIGURE A-1

VISCOSITY OF REFRIGERANT 12

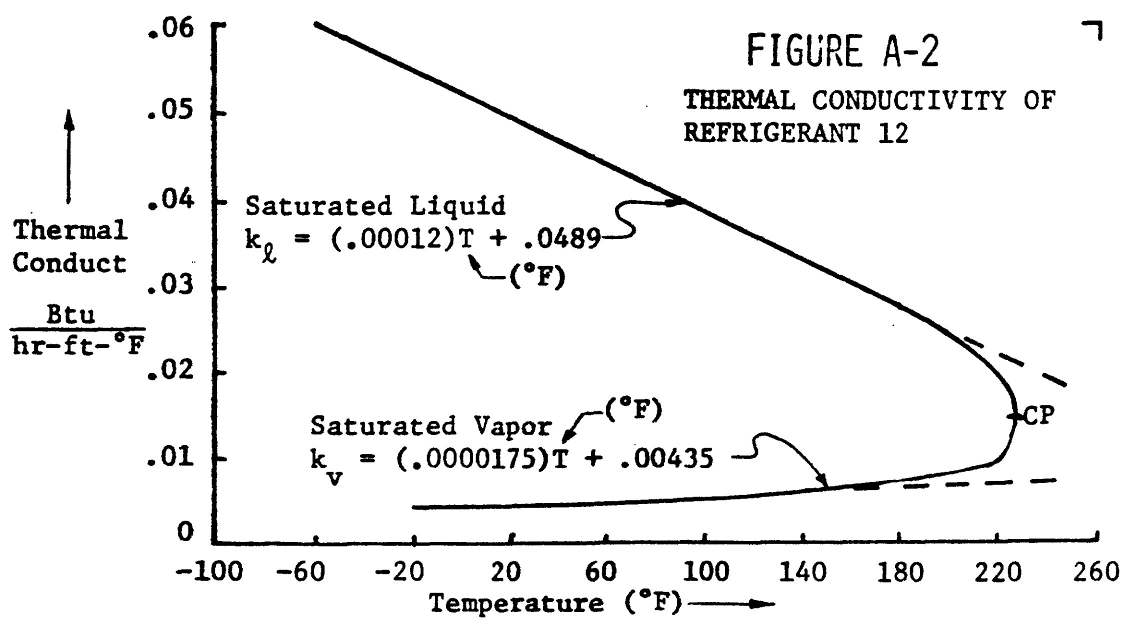


FIGURE A-2

THERMAL CONDUCTIVITY OF REFRIGERANT 12

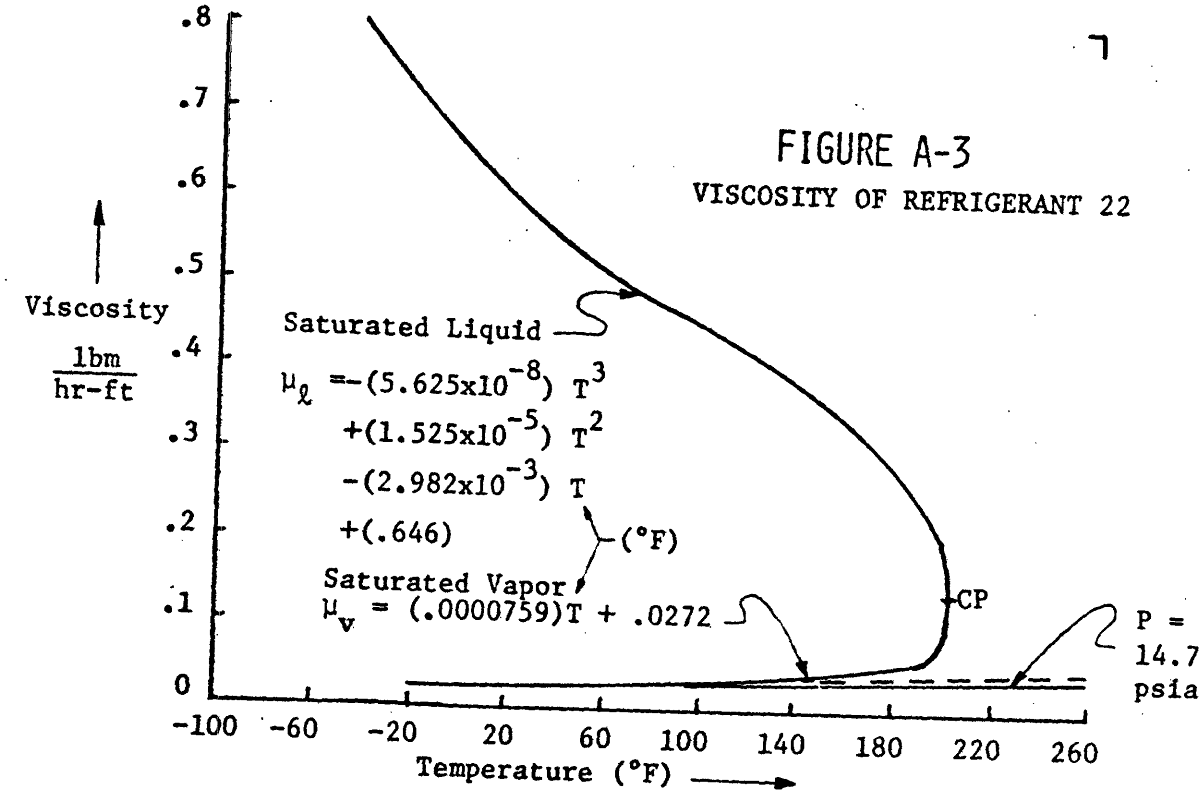


FIGURE A-3

VISCOSITY OF REFRIGERANT 22

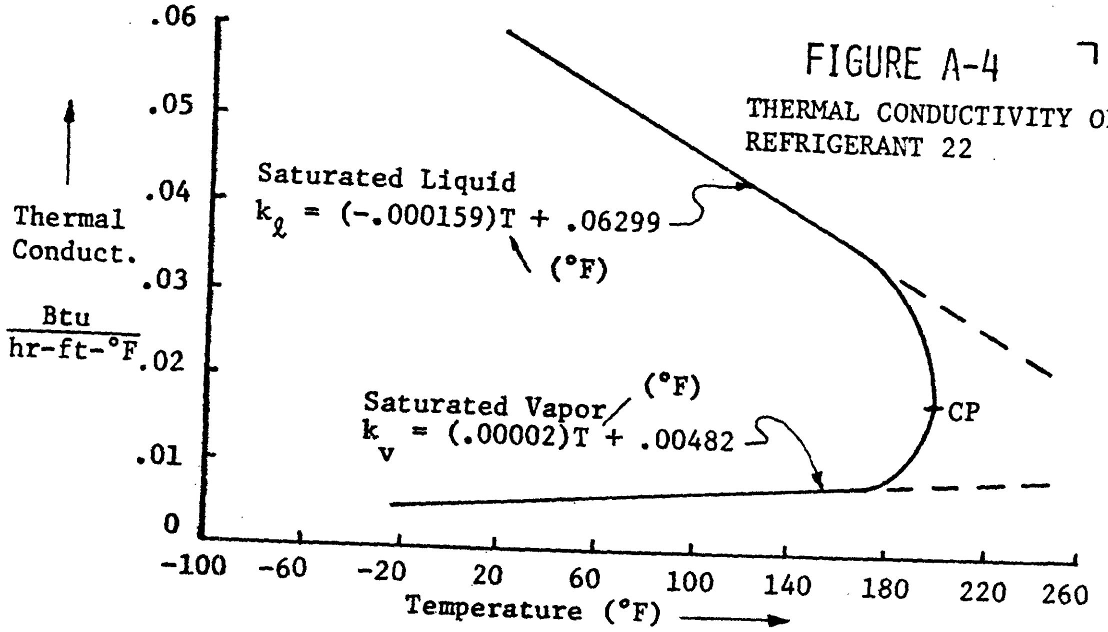
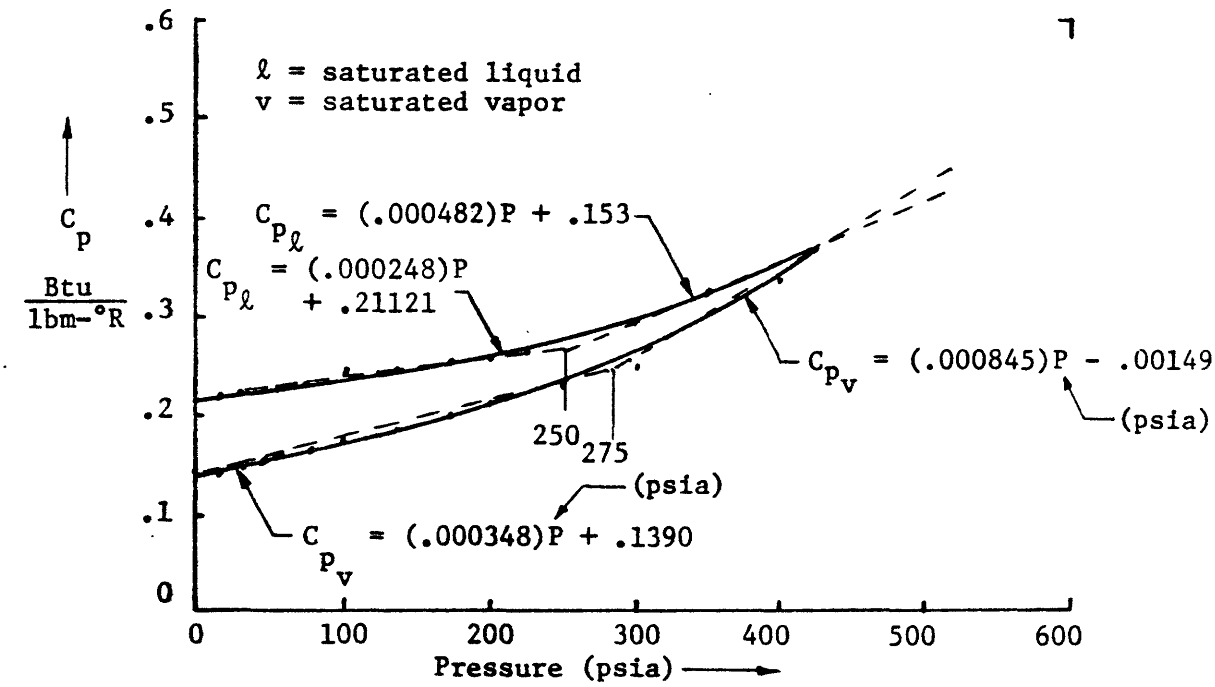


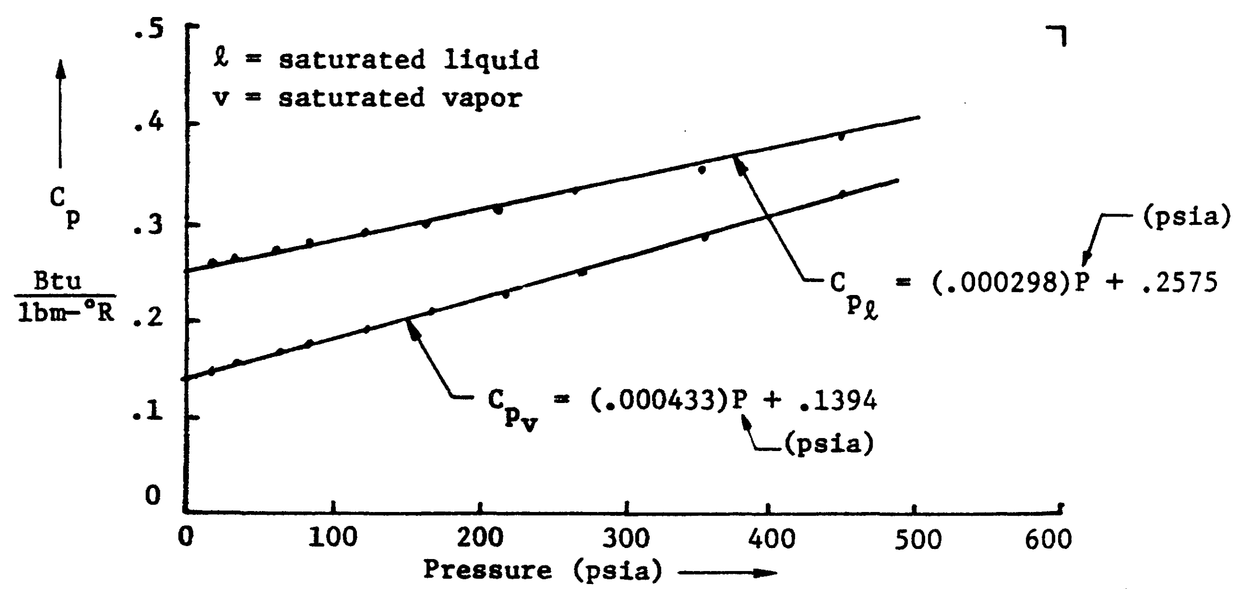
FIGURE A-4

THERMAL CONDUCTIVITY OF REFRIGERANT 22



SPECIFIC HEAT AT CONSTANT PRESSURE OF REFRIGERANT 12

FIGURE A-5



SPECIFIC HEAT AT CONSTANT PRESSURE OF REFRIGERANT 22

FIGURE A-6

```
      SUBROUTINE TABLES(NR,I)
C
C     PURPOSE
C        TO PROVIDE CORRECT VALUES FOR CONSTANTS IN THE
C        THERMODYNAMIC PROPERTIES SUBPROGRAMS, CORRESPONDING
C        TO THE DESIRED REFRIGERANT (12,22, OR 502)
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        NR    -  REFRIGERANT NUMBER (12,22, OR 502)
C      OUTPUT
C        ALL OF THE CONSTANTS HELD IN COMMON BLOCKS
C        THE REFRIGERANT INDICATOR 'I'
      REAL K,LE10,L10E,J
C
C     DESCRIPTION OF CONSTANTS
C
C        VAPOR PRESSURE CONSTANTS
      COMMON/SAT/AVP,BVP,CVP,DVP,EVP,FVP
C
C        CRITICAL POINT PROPERTIES  TC,  PC,  VC
C        INITIAL APPROXIMATION CONSTANTS  A,  B
C        MISCELLANEOUS CONSTANTS  TFR,  LE10
      COMMON/SUPER/ TC,PC,VC,A,B,TFR,LE10
C
C        EQUATION OF STATE CONSTANTS
      COMMON/STATES/R,B1,A2,B2,C2,A3,B3,C3,A4,B4,C4,A5,B5,
     1C5,A6,B6,C6,K,ALPHA,CPR
C
C        SPECIFIC HEAT AT CONSTANT VOLUME CONSTANTS
C        ACV,BCV,CCV,DCV,ECV,FCV
C        ENTHALPY AND ENTROPY OF VAPOR CONSTANTS  X,  Y
C        MISCELLANEOUS CONSTANTS  L10E,  J
      COMMON/OTHER/ACV,BCV,CCV,DCV,ECV,FCV,X,Y,L10E,J
      IWRITE = 5
C
C     SET REFRIGERANT INDICATOR 'I'
      I = 0
      IF(NR.EQ.12) I=1
      IF(NR.EQ.22) I=2
      IF(NR.EQ.502) I=3
      IF(I.EQ.0) GO TO 999
      J = .185053
      L10E = .434294
      LE10 = 2.302585
      GO TO(10,20,30) , I
C CONSTANTS FOR REFRIGERANT 12
   10 AVP = 39.883817
      BVP = -3436.63228
```

```
      CVP = -12.471522
      DVP = 4.730442E-03
      EVP = 0.0
      FVP = 0.0
      TC = 693.3
      PC = 596.9
      VC = .02870
      A = 120.0
      B = 312.0
      TFR = 459.7
      R = .088734
      B1 = 6.509389E-03
      A2 = -3.409727
      B2 = 1.594348E-03
      C2 = -56.762767
      A3 = 6.023945E-02
      B3 = -1.879618E-05
      C3 = 1.311399
      A4 = -5.487370E-04
      B4 = 0.0
      C4 = 0.0
      A5 = 0.0
      B5 = 3.468834E-09
      C5 = -2.543907E-05
      A6 = 0.0
      B6 = 0.0
      C6 = 0.0
      K = 5.475
      ALPHA = 0.0
      CPR = 0.0
      ACV = 8.0945E-03
      BCV = 3.32662E-04
      CCV = -2.413896E-07
      DCV = 6.72363E-11
      ECV = 0.0
      FCV = 0.0
      X = 39.556551
      Y = -1.653794E-02
      RETURN
C CONSTANTS FOR REFRIGERANT 22
   20 AVP = 29.357545
      BVP = -3845.193152
      CVP = - 7.861031
      DVP = 2.190939E-03
      EVP = .445747
      FVP = 686.1
      TC = 664.5
      PC = 721.906
      VC = .030525
      A = 120.0
```

```
      B = 388.0
      TFR = 459.69
      R = .124098
      B1 = .002
      A2 = -4.353547
      B2 = 2.407252E-03
      C2 = -44.066868
      A3 = -.017464
      B3 = 7.62789E-05
      C3 = 1.483763
      A4 = 2.310142E-03
      B4 = -3.025723E-06
      C4 = 0.0
      A5 = -3.724044E-05
      B5 = 5.355465E-08
      C5 = -1.845051E-04
      A6 = 1.363387E08
      B6 = -1.672612E05
      C6 = 0.0
      K = 4.2
      ALPHA = 548.2
      CPR = 0.0
      ACV = 2.812836E-02
      BCV = 2.255408E-04
      CCV = -6.509607E-08
      DCV = 0.0
      ECV = 0.0
      FCV = 257.341
      X = 62.4009
      Y = -4.53335E-05
      RETURN
C CONSTANTS FOR REFRIGERANT 502
   30 AVP = 10.644955
      BVP = -3671.153813
      CVP = -   .369835
      DVP = -1.746352E-03
      EVP = .816114
      FVP = 654.0
      TC = 639.56
      PC = 591.00
      VC = .028571
      A = 117.0
      B = 279.0
      TFR = 459.67
      R = .096125
      B1 = .00167
      A2 = -3.261334
      B2 = 2.057629E-03
      C2 = -24.24879
      A3 = 3.486675E-02
```

```
      B3 = -.867913E-05
      C3 = .332748
      A4 = -8.576568E-04
      B4 = 7.024055E-07
      C4 = 2.241237E-02
      A5 = 8.836897E-06
      B5 = -7.916809E-09
      C5 = -3.716723E-04
      A6 = -3.825373E07
      B6 = 5.581609E04
      C6 = 1.537838E09
      K = 4.2
      ALPHA = 609.0
      CPR = 7.E-07
      ACV = 2.0419E-02
      BCV = 2.996822E-04
      CCV = -1.409243E-07
      DCV = 2.210861E-11
      ECV = 0.0
      FCV = 64.058511
      X = 35.308
      Y = -.027444
      RETURN
C
C     PRINT ERROR MESSAGE IF
C     'NR' DOES NOT EQUAL 12, 22, OR 502
  999 WRITE(IWRITE,1000)
 1000 FORMAT('  ************ERROR IN SUBROUTINE -TABLES-')
      RETURN
      END
```

```
      FUNCTION TSAT(NR,PSAT)
C
C     PURPOSE
C        TO EVALUATE THE SATURATION TEMPERATURE
C        OF REFRIGERANT 12,22, OR 502
C        GIVEN THE SATURATION PRESSURE
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        NR    =  REFRIGERANT NUMBER (12,22, OF 502)
C        PSAT  =  SATURATION PRESSURE (PSIA)
C      OUTPUT
C        TSAT  =  SATURATION TEMPERATURE (F)
C     REMARKS
C        SUBROUTINE TABLES CALLED BY THIS FUNCTION
      REAL LE10
C
C     CONSTANTS
C
C        VAPOR PRESSURE CONSTANTS
      COMMON/SAT/AVP,BVP,CVP,DVP,EVP,FVP
C
C        CRITICAL POINT PROPERTIES   TC,    PC,    VC
C        INITIAL APPROXIMATION CONSTANTS   A,   B
C        MISCELLANEOUS CONSTANTS   TFR,  LE10
      COMMON/SUPER/ TC,PC,VC,A,B,TFR,LE10
      IWRITE = 3
C
C     OBTAIN VALUES OF THE CONSTANTS FOR THE
C     DESIRED REFRIGERANT FROM SUBROUTINE 'TABLES'
C     (THROUGH COMMON)
      CALL TABLES(NR,T)
C
C     CHECK 'PSAT'
      IF(PSAT.LE.0.0) GO TO 999
C
C     COMPUTE INITIAL ESTIMATE OF 'TSAT' FROM
C     LINEAR APPROXIMATION
      PLOG = ALOG10(PSAT)
      TR=A*PLOG + B
      ITER = 0
C
C     ITERATE TO WITHIN .01 USING NEWTON ITERATION
    1 TRO = TR
      ITER = ITER + 1
      IF(ITER.GT. 30) GO TO 998
      C = ALOG10(ABS(FVP - TRO))
      F=AVP+BVP/TRO+CVP*ALOG10(TRO)+DVP*TRO+EVP*((FVP-TRO)/
     1TRO)*C-PLOG
```

```
      FP=-BVP/TRO**2+CVP/(LE10*TRO)+DVP-EVP*(1./(LE10*TRO)+
     1FVP*C/TRO**2)
      TR=TRO-F/FP
      IF(ABS(TR-TRO).GT..01) GO TO 1
      TSAT=TR-TFR
      RETURN
C
C     PRINT ERROR MESSAGE IF
C        PSAT IS LESS THAN OR EQUAL TO ZERO
C        NUMBER OF ITERATIONS IS GREATER THAN 30
  998 TSAT=TR-TFR
      WRITE(IWRITE,1000)
      RETURN
  999 TSAT=0
      WRITE(IWRITE,1000)
 1000 FORMAT(10X,'ERROR IN CALLING SUBROUTINE -TSAT- ')
      RETURN
      END
```

```
      FUNCTION SPVOL(NR,TF,PPSIA)
C
C     PURPOSE
C        TO EVALUATE THE SPECIFIC VOLUME OF THE VAPOR PHASE
C        OF REFRIGERANT 12,22, OR 502
C        GIVEN THE PRESSURE AND TEMPERATURE
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        NR    =  REFRIGERANT NUMBER (12,22, OR 502)
C        TF    =  TEMPERATURE (F)
C        PPSIA=  PRESSURE (PSIA)
C      OUTPUT
C        SPVOL=  SPECIFIC VOLUME (CU FT/LBM)
C
C     REMARKS
C        FUNCTION SUBPROGRAM TSAT CALLED BY THIS FUNCTION
C        SUBROUTINE TABLES CALLED BY THIS SUBROUTINE
      REAL K,LE10
C
C     CONSTANTS
C
C        CRITICAL POINT PROPERTIES   TC,  PC,  VC
C        INITIAL APPROXIMATION CONSTANTS  A,  B
C        MISCELLANEOUS CONSTANTS   TFR,  LE10
      COMMON/SUPER/ TC,PC,VC,A,B,TFR,LE10
C
C     EQUATION OF STATE CONSTANTS
      COMMON/STATEQ/R,B1,A2,B2,C2,A3,B3,C3,A4,B4,C4,A5,B5,
     1C5,A6,B6,C6,K,ALPHA,CPR
      IWRITE = 5
C
C     OBTAIN VALUES OF THE CONSTANTS CORRESPONDING TO THE
C     DESIRED REFRIGERANT FROM SUBROUTINE TABLES
C     (THROUGH COMMON)
      CALL TABLES(NR,T)
C
C     CONVERT 'TF' TO 'T' AND CHECK VALUE
      T = TF + TFR
      IF(T.LE.0.0) GO TO 999
C
C     CALCULATE 'TFSAT' AND COMPARE WITH 'TF'
      TFSAT=TSAT(NR,PPSIA)
      IF(TF.LT.(TFSAT-0.050)) GO TO 999
C
C     CHECK 'PPSIA'
      IF(PPSIA.LE.0.0) GO TO 999
C
C     CALCULATE CONSTANTS
      ES0 = EXP(-K*T/TC)
```

```
      ES1=PPSIA
      ES2 = R*T
      ES3=A2+B2*T+C2*ES0
      ES4=A3+B3*T+C3*ES0
      ES5=A4+B4*T+C4*ES0
      ES6=A5+B5*T+C5*ES0
      ES7=A6+B6*T+C6*ES0
      ES32=2.*ES3
      ES43=3.*ES4
      ES54=4.*ES5
      ES65=5.*ES6
C
C     COMPUTE INITIAL ESTIMATE OF 'V' FROM IDEAL GAS LAW
      VN=R*T/PPSIA
      ITER = 0
C
C     COMPUTE 'V' TO WITHIN 1.0E-06 BY NEWTON ITERATION
    1 ITER = ITER + 1
      IF(ITER.GT.30) GO TO 998
      V = VN
      V2 = V**2
      V3 = V**3
      V4 = V**4
      V5 = V**5
      V6 = V**6
      Z = ALPHA*(V+B1)
      IF(Z.GT.150.0) Z=150.0
      EMAV=EXP(-Z)
      GO TO (2,2,3),I
    2 F=ES1-ES2/V-ES3/V2-ES4/V3-ES5/V4-ES6/V5-ES7*EMAV
      FV=ES2/V2+ES32/V3+ES43/V4+ES54/V5+ES65/V6+ES7*ALPHA*E
     1MAV
      GO TO 4
    3 EM2AV = EMAV**2
      F=ES1-ES2/V-ES3/V2-ES4/V3-ES5/V4-ES6/V5-ES7*EM2AV/(EM
     1AV+CPR)
      FV=ES2/V2+ES32/V3+ES43/V4+ES54/V5+ES65/V6+ES7*ALPHA*E
     1M2AV*(EMAV+2.*CPR)/(EMAV+CPR)**2
    4 VN=V-F/FV
      IF(ABS((VN-V)/V).GT.1.E-06) GO TO 1
      SPVCL = VN+B1
      RETURN
C
C     PRINT ERROR MESSAGE IF
C        TF IS LESS THAN OR EQUAL TO ZERO DEGREE R
C        TF IS LESS THAN TFSAT CORRESPONDING TO PSAT = PPSIA
C        PPSIA IS LESS THAN OR EQUAL TO ZERO
C        MORE THAN 30 ITERATIONS ARE NEEDED
  998 SPVCL = VN + B1
      WRITE(IWRITE,9)
```

```
      RETURN
  999 SPVOL=0.
      WRITE(IWRITE,9)
    9 FORMAT(' *****ERROR IN CALLING SUBROUTINE =SPVOL=')
      RETURN
      END
```

```
      SUBROUTINE SATPRP(NR,TF,PSAT,VF,VG,HF,HFG,HG,SF,SG)
C
C     DIMENSION AND TYPE STATEMENTS
      DIMENSION AL(3),BL(3),CL(3),DL(3),EL(3)
      REAL J,K,KTDTC,LE10,L10E
C
C     PURPOSE
C        TO EVALUATE THE SATURATION THERMODYNAMIC PROPERTIES
C        OF REFRIGERANT 12,22, OR 502
C        GIVEN THE SATURATION TEMPERATURE
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        NR    -  REFRIGERANT NUMBER (12,22, OR 502)
C        TF    -  TEMPERATURE (F)
C      OUTPUT
C        PSAT - SATURATION PRESSURE (PSIA)
C        VF    -  SPECIFIC VOLUME OF SATURATED LIQ.(CU FT/LBM)
C        VG    -  SPECIFIC VOLUME OF SATURATED VAP.(CU FT/LBM)
C        HF    -  ENTHALPY OF SATURATED LIQUID (BTU/LBM)
C        HFG   -  LATENT ENTHALPY OF VAPORIZATION (BTU/LBM)
C        HG    -  ENTHALPY OF SATURATED VAPOR (BTU/LBM)
C        SF    -  ENTROPY OF SATURATED LIQUID (BTU/LBM - R)
C        SG    -  ENTROPY OF SATURATED VAPOR (BTU/LBM - R)
C
C     REMARKS
C        FUNCTION SUBPROGRAM SPVOL CALLED BY THIS SUBROUTINE
C        SUBROUTINE TABLES CALLED BY THIS SUBROUTINE
C        FUNCTION SUBPROGRAM TSAT AVAILABLE FOR CALCULATING
C        THE SATURATION TEMPERATURE GIVEN THE SATURATION
C        PRESSURE
C
C     CONSTANTS
C
C        VAPOR PRESSURE CONSTANTS
      COMMON/SAT/AVP,BVP,CVP,DVP,EVP,FVP
C
C        CRITICAL POINT PROPERTIES   TC,  PC,  VC
C        INITIAL APPROXIMATION CONSTANTS A, B (NOT USED)
C        MISCELLANEOUS CONSTANTS   TFR,  LE10
      COMMON/SUPER/ TC,PC,VC,A,B,TFR,LE10
C
C        EQUATION OF STATE CONSTANTS
      COMMON/STATEQ/R,B1,A2,B2,C2,A3,B3,C3,A4,B4,C4,A5,B5,
     1C5,A6,B6,C6,K,ALPHA,CPR
C
C        SPECIFIC HEAT AT CONSTANT VOLUME CONSTANTS
C        ACV,BCV,CCV,DCV,ECV,FCV
C        ENTHALPY AND ENTROPY OF VAPOR CONSTANTS   X,  Y
C        MISCELLANEOUS CONSTANTS   L10E,  J
```

```
      COMMCN/CTHER/ACV,BCV,CCV,DCV,ECV,FCV,X,Y,L10E,J
C
C        LIGLIC DENSITY CONSTANTS
      DATA AL,BL,CL,DL,EL/34.84,32.76,35.0,.02696,54.634409
     1,53.48437,.834921,36.74892,63.86417,6.02683,-22.29256
     26,-70.08066,-.655549E-05,20.473289,48.47901/
C
C     OBTAIN VALUES OF THE CONSTANTS CORRESPONDING TO THE
C     DESIRED REFRIGERANT FROM SUBROUTINE TABLES
C     (THRCUGH COMMCN)
      CALL TABLES(NR,T)
      IWRITE = 5
C
C     CONVERT 'TF' TC 'T' AND CHECK VALUE
      T = TF + TFR
      IF(T.LE.0.0) GC TO 999
C
C     COMPARE 'T' WITH 'TC'
      IF(T.GT.TC) GC TC 999
C
C     CALCULATE 'PSAT'
      GC TC(10,11,11), I
   10 PSAT=10.**(AVP+BVP/T+CVP*ALCG10(T)+DVP*T)
      GO TC 12
   11 PSAT=10.**(AVP+BVP/T+CVP*ALCG10(T)+DVP*T+EVP*((FVP-T)
     1/T)*ALCG10(FVP-T))
C
C     CALCULATE 'VG'
   12 VG = SPVCL(NR,TF,PSAT)
C
C     CALCULATE 'VF'
      GC TC (1,2,2), I
    1 TCMT = TC-T
      VF=1./(AL(I)+BL(I)*TCMT+CL(I)*TCMT**(1./2.)+DL(I)*TCM
     1T**(1./3.)+EL(I)*TCMT**2)
      GC TC 3
    2 TR1 = 1.-T/TC
      VF=1./(AL(I)+BL(I)*TR1**(1./3.)+CL(I)*TR1**(2./3.)+DL
     1(I)*TR1+EL(I)*TR1**(4./3.))
C
C     CALCULATE 'HFG' BY CLAUSIUS CLAPEYRON EQUATION
    3 GC TC(31,32,32),I
   31 HFG=(VG-VF)*PSAT*LE10*(-BVP/T+CVP/LE10+DVP*T)*J
      GC TC 33
   32 HFG=(VG-VF)*PSAT*LE10*(-BVP/T+CVP/LE10+DVP*T-EVP*(L10
     1E+FVP*ALCG10(FVP-T)/T))*J
   33 SFG = HFG/T
C
C     CALCULATE 'HG' AND 'SG'
      T2 = T**2
```

```
      T3 = T**3
      T4 = T**4
      VR = VG-B1
      VR2 = 2.*VR**2
      VR3 = 3.*VR**3
      VR4 = 4.*VR**4
      KTDTC = K*T/TC
      EKTDTC = EXP(-KTDTC)
      Z = ALPHA*VG
      IF(Z.GT.150.0) Z = 150.0
      EMAV = EXP(-Z)
      H1=ACV*T+BCV*T2/2.+CCV*T3/3.+DCV*T4/4.-FCV/T
      H2 = J*PSAT*VG
      H3=A2/VR+A3/VR2+A4/VR3+A5/VR4
      H4=C2/VR+C3/VR2+C4/VR3+C5/VR4
      S1=ACV*ALOG(T)+BCV*T+CCV*T2/2.+DCV*T3/3.-FCV/(2.*T2)
      S2 = J*R*ALOG(VR)
      S3=B2/VR+B3/VR2+B4/VR3+B5/VR4
      S4 = H4
      GO TO(6,4,5), I
    4 H3=H3+A6/ALPHA*EMAV
      S3=S3+B6/ALPHA*EMAV
      GO TO 6
    5 H0=1./ALPHA*(EMAV-CPR*ALOG(1.+EMAV/CPR))
      H3=H3+A6*H0
      H4=H4-C6*H0
      S3 = S3+B6*H0
      S4 = S4-C6*H0
    6 HG=H1+H2+J*H3+J*EKTDTC*(1.+KTDTC)*H4+X
      SG=S1+S2-J*S3+J*EKTDTC*K/TC*S4+Y
C
C     CALCULATE 'HF' AND 'SF'
      HF = HG - HFG
      SF = SG-SFG
      RETURN
C
C     PRINT ERROR MESSAGE IF
C        TF IS LESS THAN OR EQUAL TO ZERO DEGREE R
C        TF IS GREATER THAN THE CRITICAL TEMPERATURE
  959 WRITE(IWRITE,1000)
 1000 FORMAT(' *****ERROR IN CALLING SUBROUTINE -SATPRP-')
      RETURN
      END
```

```
      SUBROUTINE SPFHT(NR,TF,PPSIA,CV,CP,GAMMA,SONIC)
C
C     PURPOSE
C        TO CALCULATE SPECIFIC HEAT AT CONSTANT VOLUME,
C        SPECIFIC HEAT AT CONSTANT PRESSURE, SPECIFIC
C        HEAT RATIO, AND SONIC VELOCITY FOR
C        REFRIGERANT 12,22, OR 502
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        NR    =  REFRIGERANT NUMBER (12,22, OR 502)
C        TF    =  TEMPERATURE (F)
C        PPSIA=  PRESSURE (PSIA)
C      OUTPUT
C        CV    =  SPECIFIC HEAT AT CONSTANT VOL. (BTU/LBM-R)
C        CP    =  SPECIFIC HEAT AT CONSTANT PRES. (BTU/LBM-R)
C        GAMMA=  SPECIFIC HEAT RATIO
C        SONIC=  SONIC VELOCITY (FPS)
C
C     REMARKS
C        FUNCTION SUBPROGRAM SPVOL CALLED BY THIS SUBROUTINE
C        FUNCTION SUBPROGRAM TSAT CALLED BY THIS SUBROUTINE
C        SUBROUTINE TABLES CALLED BY THIS SUBROUTINE
      REAL K
C
C     CONSTANTS
C
C        CRITICAL POINT PROPERTIES  TC,  PC,  VC
C        INITIAL APPROXIMATION CONSTANTS  A, B (NOT USED)
C        MISCELLANEOUS CONSTANTS  TFR,  LE10
      COMMON/SUPER/ TC,PC,VC,A,B,TFR,LE10
C
C       EQUATION OF STATE CONSTANTS
      COMMON/STATEQ/R,B1,A2,B2,C2,A3,B3,C3,A4,B4,C4,A5,B5,
     1C5,A6,B6,C6,K,ALPHA,CPR
C
C        SPECIFIC HEAT AT CONSTANT VOLUME CONSTANTS
C        ACV,BCV,CCV,DCV,ECV,FCV
C        ENTHALPY AND ENTROPY OF VAPOR CONSTANTS   X,  Y
C        MISCELLANEOUS CONSTANTS   L10E,  J
      COMMON/OTHER/ACV,BCV,CCV,DCV,ECV,FCV,X,Y,L10E,J
C
C     OBTAIN VALUES OF THE CONSTANTS CORRESPONDING TO THE
C     DESIRED REFRIGERANT FROM SUBROUTINE TABLES
C     (THROUGH COMMON)
      CALL TABLES(NR,T)
      IWRITE = 5
C
C     CONVERT 'TF' TO 'T' AND CHECK VALUE
      T = TF + TFR
```

```
      IF(T.LE.0.0) GO TO 999
C
C     CALCULATE 'TFSAT' AND COMPARE WITH 'TF'
      TFSAT = TSAT(NR,PPSIA)
      IF(TF.LT.TFSAT) GO TO 999
C
C     CHECK 'PPSIA'
      IF(PPSIA.LE.0.0) GO TO 999
C
C     CALCULATE 'VVAP'
      VVAP = SPVOL(NR,TF,PPSIA)
C
C     CALCULATION OF DERIVATIVES
      V1 = VVAP - B1
      V2 = V1**2
      V3 = V1**3
      V4 = V1**4
      V5 = V1**5
      V6 = V1**6
      EKTTC=EXP(-K*T/TC)
      Z = ALPHA*VVAP
      Z2 = 2.*Z
      Z3=3.*Z
      IF(Z.GT.150.0) Z = 150.0
      IF(Z2.GT.150.0) Z2 = 150.0
      IF(Z3.GT.150.0) Z3=150.0
      GO TO(1,2,3),I
    1 FDPDV=0.0
      FDPDT = 0.0
      GO TO 4
    2 FDPDV=-ALPHA*EXP(-Z)*(A6+B6*T)
      FDPDT=B6*EXP(-Z)
      GO TO 4
    3 FDPDV=-(ALPHA*(EXP(-Z3)+2.*CPR*EXP(-Z2))/(EXP(-Z2)+2.
     1*CPR*EXP(-Z)+CPR**2))*(A6+B6*T+C6*EKTTC)
      FDPDT=(B6-K*C6*EKTTC/TC)*EXP(-Z2)/(EXP(-Z)+CPR)
    4 DPDV=-R*T/V2-2.*(A2+B2*T+C2*EKTTC)/V3-3.*(A3+B3*T+C3*
     1EKTTC)/V4-4.*(A4+B4*T+C4*EKTTC)/V5-5.*(A5+B5*T+C5*EKT
     2TC)/V6+FDPDV
      DPDT=R/V1+(B2-K*C2*EKTTC/TC)/V2+(B3-K*C3*EKTTC/TC)/V3
     1+(B4-K*C4*EKTTC/TC)/V4+(B5-K*C5*EKTTC/TC)/V5+FDPDT
      GO TO(5,5,10),I
    5 FCCV = 0.0
      GO TO 15
   10 FCCV=C6*EXP(-Z)/ALPHA-(C6*CPR/ALPHA)*ALOG(1.+EXP(-Z)/
     1CPR)
C
C     CALCULATE 'CV'
   15 CV=ACV+BCV*T+CCV*T**2+DCV*T**3+FCV/T**2-(.185053*K**2
     1*T*EKTTC/TC**2)*(C2/V1+C3/(2.*V2)+C4/(3.*V3)+C5/(4.*V
```

```
     24)+FCCV)
C
C      CALCULATE 'CP'
       CP = CV-.185053*T*DPDT**2/DPDV
C
C      CALCULATE 'GAMMA'
       GAMMA = CP/CV
C
C      CALCULATE 'SCNIC'
       SCNIC=VVAP*SQRT(857.36091*T*DPDT**2/CV-4633.056*DPDV)
       RETURN
C
C      PRINT ERROR MESSAGE IF
C         TF IS LESS THAN OR EQUAL TO ZERO DEGREE R
C         TF IS LESS THAN TFSAT CORRESPONDING TO PSAT = PPSIA
C         PPSIA IS LESS THAN OR EQUAL TO ZERO
  599 WRITE(IWRITE,1000)
 1000 FORMAT(' *******-ERROR IN CALLING SUBROUTINE -SPFHT-')
       RETURN
       END
```

```
      SUBROUTINE VAPCR(NR,TF,PPSIA,VVAP,HVAP,SVAP)
C
C     PURPOSE
C        TO EVALUATE THE THERMODYNAMIC PROPERTIES
C        OF THE SUPERHEATED VAPOR PHASE
C        OF REFRIGERANT 12,22, OR 502
C        GIVEN THE TEMPERATURE AND PRESSURE
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        NR    -  REFRIGERANT NUMBER (12,22, OR 502)
C        TF    -  TEMPERATURE (F)
C        PPSIA-   PRESSURE (PSIA)
C      OUTPUT
C        VVAP -   SPECIFIC VOLUME OF VAPOR (CU FT/LBM)
C        HVAP -   ENTHALPY OF VAPOR (BTU/LBM)
C        SVAP -   ENTROPY OF VAPOR (BTU/LBM - R)
C     REMARKS
C        FUNCTION SUBPROGRAM SPVOL CALLED BY THIS SUBROUTINE
C        FUNCTION SUBPROGRAM TSAT CALLED BY THIS SUBROUTINE
C        SUBROUTINE TABLES CALLED BY THIS SUBROUTINE
      REAL K,J,KTCTC,LE10,L10E
C
C     CONSTANTS
C        CRITICAL POINT PROPERTIES  TC,  PC,  VC
C        INITIAL APPROXIMATION CONSTANTS  A, B (NOT USED)
C        MISCELLANEOUS CONSTANTS  TFR,  LE10
      COMMON/SUPER/ TC,PC,VC,A,B,TFR,LE10
C
C        EQUATION OF STATE CONSTANTS
      COMMON/STATEQ/R,B1,A2,B2,C2,A3,B3,C3,A4,B4,C4,A5,B5,
     1C5,A6,B6,C6,K,ALPHA,CPR
C        SPECIFIC HEAT AT CONSTANT VOLUME CONSTANTS
C        ACV,BCV,CCV,DCV,ECV,FCV
C        ENTHALPY AND ENTROPY OF VAPOR CONSTANTS   X,  Y
C        MISCELLANEOUS CONSTANTS   L10E,  J
      COMMON/OTHER/ACV,BCV,CCV,DCV,ECV,FCV,X,Y,L10E,J
C
C     OBTAIN VALUES OF THE CONSTANTS CORRESPONDING TO THE
C     DESIRED REFRIGERANT FROM SUBROUTINE TABLES
C     (THROUGH COMMON)
      CALL TABLES(NR,T)
      IWRITE = 3
C
C     CONVERT 'TF' TO 'T' AND CHECK VALUE
      T = TF + TFR
      IF(T.LE.0.0) GO TO 999
C
C     CALCULATE 'TFSAT' AND COMPARE WITH 'TF'
      TFSAT = TSAT(NR,PPSIA)
```

```
      IF(TF.LT.TFSAT) GO TO 999
C
C     CHECK 'PPSIA'
      IF(PPSIA.LE.0.0) GO TO 999
C
C     CALCULATE 'VVAP'
      VVAP = SPVCL(NR,TF,PPSIA)
C
C     CALCULATE 'HVAP' AND 'SVAP'
      T2 = T**2
      T3 = T**3
      T4 = T**4
      VR = VVAP-B1
      VR2 = 2.*VR**2
      VR3 = 3.*VR**3
      VR4 = 4.*VR**4
      KTDTC = K*T/TC
      EKTDTC = EXP(-KTDTC)
      Z = ALPHA*VVAP
      IF(Z.GT.150.0) Z = 150.0
      EMAV = EXP(-Z)
      H1=ACV*T+BCV*T2/2.+CCV*T3/3.+DCV*T4/4.-FCV/T
      H2 = J*PPSIA*VVAP
      H3=A2/VR+A3/VR2+A4/VR3+A5/VR4
      H4=C2/VR+C3/VR2+C4/VR3+C5/VR4
      S1=ACV*ALOG(T)+BCV*T+CCV*T2/2.+DCV*T3/3.-FCV/(2.*T2)
      S2 = J*R*ALOG(VR)
      S3=B2/VR+B3/VR2+B4/VR3+B5/VR4
      S4 = H4
      GO TO(6,4,5),I
    4 H3 = H3+A6/ALPHA*EMAV
      S3 = S3+B6/ALPHA*EMAV
      GO TO 6
    5 H0=1./ALPHA*(EMAV-CPR*ALOG(1.+EMAV/CPR))
      H3 = H3+A6*H0
      H4 = H4 -C6*H0
      S3 = S3+B6*H0
      S4 = S4-C6*H0
    6 HVAP=H1+H2+J*H3+J*EKTDTC*(1.+KTDTC)*H4+X
      SVAP=S1 +S2-J*S3+J*EKTDTC*K/TC*S4+Y
      RETURN
C
C     PRINT ERROR MESSAGE IF
C        TF IS LESS THAN OR EQUAL TO ZERO DEGREE R
C        TF IS LESS THAN TFSAT CORRESPONDING TO PSAT = PPSIA
C        PPSIA IS LESS THAN OR EQUAL TO ZERO
  999 WRITE(IWRITE,1000)
 1000 FORMAT(' ******.ERROR IN CALLING SUBROUTINE -VAPOR-')
      RETURN
      END
```

```
      SUBROUTINE TRIAL(NR,TI,DTI,P,N,ARG,TOL,V,H,S,T)
C
C     PURPOSE
C        TO DETERMINE REMAINING SUPERHEATED VAPOR PROPERTIES,
C        GIVEN THE PRESSURE AND ONE OTHER PROPERTY
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        NR    =  REFRIGERANT NUMBER (12,22, OR 502)
C        TI    =  INITIAL TEMPERATURE GUESS  (F)
C        DTI   =  INITIAL STEP SIZE FOR TEMP. ITERATION (F)
C        P     =  PRESSURE (PSIA)
C        N     =  ARGUMENT INDICATOR
C        IF N = 2, THE SECOND KNOWN PROPERTY IS SPECIFIC VOL.
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 TOLLERANCE
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     REMARKS
C        THIS PROGRAM CALLS SUBROUTINE VAPOR TO DETERMINE
C        THE DESIRED REFRIGERANT PROPERTIES
      T = TI
      DT = DTI
      DO 20 I = 1,40
      T = T+DT
      CALL VAPOR(NR,T,P,VVAP,HVAP,SVAP)
      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
      IF(DT.LT.0.0) DIFF = ARG - ARGN
      IF(DT.GT.0.0) DIFF = ARGN - ARG
      IF(DT.EQ.0.0) GO TO 25
      IF(ABS(DIFF).LE.TOL) GO TO 30
      IF(DIFF) 20,30,10
   10 T = T - DT
      DT = DT/2.0
   20 CONTINUE
   25 WRITE(5,100) N
   30 V = VVAP
      H = HVAP
      S = SVAP
      RETURN
  100 FORMAT(' *****TRIAL DOES NOT CONVERGE N=',I2,' ******')
      END
```

APPENDIX B

CARRIER® MODEL 50 D Q SERIES SINGLE PACKAGE HEAT PUMPS

(CARRIER CORPORATION, 1972, With Permission)

PHYSICAL DATA

Unit 50 D Q		004	006	008	016
Refrigerant		500	500	22	22
Compressor	Type	06R Hermetic	06D Semi-Hermetic		
	Cylinders	3	4	4	6
	RPM	3500	1750	1750	1750
Outdoor Air Fans	Type - Drive	Propeller - Direct Drive			
	No. - Dia (in)	1 - 18	1 - 24	1 - 26	2 - 26
	Nom CFM	1750	3700	5200	10,000
	Motor hp - RPM	1/6 - 1075	1/3-825	1/2-825	3/4-1140 (3 ph)
Indoor Air	Type - Drive	Centrifugal with Scroll - Belt Drive			
	No - Dia (in)	1 - 10 $\frac{5}{8}$	1 - 12 $\frac{5}{8}$	2 - 10 $\frac{5}{8}$	3 - 12
	Nom CFM	1300	2100	3220	6300
	Motor hp - RPM	1/3-1725	3/4-3450	3/4-3450	3-1725

PERFORMANCE DATA

Unit 50 DQ	Indoor Air (CFM)	Temperature Air Entering Outdoor Coil (^{o}F db at 85% R.H.) -10		0		10		20		30		40		45		50		60	
		CAP	kw	CAP	kw	CAP	kw	CAP	kw	CAP	kw	CAP	kw	CAP	kw	CAP	kw	CAP	kw
004	1200	12	2.6	15	2.7	19	2.9	23	3.1	28	3.3	33	3.5	36	3.6	38	3.7	43	4.6
006	2100	17	2.9	23	3.3	30	3.7	36	4.1	46	4.9	56	5.5	60	5.8	64	6.1	72	6.6
008	3220	19	3.6	31	4.0	44	4.6	56	5.3	67	5.9	80	6.8	87	7.3	94	7.5	106	8.3
016	6330	-	-	84	10.6	104	11.4	123	12.7	144	13.9	171	15.9	185	16.8	198	17.7	227	19.4

CAP - Instantaneous Heating Capacity in 1000's of BTU/HR, Including Indoor Fan Motor Heat

kw - Power Input including only Compressor and Outdoor Fan Motor Input

NOTE: Ratings are based on 70^{o}F Air Entering Indoor Coil and Without Auxiliary Heat

OUTDOOR FAN MOTOR INPUT

50 DQ	004	006	008	016
kw	.30	.38	.56	1.55

INDOOR FAN MOTOR DATA (See Also Indoor Fan Data)

Unit 50 DQ	Type	Nom HP	Max BHP
004	STD	1/3	.46
	FS	1/2	.75
006	STD	3/4	.95
	FS	3/4	1.24
008	STD	3/4	1.24
	FS	1	1.58
016	STD	3	3.50

STD = Standard Motor

FS = Field Supplied Motor

INDOOR FAN PERFORMANCE DATA

UNIT 50 DQ	CFM	.1	.2	.3	.4	.5	.6	.7	.8
		EXTERNAL STATIC PRESSURE (IN WG) — RPM - BHP							
004	900	750-.17	831-.23	896-.29	951-.33	1006*-.37	1052*-.43		
	1000	821-.25	889-.30	949-.34	1000*-.38	1057*-.43	1160*-.49+		
	1100	884-.31	947-.35	998*-.40	1057*-.46+	1104*-.51+			
	1200	947-.36	998*-.42	1062*-.48+	1122*-.52+				
	1300	1010*-.45	1062*-.51+	1125*-.54+					
	1400	1080*-.52+	1145*-.57+						
006	1800	638-.42	678-.47	720-.54	764-.60	804-.65	847-.73		930*-.87
	1900	658-.46	683-.53	735-.58	784-.65	827-.72	860-.79		950*-.92
	2000	678-.52	715-.57	764-.64	804-.71	847-.79	890-.85		967*-1.00+
	2100	695-.56	736-.64	784-.70	827-.77	867-.84	910-.91		990*-1.08+
	2200	715-.63	764-.69	804-.76	847-.84	890-.91	930*-.99+		1012*-1.15+
	2300	740-.67	784-.75	827-.84	870-.90	920*-.98+	950*-1.07+		1020*-1.21+
	2400	764-.75	804-.84	847-.89	890-.98+	935-1.08+	980-1.14+		
	2500	784-.83	827-.88	870-.97+	920*-1.06+	955*-1.12+	990*-1.18+		
008	2200				746-.48	800-.52	860-.61	905-.66	955-.71
	2400				790-.58	850-.63	893-.71	945-.77	1000*-.83
	2600		730-.57	785-.63	840-.70	894-.76	947-.83	995*-.83	1040*-.96
	2800	730-.63	780-.68	830-.75	890-.83	942-.91	990*-.98	1040*-1.03	1095*-1.08
	3000	790-.77	836-.82	890-.90	945-.99	994*-1.06	1045*-1.13	1095*-1.23	1145*-1.32+
	3200	846-.90	894-.98	945-1.06	1000*-1.16	1027*-1.20	1090*-1.38	1145*-1.48	
	3400	904-1.07	952-1.17	1005*-1.36+	1052*-1.42+	1094*-1.50+			
	3600	962-1.40+	1020*-1.50+	1069*-1.56+					
	3700	1000*-1.52+	1037*-1.58+						
016	4500				850-1.00	890-1.30	935-1.60	980-1.90	1025-2.20
	5000				890-1.50	930-1.75	975-2.05	1020-2.30	1070*-2.65
	5500		860-1.40	900-1.70	940-2.00	980-2.25	1025-2.50	1070*-2.80	1120*-3.10
	6000	860-1.55	900-1.80	940-2.10	980-2.40	1025-2.70	1075*-3.00	1120*-3.30	
	6300	890-1.90	930-2.20	970-2.45	1020-2.75	1060-3.00	1110*-3.35		

*-Field Supplied Drive Required
+-Field Supplied Motor Required

ADDITIONAL DATA ON MODEL 50 D Q 016

Indoor Coil - Plate Fin Type (See Figure B-1)

Fin Pitch - 14.1 Fins/in

Fin Thickness (δ) - .0055 in

Fin Material - Aluminum

Height of Coil (h) - 22.5 in

Width of Coil (L) - 86.5 in

Thickness of Coil (t) - 3.24 in

Number of Tubes in Direction of Air Flow (NT) - 3

Number of Tubes Normal to Air Flow (NP) - 18

Tube Spacing - Equilateral Triangular Pitch

Vertical Tube Spacing (S) = 1.25 in

Horizontal Tube Spacing (w) = 1.08 in

Outer Diameter of Tubes (D_o) - .506 in

Inner Diameter of Tubes (D_i) - .471 in

Number of Flow subsections (NSECT) - 9

<u>Outdoor Coil</u> - Plate Fin Type (See Figure <u>B-2</u>)

Fin Pitch - 15 Fins/in

Fin Thickness (δ) - .0055 in

Fin Material - Aluminum

Height of Coil (h) - 40 in

Width of Coil (L) - 86.5 in

Thickness of Coil (t) - 3.24 in

Number of Tubes in Direction of Air Flow (NT) - 3

Number of Tubes Normal to Air Flow (NP) - 32

Tube Spacing - Equilateral Triangular Pitch

Vertical Tube Spacing (S) - 1.25 in

Horizontal Tube Spacing (w) - 1.08 in

Outer Diameter of Tubes (D_o) - .506 in

Inner Diameter of Tubes (D_i) - .471 in

Number of Flow subsections (NSECT) - 15

<u>EQUIVALENT LENGTHS OF PIPING AND OTHER COMPONENTS (ON HEATING)</u>

Liquid Line $(\frac{L}{D})_{EQ}$ - 130

Suction Line $(\frac{L}{D})_{EQ}$ - 180

Discharge Line $(\frac{L}{D})_{EQ}$ - 144

Suction and Discharge Risers

Suction Riser - 1.32 in I. D.

Discharge Riser - 1.08 in I. D.

Thermal Expansion Valve, and Distributor Nozzle & Tubes

Thermal Expansion Valve - ALCO Controls Type

TNE IOHWIOO With A

4 A Superheat Setting

(i.e., 4° Superheat at 32°F Remote

Bulb Temp.)

Distributor Nozzle & Tubes - Sporlan Type

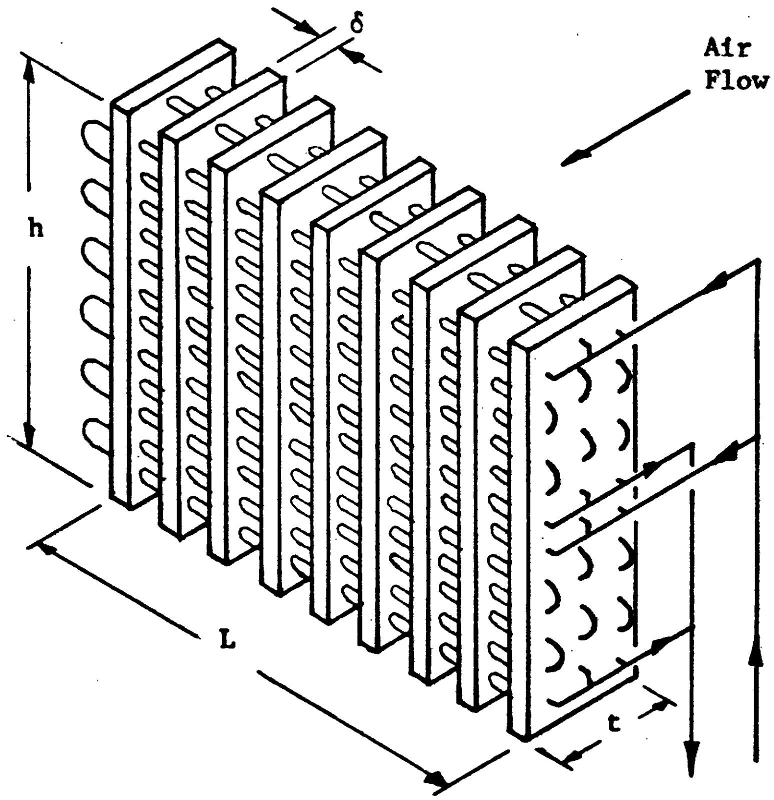
$1655\text{-}15 - \frac{3}{16} - 12 - \text{one } \frac{5}{8}$

(See Appendix D For Performance Data)

Compressor Carrier Type 06D-537

(Physical Data Given in Appendix G)

Charging Chart Data - See Figure B-3



N_{sect} = 6 as shown

D_o = tube O.D.

D_i = tube I.D.

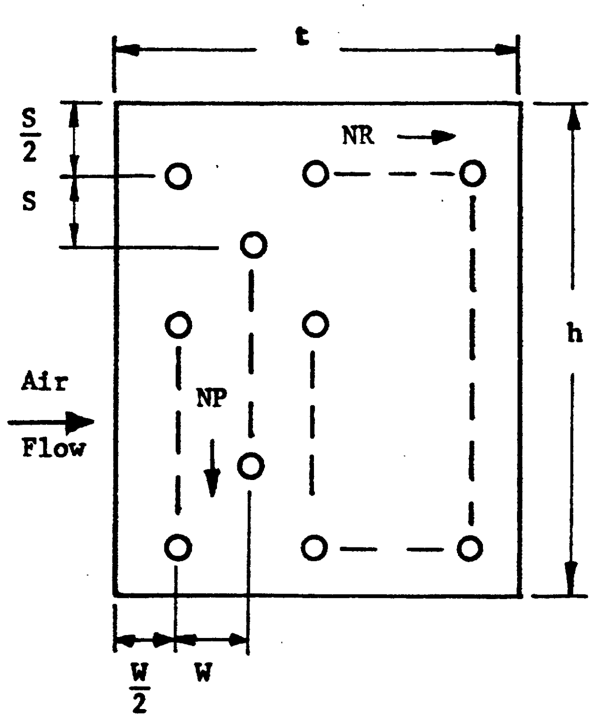
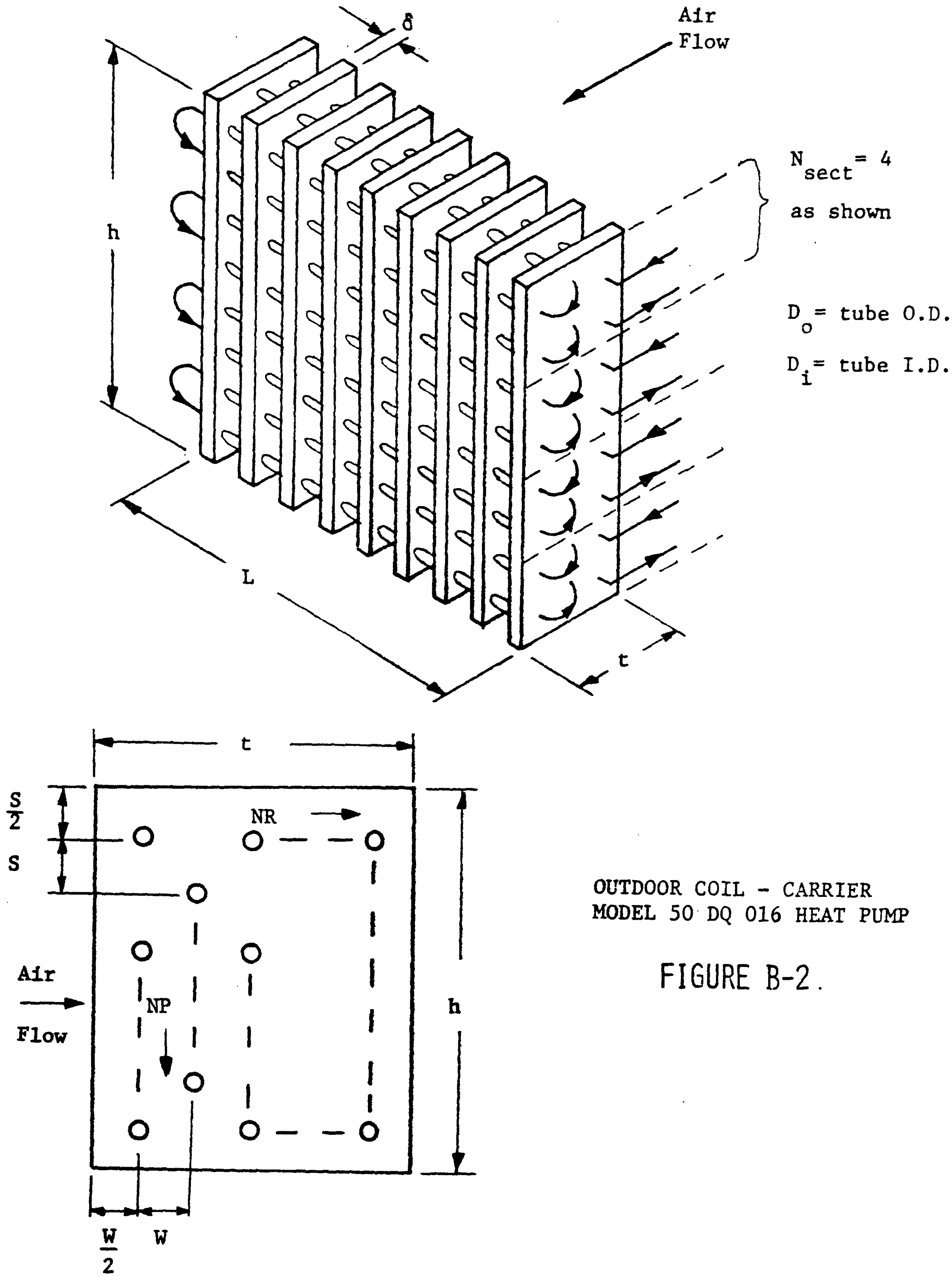


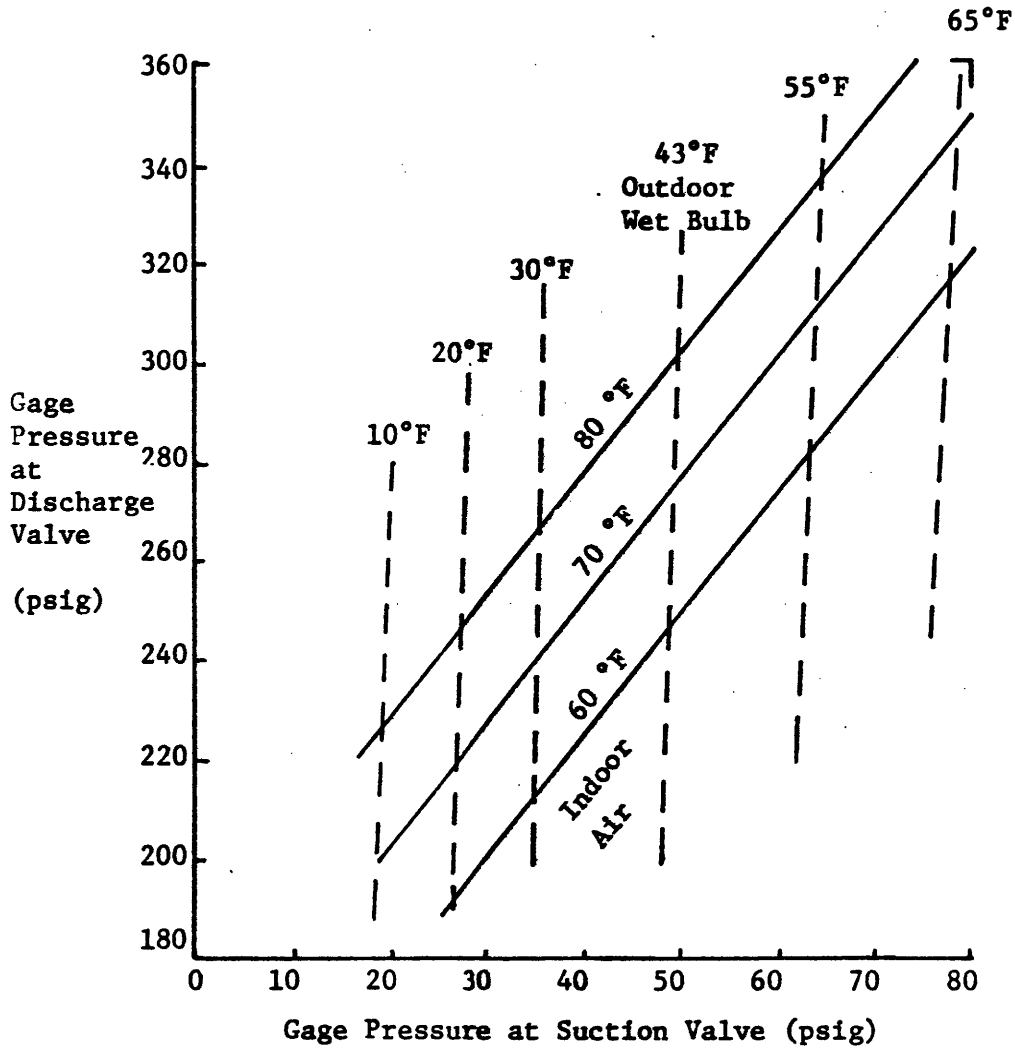
FIGURE B-1

INDOOR COIL - CARRIER
MODEL 50 DQ 016 HEAT PUMP



OUTDOOR COIL - CARRIER
MODEL 50 DQ 016 HEAT PUMP

FIGURE B-2.



