PROCEDURES FOR SIMULATING THE PERFORMANCE OF COMPONENTS AND SYSTEMS FOR ENERGY CALCULATIONS

THIRD EDITION

Compiled and published by

The Task Group on Energy Requirements for Heating and Cooling of Buildings

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR CONDITIONING ENGINEERS

Edited by

W. F. STOECKER

345 East 47th Street

New York, NY 10017

This booklet is dedicated to the memory of

ROBERT H. TULL

who served as chairman of the ASHRAE Task Group on Energy Requirements for the Heating and Cooling of Buildings from 1967 until his death in 1973.

ASHRAE TASK GROUP ON ENERGY REQUIREMENTS FOR HEATING AND COOLING OF BUILDINGS

E. Stamper, Chairman

R. S. Arnold

J. M. Ayres

B. Birdsall

R. S. Bycraft

E. Friberg

T. Kusuda

G. S. Leighton

M. Lokmanhekim

L. Nelson

R. F. Meriwether

E. P. Palmatier

C. Robart

W. Rudoy

C. Smith

D. G. Stephenson

W. F. Stoecker

SUBCOMMITTEE ON SYSTEMS AND EQUIPMENT

Richard S. Arnold
Carrier Air Conditioning Co.
Carrier Parkway
Syracuse, NY 13201 315-463-8411

Thomas Fehrenbacher
McDonnell-Douglas Automation Company
Department K678, Building 105, Box 516
St. Louis, MO 63166 314-232-8172

Robert H. Henninger
GARD-GATX
7449 N. Natchez Avenue
Niles, IL 60648 312-647-9000

Gerald S. Leighton
Utilities System Research
Department of Housing and Urban Development
Washington, DC 20410

Lorne Nelson
Honeywell, Inc.
2701 Fourth Avenue, S.
Minneapolis, MN 55408 612-332-5222

E. P. Palmatier
1011 Seagate Drive
Delray Beach, FL 33444 305-278-4594

William Rudoy
Mechanical Engineering Department
University of Pittsburgh
Pittsburgh, PA 15261 412-624-5343

Kasim Sinnamohideen
Johnson Service Company
507 E. Michigan Street
Milwaukee, WI 53201 414-276-9200

Wilbert F. Stoecker, Chairman and Editor
Department of Mechanical and Industrial Engr.
University of Illinois
Urbana, IL 61801 217-333-0916

INTRODUCTION TO THE REPORTS OF
THE ASHRAE TASK GROUP ON ENERGY REQUIREMENTS
FOR HEATING & COOLING OF BUILDINGS

The ASHRAE Task Group on Energy Requirements for Heating and Cooling of Buildings was formed by the ASHRAE Board of Directors, with the specific charge to develop a calculation procedure, which would determine the energy usage of any building and any HVAC system installed therein. The Task Group was to use modern computer technology to calculate building heating and cooling loads on an hourly basis using the best available analytical techniques. Because the calculations had to be made on an hourly basis, there was a need to use hourly weather data as input to the load program. The Task Group, therefore, concerned itself with the development of a mechanism of determining the appropriate weather year to use as input to the load program.

The output of the load program is not the building energy requirement, which is a function of the HVAC system installed, as well as the performance of the individual equipment components over a wide range of loads. It is necessary to simulate mathematically the performance of a wide range of air conditioning systems and individual components of systems in order to determine the energy usage under varying load conditions.

Since there were many companies and professional groups working on the problem of calculating the energy usage of buildings simultaneously with the ongoing work of the Task group, it was decided to use as many hourly programs as we could to calculate the energy use of the same building. This was done not to determine relative accuracy, but to see whether a wide group of programs would give reasonably close estimates of yearly and monthly energy usage.

Lastly, if the work of the Task Group was to have credibility, it was felt that an experiment field test measuring loads and energy usage on an hourly basis should be an essential part of the overall effort. A research team and building at Ohio State University was chosen to carry out this piece of the Task Group's overall effort. With these objectives in mind, the Task Group was organized into four sub-committees:

a) Load Calculation sub-committee
b) Systems Simulation sub-committee
c) Computer Users sub-committee
d) Field Test sub-committee

The reports of the work of the Task Group are organized in this way. Each sub-committee has presented a report on the results of its work.