CHARGING CHARTS - CARRIER MODEL 50 DQ 016
HEAT PUMP DURING HEATING MODE

FIGURE B-3

APPENDIX C

SAMPLE THERMODYNAMIC CYCLE DATA FOR SYSTEM SIMULATIONS - CONVENTIONAL VS CAPACITY CONTROLLED HEAT PUMP

Exaggerated P-h and P-V diagrams, showing details of the actual thermodynamic heat pump cycle are given in Figures C-1 and C-2. Comparing states in the cycle for both conventional and capacity controlled Carrier Model 50 DQ 016 heat pumps at the following conditions

Entering Indoor Air = $70^{\circ}F$

Entering Outdoor Air = $62^{\circ}F$ db, 85% rel. hum.

We find, from the computer simulations:

	CONVENTIONAL	CAPACITY CONTROLLED (14°BP)
Indoor Air Flow	6330 CFM	3165 CFM
Outdoor Air Flow	10,000 CFM	5200 CFM
% Heating Capacity Reduction	0	62%
STATE 1 - Evaporator Exit Saturation State		
T_1	$39.6^{\circ}F$	$44^{\circ}F$
P_1	82 psia	89 psia
STATE 2 - Evaporator Exit Superheated Vapor State		
T_2	$54^{\circ}F$	$59^{\circ}F$
P_2	≈82 psia	≈89 psia

PROCESS 2-3 SUCTION LINE		
ΔP_{2-3}	1 psi	.1 psi
STATE 3 - Superheated Vapor State Entering Compressor		
T_3	54°F	59°F
P_3	81 psia	89 psia
PROCESS 3-4 MOTOR COOLING		
Δh_{3-4}	3.9 Btu/lbm	3 Btu/lbm
STATE 4 - State of Vapor After Motor Cooling		
T_4	76°F	75°F
PROCESS 4-5 SUCTION-DISCHARGE HEAT TRANSFER		
Δh_{4-5}	≃ 0	≃ 0
STATE 5 - State of Vapor After Suct-Disc Heat Transfer		
T_5	76°F	75°F
PROCESS 5-6 SUCTION VALVE AND MANIFOLD		
ΔP_{5-6}	3 psi	3 psi
STATE 6 - State of Gas Entering Cylinder		
T_6	≃76°F	≃75°F
COMPRESSOR CYCLINDER PROCESSES		
STATE a - State of Re-Expansion Gas		
T_a	97°F	102°F

$(\frac{V_a}{V_D})$	.20	.15
STATE b - State at End of Intake and Mixing with Residual		
T_b	81°F	83°F
$(\frac{V_b}{V_D})$	1.05	.41
STATE b'- State at End of Expansion After Cut-Off		
$T_{b'}$	-	-.5°F
$P_{b'}$	-	29 psia
STATE c - State At End of Compression		
T_c	234°F	216°F
$(\frac{V_c}{V_D})$	.34	.14
P_c	360 psia	286 psia
STATE d and 7 - State At End of Discharge		
$T_{d \text{ and } 7}$	234°F	216°F
$P_{d \text{ and } 7}$	360 psia	286 psia
PROCESS 7-8 DISCHARGE VALVE AND MANIFOLD		
ΔP_{7-8}	25 psi	25 psi
STATE 8 - State of Gas Entering Disc. Manifold		
T_8	231°F	212°F
P_8	335 psia	261 psia

PROCESS 8-9 SUCTION-DISCHARGE HEAT TRANSFER

Δh_{8-9}	$\simeq 0$	$\simeq 0$

STATE 9 - State of Gas Leaving Compressor

T_9	231°F	212°F
P_9	335 psia	261 psia

PROCESS 9-10 DISCHARGE LINE

ΔP_{9-10}	.4 psia	<.1 psi

STATE 10 - State Entering Condenser

T_{10}	231°F	212°F
P_{10}	335 psia	261 psia

STATE 11 - Condenser Inlet Saturation State

T_{11}	136°F	116°F
P_{11}	≈335 psia	≈261 psia

PROCESS 11-12 FLOW THROUGH CONDENSER

ΔP_{11-12}	1.7 psia	.3 psia

STATE 12 - Condenser Exit Subcooled Liquid State

T_{12}	121°F	91°F

PROCESS 12-13 LIQUID LINE, TXV, DISTRIBUTOR

ΔP_{12-13}	249 psi	172 psi

STATE 13 - Evaporator Inlet State

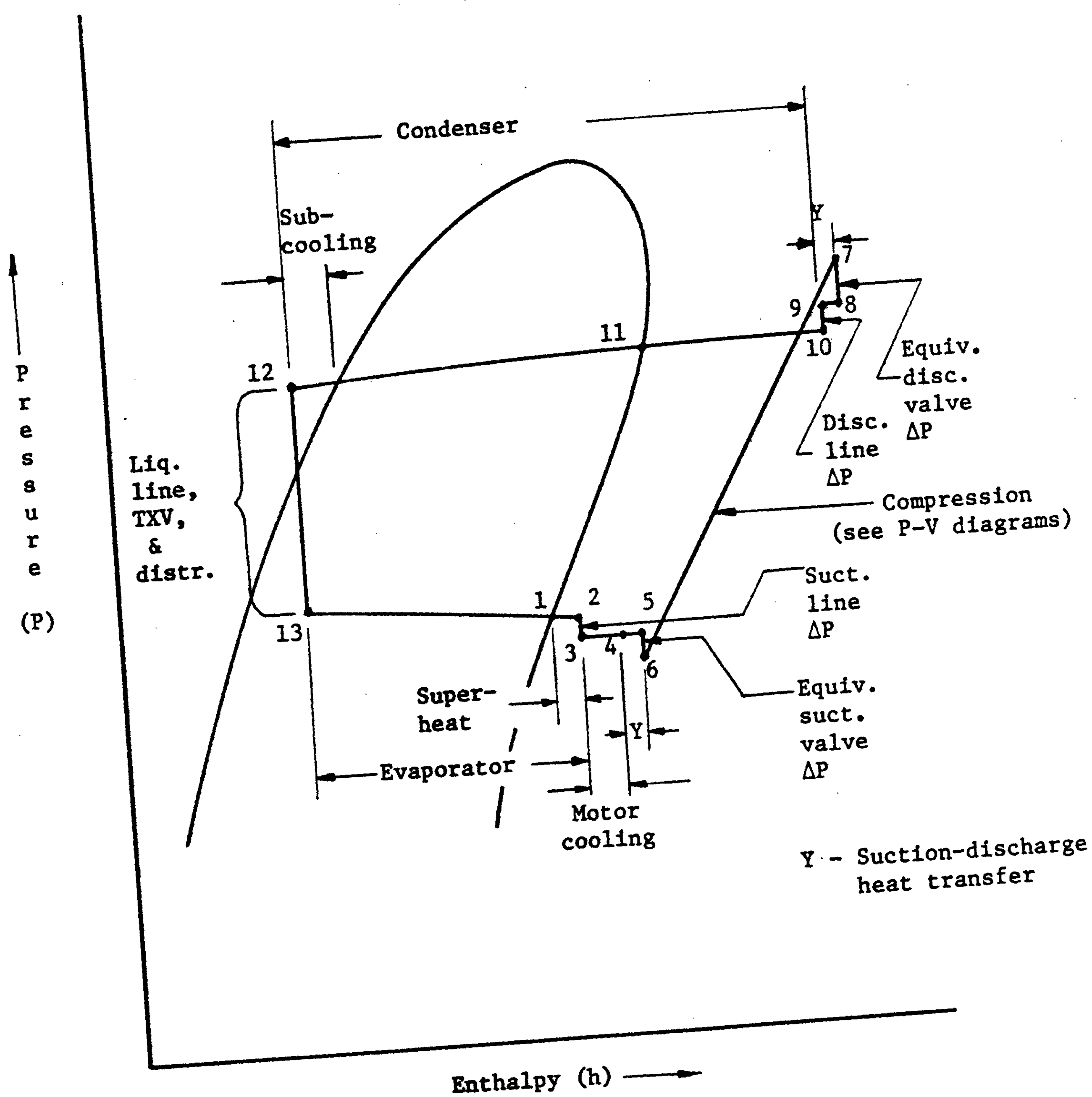
X_{13}	.28	.16
T_{13}	$40^{o}F$	$44^{o}F$
P_{13}	84.7 psi	89 psi

PROCESS 13-1 FLOW THROUGH EVAPORATOR

ΔP_{13-1}	2.7 psi	.4 psi

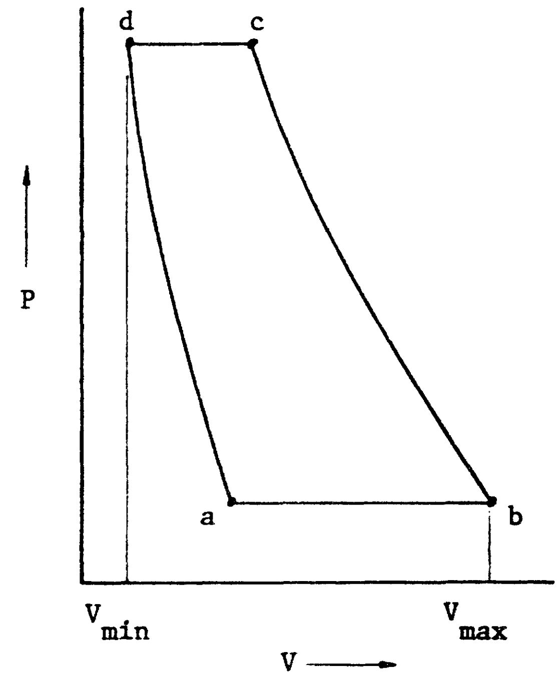
NET OUTPUT

Mass Flow	2430 lbm/hr	843 lbm/hr
Power Input to Compressor	17.6 kw	5.6 kw
Heat Rejected in Condenser	217,700 Btu/hr	82,315 Btu/hr

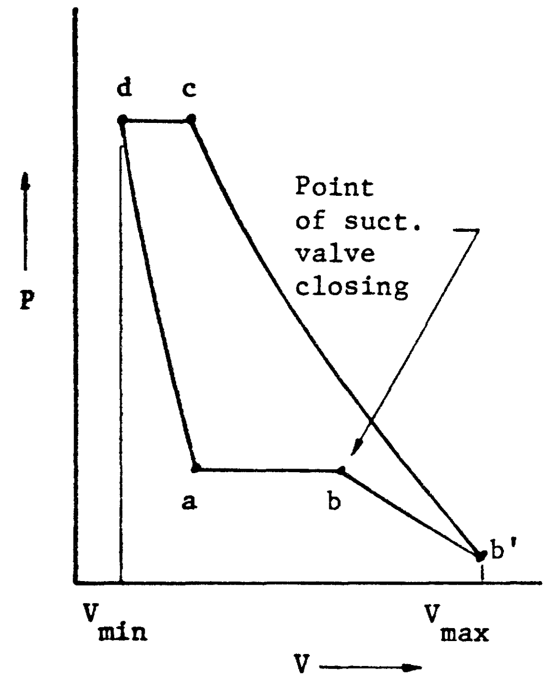


EXAGGERATED P-h DIAGRAM OF ACTUAL HEAT PUMP CYCLE

FIGURE C-1



(a)
Conventional



(b)
Capacity Controlled

CYLINDER P-V DIAGRAMS

FIGURE C-2

APPENDIX D

DETAILS OF SYSTEM FLOW BALANCE MODELING

Physical dimensions necessary for the system flow balance model outlined in Figure 2.1-3 are: diameters of flow passages and lengths or equivalent lengths of the flow paths, compressor dimensions, and expansion device data. Information on required compressor data is given in section 2.3, and Appendix E.

Pressure drops through piping and components other than the heat exchangers and expansion device are found using the equivalent length method of pressure drop calculation for incompressible flow:

$$\Delta P = 4\ f\ \left(\frac{L}{D}\right) \frac{G^2}{2\rho\ g_c} \qquad \text{(psi)}$$

Where:

f = Moody friction factor (subprogram 'FRICT', in Appendix L produces values of Moody friction factor for laminar, transition, and turbulent flow regimes, and for rough as well as smooth pipes)

G = Mass flow per unit flow area ($lbm/hr\text{-}ft^2$)

ρ = inlet fluid density (lbm/ft^3)

$\left(\frac{L}{D}\right)$ = equivalent length

g_c = conversion factor ($32.2\ lbm\text{-}ft/lbf\ sec^2$)

Subprogram "DPLINE' , in Appendix L , is used to determine the above pressure drop. Values of L/D used in simulating the Carrier model 50 DQ 016 heat pump system are given in Appendix B.

Two-phase pressure drops in the condenser and evaporator are calculated by subprogram 'PDROP', outlined in Appendix L. Subprogram 'PDROP' is capable of determining single phase liquid and vapor pressure drops in the heat exchangers, as well as two-phase pressure drops, although as used in the system flow balance model, only the two-phase capability is employed. The two-phase pressure drop correlations are from Lockhart and Martinelli[1], and are valid for laminar, transition, and turbulent flow regimes, as are all of the single phase pressure drop relations.

Expressions describing the distributor nozzle and tubes for the Carrier model 50 DQ 016 heat pump during the heating mode, developed by curve fitting published performance data[2], are as follows:

Nozzle:

$$\Delta P_{noz} = 25.0\ (\%\ CAP_{noz})^{1.8384} \quad \text{(psi)} \quad \%CAP_{noz} \leq 1.2$$

$$\Delta P_{noz} = 29.408\ (\%\ CAP_{noz})^{.954735} \quad \text{(psi)} \quad \%CAP_{noz} > 1.2$$

$$\%CAP_{noz} = \frac{CAP}{CAP_{noz}}$$

$$CAP_{noz} = (12000)\ (CORFAC)\ 10^{[.00511\ (T_{sat_{evap}}) + .944803]}$$

(Btu/hr)

$$CORFAC = 10^{[-.006444\,(TROC) + .6444]} \qquad TROC \leq 100^{\circ}F$$

$$CORFAC = 10^{[-.007133\,(TROC) + .7133]} \qquad TROC > 100^{\circ}F$$

Where:

CAP = amount of heat transfer in the evaporator (Btu/hr)

CAPNOZ = amount of heat that could be transferred in the evaporator at the rated pressure drop across the nozzle (btu/hr)

CORFAC = correction factor for non-standard rating temperature of incoming refrigerant liquid

$\%CAP_{noz}$ = percent of rated capacity

TROC = temperature of refrigerant liquid leaving condenser

ΔP_{noz} = pressure drop through nozzle under the given conditions

Tubes:

$$\Delta P_{tubes} = (10.0)\,(\%CAP_{tubes})^{1.81217} \qquad (psi)$$

$$\%CAP_{tubes} = \frac{CAP}{CAP_{tubes}}$$

$$CAP_{tubes} = (N_{sect})\,(12000)\,(CORFAC)\,10^{[.005291\,(T_{sat\,evap}) - .48733]}$$

(Btu/hr)

Where:

CAP_{tubes} = amount of heat that could be transferred in the evaporator at the rated pressure drop across the tubes (btu/hr)

N_{sect} = number of tubes or separate flow paths in evaporator

$\%CAP_{tubes}$ = percent of rated capacity

ΔP_{tubes} = pressure drop through tubes under the given conditions

The expression for the thermal expansion valve coefficient CTXV for the 50 DQ 016 unit on heating was found to be:

$$CTXV = .002128\ (T_{\substack{sat\\evap}})^2 + .2491\ (T_{\substack{sat\\evap}}) + 9.455\ \frac{(\frac{lbm}{hr})}{[(\frac{lbm}{ft^3})\ (\frac{lbf}{in^2})]^{1/2}}$$

The system flow balance model, as outlined in the flow chart of Figure D-1, requires that the pressure drop across the expansion device 'DPACT' be equal to that available across the device, 'DPSYS'. These quantities are defined as:

$$DPACT = \Delta P_{TXV} + \Delta P_{noz} + \Delta P_{tubes}$$

$$DPSYS = POC - PIE$$

where

$$POC = P_{IC} - \Delta P_{\substack{disc\\line}} - \Delta P_{cond} - \Delta P_{\substack{liq\\line}}$$

$$PIE = P1OE + \Delta P_{evap} + \Delta P_{\substack{suct\\line}}$$

PIC = Pressure at exit from compressor

$\Delta P_{\substack{disc\\line}}$ = Pressure drop in discharge line

ΔP_{cond} = Pressure drop through condenser

$\Delta P_{\substack{liq\\line}}$ = Pressure drop in liquid line

P1OE = Pressure at entrance to compressor

$\Delta P_{\substack{suct\\line}}$ = Pressure drop in suction line

ΔP_{evap} = Pressure drop through evaporator

Thermodynamic properties required for the system flow balance model are determined from basic equations, as described in Appendix A. To check for liquid line flashing it is necessary to determine the saturation pressure corresponding to the temperature of the refrigerant leaving the condenser. If the drop in pressure in the liquid line is enough to lower the pressure below the saturation pressure, then some liquid will flash into vapor.

Finally, to check for adequate oil return in suction and discharge risers, we must compute the actual vapor velocity in the risers and compare it to the minimum vapor velocity required for oil entrainment.

Data on minimum velocity for oil entrainment in suction and discharge risers, for refrigerant 22[3], has been curve fit, yielding the following:

$$VSR = (60.0)\ 10^{[-.005875\ (T_{\substack{sat\\evap}}) + .5 \log_{10}(D_{SL}) + 3.4826]}$$
(ft/hr)

$$VDR = (60.0)\ 10^{[-.00315\ (T_{\substack{sat\\cond}}) + .5 \log_{10}(D_{DL}) + 3.40]}$$
(ft/hr)

Where:

D_{SL} = Diameter of suction rister (ft)

D_{DL} = Diameter of discharge riser (ft)

VSR = Minimum velocity for oil entrainment for refrigerant 22 in suction riser (ft/hr)

VDR = Minimum velocity for oil entrainment for refrigerant 22 in discharge riser (ft/hr)

For more information concerning the specifics of the computer simulation of the system flow balance model, see comments in the program listing at the end of this section.

REFERENCES

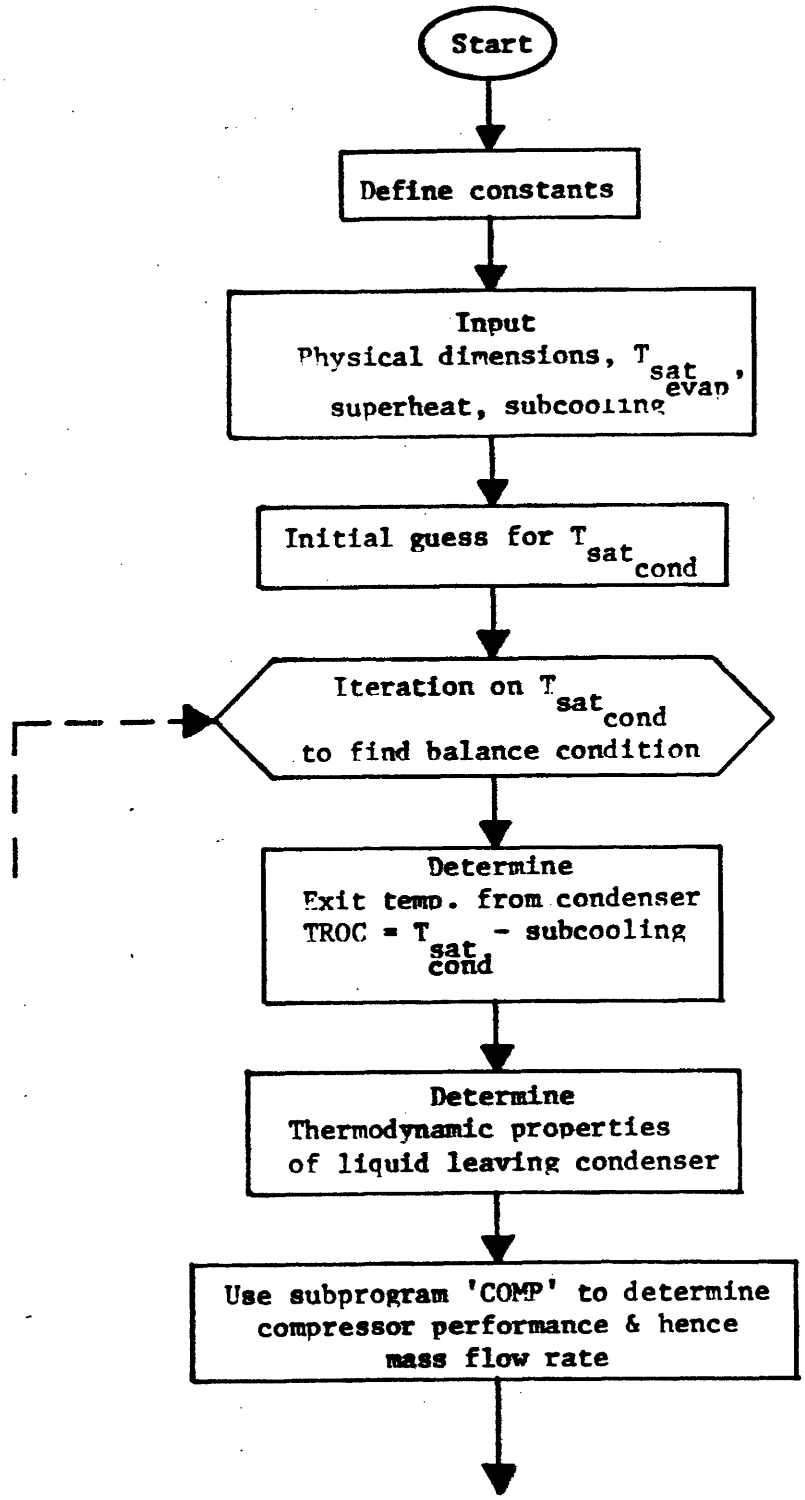
1. Lockhart, R.W. and Martinelli, R.C., "Proposed Correlation of Data For Isothermal Two-Phase, Two-Component Flow in Pipes", Chemical Engineering Process, Vol. 45, No. 1, pg. 39 (1949).

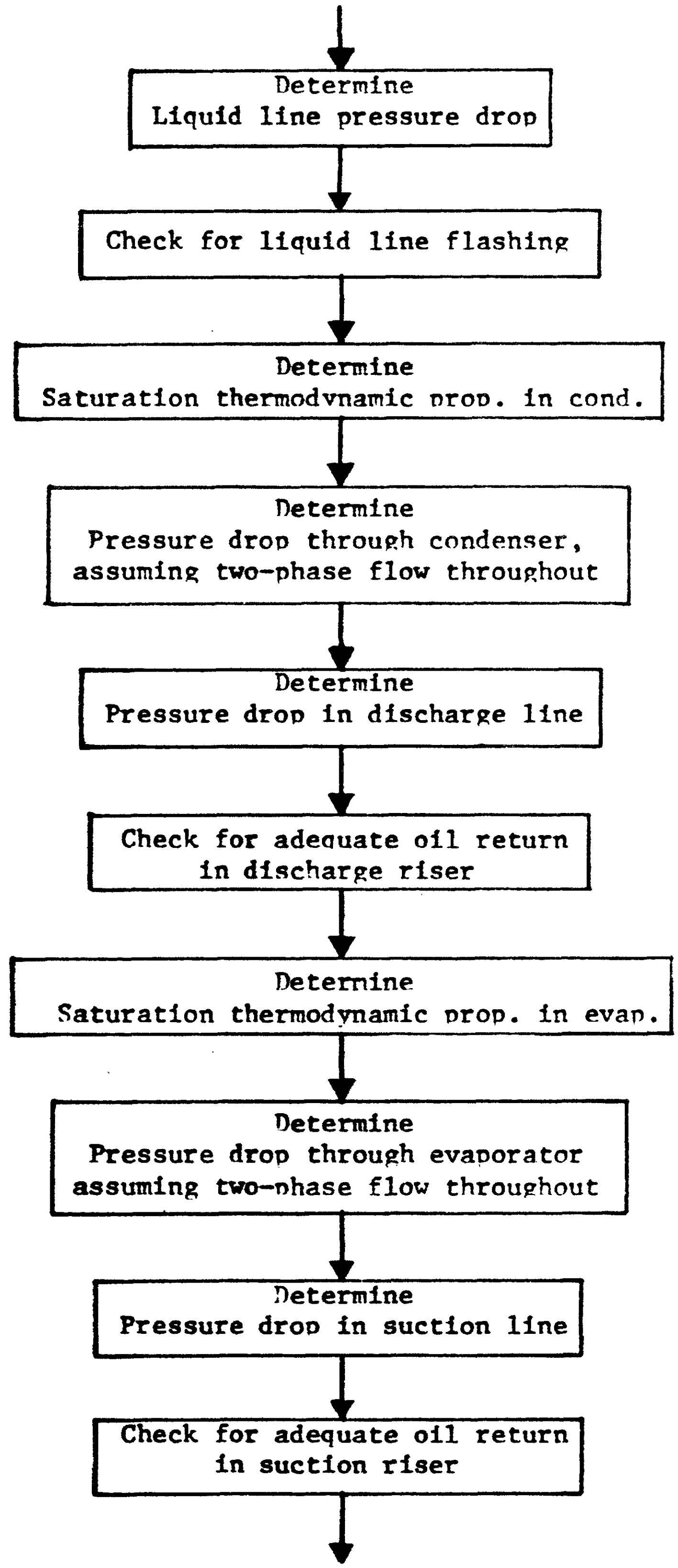
2. "Refrigerant Distributors", Bulletin 20-10, (Sporlan Valve Company, St. Louis, Missouri), July 1973.

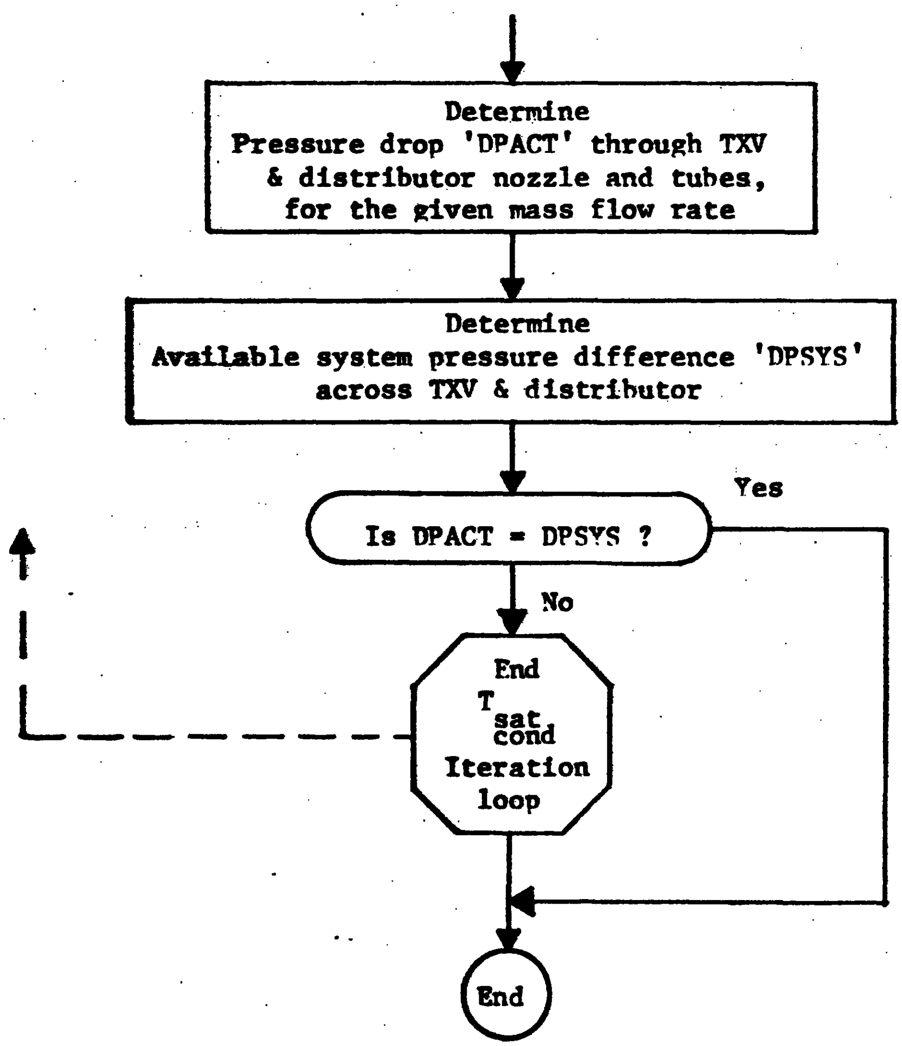
3. ASHRAE GUIDE & DATA BOOK, SYSTEMS & EQUIPMENT VOL. (New York: American Soc. of Heat., Refg., & Air-Cond. Eng., Inc., 1967) pg. 801-805.

FIGURE D-1

FLOW CHART FOR SYSTEM FLOW BALANCE MODEL




Determine
Liquid line pressure drop
Check for liquid line flashing
Determine
Saturation thermodynamic prop. in cond.
Determine
Pressure drop through condenser,
assuming two-phase flow throughout
Determine
Pressure drop in discharge line
Check for adequate oil return
in discharge riser
Determine
Saturation thermodynamic prop. in evap.
Determine
Pressure drop through evaporator
assuming two-phase flow throughout
Determine
Pressure drop in suction line
Check for adequate oil return
in suction riser


Determine
Pressure drop 'DPACT' through TXV & distributor nozzle and tubes, for the given mass flow rate
Determine
Available system pressure difference 'DPSYS' across TXV & distributor
Is DPACT = DPSYS ?
Yes
No
End
$T_{sat\ cond}$
Iteration
loop
End

```
C          SYSTEM FLOW BALANCE PROGRAM
C
C       PURPOSE
C          TO DETERMINE THE CONDENSER AND EVAPORATOR CONDITIONS
C          WHICH, WITH GIVEN THERMAL EXPANSION VALVE BEHAVIOR
C          AND A GIVEN COMPRESSOR (EITHER CONVENTIONAL OR
C          CAPACITY CONTROLLED), WILL PRODUCE A MASS FLOW
C          BALANCE IN THE SYSTEM
C
C       EXPLICIT INPUT PARAMETERS
C          DLLOC=  DIAMETER OF LIQUID LINE COMMING FROM
C                  OUTDOOR COIL (FT)
C          XLEQLO= EQUIVALENT LENGTH OF LIQUID LINE COMMING
C                  FROM OUTDOOR COIL (L/D - DIMENSIONLESS)
C          DLLIC=  DIAMETER OF LIQUID LINE COMMING FROM INDOOR
C                  COIL (FT)
C          XLEQLI= EQUIVALENT LENGTH OF LIQUID LINE COMMING
C                  FROM INDOOR COIL (L/D - DIMENSIONLESS)
C          DSL   = DIAMETER OF SUCTION LINE (VAPOR) (FT)
C          XLEQSL= EQUIVALENT LENGTH OF SUCTION LINE
C                  (L/D - DIMENSIONLESS)
C          DDL   = DIAMETER OF DISCHARGE LINE (VAPOR) (FT)
C          XLEQDL= EQUIVALENT LENGTH OF DISCHARGE LINE
C                  (L/D - DIMENSIONLESS)
C          DOC   = INSIDE DIAMETER OF TUBES IN OUTDOOR COIL (FT)
C          DZOC  = REFRIGERANT FLOW LENGTH IN EACH PARALLEL
C                  FLOW BRANCH IN THE OUTDOOR COIL (FT)
C          DIC   = INSIDE DIAMETER OF TUBES IN INDOOR COIL (FT)
C          DZIC  = REFRIGERANT FLOW LENGTH IN EACH PARALLEL
C                  FLOW BRANCH IN THE INDOOR COIL (FT)
C          M     = NUMBER OF DIFFERENT FLOW CONFIGURATIONS TO
C                  BE STUDIED
C          TSATE=  SATURATION TEMP.AT EXIT OF EVAPORATOR (F)
C          DTE   = TEMPERATURE INCREMENT FOR EVAP. (F)
C          NE    = NUMBER OF EVAP. TEMPS. EXAMINED
C          TSATC=  SATURATION TEMP. AT ENTRANCE TO CONDENSER (F)
C          DTC   = TEMPERATURE INCREMENT FOR COND.(F)
C          NC    = NUMBER OF CONDENSER TEMPS. EXAMINED
C          SUPER=  SUPERHEAT OF VAPOR LEAVING EVAPORATOR AND
C                  ENTERING THE COMPRESSOR (F)
C          DTROC=  GUESS FOR AMOUNT OF SUBCOOLING OF REFRIGERANT
C                  LEAVING CONDENSER (F)
C          NCORH=  INDICATOR FOR COOLING OR HEATING MODE
C                  IF 'NCORH' = 1 - COOLING MODE
C                  IF 'NCORH' = 2 - HEATING MODE
C          NSECT=  NUMBER OF PARALLEL FLOW SECTIONS
           TSATBF= AN INPUT TEMPERATURE USED TO SPECIFY THE
                   FLOW COEFFICIENT OF THE THERMAL EXPANSION
                   VALVE (F)
```

```
C          ICONTR= CONTROL INDICATOR
C                  'ICONTR' = 1 MEANS CONVENTIONAL SYSTEM
C                  'ICONTR' = 2 MEANS CAPACITY CONTROLLED
C                  SYSTEM
C          CUTOFF= A PARAMETER USED TO INDICATE THE AMOUNT OF
C                  CAPACITY CONTROL USED - IT IS DEFINED AS
C                  'CUTOFF' = 1.2 - (VOL.CUT - VOL.MIN)/VOL.DISP.
C                  WHERE VOL.CUT IS THE VOLUME SWEPT BY THE
C                  PISTON BEFORE THE SUCTION VALVE IS CLOSED
C                  VOL.MIN IS THE CLEARANCE VOLUME
C                  VOL.DISP. IS THE DISPLACEMENT VOLUME
C                  (ALL PER CYLINDER)
C
C       INPUT PARAMETERS IN COMMON
C          NCYL  =  NUMBER OF COMPRESSOR CYLINDERS
C          VR    =  CLEARANCE VOLUME RATIO, DEFINED AS
C                   'VR' = VOL.MIN/VOL.DISP.
C          VD    =  DISPLACEMENT VOLUME PER CYLINDER (VOLUME
C                   SWEPT BY PISTON) (CU FT)
C          SYNC  =  SYNCHRONOUS MOTOR SPEED (RPM)
C          RPM   =  INITIAL GUESS FOR ACTUAL MOTOR SPEED (RPM)
C          EFFIS=   ISENTROPIC EFFICIENCY OF THE COMPRESSION AND
C                   EXPANSION PORTIONS OF THE CYLINDER
C                   PROCESSES
C          DPDI  =  EQUIVALENT PRESSURE DROP ACROSS DISCHARGE
C                   VALVE TO ACCOUNT FOR VALVE DYNAMICS AND
C                   FLOW LOSSES (PSI)
C          DPS   =  EQUIVALENT PRESSURE DROP ACROSS SUCTION
C                   VALVE TO ACCOUNT FOR VALVE DYNAMICS AND
C                   FLOW LOSSES (PSI)
C          DPFRAC=  (NOT USED HERE)
C          SDELAY=  SUCTION VALVE CLOSING DELAY (DEGREES AFTER
C                   BOTTOM DEAD CENTER)
C          PMC   =  PERCENT OF COMPRESSOR MOTOR HEAT WHICH IS
C                   REMOVED BY THE SUCTION GAS (THE REMAINDER
C                   IS LOST BY CONVECTION TO THE AMBIENT)
C          PHT   =  (NOT USED HERE)
C          PHTD  =  (NOT USED HERE)
C          EAD   =  (NOT USED HERE)
C          EAS   =  (NOT USED HERE)
C          PWRNL=   (NOT USED HERE)
C          EFFME=   MECHANICAL EFFICIENCY OF COMPRESSOR
C          POWMAX=  MAXIMUM POWER OUTPUT OF COMPRESSOR MOTOR
C                   (KW) WHEN OPERATING AT MAXIMUM PERMISSIBLE
C                   OVERLOAD
C          EQAREA=  EQUIVALENT HEAT TRANSFER AREA BETWEEN
C                   SUCTION AND DISCHARGE MANIFOLDS - THIS IS
C                   USED TO GIVE A ROUGH APPROXIMATION OF
C                   INTERNAL HEAT TRANSFER LOSSES, IF A
C                   PARTICULAR COMPRESSOR DESIGN SHOULD REQUIRE
```

```
C                    THAT IT BE INCLUDED
C                    (THERE IS, HOWEVER, NO REPLACEMENT FOR THE
C                    ACTUAL CONDITIONS IN EACH COMPRESSOR)
C          DDELAY- DISCHARGE VALVE CLOSING DELAY (DEGREES
C                    AFTER TOP DEAD CENTER)
C          XOIL -  OIL CIRCULATION RATE (LBM OIL/LBM OF REF.+OIL)
C
C       INPUT DATA CONSTANTS
C          SLPEMV & XINMV - COEFFICIENTS FOR DETERMINING
C                           VISCOSITY OF REFRIGERANT VAPOR
C          XM1-XM4        - COEFFICIENTS FOR DETERMINING
C                           VISCOSITY OF REFRIGERANT LIQUID
C          C1-C3          - COEFFICIENTS FOR DETERMINING THE
C                           THERMAL EXPANSION VALVE BEHAVIOR
C NOTE: THE INPUT DATA CONSTANTS ARE FOR REFRIGERANT 22 ONLY
C
C       OUTPUT PARAMETERS
C          TSATE AND TSATC FOR A REFRIGERANT FLOW BALANCE (F)
C          XMR  -  REFRIGERANT MASS FLOW RATE AT BALANCE
C                    CONDITIONS (LBM/HR)
C          POW  -  TOTAL COMPRESSOR INPUT POWER AT FLOW
C                    BALANCE CONDITIONS (KW)
C          TIC  -  TEMPERATURE OF REFRIGERANT ENTERING COND.(F)
C
C       REMARKS
C          THIS PROGRAM CALLS SUBROUTINE 'SATPRP' TO DETERMINE
C          SATURATION PROPERTIES OF REFRIGERANTS
C          THIS PROGRAM CALLS SUBROUTINE 'COMP' TO DETERMINE
C          COMPRESSOR PERFORMANCE
C          THIS PROGRAM CALLS SUBROUTINE 'DPLINE' TO DETERMINE
C          PRESSURE DROPS IN SINGLE PHASE REGIONS OF
C          CONNECTING PIPING
C          THIS PROGRAM CALLS SUBROUTINE 'PDROP' TO DETERMINE
C          PRESSURE DROPS IN TWO-PHASE FLOW IN THE HEAT EXCH.
C          THIS PROGRAM USES FUNCTION SUBPROGRAM 'TSAT' TO
C          DETERMINE SATURATION TEMPERATURES CORRESPONDING TO
C          GIVEN PRESSURES
C
      COMMON/COMPR/NCYL,VR,VD,SYNC,RPM,EFFIS,DPDI,DPS,DPFRAC,
     1SDELAY,PMC,PHT,PHTC,EAD,EAS,XMR,POW,TIC,HIC,H1CE,PIC,
     2P1CE,T1CE,POWMAX,PWRNL,EGAREA,DDELAY,XOIL,EFFME
      DATA SLPEMV,XINMV/.0020759,.0272/
      DATA XM1,XM2,XM3,XM4/-5.625E-08,1.525E-05,-2.982E-03,
     11.646/
      DATA C1,C2,C3/2.128E-03,.2491,9.455/
      XOIL = 0.0
      DDELAY = 0.0
      EGAREA = 0.0
      PWRNL = 0.0
      SYNC = 1800.0
```

```
      POWMAX = 16.89
      NR = 22
      EFFIS = .94
      CPDI = 25.0
      CPS = 3.0
      CPFRAC = 0.0
      NCYL = 6
      RPM = 1750.0
      EAC = 0.0
      EAS = 0.0
      EFFME = .96
      PHT = 0.0
      PHTD = 0.0
      PMC = .85
      SDELAY = 0.0
      READ(8,620)DLLCC,XLEQLO,DLLIC,XLEQLI,DSL,XLEQSL,DCL,
     1XLEQDL
      READ(8,630) DCC,DZOC,DIC,DZIC
      READ(8,490) M
      DO 130 J = 1,M
      READ(8,600) TSATEI,DTE,NE,TSATCI,DTC,NC,SUPER
      READ(8,610)DTRCC,NCORH,NSECT
      READ(8,640) TSATBP,ICONTR,CUTOFF
      WRITE(5,550) DCC,DZOC,DIC,DZIC
      WRITE(5,560) DLLCC,XLEQLO,DLLIC,XLEQLI
      WRITE(5,570) DSL,XLEQSL,DCL,XLEQDL
      WRITE(5,590) TSATBP
C
C--------LOOP FOR ITERATING ON CONDENSER TEMPERATURE-------
C
C     ITERATE TO FIND THE CONDENSER TEMPERATURE WHICH
C     GIVES A SYSTEM FLOW BALANCE FOR THE GIVEN TSATE,
C     THERMAL EXPANSION VALVE BEHAVIOR, AND COMPRESSOR BEHAVIOR
C
      TSATC = TSATCI
      TSATE = TSATEI
      DT = 2.0
      DO 100 I = 1,20
      TSATC = TSATC + DT
      WRITE(6,515) TSATE,TSATC,CUTOFF
C
C     PROVISION FOR RUN-TIME INTERACTIVE DATA INPUT
C
      READ(6&514,ECHO=6)
C
C     CALCULATE THE GUESSED TEMP. 'TROC' FOR EXIT TEMP. OF
C     REFRIGERANT FROM CONDENSER (THIS MUST BE CHECKED
C     MANUALLY WITH THE OUTPUT OF THE CONDENSER PERFORMANCE
C     PROGRAM)
C
```

```
      TRCC = TSATC - DTRCC
C
C     DETERMINE PROPERTIES OF LIQUID REFRIGERANT LEAVING COND.
C
      CALL SATPRP(NR,TRCC,P,VF,VG,H3,HFG,HG,SF,SG)
      RHO3 = 1.0/VF
      XMUL = XM1*TRCC**3 + XM2*TRCC**2 + XM3*TRCC + XM4
      VR = .05
      VD = .003522
C
C     CALL SUBROUTINE COMP TO DETERMINE THE COMPRESSOR
C     PERFORMANCE AND REFRIGERANT FLOW RATE 'XMR'
C
      CALL COMP(NR,TSATE,DTE,NE,TSATC ,DTC,NC,SUPER,ICONTR,
     1CUTOFF)
      IF(NCORH.EQ.1) GO TO 40
      IF(NCORH.EQ.2) GO TO 50
C
C     DEFINE VARIABLES IF OPERATING IN THE COOLING MODE
C
   40 DERC = DOC
      GRC = 4.0*XMR/(15.0*3.14*DERC**2)
      DZTPC = DZOC
      DERE = DIC
      GRE = 4.0*XMR/(9.0*3.14*DERE**2)
      DZTPE = DZIC
      DLL = DLLOC
      XLEQLL = XLEQLO
      GO TO 60
C
C     DEFINE VARIABLES IF OPERATING IN THE HEATING MODE
C
   50 DERC = DIC
      GRC = 4.0*XMR/(9.0*3.14*DERC**2)
      DZTPC = DZIC
      DERE = DOC
      GRE = 4.0*XMR/(15.0*3.14*DERE**2)
      DZTPE = DZOC
      DLL = DLLIC
      XLEQLL = XLEQLI
   60 E = 5.0E-06
C
C     DETERMINE LIQUID LINE PRESSURE DROP 'DPLL' (PSI)
C
      CALL DPLINE(DLL,XLEQLL,E,XMR,RHO3,XMUL,DPLL)
C
C     DETERMINE SATURATION PROPERTIES  OF REFRIGERANT IN COND.
C
      CALL SATPRP(NR,TSATC ,P,VF,VV,HF,HFG,HG,SF,SG)
      RHOL = 1.0/VF
```

```
      RHOV = 1.0/VV
      XMUL = XM1*TSATC **3 + XM2*TSATC **2 + XM3*TSATC  + XM4
      XMUV = SLPEMV*TSATC  + XINMV
C
C     DETERMINE PRESSURE DROP IN CONDENSER 'DELPC' (PSI)
C     ASSUMING THAT THE TWO-PHASE PRESSURE DROP IS APPROX.
C     THE TOTAL CONDENSER PRESSURE DROP
C
      CALL PDROP(4,DERC,E,GRC,XMUV,XMUL,RHOV,RHOL,1.0,1.0,
     1DZTPC,0.0,1.0,VV,0.0,0.0,DELPC)
C
C     DETERMINE THE ACTUAL VAPOR VELOCITY IN THE DISCHARGE
C     RISER 'VDRACT' (FT/HR)
C
      VDRACT = XMR*4.0/(RHOV*3.14*DDL**2)
C
C     DETERMINE THE PRESSURE DROP 'DPDL' (PSI) OF VAPOR
C     IN THE DISCHARGE LINE
C
      CALL DPLINE(DDL.XLEGDL,E,XMR,RHOV,XMUV,DPDL)
C
C     DETERMINE SATURATION PROPERTIES OF REFRIGERANT IN THE
C     EVAPORATOR
C
      CALL SATPRP(NR,TSATE,P,VF,VV,HLIQ,HFG,HG,SF,SG)
      RHOL = 1.0/VF
      RHOV = 1.0/VV
      XMUL = XM1*TSATE**3 + XM2*TSATE**2 + XM3*TSATE + XM4
      XMUV = SLPEMV*TSATE + XINMV
      XI = (H3-HLIQ)/HFG
C
C     DETERMINE PRESSURE DROP IN THE EVAPORATOR 'DELPE' (PSI)
C     ASSUMING THAT THE TWO-PHASE PRESSURE DROP IS APPROX.
C     THE TOTAL EVAPORATOR PRESSURE DROP
C
      CALL PDROP(3,DERE,E,GRE,XMUV,XMUL,RHOV,RHOL,1.0,1.0,
     1DZTPE,1.0,XI,VV.2.0,0.0,DELPE)
C
C     DETERMINE ACTUAL VAPOR VELOCITY IN SUCTION RISER
C     'VSRACT' (FT/HR)
C
      VSRACT = XMR*4.0/(RHOV*3.14*DSL**2)
C
C     DETERMINE PRESSURE DROP OF VAPOR IN SUCTION LINE 'DPSL'
C     (PSI)
C
      CALL DPLINE(DSL.XLEGSL,E,XMR,RHOV,XMUV,DPSL)
      CAP = XMR*(H1CE - H3)
      PIE = P1CE - DELPE + DPSL
      POC = PIC + DELPC - DPDL - DPLL
```

```
      T = TSAT(NR,PCC)
      IF(T.LE.TRCC) WRITE(5,595)
      CAPPT = CAP/FLOAT(NSECT)
C
C     DETERMINE PRESSURE DROP THROUGH DISTRIBUTOR NOZZLE
C     AND TUBES
C
      IF(TRCC.LE.100.0) CORFAC=10.0**(-.006444*TROC+.6444)
      IF(TRCC.GT.100.0) CORFAC = 10.0**(-.007133*TROC+.7133)
      TIE = TSAT(NR,PIE)
      IF(NCORR.EQ.1)CAPNOZ=10.0**(.004842*TIE+.59162)*
     112000.0*CORFAC
      IF(NCORR.EQ.2) CAPNOZ=10.0**(.00511*TIE+.944803)*
     112000.0*CORFAC
      IF(NCORR.EQ.1) CAPTUB=10.0**(.005629*TIE-.183775)*
     112000.0*CORFAC
      IF(NCORR.EQ.2) CAPTUB = 10.0**(.005291*TIE-.487330)*
     112000.0*CORFAC
      IF(NCORR.EQ.1) CAP = CAP*4.0/9.0
      PCAPN = CAP/CAPNOZ
      PCAPT = CAPPT/CAPTUB
      IF(PCAPN.LE.1.2) DPNOZ = 25.0*PCAPN**1.8384
      IF(PCAPN.GT.1.2) DPNOZ = 29.408*PCAPN**.954735
      DPTUBE = 10.0*PCAPT**1.81217
      DPSYS = PCC - PIE
C
C     THERMAL EXPANSION VALVE
C
C     DETERMINE  THE FLOW AREA COEFFICIENT 'CTXV' FOR EITHER
C     CONVENTIONAL OR CAPACITY CONTROLLED SYSTEMS
C
      IF(ICONTR.EQ.1) CTXV = C1*TSATE**2 + C2*TSATE + C3
      IF(ICONTR .EQ.2) CTXV = C1*TSATBP**2 + C2*TSATBP + C3
C
C     DETERMINE PRESSURE DROP THROUGH TXV 'DPTXV'(PSI)
C
      IF(NCORR.EQ.2) DPTXV = (XMR/CTXV)**2/RHO3
      IF(NCORR.EQ.1)DPTXV=(XMR*4.0/(9.0*1178.1))**2*71.236*
     1100.0/RHO3
C
C     DETERMINE THE ACTUAL PRESSURE DROP 'DPACT' (PSI)
C     WHICH WOULD EXIST WITH THE GIVEN FLOW RATE
C
      DPACT = DPTXV + DPNOZ + DPTUBE
      WRITE(5,500) NCORR,NSECT,SUPER,TROC,H3,PIC
      WRITE(5,510) HIC,H1CE,POC,PIE,TIE
      WRITE(5,520) DELPE,DELPC,DPTXV,DPNOZ,DPTUBE,DPLL,
     1DPSL,DPDL
      WRITE(5,530) CAP,CAPPT,CORFAC,CAPNOZ,CAPTUB,PCAPN,PCAPT
      WRITE(5,540) TSATE,TSATC ,DPSYS,DPACT,XMR,[illegible],POW
```

```
      WRITE(6,515) TSATE,TSATC,CUTOFF,DPSYS,DPACT
      CAPE = XMR*(H1CE - H3)
      CAPC = XMR*(HIC - H3)
      WRITE(5,520) CAPE,CAPC
C
C     DETERMINE THE MINIMUM SUCTION RISER VAPOR VELOCITY
C     'VSR' (FT/HR) REQUIRED FOR OIL ENTRAINMENT
C
      VSR=60.0*10.0**(-.005875*TSATE+.5*ALOG10(DSL)+3.4826)
C
C     DETERMINE THE MINIMUM DISCHARGE  RISER VAPOR VELOCITY
C     'VDR' (FT/HR) REQUIRED FOR OIL ENTRAINMENT
C
      VDR=60.0*10.0**(-.00315*TSATC +.5*ALOG10(DDL)+3.40)
      WRITE(5,575) VSR,VSRACT,VDR,VDRACT,CTXV
C
C     WRITE WARNING MESSAGE IF THERE IS INADEQUATE OIL RETURN
C
      IF(VSRACT.LT.VSR) WRITE(5,650) VSRACT,VSR
      IF(VDRACT.LT.VDR) WRITE(5,660) VDRACT,VDR
      IF(TIC.GE.280.0) WRITE(5,670)
C
C     CHECK THE ACTUAL PRESSURE DROP ACROSS THE SYSTEM AT
C     THE GIVEN FLOW RATE 'DPACT' WITH THE AVAILABLE PRESSURE
C     DROP BETWEEN CONDENSER AND EVAPORATOR 'DPSYS', TO SEE
C     IF A FLOW BALANCE ACTUALLY EXISTS
C
      IF(ABS(DPSYS-DPACT).LT.5.0) GO TO 130
      IF(DPSYS-DPACT) 95,130,90
   90 IF(I.EQ.1) DT = -DT
      IF(DT.LT.0.0) GO TO 100
      TSATC = TSATC - DT
      DT = DT/2.0
      GO TO 100
   95 IF(DT.LT.0.0) TSATC = TSATC - DT
      IF(DT.LT.0.0) DT = DT/2.0
  100 CONTINUE
C-----------------END CONDENSER TEMPERATURE LOOP------------
  130 CONTINUE
  490 FORMAT(I10)
  500 FORMAT('  NCORR =',I3,'    NSECT=',I5,'    SUPER='
     1,F10.3,'     TROC=',F10.3,'    H3=',F10.4,'    PIC=',F10.3)
  510 FORMAT('    HIC=',F10.4,'    H1OE=',F10.4,'    POC='
     1,F10.3,'    PIE=',F10.3,'    TIE=',F10.3)
  514 NAMELIST ' INPUT VARIABLES ARE ?', ( TSATE,TSATC,CUTOFF)
  515 FORMAT(' TSATE=',F7.2,' TSATC=',F7.2,' CUTOFF=',F4.2,
     1' DPSYS=',F7.2,' DPACT=',F7.2)
  520 FORMAT('   DELPE=',F8.2,'   DELPC=',F8.2,'   DPTXV='
     1,F8.2,'  DPNOZV=',F8.2,'  DPTUBE=',F8.2,'   DPLL=',F8.2,
     2'   DPSL=',F8.2,'   DPDL='F8.2)
```

```
530  FORMAT('   CAP=',F12.3,'   CAPPT=',F10.3,'   CORFAC='
    1,F6.2,'   CAPNCZ=',F12.4,'   CAPTUB=',F12.4,'   PCAPN='
    2,F10.4,'   PCAPT=',F10.4)
540  FORMAT('    TSATE=',F10.3,'    TSATCI=',F10.3,'    DPSYS='
    1,F12.3,' DPACT=',F10.3,'    XMR=',F12.4,'    TIC=',F10.3,
    2'    PCW=',F10.3)
550  FORMAT('    DOC=',F15.5,'    DZOC=',F15.5,'    DIC='
    1,F15.5,'    DZIC=',F15.5)
560  FORMAT('    DLLCO=',F15.5,'    XLEGLO=',F15.5,'    DLLIC='
    1,F15.5,'   XLEGLI=',F15.5)
570  FORMAT('    DSL =',F15.5,'    XLEGSL=',F15.5,'    DDL='
    1,F15.5,'    XLEGDL=',F15.5)
575  FORMAT('VSR =',E12.5,' FT/HR  VSRACT=',E12.5,
    1'   FT/HR    VDR=',E12.5,'  FT/HR    VDRACT=',E12.5,
    2'   FT/HR    CTXV  =',F10.4)
580  FORMAT('    CAPE =',F15.2,'    CAPC =',F15.2)
590  FORMAT('    TSATPF =',F10.4,'  DEG. F')
595  FORMAT(' ******FLASHING OCCURS IN LIQUID LINE******')
600  FORMAT(2F10.2,I10,2F10.2,I10,F10.2)
610  FORMAT(F10.4,2I10)
620  FORMAT(8F15.5)
630  FORMAT(4F15.5)
640  FORMAT(F10.4,I10,F10.4)
650  FORMAT('**********INADEQUATE OIL RETURN IN SUCTION'
    1,,' RISER  VSRACT=',F15.5,'   VSR=',F15.5,'*********')
660  FORMAT(' *************INADEQUATE OIL RETURN IN'
    1,,' DISCHARGE RISER  VDRACT=',F15.5,'    VDR=',F15.5,
    2'*********')
670  FORMAT(' *************COMPRESSOR DISCHARGE TEMPERATURE'
    1,,' IS EXCESSIVE**********')
120  END
```

APPENDIX E

DETAILS OF THE COMPRESSOR SIMULATION MODEL

Details of the compressor model discussed in section 2.3 are given in this section, accompanied by a detailed flow chart and program listings.

Discussion of the compressor model can be divided into three major sections:

1. Cylinder processes, valve, and manifold modeling
2. Motor cooling, friction, and suction-discharge heat transfer
3. Oil circulation effect on capacity

Cylinder Processes, Valve, and Manifold Modeling

Valve dynamics, manifold pressure pulsations, and cylinder/manifold interactions have been modeled as equivalent cylinder pressure overshoots or undershoots, and valve closing delays, as discussed in section 2.3. Cylinder processes on each complete compressor stroke are:

1. Intake of suction gas and mixing with residual
2. Compression
3. Discharge
4. Re-expansion of residual mass.

Before going into details of the cylinder processes it is necessary to determine the effect of suction valve closing delay on effective displacement volume, and of discharge valve closing delay on effective clearance volume:

The expression for cylinder volume as a function of crank angle, referencing from $\theta = 0^o$ at top dead center (TDC) is:

$$V_{cyl} = V_{min} + \frac{\pi D^2}{4} [R_r \{1 + \frac{R_c}{R_r} - \frac{R_c}{R_r} \cos\theta$$

$$- \cos [\sin^{-1} (\frac{R_c}{R_r} \sin\theta)]\}$$

Where:

R_c = center-to-center length of crankshaft throw

R_r = Center-to-center length of connecting rod

V_{min} = Clearance Volume

D = Cylinder diameter

Typically,

$$\frac{R_c}{R_r} \leq .25$$

Hence for angles near TDC $(\theta = 0^o)$ and BDC $(\theta = 180^o)$ we can approximate cylinder volume as:

$$V_{cyl} = V_{min} + V_D \frac{(1 - \cos\theta)}{2}$$

Where:

$V_D = \frac{\pi D^2}{4} (2 R_c)$ = displacement volume

The effective displacement volume accounting for suction valve closing delay thus becomes:

$$V_{D_{eff}} = V_{cyl_{max_{eff}}} - V_{min}$$

or

$$V_{D_{eff}} = V_D \frac{\{1 - \cos[(1 - \frac{\theta_s}{180})\ \pi]\}}{2}$$

Where

θ_s = Suction valve closing delay in degrees after bottom dead center (ABDC)

V_{max} = Maximum cylinder volume

Then:

$$\eta_D \equiv \text{Displacement efficiency} = \frac{V_{D_{eff}}}{V_D}$$

$$\eta_D = \frac{\{1 - \cos[(1 - \frac{\theta_s}{180})\ \pi]\}}{2}$$

The effective clearance volume accounting for discharge valve closing delay becomes:

$$V_{min_{eff}} = V_{min} + V_D \frac{\{1 - \cos[\theta_D \frac{\pi}{180}]\}}{2}$$

Where:

θ_D = Discharge valve closing delay in degrees after top dead center (ATDC)

We can now define the following volume ratios:

$$VR = \frac{V_{min}}{V_D}$$

$$VR_{eff} = \frac{V_{min}}{V_{D_{eff}}} = \frac{VR}{\eta_D}$$

$$VRMAX = \frac{V_{min}}{V_{max}} = \frac{VR}{1 + VR}$$

$$VRMAX_{eff} = \frac{V_{min}}{V_{max_{eff}}} = \frac{VR_{eff}}{1 + VR_{eff}}$$

$$VRMEV = \frac{V_{min_{eff}}}{V_{min}} = 1 + \frac{\{1 - \cos[\theta_D \frac{\pi}{180}]\}}{2\ VR}$$

Derivation of relations describing the four cylinder processes then proceeds as follows:

Intake Of Suction Gas And Mixing With Residual

$$\Delta m = m_{max_{eff}} - m_{res}$$

Where:

Δm = Mass pumped per stroke

m_{max} = Mass in cylinder at maximum effective volume

$_{eff}$ = $V_{max_{eff}} / \nu_{max}$

ν_{max} = Specific volume of gas in cylinder at maximum effective volume (ft^3/lbm)

m_{res} = Mass in cylinder at minimum effective volume (residual mass)

= $V_{\substack{min\\eff}}/\nu_{res}$

ν_{res} = Specific volume of gas in cylinder at minimum effective volume (ft^3/lbm)

Considering the intake process occuring at constant cylinder pressure, the first law of thermodynamics gives:

$$\cancel{Q}^{\,0} = W_I + \Sigma\, \cancel{m_e}^{\,0} h_e - \Sigma\, m_i\, h_i + \Sigma\, m_f\, u_f - \Sigma\, m_o\, u_o$$

or

$$W_I = m_i\, h_i + m_o\, u_o - m_f\, u_f$$

Where:

$m_i = \Delta m = m_f - m_o$

m_o = Original mass in cylinder

m_f = Final mass in cylinder

h_i = Enthalpy (Btu/lbm) of incoming suction gas

at $P_{\substack{cyl\\suct}} = P_{suct} - \Delta P_s$

ΔP_s = Equivalent cylinder pressure undershoot on intake

u_o = Internal energy (Btu/lbm) of original mass in cylinder

u_f = Internal energy (Btu/lbm) of final mass in cylinder

W_I = Work produced by gas on intake

Also:

$$W_I = \int_o^f P\,dV = P_{\substack{cyl\\suct}} \left(V_{\substack{max\\eff}} - V_{R_x}\right)$$

Where:

V_{R_x} = Cylinder volume at the end of re-expansion of residual mass.