The Load sub-committee's report gives the algorithms necessary to write a program on load calculation on an hourly basis. These algorithms have been used in several programs, the National Bureau of Standards NBSLD being a notable example.

The System Simulation report gives the equations for simulating the performance of individual components of the air-conditioning systems as well as the method for simulating the performance of any system. This report is the latest version of a booklet that has been issued in two earlier versions.

The report of the Computer Users sub-committee presents the work on "Operation Cross Check", the comparison of the results from the existing computer programs used as well as a method for selecting a year of input weather data (TYWD).

The Field Test sub-committee report is a compilation of all of the ASHRAE symposium papers and technical papers presented at ASHRAE annual and semi-annual meetings.

The success of the Task Group in accomplishing its goals is due to the efforts of the many members of the Task Group and its sub-committees. These people spent many hours in a labor of love, making use of their own great expertise and using that of their colleagues. Particular thanks goes to the sub-committee chairmen, present and past. These gentlemen are Dr. T. Kusuda, Dr. W. Stoecker, C. Robart, Jr., Richard Cook, and J. Marx Ayres. The other members of the T.G. and sub-committees listed in the individual reports contributed greatly.

The major force in accomplishing the goals of the Task Group was Robert H. Tull. Bob served as chairman of the T.G. from its inception until his untimely death in June 1973. He was responsible for organizing the work, recruiting the members, cajoling them to work on a tight schedule and providing direction and inspiration. It can truly be said that without his tireless effort, his administrative and organizational skills, his vision of the scope of the work to be done, the task could not have been done. The profession and society are indebted to him for the contribution that the T.G. has made. The members of the T.G. dedicate these final reports to his memory.

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1. INTRODUCTION

Role of the Systems and Equipment Subcommittee

The responsibilities of the Systems and Equipment Subcommittee in the Energy Task Group are (1) to recommend procedures for simulating component and system performance for energy calculations, (2) to review procedures and advise the field-test contractors in the system simulation activities, and (3) to encourage participation of appropriate ASHRAE technical committees in improved methods of presenting performance data.

System simulation portion of the energy calculation

The heating- and cooling-load section of the energy calculation determines the sensible and latent gains or losses for each space, hour-by-hour. Between the point where all the loads are known and the point where a cost in dollars is calculated for a day, a month, or a year of operation lies the simulation of the system performance. The system simulation section of the energy calculation translates the instantaneous heating and cooling requirements in all the spaces into the quantity of energy required by the heating and cooling plant(s).

Purpose of this booklet

The functions intended to be served by this document are (1) to provide a guide to those preparing energy calculation programs for accommodating systems and equipment, and (2) to aid in standardizing procedures and equations representing component and system performance.

2. PRINCIPLES OF SYSTEM SIMULATION

What system simulation entails

A one-sentence definition of system simulation applicable to our purposes is that of "predicting the operating quantities within a system (pressures, temperatures, energy- and fluid-flow rates) at the condition where all energy and material balances, all equations of state of working substances, and all performance characteristics of individual components are satisfied." Numerous approaches may be used to arrive at the foregoing result, and the choice of method may remain an individual preference.

Steady-state vs. dynamic simulation

The term "system simulation" is frequently applied to the process of predicting the performance of a system undergoing changes of operating variables with respect to time. In such a simulation, the thermal capacities of equipment are crucial. The subcommittee acknowledges that a future goal will require analysis of dynamic performance, but the present booklet concentrates on steady-state simulation. By steady-state simulation, we refer to the actual operation at a certain hour, but accept the fact that the steady-state operation probably will be different at the next hour. It is essential that the dynamic characteristics of the building be considered in the calculation of the thermal loads, but the dynamic response of most systems is much more rapid than that of the building. For this reason a steady-state simulation of the system is adequate for most energy calculations.

General simulation programs

An active technical field is the development of computer programs wherein the user need only connect information-flow blocks in the proper sequence, provide the equations by which the output variables are related to the input variables, then permit the computer to proceed with the simulation. The use of these simulation methods for thermal systems in buildings is attractive, but this booklet concentrates on procedures that are applicable to specific components and systems typical of current practice.