Equating expressions for work we find after some manipulation:

$$0 = (h_i - h_{max}) + (VRMEV)\ (VRMAX_{eff})\ \left(\frac{\nu_{max}}{\nu_{res}}\right)\ (h_{R_x} - h_i)$$

Equation E-1

Where:

h_{max} = Enthalpy (Btu/lbm) of gas in cylinder at end of intake and mixing (at maximum effective volume)

h_{R_x} = Enthalpy (Btu/lbm) of residual mass in cylinder after re-expansion

To find the state at the end of the intake and mixing process, we guess T_{max} (and hence h_{max} and ν_{max}), and iterate until equation E-1 is satisfied.

Then

$$\frac{W_I}{\Delta m} = \frac{P_{\substack{cyl\\suct}} \left[\frac{1}{(VRMAX_{eff})(VRMEV)} - \frac{v_{R_X}}{v_{res}}\right] v_{res}}{\left[\frac{(v_{res})}{(v_{max})} \frac{1}{(VRMAX_{eff})(VRMEV)} - 1\right]}$$

For the initial calculation we do not know the state of the re-expansion gas. Hence, on the first calculation we ignore the effect of mixing and assume that the state at the end of intake is the same as the state of the incoming suction gas.

Compression

The first law of thermodynamics yields for the compression work 'W_c' , assuming no heat transfer:

$$W_c = m_{\substack{max\\eff}} (u_o - u_f)$$

Then, for isentropic compression:

$$\left(\frac{W_c}{V_{\substack{max\\eff}}}\right)_{is} = \frac{1}{v_{max}} [u_{max} - u_{\substack{cyl\\disc\\is}}]$$

Where:

$u_{\substack{cyl\\disc\\is}}$ = Internal energy (Btu/lbm) of gas in cylinder at the end of compression

And, for non-isentropic compression:

$$\left(\frac{W_c}{V_{max}}\right)_{\substack{eff\ non\\ is}} = \frac{\left(\frac{W_c}{V_{max}}\right)_{eff\ is}}{\eta_{is}}$$

Where

η_{is} = isentropic compression efficiency

The non-isentropic compression end state is found by guessing

$T_{\substack{cyl\\ disc}}$ (and hence $u_{\substack{cyl\\ disc\\ non-is}}$),

Evaluating

$$\left(\frac{W_c}{V_{max}}\right)_{eff\ guess} = \frac{1}{v_{max}}\left[u_{max} - u_{\substack{cyl\\ disc\\ non-is}}\right]$$

and iterating until

$$\left(\frac{W_c}{V_{max}}\right)_{eff\ guess} = \left(\frac{W_c}{V_{max}}\right)_{eff\ non-is}$$

Compression work can be found later from the expression:

$$\left(\frac{W_c}{\Delta m}\right)_{\substack{non\\ is}} = \frac{\left[u_{max} - u_{\substack{cyl\\ disc\\ non-is}}\right]}{\left[1 - (VRMEV)\ (VRMAX_{eff})\ \frac{v_{max}}{v_{res}}\right]}$$

Discharge

$$W_D = \int PdV = P_{\substack{cyl\\disc}} (V_{\substack{min\\eff}} - V_{\substack{cyl\\disc}})$$

Where:

W_D = Work required to discharge the gas from the cylinder

$P_{\substack{cyl\\disc}}$ = Cylinder pressure at end of compression (constant during discharge)

$= P_{disc} + \Delta P_D$

ΔP_D = Equivalent cylinder pressure overshoot on discharge

$V_{\substack{cyl\\disc}}$ = Cylinder volume at end of compression

And, since discharge is assumed to take place at constant pressure there is no change of state of the gas. That is, the state of the residual mass in the cylinder is the same as the state of the gas at the end of compression.

Hence:

$$\frac{W_D}{\Delta m} = - P_{\substack{cyl\\disc}} \; v_{\substack{cyl\\disc}}$$

$$= - P_{\substack{cyl\\disc}} \; v_{res}$$

Re-Expansion Of Residual Mass

As for compression, the first law of thermodynamics yields, assuming no heat transfer:

$$W_{R_x} = m_{res}\ (u_o - u_f)$$

and then

$$\left(\frac{W_{R_x}}{V_{min_{eff}}}\right)_{is} = \frac{1}{\nu_{res}}\left[u_{res} - u_{R_{x_{is}}}\right]$$

Where:

$u_{R_{x_{is}}}$ = Internal Energy (Btu/lbm) of gas in cylinder at end of re-expansion

Then

$$\left(\frac{W_{R_x}}{V_{min_{eff}}}\right)_{non\ is} = \eta_{is}\left(\frac{W_{R_x}}{V_{min_{eff}}}\right)_{is}$$

Where:

η_{is} = Isentropic expansion efficiency (assumed same as isentropic compression efficiency)

The non-isentropic re-expansion end state is found by guessing T_{R_x} (and hence $u_{R_{x_{non-is}}}$), evaluating

$$\left(\frac{W_{R_x}}{V_{min_{eff}}}\right)_{guess} = \frac{1}{\nu_{res}} [u_{res} - u_{R_{x_{non-is}}}]$$

and iterating until

$$\left(\frac{W_{R_x}}{V_{min}}\right)_{eff\ guess} = \left(\frac{W_{R_x}}{V_{min_{eff}}}\right)_{non-is}$$

Re-expansion work can be found later from the expression:

$$\left(\frac{W_{R_x}}{\Delta m}\right)_{non-is} = \frac{[u_{res} - u_{R_{x_{non-is}}}]}{[(\frac{\nu_{res}}{\nu_{max}}) \frac{1}{(VRMAX_{eff})(VRMEV)} - 1]}$$

After completing one full cycle of calculations, an estimate for the state of the re-expansion gas is available. The entire cycle is repeated until the correct valves for the re-expansion state, and for the end state after intake and mixing are determined.

For more details, see the flow chart in Figure E-2 and comments in the program listing for subroutine 'COMP' at the end of this section.

Motor Cooling, Friction, and Suction-Discharge Heat Transfer

An iterative solution is required to properly determine the amount of heat given to the suction gas by internal friction, motor waste heat, and suction-discharge heat transfer. A first guess for the temperature of the suction gas entering the cylinder is made and, using the results from the cylinder process portion of the model, the resulting refrigerant mass flow rate and motor power are calculated. The amount of waste heat due to friction is next calculated, followed by determination of actual motor speed, motor efficiency, and motor waste heat. The assumed mechanical efficiency 'η_{mech}' of the compressor, accounting for friction, has been assumed to be 96% in all of the studies done to date. Curves showing the assumed variation of motor speed 'RPM' and motor efficiency 'η_{motor}' with load on the motor are given in Figures 2.3-4 and 2.3-5 respectively. The latter curves are fairly respresentative of squirrel-cage induction motors in the 3 to 10 horsepower range[1]. Most of the heat generated by friction and motor inefficiency in hermetic and semi-hermetic compressors is given to the suction gas. A small portion, however, is lost to the ambient by convection and radiation from the compressor shell.

Next, an estimate of heat transfer between suction and discharge gas is made, using an approximate method to be discussed shortly.

After estimating friction, motor cooling, and heat transfer effects on the suction gas, a new estimate of the state of the suction gas entering the cylinder can be made and the entire process repeated until the correct state of gas entering the cylinder is found. The procedure is outlined below, and is also shown on the flow chart in Figure E-2.

1. Assume temperature of suction gas entering cylinder
2. Use cylinder process model to determine total work input to the gas 'W_{tot}', mass pumped per stroke 'Δm', and temperature of discharge gas 'T_{disc}'
3. Determine work input to compressor 'W_{comp}'

$$W_{comp} = \frac{W_{tot}}{\eta_{mech}} \qquad (\frac{Btu}{lbm})$$

4. Iterate on motor speed

→ Assume Motor Speed

$\dot{m} = (\Delta m)\ (N_{cyl})\ (RPM)$ (Total Mass Flow)

$\mathcal{P}_{comp} = (W_{comp})\ \dot{m}$ (Power Output of Motor)

$$\%\ \text{Load on Motor} = \frac{\mathcal{P}_{comp}}{\mathcal{P}_{max}} \qquad (\mathcal{P}_{max} = \text{Maximum Power Output of Motor})$$

RPM = f (% Load) (See Figure <u>2.3-4</u>)

Repeat until correct RPM is found

5. η_{motor} = f (% Load) (See Figure <u>2.3-5</u> and listing for subroutine 'EFFM' at end of this section)

6. Determine waste heat 'QMC' given to suction gas by friction and motor cooling

$$QMC = (\%\ \text{Motor Cooling})\ \left[\frac{W_{comp}}{\eta_{motor}} - W_{tot}\right]$$

(% Motor cooling = Percent of waste heat which is absorbed by the suction gas)

7. Determine temperature of suction gas after absorbing waste heat

8. Determining suction-discharge heat transfer 'QHT' as described shortly

9. Determine a new guess for the state of the suction gas entering the cylinder using

$$h_{new} = h_1 + QMC + QHT$$

where

h_{new} = enthalpy of gas entering cylinder

h_1 = enthalpy of gas leaving evaporator

10. Repeat steps 1-9 until correct amounts of motor cooling and heat transfer are found

For more information see the flow chart in Figure E-2 and comments in the program listing for subroutine 'COMP' at the end of this section.

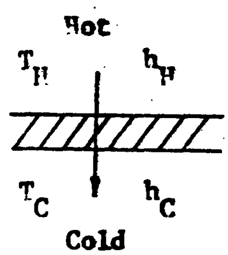
Approximate Suction-Discharge Manifold Heat transfer

As mentioned in section 2.3, some compressor designs have negligible amounts of suction-discharge heat transfer, while others have large amounts. Presented here is a very rough method of estimating suction-discharge heat transfer, which is designed to permit study of the variation of suction gas superheat due to heat transfer with the discharge gas, while pumping across different pressure ratios.

Suction-discharge heat transfer is calculated by subrountine 'HEAT' so that, if more correct heat transfer information for a particular compressor is available, subroutine 'HEAT' may be changed without affecting the remainder of the compressor model.

The present suction-discharge heat transfer model is as follows:

Let us first assume that the heat transfer occurs across a flat metal surface of negligible heat transfer resistance.



Then

$$q = U\, A_{ht}\, \Delta T_{1m}$$

$$U = \frac{1}{\frac{1}{h_c} + \frac{1}{h_H}}$$

Where:

q = Heat transfer rate per manifold

A_{ht} = Heat transfer area per manifold

U = Overall heat transfer coefficienct

h = Heat transfer coefficient

$$\Delta T_{lm} = \frac{(T_{H_{in}} - T_{c_{out}}) - (T_{H_{out}} - T_{c_{in}})}{\ln\left[\dfrac{(T_{H_{in}} - T_{c_{out}})}{(T_{H_{out}} - T_{c_{in}})}\right]} \quad \text{(Assuming counter flow)}$$

Using the following relation for the heat transfer coefficients in the manifolds[2]

$$Nu = 1.48\ Re^{.63}\ Pr^{.6}$$

and assuming

$$D_{eq_{disc}} \simeq .75\ D_{eq_{suct}}$$

$$PER \simeq 1.5\ PER_{disc}$$

We get, after some manipulation:

$$q = \frac{(\dot{m})^{.63}\ (EQAREA)\ \Delta T_{lm}}{F_s + F_D}$$

Where:

Nu = Nusselt number = $\dfrac{h\ D_{eq}}{k}$

Re = Reynolds number = $\dfrac{\rho V\ D_{eq}}{\mu}$

Pr = Prantl number = $\dfrac{\mu\ C_p}{k}$

k = Thermal conductivity of fluid

μ = Viscosity of fluid

C_p = Specific heat at constant pressure of fluid

ρ = Density of Fluid

V = Velocity of fluid

D_{eq} = Equivalent diameter of flow passage

$$= \frac{4\ A_{xs}}{PER}$$

A_{xs} = Cross-sectional or flow area of flow passage

PER = Wetted perimeter of flow passage

$\dot{m}$ = Mass flow rate per manifold

$$F_s = \frac{4}{(1.48)(k_s)(Pr_s^{.6})(\frac{4}{\mu_s})^{.63}}$$

$$F_D = \frac{(4)\ (.75)}{(1.48)\ (k_D)\ (Pr_D^{.6})\left[\frac{(4)\ (1.5)}{\mu_D}\right]^{.63}}$$

s = Subscript indicating suction gas

D = Subscript indicating discharge gas

$$EQAREA = \frac{A_{ht}}{A_{xs_s}}\ (PER_s^{.37})$$

Note that the equivalent area term 'EQAREA' does <u>not</u> have units of <u>AREA</u>.

The heat transfer between suction and discharge gases is then found by iteration as follows:

1. Guess $T_{\substack{disc\\out}}$

2. $T_{\substack{suct\\out}} = T_{\substack{suct\\in}} + \frac{C_{p_D}}{C_{p_s}} \left(T_{\substack{disc\\in}} - T_{\substack{disc\\out}}\right)$

3. $$\Delta T_{lm} = \frac{(T_{D_{in}} - T_{s_{out}}) - (T_{D_{out}} - T_{s_{in}})}{\ln\left[\frac{(T_{D_{in}} - T_{s_{out}})}{(T_{D_{out}} - T_{s_{in}})}\right]}$$

4. $$T_{s_{avg}} = \frac{T_{s_{in}} + T_{s_{out}}}{2} \qquad T_{D_{avg}} = \frac{T_{D_{in}} + T_{D_{out}}}{2}$$

5. Evaluate properties μ , k , and C_p at $T_{s_{avg}}$ and $T_{D_{avg}}$

6. $$q^* = \frac{(\dot{m})^{.63} \text{ (EQAREA) } \Delta T_{lm}}{F_s + F_D}$$

7. Repeat 1 - 6 until $q = q^*$

8. $QHT = \frac{q}{\dot{m}}$

For more information, see comments in the program listing for subroutine 'HEAT' given at the end of this section.

Oil Circulation Effect on Capacity

The effect of oil circulation on capacity can be determined once the refrigerant-oil solubility characteristics, as shown in Appendix F, are known. As illustrated in Figure E-1, the capacity of a compressor is defined as the evaporator capacity 'Q_e' of a refrigeration system:

$$Q_e = \dot{m} \, (h_{1_m} - h_{3_m})$$

Where:

h_{1_m} = Enthalpy of total mixture leaving evaporator

h_{3_m} = Enthalpy of total mixture entering evaporator

$\dot{m}$ = Total mixture mass flow rate

The enthalpy entering the evaporator can be defined as: (from Cooper[3])

$$h_{3_m} = (x)\ (h_{3_{oil}}) + (1 - x)\ (h_{3_{ref}})$$

Where:

$h_{3_{ref}}$ = Enthalpy of pure refrigerant entering evaporator

$h_{3_{oil}}$ = Enthalpy of oil entering evaporator

x = Weight percent of oil circulating in system

$$\left[\frac{\text{lbm oil}}{(\text{lbm oil} + \text{lbm ref})}\right]$$

The enthalpy of the total mixture leaving the evaporator can be defined as:

$$h_{1_m} = (z)\ (h_z) + (1 - z)\ (h_{1_{ref\ vapor}})$$

Where:

$h_{1_{ref\ vapor}}$ = Enthalpy of pure refrigerant vapor at P_{sat} and T_{super} leaving the evaporator

h_z = Enthalpy of liquid mixture leaving evaporator at T_{super}

z = Weight percent of liquid leaving evaporator

$$\left(\frac{\text{lbm liquid}}{\text{lbm total mixture}}\right)$$

The enthalpy of the liquid leaving the evaporator can be defined as:

$$h_z = (w)\,(h_{1_{\substack{ref\\liq}}}) + (1 - w)\,(h_{1_{oil}})$$

Where:

$h_{1_{oil}}$ = Enthalpy of oil leaving evaporator at T_{super}

$h_{1_{\substack{ref\\liq}}}$ = Enthalpy of refrigerant liquid at T_{super} leaving the evaporator

w = Weight percent of refrigerant in the liquid leaving the evaporator

$$\left[\frac{\text{lbm ref liq}}{(\text{lbm oil} + \text{lbm ref liq})}\right]$$

Then, from continuity we find:

$$z = \frac{x}{1 - w}$$

The enthalpy of a typical refrigeration compressor oil, referenced to a base at $-40^{o}F$, as are the refrigerants, is:

$$h_{oil} = (.403)\,T + (.00025)\,T^2 + 15.75$$

Where:

$$T = \text{Temperature } (^{o}F)$$

$$h_{oil} = \text{Btu/lbm}$$

The density of a typical refrigeration oil is about:

$$\rho_{oil} = 57.6 \text{ lbm/ft}^3$$

The work of compression of the oil 'W_{oil}' is hence:

$$W_{oil} = \frac{\Delta P}{\rho_{oil}}$$

Where

$$\Delta P = P_{suct} - P_{disc}$$

Finally, the total power '$\mathcal{P}$' required by the compressor is

$$\mathcal{P} = \mathcal{P}_{ref} + \mathcal{P}_{oil}$$

Where:

$\mathcal{P}_{ref}$ = Power required to compress refrigerant

$\mathcal{P}_{oil} = (\dot{m}_{ref}) (\frac{x}{1 - x}) W_{oil}$ = Power required to compress oil

$\dot{m}_{ref}$ = Total mass flow of pure refrigerant

For more information, see comments in the listings for subroutines 'OIL' and 'COMP' at the end of this section.

Modeling Early Suction-Valve Closing

The early suction-valve cut-off method of compressor capacity control is modeled using the same model as for a conventional compressor with two exceptions. First, the effect of late suction valve closing is eliminated when cut-off control is in use. Second, there is an expansion of the gas in the cylinder after the suction valve is closed.

Only one additional input is required to model early suction-valve cut-off control:

$$CUTOFF \equiv 1 - \frac{V_{cut} - V_{min}}{V_D}$$

Where:

V_{out} = Total volume of cylinder when suction valve is closed.

CUTOFF = Indicates the amount of capacity reduction, but is not synonomous with % flow reduction

Modifications to the compressor model are as follows:

First let us define the following volume ratio

$$VRCUT = \frac{V_{cut}}{V_D} = 1 - CUTOFF + VR$$

Let us also define the state in the cylinder at cut-off as the end state for the intake-mixing-process, and give it the subscript 'CUT'. We then have, from the first law of thermodynamics on the expansion of the gas after cut-off:

$$W_{ex} = m\ (u_o - u_f)$$

Where:

W_{ex} = work produced by adiabatic expansion of gas

$$m = m_{max} = \frac{V_{max}}{\mathcal{V}_{max}} = \frac{V_{cut}}{\mathcal{V}_{cut}}$$

Then

$$\mathcal{V}_{max} = [\frac{1 + VR}{VRCUT}]\ \mathcal{V}_{cut}$$

and

$$W_{ex} = \frac{V_{max}}{\mathcal{V}_{max}}\ (u_{cut} - u_{max})$$

Where

u_{cut} = Internal energy of the gas in the cylinder at the beginning of expansion after cut-off.

$\mathcal{V}_{cut}$ = Specific volume of the gas at the beginning of expansion

First, assume isentropic expansion and evaluate:

$$(\frac{W_{ex}}{m})_{is} = u_{cut} - u_{max_{is}}$$

The non-isentropic work of expansion is then

$$(\frac{W_{ex}}{m})_{non-is} = \eta_{is}\ (\frac{W_{ex}}{m})_{is}$$

Where

η_{is} = Isentropic compression and expansion efficiency

To find the non-isentropic end state 'max' for expansion of cut-off gas, we guess T_{max} (and hence $u_{max_{non\ is}}$) evaluate

$$\left(\frac{W_{ex}}{m}\right)_{guess} = u_{cut} - u_{max_{non-is}}$$

and iterate until

$$\left(\frac{W_{ex}}{m}\right)_{guess} = \left(\frac{W_{ex}}{m}\right)_{non-is}$$

The non-isentropic work of expansion can be evaluated later from the expression:

$$\frac{W_{ex}}{\Delta m} = \frac{(u_{cut} - u_{max_{non-is}})}{[1 - (VRMAX)(VRMEV)(\frac{\mathcal{V}_{max}}{\mathcal{V}_{res}})]}$$

Unfortunately, implementing the expansion of cut-off gas calculation into the simulation program is not as simple as presented above. The superheat of the suction gas is always great enough to prevent expansion of the suction gas into the saturation region, because of motor cooling, internal heat-transfer, and mixing with residual gas. However, during the first few iterations on the effect of motor cooling, mixing, and the like, the superheat has not been added in, and expansion of the cut-off gas does go into the saturation region. To prevent an abundance of error messages and possibly fatal errors (which would terminate the calculation), expansion into the saturation region has been accounted for.

The relations describing expansion into the saturation region are as follows:

Isentropic expansion:

$$S_{cut} = S_{max}$$

Guess various T_{sat}'s and evaluate

$$\text{Quality} = \frac{(\nu_{max} - \nu_f)}{(\nu_g - \nu_f)}$$

and

$$\text{Quality}^* = \frac{S_{cut} - S_f}{S_g - S_f}$$

Where

ν = Specific volume

S = Entropy

f = Subscript indicating saturated liquid

g = Subscript indicating saturated vapor

If the two qualities are contradictory, i.e. Quality < 1 and Quality* > 1, then the isentropic expansion is not into the saturation region. If, however, the two qualities can be made equal at some T_{sat}, the expansion was into the saturation region.

Evaluate:

$$u_{\substack{max\\non-is}} = u_{cut} - \eta_{is}\,[u_{cut} - h_{max_{is}} + P_{max_{is}}\,\nu_{max}]$$

and repeat the iteration of T_{sat} for the non-isentropic expansion:

Guess T_{sat}

Evaluate $u_g = h_g - P_{sat} v_g$

If $u_{max_{non-is}} \geq u_g$, then the non-isentropic expansion is not into the saturation region. If, however, the internal energy 'u_{mix}' of the guessed saturated mixture can be made equal to $u_{max_{non-is}}$ at some T_{sat}, then the expansion went into the saturation region.

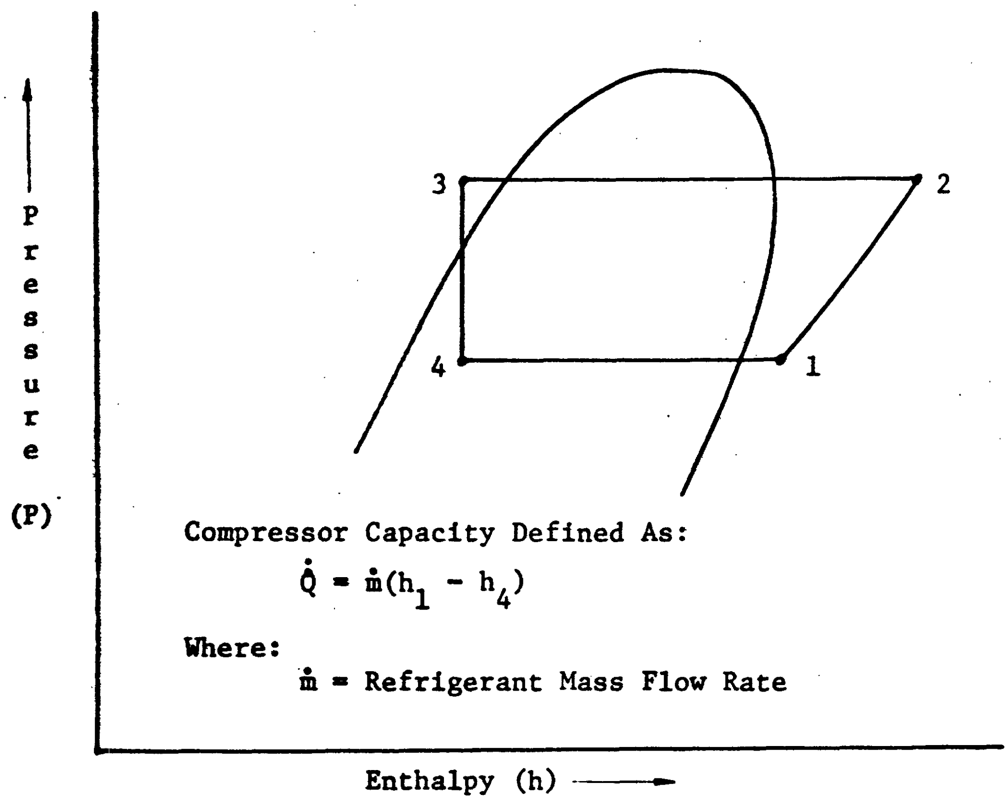
As before, the work of expansion $\frac{W_{ex}}{\Delta m}$, can be evaluated later. For more details see the program flow chart in Figure E-2 and comments in the program listing for subroutine 'COMP' at the end of this section.

References

1. Handbook of Air Conditioning System Design (New York: McGraw-Hill Inc., 1965) pg. 8-21.

2. Hughes, J.M., Qvale, E.B., Pearson, J.T., "Experimental Investigation of Some Thermodynamic Aspects of Refrigerating Compressors", Proceedings of the 1972 Purdue Compressor Technology Conference (Purdue Research Foundation, 1972) pg. 516-520.

3. Cooper, K. W., and Mount, A. G., "Oil Circulation - Its Effects on Compressor Capacity, Theory and Experiment", Proceedings of the 1972 Purdue Compressor Technology Conference (Purdue Research Foundation, 1972) pg. 52-59.

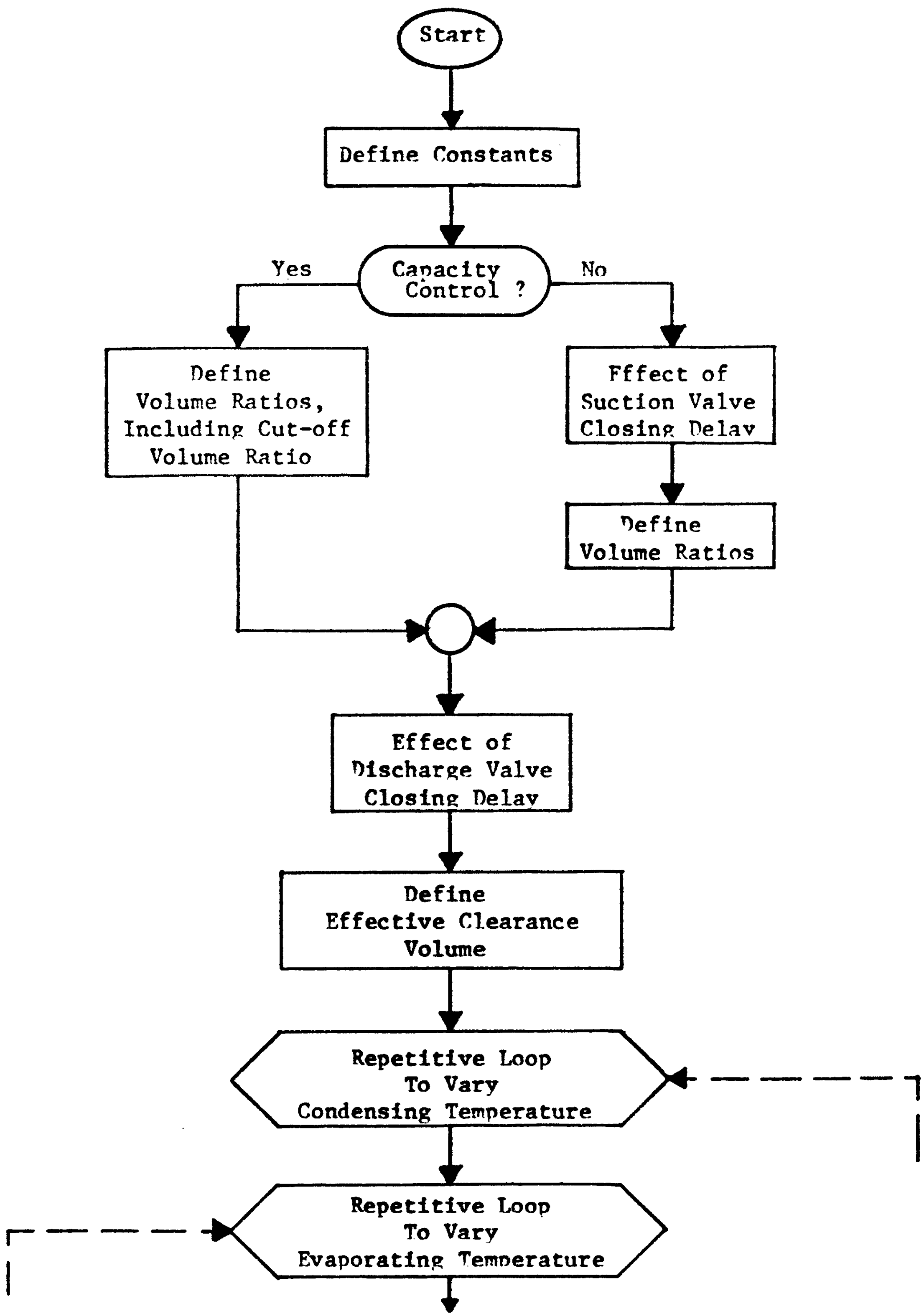


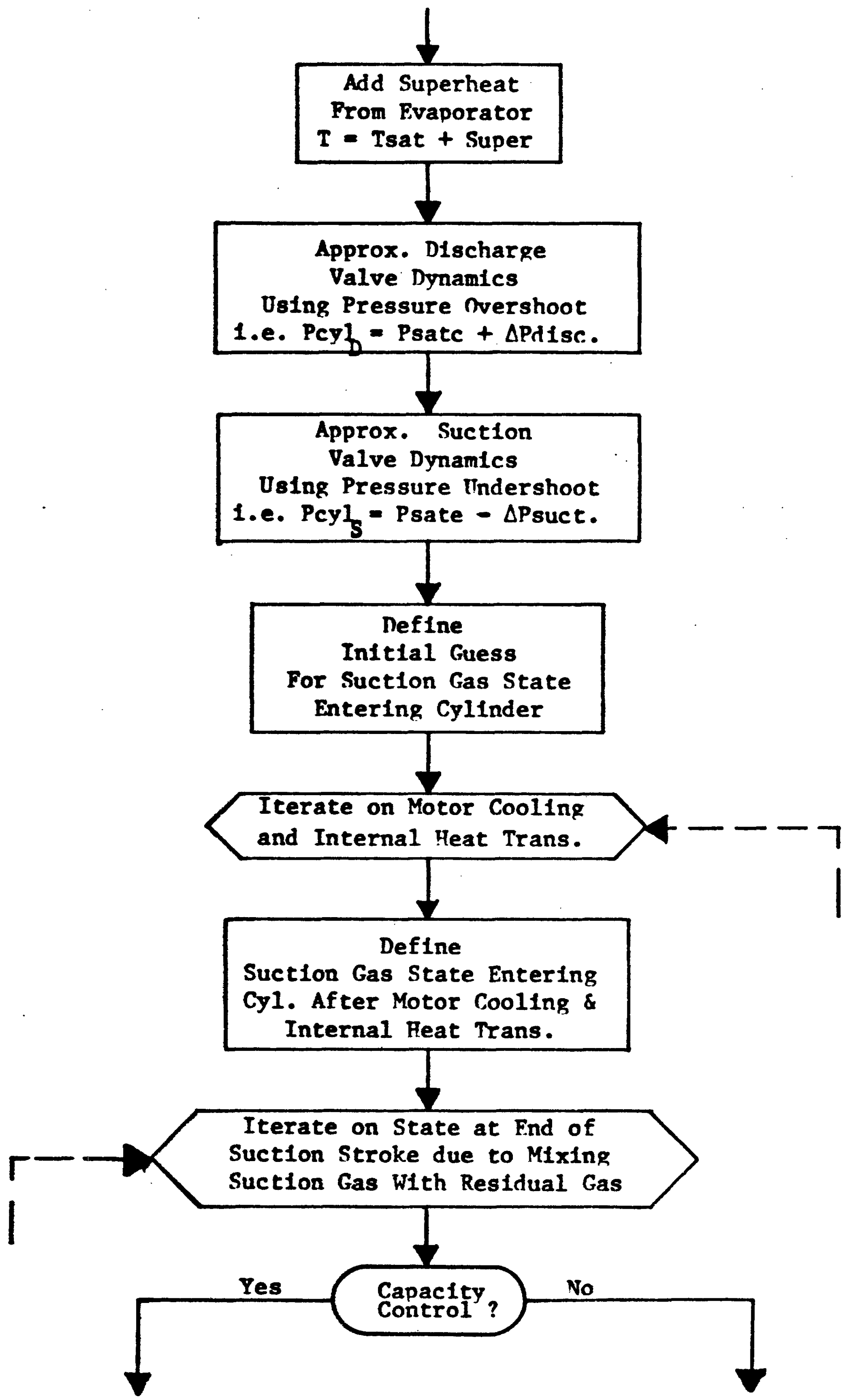
COMPRESSOR CAPACITY DEFINITION

FIGURE E-1

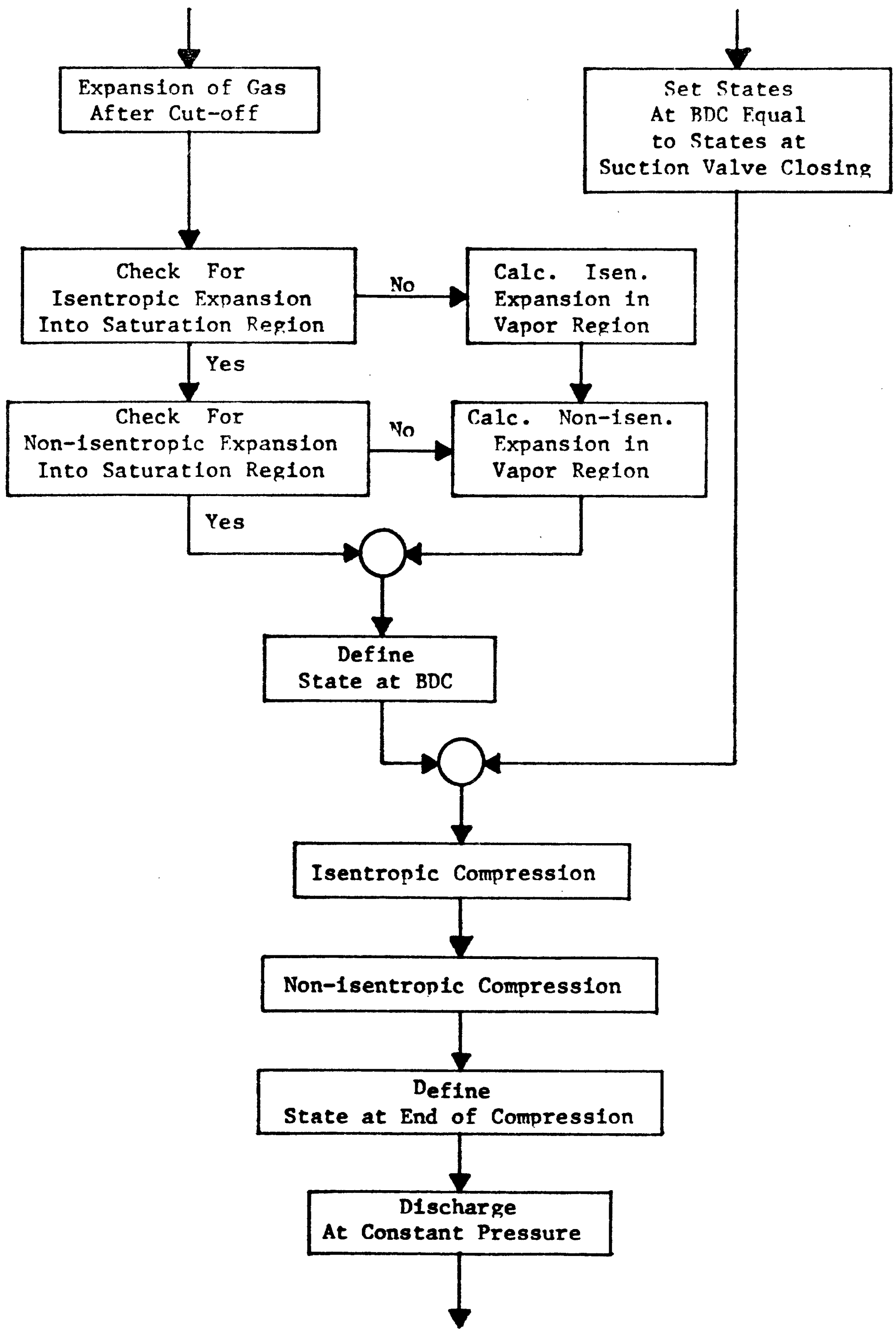
FIGURE E-2

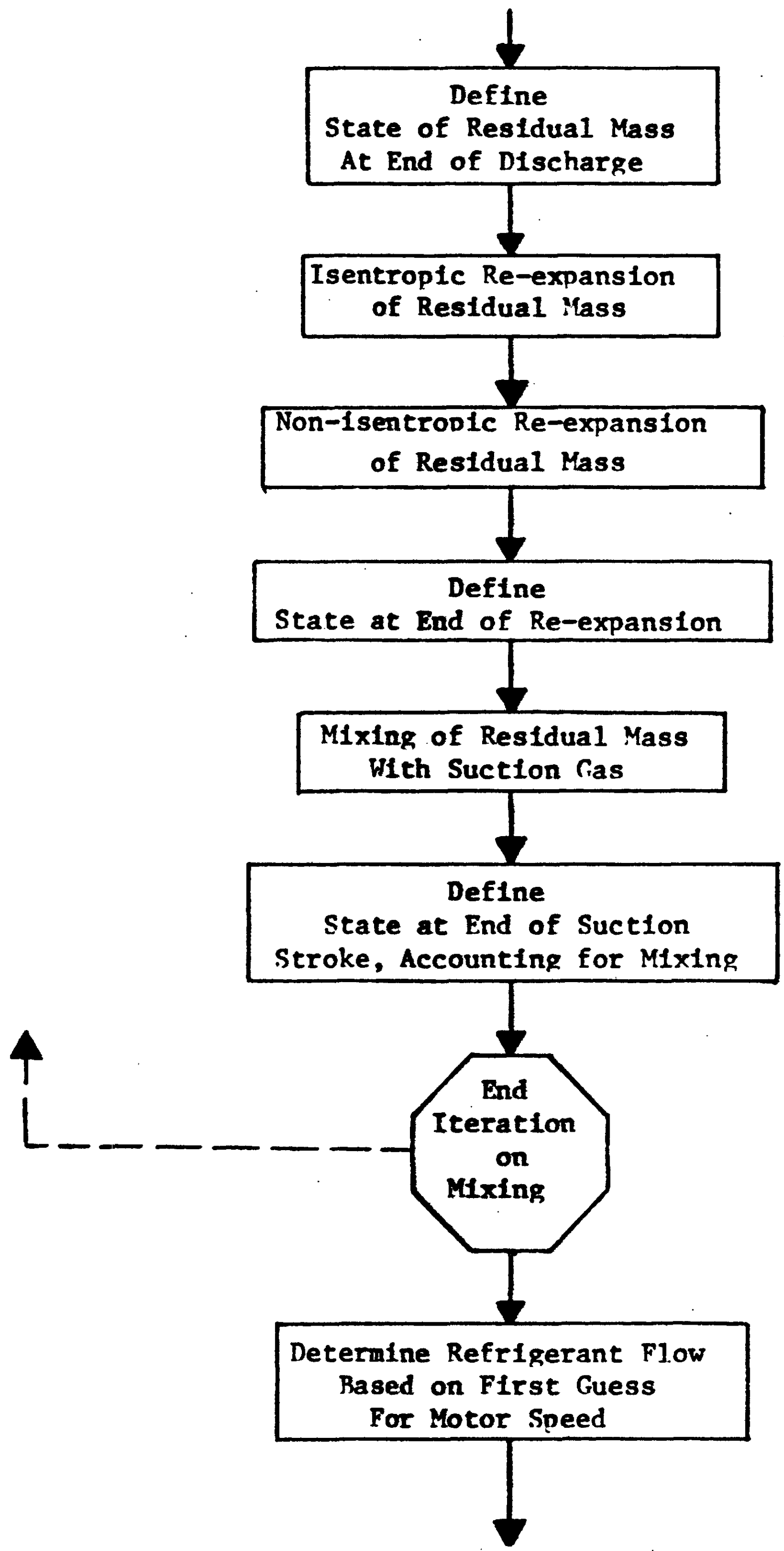
FLOW CHART FOR COMPRESSOR MODEL

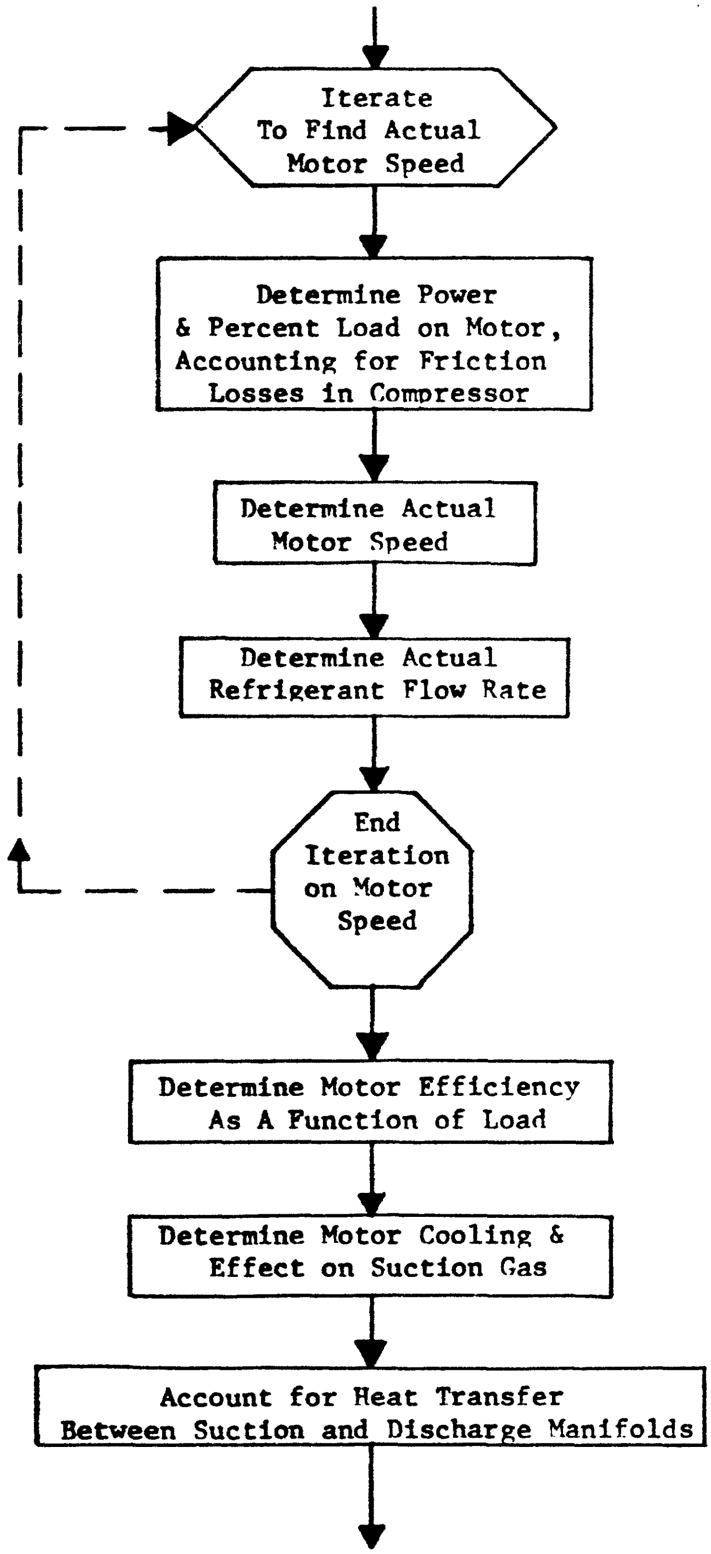


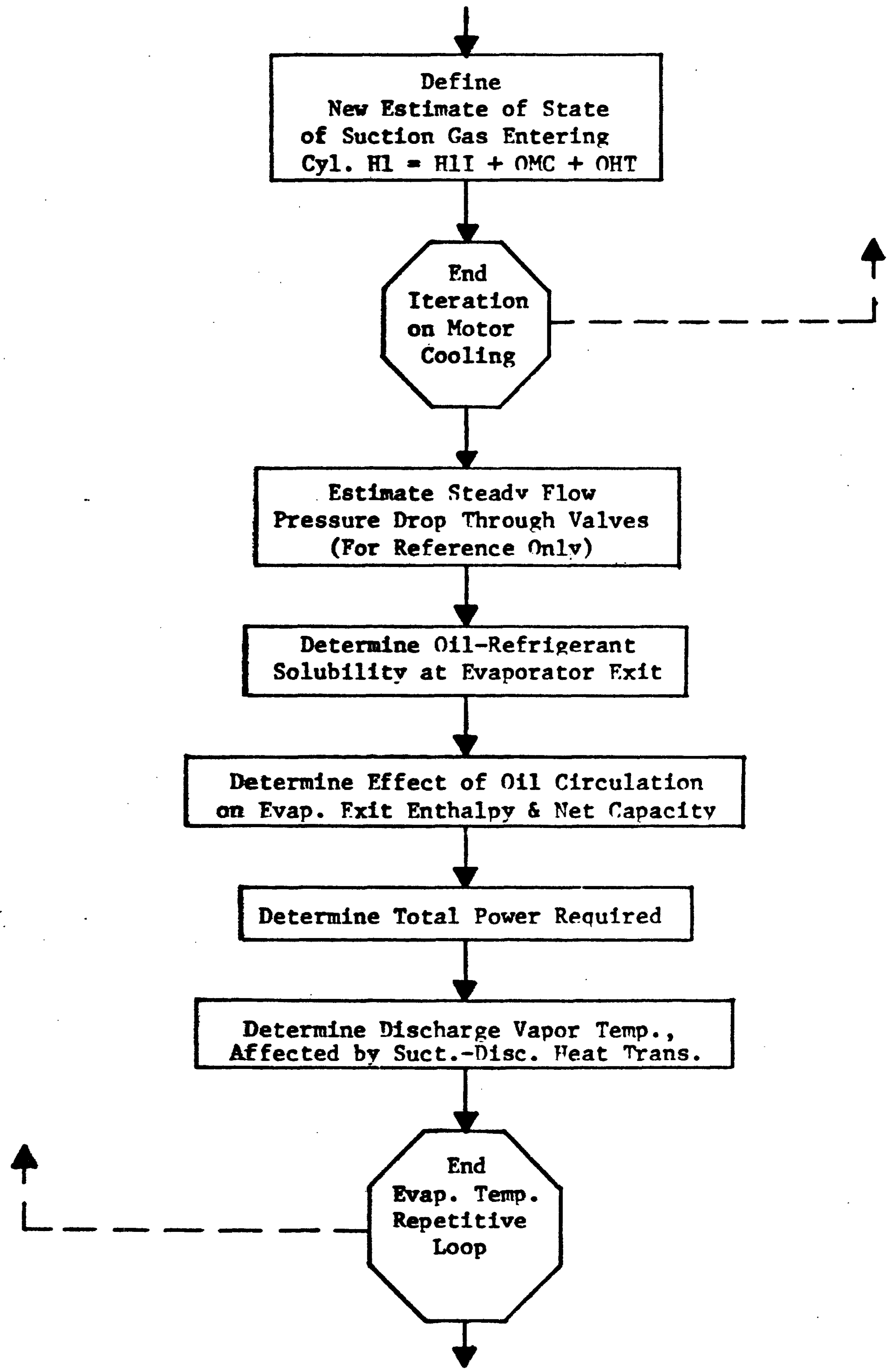


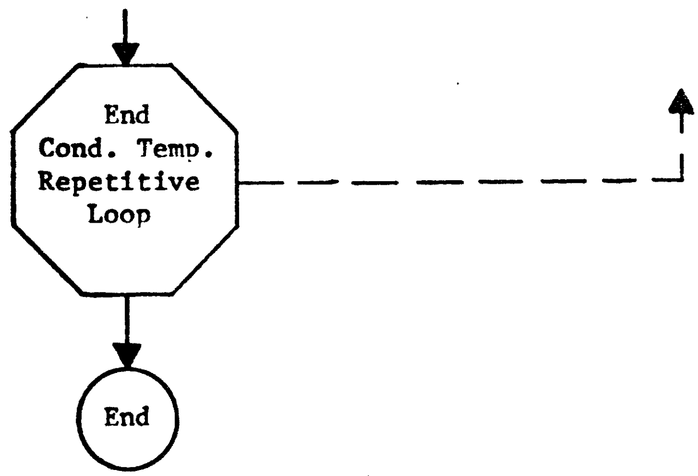
Add Superheat
From Evaporator
T = Tsat + Super
Approx. Discharge
Valve Dynamics
Using Pressure Overshoot
i.e. Pcyl D = Psatc + ΔPdisc.
Approx. Suction
Valve Dynamics
Using Pressure Undershoot
i.e. Pcyl S = Psate - ΔPsuct.
Define
Initial Guess
For Suction Gas State
Entering Cylinder
Iterate on Motor Cooling
and Internal Heat Trans.
Define
Suction Gas State Entering
Cyl. After Motor Cooling &
Internal Heat Trans.
Iterate on State at End of
Suction Stroke due to Mixing
Suction Gas With Residual Gas
Yes
Capacity
Control ?
No


Expansion of Gas After Cut-off
Set States At BDC Equal to States at Suction Valve Closing
Check For Isentropic Expansion Into Saturation Region
No
Calc. Isen. Expansion in Vapor Region
Yes
Check For Non-isentropic Expansion Into Saturation Region
No
Calc. Non-isen. Expansion in Vapor Region
Yes
Define State at BDC
Isentropic Compression
Non-isentropic Compression
Define State at End of Compression
Discharge At Constant Pressure


Define
State of Residual Mass
At End of Discharge
Isentropic Re-expansion
of Residual Mass
Non-isentropic Re-expansion
of Residual Mass
Define
State at End of Re-expansion
Mixing of Residual Mass
With Suction Gas
Define
State at End of Suction
Stroke, Accounting for Mixing
End
Iteration
on
Mixing
Determine Refrigerant Flow
Based on First Guess
For Motor Speed


Iterate To Find Actual Motor Speed
Determine Power & Percent Load on Motor, Accounting for Friction Losses in Compressor
Determine Actual Motor Speed
Determine Actual Refrigerant Flow Rate
End Iteration on Motor Speed
Determine Motor Efficiency As A Function of Load
Determine Motor Cooling & Effect on Suction Gas
Account for Heat Transfer Between Suction and Discharge Manifolds


Define
New Estimate of State
of Suction Gas Entering
Cyl. H1 = H1I + QMC + QHT
End
Iteration
on Motor
Cooling
Estimate Steady Flow
Pressure Drop Through Valves
(For Reference Only)
Determine Oil-Refrigerant
Solubility at Evaporator Exit
Determine Effect of Oil Circulation
on Evap. Exit Enthalpy & Net Capacity
Determine Total Power Required
Determine Discharge Vapor Temp.,
Affected by Suct.-Disc. Heat Trans.
End
Evap. Temp.
Repetitive
Loop


End
Cond. Temp.
Repetitive
Loop
End

```
      SUBROUTINE COMP(NR,TSATEI,DTE,NE,TSATCI,DTC,NC,SUPER,
     1ICONTR,CUTOFF)
C
C      PURPOSE
C         TO SIMULATE HERMETICALLY SEALED REFRIGERATION
C         COMPRESSORS
C         THIS PROGRAM PREDICTS SUCH THINGS AS MASS FLOW
C         RATE, DISCHARGE TEMPERATURE, POWER CONSUMPTION,
C         AND CERTAIN UNDESIRABLE OPERATING CONDITIONS SUCH
C         AS INADEQUATE MOTOR COOLING, EXCESSIVE DISCHARGE
C         TEMPERATURE,EXCESSIVE POWER CONSUMPTION, AND LOW
C         MOTOR EFFICIENCY
C    NOTE:  THIS PROGRAM CAN SIMULATE EITHER CONVENTIONAL
C         COMPRESSORS OR VARIABLE CAPACITY COMPRESSORS
C         (ACHIEVED BY EARLY SUCTION VALVE CLOSING)
C
C      GENERAL DESCRIPTION
C         THE COMPRESSOR STROKE IS SEPARATED INTO FOUR OR FIVE
C         SEPARATE PROCESSES: RE-EXPANSION OF RESIDUAL MASS,
C         INTAKE OF SUCTION GAS AND MIXING WITH RESIDUAL,
C         EXPANSION OF GAS IN CYLINDER AFTER EARLY SUCTION
C         VALVE CLOSING (ONLY ON CAPACITY CONTROL CASES)
C         COMPRESSION OF GAS IN CYLINDER TO HIGH PRESSURE
C         DISCHARGE OF GAS IN CYLINDER AT CONSTANT PRESSURE
C
C      EXPLICIT INPUT PARAMETERS
C         NR     -  NUMBER OF REFRIGERANT (12,22, OR 502)
C         TSATE-    SATURATION TEMPERATURE AT EXIT OF EVAPORATOR
C                   OR,IF SUCTION LINE PRESSURE DROP IS EXCESSIVE,
C                   SATURATION TEMP.CORRESPONDING TO THE PRES.
C                   ENTERING THE COMPRESSOR (F)
C         DTE    -  TEMPERATURE INCREMENT FOR EVAP. (F)
C         NE     -  NUMBER OF EVAP. TEMPS. EXAMINED
C         TSATC-    SATURATION TEMP.IN CONDENSER (F), OR,
C                   IF DISCHARGE LINE PRESSURE DROP IS EXCESSIVE,
C                   SAT. TEMP CORRESPONDING TO PRESSURE AT
C                   EXIT OF COMPRESSOR
C         DTC    -  TEMPERATURE INCREMENT FOR COND.(F)
C         NC     -  NUMBER OF CONDENSER TEMPS. EXAMINED
C         SUPER-    SUPERHEAT OF VAPOR ENTERING THE COMPRESSOR
C                   ABOVE THE SATURATION TEMP. AT THE ENTERING
C                   PRESSURE (F)
C         ICONTR-   CONTROL INDICATOR
C                   'ICONTR' = 1 MEANS CONVENTIONAL COMPRESSOR
C                   'ICONTR' = 2 MEANS CAPACITY CONTROLLED
C                   COMPRESSOR
C         CUTOFF-   A PARAMETER USED TO INDICATE THE AMOUNT OF
C                   CAPACITY CONTROL USED - IT IS DEFINED AS
C                   'CUTOFF' = 1.0 - (VOL.CUT - VOL.MIN)/VOL.DISP.
```

```
C                   WHERE VCL.CUT IS THE VOLUME SWEPT BY THE
C                   PISTON BEFORE THE SUCTION VALVE IS CLOSED
C                   VCL.MIN IS THE CLEARANCE VOLUME
C                   VCL.DISP. IS THE DISPLACEMENT VOLUME
C                   (ALL PER CYLINDER)
C
C       INPUT PARAMETERS FROM COMMON
C          NCYL  =  NUMBER OF COMPRESSOR CYLINDERS
C          VR    =  CLEARANCE VOLUME RATIO, DEFINED AS
C                   'VR' = VOL.MIN/VOL.DISP.
C          VD    =  DISPLACEMENT VOLUME PER CYLINDER (VOLUME
C                   SWEPT BY PISTON) (CU FT)
C          SYNC  =  SYNCHRONOUS MOTOR SPEED (RPM)
C          RPM   =  INITIAL GUESS FOR ACTUAL MOTOR SPEED (RPM)
C          EFFIS=   ISENTROPIC EFFICIENCY OF THE COMPRESSION AND
C                   EXPANSION PORTIONS OF THE CYLINDER
C                   PROCESSES
C          DPDI  =  EQUIVALENT PRESSURE DROP ACROSS DISCHARGE
C                   VALVE TO ACCOUNT FOR VALVE DYNAMICS AND
C                   FLOW LOSSES (PSI)
C          DPS   =  EQUIVALENT PRESSURE DROP ACROSS SUCTION
C                   VALVE TO ACCOUNT FOR VALVE DYNAMICS AND
C                   FLOW LOSSES (PSI)
C          DPFRAC=  (NOT USED HERE)
C          SDELAY=  SUCTION VALVE CLOSING DELAY (DEGREES AFTER
C                   BOTTOM DEAD CENTER)
C          PMC   =  PERCENT OF COMPRESSOR MOTOR HEAT WHICH IS
C                   REMOVED BY THE SUCTION GAS (THE REMAINDER
C                   IS LOST BY CONVECTION TO THE AMBIENT)
C          PHT   =  (NOT USED HERE)
C          PHTD  =  (NOT USED HERE)
C          EAD   =  (NOT USED HERE)
C          EAS   =  (NOT USED HERE)
C          POWMAX=  MAXIMUM POWER OUTPUT OF COMPRESSOR MOTOR
C                   (KW) WHEN OPERATING AT MAXIMUM PERMISSIBLE
C                   OVERLOAD
C          PWRNL=   NOT USED HERE
C          EQAREA=  EQUIVALENT HEAT TRANSFER AREA BETWEEN
C                   SUCTION AND DISCHARGE MANIFOLDS (FT**.37)
C                   - THIS IS USED TO GIVE A ROUGH APPROXIMATION
C                   OF INTERNAL HEAT TRANSFER LOSSES, IF A
C                   PARTICULAR COMPRESSOR DESIGN SHOULD REQUIRE
C                   THAT IT BE INCLUDED
C                   (THERE IS, HOWEVER, NO REPLACEMENT FOR THE
C                   ACTUAL CONDITIONS IN EACH COMPRESSOR)
C          DDELAY=  DISCHARGE VALVE CLOSING DELAY (DEGREES
C                   AFTER TOP DEAD CENTER)
C          XOIL  =  OIL CIRCULATION RATE (LBM OIL/LBM OF REF.+OIL)
C          EFFME=   MECHANICAL EFFICIENCY OF COMPRESSOR
C
```

```
C      INPUT DATA CONSTANTS
C         NREF =  REFRIGERANT NUMBER (USUALLY SAME AS NR)
C         SLPEMV & XINMV - COEFFICIENTS FOR DETERMINING
C                          VISCOSITY OF REFRIGERANT VAPOR
C         SLPEKV & XINKV - COEFFICIENTS FOR DETERMINING
C                          THERMAL CONDUCTIVITY OF REFRIGERANT
C                          VAPOR
C         CPRV1 & CPRV2  - COEFFICIENTS FOR DETERMINING
C                          SPECIFIC HEAT AT CONS. PRES.
C                          OF VAPOR
C
C      OUTPUT PARAMETERS
C         TIC   =  TEMP.OF REFRIGERANT LEAVING COMP.(F)
C         HIC   =  ENTHALPY OF REFRG.LEAVING COMP.(BTU/LBM)
C         XMR   =  MASS FLOW RATE OF REFRG.LEAVING COMP.(LBM/HR)
C         POW   =  POWER INPUT (KW) REQUIRED BY COMPRESSOR MOTOR
C         QE    =  COOLING CAPACITY OF COMPRESSOR (BTU/HR)
C                  BASED ON ZERO SUBCOOLING AND NO PRESSURE
C                  LOSSES  ON HIGH OR LOW SIDE, WITH THE
C                  EFFECT OF OIL CIRCULATION INCLUDED
C
C      REMARKS
C         THIS PROGRAM CALLS SUBROUTINE TRIAL TO DETERMINE
C         VAPOR PROPERTIES WHICH MUST BE FOUND BY ITERATION
C         THIS PROGRAM CALLS SUBROUTINE VAPOR TO DETERMINE
C         VAPOR PROPERTIES
C         THIS PROGRAM USES FUNCTION SUBPROGRAM TSAT TO
C         DETERMINE SATURATION TEMPERATURES CORRESPONDING
C         TO GIVEN PRESSURES
C         THIS PROGRAM CALLS SUBROUTINE SATPRP TO DETERMINE
C         SATURATION STATE PROPERTIES
C         THIS PROGRAM USES SUBROUTINE OIL TO DETERMINE THE
C         REFRIGERANT-OIL SOLUBILITY BEHAVIOR
C         THIS PROGRAM USES FUNCTION SUBPROGRAM EFFM TO
C         DETERMINE COMPRESSOR MOTOR EFFICIENCY AS A FUNCTION
C         OF LOAD
C         THIS PROGRAM USES SUBROUTINE HEAT TO DETERMINE
C         SUCTION-DISCHARGE HEAT TRANSFER
C   NOTE:  INPUT DATA CONSTANTS FOR VAPOR VISCOSITY, ETC. ARE
C          FOR REFRIGERANT 22 ONLY, AND SUBROUTINE OIL, FOR
C          DETERMINING REFRIGERANT-OIL SOLUBILITY,IS FOR
C          REFRIGERANTS 12, AND 22 ONLY
C
      COMMON/COMPR/NCYL,VR,VD,SYNC,RPM,EFFIS,DPDI,DPS,DPFRAC,
     1SDELAY,PMC,PHT,PHTD,EAD,EAS,XMR,POW,TIC,HIC,H1CE,PIC,
     2P1CE,T1CE,POWMAX,PWRNL,EGAREA,DDELAY,XOIL,EFFME
      DATA NREF,SLPEMV,XINMV,SLPEKV,XINKV/22,.0000759,.0272,
     1.00002,.00482/
      DATA CPRV1,CPRV2/.000433,.1394/
      WRITE(5,620) EFFIS,EFFME,DPDI,DPS,DPFRAC
```

```
      WRITE(5,650) NCYL,VR,VD,RPM,SUPER
      WRITE(5,700) EAD,EAS,PHT,EFFD,PHTD,PMC
      WRITE(5,630) SYNC,POWMAX,PWRNL,DDELAY,SDELAY
      WRITE(5,750) ICONTR,CUTOFF
      VRACT = VR
      IF(ICONTR.EQ.2) GO TO 1
C
C     CONVENTIONAL COMPRESSOR
C
C     CALCULATE THE DISPLACEMENT EFFICIENCY 'EFFD' AS AFFECTED
C     BY THE CLOSING DELAY OF THE SUCTION VALVE
C     (NOTE THAT WITH EARLY SUCTION VALVE CUT-OFF CONTROL,
C     THIS EFFECT IS NOT PRESENT)
C
      EFFD =(1.0-COS(3.14159*(1.0-SDELAY/180.0)))/2.0
C
C     CALCULATE THE EFFECTIVE CLEARANCE VOLUME RATIO 'VR'
C
      VR = VR/EFFD
C
C     CALCULATE THE EFFECTIVE MAX.VOLUME RATIO 'VRMAX',
C     DEFINED AS 'VRMAX'=VOL.MIN/VOL.MAX.(EFFECTIVE VOLUMES)
C
      VRMAX = VR/(1.0 + VR)
C
C     CALCULATE EFFECTIVE DISPLACEMENT VOLUME
C
      VD = EFFD*VD
      GO TO 2
C
C     CAPACITY CONTROLLED COMPRESSOR
C
C     CALCULATE THE MAX.VOLUME RATIO  'VRMAX' DEFINED AS
C     'VRMAX' = VOL.MIN/VOL.MAX (ACTUAL VOLUMES)
C
  1   VRMAX = VR/(1.0+VR)
C
C     CALCULATE THE VOLUME RATIO OF CUT-OFF CONTROL,DEFINED
C     AS 'VRCUT' = VOL.OF CYL.WHEN SUCTION VALVE IS CLOSED/
C     VOL.DISP. (ACTUAL VOLUMES)
C
      VRCUT = 1.0-CUTOFF + VR
  2   TSATC = TSATCI
C
C     CALCULATE VOLUME RATIO AT MIN.EFFECTIVE VOLUME 'VRMEV',
C     AS AFFECTED BY THE CLOSING DELAY OF THE DISCHARGE VALVE
C     'VRMEV' = EFFECTIVE MINIMUM VOL./ACTUAL MINIMUM VOL.
C
      VRMEV=1.0+(1.-COS(DDELAY*3.14159/180.))/(2.*VRACT)
C
```

```
C      LOOP FOR VARYING CONDENSER SATURATION TEMP.
C
       DO 450 III = 1,NC
       TSATC = TSATC + DTC
C
C      DETERMINE SATURATION PROPERTIES AT 'TSATC'
C
       CALL SATPRP(NR,TSATC,PSATC,VF,VG,HSATLC,HFG,HG,SF,SG)
C
C      LOOP FOR VARYING EVAPORATOR SATURATION TEMP.
C
       TSATE = TSATEI
       DO 350  JJJ = 1,NE
       DPDV = 0.0
       DPSV = 0.0
       TSATE = TSATE + DTE
C
C      DETERMINE SATURATION PROPERTIES AT 'TSATE'
C
       CALL SATPRP(NR,TSATE,P1OE,VF,VG,HF,HFG,HG,SF,SG)
       T1 = TSATE + SUPER
       T1OE = T1
C
C      DETERMINE REFRIGERANT PROPERTIES ENTERING COMPRESSOR
C
       CALL VAPOR(NR,T1OE,P1OE,VVAP,H1OE,SVAP)
C
C      LOOP FOR STUDYING THE EFFECT OF THE EQUIVALENT PRESSURE
C      DROP MODEL FOR THE DISCHARGE VALVE (NOT IN USE HERE)
C
       DPD = DPDI
       DPDR = DPDI
       DO 200 IDPD=1,20
C
C      DETERMINE THE ACTUAL CYL.PRES.AT DISCHARGE 'P2' (PSIA)
C
       P2 = PSATC + DPD + DPDV
C
C      DETERMINE ACTUAL CYL.PRES.ON SUCTION STROKE'P1I' (PSIA)
C
       P1I = P1OE - DPS
C
C      MAKE AN INITIAL GUESS FOR THE ACTUAL TEMPERATURE OF
C      THE VAPOR ENTERING THE CYL.ON SUCTION STROKE, AS
C      AFFECTED BY MOTOR COOLING  'T1 = T1OE + 20 (F)
C
       T1 = T1OE + 20.0
       CALL VAPOR(NR,T1,P1I,V1,H1,S1)
       P1 = P1I- DPSV
       DT1AVG = 0.0
```

```
C
C -----ITERATE ON MOTOR COOLING AND INTERNAL HEAT TRANSFER---
C
      HPREV = H1
C
C     SET THE ENTHALPY AND INTERNAL ENERGY CONVERGENCE
C     TOLLERANCE 'HTOL' (BTU/LBM)
C
      HTOL = .1
      DO 120  K= 1,20
      HREF = 0.0
      VREF = 0.0
      TI1 = T1 + 3.0
C
C     USE THE GENERAL PROPERTY CONVERGENCE SUBROUTINE 'TRIAL'
C     TO DETERMINE VAPOR PROPERTIES GIVEN H, AND P
C
      CALL TRIAL(NR,TI1,-3.0,P1,3,H1,HTOL,V,H,SS,T)
C
C     SET INITIAL VALUES FOR VAPOR PROPERTIES AT EARLY
C     SUCTION VALVE CLOSING (CUT-OFF) VCUT (CU FT/LBM ),
C     HCUT (BTU/LBM), SCUT (BTU/LBM-R),TCUT (F)
C
      VCUT = V
      HCUT = H
      SCUT = SS
      TCUT = T
C
C----------THERMODYNAMIC STATE BALANCE AND MIXING----------
C
C     ITERATE ON A THERMODYNAMIC STATE BALANCE AT THE END
C     OF THE SUCTION STROKE
C
      DO 120 J = 1,20
      IF(ICONTR.EQ.2) GO TO 3
C
C     IF 'ICONT' = 1, WE ARE STUDYING A CONVENTIONAL COMP.
C     AND STATES AT CUT-OFF ARE SAME AS STATES AT BOTTOM
C     DEAD CENTER - SKIP EXPANSION OF SUCTION GAS SECTION
C
      VMAX = VCUT
      HMAX = HCUT
      SMAX = SCUT
      PMAX = P1
      WEXCUT = 0.0
      GO TO 4
C
C------EFFECT OF EXPANSION OF SUCTION GAS AFTER CUT-OFF------
C
C     DETERMINE SPECIFIC VOLUME OF VAPOR AT BOTTOM DEAD CENTER
```

```
C      (BCC) 'VMAX', USING CONSERVATION OF MASS
C
   3   VMAX = VCUT*(1.0+VR)/VRCUT
C
C      USE IDEAL GAS LAW TO OBTAIN FIRST GUESS FOR END PRESSURE
C
       P = P1*VCUT/VMAX
C
C      CHECK FOR ISENTROPIC EXPANSION INTO SATURATION REGION
C
       NCUT = 30
       T = TSAT(NR,P) + 30.0
       DT = -10.0
       DO 250 I = 1,30
       T = T + DT
       CALL SATPRP(NR,T,P,VF,VG,HF,HFG,HG,SF,SG)
       QUALS= (SCUT- SF)/(SG-SF)
       QUALV = (VMAX - VF)/( VG - VF)
       IF((QUALS.GT.1.0) .AND. (QUALV.GT.1.0)) GO TO 250
C
C      EXPANSION IS NOT INTO SATURATION REGION IF THE INDICATED
C      QUALITIES 'QUALS' AND 'QUALV' ARE CONTRADICTORY- GO TO
C      VAPOR REGION ISENTROPIC EXPANSION AT STEP 272
C
       IF((QUALS.GT.1.0).AND.(QUALV.LT.1.0)) GO TO 270
       IF(ABS(QUALS-QUALV).LE..01) GO TO 260
       IF(QUALS - QUALV) 250,260,240
 240   T = T - DT
       DT = DT/2.0
 250   CONTINUE
       WRITE(5,250)
       GO TO 1000
 260   HMAX = QUALS*HFG + HF
       SMAX = SCUT
       TMAX = T
       PMAX = P
       UMAX=HCUT-P1*VCUT*144.0/778.0-EFFIS*(HCUT-HMAX-(P1*
      1VCUT-PMAX*VMAX)*144.0/778.0)
C
C      IF THE ISENTROPIC EXPANSION WENT INTO THE SATURATION
C      REGION, THEN CHECK TO SEE IF THE NON-ISENTROPIC EXPANSION
C      GOES INTO THE SATURATION REGION, BASED ON RESULTS FROM
C      THE ISENTROPIC EXPANSION
C
       DT = 2.0
       T = T - DT
       DO 290 I = 1,30
       T = T + DT
       CALL SATPRP(NR,T,P,VF,VG,HF,HFG,HG,SF,SG)
       QUALV = (VMAX - VF)/(VG - VF)
```

```
      IF(QUALV.GT.1.0) GO TO 280
      UG = HG -P*VG*144.0/778.0
C
C     IF THE INTERNAL ENERGIES 'UMAX' AND 'U' (BTU/LBM)
C     ARE CONTRADICTORY, THEN NON-ISENTROPIC EXPANSION IS NOT
C     INTO THE SATURATION REGION - GO TO STEP 275
C
      IF(UMAX.GE.UG) GO TO 275
      H = QUALV*HFG + HF
      U = H - P*VMAX*144.0/778.0
      IF(ABS(UMAX-U).LE.HTOL) GO TO 295
      IF(UMAX - U) 280,295,290
 280  T = T - DT
      DT = DT/2.0
 290  CONTINUE
      WRITE(5,860)
      GO TO 1000
 295  SMAX = QUALV*(SG-SF) + SF
      WRITE(5,870) QUALV,T,P
      GO TO 10
C
C     NON-SATURATED EXPANSION SECTION
C
 270  P = P1*VCUT/VMAX
C
C     ITERATE ON PRESSURE  TO FIND ISENTROPIC EXPANSION
C     END STATE AT BDC
C
      DP = -5.0
      P = P - DP
      DO 6 I = 1,NCUT
      P = P + DP
      T = TSAT(NR,P)
      TSTART = T - 5.0
      CALL SATPRP(NR,T,PSAT,VF,VG,HF,HFG,HG,SF,SG)
      IF(VMAX.LT.VG) GO TO 5
C
C     USE THE GENERAL PROPERTY CONVERGENCE SUBROUTINE 'TRIAL'
C     TO DETERMINE VAPOR PROPERTIES GIVEN P, AND V
C
      CALL TRIAL(NR,TSTART,5.0,P,2,VMAX,.0001,V,H,SS,T)
      IF(ABS(SCUT-SS).LE..00005) GO TO 7
      IF(SCUT-SS) 6,7,5
  5   P = P-DP
      DP = DP/2.0
  6   CONTINUE
      WRITE(5,730)
      WRITE(5,840) VMAX,SCUT,P,VG,V,SS
      WRITE(5,731) IDPD,K,J,I
      WRITE(5,732)
```

```
      GO TO 1000
    7 HMAX = H
      SMAX = SS
      TMAX = T
      PMAX = P
      UMAX = HCUT-P1*VCUT*144.0/778.0-EFFIS*(HCUT-HMAX-
     1(P1*VCUT-PMAX*VMAX)*144.0/778.0)
  275 P = P1*VCUT/VMAX
C
C     ITERATE ON PRESSURE TO FIND NON-ISENTROPIC EXPANSION
C     END STATE AT BDC, USING RESULTS FROM THE ISENTROPIC
C     EXPANSION CASE
C
      DP = -5.0
      P = P - DP
      DO 9 I = 1,NCUT
      P = P + DP
      T = TSAT(NR,P)
      TSTART = T - 5.0
      CALL SATPRP(NR,T,PSAT,VF,VG,HF,HFG,HG,SF,SG)
      IF(VMAX.LT.VG) GO TO 8
C     TO DETERMINE VAPOR PROPERTIES GIVEN P, AND V
      CALL TRIAL(NR,TSTART,5.0,P,2,VMAX,.0001,V,H,SS,T)
      U = H - P*V*144.0/778.0
      IF(ABS(UMAX-U).LE.HTOL) GO TO 11
      IF(UMAX-U) 9,10,8
    8 P = P-DP
      DP = DP/2.0
    9 CONTINUE
      WRITE(5,740)
      WRITE(5,840) VMAX,UMAX,U,P,VG
      WRITE(5,731) IDPC,K,J,I
      WRITE(5,732)
      GO TO 1000
   11 WRITE(5,875)
      WRITE(5,876) T,P,H,SS,V
      SMAX = SS
C
C     DEFINE STATE PROPERTIES AT BDC (MAXIMUM VOLUME)
C     'HMAX' (BTU/LBM), 'VMAX' (CU FT/LBM), 'SMAX' (BTU/LBM-R)
C     'TMAX' (F), 'PMAX' (PSIA)
C
   10 HMAX = H
      TMAX = T
      PMAX = P
C
C-------END OF EXPANSION OF SUCTION GAS AFTER CUT-OFF-------
C
C     DETERMINE STATE PROPERTIES IN CYLINDER AFTER COMPRESSION,
C     ASSUMING ISENTROPIC COMPRESSION
```

```
C
    4  CALL TRIAL(NR,TSATC,20.0,P2,4,SMAX,.00005,V,H,SS,T)
       V2 = V
       H2 = H
       S2 = SS
       T2 = T
C
C      CALCULATE NON-ISENTROPIC COMPRESSION WORK 'WC' (BTU/LBM)
C
       WC=(HMAX-H2+(P2*V2-PMAX*VMAX)*144.0/778.0)/(EFFIS*
      1(1.0-VRMAX*VMAX*VRMEV/V2))
C
C      DETERMINE STATE PROPERTIES IN CYLINDER AT END OF
C      COMPRESSION FOR NON-ISENTROPIC COMPRESSION, BASED ON
C      RESULTS FROM ISENTROPIC COMPRESSION CASE
C
       T = T2
       DT = 5.0
       DO 20 I = 1,30
       T = T + DT
       CALL VAPOR(NR,T,P2,VVAP,HVAP,SVAP)
       Z=(HMAX-HVAP+(P2*VVAP-PMAX*VMAX)*144.0/778.0)/(1.0-
      1VRMAX*VMAX*VRMEV/VVAP)
       IF(ABS(Z-WC).LE.HTOL) GO TO 25
       IF(WC-Z) 20,25,15
   15  T = T-DT
       DT = DT/2.0
   20  CONTINUE
       WRITE(5,400)
C
C      DEFINE ACTUAL STATE PROPERTIES AT END OF COMPRESSION
C      PORTION OF STROKE
C
   25  H2 = HVAP
       V2 = VVAP
       S2 = SVAP
       T2 = T
C
C      DETERMINE WORK REQUIRED FOR DISCHARGE PORTION OF STROKE
C      'WD' (WHICH IS ASSUMED TO OCCUR AT CONSTANT PRESSURE)
C
       WD = -P2*V2*144.0/778.0
C
C      DEFINE STATE PROPERTIES OF RESIDUAL MASS AT END OF
C      DISCHARGE STROKE
C
       HRES = H2
       VRES = V2
       SRES = S2
       TRES = T2
```

```
C
C     USE THE GENERAL PROPERTY CONVERGENCE SUBROUTINE 'TRIAL'
C     TO DETERMINE STATE PROPERTIES AT END OF THE RE-EXPANSION
C     PORTION OF STROKE, GIVEN P AND S, AND ASSUMING
C     ISENTROPIC RE-EXPANSION
C
      CALL TRIAL(NR,T1,10.0,P1,4,SRES,.00005,V,H,SS,T)
      VRX = V
      HRX = H
      TRX = T
C
C     CALCULATE THE NON-ISENTROPIC WORK DONE BY RE-EXPANSION
C
      WRX=EFFIS*(HRES-HRX-(P2*VRES-P1*VRX)*144.0/778.0)/
     1(VRES/(VMAX*VRMAX*VRMEV)-1.0)
C
C     ITERATE TO FIND THE ACTUAL STATE PROPERTIES AT THE END
C     OF NON-ISENTROPIC RE-EXPANSION
C
      T = TRX
      DT = 10.0
      DO 35 I= 1,30
      T = T + DT
      CALL VAPOR(NR,T,P1,VVAP,HVAP,SVAP)
      Z=(HRES-HVAP-(P2*VRES-P1*VVAP)*144.0/778.0)/(VRES/
     1(VMAX*VRMAX*VRMEV)-1.0)
      IF(ABS(Z-WRX).LE.HTOL) GO TO 40
      IF(Z-WRX) 30,40,35
   30 T = T-DT
      DT = DT/2.0
   35 CONTINUE
      WRITE(5,410)
C
C     DEFINE STATE PROPERTIES IN CYLINDER AFTER NON-ISENTROPIC
C     RE-EXPANSION
C
   40 VRX = VVAP
      HRX = HVAP
      SRX = SVAP
      TRX = T
C
C     ACCOUNT FOR MIXING OF THE RESIDUAL GAS AND THE INCOMMING
C     SUCTION GAS TO FIND THE ACTUAL STATE AT THE BEGINNING
C     OF CUT-OFF
C
      T = T1
      DT = 5.0
      DO 50 I = 1,30
      T = T + DT
      CALL VAPOR(NR,T,P1,VVAP,HVAP,SVAP)
```

```
      IF(ICONT.NE.2)Z=H1-HVAP+VRMAX*VVAP*VRMEV*(HRX-H1)/VRES
      IF(ICONTR.EQ.2)Z=H1-HVAP+VR*VRMEV/VRCUT*VVAP*(HRX-H1)
     1/VRES
      IF(ABS(Z).LE.HTOL) GO TO 55
      IF(Z) 45,55,50
   45 T = T-DT
      DT = DT/2.0
   50 CONTINUE
      WRITE(5,432)
   55 HCUT = HVAP
      VCUT = VVAP
      SCUT = SVAP
      TCUT = T
      IF((ABS(HCUT-HREF).LE.HTOL).AND.(ABS(VCUT-VREF).LE.
     1.25)) GO TO 110
      HREF = HCUT
      VREF = VCUT
  100 CONTINUE
C
C-------END OF THERMODYNAMIC STATE BALANCE AND MIXING-------
C
      WRITE(5,440)
C
C     CALCULATE WORK PRODUCED ON VAPOR INTAKE 'WI' (BTU/LBM),
C     WORK PRODUCED BY EXPANSION OF GAS AFTER CUT-OFF 'WEXCUT'
C     (BTU/LBM), AND MASS PUMPED PER STROKE 'XMFLOW' (LBM/STROKE)
C
  110 IF(ICONTR.NE.2)WI=P1*(1.0/(VRMAX*VRMEV) -VRX/VRES)*
     1VRES*144.0/(778.0*(VRES/(VRMAX*VRMEV*VMAX)-1.0))
      IF(ICONTR.EQ.2)WI=P1*(VRCUT/(VR*VRMEV)-VRX/VRES)*
     1VRES*144.0/(778.0*(VRES/(VRMAX*VRMEV*VMAX)-1.0))
      IF(ICONTR.EQ.2)WEXCUT=(HCUT-HMAX-(P1*VCUT-PMAX*VMAX)
     1*144.0/778.0)/(1.0-VRMAX*VRMEV*VMAX/VRES)
      XMFLOW = VR*VRMEV*VD*(1.0/(VRMAX*VRMEV*VMAX)-1.0/VRES)
      ICOUNT = 1
  140 ICOUNT = ICOUNT + 1
      IF(ICOUNT .GT. 10) GO TO 150
C
      WTOT = ABS(WC+WR+WRX+WI+WEXCUT)
C     CALCULATE TOTAL REFRIGERANT FLOW RATE 'XMR' (LBM/HR)
C
      XMR = XMFLOW*RPM*60.0*FLOAT(NCYL)
C
C     DETERMINE 'WCOMP', THE ACTUAL WORK INPUT TO THE
C     COMPRESSOR (BTU/LBM), ACCOUNTING FOR MECHANICAL
C     EFFICIENCY OF COMPRESSOR
C     DETERMINE 'POWER', THE ACTUAL POWER REQUIRED TO RUN
C     THE COMPRESSOR (KW), AND THEN DETERMINE 'PP', THE
C     PERCENT LOAD ON THE MOTOR, AND ITERATE ON MOTOR SPEED
C
```

```
      WCOMP = WTOT/EFFME
      POWER = XMR*WCOMP*.0002928
      PP = POWER/POWMAX
      IF((ICONTR.EQ.2).AND.(PP.LE..4)) PP = .4
      RPMN=SYNC*(.21064*PP**4-.26395*PP**3+.01930*PP**2-
     1.01299*PP+.994)
      IF(ABS(RPM-RPMN).LT.(.01*SYNC)) GO TO 160
      RPM = RPMN
      GO TO 140
 150  WRITE(5,310) PP,RPM,RPMN
C
C     DETERMINE MOTOR EFFICIENCY 'EFFMO' AS A FUNCTION OF LOAD
C
 160  EFFMO = EFFM(PP)
      WACT = WCOMP/EFFMO
      WRITE(5,822) WO,WC,WRX,WI,WEXCUT,WACT
      WRITE(5,825) EFFMO, RPM,PP
C
C     ACCOUNT FOR MOTOR COOLING AND INTERNAL HEAT TRANSFER
C
C     DETERMINE THE AMOUNT OF HEAT 'QMC' GIVEN TO THE SUCTION
C     GAS BY MOTOR COOLING
C
      QMC = PMC*(WACT-WTOT)
      H1MC = H1CE + QMC
      IF(H1MC.GT.H1) DTRIAL = 3.0
      IF(H1MC.LT.H1) DTRIAL = -3.0
C
C     USE THE GENERAL PROPERTY CONVERGENCE SUBROUTINE 'TRIAL'
C     TO DETERMINE A NEW VALUE FOR THE TEMPERATURE OF THE
C     SUCTION GAS  ENTERING THE CYLINDER, USING P, AND H
C
      CALL TRIAL(NR,T1,DTRIAL,P1I,3,H1MC,HTOL,V,H,SS,T)
      T1 = T
      T1I = T1
C
C     CALCULATE REFRIGERANT PROPERTIES, AND THEN MAKE A ROUGH
C     ESTIMATE OF THE HEAT TRANSFER 'QHT' BETWEEN SUCTION AND
C     DISCHARGE MANIFOLDS
C
      IF(EQAREA.LE..01) QHT = 0.
      IF(EQAREA.LE..01) GO TO 162
      XMPC = XMR/FLOAT(NCYL)
      CPS = CPRV1*P1 + CPRV2
      CPD = CPRV1*P2 + CPRV2
      CALL HEAT(CPS,CPD,SLPEMV,XINMV,SLPEKV,XINKV,TRES,T1,
     1EQAREA,XMPC,QHT,TDISCO,TSUCTO)
      T1 = TSUCTO
 162  H1 = H1MC + QHT
      IF(ABS(H1-HPREV).LE.HTOL) GO TO 180
```

```
      HPREV = H1
C
C     USE THE GENERAL PROPERTY CONVERGENCE SUBROUTINE 'TRIAL'
C     TO DETERMINE  A NEW GUESS FOR PROPERTIES OF THE REFRIG.
C     ENTERING THE CYLINDER, GIVEN P, AND H
C
      CALL TRIAL(NR,T1I,3.0,P1I,3,H1,HTCL,V,H,SS,T)
      V1 = V
      S1 = SS
      T1 = T
      WRITE(5,830) T1I,T1,DT1AVG,GHT,GMC,EGAREA
 120  CONTINUE
C
C--------END MOTOR COOLING AND INTERNAL HEAT TRANSFER-------
C
      WRITE(5,504)
 180  IF(ABS(DPD-DPDR).LE.5.0) GO TO 210
 200  CONTINUE
      WRITE(5,560)
 210  DPDV = 0.0
      DPSV = 0.0
      WRITE(5,733) IDPD,K,J
      WRITE(5,301) T2,P2,V2,H2,HVAP,S2,SVAP
      WRITE(5,570) DPD,DPDR,DPFRAC,XMR
      WRITE(5,650) DPDV,DPSV
C
C     DETERMINE THE EFFECT OF OIL CIRCULATION RATE ON
C     COMPRESSOR PERFORMANCE
C
C     CALCULATE THE WORK NECESSARY TO COMPRESS THE OIL BEING
C     PUMPED WITH THE GAS 'WOIL' (BTU/LBM),AND THE TOTAL
C     COMPRESSOR POWER REQUIREMENT 'POW' (KW)
C
      WOIL = (PSATC-P1CE)*144.0/(778.0*57.6)
      POW = XMR*(WACT + XOIL*WOIL/(1.0-XOIL))*.0002928
C
C     FINALLY, DETERMINE THE EFFECT OF THE OIL ON THE ENTHALPY
C     OF THE REFRIGERANT, AND FIND THE NET REFRIGERATION
C     EFFECT POSSIBLE 'QE' (BTU/HR)
C
      CALL OIL(NR,P1CE,T1CE,XOIL,H1CE,W,H1M)
      H3OIL = .403*TSATC + .00025*TSATC**2 + 15.75
      H3M = XOIL*H3OIL + (1.0-XOIL)*HSATLC
      QE = XMR/(1.0 - XOIL)*(H1M-H3M)
C
C     PRINT RESULTS
C
      WRITE(5,320) XOIL,W,H1M,H3M
      WRITE(5,670) XMR,QE,POW
C
```

```
C      DETERMINE THE EXIT TEMPERATURE FROM THE COMPRESSOR, AS
C      AFFECTED BY INTERNAL HEAT TRANSFER
C
       HRES = HRES - GHT
       TI = TRES + 2.0
       CALL TRIAL(NR,TI,-2.0,PSATC,3,HRES,.01,V,H,SS,T)
       TIC = T
       HIC = HRES
       PIC = PSATC
       IF(TIC.GE.280.0) WRITE(5,302)
       WRITE(5,540)
       WRITE(5,550) TSATE,TSATC,PSATC,TIC,HIC,H1OE,WACT
  350  CONTINUE
  450  CONTINUE
  301  FORMAT('0','T=',F6.1,3X,'P2=',F6.2,3X,'V2=',F6.4,3X,
      1'H2=',F6.2,3X,'  HVAP  =',F6.2,3X,'S2=',F6.4,3X,
      2'  SVAP  =',F6.4)
  302  FORMAT(' *********DISCHARGE TEMPERATURE EXCESSIVE***')
  310  FORMAT(' *****ITERATION ON MOTOR SPEED DOES NOT '
      1,,'CONVERGE PF=',E12.5,' RPM=',E12.5,' RPMN=',E12.5 )
  320  FORMAT('   XOIL=',F10.5,' LBM OIL/LBM MIX   W=',F6.4,
      1'     LBM REFL/LBM OIL + REFL   H1M=',F7.2,' BTU/LBM'
      2,,' MIX   H3M=',F7.2,' BTU/LBM MIX')
  400  FORMAT(' ******NON-ISENTROPIC COMPRESSION DOES NOT'
      1,,' CONVERGE************')
  410  FORMAT(' *****RE-EXPANSION OF RESIDUAL DOES NOT'
      1,,' CONVERGE *************')
  430  FORMAT(' *****ACTUAL HMAX AND VMAX DOES NOT'
      1,,' CONVERGE********')
  440  FORMAT('******ITERATION ON STATES DOES NOT CONVERGE'
      1,,' **********')
  504  FORMAT('0********** ITERATION ON MOTOR COOLING EFFECT'
      1,,' ON STATE 1 DOES NOT CONVERGE ***********')
  540  FORMAT('0',' TSATE       TSATC       PSATC        TIC'
      1,,'      HIC          H1OE          W')
  550  FORMAT(7F10.2)
  560  FORMAT(' *****ITERATION ON DPD DOES NOT CONVERGE*****')
  570  FORMAT('  DPD=',F10.4,'   DPDR=',F10.4,'   DPFRAC='
      1,F10.5,'  XMR=',F15.5)
  620  FORMAT('0','EFF.S=',F10.2,5X,'EFFME=',F10.2,'  DPDI='
      1,F10.2,5X,'DPS=',F10.2,'   DPFRAC =',F10.5)
  630  FORMAT(5X,' SYNC=',F10.2,' RPM        POWMAX=',F10.5,' KW
      1PWRNL=',F10.5,'  DDELAY=',F10.5,' DEGREES  SDELAY='
      2,F10.5,'  DEGREES')
  650  FORMAT('0',10X,'NCYL=',I2,5X,'VR=',F10.3,5X,'VD='
      1,F10.4,'CU.FT.',5X,'RPM=',F10.2,5X,'SUPER=',F10.2,' F')
  670  FORMAT('0',10X,'XMR='F15.2,' LBM/HR',5X,'QE=',F15.2,
      1' BTU/HR',5X,'POWER=',F10.3,' KW')
  690  FORMAT('0',10X,'CPDV=',F10.4,5X,'DPSV=',F10.4)
  700  FORMAT('   EAD=',F10.4,' SG.IN   EAS=',F10.4,' SG.IN.'
```

```
     1,,'     PHT=',F10.4,'    EFFC=',F10.4,'    PHTD=',F10.4,
     2'    PMC=',F10.4)
720  FORMAT(' *********ITERATION ON DISCHARGE DOES NOT'
     1,,' CONVERGE ***********')
730  FORMAT(' *********ISENTROPIC CUTOFF NOT CONVERGED'
     1,,' **********')
731  FORMAT('     ICPD=',I10,'    K=',I10,'    J=',I10,'    I='
     1,I10)
732  FORMAT('  THIS FAILURE TO CONVERGE IS NON-CRITICAL'
     1,,' UNLESS THE VALUES OF ICPD,K,AND J ARE THE SAME AS'
     2,,' THE FINAL VALUES')
733  FORMAT('  FINAL ICPD=',I10,'    FINAL K=',I10,'    FINAL'
     1,,' J=',I10)
740  FORMAT(' **********NON-ISENTROPIC CUTOFF NOT CONVERGED'
     1,,' **********')
750  FORMAT('    ICONTR=',I10,'    CUTOFF=',F15.5)
760  FORMAT('    VCUT=',F10.4,'    HCUT=',F10.4,'    SCUT='
     1,F10.5,'    TCUT=',F10.4)
770  FORMAT('    P=',F10.4,'    T=',F10.4,'    VMAX=',F10.4,
     1'    V=',F10.4,'    H=',F10.4,'    SS=',F10.4,'    T='
     2,F10.4,'    SCUT=',F10.4)
780  FORMAT(' HMAX=',F8.4,' SMAX=',F8.5,' VMAX=',F8.4,
     1' TMAX=',F8.2,' PMAX=',F8.4,' WEXCUT=',F10.4,' UMAX='
     2,F10.4)
790  FORMAT(' VCUT='.F10.4,' HCUT=',F10.4,' SCUT=',F10.4,
     1'TCUT=',F10.4,' HREF=',F10.4,' VREF=',F10.4)
800  FORMAT('   P=',F8.4,' T=',F8.2,' VMAX=',F10.4,' V='
     1,F10.4,' H=',F10.4,' SS=',F10.5,' T=',F8.2,' U='
     2,F10.4,' UMAX=',F10.4)
810  FORMAT(' HMAX=',F8.4,' SMAX=',F8.5,' VMAX=',F8.4,
     1' TMAX=',F8.2,' PMAX=',F8.4)
820  FORMAT('    WC=',F10.4,'    WD=',F10.4,'    WRX=',F10.4,
     1'    WI=',F10.4,' WEXCUT=',F10.4,'    WACT=',F10.4)
825  FORMAT(5X,'EFFMC=',F7.3,'    RPM=',F10.2,' PR=', F7.3)
830  FORMAT('    T1I=',F7.2,' F    T1=',F7.2,' F    DT1AVG ='
     1,F7.2,' F   GHT =',F8.3,' BTU/LBM    GMC =',F8.3,
     2'   BTU/LBM   EGAREA=',F8.4,'   FT/FT**.63')
840  FORMAT(6E14.5)
850  FORMAT(' ********ISENTROPIC CUT-OFF SATURATION SEARCH'
     1,,' FAILS TO CONVERGE***********')
860  FORMAT(' ***********NON-ISENTROPIC CUT-OFF SATURATION'
     1,,'SEARCH FAILS TO CONVERGE**********')
870  FORMAT('    QUALV=',F10.5,'   T = ',F10.5,'    P =',F10.5)
875  FORMAT('    PROPERTIES AT END OF CUT-OFF STROKE'
     1,,'  T, P, H, SS, V')
876  FORMAT( 5X,5F10.5)
1000 END
```

```
      SUBROUTINE OIL(NR,P,T,X,H1CE,W,HM)
C
C     PURPOSE
C        TO CALCULATE THE ENTHALPY OF REFRIGERANT-OIL
C        MIXTURES FOR REFRIGERANTS 12, AND 22
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        NR    -  REFRIGERANT NUMBER (12, OR 22)
C        P     -  PRESSURE (PSIA)
C        T     -  TEMPERATURE OF MIXTURE (F)
C        X     -  WEIGHT PERCENT OF OIL IN THE TOTAL MIXTURE
C        H1CE  -  ENTHALPY OF PURE REFRIGERANT VAPOR (BTU/LBM)
C                 AT EVAPORATOR EXIT TEMP. AND PRES.
C      OUTPUT
C        W     -  WEIGHT PERCENT OF REFRIGERANT LEFT IN
C                 THE LIQ.PHASE REFRIGERANT-OIL MIXTURE
C                 LEAVING THE EVAPORATOR
C        HM    -  ENTHALPY (BTU/LBM) OF TOTAL REFRIGERANT-OIL
C                 MIXTURE LEAVING THE EVAPORATOR
C
C     REMARKS
C        SUBROUTINE SATPRP IS CALLED TO PROVIDE SATURATION
C        STATE PROPERTIES OF THE REFRIGERANT
      IWRITE = 5
      TR = T + 460.0
      IF(NR.EQ.22) GO TO 10
      IF(NR.EQ. 12) GO TO 15
      WRITE(IWRITE,100)
      GO TO 200
   10 W = ((ALOG10(P)-6.293+2136.0/TR)/(169.6/TR-.4448))
     1**(-2.0)
      GO TO 20
   15 PK = P*.07031
      TK = 5.0*(T-32.0)/9.0 + 273.16
      W = ((ALOG10(PK)-5.0057+1177.67/TK)/(98.753/TK -.558)
     1)**(-2.0)
   20 IF(W.GE.1.0) W = .9999
C
C     Z IS THE AMOUNT OF OIL AND REFRIGERANT LIQUID PER
C     POUND OF TOTAL MIXTURE LEAVING THE EVAPORATOR
      Z = X/(1.0-W)
      HOIL = .403*T + .00025*T**2 + 15.75
      CALL SATPRP(NR,T,PSAT,VF,VG,HLIQ,HFG,HG,SF,SG)
      HZ = (1.0-W)*HOIL + W*HLIQ
      HM = Z*HZ + (1.0-Z)*H1CE
  100 FORMAT(' *******ERROR IN SUBROUTINE OIL ********')
  200 RETURN
      END
```

```
      FUNCTION EFFM(PP)
C
C     PURPOSE
C        TO ESTIMATE MOTOR EFFICIENCY AS A FUNCTION OF
C        PERCENT OF MAXIMUM LOAD FOR SQUIREL CAGE
C        INDUCTION MOTORS
C
C     DESCRIPTION OF PARAMETERS
C      INPUT  =  PERCENT OF MAXIMUM POWER 'PP'
C      OUTPUT =  MOTOR EFFICIENCY 'EFFM'
C
      IWRITE = 5
      IF(PP.LT.0.0) GO TO 50
      IF(PP.GT.1.0) WRITE( IWRITE,100) PP
      IF(PP.LT..200) WRITE(IWRITE,110) PP
      IF(PP.LT..100) EFFM  = 6.65*PP
      IF(PP.GT..100) EFFM  = 1.3*PP + .555
      IF(PP.GT..200) EFFM  = .375*PP + .740
      IF(PP.GT..400) EFFM  = .89
      IF(PP.GT..600) EFFM  = -.1*PP +.95
      IF(PP.GT..8) EFFM  = -.25*PP + 1.07
      RETURN
   50 WRITE(IWRITE,120) PP
      RETURN
  100 FORMAT(' *****MOTOR POWER EXCESSIVE PP=',E12.5,'***')
  110 FORMAT(' *****MOTOR POWER TOO LOW - MOTOR INEFFICIENT'
     1, ,'  PP =',E12.5,' ********')
  120 FORMAT(' *****MOTOR POWER IS NEGATIVE PP=',E12.5,'**')
      END
```

```
      SUBROUTINE HEAT(CPS,CPD,SLPEMV,XINMV,SLPEKV,XINKV,
     1TDISCI, TSUCTI,EGAREA,XMPC,GHT,TDISCO,TSUCTO)
C
C     PURPOSE
C        TO ESTIMATE SUCTION-DISCHARGE MANIFOLD HEAT TRANSFER
C        IN THE COMPRESSOR
C     DESCRIPTION OF PARAMETERS
C      INPUTS
C          CPS    -  SPECIFIC HEAT AT CONSTANT PRESSURE OF THE
C                    GAS IN THE SUCTION MANIFOLD (BTU/LBM-R)
C          CPD    -  SPECIFIC HEAT AT CONSTANT PRESSURE OF
C                    GAS IN THE DISCHARGE MANIFOLD (BTU/LBM-R)
C          SLPEMV & XINMV  -  COEFFICIENTS FOR VISCOSITY OF
C                             VAPOR
C          SLPEKV & XINKV  -  COEFFICIENTS FOR THERMAL
C                             CONDUCTIVITY OF VAPOR
C          TSUCTI-   TEMP. OF GAS ENTERING SUCT. MANIFOLD (F)
C          TDISCI-   TEMP. OF GAS ENTERING DISC. MANIFOLD (F)
C          EGAREA-   EQUIVALENT AREA FOR HEAT TRANSFER - SEE
C                    EXPLANATION FOR UNITS (NOT UNITS OF AREA)
C          XMPC   -  MASS FLOW RATE OF REFRIGERANT PER
C                    CYLINDER (LBM/HR)
C      OUTPUTS
C          TSUCTO-   TEMP. OF GAS LEAVING SUCT. MANIFOLD. (F)
C          TDISCO-   TEMP.OF GAS LEAVING DISC. MANIFOLD (F)
C          GHT    -  SUCTION-DISCHARGE MANIFOLD HEAT TRANSFER
C                    (BTU/LBM)
      IWRITE = 5
      T = TDISCI - 20.
      DT = 5.
C
C     ITERATE ON TEMP. OF GAS LEAVING DISC. MANIFOLD UNTIL
C     THE CORRECT HEAT TRANSFER RATE IS FOUND
      DO 52 I = 1,32
      T = T - DT
      TSUCTO = TSUCTI + (TDISCI-T)*CPD/CPS
      DTA = TDISCI - TSUCTO
      DTB = T - TSUCTI
      DTLM = (DTA-DTB)/ALOG(DTA/DTB)
      TSAVG = (TSUCTI + TSUCTO)/2.
      TDAVG = (TDISCI + T)/2.
C
C     EVALUATE REFRIGERANT PROPERTIES
C        XMUS & XMUD - VISCOSITY (LBM/HR-FT)
C        XKS & XKD - THERMAL CONDUCTIVITY (BTU/HR-FT-F)
C        PRD & PRS - PRANDTL NUMBER
      XMUS = SLPEMV*TSAVG + XINMV
      XMUD = SLPEMV*TDAVG + XINMV
```

```
      XKS = SLPEKV*TSAVG +XINKV
      XKC = SLPEKV*TCAVG + XINKV
      PRS = XMUS*CPS/XKS
      PRC = XMUC*CPC/XKC
      FST=4./(1.48*XKS*PRS**.6*(4./XMUS)**.63)
      FCT=3./(1.48*XKC*PRC**.6*(6./XMUC)**.63)
      G = XMPC*CPC*(TCISCI - T)
      GS =XMPC**.63*EQAREA*CTLM/(FST+FCT)
      IF(ABS(G-GS).LE.(.01*G)) GO TO 62
      IF(((G-GS).GE.Ø.).AND.(I.EQ.1)) CT = -DT
      IF(CT.GT.Ø.) GO TO 35
      IF(G-GS) 4Ø,6Ø,5Ø
 35   IF(G-GS) 5Ø,6Ø,4Ø
 4Ø   T = T + CT
      CT = CT/2.
 5Ø   CONTINUE
      WRITE(IWRITE,2ØØ)
      GO TO 1ØØ
 6Ø   GHT = G/XMPC
      TCISCO = T
      RETURN
2ØØ   FORMAT(' *****SUBROUTINE HEAT DOES NOT CONVERGE****')
1ØØ   END
```

APPENDIX F

REFRIGERANT - OIL SOLUBILITY

Solubility of oil-refrigerant mixtures has been discussed by Bambach[1], Spauschus[2], and Cooper[3]. Many refrigerant-oil mixtures, such as R12 and R22 are miscible over the entire range of concentrations from 0 to 100%. Bambach[1] has found that the solubility of R12-oil mixtures may be described by the following expressions:

$$\text{(A)}\quad \text{Log}_{10}\,(P) = \left[5.0057 - .550\,w^{-1/2} - \frac{(1177.67 - 98.753\,w^{-1/2})}{T}\right] \quad T \leq 0°C$$

$$\text{(B)}\quad \text{Log}_{10}\,(P) = [\ (A)\ ] - [.002338\,(w - .6)^2 - .000075](T - 273.16) \quad T > 0°C$$

Where:

P = pressure (kg/cm^2)

T = temperature (^oK)

w = lbm refrigerant/lbm liquid mixture

Refrigerant 22 behaves slightly differently than refrigerant 12 in that refrigerant 12-oil mixtures remain a single phase throughout the entire range 0 to 100% concentration. R22-oil mixtures, however, separate into two distinct liquid phases above certain concentration limits. One phase is oil-rich, while the other phase is refrigerant-rich.

We are concerned here with the amount of liquid refrigerant left in the oil at the exit from the evaporator, and the fact that it is in two phases is of secondary importance. For this reason, it has been assumed that an expression similar to the one described by Bambach for R12 also holds for R22. That is:

$$\text{(C)} \quad \mathrm{Log}_{10}\,(P) = a - bw^{-1/2} - \frac{[C + dw^{-1/2}]}{T}$$

The necessary constants have been determined to be as follows:

$a = 6.293$

$b = .4448$

$c = 2136.0$

$d = -169.6$

Where

P = pressure (psia)

T = Temperature (^oR)

For simplicity we shall assume, even for R12, that the solubility may be described by a single expression of the form of equations (A) or (C). Comparisons of predicted results with actual solubility curves are given in Figures F-1 and F-2.[4] Expressions (A) and (C) may be rearranged to solve explicitly for an expression of w as a function of P and T

$$w = \left\{ \frac{[\mathrm{Log}_{10}\,(P) - 6.293 + \frac{2136.0}{T}]}{[\,\frac{169.6}{T} - .4448]} \right\}^{-2} \quad \text{for R22}$$

P = psia

T = ^oR

$$w = \left\{ \frac{[\operatorname{Log}_{10}(P) - 5.0057 + \frac{1177.67}{T}]}{[\frac{98.753}{T} - .558]} \right\}^{-2} \quad \text{for R12}$$

$$P = \frac{k_g}{cm^2}$$

$$T = {}^{o}K$$

References

1. Bambach, G., "Das Verhalten Von Mineralol -F12-Gemischen in Kaltmaschinen", C.F. Muller, Karlsruhe, 1955.

2. Spauschus, "Vapor Pressures, Volumes, & Miscibility Limits of R22 Oil Solutions", ASHRAE JOURNAL, Dec. '64, pg. 65 and also ASHRAE TRANSACTIONS, Vol. 70, 1964.

3. Cooper, K. W., and Mount, A. G., "Oil Circulation - Its Effect on Compressor Capacity, Theory and Experiment", 1972 Purdue Compressor Technology Conference Proceedings (Purdue Research Foundation, 1972.)

4. ASHRAE GUIDE & DATA BOOK, Systems & Equipment (New York: American Soc. of Heat, Refg., & Air Cond. Eng., Inc., 1967) pg. 265-285.

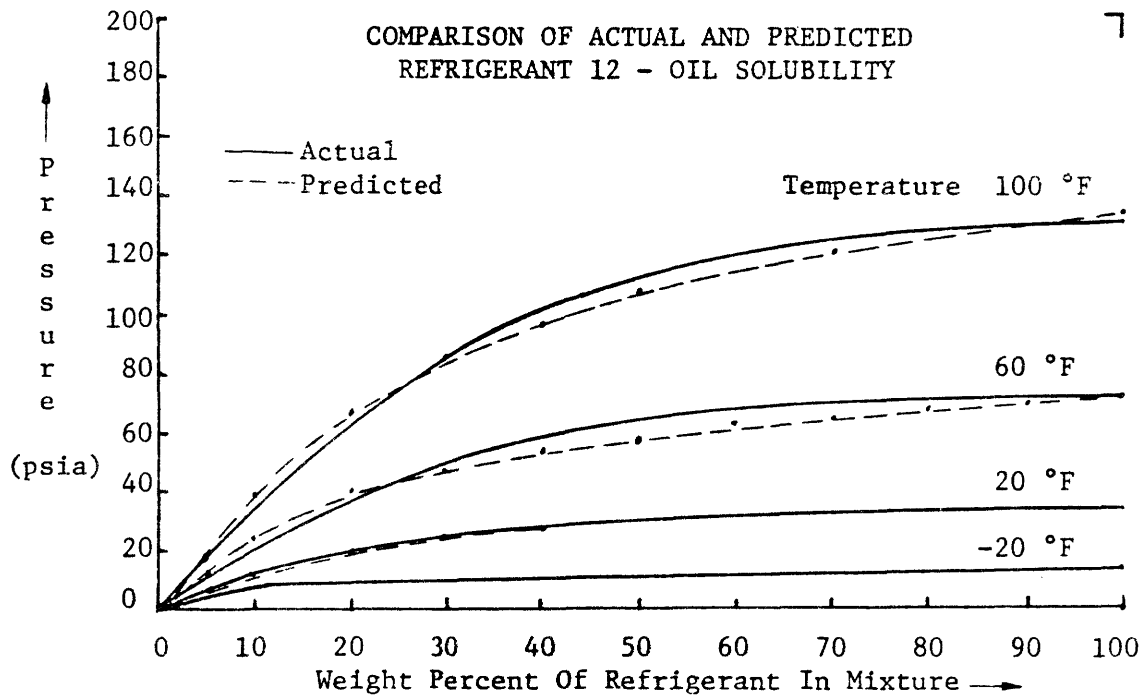


FIGURE F-1

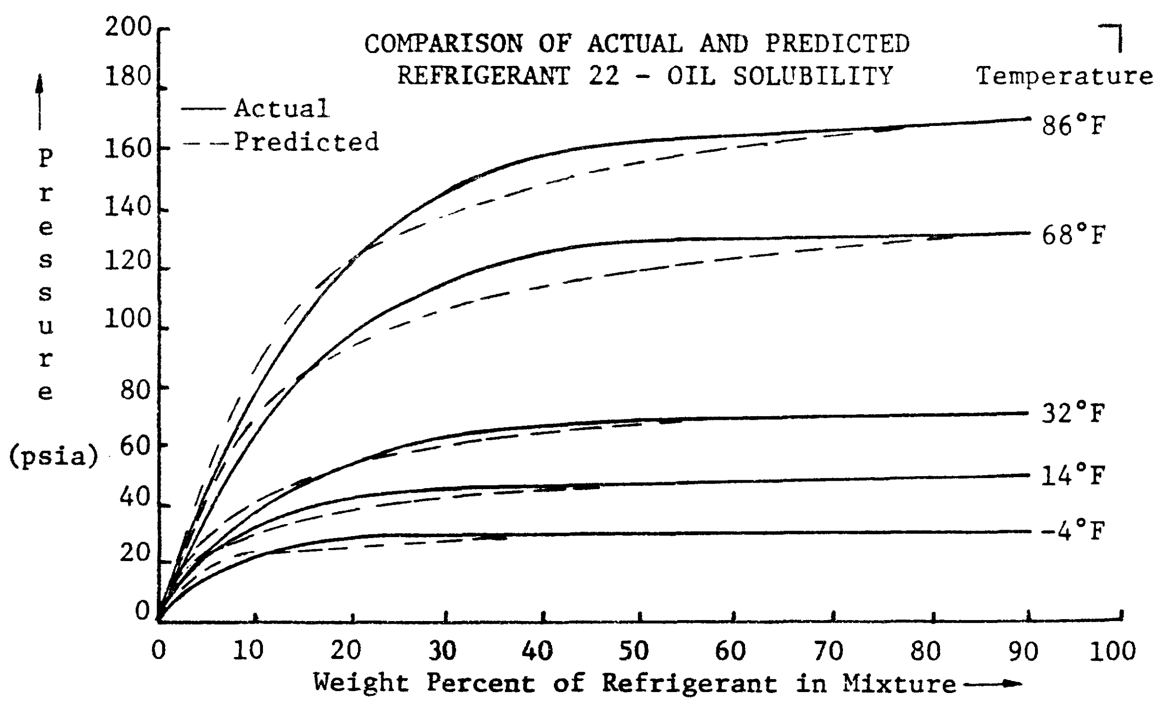


FIGURE F-2

APPENDIX G

COMPRESSOR DATA

Carrier 06D-824 Compressor

The 06D-824 is a relatively large, semi-hermetic refrigeration compressor, having a surface to volume ratio $(\frac{S}{A})$ of about 2.76 $(in)^{-1}$ per cylinder. The difference between a semi-hermetic and a welded hermetic compressor is that a semi-hermetic unit is bolted together, and can be disassembled for servicing. A welded hermetic, as most smaller refrigeration compressors are, cannot be serviced, but rather is replaced if failure should occur. Information concerning this nominal 9 ton compressor has been provided by Carlyle Compressor Company, a Division of Carrier Corporation.

Data for use in the compressor simulation was as follows:

Synchronous Motor Speed = 1800 RPM

Initial Guess for Actual Motor Speed = 1750 RPM

Refrigerant = 22

Displacement Volume $(V_D) = 2.273 \times 10^{-3}$ ft^3

Clearance Volume Ratio V_{min}/V_D = VR = .05

Number of Cylinders = 6

Superheat Base for Capacity Rating = 15^oF

Subcooling Base for Capacity Rating = 0^oF

η_{is} = 94%

η_{mech} = 96%

% Motor Cooling = 85%

$\Delta P_D = 25$ psi

$\Delta P_S = 5$ psi

$\theta_D = 0$

$\theta_S = 0$

$\Delta T_{\substack{\text{suct-disc}\\ \text{H.T.}}} = 0$

% Oil Circulation = 0

Maximum Power Output of Compressor Motor = 11.64 kw

Carrier 06D-537 Compressor

The 06D-537 is a large, semi-hermetic refrigeration compressor, having a surface to volume ratio of about 2.49 $(\text{in})^{-1}$ per cylinder. It should be noted that this nominal 14 ton compressor is a larger version of the 06D-824 compressor, having the same bore, but a longer stroke. The 06D-537 compressor is used in the Carrier Model 50 D Q 016 Heat Pump.

Data for use in the compressor simulation was as follows:

Synchronous Motor Speed = 1800 RPM

Initial Guess for Actual Motor Speed = 1750 RPM

Refrigerant = 22

Displacement Volume (V_D) = 3.522×10^{-3} ft^3

Clearance Volume Ratio V_{min}/V_D = VR =.05

Number of Cylinders = 6

Superheat Base for Capacity Rating = $15^{o}F$

Subcooling Base for Capacity Rating = $0^{o}F$

η_{is} = 94%

η_{mech} = 96%

% Motor Cooling = 85%

ΔP_D = 25 psi

ΔP_S = 3 psi

θ_D = 0

θ_S = 0

$\Delta T_{\substack{suct-disc \\ H.T}}$ = 0

% Oil Circulation = 0

Maximum Power Output of Compressor Motor = 16.89 kw

3-Ton Welded Hermetic Compressor

The manufacturer of this compressor wished to remain unidentified, but has provided the necessary technical information on this nominal 3 ton refrigeration compressor having an $(\frac{S}{A})$ of about 3.44 $(in)^{-1}$.

Data for use in the compressor simulation was as follows:

Synchronous Motor Speed = 3600 RPM

Initial Guess for Actual Motor Speed = 3500 RPM

Refrigerant = 22

$V_D = 1.15 \times 10^{-3}$ ft^3

VR = .062

Number of Cylinders = 2

Superheat Base for Capacity Rating = $20^{\circ}F$

Subcooling Base for Capacity Rating = $0^{\circ}F$

η_{is} = 90%

η_{mech} = 96%

% Motor Cooling = 85%

ΔP_D = 25 psi

ΔP_S = 5 psi

θ_D = 10° ATDC

θ_S = 15° ABDC

$\Delta T_{suct-Disc}$ = $50^{\circ}F$

H.T. @ $T_{evap\ sat} = 10^{\circ}F$, $T_{cond\ sat} = 120^{\circ}F$

% Oil Circulation = 5%

Max. Power Output of Compressor Motor = 3.9 kw

APPENDIX H

PARAMETRIC STUDIES ON CARRIER 06D-537 COMPRESSOR

EFFECT OF VARYING ΔP_s FROM 1 PSI TO 5 PSI WITH:

ΔP_D = 10 psi $\theta_s = 0$ $\eta_{mech} = .96$ $\Delta T_{suct} = 0$
$\theta_D = 0$ $\eta_{is} = .94$ % Oil = 0 % Motor cooling = .85

$T_{sat\ evap} - T_{sat\ cond}$	Change in Flow or Capacity (%)	Change in Power (%)	Change in Overall Compressor Efficiency (%)
50 - 145	-5.0	-1.7	-2
50 - 120	-5.8	-1.6	-2
50 - 80	-4.7	+2.6	-5
20 - 145	-9.3	-4.8	-3
20 - 120	-9.1	-3.8	-4
20 - 80	-8.3	-.9	-5
-10 - 120	-19.9*	-11.5	-6
-10 - 80	-17.5*	-6.4	-8

* - Due Mostly to % Density Change of Suction Gas at Low Pressure

EFFECT OF VARYING ΔP_D FROM 10 PSI to 30 PSI WITH:

ΔP_s = 1 psi $\theta_s = 0$ $\eta_{mech} = .96$ $\Delta T_{suct} = 0$
$\theta_D = 0$ $\eta_{is} = .94$ % Oil = 0 % Motor cooling = .85

$T_{sat\ evap} - T_{sat\ cond}$	Change in Flow or Capacity (%)	Change in Power (%)	Change in Overall Compressor Efficiency (%)
50 - 145	-1.8	+3.7	-2
50 - 120	-2.9	+5.3	-5
50 - 80	-1.3	+19.9*	-12
20 - 145	-2.1	+1.4	-3
20 - 120	-2.1	+3.2	-3
20 - 80	-1.9	+9.3*	-7
-10 - 120	-3.8	+.9	-3
-10 - 80	-3.6	+4.7	-6

* - Due Mostly to % Increase in Compression Work At low Pressure Ratio

EFFECT OF VARYING θs FROM 0^{o} ABDC TO 20^{o} ABDC WITH:

ΔP_s = 1 psi $\theta_D = 0$ η_{mech} = .96 ΔT_{suct} = 0

ΔP_D = 10 psi η_{is} = .94 % Oil = 0 % Motor cooling = .85

$T_{sat\ evap}$	-	$T_{sat\ cond}$	Change in Flow Or Capacity (%)	Change in Power (%)	Change in Overall Compressor Efficiency
50	-	145	-2.7	-3.9	0
50	-	120	-3.0	-3.6	0
50	-	80	-3.1	-3.2	0
20	-	145	-3.7	-3.9	0
20	-	120	-3.4	-3.6	0
20	-	80	-3.2	-3.2	0
-10	-	120	-4.1	-3.8	0
-10	-	80	-3.8	-3.0	-1

EFFECT OF VARYING θ_D FROM 0^{o} ATDC TO 10^{o} ATOC WITH:

ΔP_s = 1 psi $\theta_s = 0$ η_{mech}=.96 ΔT_{suct} = 0

ΔP_D = 10 psi η_{is}= .94 % Oil = 0 % Motor Cooling =.85

$T_{sat\ evap}$		$T_{sat\ cond}$	Change in Flow or Capacity (%)	Change in Power (%)	Change in Overall Compressor Efficiency
50	-	145	-2.8	-2.8	0
50	-	120	-2.0	-2.1	0
50	-	80	-1.2	-1.2	0
20	-	145	-5.4	-4.5	-1
20	-	120	-3.9	-2.6	0
20	-	80	-2.1	-1.8	0
-10	-	120	-8.1	-5.8	-1
-10	-	80	-4.6	-2.7	-2

EFFECTS OF VARYING η_{is} FROM 94% TO 98% WITH:

ΔP_s = 1 psi $\theta_s = 0$ $\eta_{mech} = .96$ $\Delta T_{suct} = 0$

ΔP_D = 10 psi $\theta_D = 0$ % Oil = 0 % Motor cooling = .85

$T_{sat\ evap}$	-	$T_{sat\ cond}$	Change in Flow or Capacity (%)	Change in Power (%)	Change in Overall Compressor Efficiency (%)
50	-	145	+1.3	-7.1	+6
50	-	120	+.6	-6.3	+5
50	-	80	+.1	-4.8	+4
20	-	145	+1.5	-7.9	+6
20	-	120	+1.0	-6.7	+6
20	-	80	+.3	-5.3	+4
-10	-	120	+1.9	-8.9	+8
-10	-	80	+.3	-5.5	+4

EFFECTS OF VARYING η_{mech} FROM 94% TO 98% WITH:

ΔP_s = 1 psi $\theta_s = 0$ $\eta_{is} = .94$ $\Delta T_{suct} = 0$

ΔP_D = 10 psi $\theta_D = 0$ % Oil = 0 % Motor cooling = .85

$T_{sat\ evap}$		$T_{sat\ cond}$	Change in Flow or Capacity (%)	Change in Power (%)	Change in Overall Efficiency (%)
50	-	145	+1.7	-4.8	+4
50	-	120	+1.3	-4.6	+5
50	-	80	+.4	-3.9	+3
20	-	145	+2.2	-4.0	+4
20	-	120	+1.6	-4.3	+4
20	-	80	+ .9	-4.2	+4
-10	-	120	+2.4	-4.1	+4
-10	-	80	+1.4	-3.7	+3

EFFECTS OF VARYING % MOTOR COOLING FROM 80% TO 100% WITH:

ΔP_s = 1 psi $\theta_s = 0$ $\eta_{is} = .94$ % Oil = 0

ΔP_D = 10 psi $\theta_D = 0$ $\eta_{mech} = .96$ $\Delta T_{suct} = 0$

$T_{sat\ evap}$ - $T_{sat\ cond}$	Change in Flow Or Capacity (%)	Change in Power (%)	Change in Overall Compressor Efficiency (%)
50 - 145	-1.5	+ .3	-1
50 - 120	-.8	+.3	-1
50 - 80	-.3	-.2	0
20 - 145	-1.7	0	-1
20 - 120	-1.1	+.1	0
20 - 80	-.8	+.1	0
-10 - 120	-1.7	-.3	-1
-10 - 80	-.9	+.1	-1

EFFECTS OF VARYING % OIL CIRCULATION FROM 0% TO 10% BY WEIGHT

ΔP_s = 1 psi $\theta_s = 0$ $\eta_{is} = .94$ % Motor Cooling = .85

ΔP_D = 10 psi $\theta_D = 0$ $\eta_{mech} = .96$ $\Delta T_{suct} = 0$

$T_{sat\ evap}$ - $T_{sat\ cond}$	Change in * Capacity (%)	Change in * Power (%)	Change in Overall Compressor Efficiency (%)
50 - 145	-13.3	+.4	0
50 - 120	-9.6	+.4	0
50 - 80	-5.5	+.3	0
20 - 145	-15.2	+.4	0
20 - 120	-11.1	+.3	0
20 - 80	-6.6	+.2	0
-10 - 120	-12.2	+.2	0
-10 - 80	-7.5	+.2	0

* Oil Circulation Affects Evaporator Capacity, But Has Little Effect On Flow or Power

EFFECT OF INCREASING SUCTION GAS SUPERHEAT BY 30°F

ABOVE THAT DUE TO MOTOR COOLING AND OTHER EFFECTS WITH:

ΔP_s = 1 psi	$\theta_s = 0$	η_{is} = .94	% Oil = 0
ΔP_D = 10 psi	$\theta_D = 0$	η_{mech} = .96	% Motor Cooling = .85

$T_{sat\ evap}$ - $T_{sat\ cond}$	Change in Flow Or Capacity (%)	Change in Power (%)	Change in Overall Compressor Efficiency (%)
-10 - 120	-6.9	-.5	-4
-10 - 80	-6.6	0	-5

$T_{sat\ evap} > 0°F$

Amount of suction gas superheat due to suction-discharge heat transfer is considerably less than that at the low suction-high discharge pressure (high pressure ratio) condition given above.

APPENDIX I

DETAILS OF AIR-COOLED, CROSS-FLOW CONDENSER MODELING

Details of the general air-cooled, cross-flow condenser model, 'EXCH', and of the special case, finned tube condenser model, are given in this section, followed by computer program listings for each.

General Model 'EXCH'

The effectiveness-NTU method of heat transfer analysis is described below:

$$\varepsilon \equiv \frac{\dot{q}_{actual}}{\dot{q}_{max\ possible}} = \frac{C_H\ (T_{H_{in}} - T_{H_{out}})}{C_{min}\ (T_{H_{in}} - T_{c_{in}})} = \frac{C_c\ (T_{c_{out}} - T_{c_{in}})}{C_{min}\ (T_{H_{in}} - T_{c_{in}})}$$

Where:

$\dot{q}$ = Heat transfer rate

ε = Effectiveness

C_H = $(\dot{m}\ C_p)$ of the hotter fluid

C_C = $(\dot{m}\ C_p)$ of the colder fluid

$\dot{m}$ = Mass flow rate

C_p = Specific heat at constant pressure

C_{min} = the smaller of C_H and C_C

$T_{H_{in}}$ = entering temperature of the hotter fluid

$T_{H_{out}}$ = exit temperature of the hotter fluid

$T_{C_{in}}$ = entering temperature of the colder fluid

$T_{C_{out}}$ = exit temperature of the colder fluid

In general, effectiveness can be expressed by a relation of the form:

$$\varepsilon = f\left(\frac{C_{min}}{C_{max}},\ NTU\right)$$

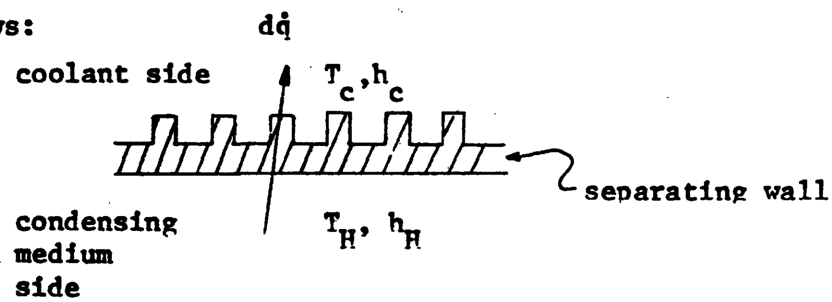
Where:

C_{max} = The larger of C_H and C_C

$NTU = \dfrac{AU}{C_{min}}$

AU = Overall conductance for heat transfer

A general expression for overall conductance, allowing for the possibility of an extended surface on the coolant side can be developed as follows:



$$d\dot{q} = U dA\ \Delta T = U\ dA\ (T_H - T_c)$$

$$\frac{1}{U\ dA} = \frac{1}{\eta_o\ h_c\ dA_{C_{H.T.}}} + \frac{1}{h_H\ dA_{H_{H.T.}}}$$ assuming resistance of material separating fluids is negligible

Where:

η_o = overall surface efficiency of extended surfaces, including contact resistance, as described in Appendix J.

$d\dot{q}$ = local heat transfer (Btu/hr)

$dA_{c_{H.T.}}$ = unit heat transfer area on coolant side (ft^2)

$dA_{H_{H.T.}}$ unit heat transfer area on condensing medium side (ft^2)

$h_C \equiv$ heat transfer coefficient on coolant side (Btu/hr-ft^2-^oF)

$$\frac{1}{h_C} = \frac{1}{h_{fluid}} + \frac{1}{h_{scale}}$$

$h_H \equiv$ heat transfer coefficient on condensing medium side (Btu/hr-ft^2-^oF)

$$\frac{1}{h_H} = \frac{1}{h_{fluid}} + \frac{1}{h_{scale}}$$

The general expression for NTU is hence:

$$NTU = \frac{1}{C_{min}\left[\frac{1}{\eta_o\ h_c\ dA_{c_{HT}}} + \frac{1}{h_H\ dA_{H_{HT}}}\right]}$$

or

$$NTU = \frac{dA_{H_{HT}}}{C_{min}\left[\frac{\alpha_H}{\eta_o\ h_c\ \alpha_c} + \frac{1}{h_H}\right]}$$

where:

$\alpha_c \equiv$ heat transfer area on coolant side/total heat exchanger volume (1/ft)

$\alpha_H \equiv$ heat transfer area on condensing medium side/total heat exchanger volume (1/ft)

$C_{min} = (d\dot{m}\ C_P)_{min}$

The procedure for determining desuperheating, condensing, and subcooling region performance, as shown in the flow diagram in Figure I-2, is as follows:

Determine NTU_{tp}

$$NTU_{tp} = \frac{AOM}{C_{p_c}\left[\frac{\alpha_H}{\eta_o\ h_c \alpha_c} + \frac{1}{h_{H_{tp}}}\right]}$$

Where:

$$AOM \equiv \frac{dA_{H_{HT}}}{d\ \dot{m}_c}$$

'tp' = subscript indicating two-phase region

C_{p_c} = specific heat at constant pressure for the colder fluid

$d\ \dot{m}_c$ = local flow rate of coolant (LBM/HR)

Then:

$$\varepsilon_{tp} = 1 - e^{-NTU_{tp}}$$

Next, determine the bulk superheated vapor temperature 'T_{ds}' at the end of the desuperheating region when condensation begins.

$$d\dot{q} = h_{H_v}\ dA_{H_{HT}}\ (T_{ds} - T_{wall})$$

also

$$d\dot{q} = UdA\ (T_{ds} - T_{c_{in}})$$

Where:

h_{H_v} = single phase vapor (superheated vapor) heat transfer coefficient on condensing medium side (Btu/hr-ft^2-^{o}F)

$T_{wall} \equiv T_{H_{sat}}$ = saturation temperature of the condensing medium (^{o}F)

$T_{C_{in}}$ = entering temperature of the coolant (^{o}F)

Equating the two expressions for $d\dot{q}$ we get:

$$h_{H_v}\ dA_{H_{HT}}\ (T_{ds} - T_{H_{sat}}) = \frac{dA_{H_{HT}}}{\left[\frac{\alpha_H}{\eta_o\ \alpha_c\ h_c} + \frac{1}{h_{H_v}}\right]}\ (T_{ds} - T_{c_{in}})$$

rearranging we find:

$$T_{ds} = \frac{[(T_{H_{sat}})\ (RES) - T_{c_{in}}]}{[RES - 1]}$$

Where:

$$RES \equiv \left[1 + \frac{\alpha_H\ h_{H_v}}{\alpha_c\ h_c\ \eta_o}\right]$$

Now the driving enthaply differential for heat transfer in the two-phase region h_{fg}'' can be determined:

$$h_{fg}'' = [h_{fg} + C_{p_{H_v}}\ (T_{ds} - T_{H_{sat}})]\ (1 - X_3)$$

Where:

h_{fg} ≡ latent heat of vaporization of the condensing medium (Btu/lbm)

$C_{p_{H_v}}$ = specific heat at constant volume of superheated vapor on condensing medium side (Btu/lbm-^{o}R)

X_3 = exit quality of condensing medium, if we choose to study cases of incomplete condensation.

Next, we use the definition of effectiveness to determine the amount of coolant passing over the two-phase region of the heat exchanger:

$$\dot{m}_{c_{tp}} = \frac{(\dot{m}_H)\,(h_{fg}{''})}{(\varepsilon_{tp})\,(C_{p_c})\,(T_{H_{sat}} - T_{c_{in}})}$$

Where:

$\dot{m}_H$ = total flow rate of condensing medium (lbm/hr)

The fraction of the heat exchanger 'F_{tp}' which is used for the two-phase region is hence:

$$F_{tp} = \frac{\dot{m}_{c_{tp}}}{\dot{m}_c}$$

Where $\dot{m}_c$ = total coolant flow rate (lbm/hr)

Now we seek to determine the fraction of heat exchanger 'F', which is required for desuperheating. An iterative procedure is required, as follows:

Guess F

$$\dot{m}_{c_{sp}} = (F)\ (\dot{m}_c)$$

$$C_{c_{sp}} = (\dot{m}_{c_{sp}})\ (C_{p_c})$$

$$C_{H_{sp}} = (\dot{m}_H)\ (C_{p_{H_v}})$$

$$C_{min} = \text{Smaller of } C_{c_{sp}} \text{ and } C_{H_{sp}}$$

$$C_{max} = \text{Larger of } C_{c_{sp}} \text{ and } C_{H_{sp}}$$

$$R_{tot} = \frac{1}{UA} = \frac{\left[\dfrac{\alpha_H}{\eta_o\ \alpha_c\ h_c} + \dfrac{1}{h_{H_{sp_v}}}\right]}{(F)\ (A_{H_{ht}})}$$

$$NTU = \frac{1}{(R_{tot})\ (C_{min})}$$

$$\varepsilon_{XF} = f\left(\frac{C_{min}}{C_{max}},\ NTU\right)$$

$$\varepsilon_{XF}^{*} = \frac{C_H (T_{H_{in}} - T_{ds})}{C_{min}\ (T_{H_{in}} - T_{c_{in}})}$$

If $\varepsilon_{XF} = \varepsilon_{XF}^{*}$, repeat

Where:

$\dot{m}_{C_{sp}}$ = flow rate of coolant passing over the desuperheating region (lbm/hr)

$T_{H_{in}}$ = temperature of superheated vapor entering condenser (^oF)

R_{tot} = overall resistance to heat transfer in the desuperheating region

ε_{XF} = cross-flow effectiveness as determined from the proper expression or chart. (See Appendix J for cross-flow effectiveness, both fluids unmixed)

ε_{XF}^* = effectiveness as determined from the definition of effectiveness

$A_{H_{ht}}$ = total condensing side heat transfer area (ft^2)

'sp' = subscript indicating single phase vapor region

Then:

$$F_{sc} = 1 - F - F_{tp}$$

Where:

'sc' = subscript indicating subcooled liquid region

If $F_{sc} \leq 0$, then there was incomplete condensation in the two-phase region. In the present model, calculations are terminated if incomplete condensation occurs. It is possible, however, with only slight modifications, to use the present model for the case of incomplete condensation. This is done merely by iterating on X_3 , the exit quality.

If $F_{sc} > 0$, determine exit temperature of the subcooled liquid:

$$C_{H_{sc}} = (\dot{m}_H)\,(C_{p_{H_\ell}})$$

$$C_{C_{sc}} = (F_{sc})\,(\dot{m}_C)\,(C_{p_C})$$

C_{min} = smaller of $C_{H_{sc}}$ and $C_{C_{sc}}$

C_{max} = larger of $C_{H_{sc}}$ and $C_{C_{sc}}$

$$R_{tot} = \frac{\left[\dfrac{\alpha_H}{\eta_o\ \alpha_c\ h_c} + \dfrac{1}{h_{H_{sc}}}\right]}{(F_{sc})\ (A_{H_{ht}})}$$

$$NTU = \frac{1}{(R_{tot})(C_{min})}$$

$$\varepsilon_{\chi F_{sc}} = f\left(\frac{C_{min}}{C_{max}}\ ,\ NTU\right)$$

$$T_{H_{out}} = T_{H_{sat}} = \frac{(\varepsilon_{\chi F_{sc}})\ (C_{min})\ (T_{H_{sat}} - T_{c_{in}})}{C_{H_{sc}}}$$

Where:

$T_{H_{out}}$ = temperature of subcooled liquid leaving condenser (^oF)

$C_{p_{H_\ell}}$ = specific heat at constant pressure of subcooled liquid (Btu/lbm-^oR)

Finally, we can determine heat transfer rates and exit coolant temperatures for each region.

$$Q_{sc} = C_{H_{sc}} (T_{H_{sat}} - T_{H_{out}}) \qquad \text{(Btu/hr)}$$

$$Q_{tp} = \dot{m}_H \, h_{fg}'' \qquad \text{(Btu/hr)}$$

$$Q_{sp} = C_{H_{sp}} (T_{H_{in}} - T_{ds}) \qquad \text{(Btu/hr)}$$

$$Q_{tot} = Q_{sc} + Q_{tp} + Q_{sp} \qquad \text{(Btu/hr)}$$

$$T_{c_{out_{sp}}} = \frac{Q_{sp}}{C_{c_{sp}}} + T_{c_{in}} \qquad (^{o}F)$$

$$T_{c_{out_{tp}}} = \frac{Q_{tp}}{(\dot{m}_{c_{tp}})(C_{p_c})} + T_{c_{in}} \qquad (^{o}F)$$

$$T_{c_{out_{sc}}} = \frac{Q_{sc}}{C_{c_{sc}}} + T_{c_{in}} \qquad (^{o}F)$$

$$T_{c_{out_{avg}}} = \frac{Q_{tot}}{(\dot{m}_c)\,(C_{p_c})} + T_{c_{in}} \qquad (^{o}F)$$

For more information, see comments in the program listing for subroutine 'EXCH' at the end of this section.

Modeling a Finned Tube Condenser

The geometry factors necessary for use of general model 'EXCH' are: α_H , α_c , AOM , $A_{H_{ht}}$, $A_{C_{ht}}$, $A_{C_{flow}}$, and FAR. For the finned tube condenser case, shown in Figure I-1, the above geometry factors are developed as follows:

$$h = (NP - 1)\ s + 2\ (\tfrac{s}{2}) = (NP)\ (s)$$

$$t = (NT - 1)\ w + 2\ (\tfrac{w}{2}) = (NT)\ (w)$$

$$A_{c_{flow}} = A_{\substack{air\\flow}} = NP\ (s - D_o)\ L\ [1 - (\delta)\ (FP)]$$

$$A_{\substack{c\\frontal}} = A_{\substack{air\\frontal}} = (h)\ (L) = (NP)\ (s)\ (L)$$

$$\sigma_c = \sigma_{air} = \frac{A_{\substack{air\\flow}}}{A_{\substack{air\\frontal}}} = \frac{dA_{\substack{air\\flow}}}{dA_{\substack{air\\frontal}}}$$

$$\sigma_A = \frac{s - D_o}{s}\ [1 - \delta\ (FP)]$$

$$A_{c_{\substack{heat\\trans}}} = A_{\substack{air\\heat\\trans}} = (NP)(NT)(L)\ \{[(s)(w) - \frac{\pi\ D_o^2}{4}](2)(FP) + [1-(FP)(\delta)](\pi\ D_o)\}$$

$$\text{Volume of heat exchanger} = (h)\ (L)\ (t) = (NP)\ (s)\ (L)\ (NT)\ (w)$$

$$\alpha_c = \frac{A_{air_{ht}}}{\text{volume}} = \frac{(2)\ (FP)\ [(s)(w) - \frac{\pi\ D_o^2}{4}] + [1 - (FP)\ (\delta)]\ (\pi\ D_o)}{(s)\ (w)}$$

$$A_{\substack{air\\ht\ fins\\only}} = (2)\ (FP)\ [(s)(w) - \frac{\pi\ D_o^2}{4}]$$

$$\mathrm{FAR} \equiv \frac{A_{\substack{\text{fins}\\ \text{ht}}}}{A_{\substack{\text{air ht}\\ \text{total}}}} = \frac{(2)\ (FP)\ [(s)(w) - \frac{\pi\ D_o^{\ 2}}{4}]}{\{(2)\ (FP)\ [(s)(w) - \frac{\pi\ D_o^{\ 2}}{4}] + [1 - (FP)(\delta)](\pi\ D_o)\}}$$

$$A_{H_{ht}} = A_{\substack{\text{cond. ht}\\ \text{inside}}} = (NT)\ (\pi)\ (D_i)\ (L)\ (NP)$$

$$\alpha_H \equiv \frac{A_{H_{ht}}}{\text{volume}} = \frac{\pi\ D_i}{(s)(w)}$$

$$A_{H_{flow}} = \frac{\pi\ D_i^{\ 2}}{4}$$

$$\mathrm{AOM} \equiv \frac{dA_{H_{ht}}}{d\ \dot{m}_c} = \frac{(NT)\ (\pi)\ (D_i)\ (NP)\ (dL)}{(\rho_{air})(CFM_{air})\left(\frac{dA_{\substack{air\\ flow}}}{A_{air}}\right)}$$

But

$$G_A \equiv \frac{\dot{m}_{air}}{A_{\substack{air\\ flow}}} = \frac{(\rho_{air})(CFM_{air})}{(NP)\ (s - D_o)[1 - \delta\ (FP)]L}$$

$$\sigma_A \equiv \frac{A_{\substack{air\\ flow}}}{A_{\substack{air\\ frontal}}} = \frac{(NP)\ (s - D_o)\ [1 - \delta(FP)]L}{(NP)\ (s)\ (L)}$$

Hence

$$\mathrm{AOM} = \frac{(NT)\ (\pi)\ (D_i)\ (NP)\ (dL)}{G_A\ \sigma_A\ (\frac{dL}{L})\ (NP)\ (s)\ (L)}$$

$$AOM = \frac{(NT)\ (\pi)\ (D_i)}{(G_A)\ (\sigma_A)\ (s)}$$

Having determined the geometry factors, the procedures for determining total performance, as outlined in the flow chart of Figure I-3, is as follows:

Split the condenser up into equivalent sub-circuits.

$$\dot{m}_c = \frac{\dot{m}_c}{N_{sect}}$$

$$\dot{m}_H = \frac{\dot{m}_H}{N_{sect}}$$

$$A_{H_{ht}} = \frac{A_{H_{ht}}}{N_{sect}}$$

Where:

N_{sect} = Number of parallel flow sub-circuits in the heat exchanger

Then:

Using thermodynamic properties corresponding to the states of interest, determine the heat transfer coefficients as described in Appendix K.

Next, use general condenser model 'EXCH' to determine performance, for the given geometry factors, temperatures, and flow rates.

And, using results from 'EXCH',

Determine the length of the desuperheating, two-phase, and subcooling regions:

$$DZTP = \frac{(F_{tp})\ (A_{H_{ht}})}{(\pi\ D_i)} \text{ (ft) (two-phase)}$$

$$DZV = \frac{(F)\ (A_{H_{ht}})}{(\pi\ D_i)} \text{ (ft) (desuperheating)}$$

$$DSL = \frac{(F_{sc})\ (A_{H_{ht}})}{(\pi\ D_i)} \text{ (ft) (subcooling)}$$

Where:

D_i = inside diameter of tubes in heat exchanger

Determine total pressure drop 'PD' as described in Appendix L.

Convert results back to total flow notation

$$\dot{m}_c = \dot{m}_c\ (N_{sect})$$

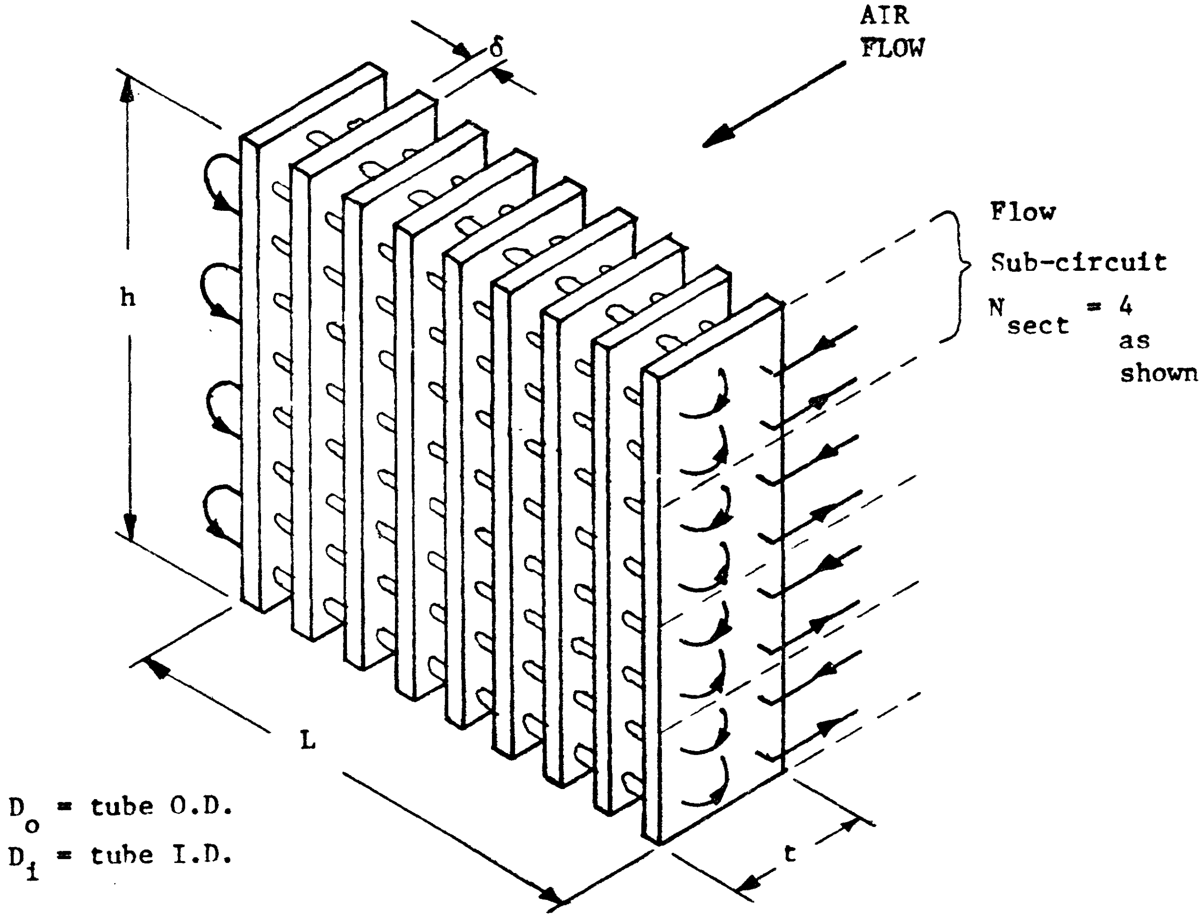
$$\dot{m}_H = \dot{m}_H\ (N_{sect})$$

$$Q_{tot} = Q_{tot}\ (N_{sect})$$

Finally, using the value of total pressure drop through the coil, determine the saturation temperature of the condensing medium leaving the condenser. If the drop in condensing temperature is greater than $2^{o}F$, repeat the analysis using

$$T_{sat_{avg}} = \frac{(T_{sat_{in}} + T_{sat_{out}})}{2}$$

For more information, see comments in the program listing for the finned tube condenser simulation at the end of this section.



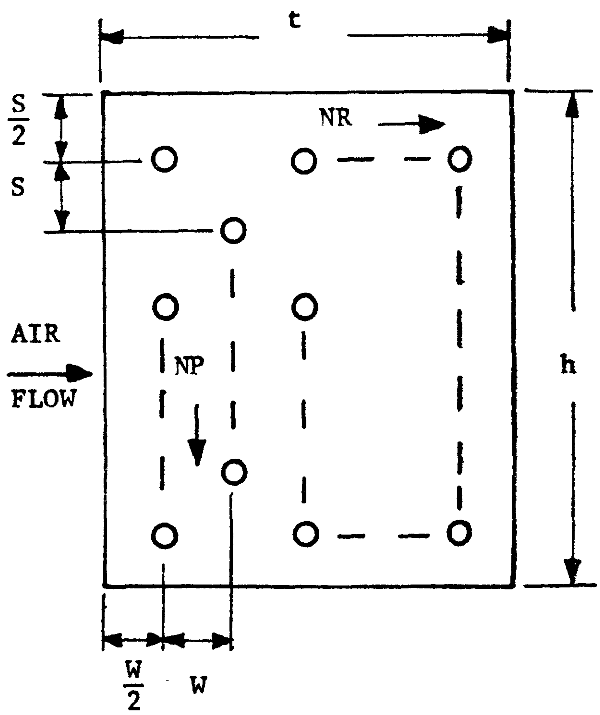


FIGURE I-1

DIMENSIONS FOR FINNED TUBE HEAT EXCHANGER WITH STAGGERED ROUND TUBES

FIGURE I-2

FLOW CHART FOR GENERAL CONDENSER MODEL - 'EXCH'

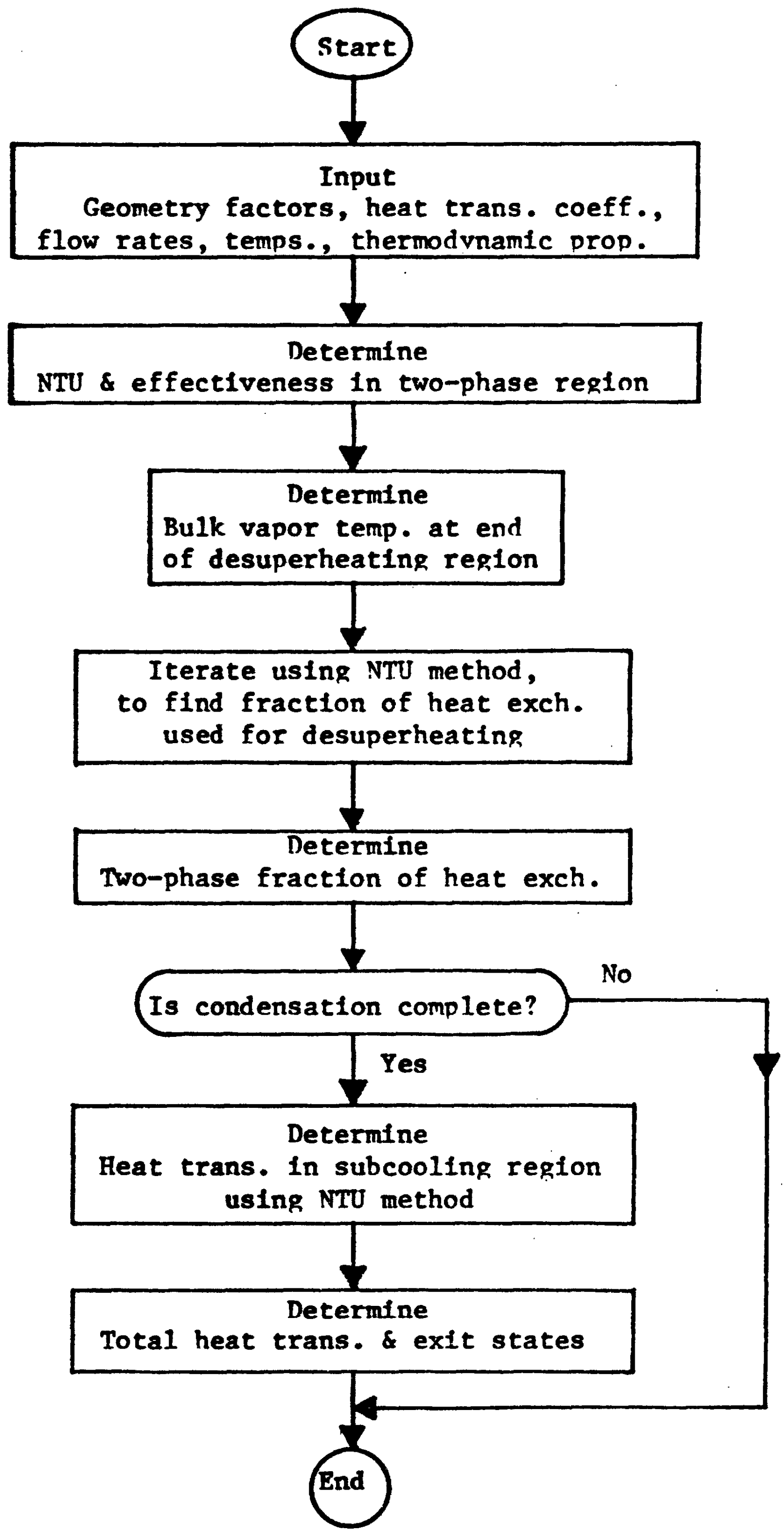
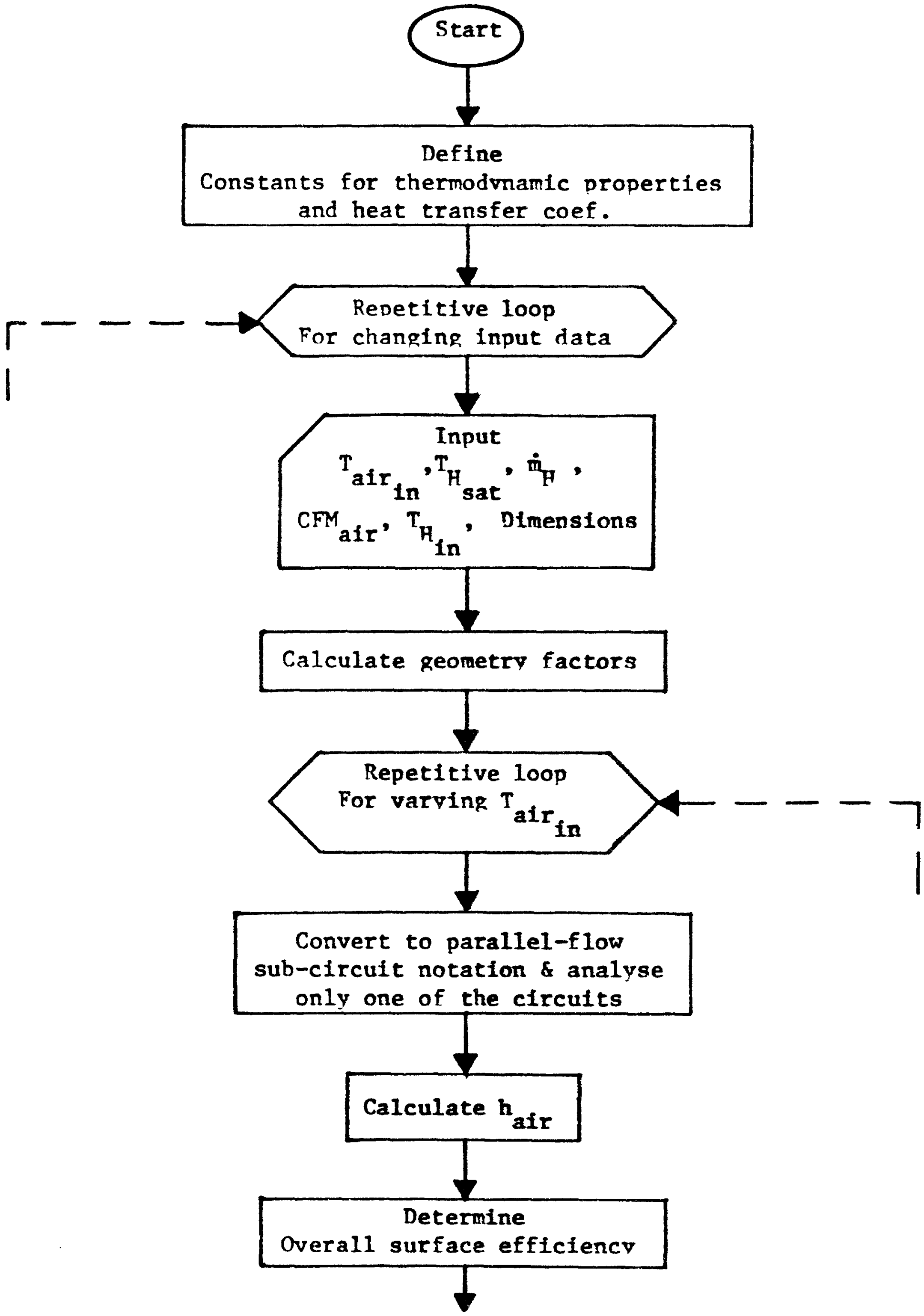
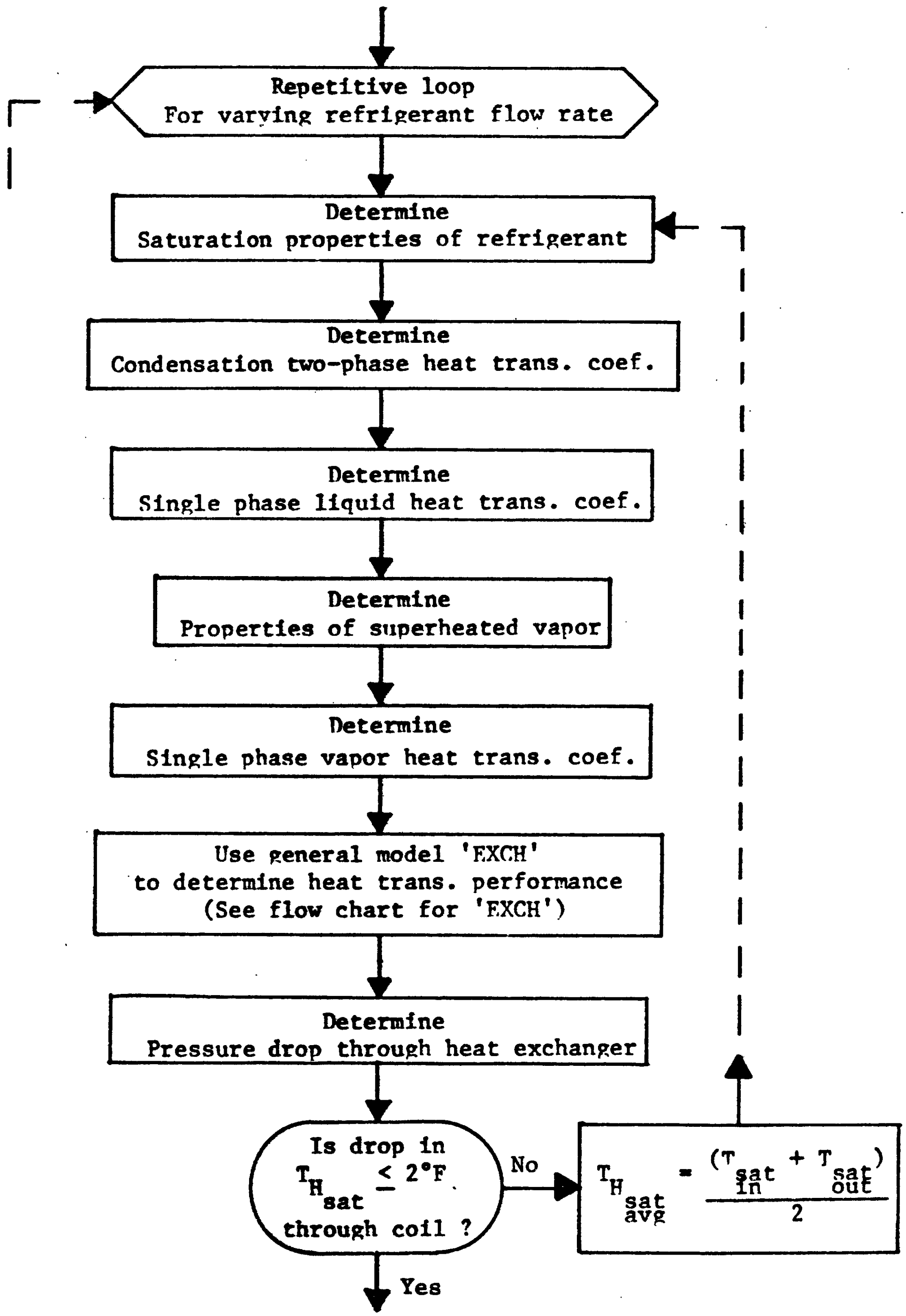
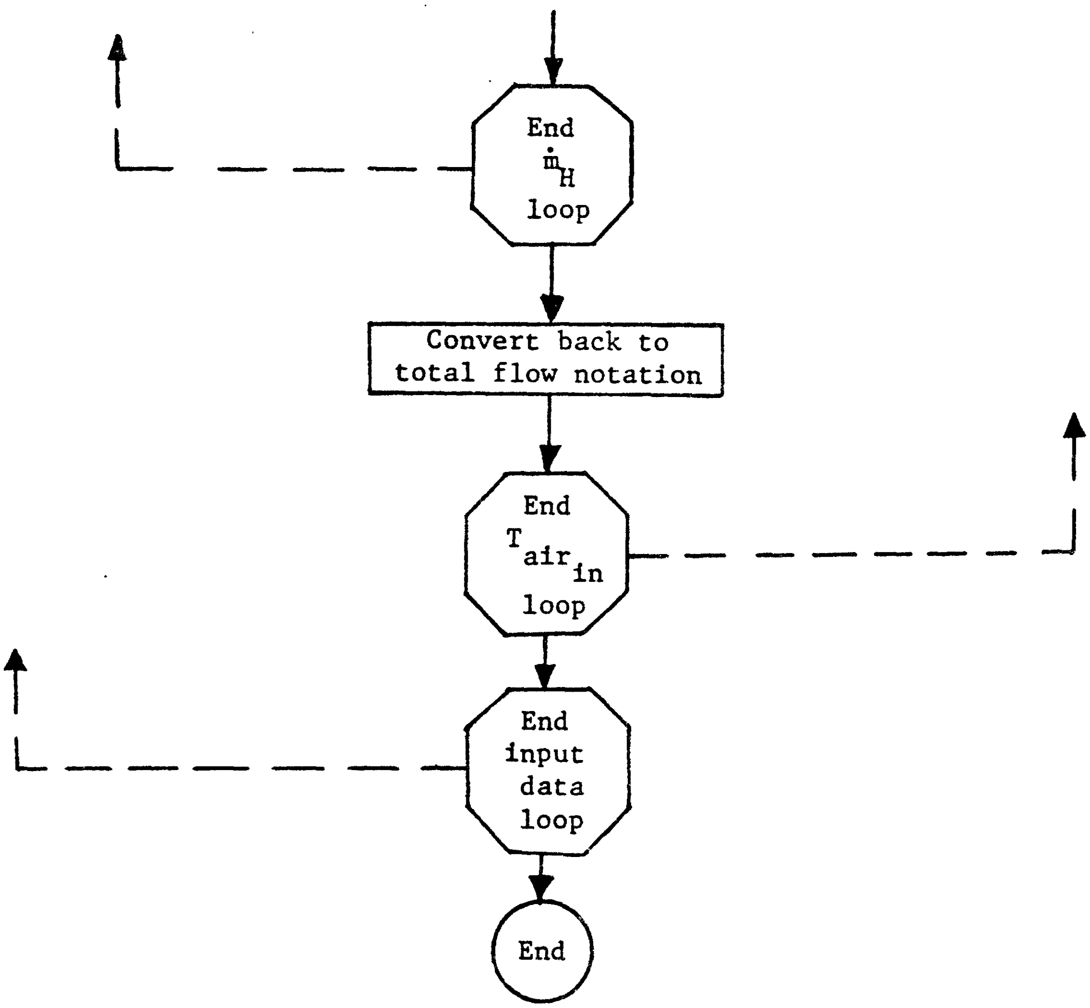


FIGURE I-3

FLOW CHART FOR FINNED TUBE CONDENSER MODEL




Repetitive loop
For varying refrigerant flow rate
Determine
Saturation properties of refrigerant
Determine
Condensation two-phase heat trans. coef.
Determine
Single phase liquid heat trans. coef.
Determine
Properties of superheated vapor
Determine
Single phase vapor heat trans. coef.
Use general model 'EXCH'
to determine heat trans. performance
(See flow chart for 'EXCH')
Determine
Pressure drop through heat exchanger
Is drop in
$T_{H_{sat}} \leq 2°F$
through coil ?
No
$T_{H_{sat\,avg}} = \frac{(T_{sat_{in}} + T_{sat_{out}})}{2}$
Yes


End
$\dot{m}_H$
loop
Convert back to total flow notation
End
$T_{air_{in}}$
loop
End
input
data
loop
End

```
      SUBROUTINE EXCH(M,ACM,ALFAR,ALFAA,TH,TC,HFG)
C
C     PURPOSE
C        TO DETERMINE THE HEAT TRANSFER AND RESULTING
C        TEMPERATURES IN THE CONDENSER, GIVEN ALL OF THE
C        NECESSARY COEFFICIENTS AND OTHER DETAILS
C
C     DESCRIPTION OF PARAMETERS
C      INPUTS
C        HEAT EXCHANGER GEOMETRY
C          ACM    -  UNIT REFRIGERANT SIDE HEAT TRANSFER
C                    AREA/UNIT AIR FLOW RATE
C                    (SQ FT-HR/LBM DRY AIR)
C          ALFAR  -  ALPHA REFRIGERANT (REFRIGERANT SIDE
C                    HEAT TRANSFER AREA/TOTAL VOLUME OF HEAT
C                    EXCHANGER-1/FT)
C          ALFAA  -  ALPHA AIR (AIR SIDE HEAT TRANSFER AREA/
C                    TOTAL VOLUME OF HEAT EXCHANGER-1/FT)
C          ARHT   -  TOTAL REFG. SIDE HEAT TRANS. AREA (SQ FT)
C
C        HEAT TRANSFER COEFFICIENTS
C          HA     -  AIR SIDE HEAT TRANS.COEF.(BTU/HR-SQ FT-F)
C          HRTP   -  REF. SIDE TWO-PHASE HEAT TRANS.
C                    COEF. (BTU/HR-SQ FT-F)
C          HRSPV  -  REF. SIDE SINGLE PHASE VAPOR HEAT
C                    TRANS. COEF. (BTU/HR-SQ FT-F)
C          HRSPL  -  REF. SIDE SINGLE PHASE LIQUID HEAT
C                    TRANS. COEF. (BTU/HR-SQ FT-F)
C
C        REFRIGERANT PROPERTIES
C          TRI    -  TEMP. OF REFG. ENTERING CONDENSER (F)
C          TH     -  SATURATED CONDENSING TEMP. OF REFG. (F)
C          HFG    -  LATENT ENTHALPY OF VAPORIZATION
C                    THE REFRIGERANT (BTU/LBM)
C          CPRV   -  SPECIFIC HEAT AT CONSTANT PRESSURE
C                    OF THE REFRIGERANT VAPOR (BTU/LBM-R)
C          CPRL   -  SPECIFIC HEAT AT CONSTANT PRESSURE OF
C                    THE REFRIGERANT LIQUID (BTU/LBM-R)
C          X3     -  EXIT QUALITY FROM THE CONDENSER
C          XMR    -  MASS FLOW RATE OF REFRIGERANT (LBM/HR)
C
C        AIR PROPERTIES
C          CPA    -  SPECIFIC HEAT AT CONSTANT PRESSURE OF
C                    THE AIR (BTU/LBM-R)
C          TC     -  TEMP. OF AIR ENTERING CONDENSER (F)
C          XMA    -  MASS FLOW RATE OF AIR (LBM/HR)
C
C        OTHER INPUTS
C          M      -  AN INDICATOR (NOT USED HERE)
```

```
C            SEFFX  -  SURFACE EFFICIENCY OF FINNED SURFACE
C
C       OUTPUTS
C            F      -  SINGLE PHASE VAPOR FRACTION OF TOTAL
C                      HEAT EXCHANGER SURFACE
C            FTP    -  TWO-PHASE FRACTION OF TOTAL HEAT
C                      EXCHANGER SURFACE
C            FSC    -  SUBCOOLING FRACTION OF TOTAL HEAT
C                      EXCHANGER SURFACE
C            GSP    -  HEAT TRANSFER RATE IN SINGLE PHASE
C                      VAPOR REGION (BTU/HR)
C            GTP    -  HEAT TRANSFER RATE IN TWO-PHASE
C                      REGION (BTU/HR)
C            GSC    -  HEAT TRANSFER RATE IN SUBCOOLING
C                      REGION (BTU/HR)
C            TACSP  -  AIR TEMPERATURE OUT OF SINGLE PHASE
C                      VAPOR REGION (F)
C            TACTP  -  AIR TEMPERATURE OUT OF TWO-PHASE
C                      REGION (F)
C            TACSC  -  AIR TEMPERATURE OUT OF SUBCOOLING
C                      REGION (F)
C            TAC    -  AVERAGE, MIXED AIR TEMPERATURE LEAVING
C                      THE CONDENSER (F)
C            TRC    -  TEMP.OF REFRIGERANT LEAVING HEAT EXCH.(F)
C
C      REMARKS
C         SUBROUTINE EXF IS CALLED BY THIS PROGRAM TO
C         DETERMINE THE EFFECTIVENESS IN CROSSFLOW
C         (THIS PROGRAM USES THE EFFECTIVENESS-NTU METHOD
C         FOR CALCULATING HEAT TRANSFER PERFORMANCE)
C
       COMMON CPA,HA,SEFFX,HRSPV,HRSPL,CPRL,CPRV,XMR,XMA,
      1X3,TRI,HRTP,F,FTP,FSC,GSP,GTP,GSC,TAOSP,TAOTP,TAOSC,
      1TAC,TRC,ARHT
C
C      DETERMINE NTU AND EFFECTIVENESS FOR TWO-PHASE REGION
C
       XNTUTP=ACM/(CPA*(ALFAR/(SEFFX*HA*ALFAA)+1.0/HRTP))
       ETP = 1.0 - EXP(-XNTUTP)
       RES = 1.0 + HRSPV*ALFAR/(HA*ALFAA*SEFFX)
C
C      FIND THE REFG.TEMP.'TRVDS' (F) AT THE END OF THE
C      DESUPERHEATING OR SINGLE PHASE VAPOR REGION
C
       TRVDS = (TH*RES - TC)/(RES-1.0)
       IF(TRVDS.GE.TRI) TRVDS = TRI
C
C      CALCULATE HFG DOUBLE PRIME 'HFGDP' - THE EFFECTIVE
C      DRIVING ENTHALPY DIFFERENCE IN THE TWO-PHASE REGION
C
```

```
      HFGDP = (HFG + CPRV*(TRVDS - TH))*(1.0-X3)
      XMATP = XMR*HFGDP/(ETP*CPA*(TH-TC))
      WRITE(5,570) XNTUTP,ETP,TRVDS,HFGDP,XMATP
      F = 0.0
      CA = 1.0
      CR = 1.0
      IF(ABS(TRVDS-TRI).LE.2.0) GO TO 60
      IF(X3.GT.0.0) GO TO 100
      PS = 0.0
      PR = PS
      N = 0
C
C     ITERATE TO FIND THE FRACTION OF TOTAL HEAT EXCHANGER
C     SURFACE USED FOR THE DESUPERHEATING OR SINGLE PHASE
C     VAPOR REGION
C
      DF = .01
      DO 50 I=1,100
    5 F = F + DF
      XMASP = F*XMA
      CA = XMASP*CPA
      CR = XMR*CPRV
      RTCT = (ALFAR/(EEFFX*HA*ALFAA) + 1.0/HRSPV)/(F*ARHT)
      CALL EXF(RTCT,CA,CR,CMIN,EXFR)
      EXFS=CR*(TRI-TRVDS)/(CMIN*(TRI-TC))
      WRITE(5,520) F,XMA,ARHT,XMASP
      WRITE(5,530) CA,CR,CMIN
      WRITE(5,535) EXFR,EXFS,PR,PS
      IF(ABS(EXFR-EXFS).LE.(.03*EXFR)) GO TO 60
      IF(I.EQ.1) GO TO 20
      IF((PR-PS)/(EXFR-EXFS)) 60,15,15
   15 IF((ABS(PR-PS).LT.ABS(EXFR-EXFS)).AND.(I.EQ.2))GO TO 12
      IF(ABS(PR-PS).LT.ABS(EXFR-EXFS)) GO TO 55
      GO TO 20
   12 F = F - 2.0*DF
      DF = DF/2.0
      I = 1
      N = N + 1
      IF(N.GT.10) GO TO 55
      GO TO 5
   20 PR = EXFR
      PS = EXFS
   50 CONTINUE
   55 WRITE(5,500) M
      GO TO 200
   60 GSC = 0.0
      CASC = 1.0
      TACSC = 5000.0
      TRC = 5000.0
C
```

```
C      CALCULATE THE TWO-PHASE AND SUBCOOLING FRACTIONS
C      OF TOTAL HEAT EXCHANGER SURFACE
C
C      IF THE SUBCOOLING FRACTION IS LESS THAN ZERO - PRINT
C      AN ERROR MESSAGE BECAUSE CONDENSATION IS INCOMPLETE
C
 100   FTP = XMATP/XMA
       FSC = 1.0-F-FTP
       WRITE(5,590) FTP,FSC
       IF(FSC) 105,120,110
  105  WRITE(5,595)
       GO TO 250
 110   IF(X3.GT.0.0) GO TO 120
       CRSC = XMR*CPRL
       CASC = FSC*XMA*CPA
       RTOT = (ALFAR/(SEFFX*HA*ALFAA) + 1.0/HRSPL)/(FSC*ARHT)
       CALL EXF(RTOT,CASC,CRSC,CMIN,EXFR)
C
C      CALCULATE HEAT TRANSFER RATES AND TEMPERATURES
C
       TRC = TH - EXFR*CMIN*(TH-TC)/CRSC
       QSC = CRSC*(TH-TRC)
       WRITE(5,620) CRSC,CASC,EXFR,TRC,QSC
 120   QSP = CR*(TRI-TRVDS)
       QTP = XMR*HFGDP
       TACSP = QSP/CA + TC
       TACTP = QTP/(XMATP*CPA) + TC
       TACSC = QSC/CASC + TC
       TAC = (QSC + QSP + QTP)/(XMA*CPA) + TC
       WRITE(5,610) QSP,QTP
       WRITE(5,615) TACSP,TACTP,TACSC,TAC
 200   WRITE(5,650) M,SEFFX,CPA
       WRITE(5,660) ALFAR,ALFAA
       WRITE(5,665) HA,HRSPV,HRSPL,HRTP
       WRITE(5,690) F,FTP,FSC
       WRITE(5,700) QSP,QTP,QSC
       WRITE(5,710) TACSP,TACTP,TACSC,TAC,TRC
 250   WRITE(5,670) CPRL,CPRV,TH,TC,HFG
       WRITE(5,680) XMR,XMA,X4,X3,ARHT
       RETURN
 500   FORMAT('0',10X,'ITERATION ON VAPOR FRACTION OF H.E.'
     1, ,' DOES NOT CONVERGE   M='I2)
 520   FORMAT(' ',10X,'F=',F5.3,5X,'XMA=',F10.4,5X,'ARHT='
     1,F10.4,5X,'XMASP=',F10.4)
 530   FORMAT(' ',10X,'CA=',F10.4,5X,'CR=',F10.4,5X,'CMIN='
     1,F10.4)
 535   FORMAT(' ',10X,'EXFR=',F4.2,5X,'EXFS=',F4.2,5X,'PR='
     1,F4.2,5X,'PS=',F4.2)
 570   FORMAT('1',5X,'XNTUTP=',F8.3,5X,'ETP=',F4.2,5X,'TRVDS='
     1,F7.2,5X,'HFGDP=',F10.4,5X,'XMATP=',F10.4)
```

```
590  FORMAT(' ',10X,'FTP=',F4.2,5X,'FSC=',F4.2)
595  FORMAT('0','******INCOMPLETE CONDENSATION******')
600  FORMAT(' ',8X,'CRSC=',F10.4,5X,'CASC=',F10.4,5X,'EXFR='
    1,F4.2,5X,'TRC=',F10.4,5X,'GSC=',F10.4)
610  FORMAT(' ',10X,'GSP=',F10.4,5X,'GTP=',F10.4)
615  FORMAT(' ',10X,'TAOSP=',F10.4,5X,'TAOTP=',F10.4,5X,
    1'TACSC=',F10.4,5X,'TAC=',F10.4)
650  FORMAT(' ',10X,'M=',I2,5X,'SEFFX=',F4.2,5X,'CPA=',F5.3)
660  FORMAT('',10X,'ALFAR=',F8.3,5X,'ALFAA=',F8.3)
665  FORMAT(' ',10X,'HA=',F10.4,5X,'HRSPV=',F10.4,5X,
    1'HRSPL=',F10.4,5X,'HRTP=',F10.4)
670  FORMAT(' ',10X,'CPRL=',F8.3,5X,'CPRV=',F8.3,5X,'TH='
    1,F7.2,5X,'TC=',F7.2,5X,'HFG=',F8.4)
680  FORMAT(' ',10X,'XMR=',F10.4,5X,'XMA=',F10.4,5X,'X4='
    1,F4.2,5X,'X3=',F4.2,5X,'ARHT=',F10.4)
690  FORMAT(' ',10X,'F=',F6.3,5X,'FTP=',F6.3,5X,'FSC=',F6.3)
700  FORMAT(' ',10X,'GSP=',F10.4,5X,'GTP=',F10.4,5X,
    1'GSC=',F10.4)
710  FORMAT(' ',10X,'TAOSP=',F7.2,5X,'TAOTP=',F7.2,5X,
    1'TACSC=',F7.2,5X,'TAC=',F7.2,5X,'TRO=',F7.2)
     END
```

```
C           CONDENSER SIMULATION PROGRAM
C
C      PROGRAM FOR COMPUTING CONDENSER PERFORMANCE
C      AIR COOLED, PLATE-FIN, CROSSFLOW TYPE
C
C      INPUT DATA FROM CARD READER (DESCRIBED FULLY BELOW)
C         NRUN,DEA,DER,DELTA,FP,XKF,AAF,GA,NT,NSECT,HCONT,
C         ST,WT,TAII,DTA,NTEMP,TSA,DXMRI,XMRI,NXMR,TRI
C
C      OUTPUT
C         QC    -  TOTAL HEAT TRANSFER RATE (BTU/HR)
C         TAC   -  AVERAGE AIR TEMPERATURE LEAVING COND. (F)
C         TRC   -  TEMPERATURE OF REFRIGERANT LEAVING COND. (F)
C         HRC   -  ENTHALPY OF REFRIGERANT LEAVING COND.(BTU/LBM)
C
C      REMARKS
C         THIS PROGRAM CALLS SUBROUTINE SPHTC TO DETERMINE
C         SINGLE PHASE HEAT TRANSFER COEFFICIENTS
C         THIS PROGRAM CALLS SUBROUTINE SEFF TO DETERMINE
C         SURFACE EFFICIENCY OF FINNED SURFACE
C         THIS PROGRAM CALLS SUBROUTINE SATPRP TO DETERMINE
C         SATURATION THERMODYNAMIC PROPERTIES
C         THIS PROGRAM CALLS SUBROUTINE CHTC TO DETERMINE
C         THE CONDENSATION TWO-PHASE HEAT TRANSFER COEFFICIENT
C         FOR FORCED CONVECTION CONDENSATION INSIDE TUBES
C         THIS PROGRAM CALLS SUBROUTINE VAPOR TO DETERMINE
C         THERMODYNAMIC PROPERTIES OF SUPERHEATED REFRIGERANT
C         VAPOR
C         THIS PROGRAM CALLS SUBROUTINE EXCH TO DETERMINE
C         THE OVERALL HEAT EXCHANGER PERFORMANCE, HEAT TRANS.
C         RATES, AIR TEMPERATURES ETC.
C         THIS PROGRAM CALLS SUBROUTINE PDROP TO DETERMINE
C         PRESSURE DROP OF REFRIGERANT FLOWING IN THE COIL
C         THIS PROGRAM CALLS FUNCTION SUBPROGRAM TSAT TO
C         DETERMINE SATURATION TEMPERATURES CORRESPONDING
C         TO GIVEN PRESSURES
C
       COMMON CPA,HA,SEFFX,HRV,HRL,CPRL,CPRV,XMR,XMA,X3,
      1TRI,HTP,F,FTP,FSC,GSF,GTP,GSC,TAOSP,TAOTP,TAOSC,TAO,
      2TRC,ARHT
C
C      ---------------- INPUT DATA CONSTANTS -------------
C
C AIR PROPERTIES
C      PRA   -  PRANDTL NUMBER OF AIR
C      XMUA  -  VISCOSITY OF AIR (LBM/HR-FT)
C      RAL   -  UNIVERSAL GAS CONSTANT FOR AIR (FT-LBF/LBM-R)
C      PA    -  ATMOSPHERIC PRESSURE (PSIA)
```

```
C     CPA    -  SPECIFIC HEAT AT CONST.PRES.OF AIR (BTU/LBM-R)
      DATA PRA,XMUA,RAU,PA/.714,.043,53.34,14.7/
      CPA = .24
C
C REFRIGERANT PROPERTY VARIATION COEFFICIENTS
C     NR     -  NUMBER OF REFRIGERANT (12,22, OR 502)
C     NREF   -  NUMBER OF REFRIGERANT (USUALLY SAME AS NR)
C     SLPEMV & XINMV  -  COEFFICIENTS FOR VISCOSITY OF VAPOR
C     SLPEKV & XINKV  -  COEFFICIENTS FOR THERMAL
C                        CONDUCTIVITY OF VAPOR
C     SLPEKL & XINKL  -  COEFFICIENTS FOR THERMAL
C                        CONDUCTIVITY OF LIQUID
C     XM1 - XM4       -  COEFFICIENTS FOR VISCOSITY OF LIQ.
C     CP1 & CP2       -  COEFFICIENTS FOR SPECIFIC HEAT
C                        AT CONST. PRES. OF LIQUID
      NR = 22
      DATA NREF,SLPEMV,XINMV,SLPEKV,XINKV,SLPEKL,XINKL
     1/22,.0000759,.0272,.00002,.00482,-.000159,.06299/
      DATA XM1,XM2,XM3,XM4/-5.625E-08,1.525E-05,-2.982E-03,
     1.646/
      DATA CP1,CP2/2.98E-04,.2575/
C ****NOTE - THE ABOVE REFRIGERANT PROPERTY COEFFICIENTS
C     ARE FOR REFRIGERANT 22 ONLY
C
C AIR SIDE FLOW CHARACTERISTICS (SAME FOR BOTH EVAP.&COND.
C IF THEY ARE OF THE SAME TYPE)
C     C1A-C6A         -  COEFFICIENTS FOR EXPRESSING THE
C                        AIR SIDE HEAT TRANSFER COEFFICIENT
C     XLLA   -  LOWER REYNOLDS NUMBER LIMIT FOR LAMINAR
C               FLOW ON AIR SIDE
C     ULA    -  UPPER REYNOLDS NUMBER LIMIT FOR TURBULENT
C               FLOW ON AIR SIDE
      DATA C1A,C2A,C3A,C4A,C5A,C6A,XLLA,ULA
     1/.2243,-.385,.2243,-.385,.2243,-.385,1000.0,2000.0/
C
C REFRIGERANT SIDE FLOW CHARACTERISTICS (SAME FOR BOTH
C EVAP.& COND. IF THEY ARE OF SAME TYPE)
C     C1R-C6R         -  COEFFICIENTS FOR EXPRESSING THE
C                        REFRIGERANT SIDE SINGLE PHASE HEAT
C                        TRANSFER COEFFICIENTS
C     XLLR   -  LOWER REYNOLDS NUMBER LIMIT FOR LAMINAR
C               FLOW ON REFRIGERANT SIDE (SINGLE PHASE)
C     ULR    -  UPPER REYNOLDS NUMBER LIMIT FOR TURBULENT
C               FLOW ON REFRIGERANT SIDE (SINGLE PHASE)
      DATA C1R,C2R,C3R,C4R,C5R,C6R,XLLR,ULR
     1/1.164,-.7824,.000254,.49985,.00667,-.0897,2400.,3500./
C
C     -----------END OF  INPUT DATA CONSTANTS ------------
C
```

```
C      CUTER LOOP FOR MULTIPLE RUNS WHILE VARYING HEAT
C      EXCHANGER CHARACTERISTICS
C
       READ(8,590) NRUN
       DO 580  IN = 1,NRUN
C
C      ----------------HEAT EXCHANGER CHARACTERISTICS-------
C
C      DEA    -  OUTSIDE DIAMETER OF TUBES (FT)
C      DER    -  INSIDE DIAMETER OF TUBES (FT)
C      DELTA  -  FIN THICKNESS (FT)
C      FP     -  FIN PITCH (FINS/FT)
C      XKF    -  THERMAL CONDUCTIVITY OF FINS (BTU/HR-FT-F)
C      AAF    -  HEAT EXCHANGER FRONTAL AREA (SQ FT)
C      QA     -  AIR FLOW RATE (CU FT/MIN)
C      NT     -  NUMBER OF TUBES IN DIRECTION OF AIR FLOW
C      NSECT  -  NUMBER OF PARALLEL CIRCUITS IN HEAT EXCHANGER
C      HCONT  -  CONTACT RESISTANCE BETWEEN FINS AND TUBES
C                (BTU/HR-SQ FT-F)
C      ST     -  VERTICAL SPACING OF TUBE PASSES (FT)
C      WT     -  SPACING OF TUBE ROWS IN DIR.OF AIR FLOW (FT)
C      SIGA   -  SIGMA AIR (AIR FLOW AREA/FRONTAL AREA)
C      ATBO   -  CROSS-SECTIONAL AREA OCCUPIED BY TUBE (SQ FT)
C      PTBO   -  OUTTER PERIMETER OF TUBE (FT)
C      ALFAA  -  ALPHA AIR (AIR SIDE HEAT TRANSFER AREA/TOTAL
C                VOLUME OF HEAT EXCHANGER -1/FT)
C      ARFT   -  CROSS-SECTIONAL FLOW AREA INSIDE TUBES (SQ FT)
C      P      -  INSIDE PERIMETER OF TUBES (FT)
C      ALFAR  -  ALPHA REFRIGERANT (REFRIGERANT SIDE HEAT
C                TRANS.AREA/TOTAL VOLUME OF HEAT EXCHANGER-1/FT)
C      FAR    -  RATIO - FIN HEAT TRANS.AREA/TOTAL H.T. AREA
C      XLF    -  LENGTH OF FINS (FT)
C      ARHT   -  TOTAL REFG.SIDE HEAT TRANS. AREA/NSECT (SQ FT)
C      CAR    -  RATIO - FIN HEAT TRANS.AREA/CONTACT AREA
C                TO ACCOUNT FOR CONTACT RESISTANCE BETWEEN
C                FINS AND TUBES
       READ(8,600)  DEA,DER,DELTA,FP,XKF,AAF,QA,NT,NSECT
       READ(8,611) HCONT, ST,WT
       SIGA = (ST-DEA)*(1.0-DELTA*FP)/ST
       ATBO = 3.14*DEA**2/4.0
       PTBO = 3.14*DEA
       ALFAA=(2.0*(ST*WT-ATBO)*FP +(1.0-DELTA*FP)*PTBO)/(ST*WT)
       ARFT = 3.14*DER**2/4.0
       P = 3.14*DER
       ALFAR = 3.14*DER/(ST*WT)
       FAR=2.0*FP*(ST*WT-ATBO)/(2.0*FP*(ST*WT-ATBO)+PTBO*
      1(1.0-FP*DELTA))
       XLF = ST/2.0
       ARHT = FLOAT(NT)*3.14*DER*AAF/(ST*FLOAT(NSECT))
```

```
      CAR =2.0*(ST*WT-3.14*DEA**2/4.0)/(3.14*DEA*DELTA)
      WRITE(5,510)
      WRITE(5,500)DEA,DER,DELTA,FP,XKF,AAF,GA,ARHT,NT,NSECT
      WRITE(5,501) HCONT,ST,WT
C
C     ---------END OF HEAT EXCHANGER CHARACTERISTICS----------
C
C     INITIAL VALUES FOR AIR AND REFRIGERANT FLOW CONDITIONS
C
C     TAII  -  AIR TEMPERATURE ENTERING CONDENSER (F)
C     DTA   -  AIR TEMP.INCREMENT (F)
C     NTEMP -  NUMBER OF AIR TEMPS. EXAMINED
C     TSA   -  REFRIGERANT SATURATION TEMP. (F)
C     DXMRI -  REFRIGERANT FLOW RATE INCREMENT (LBM/HR)
C     XMRI  -  INITIAL TOTAL REFRIGERANT FLOW RATE (LBM/HR)
C     NXMR  -  NUMBER OF REFRIGERANT FLOW RATES EXAMINED
C     TRI   -  TEMP.OF REFRIGERANT VAPOR ENTERING COND. (F)
C
      READ(8,610) TAII,DTA,NTEMP,TSA ,DXMRI,XMRI,NXMR,TRI
      VA = GA*60.0/(SIGA*AAF)
C
C     LOOP FOR VARYING AIR TEMPERATURE ENTERING CONDENSER
C
      TAI = TAII
      DO 400 I=1,NTEMP
      WRITE(5,540) I
      TAI = TAI +DTA
      GA = VA*PA*144.0/(RAU*(TAI + 460.0))
C     ACM    -  UNIT REFRIGERANT SIDE HEAT TRANSFER AREA/ UNIT
C     AIR FLOW RATE (SQ FT-HR/LBM DRY AIR)
      ACM = FLOAT(NT)*P/(ST*GA*SIGA)
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*GA*PA*144.0/(RAU*(TAI + 460.0)*FLOAT(NSECT))
C
C     DETERMINE AIR SIDE HEAT TRANS.COEF.'HA'(BTU/HR-SQ FT-F)
C
      CALL SPHTC(DEA,GA,C1A,C2A,C3A,C4A,C5A,C6A,XLLA,ULA,
     1XMLA,CPA,PRA,REA,HA)
C
C     DETERMINE OVERALL SURFACE EFFICIENTCY 'SEFFX'
C
      CALL SEFF(XKF,DELTA,HA,XLF,FAR,CAR,HCONT,SEFFX)
      ICNT = 1
C
C     LOOP FOR VARYING REFRIGERANT FLOW RATE
```

```
C
      XMR = XMRI/FLOAT(NSECT)
      DXMR = DXMRI/FLOAT(NSECT)
      DO 200 K=1,NXMR
      WRITE(5,540) K
      XMR = XMR + DXMR
C
C     DETERMINE SATURATION PROPERTIES OF REFRIGERANT
C
   20 CALL SATPRP(NR,TSA ,PSAT,VF,VG,HSATL,HFG,HSATV,SF,SG)
      RHOV = 1.0/VG
      RHOL = 1.0/VF
      IF(ICNT.EQ.1) P2 = PSAT
      XMUL = XM1*TSA **3 + XM2*TSA **2 + XM3*TSA  + XM4
      CPRL = CP1*PSAT + CP2
      XKRL = SLPEKL*TSA  + XINKL
      XMURV = SLPEMV*TSA  + XINMV
      X3 = 0.0
      PRRL = XMUL*CPRL/XKRL
      GR = XMR/AREFT
C
C     DETERMINE CONDENSATION TWO-PHASE HEAT TRANS.COEF.'HTP'
C     (BTU/HR-SQ FT-F)
C
      CALL CHTC(DER,GR,X3,PRRL,XKRL,XMURV,XMUL,RHOL,RHOV,HTP)
C
C     DETERMINE SINGLE PHASE LIQUID HEAT TRANSFER COEF.
C     'HRL' (BTU/HR-SQ FT-F)
C
      CALL SPHTC(DER,GR,C1R,C2R,C3R,C4R,C5R,C6R,XLLR,ULR,
     1XMUL,CPRL,PRRL,RERL,HRL)
C
C     DETERMINE SINGLE PHASE VAPOR PROPERTIES
C
      CALL VAPOR(NR,TRI,P2,V2I,H2I,S2I)
      XMURV = SLPEMV*(TSA  + TRI)/2.0 + XINMV
      XKRV = SLPEKV*(TSA  + TRI)/2.0 + XINKV
      CPRV = (H2I-HSATV)/(TRI-TSA )
      PRRV = XMURV*CPRV/XKRV
C
C     DETERMINE SINGLE PHASE VAPOR HEAT TRANSFER COEF.
C     'HRV' (BTU/HR-SQ FT-F)
C
      CALL SPHTC(DER,GR,C1R,C2R,C3R,C4R,C5R,C6R,XLLR,ULR,
     1XMURV,CPRV,PRRV,RERV,HRV)
C
C     USE SUBROUTINE EXCH TO DETERMINE CONDENSER HEAT
C     TRANSFER PERFORMANCE AND RETURN ALL RESULTS THROUGH
C     COMMON
```

```
C
      CALL EXCH(4,ACM,ALFAR,ALFAA,TSA,TAI,HFG)
      DZTP=FTP*ARHT/P
      DZV = F*ARHT/P
      DZL = FSC*ARHT/P
      XIC = 1.0
      E = 5.0E-06
C
C     USE SUBROUTINE PDROP TO DETERMINE PRESSURE DROP OF
C     REFRIGERANT THROUGH CONDENSER 'PD' (PSI)
C
      CALL PDROP(4,DER,E,GR,XMURV,XMUL,RHOV,RHOL,RERV,
     1RERL,DZTP,X3,XIC,V2I,DZV,DZL,PD)
      GC = GSP + GTP + GSC
      XMA = XMA*FLOAT(NSECT)
      XMR = XMR*FLOAT(NSECT)
C
C     CONVERT BACK TO TOTAL FLOW AND OVERALL PERFORMANCE
C     AND PRINT RESULTS
C
      GC = GC*FLOAT(NSECT)
      WRITE(5,525)
      WRITE(5,520) TAI,TSA ,XMR,TRI
      WRITE(5,535)
      WRITE(5,530) XMA,H2I,V2I,S2I,PD,GC
      WRITE(5,850) GA,REA,HA,SEFFX
      WRITE(5,860) GR,RERV,HRV
      WRITE(5,865) RERL,HRL,HTP
      WRITE(5,870) F,FTP,FSC
      WRITE(5,880) GSP,GTP,GSC
      WRITE(5,890) TAOSP,TAOTP,TAOSC
      CALL SATPRP(NR,TRO,P,VF,VG,HRO,HFG,HG,SF,SG)
      WRITE(5,900) TAO,TRO,ARHT,HRO
      XMA = XMA/FLOAT(NSECT)
      XMR = XMR/FLOAT(NSECT)
      IF(ICNT.NE.1) GO TO 195
C
C     CHECK DROP IN SATURATION TEMPERATURE DUE TO PRESSURE
C     DROP IN COIL - IF THE DROP IN SATURATION TEMPERATURE
C     IS GREATER THAN 2 DEGREES F - REPEAT ALL CALCULATIONS,
C     USING AN AVERAGE VALUE OF SATURATION TEMPERATURE
C
      POUT = PSAT + PD
      TSATO = TSAT(NR,POUT)
      WRITE(5,830) POUT,TSATO
      IF((TSA -TSATO).LE.2.0) GO TO 200
      TSA  = (TSA  + TSATO)/2.0
      ICNT = 2.0
      GO TO 20
```

```
195   TSA   = 2.0*TSA   - TSATO
      ICNT = 1
200   CONTINUE
400   CONTINUE
500   CONTINUE
502   FORMAT('0',8F12.6,2I4)
501   FORMAT('   HCONT=',F10.3,'   ST=',F10.5,'   WT=',F10.5)
510   FORMAT('0',' DEA (FT)     DER (FT)    DELTA (FT)   FP'
     1,,' (FINS/FT)XKF(BTU/HRFT) AAF (SQFT) QA(CUFT/MIN)'
     2,,' ARHT (SQFT)     NT    NSECT')
520   FORMAT('0',4F10.4)
525   FORMAT('1','  TAI (F)  TSAT (F) XMR (LBM/HR) TRI (F)')
530   FORMAT('0',6F15.4)
535   FORMAT('0',' XMA (LBM/HR)  H2I (BTU/LBM)  V2I (CU-FT/'
     1,,'LBM)S2I (BTU/LBM-R) PC (PSIA)       QC (BTU/HR)')
540   FORMAT(I4)
590   FORMAT(I10)
600   FORMAT(7F12.6,2I10)
610   FORMAT(2F10.4,I10,3F10.4,I10,F10.4)
611   FORMAT(3F15.5)
830   FORMAT('     FCUT=',F10.5,'    TSATO=',F8.2)
850   FORMAT('0',5X,'GA=',F10.2,' (LBM/HR-SQ-FT)',5X,'REA='
     1,F10.2,5X,'HA=',F10.2,' (BTU/HR-SQ FT-R)',5X,'SEFFX='
     2,F6.3)
860   FORMAT('0',5X,'GR=',F10.3,' (LBM/HR-SQ-FT)',5X,'RERV='
     1,F10.2,5X,'HRV=',F7.1,' (BTU/HR-SQ FT-R)')
865   FORMAT('0',10X,'RERL=',F10.2,5X,'HRL=',F7.1,' (BTU/'
     1,,'HR-SQ FT-R)',5X,'HTP=',F7.1,' (BTU/HR-SQ FT-R)')
870   FORMAT('0',10X,'F=',F6.4,5X,'FTP=',F6.4,5X,'FSC=',F6.4)
880   FORMAT('0',10X,'QSP=',F10.2,' (BTU/HR)',5X,'QTP='
     1,F10.2,' (BTU/HR)',5X,'QSC=',F10.2,' (BTU/HR)')
890   FORMAT('0',10X,'TACSP=',F7.2,' (F)',5X,'TACTP=',F7.2,
     1' (F)',5X,'TACSC=',F7.2,' (F)')
900   FORMAT('0',10X,'TAC=',F7.2,' (F)',5X,'TRO=',F7.2,
     1' (F)',5X,'ARHT=',F10.4,' (SQ FT)   HRO=',F10.4,
     2' (BTU/LBM)')
      END
```

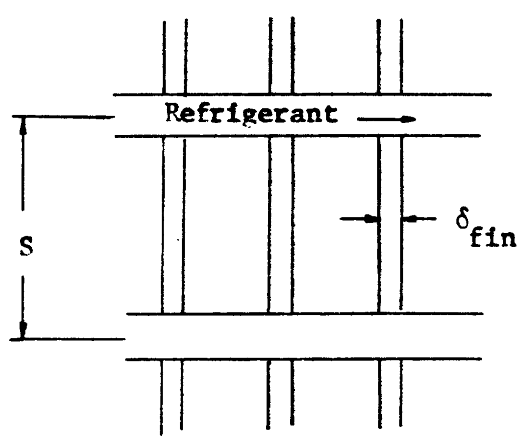
APPENDIX J

COMPLEMENTS TO HEAT EXCHANGER ANALYSIS

Presented here are derivations of an expression describing overall surface efficiency of finned circular tubes, including contact resistance between fins and tubes, and of a closed-form expression for cross-flow effectiveness, both fluids unmixed, when using the effectiveness-NTU method of heat exchanger design. Program listings are included.

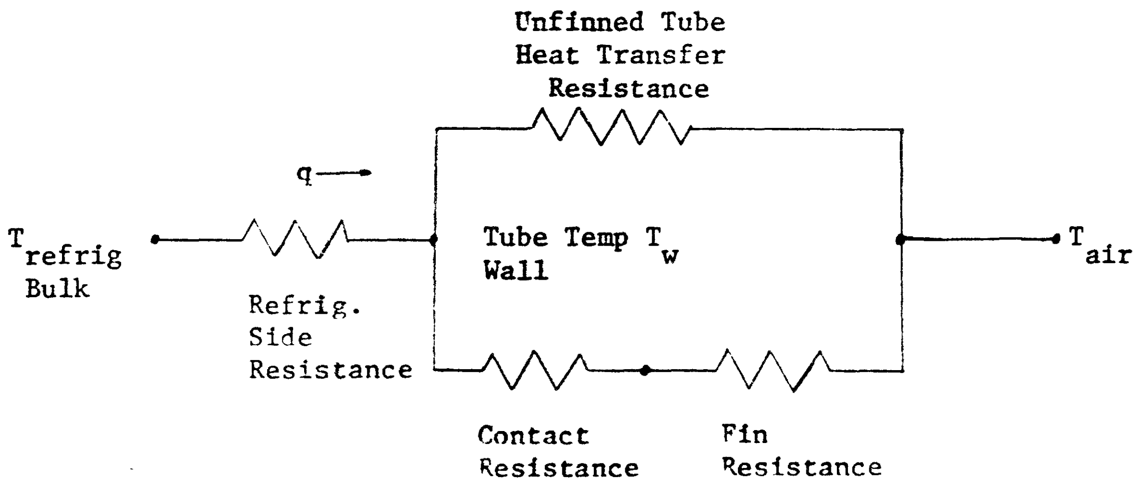
Overall Surface Efficiency

Figure J-1 gives a drawing of a typical finned tube heat exchanger, while Figure J-2 shows an electrical analogy to the heat flow of such a configuration.



FINNED SURFACE

FIGURE J-1



ELECTRICAL ANALOGY OF FINNED SURFACE

FIGURE J-2

From the electrical analog, we see that

$$R_{total} = R_{refrig} + \frac{1}{\dfrac{1}{R_{unfinned}} + \dfrac{1}{(R_{contact} + R_{finned})}}$$

We know however, that

$$q = hdA\ \Delta T \qquad\qquad V = IR$$

$$q \rightleftharpoons I$$

$$\Delta T \rightleftharpoons V$$

$$\therefore \quad R \rightleftharpoons \frac{1}{hdA}$$

negligible

$$R_{refrig} = \frac{1}{h_{refrig}\ dA_{inside\ tubes}} + \cancel{\text{Resistance of tube wall}}$$

$$R_{unfinned} = \frac{1}{h_{air}\ dA_{unfinned\ tubes}}$$

$$R_{contact} = \frac{1}{h_{contact}\ dA_{contact}}$$

$$R_{fin} = \frac{1}{\eta_{fin}\ h_{air}\ dA_{fin\ heat\ transfer}}$$

As discussed in Kreith, Principles of Heat Transfer[1], and Rohsenow & Choi, Heat, Mass, and Momentum Transfer[2], fin efficiency η_{fin} for plate fins can be expressed as:

$$\eta_{fin} = \frac{\mathrm{Tanh}\ (m\ \ell)}{m\ \ell}$$

$$m \equiv \left[\frac{2\ h_{air}}{k_{fin}\ \delta_{fin}}\right]^{1/2}$$

where k_{fin} = Thermal Conductivity of Fin Material

δ_{fin} = Fin Thickness

h_{air} = Air Side Heat Transfer Coefficient

ℓ = Effective Fin Length

= Tube Spacing/2

ℓ = S/2

As discussed in Rohsenow & Choi[2], pg. 307, overall surface efficiency η_o is defined as

$$\eta_o \equiv \frac{dq}{dA_{tot}\ h_{air}(T_{air} - T_{wall})}$$

where A_{tot} = Total Air Side Heat Transfer Area = $A_{unfinned} + A_{finned}$

Hence, for the present case, we have

$$\eta_o = \frac{(hA)_{equivalent}}{A_{tot}\ h_{air}}$$

$$\eta_o = \frac{\left[\dfrac{1}{R_{unfinned}} + \dfrac{1}{(R_{contact} + R_{finned})}\right]}{dA_{tot}\ h_{air}}$$

$$\eta_o = \frac{h_{air}\ dA_{unfinned} + \dfrac{1}{\dfrac{1}{h_{cont}\ dA_{cont}} + \dfrac{1}{\eta_{fin}\ h_{air}\ dA_{fin}}}}{dA_{tot}\ h_{air}}$$

$$\eta_o = \frac{dA_{unfinned}}{dA_{tot}} + \frac{1}{\dfrac{h_{air}}{h_{cont}}\ \dfrac{dA_{tot}}{dA_{cont}} + \dfrac{1}{\eta_{fin}}\ \dfrac{dA_{tot}}{dA_{fin}}}$$

or similarly

$$\eta_o = 1 - \frac{dA_{finned}}{dA_{tot}} + \frac{dA_{finned}}{dA_{tot}} \left[\frac{1}{\left(\dfrac{h_{air}}{h_{cont}}\right) \dfrac{dA_{fin}}{dA_{cont}} + \dfrac{1}{\eta_{fin}}}\right]$$

$$\eta_o = 1 - \frac{dA_{finned}}{dA_{tot}} \left\{1 - \frac{1}{\left[\left(\dfrac{dA_{fin}}{dA_{cont}}\right)\left(\dfrac{h_{air}}{h_{cont}}\right) + \dfrac{1}{\eta_{fin}}\right]}\right\}$$

Using this expression for η_o, we find

$$R_{tot} = R_{refrig} + \frac{1}{\eta_o\ dA_{tot}\ h_{air}}$$

$$R_{tot} = \frac{1}{h_{ref}\ dA_{\substack{inside\\tubes}}} + \frac{1}{\eta_o\ dA_{tot}\ h_{air}}$$

and

$$dq = \frac{(T_{air} - T_{refrig\ bulk})}{R_{tot}}$$

Subroutine SEFF, when given the necessary inputs, produces a value for overall surface efficiency, using the expression for fin efficinecy and the latter expression for overall surface efficiency. For further details, see comments in the program listing at the end of this section.

Cross-Flow Effectiveness

The analytical expressions for cross-flow effectiveness with both fluids unmixed are not in closed form, and are hence, normally presented in graphical form as in Kays & London[3]. A correction factor to counterflow effectiveness has been empirically determined which, as shown in Figure J-3, approximates the actual cross-flow effectiveness within about 3% over the entire range. The resulting expression is:

$$\varepsilon_{XF} = \frac{\varepsilon_{\text{counter flow}}}{\left[1 + \left(\frac{C_{min}}{C_{max}}\right)(.047)\right] NTU^{.036\left(\frac{C_{min}}{C_{max}}\right)}}$$

where:

$$\varepsilon_{cf} = \frac{NTU}{1 + NTU} \text{ (counter flow)} \qquad \text{for } \frac{C_{min}}{C_{max}} = 1$$

$$\varepsilon_{cf} = \frac{1 - e^{-NTU\left(1-\frac{C_{min}}{C_{max}}\right)}}{\left[1-\left(\frac{C_{min}}{C_{max}}\right) e^{-NTU\left(1-\frac{C_{min}}{C_{max}}\right)}\right]} \qquad \text{For } \frac{C_{min}}{C_{max}} < 1$$

cf = counter flow

$C = \dot{m}\, C_p$ = Heat Capacity Rate

C_{min} = Smaller of the Two Heat Capacity Rates

C_{max} = Larger of the Two Heat Capacity Rates

$NTU = \frac{AU}{C_{min}}$ (Dimensionless)

$AU = \frac{1}{R_{tot}}$ = Overall Conductance

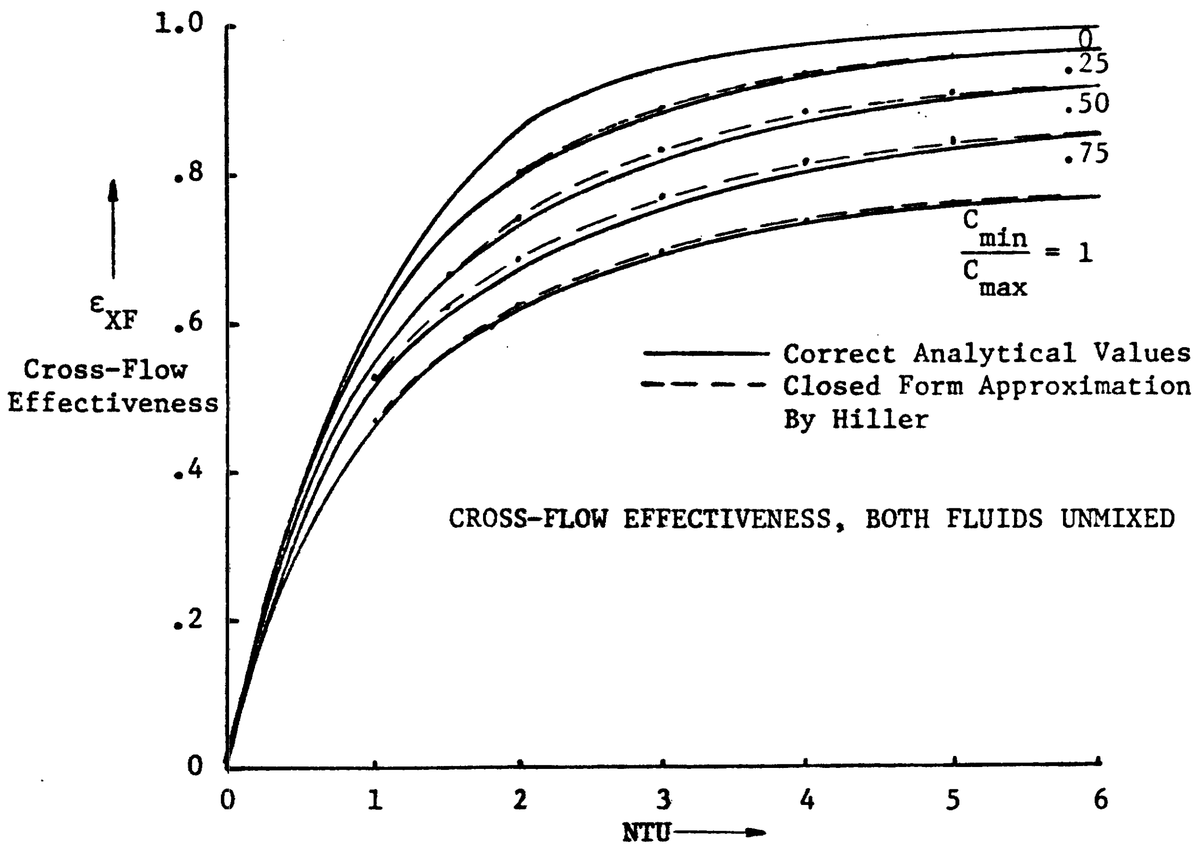
Subroutine EXF uses the above expressions to evaluate cross-flow effectiveness for given operating conditions. See comments in the program listing at the end of this section for more details.

REFERENCES

1. Kreith, Frank, Principles of Heat Transfer (2nd ed.; Scranton, Penn.: International Textbook Co., 1969) pg. 62.

2. Rohsenow, Warren M. and Choi, Harry Y., Heat, Mass, and Momentum Transfer (Englewood Cliffs, New Jersey: Prentice-Hall, Inc., 1961), pg. 109, 307.

3. Kays, W. M. and London, A. L., Compact Heat Exchangers (Palo Alto, California: The National Press, 1955) pg. 27, 33.



1.0
.8
.6
.4
.2
0
ε_{XF}
Cross-Flow
Effectiveness
0
1
2
3
4
5
6
NTU
0
.25
.50
.75
$\frac{C_{min}}{C_{max}} = 1$
Correct Analytical Values
Closed Form Approximation
By Hiller
CROSS-FLOW EFFECTIVENESS, BOTH FLUIDS UNMIXED

FIGURE J-3

```
      SUBROUTINE SEFF(XK,DELTA,HA,XL,FAR,CAR,HCONT,SEFFR)
C
C     PURPOSE
C        TO DETERMINE THE SURFACE EFFICIENCY
C        FOR A FINNED SURFACE, ACCOUNTING FOR CONTACT
C        RESISTANCE BETWEEN FINS AND BASE
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        XK    =  THERMAL CONDUCTIVITY OF FIN (BTU/HR-FT-F)
C        DELTA=  FIN THICKNESS (FT)
C        HA    =  EXTERNAL HEAT TRANS.COEF.(BTU/HR-SQ FT-F)
C        XL    =  LENGTH OF FIN (FT)
C        FAR   =  RATIO - FIN HEAT TRANS.AREA/TOTAL H.T. AREA
C        CAR   =  RATIO - FIN HEAT TRANS.AREA/CONTACT AREA
C        TO ACCOUNT FOR CONTACT RESISTANCE BETWEEN FINS
C        AND BASE
C        HCONT=  CONTACT RESISTANCE (BTU/HR-SQ FT-F)
C      OUTPUT
C        SEFFR=  OVERALL SURFACE EFFICIENCY
C
C     FIN EFFICIENCY
      XM = SQRT(2.0*HA/(XK*DELTA))
      FINEF = (EXP(XM*XL)-EXP(-XM*XL))/((EXP(XM*XL)+EXP(-XM
     !*XL))*XM*XL)
C
C     SURFACE EFFICIENCY
      SEFFR=1.0-FAR*(1.0-1.0/(CAR*HA/HCONT+1.0/FINEF))
      RETURN
      END
```

```
      SUBROUTINE EXF(RTOT,CA,CR,CMIN,EXFR)
C
C     PURPOSE
C        TO DETERMINE THE EFFECTIVENESS OF A CROSS FLOW HEAT
C        EXCHANGER USING THE EFFECTIVENESS-NTU METHOD
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        RTOT =  TOTAL RESISTANCE TO HEAT FLOW
C        BETWEEN FLUIDS ((HR-F)/BTU)
C        CA   =  TOTAL HEAT CAPACITY OF FLUID A (BTU/R)
C        CR   =  TOTAL HEAT CAPACITY OF FLUID R (BTU/R)
C      OUTPUT
C        CMIN =  THE SMALLER OF CA AND CR (BTU/R)
C        EXFR =  EFFECTIVENESS OF CROSS FLOW HEAT EXCHANGER
C
C     DETERMINE CMIN AND NTU
      IF(CR/CA.LT..999) GO TO 5
      IF(CA/CR.LT..999) GO TO 10
      CMIN = CA
      CMAX = CR
      XNTU = 1.0/(CMIN*RTOT)
      ECF = XNTU/(1.0 + XNTU)
      GO TO 20
    5 CMIN = CR
      CMAX = CA
      GO TO 15
   10 CMIN = CA
      CMAX = CR
   15 XNTU = 1.0/(CMIN*RTOT)
C
C     EVALUATE COUNTER FLOW EFFECTIVENESS
      ECF = (1.0-EXP(-XNTU*(1.0-CMIN/CMAX)))/(1.0-CMIN/CMAX
     1*EXP(-XNTU*(1.0 - CMIN/CMAX)))
C
C     APPLY CORRECTION FACTOR TO COUNTER FLOW EFFECTIVENESS
C     TO OBTAIN CROSS FLOW EFFECTIVENESS
   20 EXFR= ECF/((1.0 + .047*CMIN/CMAX)*XNTU**
     1(.036*CMIN/CMAX))
      RETURN
      END
```

APPENDIX K

HEAT TRANSFER COEFFICIENTS

This section outlines the basis for all heat transfer coefficients used in the present study. Briefly, these are: Condensation (two-phase) heat transfer coefficient, evaporation (two-phase) heat transfer coefficient, and single phase refrigerant and air side heat transfer coefficients. The three subroutines used for computing these coefficients are described, and listings are included at the end of the discussion.

Condensation Heat Transfer Coefficient

Condensation of R-12 and R-22 in forced convection was studied by Traviss, Baron, and Rohsenow[1] using the Lockhart-Martinelli two-phase flow pressure drop correlation. (See Appendix L). The following relations are found from a quite general derivation, and are hence believed applicable to condensation of other fluids.

The local heat transfer coefficient h_z is correlated within $\pm$ 15% by

$$.1 \leq F\ (X_{tt}) < 1 \qquad \frac{Nu_z F_2}{Pr_\ell Re_\ell^{.9}} = F\ (X_{tt})$$

$$1 < F\ (X_{tt}) < 15 \qquad \frac{Nu_z F_2}{Pr_\ell Re_\ell^{.9}} = [F(X_{tt})]^{1.15}$$

where

$$F(X_{tt}) \equiv .15\ [X_{tt}^{-1} + 2.85\ X_{tt}^{-.476}]$$

$$X_{tt} = \left(\frac{\mu_\ell}{\mu_v}\right)^{.1} \left(\frac{1-x}{x}\right)^{.9} \left(\frac{\rho_v}{\rho_\ell}\right)^{.5}$$

$$Re_\ell < 50 \qquad F_2 = .707\ Pr_\ell\ Re_\ell^{.5}$$

$$50 < Re_\ell < 1125 \qquad F_2 = 5\ Pr_\ell + 5\ \ln\ [1 + Pr_\ell(.09636\ Re_\ell^{.585} - 1)]$$

$$Re_\ell > 1125 \qquad F_2 = 5\ Pr_\ell + 5\ \ln(1 + 5\ Pr_\ell) + 2.5\ \ln(.00313\ Re_\ell^{.812})$$

$$Re_\ell \equiv \frac{G\ (1-X)D}{\mu_\ell}$$

X = Local Quality

The length of tube Δz required to change the quality Δx from an energy balance is

$$\Delta z = \frac{G\ h_{fg}\ D\Delta x}{4\ h_z\ \Delta T}$$

Using this expression, the length of tube z to change the quality from x_i at inlet to x_e at exit may be found by dividing the calculation into steps of $\Delta x = .05$ or $.10$. However, for cases of approximately constant ΔT, as in many air-cooled condensers, the above expressions may be combined and integrated to yield an expression for the overall average heat transfer coefficient[1]:

$$\frac{1}{h_{avg}} = \frac{1}{(x_i - x_e)} \int_{x_e}^{x_i} \frac{dx}{h_z}$$

As can be seen, this expression is a function only of quality. The average heat transfer coefficient may thus be obtained by dividing the calculation into septs of $\Delta x = .05$ or $.10$, and performing a simple step-wise-constant integration. Subroutine CHTC has been programmed to do such an integration and return a value of the average condensation heat transfer coefficient, independent of length. See comments in the program listing at the end of this section for more details.

Evaporation Heat Transfer Coefficient

A good discussion of two-phase boiling and evaporation is given in Tong, Boiling Heat Transfer and Two-Phase Flow[2]. An expression for the average evaporation two-phase heat transfer coefficient from entering quality x_i to exit quality x_e, assuming constant ΔT between tube wall and fluid is as follows.

$$h_{avg} = (.023)(.325)(2.50)\frac{k_\ell}{D_\ell^{.2}}\left(\frac{G}{\mu_\ell}\right)^{.8}\left(\frac{\mu_\ell C_{p\ell}}{k_\ell}\right)^{.4}\left(\frac{\rho_\ell}{\rho_v}\right)\left(\frac{\mu_v}{\mu_\ell}\right)^{.075}\frac{(x_e - x_i)}{(x_e^{.325} - x_i^{.325})}$$

The length of the evaporating region can then be found from an energy balance yielding:

$$L_{tot} = \frac{\dot{m}\, h_{fg}}{\Delta T_{avg}\, P\, h_{avg}} (x_e - x_i)$$

where P is perimeter of flow passage. Subroutine EHTC uses the above expression to compute the average two-phase evaporation heat transfer coefficient. See comments in the program listing at the end of this section for more details.

Single Phase Heat Transfer Coefficients

Kays & London, Compact Heat Exchangers[3] presents heat transfer information on a number of different heat exchanger configurations. The correlations used in the present work for single phase refrigerant-side coefficients are developed from data on flow inside circular tubes, shown in [Figure K-1]. (Kays & London type ST-1). Kays & London suggest that for air-side coefficients, better correlation of data is achieved using outside tube diameter, rather than the equivalent diameter as is used in their book. For this reason, the Kays & London data has been replotted, correlated on the basis of tube outside diameter, for heat exchangers with round tubes in cross-flow. The results are shown in [Figure K-2].

The heat transfer coefficients obtained from the above correlations can be modeled in general form by the following expression:

$$S_T \, Pr^{2/3} = A \, Re^B$$

Where:

$$S_T \equiv \frac{h}{G \, C_p} \qquad Re \equiv \frac{G \, D_o}{\mu} \qquad D_o = \text{Tube Outside Diameter}$$

We find from [Figures K-1 and K-2]

Refrigerant-side

Laminar flow

Re < 3500 $\qquad S_T \, Pr^{2/3} = 1.10647 \, Re^{-.78992}$

Transition flow

$3500 \leq Re \leq 6000$ $\qquad S_T \, Pr^{2/3} = 3.5194 \times 10^{-7} \, Re^{1.03804}$

Turbulent flow

Re > 6000 $\qquad S_T \, Pr^{2/3} = .01080 \, Re^{-.13750}$

Air-side

All flow regimes

$$S_T \, Pr^{2/3} = .2243 \, Re^{-.385}$$

Subroutine SPHTC is structured to accept values of the constants in the various flow regimes and to produce a value for a single phase heat transfer coefficient based on the computed Reynolds number. The inputs for this program are structured as follows:

Laminar flow

Re < XLL $\qquad S_T \, Pr^{2/3} = C_1 \, Re^{C_2}$

Transition flow

$XLL \leq Re \leq UL$ $\qquad S_T \, Pr^{2/3} = C_3 \, Re^{C_4}$

Turbulent flow

Re > UL $$S_T \, Pr^{2/3} = C_5 \, Re^{C_6}$$

For more information, see comments in the program listing.

REFRENCES

1. Traviss, D. P., Baron, A. G., and Rohsenow, W.M., "Forced Convection Condensation Inside Tubes", Report No. 72591-74; Heat Transfer Laboratory, Massachusetts Institute of Technology, Cambridge, Mass. (ASHRAE Contract No. RP63).

2. Tong, L.S., Boiling Heat Transfer And Two-Phase Flow (New York: John Wiley & Sons, Inc., 1965) CPT 5.

3. Kays, W. M. and London, A.L., Compact Heat Exchangers, (Palo Alto, California: The National Press, 1955).

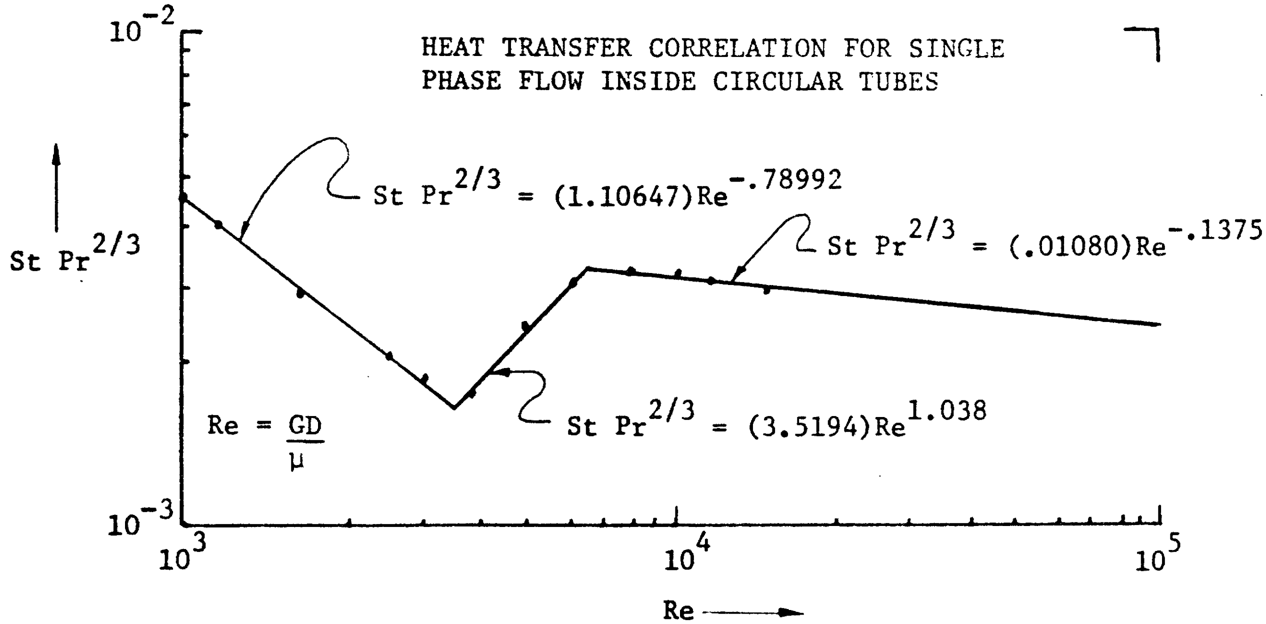


FIGURE K-1

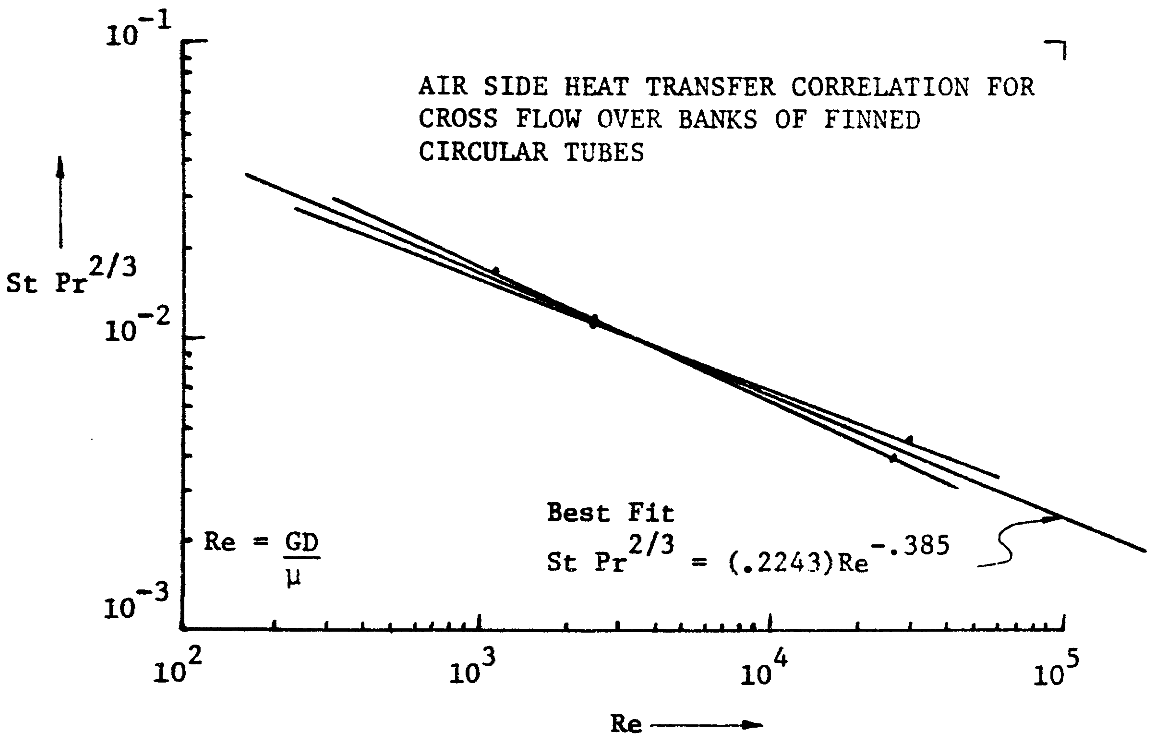


FIGURE K-2

```
      SUBROUTINE  CHTC(DE,G,XE,PRL,XKL,XMUV,XMUL,RHOL,
     1RHOV,HAVG)
C
C      PURPOSE
C         TO DETERMINE THE FORCED CONVECTION CONDENSATION
C         TWO-PHASE HEAT TRANS. COEF. FOR FLOW IN TUBES
C         (BASED ON CORRELATIONS BY TRAVIS)
C
C      DESCRIPTIONS OF PARAMETERS
C        INPUT
C         DE    =  EQUIVALENT DIAMETER OF FLOW PASSAGE (FT)
C         G     =  MASS FLOW PER UNIT AREA (LBM/HR-SQ FT)
C         XE    =  EXIT QUALITY
C         PRL   =  PRANDTL NUMBER OF THE LIQUID PHASE
C         XKL   =  THERMAL COND. OF LIQ. PHASE(BTU/HR-FT-F)
C         XMUV  =  VISCOSITY OF VAPOR PHASE (LBM/HR-FT)
C         XMUL  =  VISCOSITY OF LIQ. PHASE (LBM/HR-FT)
C         RHOL  =  DENSITY OF LIQ. PHASE (LBM/CU FT)
C         RHOV  =  DENSITY OF VAPOR PHASE (LBM/CU FT)
C        OUTPUT
C         HAVG  =  AVERAGE CONDENSATION TWO-PHASE
C         HEAT TRANSFER COEF. (BTU/HR-SQ FT-F)
C
C      INITIAL CONDITIONS
       HP = 5000.0
       HIINT = 0.0
C
C      INTEGRATE FROM QUALITY EQUALS 1 TO QUALITY EQUALS XE
       X = 1.0
       DX = .05
       DO 10 I=1,20
       X = X-DX
       IF(X.LE.XE) GO TO 15
       XTT = (XMUL/XMUV)**.1*(RHOV/RHOL)**.5*((1.0-X)/X)**.9
       FXTT = .15*(1.0/XTT + 2.85*XTT**(-.476))
       REL = G*DE*(1.0-X)/XMUL
       IF(REL.LT.50.0) F2 = .707*PRL*REL**.5
       IF((REL.GE.50.0).AND.(REL.LE.1125.0)) F2=5.0*PRL + 5.
      10*ALOG(1.0+PRL*(.09636*REL**.585-1.0))
       IF(REL.GT.1125.0) F2 = 5.0*PRL + 5.0*ALOG(1.0 + 5.0*
      1PRL)+2.5*ALOG(.00313*REL**.812)
       IF(FXTT.LE..1) GO TO 9
C
C      EVALUATION OF LOCAL HEAT TRANS. COEF.
       IF(FXTT.LT.1.0) HLOC=XKL*PRL*REL**.9*FXTT/(DE*F2)
       IF((FXTT.GE.1.0).AND.(FXTT.LT.15.0))HLOC=XKL*PRL*REL
      1**.9*FXTT**1.15/(DE*F2)
       IF(FXTT.GE.15.0) GO TO 9
       HINVM = (1.0/HLOC + 1.0/HP)/2.0
       HP = HLOC
```

```
      HIINT =-DX*HINVM + HIINT
      GO TO 10
    5 IF(I.GT.19) GO TO 10
      WRITE(5,500) FXTT
   10 CONTINUE
C
C     INTEGRATED AVERAGE HEAT TRANS. COEF.
   15 HAVG = (XE-1.0)/HIINT
      RETURN
  500 FORMAT('0',10X,'FXTT LIMIT EXCEEDED FXTT=',F10.2)
      END
```

```
      SUBROUTINE EHTC(DE,G,XI,XE,PRL,XKL,XMUV,XMUL,RHOL,
     1RHOV,HTC)
C
C      PURPOSE
C         TO DETERMINE THE EVAPORATION TWO-PHASE HEAT TRANS.
C         COEFFICIENT FOR FORCED CONVECTION FLOW
C         INSIDE TUBES
C
C      DESCRIPTION OF PARAMETERS
C       INPUT
C         DE    -  EQUIVALENT DIAMETER OF FLOW PASSAGE (FT)
C         G     -  MASS FLOW PER UNIT AREA (LBM/HR-SQ FT)
C         XI    -  INITIAL QUALITY
C         XE    -  EXIT QUALITY
C         PRL   -  PRANDTL NUMBER OF THE LIQUID PHASE
C         XKL   -  THERMAL COND. OF LIQ. PHASE (BTU/HR-FT-F)
C         XMUV  -  VISCOSITY OF VAPOR PHASE (LBM/HR-FT)
C         XMUL  -  VISCOSITY OF LIQUID PHASE (LBM/HR-FT)
C         RHOL  -  DENSITY OF LIQUID PHASE (LBM/CU FT)
C         RHOV  -  DENSITY OF VAPOR PHASE (LBM/CU FT)
C       OUTPUT
C         HTC   -  EVAP.HEAT TRANS. COEF.(BTU/HR-SQ FT-F)
C
      HTC = .023*.325*2.50*XKL*(G/XMUL)**.8*DE**(-.2)*PRL**
     1.4*(RHOL/RHOV)**.375*(XMUV/XMUL)**.075*(XE-XI)/
     2(XE**.325-XI**.325)
      RETURN
      END
```

```
      SUBROUTINE SPHTC(DE,G,C1,C2,C3,C4,C5,C6,XLL,UL,XMU,
     1CP,PR,RE,HTC)
C
C      PURPOSE
C         TO DETERMINE SINGLE PHASE HEAT TRANSFER
C         COEFFICIENTS AND REYNOLDS NUMBERS
C         IN LAMINAR,TRANSITION, OR TURBULENT FLOW
C
C      DESCRIPTION OF PARAMETERS
C       INPUT
C         DE    -   EQUIVALENT DIAMETER OF FLOW PASSAGE (FT)
C         G     -   MASS FLOW PER UNIT AREA (LBM/HR-SQ FT)
C         C1-C6-   CONSTANTS DESCRIBING DESIRED HEAT
C                  TRANSFER COEFFICIENT RELATION
C         XLL   -   LOWER REYNOLDS NUMBER LIMIT FOR LAMINAR FLOW
C         UL    -   UPPER REYNOLDS NUMBER LIMIT FOR TURB. FLOW
C         XMU   -   VISCOSITY OF FLUID (LBM/HR-FT)
C         CP    -   SPECIFIC HEAT AT CONSTANT PRES.(BTU/LBM-R)
C         PR    -   PRANDTL NUMBER
C       OUTPUT
C         RE    -   REYNOLDS NUMBER
C         HTC   -   SINGLE PHASE HEAT TRANS.COEF.(BTU/HR-SQ FT-F)
C
      RE = DE*G/XMU
C
C      LAMINAR FLOW REGIME
      IF(RE.LT.XLL) HTC = C1*G*CP*PR**(-.667)*RE**C2
C
C      TRANSITION FLOW REGIME
      IF((RE.GE.XLL).AND.(RE.LT.UL)) HTC=C3*G*CP*PR**(-.667
     1)*RE**C4
C
C      TURBULENT FLOW REGIME
      IF(RE.GE.UL) HTC = C5*G*CP*PR**(-.667)*RE**C6
      RETURN
      END
```

APPENDIX L

PRESSURE DROP RELATIONS

Discussed in this section are refrigerant two-phase and single phase pressure drops in the heat exchangers, single phase pressure drops in connecting piping, and heat exchanger air-side pressure drop. Derivations and program listings are included.

Two-Phase Pressure Drops

As described in Traviss[1] and Tong[2], the method of Lockhart and Martinelli[3] can be used to express the total two-phase pressure drop in either evaporation or condensation in the following manner:

$$\frac{dP}{dz} = \left(\frac{dP}{dz}\right)_f + \left(\frac{dP}{dz}\right)_g + \left(\frac{dP}{dz}\right)_m$$

Where $\left(\frac{dP}{dz}\right)_f$ is the component due to friction

$\left(\frac{dP}{dz}\right)_g$ is the component due to gravity and static head

$\left(\frac{dP}{dz}\right)_m$ is the component due to momentum change

$$\left(\frac{dP}{dz}\right)_f = \frac{-\left(\frac{G_v^{\,2}}{\rho_v}\right)}{g_o\, D} (.09) \left(\frac{\mu_v}{G_v D}\right)^{.2} [1 + 2.85\ (X_{tt}{}^{.523})]^2$$

$G_v \equiv (G)(x)$ subscript v for vapor

subscript ℓ for liquid

x = Quality D = Tube Equivalent Diameter

$$X_{tt} = \left(\frac{\mu_\ell}{\mu_v}\right)^{.1} \left(\frac{\rho_v}{\rho_\ell}\right)^{.5} \left(\frac{1-x}{x}\right)^{.9}$$

$$\left(\frac{dP}{dz}\right)_g = \frac{\left(\frac{G^2}{\rho_v}\right)}{g_o D} \frac{1}{F_r^{\,2}} \left[\left(\frac{\rho_\ell}{\rho_v}\right) - B\,\alpha\right]$$

$$F_r^{\,2} \equiv \frac{\frac{G^2}{\rho_v}}{a\,D} \qquad B \equiv \frac{\rho_\ell - \rho_v}{\rho_v}$$

a = Axial acceleration due to external force i.e. Gravity

$$\alpha \equiv \frac{1}{1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_v}{\rho_\ell}\right)^{2/3}}$$

$$\left(\frac{dP}{dz}\right)_m = -\frac{\left(\frac{G^2}{\rho_v}\right)}{g_o} \left(\frac{dx}{dz}\right) \left[2x + (1-2x)\left(\frac{\rho_v}{\rho_\ell}\right)^{1/3} + (1-2x)\left(\frac{\rho_v}{\rho_\ell}\right)^{2/3} - 2(1-x)\left(\frac{\rho_v}{\rho_\ell}\right)\right]$$

Now, using a given or assumed quality vs length profile (usually assumed linear), the calculations can be separated into steps of $\Delta x = .05$ or $.01$ and the results summed to give the total pressure drop. Alternatively we may integrate the expressions as follows:

Rearrange to obtain

$$\left(\frac{dP}{dz}\right)_f =$$

$$\frac{-(.09)}{g_o}\left(\frac{\mu_v^{.2}}{\rho_v}\right)\left(\frac{G^{1.8}}{D^{1.2}}\right) x^{1.8}\left\{1 + 2.85\left[\left(\frac{\mu_\ell}{\mu_v}\right)^{.1}\left(\frac{\rho_v}{\rho_\ell}\right)^{.5}\right]^{.523}\left(\frac{1-x}{x}\right)^{(.9)(.523)}\right\}^2$$

$$\left(\frac{dP}{dz}\right)_g = \frac{a}{g_o}\left\{\frac{\rho_\ell}{\rho_v} - \frac{(\rho_\ell - \rho_v)}{\rho_v}\left[\frac{1}{1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_v}{\rho_\ell}\right)^{2/3}}\right]\right\}$$

$$\left(\frac{dP}{dz}\right)_m =$$

$$\frac{-G^2}{\rho_v\ g_o}\left[2x + (1-2x)\left(\frac{\rho_v}{\rho_\ell}\right)^{1/3} + (1-2x)\left(\frac{\rho_v}{\rho_\ell}\right)^{2/3} - 2(1-x)\left(\frac{\rho_v}{\rho_\ell}\right)\right]\frac{dx}{dz}$$

Let us integrate to find the pressure drop due to momentum change

$$\int_{P_i}^{P_f} dP_m = -\int_{x_i}^{x_f}\frac{G^2}{\rho_v g_o}[\ ]dx$$

$$(P_f - P_i)_m =$$

$$\frac{-G^2}{\rho_v\ g_o}\left[x^2 + \left(\frac{\rho_v}{\rho_\ell}\right)^{1/3}(x - x^2) + \left(\frac{\rho_v}{\rho_\ell}\right)^{2/3}(x - x^2) - 2\frac{\rho_v}{\rho_\ell}\left(x - \frac{x^2}{2}\right)\right]\Bigg|_{x_i}^{x_f}$$

$$(P_f - P_i)_m = \frac{G^2}{\rho_v\ g_o}\left\{\left[1 + \left(\frac{\rho_v}{\rho_\ell}\right) - \left(\frac{\rho_v}{\rho_\ell}\right)^{1/3} - \left(\frac{\rho_v}{\rho_\ell}\right)^{2/3}\right](x_f^2 - x_i^2)\right.$$

$$\left. - \left[2\left(\frac{\rho_v}{\rho_\ell}\right) - \left(\frac{\rho_v}{\rho_\ell}\right)^{1/3} - \left(\frac{\rho_v}{\rho_\ell}\right)^{2/3}\right](x_f - x_i)\right\}$$

Next consider the friction term.

Assume that quality varies linearly with length.

$$x = C_1 \, dz + B$$

$$dx = C_1 \, dz$$

$$\int_{P_i}^{P_f} d\,P_f = - \int_{x_i}^{x_f} C_2 \, \{1 + C_3 \, (\frac{1-x}{x})^{.47}\}^2 \, x^{1.8} \, dx$$

where

$$C_2 \equiv \frac{(.09)\, \mu_v^{.2}\, G^{1.8}}{C_1 \, g_o \, \rho_v \, D^{1.2}}$$

$$C_3 \equiv 2.85 \, (\frac{\mu_\ell}{\mu_v})^{.0523} \, (\frac{\rho_v}{\rho_\ell})^{.262}$$

$$(P_f - P_i)_f = - \int_{x_i}^{x_f} C_2 \, [1 + 2\,C_3 \, (\frac{1-x}{x})^{.47} + C_3^2 \, (\frac{1-x}{x})^{.94}] \, x^{1.8} \, dx$$

$$= - C_2 \, \frac{x^{2.8}}{2.8} \Big|_{x_i}^{x_f} - 2\, C_2 \, C_3 \int_{x_i}^{x_f} (1-x)^{.47} \, x^{1.33} \, dx$$

$$- C_2 \, C_3^2 \int_{x_i}^{x_f} (1-x)^{.94} \, x^{.86} \, dx$$

Using the binomial expansion

$$(1 \pm x)^m = 1 \pm mx + \frac{m\,(m-1)\,x^2}{2!} \pm \frac{m\,(m-1)\,(m-2)\,x^3}{3!} \quad \ldots .$$

$$0 \leq x \leq 1$$

We may evaluate the last two terms above. Consider first:

$$\int (1 - x)^{.47} x^{1.33} dx =$$

$$\int x^{1.33} [1 - .47x + \frac{(.47)\ (-.53)}{2} x^2 - \frac{(.47)\ (-.53)\ (-1.53)\ x^3}{6} + \ldots]dx$$

$$\int (1 - x)^{.47} x^{1.33} dx =$$

$$\{ \frac{x^{2.33}}{2.33} \frac{- .47\ x^{3.33}}{3.33} - \frac{(.47)\ (.53)}{(2)\ (4.33)} x^{4.33} - \frac{(.47)\ (.53)(1.53)}{(6)\ (5.33)} x^{5.33} + \ldots\}\Big|_{x_i}^{x_f}$$

Now calculate the ratio of the first and second terms, the second and third terms, and the third and first terms

$$R_{2-1} = \frac{\frac{.47}{3.33} x^{3.33}}{\frac{x^{2.33}}{2.33}} = .329\ x$$

$$R_{3-2} = \frac{\frac{(.47)\ (.53)}{(2)\ (4.33)} x^{4.33}}{\frac{(.47)}{(3.33)} x^{3.33}} = .204\ x$$

$$R_{3-1} = (.204)\ (.329)\ x^{.2} = .0672\ x^{2}$$

Since $0 \leq x \leq 1$

We see that we may truncate after the first 3 terms to get

$$\int_{x_i}^{x_f} (1 - x)^{.47} x^{1.53} dx =$$

$$\{\frac{1}{2.33} - \frac{.47}{3.33} x - \frac{(.47)\ (.53)}{(2)\ (4.33)} x^2\} x^{2.33}\Big|_{x_i}^{x_f}$$

Performing a similar expansion on the second integral and truncating after the second term we find

$$\int_{x_i}^{x_f} (1-x)^{.94}\, x^{.86}\, dx = \left\{\frac{1}{1.86} - \frac{.94}{2.86}\, x\right\} x^{1.86} \Big|_{x_i}^{x_f}$$

If we neglect the gravity or external acceleration force term, $\left(\frac{dP}{dz}\right)_g = 0$, we arrive at the following expression

$$(P_f - P_i)_{total} = (P_f - P_i)_m + (P_f - P_i)_f$$

$$(P_f - P_i)_m = \frac{-G}{\rho_v\, g_o} \left\{\left[1 + \left(\frac{\rho_v}{\rho_\ell}\right) - \left(\frac{\rho_v}{\rho_\ell}\right)^{1/3} - \left(\frac{\rho_v}{\rho_\ell}\right)^{2/3}\right] (x_f^2 - x_i^2)\right.$$

$$\left. - \left[2\left(\frac{\rho_v}{\rho_\ell}\right) - \left(\frac{\rho_v}{\rho_\ell}\right)^{1/3} - \left(\frac{\rho_v}{\rho_\ell}\right)^{2/3}\right] (x_f - x_i)\right\}$$

$$(P_f - P_i)_f = -C_2\{.357\, x^{2.8} + 2\, C_3[.429 - .141\, x - .0288\, x^2]\, x^{2.33}$$

$$+ C_3^2\, [.538 - .329\, x]\, x^{1.86}\} \Big|_{x_i}^{x_f}$$

$$C_3 \equiv 2.85 \left(\frac{\mu_\ell}{\mu_v}\right)^{.0523} \left(\frac{\rho_v}{\rho_\ell}\right)^{.262}$$

$$C_2 \equiv \frac{(.09)\, \mu_v^{.2}\, G^{1.8}}{C_1\, g_o\, \rho_v\, D^{1.2}}$$

$$C_1 = \frac{(x_f - x_i)}{(z_f - z_i)}$$

The above expressions for total two-phase pressure drop, along with expressions for single phase region pressure drop, are used in subroutine PDROP to determine total pressure drop in the evaporator and condenser. See comments in the program listing at the end of this section for more details.

Single Phase Pressure Drops in Heat Exchangers

Derivation of the vapor region pressure drop in the heat exchangers, accounting for density change, is as follows:

$$P_i - P_f = \frac{G^2}{\alpha} \left(\frac{1}{\rho_f} - \frac{1}{\rho_i}\right) + 4f \frac{L}{D} \frac{G^2}{2\,\rho_m} + \cancelto{0}{g\rho_m (h_f - h_i)}$$

$$\frac{1}{\rho_m} = \frac{\left(\frac{1}{\rho_f} + \frac{1}{\rho_i}\right)}{2}$$

$$\alpha \simeq 1$$

$$v = \frac{1}{\rho}$$

$$(P_i - P_f)_{vapor} = G^2 \left[(v_f - v_i) + f \frac{L}{D} (v_f + v_i)\right]$$

where

f = Moody Friction Factor

The expression of the liquid phase pressure drop in the heat exchangers is the normal incompressible flow relation

$$\Delta P = 4\,f\,\frac{L}{D}\,\frac{G^2}{2\,\rho_\ell}$$

Where f = Moody Friction Factor.

See comments in the PDROP program listing at the end of this section for more details.

Single Phase Line Pressure Drops

The equivalent length method is used to account for pressure drop in the connecting piping. The standard incompressible flow relation is used except that an estimated equivalent value of L/D is used, instead of the actual L and D.

$$\Delta P = 4\ f\ \left(\frac{L}{D}\right)\ \frac{G^2}{2\rho}$$

Where f = Moody friction factor.

Subroutine DPLINE is used to calculate pressure drops in this manner. For more details, see comments in the program listing at the end of this section.

Note that the programs previously described used subroutine FRICT to estimate the Moody friction factor. This subroutine accepts, among other inputs, a value for the tube wall surface roughness, and it can reproduce the entire Moody friction factor plot as shown in Figure L-1 [4]. This has been used instead of the standard laminar and turbulent limits for smooth pipe because in some applications, the connecting piping can be far from smooth. See comments in the FRICT program listing at the end of this section for more details.

Air-Side Pressure Drops

Kays & London suggest in, Compact Heat Exchangers[5], that for air-side coefficients, better correlation of data is achieved using outside tube diameter, rather than the equivalent diameter as is used in their book. For this reason, the Kays & London data for tubes with plate fins has been replotted, correlated on the basis of tube outside diameter, for heat exchangers with round tubes in cross-flow. As seen in Figure L -2 there is generally poor correlation between different heat exchangers for the friction factor. The author has developed a new method of correlation which accounts for $\sigma \equiv$ air flow area/total frontal area. As can be seen in Figure L-3, accounting for this effect in the indicated manner produces excellent correlation between different heat exchangers. The resulting expression is

$$f_{air}^{\sigma} = .367 \ Re^{-.108}$$

and the total air side pressure drop expression becomes:

$$\Delta P_{air} = f_{air} \frac{t}{\left[\frac{4\,\sigma}{\alpha_{air}}\right]} \frac{G^2}{2\,\rho_{air}\,g_o}$$

Where:

$$\left.\begin{aligned} \sigma &\equiv \frac{\text{Air Flow Area}}{\text{Total Frontal Area}} \\ \alpha_{air} &= \frac{\text{Total Air-Side Heat Transfer Area}}{\text{Total Heat Exchanger Volume}} \end{aligned}\right\} \text{See Discussion of Evaporator and Condenser}$$

t = Total Thickness of Heat Exchanger

$$Re = \frac{G\ D}{\mu}$$

D = Tube Outside Diameter

These relations can be used to estimate the magnitude of air-side flow losses through the coils.

References

1. Traviss, D.P., Baron, A.E., and Rohsenow, W. M., "Forced Convection Condensation Inside Tubes", Report No. 72591-74; Heat Transfer Laboratory, Massachusetts Institute of Technology, Cambridge, Mass. (ASHRAE Contract No. RP63).

2. Tong, L. S., Boiling Heat Transfer And Two-Phase Flow (New York: John Wiley & Sons, Inc., 1965) Cpt. 4.

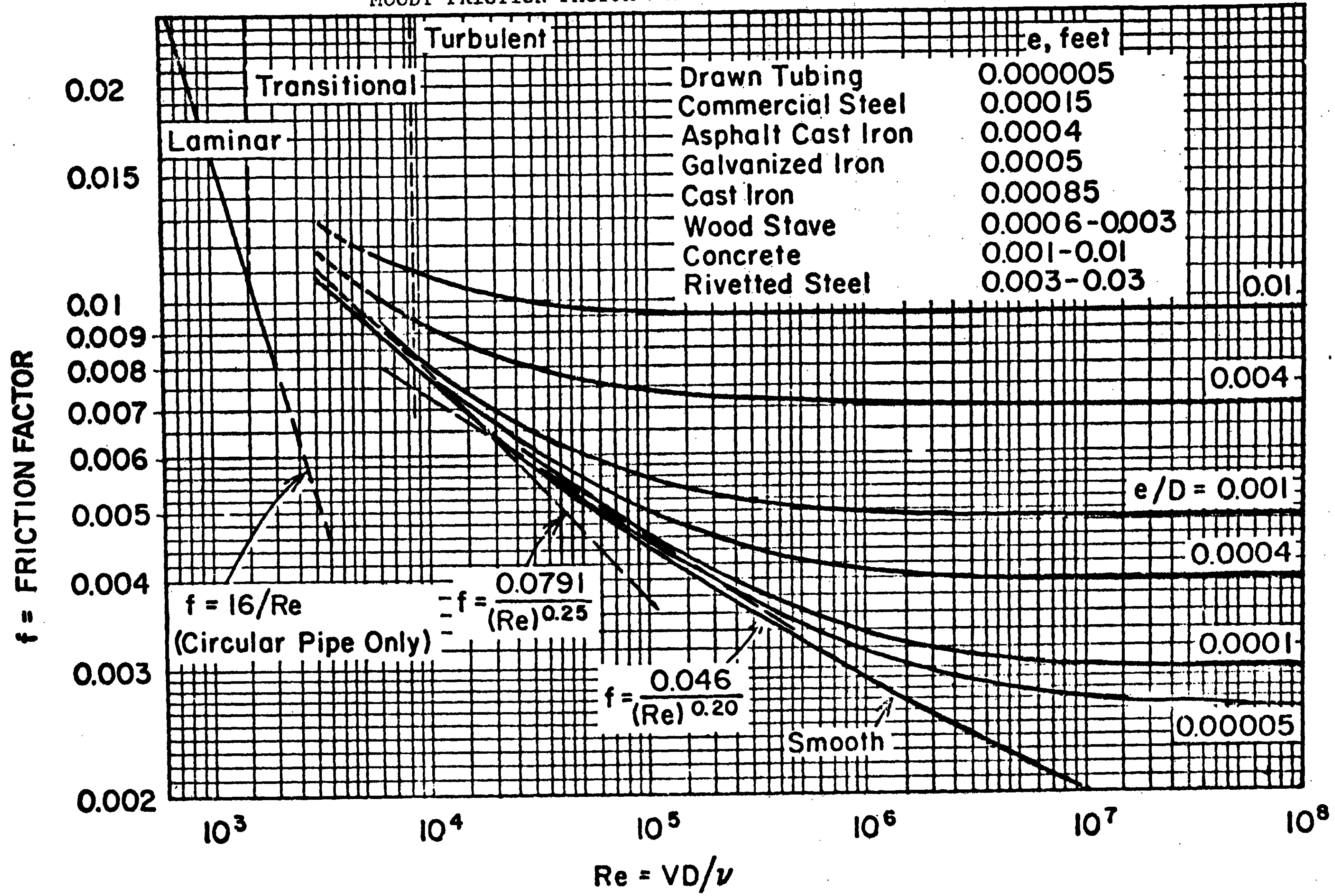
3. Lockhart, R. W. and Martinelli, R.C., "Proposed Correlation of Data for Isothermal Two-Phase, Two-Component Flow in Pipes", Chemical Engineering Progress, Vol. 45, No. 1, pg. 39 (1949).

4. Miller, D. S., Internal Flow: A Guide To Losses In Pipe And Duct Systems, (Cranfield, Belford, England: The British Hydromechanics Research Assoc., 1971).

5. Kays, W. M. and London, A. L., Compact Heat Exchangers (Palo Alto, California: The National Press, 1955).

FIGURE L-1

MOODY FRICTION FACTOR FOR FLOW IN CIRCULAR PIPES



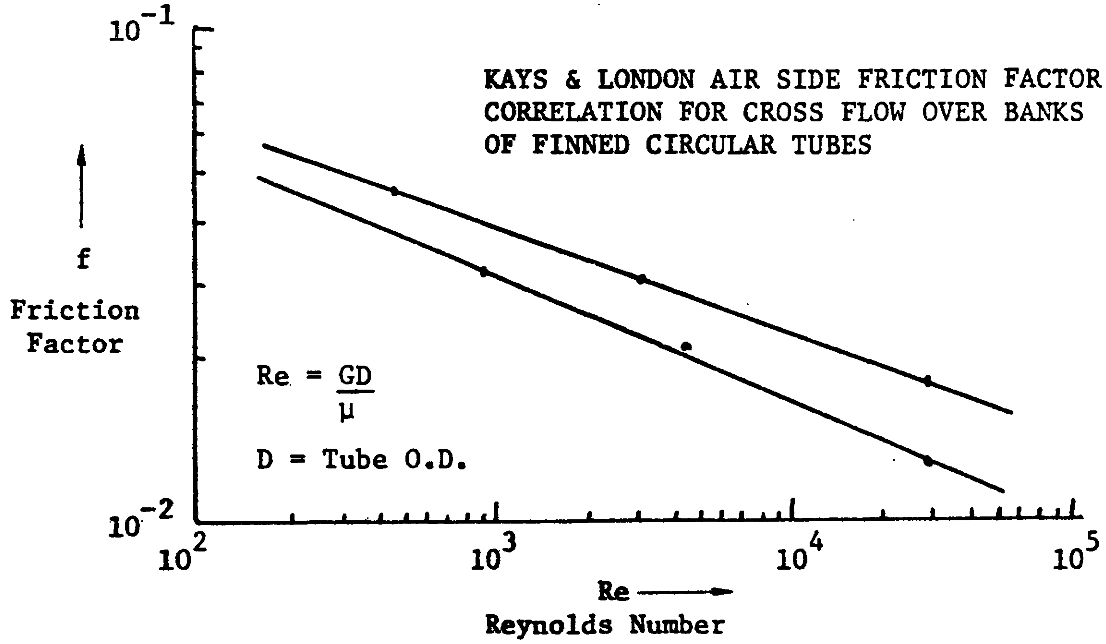


FIGURE L-2

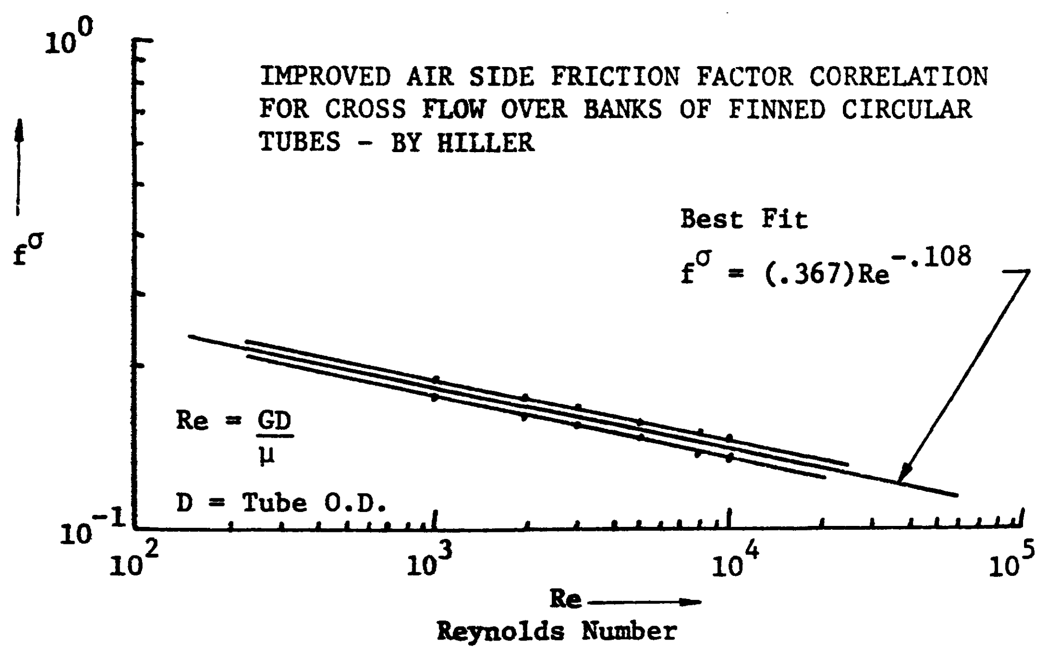


FIGURE L-3

```
      SUBROUTINE PDROP(N,D,E,G,XMUV,XMUL,RHOV,RHOL,REV,REL,
     1DZTP,XF,XI,VV,DZV,DZL,PD)
C
C     PURPOSE
C        TO DETERMINE BOTH SINGLE PHASE AND TWO-PHASE
C        PRESSURE DROPS FOR FLOW IN TUBES
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        N     -  INDICATOR (2 OR 3 MEANS EVAPORATOR)
C        D     -  EQUIVALENT DIAMETER OF FLOW PASSAGE (FT)
C        E     -  SURFACE ROUGHNESS OF FLOW PASSAGE (FT)
C        G     -  MASS FLOW PER UNIT AREA (LBM/HR-SQ FT)
C        XMUV  -  VISCOSITY OF VAPOR PHASE (LBM/HR-FT)
C        XMUL  -  VISCOSITY OF LIQ. PHASE (LBM/HR-FT)
C        RHOV  -  DENSITY OF VAPOR PHASE (LBM/CU FT)
C        RHOL  -  DENSITY OF LIQ. PHASE (LBM/CU FT)
C        REV   -  REYNOLDS NUMBER OF VAPOR PHASE REGION
C        REL   -  REYNOLDS NUMBER OF LIQUID PHASE REGION
C        DZTP  -  LENGTH OF TWO-PHASE REGION (FT)
C        XF    -  FINAL QUALITY
C        XI    -  INITIAL QUALITY
C        VV    -  EXIT SPECIFIC VOL.OF VAPOR PHASE (CU FT/LBM)
C        DZV   -  LENGTH OF SINGLE PHASE VAPOR REGION (FT)
C        DZL   -  LENGTH OF SINGLE PHASE LIQ. REGION (FT)
C      OUTPUT
C        DPV   -  PRES.DROP IN SINGLE PHASE VAPOR REGION (PSI)
C        DPL   -  PRES. DROP IN SINGLE PHASE LIQ. REGION (PSI)
C        DPTP  -  PRES.DROP IN TWO-PHASE REGION (PSI)
C        PD    -  TOTAL PRESSURE DROP (PSI)
C **** CAUTION - WATCH SIGN CONVENTION ***********
C
C     REMARKS
C        THIS PROGRAM CALLS SUBROUTINE FRICT, FOR
C        DETERMINING THE GENERAL MOODY FRICTION FACTOR
C        FOR SINGLE PHASE FLOW IN TUBES
C
C     MOMENTUM COMPONENT OF TWO-PHASE PRES. DROP
      DPM =((XF**2-XI**2)*(1.0+RHOV/RHOL -(RHOV/RHOL)**.333
     1-(RHOV/RHOL)**.667)-(XF-XI)*(2.0*RHOV/RHOL-(RHOV/
     2RHOL)**.333-(RHOV/RHOL)**.667))*G**2/(RHOV*32.2*
     33600.0**2*144.0)
      C1 = (XF-XI)/DZTP
      C2 = .09*XMUV**.2*G**1.8/(C1*RHOV*D**1.2*32.2*3600.0
     1**2*144.0)
      C3 = 2.85*(XMUL/XMUV)**.0523*(RHOV/RHOL)**.262
C
C     FRICTION COMPONENT OF TWO-PHASE PRES. DROP
      DPF =C2*(.357*(XF**2.8-XI**2.8)+2.0*C3*(.429*(XF**2.33
```

```
     1-XI**2.33)-.141*(XF**3.33-XI**3.33)-.0287*(XF**4.33
     2-XI**4.33))+C3**2*(.538*(XF**1.86-XI**1.86)-.329*(XF
     3**2.86-XI**2.86)))
C
C     TOTAL TWO-PHASE PRESSURE DROP
      DPTP = DPM + DPF
      CALL FRICT(REV,E,D,FFV)
      IF((N.EQ.2).OR.(N.EQ.3)) GO TO 20
C
C     CONDENSER SINGLE PHASE PRESSURE DROPS
      DPV=G**2*(1.0/RHOV-VV+FFV*DZV*(1.0/RHOV+VV)/D)/
     1(32.2*3600.0**2*144.0)
      CALL FRICT(REL,E,D,FFL)
      DPL = 2.0*FFL*DZL*G**2/(D*RHOL*32.2*3600.0**2*144.0)
      GO TO 40
C
C     EVAPORATOR SINGLE PHASE PRESSURE DROPS
   20 DPV=G**2*(VV-1.0/RHOV+FFV*DZV*(VV+1.0/RHOV)/D)/
     1(32.2*3600.0**2*144.0)
      DPL = 0.0
C
C     TOTAL PRESSURE DROP
   40 PD = -(DPTP + DPV + DPL)
      WRITE(5,500) DZV,DZL,DZTP,DPV,DPL,DPTP
      RETURN
  500 FORMAT('    DZV=',F10.4,'    DZL=',F10.4,'DZTP=',F10.4,
     1'    DPV=',F10.4,'    DPL=',F10.4,'    DPTP=',F10.4)
      END
```

```
      SUBROUTINE DPLINE(D,XLEQ,E,XMR,RHO,XMU,DPLNE)
C
C      PURPOSE
C         TO DETERMINE SINGLE PHASE PRESSURE DROPS
C
C      DESCRIPTION OF PARAMETERS
C       INPUT
C         D     =  EQUIVALENT DIAMETER OF FLOW PASSAGE (FT)
C         XLEQ  =  EQUIVALENT LENGTH (L/D - NON DIMENSIONAL)
C         E     =  SURFACE ROUGHNESS OF FLOW PASSAGE (FT)
C         XMR   =  MASS FLOW RATE (LBM/HR)
C         RHO   =  DENSITY OF FLUID (LBM/CU FT)
C         XMU   =  VISCOSITY OF FLUID (LBM/HR-FT)
C       OUTPUT
C         DPLNE=   SINGLE PHASE PRESSURE DROP (PSI)
C
C      REMARKS
C         THIS PROGRAM CALLS SUBROUTINE FRICT FOR
C         DETERMINING THE GENERAL MOODY FRICTION FACTOR
C         FOR SINGLE PHASE FLOW IN TUBES
      RE = 4.0*XMR/(3.14*D*XMU)
      CALL FRICT(RE,E,D,FF)
      DPLNE =4.0*FF*XLEQ*(4.0*XMR/(3.14*D**2))**2/
     1(2.0*RHO*32.2*3600.0**2*144.0)
      RETURN
      END
```

```
      SUBROUTINE FRICT(RE,E,DI,FF)
C
C     PURPOSE
C        TO DETERMINE THE GENERAL MOODY FRICTION FACTOR
C        FOR SINGLE PHASE FLOW IN TUBES
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        RE    -   REYNOLDS NUMBER
C        E     -   SURFACE ROUGHNESS OF FLOW PASSAGE (FT)
C        DI    -   EQUIVALENT DIAMETER OF FLOW PASSAGE (FT)
C      OUTPUT
C        FF    -   MOODY FRICTION FACTOR
C
C     LAMINAR FLOW REGIME
      IF(RE.LE.2300.0) FF = 16.0/RE
      IF(RE.LE.2300.0) GO TO 30
C
C     TRANSITION AND TURBULENT FLOW REGIMES
      D = DI
      FF = .020
      DF = -.005
      DO 20 I = 1,30
      FF = FF + DF
      IF(FF.LE.0.0) FF = .0001
      A = -2.0*ALOG10(2.51/(RE*SQRT(4.0*FF)) + E/(3.7*D))
      B = 1.0/SQRT(4.0*FF)
      IF(ABS(A-B).LE.(.001*B)) GO TO 30
      IF(A-B) 15,30,20
   15 FF = FF - DF
      DF = DF/2.0
   20 CONTINUE
      WRITE(5,100)
   30 RETURN
  100 FORMAT(' *****FRICTION FACTOR FAILS TO CONVERGE*****')
      END
```

APPENDIX M

DETAILS OF CROSS-FLOW EVAPORATOR MODELING

Details of the general air-conditioning or heat pump type cross-flow evaporator model 'EVAP', and of the special case, finned tube evaporator model, are given in this section, followed by computer program listings for each.

General Model 'EVAP'

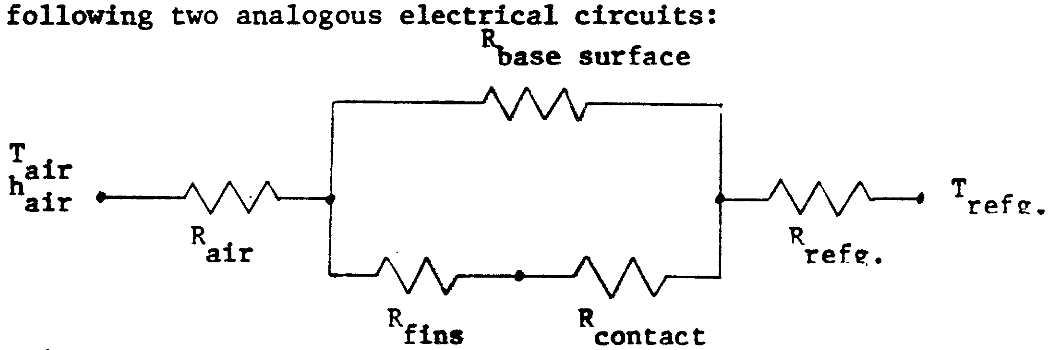
The effectiveness - NTU method of heat exchanger analysis is applicable in the single phase region of the evaporator, or for the entire evaporator if no moisture removal occurs. The analysis is exactly the same as for the general condenser model 'EXCH' of Appendix I, except that in the evaporator, the air is the hotter fluid instead of the colder fluid.

In the event of moisture removal, which is determined automatically by the model, a modified version of the effective surface temperature approach discussed by McElgin and Wiley[1] is used. It is assumed that all moisture removal, if it occurs, takes place only in the two-phase region. The effective surface temperature approach assumes that the air side heat transfer coefficient is unaffected by the presence of water on the surface of the coil, and uses a heat transfer-mass transfer analogy to determine the amount of moisture removed. A total driving enthalpy difference between bulk air and effective surface conditions, accounting for enthalpy of the moisture in the air, is used to determine the heat transfer rate. Full psychrometric chart data is used in the modified method presented here, as opposed to the approximate method used by McElgin and Wiley, which linearized local sections of the psychrometric charts. Subroutine 'XMOIST', as listed at the end of this section, is a program which produces psychrometric chart data in the range $-30 \leq T_{wet\ bulb} \leq 100^{o}F$.

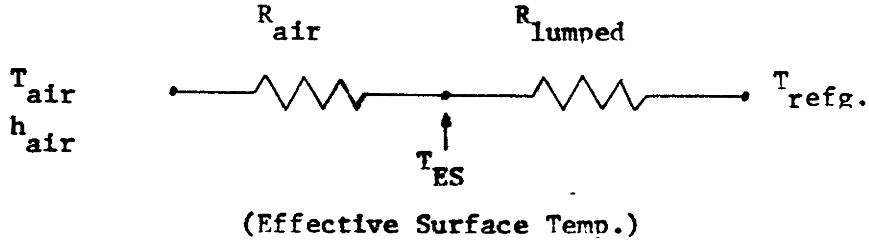
The procedure for determining sensible and latent heat transfer characteristics, and fractions of coil used for evaporating and superheating, as shown in the flow diagram in Figure M-1, is as follows:

Determine the representative coil characteristic 'COIL' using the representative surface temperature technique.

The heat transfer through the coil surface can be modeled by the following two analogous electrical circuits:



and



Now, since it is assumed that the presence of water droplets on the coil surface does not affect the resistance to heat transfer we can define 'R_{tot}' , the total resistance to heat transfer:

$$R_{tot} = \frac{1}{dA_{R_{ht}}} \left[\frac{1}{\eta_o h_a \frac{\alpha_{air}}{\alpha}} + \frac{1}{h_R} \right]$$

also

$$R_{tot} = \frac{1}{h_a \, dA_{a_{ht}}} + R_{lumped}$$

where:

η_o = Overall surface efficiency, allowing for an extended surface on the air side, and including contact resistance, as described in Appendix J.

h = Heat transfer coefficienct (Btu/hr-ft^2-^{o}F)

α_a = Air side heat transfer area/total heat exchanger volume $(\frac{1}{ft})$

α_R = Refrigerant side heat transfer area/total heat exchanger volume $(\frac{1}{ft})$

$dA_{R_{ht}}$ = Unit Heat transfer area on refrigerant side

$dA_{a_{ht}}$ = Unit heat transfer area on air side

a = Subscript indicating air side

R = Subscript indicating refrigerant side

Hence:

$$R_{lumped} = R_{tot} - \frac{1}{h_a \, dA_{a_{ht}}}$$

Then the total heat 'dq' lost by the air in passing over an element of wetted area dA_w is:

$$dq = h_a \, dA_w \, \frac{(i - i_s)}{Cp_w}$$

Where

i = Bulk enthalpy of the air (including moisture) (Btu/lbm dry air)

i_s = Enthalpy of the air at the surface of the coil (Btu/lbm dry air)

Cp_w = Moist air specific heat (Btu/lbm dry air -^{o}R)

Similarly

$$dq = \frac{1}{R_{lumped}} \, dA_w \, (T_{ES} - T_R)$$

Equating we find:

$$\text{'COIL'} \equiv \frac{(T_{ES} - T_R)}{(i - i_s)} = \frac{h_a \, R_{lumped}}{Cp_w}$$

Where Cp_w can be approximated by the normal dry air specific heat over our range of interest,

$$Cp_w = Cp_a = .24 \text{ (Btu/lbm dry air -}^o\text{R)}$$

providing we use wet bulk temperatures when computing enthalpy change of the air.

Next determine the amount of heat 'Q_{tp}' transferred in the two-phase region, assuming complete evaporation

$$Q_{tp} = \dot{m}_R (1 - X_4)\, h_{fg} \qquad \text{(Btu/hr)}$$

where

h_{fg} = Latent heat of vaporization (Btu/lbm)

X_4 = Quality of mixture entering evaporator

$\dot{m}_R$ = Mass flow rate of refrigerant (lbm/hr)

Then, using subroutine 'XMOIST' for psychrometric chart data, determine the dew point temperature of the entering air.

Determine the bulk air dry bulb temperature when the surface temperature 'T_{wall}' drops below the dew point.

$$T_{db_{MR}} = \frac{\{T_{wall}\,[\frac{1}{\eta_o} + \frac{\alpha_a}{\alpha_R}\,\frac{h_a}{h_{R_{tp}}} - T_{R_{sat}}\}}{\{\,[\frac{1}{\eta_o} + \frac{\alpha_a}{\alpha_R}\,\frac{h_a}{h_{R_{tp}}}] - 1\}}$$

Where subscripts mean:

db = Dry bulb

sat = Saturation condition

tp = Two-phase region

MR = moisture removal

Determine NTU and effectiveness in two-phase region, assuming no moisture removal.

$$NTU_{tp} = \frac{AOM}{Cp_a \left[\frac{\alpha_R}{\eta_o \; h_a \; \alpha_a} + \frac{1}{h_{R_{tp}}} \right]}$$

$$\varepsilon_{tp} = 1 - e^{-NTU_{tp}}$$

Where:

$$AOM \equiv \frac{dA_{R_{ht}}}{d \; \dot{m}_a}$$

$d \; \dot{m}_a =$ Local flow rate of air (lbm/hr)

Using the above effectiveness, determine exit air dry bulb temperature, assuming no moisture removal.

$$T_{a_{out_{tp}}} = T_{a_{in}} - \varepsilon_{tp} \; (T_{a_{in_{db}}} - T_{R_{sat}})$$

If $T_{a_{out_{tp}}} \leq T_{db_{MR}}$, then moisture removal occurs, hence do moisture removal analysis. If moisture removal did not occur:

$$F_{tp} = \frac{Q_{tp}}{\dot{m}_a \; \varepsilon_{tp} \; Cp_a \; (T_{a_{in}} - T_{R_{sat}})}$$

where:

F_{tp} = Two-phase fraction of total heat exchanger surface

and go to the superheating region analysis

Moisture Removal

If $T_{a_{in_{db}}} < T_{db_{MR}}$, then moisture removal begins on the leading edge of the coil hence find the effective surface temperature at the leading edge. This is an iterative process

Guess T_{ES}

Find i_{ES}

$T_{ES}^{*} = T_{R_{sat}} + (i_{in} - i_{ES})$ (COIL)

If $T_{ES}^{*} \neq T_{ES}$

If $T_{a_{in_{db}}} > T_{db_{MR}}$, find fraction of thickness of coil 'F_{SENS}' which is used only for sensible heat transfer.

$$F_{SENS} = \frac{\dot{m}_a \, Cp_a \left[\dfrac{1}{\eta_o h_a} + \dfrac{\alpha_a}{\alpha_R \, h_{R_{tp}}}\right] \ln\left[\dfrac{(T_{a_{in_{db}}} - T_{R_{sat}})}{(T_{db_{MR}} - T_{R_{sat}})}\right]}{A_{R_{ht}} \quad \dfrac{\alpha_a}{\alpha_R}}$$

where

$A_{R_{ht}}$ = Total refrigerant side heat transfer area (ft^2)

$\dot{m}_a$ = Mass flow rate of air (lbm/hr)

The fraction of coil thickness 'F_{MOIST}' used for mositure removal is thus

$$F_{MOIST} = 1 - F_{SENS}.$$

Next, deivide the moisture removal fraction F_{MOIST} into two equal parts and analyze the moisture removal in two steps.

Iterate to find i_2 and T_{ES_2}, the bulk air enthalpy and effective surface temperature at the end of the first moisture removal section.

Guess T_{ES_2} and find T_{wall_2} and i_{wall_2} from psychrometric data

$$i_2 = i_{wall_2} + \frac{(T_{ES_2} - T_{R_{sat}})}{COIL}$$

$$i_2^* = \frac{\left\{i_1 - \left(\frac{F_{MOIST}}{2}\right) (A_{R_{ht}}) \left(\frac{\alpha_a}{\alpha_R}\right) (h_a) \left(T_{wall_1} - T_{wall_2}\right)\right\}}{\left\{(COIL)\ (\dot{m}_2)\ (C_{p_2})\ \ln\left[\frac{(T_{wall_1} - T_{R_{sat}})}{(T_{wall_2} - T_{R_{sat}})}\right]\right\}}$$

If $i_2 \neq i_2^*$

Repeat the above for the second moisture removal section. Then, determine exit air wet bulb temperature and the fraction of the total heat exchanger surface occupied by the two-phase region,

$$F_{tp} = \frac{Q_{tp}}{\dot{m}_a\ (i_{in} - i_{out})}$$

and complete the moisture removal section by finding the exit air dry bulb temperature, and the total amount of water removal from the air

$$\dot{m}_{H_2O} = (w_{a_{in}} - w_{a_{out}})\,(F_{tp})\,\dot{m}_a$$

where

$\dot{m}_{H_2O}$ = Rate of moisture removal (lbm/hr)

$w_{a_{in}}$ = Entering moisture content of air (lbm water/lbm dry air)

$w_{a_{out}}$ = Exit moisture content of air (lbm water/lbm dry air)

Superheating Region

If $F_{tp} \geq 1$, then evaporation is incomplete, and there is no superheating region. In the present model, if such is the case, calculations are terminated. The case of incomplete condensation could easily be handled, however, merely by iterating on exit quality.

If $F_{tp} < 1$, then the superheating fraction 'F_s' of total heat exchanger surface is:

$$F_s = 1 - F_{tp}$$

and we can determine exit refrigerant temperature and heat transfer in the single phase region:

$$\dot{m}_{a_s} = (F_s)\ \dot{m}_a$$

$$C_{a_s} = (\dot{m}_{a_s})\ (Cp_a)$$

$$C_{R_s} = (\dot{m}_R)\ (Cp_{R_V})$$

$$C_{min} = \text{smaller of } C_{a_s} \text{ and } C_{R_s}$$

$$C_{max} = \text{larger of } C_{a_s} \text{ and } C_{R_s}$$

$$R_{tot} = \frac{\left[\dfrac{\alpha_R}{\eta_o\ \alpha_a\ h_a} + \dfrac{1}{h_{R_s}}\right]}{(F_s)\ (A_{R_{ht}})}$$

$$NTU_s = \frac{1}{(R_{tot})\ (C_{min})}$$

$$\epsilon_{XF_s} = f\left(\frac{C_{min}}{C_{max}},\ NTU_s\right)$$

$$T_{R_{out}} = T_{R_{sat}} + (\epsilon_{XF_s})\ \frac{(C_{min})}{(C_{R_s})}\ (T_{a_{in_{db}}} - T_{R_{sat}})$$

$$Q_{SP} = C_{R_s}\ (T_{R_{out}} - T_{R_{sat}})$$

Where

$T_{R_{out}}$ = Temperature of superheated vapor leaving evap. (^oF)

Cp_{R_V} = Specific heat at constant pressure of superheated vapor (Btu/lbm - ^oR)

s = Superheated region

XF = cross-flow

sp = single phase

For more information, see comments in the program listing for subroutine 'EVAP' at the end of this section.

Modeling a Finned Tube Evaporator

The geometry factors necessary for use of general model 'EVAP' are the same as those required in the general condenser model 'EXCH', discussed in section 2.3 and Appendix I, and will not be repeated here.

Having determined the geometry factors, the procedure for determining total evaporator performance, as outlined in the flow chart of Figure M-2, is as follows:

Split the evaporator up into equivalent sub-circuits

$$\dot{m}_a = \frac{\dot{m}_a}{N_{sect}}$$

$$\dot{m}_R = \frac{\dot{m}_R}{N_{sect}}$$

$$A_{R_{ht}} = \frac{A_{R_{ht}}}{N_{sect}}$$

Where:

N_{sect} = Number of parallel flow sub-circuits in the heat exchanger

Then:

Using thermodynamic properties corresponding to the states of interest, determine the heat transfer coefficients as described in Appendix K.

Next, use general evaporator model 'EVAP' to determine performance, for the given geometry factors, temperature, and flow rates.

Using the results from 'EVAP', determine the length of the two-phase and superheating regions:

$$DZTP = \frac{(F_{tp})\ (A_{R_{ht}})}{\pi\ D_i}$$

$$DZV = \frac{(F_s)\ (A_{R_{ht}})}{\pi\ D_i}$$

Where:

D_i = Inside diameter of tubes in the heat exchanger

And then determine the total pressure drop 'PD' as described in Appendix L.

Convert results back to total flow notation:

$$\dot{m}_a = (\dot{m}_a)\ (N_{sect})$$

$$\dot{m}_R = (\dot{m}_R)\ (N_{sect})$$

$$Q_{tot} = (Q_{tot})\ (N_{sect})$$

Finally, using the value of total pressure drop through the coil, determine the drop in saturation temperature through the coil corresponding to the pressure drop. If the drop is greater than 2°F, repeat the analysis using

$$T_{sat_{avg}} = \frac{(T_{R_{sat_{in}}} + T_{R_{sat_{out}}})}{2}$$

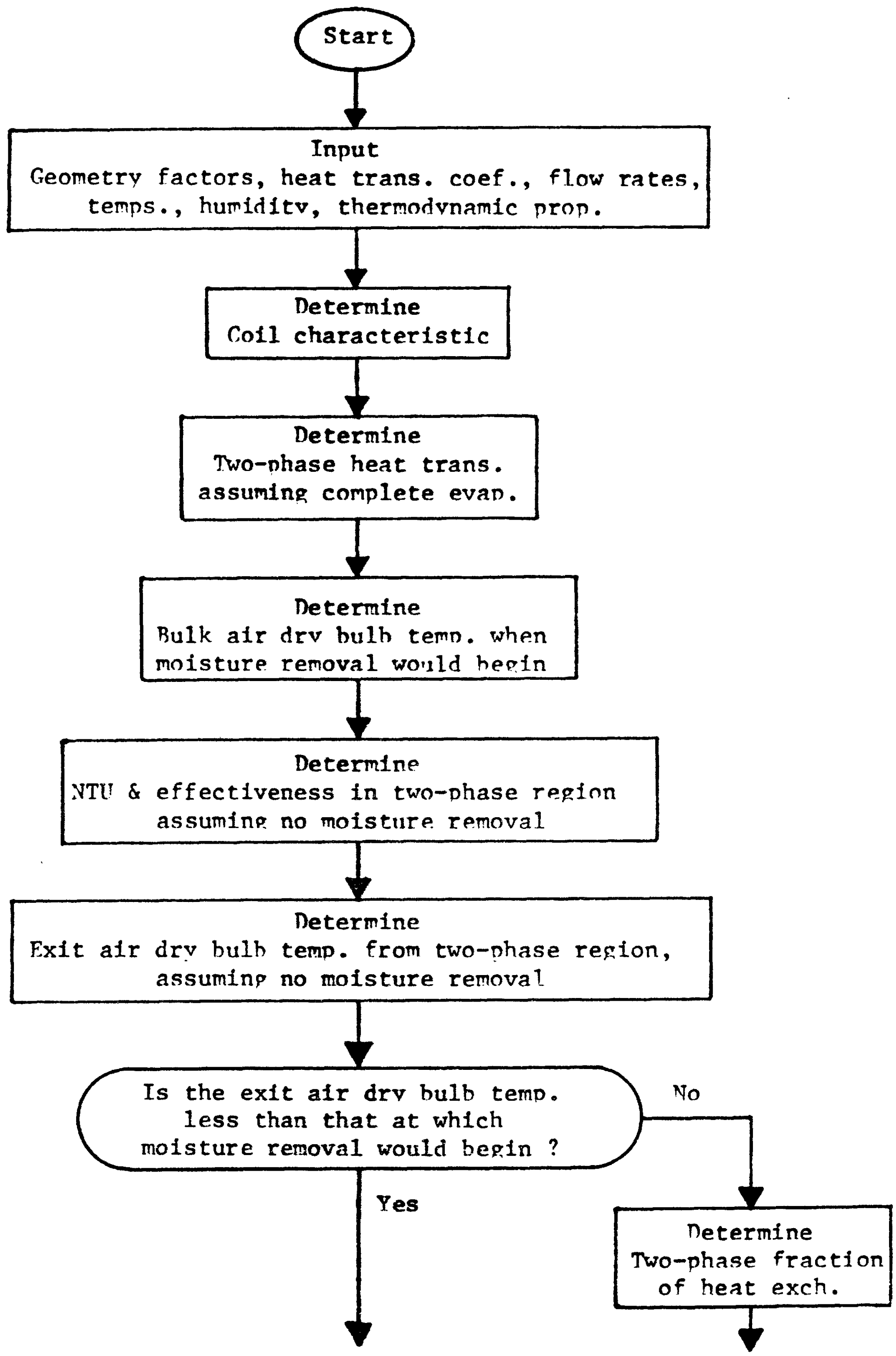
For more information, see comments in the program listing for the finned tube evaporator simulation at the end of this section.

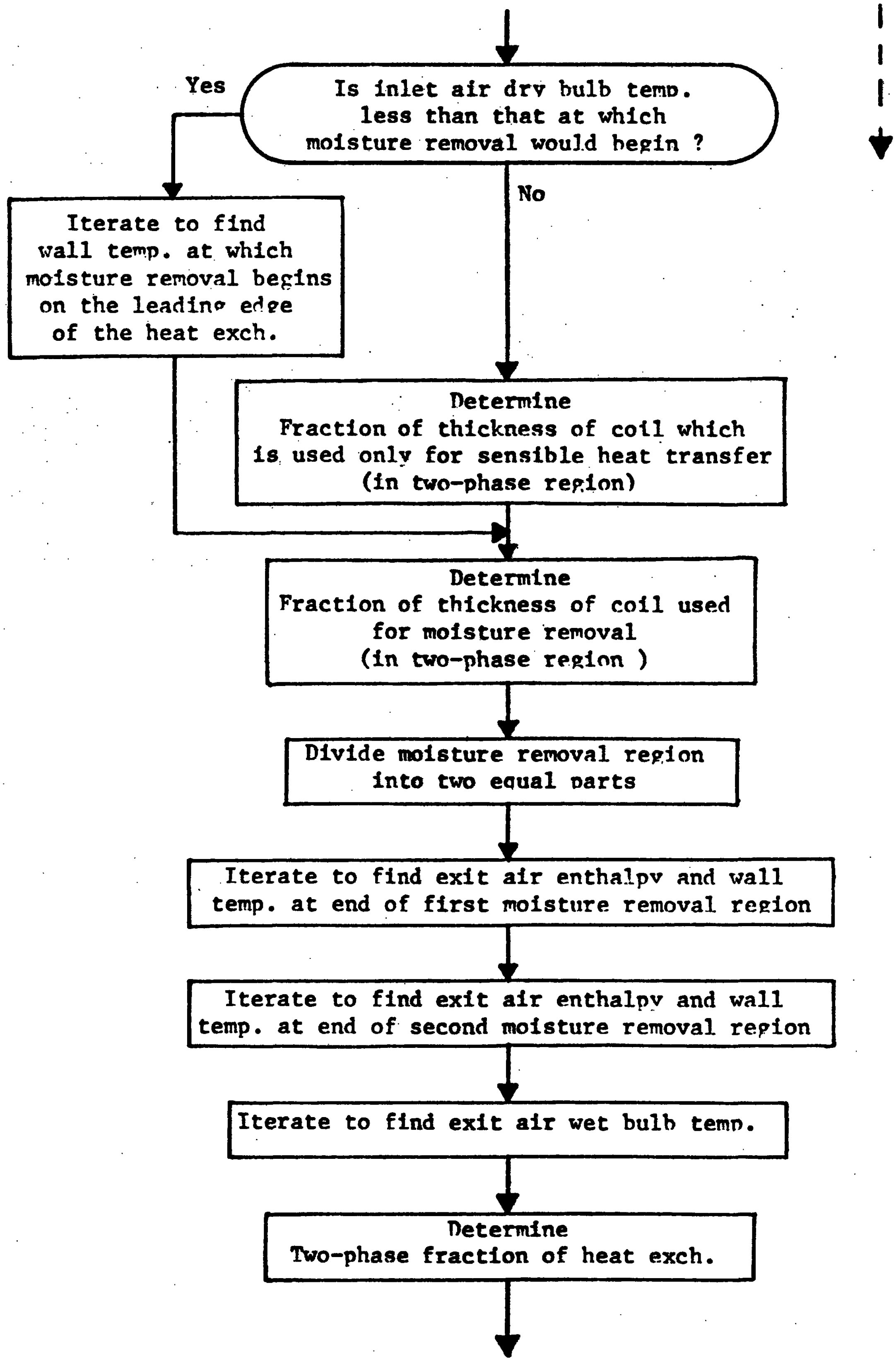
REFERENCES

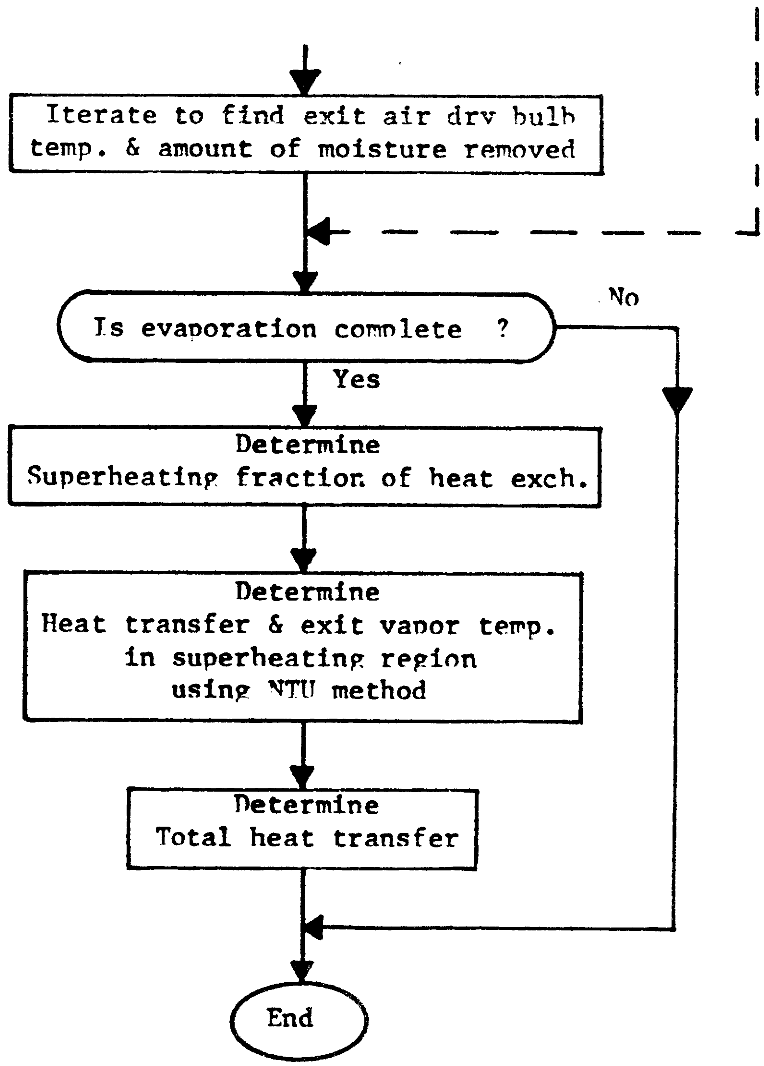
1. McElgin, J., and Wiley, D.C., "Calculation of Coil Surface Areas for Air Cooling and Dehumidification", Heating, Piping, & Air Conditioning, (March, 1940) pg. 195-201.

FIGURE M-1

FLOW CHART FOR GENERAL EVAPORATOR MODEL 'EVAP'



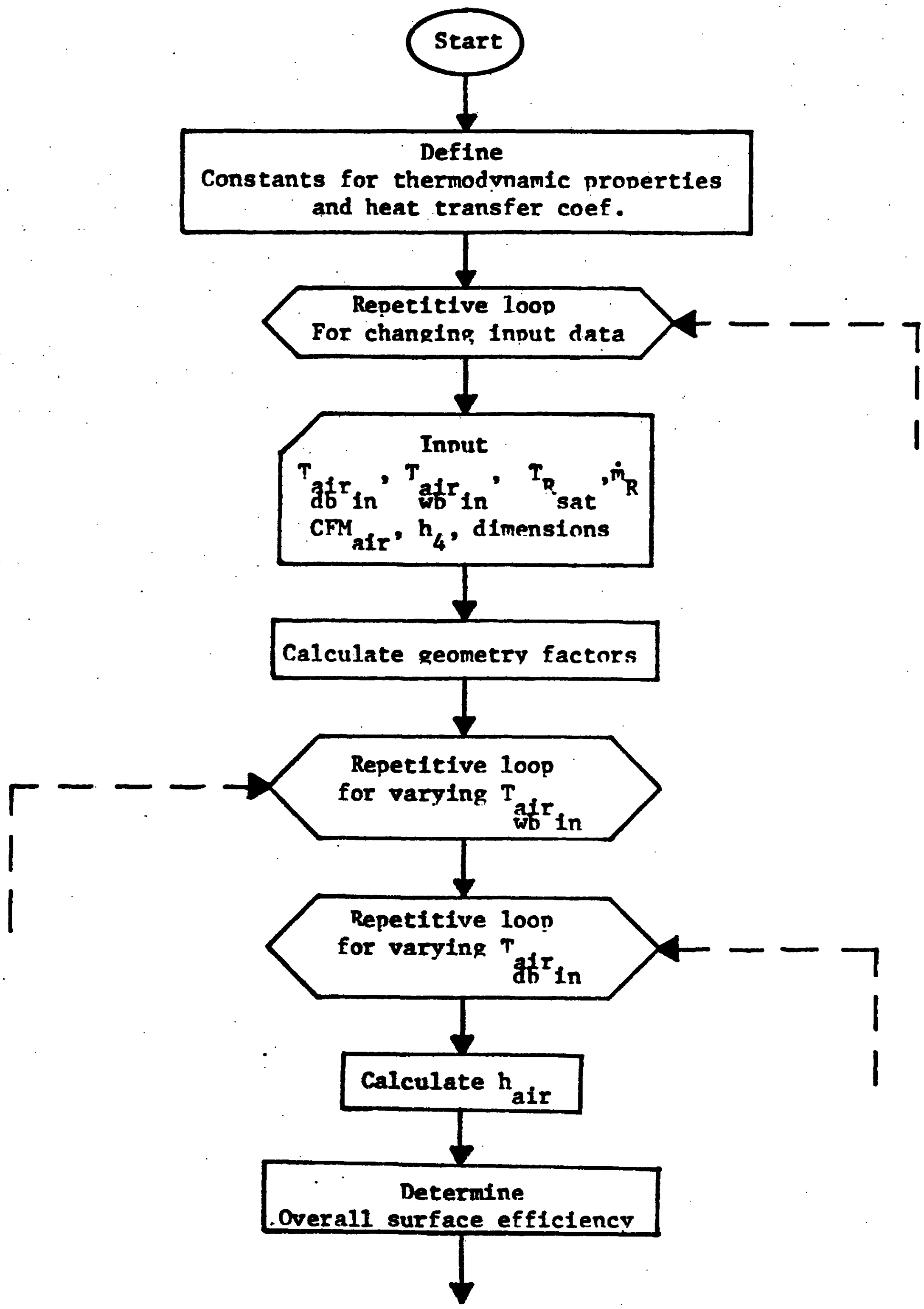

Yes
Is inlet air dry bulb temp. less than that at which moisture removal would begin ?
No
Iterate to find wall temp. at which moisture removal begins on the leading edge of the heat exch.
Determine Fraction of thickness of coil which is used only for sensible heat transfer (in two-phase region)
Determine Fraction of thickness of coil used for moisture removal (in two-phase region)
Divide moisture removal region into two equal parts
Iterate to find exit air enthalpy and wall temp. at end of first moisture removal region
Iterate to find exit air enthalpy and wall temp. at end of second moisture removal region
Iterate to find exit air wet bulb temp.
Determine Two-phase fraction of heat exch.

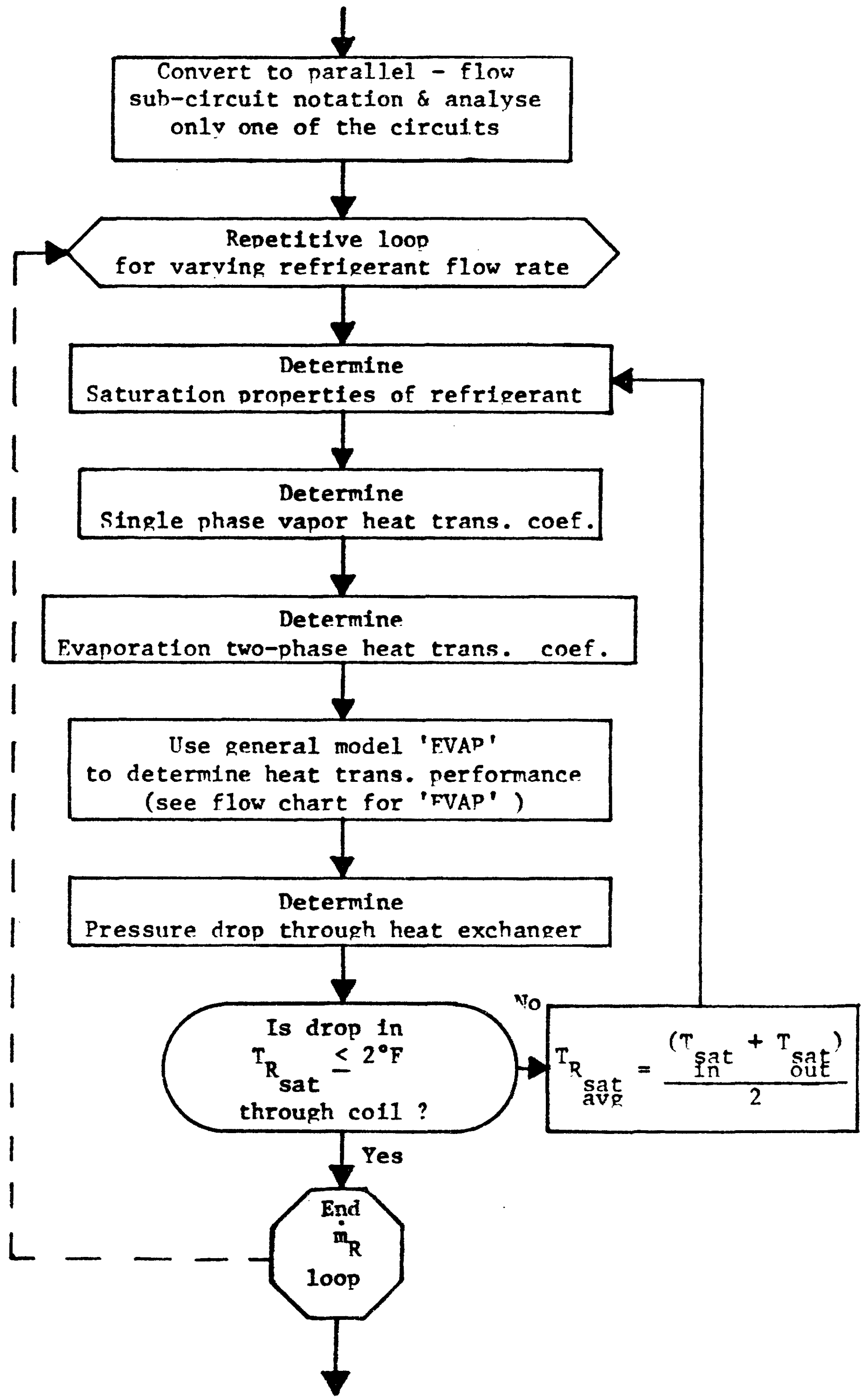


Iterate to find exit air dry bulb temp. & amount of moisture removed
Is evaporation complete ?
No
Yes
Determine
Superheating fraction of heat exch.
Determine
Heat transfer & exit vapor temp. in superheating region using NTU method
Determine
Total heat transfer
End

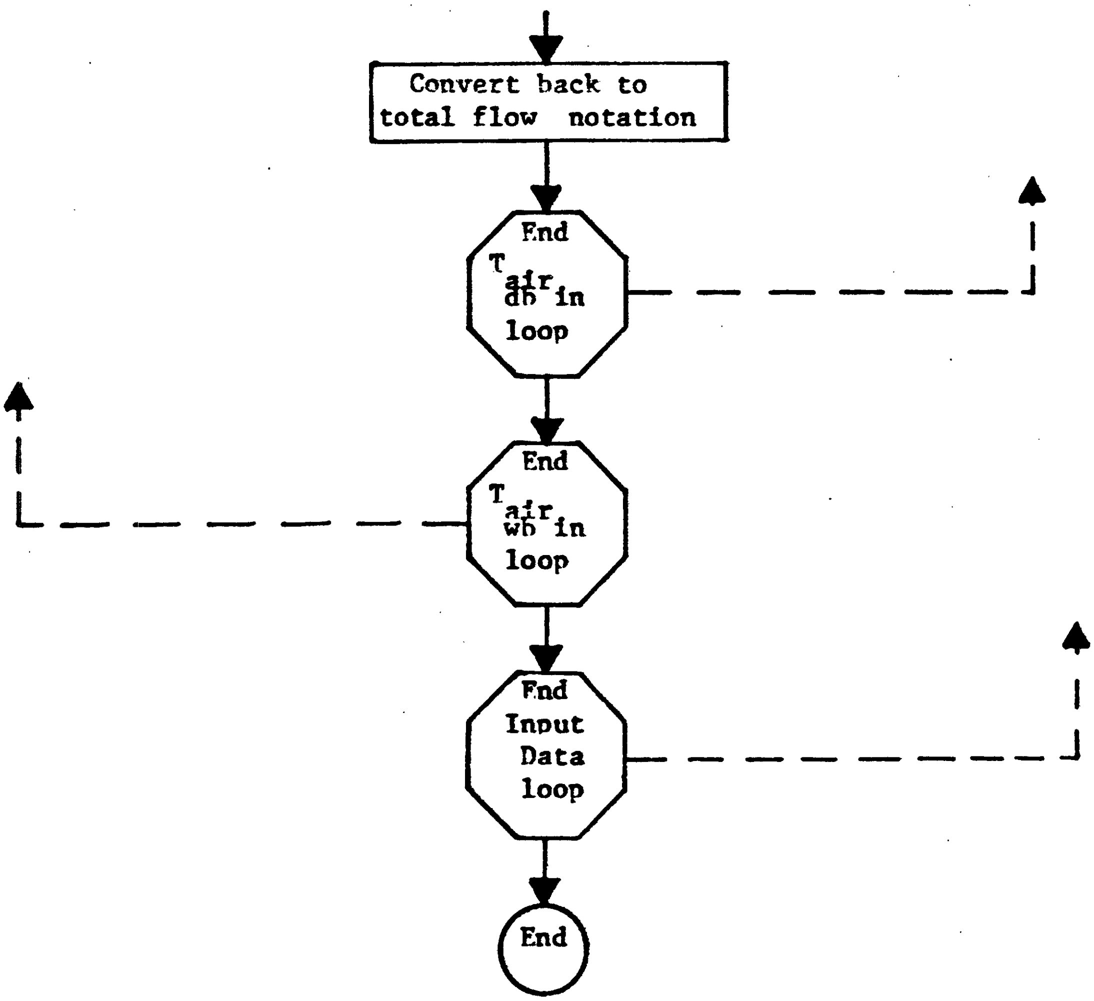
FIGURE M-2

FLOW CHART FOR FINNED TUBE EVAPORATOR MODEL





Convert to parallel - flow sub-circuit notation & analyse only one of the circuits
Repetitive loop for varying refrigerant flow rate
Determine Saturation properties of refrigerant
Determine Single phase vapor heat trans. coef.
Determine Evaporation two-phase heat trans. coef.
Use general model 'EVAP' to determine heat trans. performance (see flow chart for 'EVAP')
Determine Pressure drop through heat exchanger
Is drop in $T_{R_{sat}} \leq 2°F$ through coil ?
No
$T_{R_{sat\,avg}} = \frac{(T_{sat\,in} + T_{sat\,out})}{2}$
Yes
End $\dot{m}_R$ loop


Convert back to total flow notation
End $T_{air\,db}$ in loop
End $T_{air\,wb}$ in loop
End Input Data loop
End

```
C        EVAPORATOR  SIMULATION PROGRAM
C
C       PROGRAM FOR COMPUTING EVAPORATOR PERFORMANCE, INCLUDING
C       MOISTURE REMOVAL, FOR AIR IN CROSS FLOW, PLATE-FIN TYPE
C
C       INPUT DATA FROM CARD READER (DESCRIBED FULLY BELOW)
C          NRUN,DEA,DER,DELTA,FP,XKF,AAF,GA,NT,NSECT,HCONT,
C          ST,WT,TAII,DTA,NTEMP,TSA,DXMRI,XMRI,NXMR,H4,TWBII,
C          DTWBI,NTWB,RHI,INDIC
C
C       OUTPUT
C          QTOT  -  TOTAL HEAT TRANSFER RATE (BTU/HR)
C          QLAT  -  LATENT HEAT REMOVAL RATE (BTU/HR)
C          TDB3  -  AIR DRY BULB TEMP. (F) LEAVING EVAP.
C          TWB3  -  AIR WET BULB TEMP. (F) LEAVING EVAP.
C          TRC   -  TEMP.OF REFRIGERANT VAPOR LEAVING EVAP.(F)
C
C       REMARKS
C          THIS PROGRAM CALLS SUBROUTINE SPHTC TO DETERMINE
C          SINGLE PHASE HEAT TRANSFER COEFFICIENTS
C          THIS PROGRAM CALLS SUBROUTINE SEFF TO DETERMINE
C          SURFACE EFFICIENCY OF FINNED SURFACE
C          THIS PROGRAM CALLS SUBROUTINE SATPRP TO DETERMINE
C          SATURATION THERMODYNAMIC PROPERTIES
C          THIS PROGRAM CALLS SUBROUTINE EHTC TO DETERMINE
C          THE EVAPORATION TWO-PHASE HEAT TRANSFER COEFFICIENT
C          FOR FORCED CONVECTION EVAPORATION INSIDE TUBES
C          THIS PROGRAM CALLS SUBROUTINE VAPOR TO DETERMINE
C          THERMODYNAMIC PROPERTIES OF SUPERHEATED REFRIGERANT
C          VAPOR
C          THIS PROGRAM CALLS SUBROUTINE PDROP TO DETERMINE
C          PRESSURE DROP OF REFRIGERANT FLOWING IN THE COIL
C          THIS PROGRAM CALLS FUNCTION SUBPROGRAM TSAT TO
C          DETERMINE SATURATION TEMPERATURES CORRESPONDING
C          TO GIVEN PRESSURES
C          THIS PROGRAM CALLS SUBROUTINE EVAP TO DETERMINE
C          THE OVERALL HEAT EXCHANGER PERFORMANCE, HEAT TRANSFER
C          RATES, AIR TEMPERATURES, ETC.
C       ALL TEMPERATURES ARE IN DEGREES F
C       ALL HEAT TRANSFER RATES ARE IN BTU/HR
C       ALL MASS FLOW RATES ARE IN LBM/HR
        COMMON CPA,HA,SEFFX,HRV,CPRV,XMR,XMA,X4,HTP,F,FTP,QSP,
       1QTP,TACSP,TACTP,TRC,ARHT,XMH20,TWBI,RHI,INDIC,PA,J
C
C       ----------------- INPUT DATA CONSTANTS -------------
C
C AIR PROPERTIES
C       PRA    -   PRANDTL NUMBER OF AIR
```

```
C      XMUA   -  VISCOSITY OF AIR (LBM/HR-FT)
C      RAU    -  UNIVERSAL GAS CONSTANT FOR AIR (FT-LBF/LBM-R)
C      PA     -  ATMOSPHERIC PRESSURE (PSIA)
C      CPA    -  SPECIFIC HEAT AT CONST.PRES.OF AIR (BTU/LBM-R)
       DATA PRA,XMUA,RAU/.714,.Ø43,53.34/
       PA = 14.7
       CPA = .24
C
C REFRIGERANT PROPERTY VARIATION COEFFICIENTS
C      NR     -  NUMBER OF REFRIGERANT (12,22, OR 5Ø2)
C      NREF   -  NUMBER OF REFRIGERANT (USUALLY SAME AS NR)
C      SLPEMV & XINMV    -  COEFFICIENTS FOR VISCOSITY OF VAPOR
C      SLPEKV & XINKV    -  COEFFICIENTS FOR THERMAL
C                           CONDUCTIVITY OF VAPOR
C      SLPEKL & XINKL    -  COEFFICIENTS FOR THERMAL
C                           CONDUCTIVITY OF LIQUID
C      SLPCPV & XINCPV -    COEFFICIENTS FOR SPECIFIC HEAT
C                           AT CONST.PRES. OF VAPOR
C      XM1 - XM4         -  COEFFICIENTS FOR VISCOSITY OF LIQ.
C      CP1 & CP2         -  COEFFICIENTS FOR SPECIFIC HEAT
C                           AT CONST. PRES. OF LIQUID
C      VISCOSITIES IN LBM/(HR-FT)
C      THERMAL CONDUCTIVITIES IN BTU/(HR-FT-F)
C      SPECIFIC HEATS IN BTU/(LBM-R)
       NR = 22
       DATA NREF,SLPEMV,XINMV,SLPEKV,XINKV,SLPEKL,XINKL
      1/22,.ØØØØ759,.Ø272,.ØØØØ2,.ØØ482,-.ØØØ159,.Ø6299/
       DATA SLPCPV,XINCPV/.ØØØ433,.1394/
       DATA XM1,XM2,XM3,XM4/-5.625E-Ø8,1.525E-Ø5,-2.982E-Ø3,
      1.646/
       DATA CP1,CP2/ 2.98E-Ø4,.2575/
C*****NOTE - THE ABOVE REFRIGERANT COEFFICIENTS ARE FOR
C      REFRIGERANT 22 ONLY
C
C AIR SIDE FLOW CHARACTERISTICS (SAME FOR BOTH EVAP.&COND.
C IF THEY ARE OF THE SAME TYPE)
C      C1A-C6A           -  COEFFICIENTS FOR EXPRESSING THE
C                           AIR SIDE HEAT TRANSFER COEFFICIENT
C      XLLA   -  LOWER REYNOLDS NUMBER LIMIT FOR LAMINAR
C                FLOW ON AIR SIDE
C      ULA    -  UPPER REYNOLDS NUMBER LIMIT FOR TURBULENT
C                FLOW ON AIR SIDE
       DATA C1A,C2A,C3A,C4A,C5A,C6A,XLLA,ULA
      1/.2243,-.385,.2243,-.385,.2243,-.385,1ØØØ.Ø,2ØØØ.Ø/
C
C REFRIGERANT SIDE FLOW CHARACTERISTICS (SAME FOR BOTH
C EVAP.& COND. IF THEY ARE OF SAME TYPE)
C      C1R-C6R           -  COEFFICIENTS FOR EXPRESSING THE
C                           REFRIGERANT SIDE SINGLE PHASE HEAT
```

```
C                           TRANSFER COEFFICIENTS
C     XLLR   -  LOWER REYNOLDS NUMBER LIMIT FOR LAMINAR
C               FLOW ON REFRIGERANT SIDE (SINGLE PHASE)
C     ULR    -  UPPER REYNOLDS NUMBER LIMIT FOR TURBULENT
C               FLOW ON REFRIGERANT SIDE (SINGLE PHASE)
      DATA C1R,C2R,C3R,C4R,C5R,C6R,XLLR,ULR
     1/1.164,-.7824,.000254,.49585,.02667,-.0897,2400.,3500./
C
C     ----------END OF  INPUT DATA CONSTANTS -------------
C
C     OUTER LOOP FOR MULTIPLE RUNS WHILE VARYING HEAT
C     EXCHANGER CHARACTERISTICS
C
      J = 6
      READ(8,590) NRUN
      DO 582  IN = 1,NRUN
C
C     ----------------HEAT EXCHANGER CHARACTERISTICS-------
C
C     DEA    -  OUTSIDE DIAMETER OF TUBES (FT)
C     DER    -  INSIDE DIAMETER OF TUBES (FT)
C     DELTA  -  FIN THICKNESS (FT)
C     FP     -  FIN PITCH (FINS/FT)
C     XKF    -  THERMAL CONDUCTIVITY OF FINS (BTU/HR-FT-F)
C     AAF    -  HEAT EXCHANGER FRONTAL AREA (SQ FT)
C     GA     -  AIR FLOW RATE (CU FT/MIN)
C     NT     -  NUMBER OF TUBES IN DIRECTION OF AIR FLOW
C     NSECT  -  NUMBER OF PARALLEL CIRCUITS IN HEAT EXCHANGER
C     HCONT  -  CONTACT RESISTANCE BETWEEN FINS AND TUBES
C               (BTU/HR-SQ FT-F)
C     ST     -  VERTICAL SPACING OF TUBE PASSES (FT)
C     WT     -  SPACING OF TUBE ROWS IN DIR.OF AIR FLOW (FT)
C     SIGA   -  SIGMA AIR (AIR FLOW AREA/FRONTAL AREA)
C     ATBO   -  CROSS-SECTIONAL AREA OCCUPIED BY TUBE (SQ FT)
C     PTBO   -  OUTTER PERIMETER OF TUBE (FT)
C     ALFAA  -  ALPHA AIR (AIR SIDE HEAT TRANSFER AREA/TOTAL
C               VOLUME OF HEAT EXCHANGER -1/FT)
C     ARFT   -  CROSS-SECTIONAL FLOW AREA INSIDE TUBES (SQ FT)
C     P      -  INSIDE PERIMETER OF TUBES (FT)
C     ALFAR  -  ALPHA REFRIGERANT (REFRIGERANT SIDE HEAT
C               TRANS.AREA/TOTAL VOLUME OF HEAT EXCHANGER-1/FT)
C     FAR    -  RATIO - FIN HEAT TRANS.AREA/TOTAL H.T. AREA
C     XLF    -  LENGTH OF FINS (FT)
C     ARHT   -  TOTAL REFG.SIDE HEAT TRANS. AREA/NSECT (SQ FT)
C     CAR    -  RATIO - FIN HEAT TRANS.AREA/CONTACT AREA
C               TO ACCOUNT FOR CONTACT RESISTANCE BETWEEN
C               FINS AND TUBES
      READ(8,602) DEA,DER,DELTA,FP,XKF,AAF,GA,NT,NSECT
      READ(8,611) HCONT, ST,WT
```

```
      SIGA = (ST-DEA)*(1.0-DELTA*FP)/ST
      ATBO = 3.14*DEA**2/4.0
      PTBO = 3.14*DEA
      ALFAA=(2.0*(ST*WT-ATBO)*FP +(1.0-DELTA*FP)*PTBO)/(ST*WT)
      ARFT = 3.14*DER**2/4.0
      P = 3.14*DER
      ALFAR = 3.14*DER/(ST*WT)
      FAR=2.0*FP*(ST*WT-ATBO)/(2.0*FP*(ST*WT-ATBO)+PTBO*
     1(1.0-FP*DELTA))
      XLF = ST/2.0
      ARHT = FLOAT(NT)*3.14*DER*AAF/(ST*FLOAT(NSECT))
      CAR=2.0*(ST*WT-3.14*DEA**2/4.0)/(3.14*DEA*DELTA)
      WRITE(5,510)
      WRITE(5,520)DEA,DER,DELTA,FP,XKF,AAF,GA,ARHT,NT,NSECT
      WRITE(5,521) HCONT,ST,WT
C
C     ---------END OF HEAT EXCHANGER CHARACTERISTICS----------
C
C     INITIAL VALUES FOR AIR AND REFRIGERANT FLOW CONDITIONS
C
C     TAII   -  AIR DRY BULB TEMP. ENTERING EVAPORATOR (F)
C     DTA    -  AIR DRY BULB TEMP. INCREMENT (F)
C     NTEMP  -  NUMBER OF AIR DRY BULB TEMPS. EXAMINED
C     TSA    -  REFRIGERANT SATURATION TEMP. (F)
C     DXMRI  -  REFRIGERANT FLOW RATE INCREMENT (LBM/HR)
C     XMRI   -  INITIAL TOTAL REFRIGERANT FLOW RATE (LBM/HR)
C     NXMR   -  NUMBER OF REFRIGERANT FLOW RATES EXAMINED
C     H4     -  ENTHALPY OF REFRIGERANT ENTERING EVAP.(BTU/LBM)
C     TWBII  -  AIR WET BULB TEMP. ENTERING EVAP. (F)
C     DTWBI  -  AIR WET BULB TEMP. INCREMENT (F)
C     NTWB   -  NUMBER OF WET BULB TEMPERATURES EXAMINED
C     RHI    -  RELATIVE HUMIDITY OF AIR ENTERING EVAPORATOR
C     INDIC  -  INPUT INDICATOR
C     IF 'INDIC' EQUALS 1, INPUTS ARE TDB, AND TWB
C     IF 'INDIC' EQUALS 2, INPUTS ARE TDB, AND RH
C
      READ(8,610) TAII,DTA,NTEMP,TSA ,DXMRI,XMRI,NXMR,H4
      READ(8,620) TWBII,DTWBI,NTWB,RHI,INDIC
C
C     LOOP FOR VARYING AIR WET BULB TEMP. ENTERING EVAP.
C
      TWBI = TWBII
      DO 410 IWB = 1,NTWB
      WRITE(5,540) IWB
      TWBI = TWBI + DTWBI
C
C     LOOP FOR VARYING AIR DRY BULB TEMP. ENTERING EVAP.
C
      TAI = TAII
```

```
      DO 400 I=1,NTEMP
      WRITE(5,540) I
      TAI = TAI +DTA
      WRITE(6,670) TAI,GA
C
C     PROVISION FOR RUN-TIME INTERACTIVE DATA INPUT
C
      READ(6&675,ECHO=6)
      WRITE(5,670) TAI,GA
      VA = GA*60.0/(SIGA*AAF)
      GA = VA*PA*144.0/(RAU*(TAI + 460.0))
      ACM = FLOAT(NT)*P/(ST*GA*SIGA)
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*GA*PA*144.0/(RAU*FLOAT(NSECT)*(TAI+460.0))
C
C     DETERMINE AIR SIDE HEAT TRANS.COEF.'HA'(BTU/HR-SQ FT-F)
C
      CALL SPHTC(DEA,GA,C1A,C2A,C3A,C4A,C5A,C6A,XLLA,ULA,
     1XMUA,CPA,PRA,REA,HA)
C
C     DETERMINE OVERALL SURFACE EFFICIENTCY 'SEFFX'
C
      CALL SEFF(XKF,DELTA,HA,XLF,FAR,CAR,HCONT,SEFFX)
      ICNT = 1
C
C     LOOP FOR VARYING REFRIGERANT FLOW RATE
C
      XMR = XMRI/FLOAT(NSECT)
      DXMR = DXMRI/FLOAT(NSECT)
      DO 200 K=1,NXMR
      WRITE(5,540) K
      XMR = XMR + DXMR
C
C     DETERMINE SATURATION PROPERTIES OF REFRIGERANT
C
   20 CALL SATPRP(NR,TSA,PSAT,VF,VG,HSATL,HFG,HSATV,SF,SG)
      RHOV = 1.0/VG
      RHOL = 1.0/VF
      CPRL = CP1*PSAT + CP2
      XMUL = XM1*TSA**3 + XM2*TSA**2 + XM3*TSA + XM4
      XKRL = SLPEKL*TSA  + XINKL
      XKRV = SLPEKV*TSA  + XINKV
      XMURV = SLPEMV*TSA  + XINMV
      PRRL = XMUL*CPRL/XKRL
      CPRV = SLPCPV*PSAT + XINCPV
```

```
      PRRV = XMURV*CPRV/XKRV
      GR = XMR/ARFT
C
C     DETERMINE SINGLE PHASE VAPOR HEAT TRANSFER COEF.
C     'HRV' (BTU/HR-SQ FT-F)
C
      CALL SPHTC(DER,GR,C1R,C2R,C3R,C4R,C5R,C6R,XLLR,ULR,
     1XMURV,CPRV,PRRV,RERV,HRV)
      X4 = (H4 - HSATL)/HFG
      IF(X4.LT.0.0) WRITE(J,700) X4
C
C     DETERMINE EVAPORATION TWO-PHASE HEAT TRANS. COEF.'HTP'
C     (BTU/HR-SQ FT-F)
C
      CALL EHTC(DER,GR,X4,1.0,PRRL,XKRL,XMURV,XMUL,RHOL,
     1RHOV,HTP)
C     USE SUBROUTINE EVAP TO DETERMINE EVAPORATOR HEAT
C     TRANSFER PERFORMANCE AND RETURN ALL RESULTS THROUGH COMMON
      CALL EVAP(AOM,ALFAR,ALFAA,TSA,TAI,HFG)
C
C     RETURN TO TOTAL FLOW RATE REPRESENTATION AND
C     PRINT OVERALL RESULTS
C
      QTOT=FLOAT(NSECT)*(QSP+QTP)
      QLAT = FLOAT(NSECT)*XMH20*1057.0
      XMA=XMA*FLOAT(NSECT)
      XMR=XMR*FLOAT(NSECT)
      WRITE(J,810) QTOT,XMR,XMA,QLAT
      WRITE(J,650)K,SEFFX,NT,P,ST,CPA,GA
      WRITE(J,660) SIGA,ALFAR,ALFAA
      WRITE(J,665) HA,HRV,HRV,HTP
      WRITE(J,820) ARHT,H4,X4
      DZTP = FTP*ARHT/P
      DZV = F*ARHT/P
      DZL = 0.0
      E = 5.0E-06
C
C     USE SUBROUTINE PDROP TO DETERMINE PRESSURE DROP OF
C     REFRIGERANT THROUGH EVAPORATOR 'PD' (PSI)
C
      CALL PDROP(3,DER,E,GR,XMURV,XMUL,RHOV,RHOL,RERV,
     1RERL,DZTP,1.0,X4,VG,DZV,DZL,PD)
      WRITE(J,535)
      WRITE(J,530) XMA,PD,QTOT
      WRITE(J,850) GA,REA,HA,SEFFX
      WRITE(J,860) GR,RERV,HRV
      XMA = XMA/FLOAT(NSECT)
      XMR=XMR/FLOAT(NSECT)
      IF(ICNT.NE.1) GO TO 195
```

```
C
C     CHECK DROP IN SATURATION TEMPERATURE DUE TO PRESSURE
C     DROP IN COIL - IF THE DROP IN SATURATION TEMPERATURE
C     IS GREATER THAN 2 DEGREES F - REPEAT ALL CALCULATIONS,
C     USING AN AVERAGE VALUE OF SATURATION TEMPERATURE
C
      POUT = PSAT - FP
      TSATO = TSAT(NR,POUT)
      WRITE(6,830) POUT,TSATO
      IF(ABS(TSA -TSATO).LE.2.0) GO TO 200
      TSA  = (TSA  + TSATO)/2.0
      ICNT = 2
      GO TO 20
 195  TSA  = 2.0*TSA  - TSATO
      ICNT = 1
 200  CONTINUE
 400  CONTINUE
 410  CONTINUE
 500  CONTINUE
 500  FORMAT('0',8F12.6,2I4)
 501  FORMAT('   HCONT=',F10.3,'   ST=',F10.5,'   WT=',F10.5)
 510  FORMAT('0',' DEA (FT)    DER (FT)   DELTA (FT)  FP'
     1,,' (FINS/FT)XKF(BTU/HRFT) AAF (SQFT) QA(CUFT/MIN)'
     2,,' ARHT (SQFT)     NT    NSECT')
 530  FORMAT('0',3F15.4)
 535  FORMAT('0',' XMA (LBM/HR)     PD (PSIA)    QE (BTU/HR)')
 540  FORMAT(I4)
 590  FORMAT(I10)
 600  FORMAT(7F12.6,2I10)
 610  FORMAT(2F10.4,I10,3F10.4,I10,F10.4)
 611  FORMAT(3F15.5)
 620  FORMAT(2F10.2,I10,F10.5,I10)
 650  FORMAT(' ',10X,'K=',I2,5X,'SEFFX=',F4.2,5X,'NT=',I2,
     15X,'P=',F5.3,4X,'S=',F6.3,5X,'CPA=',F5.3,5X,'GA='
     2,F10.4)
 660  FORMAT(' ',10X,'SIGA=',F8.3,5X,'ALFAR=',F8.3,5X,
     1'ALFAA=',F8.3)
 665  FORMAT(' ',10X,'HA=',F10.4,5X,'HRSPV=',F10.4,5X,
     1'HRSPV=',F10.4,5X,'HRTP=',F10.4)
 670  FORMAT('   TAI = ',F7.2,'   GA = ',F10.2)
 675  NAMELIST ' INPUT VARIABLES ARE ?', ( TAI,GA,J)
 700  FORMAT(' ******X4 IS NEGATIVE - X4=',F10.5,'*******')
 810  FORMAT('   GTOT=',F15.4,'   XMR=',F10.4,'   XMA='
     1,F10.4,'   GLAT=',F15.4)
 820  FORMAT('   ARHT=',F10.5,'   H4=',F10.5,'   X4=',F10.5)
 830  FORMAT('    POUT=',F10.5,'   TSATO=',F8.2)
 850  FORMAT('0',5X,'GA=',F10.2,' (LBM/HR-SQ-FT)',5X,'REA='
     1,F10.2,5X,'HA=',F10.2,' (BTU/HR-SQ FT-R)',5X,'SEFFX='
     2,F6.3)
```

```
860  FORMAT('0',5X,'GR=',F10.3,' (LBM/HR-SQ-FT)',5X,'RERV='
    1,F10.2,5X,'HRV=',F7.1,' (BTU/HR-SQ FT-R)')
     END
```

```
      SUBROUTINE EVAP(ACM,ALFAR,ALFAA,TSA,TAI,HFG)
C
C     PURPOSE
C        TO DETERMINE HEAT TRANSFER, MOISTURE REMOVAL,
C        AND RESULTING TEMPERATURES AND HUMIDITIES
C        IN THE EVAPORATOR, GIVEN ALL OF THE NECESSARY
C        COEFFICIENTS AND OTHER DETAILS
C
C     DESCRIPTION OF PARAMETERS
C      INPUTS
C        HEAT EXCHANGER GEOMETRY
C          ALFAR  -  ALPHA REFRIGERANT (REFRIGERANT SIDE
C                    HEAT TRANSFER AREA/TOTAL VOLUME OF HEAT
C                    EXCHANGER - 1/FT)
C          ALFAA  -  ALPHA AIR (AIR SIDE HEAT TRANSFER AREA/
C                    TOTAL VOLUME OF HEAT EXCHANGER -1/FT)
C          ARHT   -  TOTAL REFG. SIDE HEAT TRANS. AREA (SQ FT)
C          ACM    -  UNIT REFRIGERANT SIDE HEAT TRANSFER
C                    AREA/UNIT AIR FLOW RATE
C                    (SQ FT-HR/LBM DRY AIR)
C        HEAT TRANSFER COEFFICIENTS
C          HA     -  AIR SIDE HEAT TRANS.COEFF.(BTU/HR-SQ FT-F)
C          HTP    -  REF. SIDE TWO-PHASE HEAT TRANS.
C                    COEF. (BTU/HR-SQ FT-F)
C          HRV    -  REF. SIDE SINGLE PHASE VAPOR HEAT
C                    TRANS.COEF. (BTU/HR-SQ FT-F)
C
C        REFRIGERANT PROPERTIES
C          TSA    -  REFRIGERANT SATURATION TEMP. (F)
C          HFG    -  LATENT ENTHALPY OF VAPORIZATION OF
C                    THE REFRIGERANT (BTU/LBM)
C          X4     -  ENTERING QUALITY OF THE REFRIGERANT
C          XMR    -  MASS FLOW RATE OF REFRIGERANT (LBM/HR)
C          CPRV   -  SPECIFIC HEAT AT CONSTANT PRESSURE
C                    OF THE REFRIGERANT VAPOR (BTU/LBM-R)
C        AIR PROPERTIES
C          CPA    -  SPECIFIC HEAT AT CONSTANT PRESSURE OF
C                    THE AIR (BTU/LBM-R)
C          XMA    -  MASS FLOW RATE OF AIR (LBM/HR)
C          TAI    -  DRY BULB TEMP.OF AIR ENTERING EVAP. (F)
C          TWBI   -  WET BULB TEMP.OF AIR ENTERING EVAP. (F)
C          RHI    -  RELATIVE HUMIDITY OF AIR ENTERING EVAP.
C          INDIC  -  INPUT INDICATOR
C          IF 'INDIC' EQUALS 1, INPUTS ARE TAI, AND TWBI
C          IF 'INDIC' EQUALS 2, INPUTS ARE TAI, AND RHI
C          PA     -  ATMOSPHERIC PRESSURE (PSIA)
C        OTHER INPUTS
C          SEFFX  -  SURFACE EFFICIENCY OF FINNED SURFACE
C        OUTPUTS
```

```
C          F       -  SINGLE PHASE VAPOR FRACTION OF TOTAL
C                     HEAT EXCHANGER SURFACE
C          FTP     -  TWO-PHASE FRACTION OF TOTAL HEAT
C                     EXCHANGER SURFACE
C          QSP     -  HEAT TRANSFER RATE IN SINGLE PHASE
C                     VAPOR REGION (BTU/HR)
C          QTP     -  HEAT TRANSFER RATE IN TWO-PHASE
C                     REGION (BTU/HR)
C          TRO     -  TEMP.OF REFRIGERANT LEAVING EVAP. (F)
C          XMH2O   -  RATE OF MOISTURE REMOVAL FROM AIR (LBM/HR)
C          TACSP   -  EXIT AIR TEMP.FROM SINGLE PHASE REGION (F)
C          TACTP   -  EXIT AIR DRY BULB TEMP. ASSUMING
C                     NO MOISTURE REMOVAL OCCURS (F)
C          TWB3    -  WET BULB TEMP. LEAVING EVAP. IF MOISTURE
C                     REMOVAL OCCURS (F)
C          TDB3    -  EXIT AIR DRY BULB TEMP.LEAVING EVAP.,
C                     ASSUMING MOISTURE REMOVAL OCCURS (F)
C     REMARKS
C       . THIS PROGRAM CALLS SUBROUTINE XMOIST TO DETERMINE
C         THE HUMIDITY , AND DEHUMIDIFICATION BEHAVIOR
C         OF THE EVAPORATOR
C         THIS PROGRAM CALLS SUBROUTINE EXF TO DETERMINE
C         THE EFFECTIVENESS IN CROSS FLOW
C         (THIS PROGRAM USES THE EFFECTIVENESS-NTU METHOD
C         OF CALCULATING HEAT TRANSFER PERFORMANCE IF NO
C         MOISTURE REMOVAL OCCURS, AND IN THE SINGLE PHASE
C         VAPOR REGION.
      COMMON CPA,HA,SEFFX,HRV,CPRV,XMR,XMA,X4,HTP,F,FTP,QSP,
     1QTP,TACSP,TACTP,TRO,ARHT,XMH2O,TWBI,RHI,INDIC,PA,J
C
C     DETERMINE THE REPRESENTATIVE COIL CHARACTERISTIC 'COIL'
C
      COIL=HA*(ALFAA/(HTP*ALFAR)+1.0/(SEFFX*HA)-1.0/HA)/CPA
      QTP = XMR*(1.0 - X4)*HFG
C
C     DETERMINE BULK AIR DRY BULB TEMP.'TADHI' WHEN
C     DEHUMIDIFICATION OR MOISTURE REMOVAL BEGINS
C
      CALL XMOIST(TAI,TWBI,RHI,INDIC,PA,HAIRI,WSATI,WAIRI,
     1TWALLI)
      WRITE(6,715)TAI,TWBI,RHI,INDIC,HAIRI,WSATI,WAIRI,TWALLI
      TADHI=(HA*TWALLI*(1.0/(SEFFX*HA)+ALFAA/(HTP*ALFAR))
     1-TSA)/(HA*(1.0/(SEFFX*HA)+ALFAA/(HTP*ALFAR))-1.0)
C
C     DETERMINE NTU AND EFFECTIVENESS FOR SENSIBLE HEAT
C     TRANSFER IN THE TWO-PHASE REGION, ASSUMING NO MOISTURE
C     REMOVAL OCCURS
C
      XNTUTP=ACM/(CPA*(ALFAR/(SEFFX*HA*ALFAA)+1.0/HTP))
```

```
      ETP=1.0-EXP(-XNTUTP)
      XMATP=QTP/(ETP*CPA*(TAI-TSA ))
      FTP=XMATP/XMA
      IF(FTP.GT.1.0) WRITE(J,710) FTP
C
C     DETERMINE THE EXIT AIR TEMP.'TAOTP' FROM THE TWO-PHASE
C     REGION, ASSUMING NO MOISTURE REMOVAL OCCURS
C
      TAOTP=TAI-QTP/(XMATP*CPA)
      WRITE(J,705) TADHI,TAOTP
C
C     CHECK TO SEE IF MOISTURE REMOVAL ACTUALLY OCCURS
C     IF MOISTURE REMOVAL DOES NOT OCCUR, I.E. TADHI IS LESS
C     THAN TAOTP, THEN SKIP THE MOISTURE REMOVAL SECTION
C
      IF(TADHI.LT.TAOTP) TDB3=TAOTP
      IF(TADHI.LT.TAOTP) GO TO 140
C
C ---------------MOISTURE REMOVAL SECTION----------------
C
C     NOTE: IT IS ASSUMED THAT MOISTURE REMOVAL ONLY OCCURS
C     IN THE TWO-PHASE REGION OF THE COIL
C
C     IF TADHI IS GREATER THAN THE INITIAL DRY BULB
C     TEMPERATURE 'TAI', THEN MOISTURE REMOVAL BEGINS AT THE
C     LEADING EDGE OF THE COIL - HENCE SKIP TO STEP 36
C
      IF(TADHI.GT.TAI) GO TO 36
C
C     DETERMINE THE FRACTION 'FSENS' OF THE LEADING EDGE OF
C     THE COIL SURFACE WHICH IS USED ONLY FOR SENSIBLE
C     HEAT TRANSFER
C
      FSENS=XMA*CPA*(1.0/(SEFFX*HA)+ALFAA/(HTP*ALFAR))*
     1ALOG((TAI-TSA)/(TADHI-TSA))/(ARHT*ALFAA/ALFAR)
      IF((FSENS.GT.1.0).OR.(FSENS.LT.0.0))WRITE(J,720) FSENS
      GO TO 37
C
C     ITERATE TO FIND THE CORRECT WALL TEMP.'TWALLI' AT WHICH
C     MOISTURE REMOVAL BEGINS ON LEADING EDGE OF COIL
C
   36 T = TSA
      DT = 1.0
      DO 30 L = 1,30
      T = T + DT
      CALL XMOIST(T,T,RH,1,PA,HWALLI,WWALLI,WAIR,TWALLI)
      TS = TSA  + (HATRI - HWALLI)*COIL
      WRITE(J,716) T,HWALLI,WWALLI,WAIR,TWALLI,TS
      IF(ABS(T-TS).LE..1) GO TO 35
```

```
      IF(T-TS) 30,35,25
   25 T = T-DT
      DT = DT/2.0
   30 CONTINUE
      WRITE(5,717)
   35 TWALLI = T
      FSENS = 0.0
      TADHI = TAI
   37 HAIR1 = HAIRI - CPA*(TAI-TADHI)
      FMOIST = 1.0 -FSENS
      WRITE(J,716) GTP,XMA,XMR,FSENS,FMOIST,HAIR1,FTP
C
C     SPLIT MOISTURE REMOVAL REGION INTO 2 PARTS
C
C
C     ITERATE TO DETERMINE EXIT AIR ENTHALPY 'HAIR2', AND
C     WALL TEMP.'TWALL2' AT THE END OF THE FIRST MOISTURE
C     REMOVAL REGION
C
      WRITE(J,730)
      T = TSA
      DT = 1.0
      DO 50 L = 1,30
      T = T + DT
      CALL XMOIST(T,T,RH,1,PA,HWALL2,WWALL2,WAIR2,TWALL2)
      HAIR2 = HWALL2 + (T-TSA )/COIL
      HAIR2S= HAIR1-FMOIST/2.0*ARHT*ALFAA/ALFAR*HA*(TWALLI
     1-TWALL2)/(COIL*XMA*ALOG((TWALLI-TSA)/(TWALL2-TSA))*CPA)
      WRITE(J,735) T,DT,RH,HWALL2,WWALL2,WAIR2,TWALL2,
     1HAIR2,HAIR2S
      IF(ABS(HAIR2-HAIR2S).LE..05) GO TO 60
      IF(HAIR2-HAIR2S) 50,60,45
   45 T = T - DT
      DT = DT/2.0
   50 CONTINUE
      WRITE(J,740)
   60 TWALL2 = T
      WRITE(J,750)
C
C     ITERATE TO DETERMINE EXIT AIR ENTHALPY 'HAIR3', AND
C     WALL TEMP.'TWALL3' AT THE END OF THE SECOND MOISTURE
C     REMOVAL REGION
C
      T = TSA
      DT = 1.0
      DO 90 L = 1,30
      T = T + DT
      CALL XMOIST(T,T,RH,1,PA,HWALL3,WWALL3,WAIR3,TWALL3)
      HAIR3 = HWALL3 + (T - TSA )/COIL
```

```
      HAIR3S=HAIR2-FMOIST/2.0*ARHT*ALFAA/ALFAR*HA*(TWALL2
     1-TWALL3)/(COIL*XMA*ALOG((TWALL2-TSA)/(TWALL3-TSA))
     2*CPA)
      WRITE(6,735) T,DT,RH,HWALL3,WWALL3,WAIR3,TWALL3,
     1HAIR3,HAIR3S
      IF(ABS(HAIR3-HAIR3S).LE..05) GO TO 100
      IF(HAIR3 - HAIR3S) 90,100,80
   80 T = T - DT
      DT = DT/2.0
   90 CONTINUE
      WRITE(6,760)
  100 TWALL3 = T
C
C     ITERATE TO DETERMINE THE WET BULB TEMP.'TWB3' LEAVING
C     THE EVAPORATOR
C
      T = TSA
      DT = 1.0
      DO 105 L = 1,30
      T = T + DT
      CALL XMOIST(T,T,RH,1,PA,HAIR3S,WSAT3,WAIR,TWALL)
      WRITE(6,763) T,DT,HAIR3,HAIR3S,WSAT3,WAIR
      IF(ABS(HAIR3S-HAIR3).LE..05) GO TO 106
      IF(HAIR3S-HAIR3) 105,106,104
  104 T = T - DT
      DT = DT/2.0
  105 CONTINUE
      WRITE(6,764)
  106 TWB3 = T
C
C     DETERMINE THE FRACTION 'FTP' OF THE HEAT EXCHANGER
C     WHICH IS IN TWO-PHASE FLOW
C
      FTP=GTP/(XMA*(HAIRI-HAIR3))
      WRITE(6,770) TWB3,FTP
      IF((1.0-FTP).LT.0.0) GO TO 190
C
C     ITERATE TO DETERMINE THE AMOUNT OF WATER REMOVED FROM
C     THE AIR 'XMH20' , AND THE FINAL DRY BULB TEMP. 'TDB3'
C     LEAVING THE EVAPORATOR
C
      T = TWB3
      DT = .50
      DO 120 L = 1,50
      T = T + DT
      CALL XMOIST(T,TWB3,RH3,1,PA,HAIR,WSAT,WAIR3,TWALL)
      XMH20 = (WAIRI-WAIR3)*FTP*XMA
      HAIR3S=.24*(T-32.0) +WAIR3*(1060.9 + .444*T)
      WRITE(6,775)T,DT,TWB3,RH3,HAIR,HAIR3S,HAIR3,WAIR3,XMH20,WSAT,T
```

```
      IF(ABS(HAIR3S-HAIR3).LE..05) GO TO 130
      IF(HAIR3S-HAIR3) 110,130,120
 110  T = T - DT
      DT = DT/2.0
 120  CONTINUE
      WRITE(6,780)
 130  TDB3 = T
C
C ------------- END OF MOISTURE REMOVAL SECTION -----------
C
C     DETERMINE FRACTION OF HEAT EXCHANGER 'F' USED FOR
C     SINGLE PHASE VAPOR (SUPERHEATING) REGION
C
 140  F = 1.0 - FTP
      WRITE(6,790) TDB3,F
C
C     IF 'F' IS LESS THAN ZERO, INCOMPLETE EVAPORATION OCCURS
C     HENCE, PRINT AN ERROR MESSAGE
C
      IF(F) 190,145,150
 145  F = .000001
 150  XMASP = F*XMA
C
C     USE THE EFFECTIVENESS-NTU METHOD TO DETERMINE HEAT
C     TRANSFER IN THE SINGLE PHASE VAPOR (SUPERHEATING)
C     REGION
C
      CA = XMASP*CPA
      CR = XMR*CPRV
      RTOT = (ALFAR/(SEFFX*HA*ALFAA)+1.0/HRV)/(F*ARHT)
      CALL EXF(RTOT,CA,CR,CMIN,EXFR)
C
C     CALCULATE AND PRINT THE HEAT TRANSFER RATES
C     AND TEMPERATURES OF INTEREST
C
      TRO = TSA +EXFR*CMIN*(TAI-TSA )/CR
      QSP = CR*(TRO-TSA )
      WRITE(6,630) FTP,F,XMASP
      WRITE(6,635) CA,CR,EXFR,TRO,QSP
      TAOSP = TAI-QSP/CA
      WRITE(6,640) QSP,QTP
      WRITE(6,795)
      WRITE(6,800) TAI,TWBI,TDB3,TWB3,TSA ,TRO,TAOTP
      RETURN
 190  WRITE(6,840) FTP
      RETURN
 630  FORMAT(' ',10X,'FTP=',F4.2,5X,'F=',F4.2,5X,'XMASP='
     1,F10.4)
 635  FORMAT(' ',5X,'CA=',F10.4,5X,'CR=',F10.4,5X,'EXFR='
```

```
     1,F4.2,5X,'TRC=',F10.4,5X,'GSP=',F10.4)
 640  FORMAT(' ',10X,'GSP=',F10.4,5X,'GTP=',F10.4)
 705  FORMAT('     TACHT=',F10.2,'       TAOTP=',F10.2)
 710  FORMAT(' ****FTP IS LARGER THAN 1 FTP=',F10.5,'****')
 715  FORMAT(3F10.4,I10,4F12.5)
 716  FORMAT(7F15.4)
 717  FORMAT('      **********NO SENSIBLE HEAT REMOVAL'
     1,,' ITERATION DOES NOT CONVERGE**********')
 720  FORMAT(' *****FSENS IS IN ERROR FSENS=',F10.5,'*****')
 730  FORMAT('       T          DT           RH          HWALL2    '
     1,,'WWALL2     WAIR2      TWALL2     HAIR2       HAIR2S')
 735  FORMAT(9F10.5)
 740  FORMAT(' ***********FIRST MOISTURE REMOVAL REGION ITER'
     1,,'ATION DOES NOT CONVERGE************')
 750  FORMAT('       T           DT            RH       HWALL3     '
     1,,'WWALL3      WAIR3      TWALL3     HAIR3       HAIR3S')
 760  FORMAT(' *********SECOND MOISTURE REMOVAL REGION ITERA'
     1,,'TION DOES NOT CONVERGE*********')
 763  FORMAT(6F15.5)
 764  FORMAT(' **********ITERATION ON TWB3 DOES NOT '
     1,,'CONVERGE *********')
 770  FORMAT('      TWB3=',F7.2,'     FTP=',F10.5)
 775  FORMAT(4F8.2,7G15.8)
 780  FORMAT(' **********EXIT DRY BULB TEMP DOES NOT '
     1,,'CONVERGE ********')
 790  FORMAT('        TDB3=',F8.2,'     F=',F10.5)
 795  FORMAT('    TAI      TWBI      TDB3      TWB3      TSAT       '
     1,,'TRC        TACTP')
 800  FORMAT(7F8.2)
 840  FORMAT(' *****INCOMPLETE EVAPORATION FTP=',F10.5,'**')
      END
```

```
      SUBROUTINE XMOIST(TDB,TWB,RH,INDIC,PATM,HAIR,WSAT,
     1WAIR,TWALL)
C
C     PURPOSE
C        TO DETERMINE THE ENTHALPY, SATURATION MOISTURE
C        CONTENT, AND ACTUAL MOISTURE CONTENT OF MOIST AIR,
C        AND ALSO, THE NECESSARY WALL TEMPERATURE TO INDUCE
C        MOISTURE REMOVAL, GIVEN DRY BULB TEMPERATURE AND
C        EITHER WET BULB TEMPERATURE OR RELATIVE HUMIDITY
C        (NOTE - THIS PROGRAM ESSENTIALLY REPRODUCES
C        PSYCHROMETRIC CHART DATA)
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        TDB    -   DRY BULB TEMPERATURE (F)
C        TWB    -   WET BULB TEMPERATURE (F)
C        RH     -   RELATIVE HUMIDITY
C        INDIC-     INPUT INDICATOR
C        IF 'INDIC' = 1, INPUTS ARE TDB, AND TWB
C        IF 'INDIC' = 2, INPUTS ARE TDB, AND RH
C        PATM -     ATMOSPHERIC PRESSURE (PSIA)
C      OUTPUTS
C        HAIR -     ENTHALPY OF MOIST AIR (BTU/LBM DRY AIR)
C        WSAT -     SATURATION HUMIDITY (LBM WATER/LBM DRY AIR)
C                   CORRESPONDING TO THE EXISTING WET BULB TEMP.
C        WAIR -     ACTUAL HUMIDITY (LBM WATER/LBM DRY AIR)
C                   CORRESPONDING TO THE GIVEN DRY BULB TEMP.,
C                   PRES., AND REL.HUMIDITY OR WET BULB TEMP.
C        TWALL-     SATURATION OR DEW POINT TEMPERATURE (F)
C                   CORRESPONDING TO THE GIVEN TDB,PATM, AND
C                   TWB, OR RH
      K = 0
      I = 1
      IF(INDIC.NE.1) GO TO 30
      T = TWB
C
C     DETERMINING SATURATION PARTIAL PRESSURE 'PS' (PSIA)
C     OF WATER VAPOR AT THE GIVEN TEMPERATURE
   10 IF(T.LE.0.0) PS=.00277*T + .0185
      IF(T.GT.0.0) PS=.00124*T + .0185
      IF(T.GT.10.0) PS=.00196*T + .0113
      IF(T.GT.20.0) PS=.00317*T - .0129
      IF(T.GT.32.0) PS= .004145*T - .0441
      IF(T.GT.40.0) PS = .005641*T - .10394
      IF(T.GT.50.0) PS = .007019*T - .21284
      IF(T.GT.60.0) PS = .01068*T - .3845
      IF(T.GT.70.0) PS = .01438*T - .6435
      IF(T.GT.80.0) PS = .01913*T - 1.0235
      IF(T.GT.90.0) PS = .0251*T - 1.5608
```

```
      W     = 1.004*18.01*PS/(28.967*(PATM-PS))
      IF(K.NE.0) GO TO 50
      IF(INDIC.EQ.2) GO TO 40
      IF(I.NE.1) GO TO 20
      I = 2
      WSAT = W
      WAIR = WSAT -.000236*(TDB -T)
      HAIR = .24*(TWB-32.0) + WSAT*(1060.9 + .444*TWB)
      P=PATM/(1.004*18.01/(28.967*WAIR)+1.0)
      T = TDB
      GO TO 10
C
C     FINDING THE CORRESPONDING RELATIVE HUMIDITY,
C     GIVEN THE WET BULB TEMPERATURE
   20 RH = P/PS
      GO TO 92
   30 T = TDB
      GO TO 10
   40 P = RH*PS
      WAIR = RH*W*(PATM-PS)/(PATM-P)
C
C     FINDING THE CORRESPONDING WET BULB TEMPERATURE,
C     GIVEN THE RELATIVE HUMIDITY
      DT = -10.0
      DO 70 K=1,30
      T = T+DT
      GO TO 10
   50 WS = W     - .000236*(TDB-T)
      IF(ABS(WS-WAIR).LE..00005) GO TO 80
      IF(WS-WAIR) 60,80,70
   60 T = T-DT
      DT = DT/2.0
   70 CONTINUE
      WRITE(5,100)
   80 TWB = T
      WSAT = W
      HAIR = .24*(TWB-32.0) + WSAT*(1060.9 + .444*TWB)
C
C     DETERMINING THE SATURATION OR DEW POINT TEMP. 'TWALL'
C     CORRESPONDING TO THE GIVEN PRESSURE, DRY BULB
C     TEMPERATURE, AND RELATIVE HUMIDITY OR WET BULB TEMP.
   90 IF(P.LE..0185) TWALL=(P-.0185)/.00077
      IF(P.GT..0185) TWALL = (P-.0185)/.00124
      IF(P.GT..0309) TWALL=(P-.0113)/.00196
      IF(P.GT..0505) TWALL=(P+.0129)/.00317
      IF(P.GT..0885) TWALL=(P+.0441)/.004145
      IF(P.GT..12170) TWALL=(P+.10394)/.005641
      IF(P.GT..17611) TWALL=(P+.21284)/.007819
      IF(P.GT..2563) TWALL=(P+.3845)/.01068
      IF(P.GT..3631) TWALL = (P+.6435)/.01438
```

```
      IF(P.GT..5069) TWALL = (P+1.0235)/.01913
      IF(P.GT..6982) TWALL = (P+1.5608)/.0251
100   FORMAT(' ***** ITERATION IN XMOIST DOES NOT CONVERGE')
      RETURN
      END
```

APPENDIX N

CAPACITY CONTROLLED 50 DQ 016 STUDIES

Condenser 6330 CFM , Entering Air $70^{o}F$

Evaporator 10000 CFM , Entering Air 85% R.H.

P_{IF} = 9425 Btu/hr

P_{OF} = 5300 Btu/hr

OUTDOOR AIR TEMP (^{o}F db)	Q HP* (1000's of Btu/hr) CONV	$53^{o}BP$	$37^{o}BP$	$10^{o}BP$	COP No Fans CONV	$53^{o}BP$	$37^{o}BP$	$10^{o}BP$
62.5	218	195	144	72	3.62	3.95	4.50	4.93
57.5	205	196	145	74	3.60	3.73	4.25	4.68
52.5	192		146	75	3.56		4.05	4.43
47.5	180		148	77	3.53		3.85	4.25
42.5	168		152	78	3.49		3.67	4.05
37.5	155		153	81	3.45		3.48	3.87
32.5	144			82	3.41			3.67
27.5	132			84	3.36			3.52
22.5	119			85	3.30			3.35
17.5	108			86	3.20			3.22
12.5	95.6			88	3.10			3.10
7.5	83.1				2.90			
2.5	71.1				2.65			
-2.5	0				1.0			
↓	↓				↓			

OUTDOOR AIR TEMP (^{o}F)	COP OUTDOOR FANS CONV	$53^{o}BP$	$37^{o}BP$	$10^{o}BP$	COP BOTH FANS CONV	$53^{o}BP$	$37^{o}BP$	$10^{o}BP$
62.5	3.28	3.56	3.85	3.67	3.01	3.15	3.30	2.82
57.5	3.25	3.37	3.68	3.55	2.97	3.05	3.18	2.77
52.5	3.20		3.54	3.42	2.93		3.06	2.72
47.5	3.18		3.38	3.30	2.87		2.95	2.67
42.5	3.14		3.25	3.20	2.80		2.85	2.62
37.5	3.08		3.11	3.08	2.77		2.77	2.55
32.5	3.03			2.98	2.67			2.48
27.5	2.95			2.87	2.60			2.43
22.5	2.88			2.80	2.52			2.37
17.5	2.78			2.70	2.42			2.33
12.5	2.63			2.60	2.30			2.26
7.5	2.43				2.15			
2.5	2.16				1.93			
-2.5	1.0				1.0			
↓	↓				↓			

* Q HP – Does Not Contain Indoor Fan Motor Heat

Air Cond. = 4500 CFM $T_{Air\ Cond.}$ = 70°F P_{IF} = 6919 Btu/hr

Air Evap. = 7500 CFM   R.H. = 85% P_{OF} = 2650 Btu/hr

OUTDOOR AIR TEMP (°F)	Q HP* (1000's Of Btu/hr) CONV.	30°BP	20°BP	10°BP	COP No Fans CONV	30°BP	20°BP	10°BP
62.5	200	120	100	72	3.30	4.30	4.53	4.67
57.5	191	122	102	74	3.30	4.12	4.32	4.45
52.5	179	124	103	75	3.29	3.93	4.12	4.25
47.5	169	126	104	76	3.26	3.75	3.92	4.05
42.5	158	127	105	78	3.23	3.57	3.75	3.85
37.5	147	128	106	80	3.21	3.40	3.56	3.67
32.5	135	130	107	82	3.18	3.24	3.40	3.50
27.5	125		108	83	3.13		3.24	3.36
22.5	114		108.5	84	3.10		3.12	3.25
17.5	103			86	3.05			3.15
12.5	92			88	2.97			3.02
7.5	81				2.85			
2.5	71				2.65			
-2.5	0				1.0			
↓					↓			

OUTDOOR AIR TEMP (°F)	COP OUTDOOR FAN CONV	30°BP	20°BP	10°BP	COP BOTH FANS CONV	30°BP	20°BP	10°BP
62.5	3.24	3.96	4.08	4.00	2.98	3.40	3.40	3.15
57.5	3.20	3.80	3.90	3.83	2.95	3.28	3.28	3.09
52.5	3.17	3.62	3.73	3.68	2.93	3.17	3.17	3.01
47.5	3.15	3.45	3.57	3.55	2.89	3.05	3.05	2.95
42.5	3.11	3.30	3.42	3.41	2.86	2.95	2.95	2.87
37.5	3.06	3.15	3.27	3.27	2.80	2.85	2.85	2.80
32.5	3.00	3.04	3.12	3.16	2.75	2.77	2.77	2.72
27.5	2.95		3.00	3.05	2.69		2.69	2.65
22.5	2.87		2.87	2.95	2.65		2.65	2.63
17.5	2.83			2.85	2.55			2.50
12.5	2.75			2.75	2.45			2.44
7.5	2.65				2.30			
2.5	2.53				2.15			
-2.5	1.0				1.0			
↓	↓				↓			

*Q HP - Does Not Contain Indoor Fan Motor Heat

Air Cond. = 3165 $T_{Air\ Cond} = 70^oF$ $P_{IF} = 3535$

Air Evap = 5200 R.H. = 85% $P_{OF} = 1818$

OUTDOOR AIR TEMP (^{o}F)	Q HP* (1000's Btu/hr) CONV	23^oBP	14^oBP	COP No Fans CONV	23^oBP	14^oBP
62.5	184	101	82	2.89	4.06	4.35
57.5	176	102	83	2.89	3.88	4.17
52.5	167	103	84	2.89	3.70	3.98
47.5	158	104	85	2.89	3.56	3.80
42.5	148	105	87	2.89	3.40	3.62
37.5	139	106	88	2.89	3.25	3.45
32.5	128	108	89	2.89	3.12	3.30
27.5	120	110	90	2.89	2.97	3.15
22.5	110		91	2.89		3.03
17.5	101		92	2.88		2.90
12.5	90			2.82		
7.5	83			2.74		
2.5	72			2.64		
-2.5	0			1.0		
↓	↓			↓		

OUTDOOR AIR TEMP (^{o}F)	COP OUTDOOR FAN CONV	23^oBP	14^oBP	COP BOTH FANS CONV	23^oBP	14^oBP
62.5	2.80	3.87	3.95	2.75	3.47	3.50
57.5	2.80	3.68	3.78	2.75	3.35	3.38
52.5	2.80	3.52	3.63	2.75	3.21	3.27
47.5	2.80	3.37	3.47	2.75	3.10	3.15
42.5	2.80	3.20	3.35	2.75	2.97	3.05
37.5	2.80	3.07	3.20	2.73	2.85	2.92
32.5	2.80	2.92	3.07	2.70	2.75	2.82
27.5	2.80	2.83	2.95	2.67	2.70	2.75
22.5	2.77		2.85	2.63		2.65
17.5	2.72		2.77	2.58		2.60
12.5	2.65			2.52		
7.5	2.56			2.45		
2.5	2.38			2.33		
-2.5	1.0			1.0		
↓	↓			↓		

* Q HP - Does Not Contain Indoor Fan Motor Heat

APPENDIX O

HEAT PUMP PERFORMANCE DATA FOR LOAD LINE "D" STUDIES*

SAN FRANCISCO

	46^oBP 50 DQ 004 - CONV AIR COND =1200 CFM AIR EVAP = 1750 CFM P_{IF} = 2038 BTU/HR P_{OF} = 1024 BTU/HR			39^oBP 50 DQ 006-CAP. CONT. AIR COND = 1050 CFM AIR EVAP = 1850 CFM P_{IF} = 1544 BTU/HR P_{OF} = 1082 BTU/HR			32^oBP 50 DQ 008 = CAP. CONT. AIR COND = 1610 CFM AIR EVAP = 2600 CFM P_{IF} = 2001 BTU/HR P_{OF} = 1126 BTU/HR		
AMBIENT TEMP (^oF)	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}
62.5	42.7	3.37	2.84	44.2	3.29	2.85	59.0	3.53	3.07
57.5	40.7	3.32	2.79	44.5	3.15	2.75	59.3	3.38	2.97
52.5	38.5	3.26	2.73	44.8	3.01	2.65	59.5	3.22	2.85
47.5	36.0	3.22	2.67	45.2	2.89	2.56	59.8	3.10	2.76
42.5	33.3	3.13	2.58	45.5	2.77	2.47	60.2	2.96	2.65
37.5	30.5	3.03	2.48	45.0	2.72	2.43	60.5	2.82	2.54
32.5	28.0	2.94	2.39	41.5	2.72	2.41	60.2	2.72	2.46
27.5	25.5	2.82		38.0	2.72		55.2	2.72	
22.5	23.2	2.70		34.2	2.72		50.0	2.72	
17.5	21.0	2.58		30.7	2.71		45.0	2.71	
12.5	18.7	2.45		27.0	2.65		39.5	2.65	
7.5	16.7	2.29		23.5	2.58		34.5	2.58	
2.5	14.7	2.14		21.0	2.48		29.0	2.48	
-2.5	13.5	1.93		18.4	2.25		24.0	2.25	
-7.5	12.0	1.60		15.5	1.90		18.5	1.90	
↓	0	1.0	1.0	0	1.0	1.0	0	1.0	1.0

*All COP_{NF} values shown have been computed as outlined in Section 3.4, and are based in part, or in total on computer simulations of the 50 DQ 016. All COP_{NF} values derived from the 016 simulations were reduced by a uniform 6% before use in computing the above values, to correct for the slight simulation inaccuracy noted in Chapter 2.

HEAT PUMP PERFORMANCE DATA FOR LOAD LINE "D" STUDIES*

CHARLESTON

	46^{o}BP 50 DQ 004 - CONV. AIR COND = 1200 CFM AIR EVAP = 1750 CFM P_{IF} = 1775 BTU/HR P_{OF} = 1024 BTU/HR			39^{o}BP 50 DQ 006 - CAP. CONT. AIR COND = 1200 CFM AIR EVAP = 1850 CFM P_{IF} = 1775 BTU/HR P_{OF} = 1082 BTU/HR			32^{o}BP 50 DQ 008 - CAP. CONT. AIR COND = 1610 CFM AIR EVAP = 2600 BTU/HR P_{IF} = 1158 BTU/HR P_{OF} = 1126 BTU/HR			21^{o}BP 50 DQ FICT - CAP. CONT. AIR COND = 2250 CFM AIR EVAP = 3750 CFM P_{IF} = 2830 BTU/HR P_{OF} = 1297 BTU/HR		
AMBIENT TEMP (^{o}F)	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}
62.5	42.7	3.37	2.87	45.8	3.46	2.96	59.0	3.53	3.17	75.5	3.82	3.28
57.5	40.7	3.32	2.82	46.0	3.31	2.85	59.3	3.38	3.05	76.2	3.65	3.16
52.5	38.5	3.26	2.76	46.2	3.16	2.74	59.5	3.22	2.92	76.7	3.48	3.04
47.5	36.0	3.22	2.70	46.2	3.02	2.64	59.8	3.10	2.83	77.5	3.30	2.91
42.5	33.3	3.13	2.61	46.8	2.89	2.55	60.2	2.96	2.71	78.1	3.18	2.82
37.5	30.5	3.03	2.51	46.2	2.81	2.49	60.5	2.82	2.60	78.8	3.06	2.73
32.5	28.0	2.94	2.42	42.0	2.81	2.46	60.2	2.72	2.51	79.5	2.93	2.63
27.5	25.5	2.82	2.30	39.0	2.80	2.43	55.2	2.72	2.50	80.0	2.79	2.53
22.5	23.2	2.70	2.19	35.0	2.77	2.37	50.0	2.72	2.48	80.7	2.72	2.47
17.5	21.0	2.58	2.08	31.5	2.74	2.32	45.0	2.71	2.44	75.5	2.71	2.45
12.5	18.7	2.45		27.7	2.69		39.5	2.65		67.5	2.65	
7.5	16.7	2.29		24.2	2.65		34.5	2.58		60.0	2.58	
2.5	14.7	2.14		21.5	2.49		29.0	2.48		51.5	2.48	
-2.5	13.5	1.93		18.5	2.25		24.0	2.25		44.0	2.25	
-7.5	12.0	1.60		15.5	1.90		18.5	1.90		35.5	1.90	
↓	0	1.0	1.0	0	1.0	1.0	0	1.0	1.0	0	1.0	1.0

*All COP_{NF} values shown have been computed as outlined in Section 3.4, and are based in part, or in total on computer simulations of the 50 DQ 016. All COP_{NF} values derived from the 016 simulations were reduced by a uniform 6% before use in computing the above values, to correct for the slight simulation inaccuracy noted in Chapter 2.

HEAT PUMP PERFORMANCES DATA FOR LOAD LINE "D" STUDIES*

NEW YORK & BOSTON

AMBIENT TEMP	37^{o}BP 50 DQ 006 - CONV AIR COND = 2100 CFM AIR EVAP = 3700 CFM P_{IF} = 2724 BTU/HR P_{OF} = 1297 BTU/HR			32^{o}BP 50 DQ 008 - CAP. CONT AIR COND = 1700 CFM AIR EVAP = 2600 CFM P_{IF} = 1158 BTU/HR P_{OF} = 1126 BTU/HR			21^{o}BP 50 DQ FICT-CAP.CONT. AIR COND = 2250 CFM AIR EVAP = 3750 CFM P_{IF} = 2001 BTU/HR P_{OF} = 1297 BTU/HR			14^{o}BP 50 DQ 016 - CAP.CONT. AIR COND = 3165 CFM AIR EVAP = 5200 CFM P_{IF} = 6049 BTU/HR P_{OF} = 1911 BTU/HR		
	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}
62.5	71.0	3.40	2.96	59.0	3.53	3.17	75.5	3.82	3.36	82.0	4.09	3.14
57.5	67.0	3.38	2.92	59.3	3.38	3.05	76.2	3.65	3.23	83.0	3.92	3.06
52.5	63.5	3.35	2.88	59.5	3.22	2.92	76.7	3.48	3.11	84.0	3.74	2.96
47.5	60.0	3.32	2.84	59.8	3.10	2.83	77.5	3.30	2.97	85.0	3.57	2.87
42.5	55.0	3.28	2.78	60.2	2.96	2.71	78.1	3.18	2.88	87.0	3.40	2.77
37.5	50.5	3.24	2.71	60.5	2.82	2.60	78.8	3.06	2.78	88.0	3.24	2.68
32.5	46.0	3.21	2.66	60.2	2.72	2.51	79.5	2.93	2.68	89.0	3.10	2.59
27.5	41.0	3.16	2.57	55.2	2.72	2.50	80.0	2.79	2.56	90.0	2.96	2.50
22.5	36.0	3.10	2.48	50.0	2.72	2.48	80.7	2.72	2.51	91.0	2.85	2.43
17.5	32.5	3.01	2.38	45.0	2.71	2.44	75.5	2.71	2.49	92.0	2.73	2.35
12.5	28.5	2.91	2.26	39.5	2.65	2.37	67.5	2.65	2.42	90.0	2.65	2.29
7.5	25.0	2.73	2.10	34.5	2.58	2.28	60.0	2.58	2.33	83.0	2.58	2.22
2.5	22.0	2.49	1.92	29.0	2.48	2.16	51.5	2.48	2.22	72.0	2.48	2.11
-2.5	19.0	2.25	1.74	24.0	2.25	1.94	44.0	2.25	2.01	0	1.0	1.0
-7.5	16.0	1.90	1.50	18.5	1.90	1.64	35.5	1.90	1.71	0	1.0	1.0
↓	0	1.0	1.0	0	1.0	1.0	0	1.0	1.0	0	1.0	1.0

*All COP_{NF} values shown have been computed as outlined in Section 3.4, and are based in part, or in total on computer simulations of the 50 DQ 016. All COP_{NF} values derived from the 016 simulations were reduced by a uniform 6% before use in computing the above values, to correct for the slight simulation inaccuracy noted in Chapter 2.

HEAT PUMP PERFORMANCE DATA FOR LOAD LINE "D" STUDIES*

OMAHA

AMBIENT TEMP (^oF)	37^oBP 50 DQ 006 - CONV, AIR COND = 2100 CFM, AIR EVAP = 3700 CFM, P_{IF} = 2512 BTU/HR, P_{OF} = 1297 BTU/HR: Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	32^oBP 50 DQ 008 - CAP.CONT., AIR COND = 1900 CFM, AIR EVAP = 2600 CFM, P_{IF} = 1158 BTU/HR, P_{OF} = 1126 BTU/HR: Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	21^oBP 50 DQ FICT - CAP.CONT., AIR COND = 2250 CFM, AIR EVAP = 3750 CFM, P_{IF} = 1737 BTU/HR, P_{OF} = 1297 BTU/HR: Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	14^oBP 50 DQ 016 - CAP.CONT., AIR COND = 3175 CFM, AIR EVAP = 5200 CFM, P_{IF} = 4191 BTU/HR, P_{OF} = 1911 BTU/HR: Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}
62.5	71.0	3.40	2.98	60.3	3.74	3.34	75.5	3.82	3.39	82.0	4.09	3.30
57.5	67.0	3.38	2.94	60.5	3.59	3.22	76.2	3.65	3.26	83.0	3.92	3.20
52.5	63.5	3.35	2.90	60.8	3.42	3.09	76.7	3.48	3.13	84.0	3.74	3.09
47.5	60.0	3.32	2.86	61.2	3.28	2.98	77.5	3.30	2.99	85.0	3.57	2.98
42.5	55.0	3.28	2.79	61.4	3.13	2.86	78.1	3.18	2.89	87.0	3.40	2.88
37.5	50.5	3.24	2.73	61.7	2.98	2.73	78.8	3.06	2.80	88.0	3.24	2.77
32.5	46.0	3.21	2.67	62.0	2.86	2.64	79.5	2.93	2.69	89.0	3.10	2.68
27.5	41.0	3.16	2.59	56.7	2.81	2.58	80.0	2.79	2.58	90.0	2.96	2.58
22.5	36.0	3.10	2.50	51.3	2.80	2.55	80.7	2.72	2.52	91.0	2.85	2.50
17.5	32.5	3.01	2.40	45.7	2.77	2.49	75.5	2.71	2.50	92.0	2.73	2.42
12.5	28.5	2.91	2.28	40.0	2.69	2.40	67.5	2.65	2.43	90.0	2.65	2.35
7.5	25.0	2.73	2.12	34.8	2.62	2.31	60.0	2.58	2.35	83.0	2.58	2.28
2.5	22.0	2.49	1.94	29.2	2.48	2.16	51.5	2.48	2.24	72.0	2.48	2.17
-2.5	19.0	2.25	1.76	24.0	2.25	1.94	44.0	2.25	2.02	0	1.0	1.0
-7.5	16.0	1.90	1.51	18.5	1.90	1.64	35.5	1.90	1.71	0	1.0	1.0
↓	0	1.0	1.0	0	1.0	1.0	0	1.0	1.0	0	1.0	1.0

*All COP_{NF} values shown have been computed as outlined in Section 3.4, and are based in part, or in total on computer simulations of the 50 DQ 016. All COP_{NF} values derived from the 016 simulations were reduced by a uniform 6% before use in computing the above values, to correct for the slight simulation inaccuracy noted in Chapter 2.

HEAT PUMP PERFORMANCE DATA FOR LOAD LINE "D" STUDIES*

MINNEAPOLIS

	37°BP			32°BP			21°BP			14°BP		
	50 DQ 006 - CONV AIR COND = 2200 CFM AIR EVAP = 3700 CFM P_{IF} = 2582 BTU/HR P_{OF} = 1297 BTU/HR			50 DQ 008 - CAP. CONT. AIR COND = 2200 CFM AIR EVAP = 2600 CFM P_{IF} = 1544 BTU/HR P_{OF} = 1126 BTU/HR			50 DQ FICT - CAP. CONT. AIR COND = 2250 CFM AIR EVAP = 3750 CFM P_{IF} = 1544 BTU/HR P_{OF} = 1297 BTU/HR			50 DQ 016 - CAP. CONT. AIR COND = 3175 CFM AIR EVAP = 5200 CFM P_{IF} = 2653 BTU/HR P_{OF} = 1911 BTU/HR		
AMBIENT TEMP (°F)	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}	Q HP 1000's OF BTU/HR	COP_{NF}	COP_{BF}
62.5	71.0	3.40	2.97	61.0	3.89	3.41	75.5	3.82	3.41	82.0	4.09	3.44
57.5	67.0	3.38	2.94	61.2	3.73	3.29	76.2	3.65	3.28	83.0	3.92	3.33
52.5	63.5	3.35	2.89	61.5	3.56	3.16	76.7	3.48	3.14	84.0	3.74	3.21
47.5	60.0	3.32	2.85	61.7	3.35	3.00	77.5	3.30	3.00	85.0	3.57	3.09
42.5	55.0	3.28	2.79	62.0	3.24	2.91	78.1	3.18	2.91	87.0	3.40	2.97
37.5	50.5	3.24	2.73	62.2	3.08	2.79	78.8	3.06	2.81	88.0	3.24	2.86
32.5	46.0	3.21	2.67	62.5	2.95	2.68	79.5	2.93	2.70	89.0	3.10	2.75
27.5	41.0	3.16	2.59	57.5	2.88	2.61	80.0	2.79	2.59	90.0	2.96	2.65
22.5	36.0	3.10	2.49	52.0	2.86	2.57	80.7	2.72	2.53	91.0	2.85	2.57
17.5	32.5	3.01	2.39	46.5	2.82	2.51	75.5	2.71	2.51	92.0	2.73	2.47
12.5	28.5	2.91	2.27	41.0	2.75	2.42	67.5	2.65	2.44	90.0	2.65	2.40
7.5	25.0	2.73	2.12	35.0	2.65	2.30	60.0	2.58	2.36	83.0	2.58	2.33
2.5	22.0	2.49	1.93	29.3	2.49	2.14	51.5	2.48	2.25	72.0	2.48	2.22
-2.5	19.0	2.25	1.75	24.0	2.25	1.92	44.0	2.25	2.03	0	1.0	1.0
-7.5	16.0	1.90	1.51	18.5	1.90	1.62	35.5	1.90	1.72	0	1.0	1.0
↓	0	1.0	1.0	0	1.0	1.0	0	1.0	1.0	0	1.0	1.0

*All COP_{NF} values shown have been computed as outlined in Section 3.4, and are based in part, or in total on computer simulations of the 50 DQ 016. All COP_{NF} values derived from the 016 simulations were reduced by a uniform 6% before use in computing the above values, to correct for the slight simulation inaccuracy noted in Chapter 2.

COMPARISON OF SEASONAL ENERGY SAVINGS (PERCENT PER YEAR) OVER CONVENTIONALLY SIZED HEAT PUMPS
CAPACITY CONTROLLED HEAT PUMPS VS OVERSIZED CONVENTIONAL HEAT PUMPS
(DUCTS SIZED USING CONVENTIONAL PRACTICE FOR AIR FLOWS OF OVERSIZED UNITS)*

LOCATION

HEAT PUMP		SAN FRANCISCO 1 speed fans	SAN FRANCISCO 2 speed fans+	CHARLESTON 1 speed fans	CHARLESTON 2 speed fans+	NEW YORK 1 speed fans	NEW YORK 2 speed fans+	BOSTON 1 speed fans	BOSTON 2 speed fans+	OMAHA 1 speed fans	OMAHA 2 speed fans+	MINNEAPOLIS 1 speed fans	MINNEAPOLIS 2 speed fans+
46°BP	Conv.	0		0		-		-		-		-	
	Cap.Cont.	.9		.4		-		-		-		-	
37°BP	Conv.	10.7		21.3		0		0		0		0	
	Cap.Cont.	6.1	6.2	15.4	18.7	-8.0	-1.4	-7.4	- .6	-5.7	1.0	-4.4	1.5
32°BP	Conv.	11.0		27.0		12.4		13.4		13.8		12.5	
	Cap.Cont.	12.4	12.4	25.4	27.2	7.9	13.3	8.6	15.0	9.0	16.5	8.2	15.7
21°BP	Conv.			28.5		18.6		21.8		27.0		26.9	
	Cap.Cont.			32.6	32.7	21.1	22.9	23.2	26.3	26.1	32.6	25.2	33.3
14°BP	Conv.					16.5		19.9		26.1		25.9	
	Cap.Cont.					26.0	26.6	28.5	29.3	32.3	34.7	30.7	34.2

* Results do not indicate any optimization of air flow rates, rather, all capacity controlled heat pumps except the 46°BP cases are assumed to have air flows 1/2 of those in the conventional unit of comparable size.

\+ Reflects full conventional air flows used below the balance point temperature, and 1/2 air flows above the balance point temperature.

APPENDIX P

WEATHER DATA

Hours Spent in $5^{o}F$ Temperature Bands - 10 Year Average[1]

TEMP (^{o}F)	SAN FRANCISCO (HOURS)	CHARLESTON (HOURS)	NEW YORK (HOURS)	BOSTON (HOURS)	OMAHA (HOURS)	MINNEAPOLIS (HOURS)
105-109					1	
100-104	+	1	+	+	13	+
95-99	1	20	5	10	44	8
90-94	5	137	28	39	149	54
85-89	15	425	96	127	288	147
80-84	40	724	265	245	445	295
75-79	99	1143	604	433	610	468
70-74	285	1267	926	676	726	621
65-69	665	1090	877	819	721	690
60-64	1264	889	754	804	606	695
55-59	2341	787	745	781	558	602
50-54	2341	651	722	766	539	538
45-49	1153	576	796	757	543	482
40-44	449	434	838	828	543	500
35-39	99	321	858	848	655	560
30-34	10	192	603	674	663	632
25-29		79	330	429	511	609
20-24		27	188	256	390	514
15-19		5	96	151	287	383
10-14			26	74	189	311
5-9			10	35	135	246
0-4			+	4	93	186
-1- -5			1	9	40	119
-6 - -10				1	15	62
-11 - -15				+	3	31
-16 - -20						10
-21 - -25						4
-26 - -30						2
-31 - -35						
Total Hours	8767	8768	8768	8766	8767	8769

+ Indicates $0 < t < .5$ Hours

[1] U.S. Dept. of Commerce Weather Bureau, Climatography of the U.S.-Series No. 82, Decennial Census of U. S. Climate, Summary of Hourly Observations.