3. PROCEDURES IN SYSTEM SIMULATION

Sequential calculations

The calculations required for a system simulation frequently can be classified as "sequential" or "simultaneous." In a sequential simulation the calculation can begin at one point in the system and move progressively through a series of components to the final result. In a simultaneous calculation, on the other hand, no toehold can be found to begin the calculation because the calculation at some point in the series is dependent upon a later calculation. An example of a sequential calculation is that of calculating the fuel input rate at a hot-water boiler for a given heating requirement at an air-heating coil. Starting with the heating requirement, the first calculation could be the temperature drop of hot water through the coil, next the water temperature returning to the boiler after mixing with the streams from other air-heating coils, followed by the calculation of the rate of heat input to the water at the boiler, and finally, the rate of fuel input using the instantaneous conversion efficiency of the boiler.

Simultaneous calculations

Simultaneous calculations, which are normally more complex than sequential, result from recirculating loops such as from return air, or the water loop in a condenser and cooling tower, or the refrigerant loop in a chilling plant. In the calculation of the humidity ratio of the air in a loop where some or all of the return air is recirculated, there is no place in the loop where the humidity ratio can be specified initially. The humidity ratio in the space cannot be initially determined even though the latent heat load is known, because the humidity ratio of the supply air is unknown. The humidity ratio of the supply air is the same as that leaving the cooling coil, but that quantity is dependent upon the humidity ratio entering the coil, which in turn is dependent upon that of the return air.

This calculation must either be performed by solving simultaneous equations or by some iterative process of assuming a trial value of the humidity ratio at one point and calculating through the loop several times until the values stabilize to within a specified tolerance. Iterations will be an expected procedure in system simulation. An objective of this booklet is to recommend approximations that convert simulations that are truly simultaneous into ones that are sequential. These approximations result in only a small sacrifice in accuracy.

Need for component simulation

If system simulation presupposes the satisfaction of the performance characteristics of each component in the system, it is expected that performance data of the components must be available for the entire range of operating conditions that might be experienced during the year of operation. For example, the energy program must be capable of computing the power required by a water-chilling plant for a variety of return chilled water flows and temperatures as well as various condenser cooling water flow rates and temperatures. Part-load performance data on the performance of components and subsystems is a crucial need for energy studies.

Manufacturer's data usually appear in catalogs in the form of tables or graphs--a form which has been convenient for engineers. For computer programs that perform energy calculations, however, performance characteristics represented in equation form will be most convenient. The next two chapters in this booklet on equation-fitting and component simulation are an attempt to help standardize the presentation of performance equations used in energy calculations. Manufacturers are encouraged to provide in their catalog the equation that represents the performance data also shown in tabular or graphical form.

4. EXPRESSING PERFORMANCE DATA IN EQUATION FORM

Equation fitting is, to a great extent, an art. There are countless forms of equations; for example, polynomials, exponentials, trignometric functions, and there is no methodical procedure for selecting the most applicable one. It is assumed that tabular or graphical data are available from such sources as manufacturer's catalogs for the component. It is further understood that all physical laws or mathematical relations should be used to advantage. For example, in simulating the performance of heat exchangers, heat-transfer laws should be used wherever possible.

Polynomial representations - one independent variable

When no unique insight into the component performance is available, a polynomial representation is probably the best choice,

$$y = a_o + a_1x + a_2x^2 + \,..\, + a_nx^n \qquad \text{(Eq. 1)}$$

also, for simplicity, no higher degree should be used than necessary. For a first degree equation,

$$y = a_o + a_1x \qquad \text{(Eq. 2)}$$

two data points are needed, for example (x_1, y_1) and (x_2, y_2), which when substituted into Eq. 2 provide two simultaneous equations which can be solved for a_o and a_1:

$$y_1 = a_o + a_1x_1$$

$$y_2 = a_o + a_1x_2$$

For the second-degree equation

$$y = a_o + a_1x + a_2x^2 \qquad \text{(Eq. 3)}$$

three points relating values of x and y are substituted in order to solve for a_0, a_1, and a_2.

If the degree of the equation is n, then n+1 points are needed to describe the equation. If more than n+1 points are available, a best-fit technique can be used as described later.

Computer program for determining coefficients in polynomial equati

In the second-degree equation, Eq. 3, it is necessary to perform some computations to set up the three linear equations and then solve these equations for a_0, a_1, and a_2. This task can be done by hand, but to facilitate the solution and especially to avoid solving sets of four and five simultaneous equations needed for cubic and fourth-degree polynomials, respectively, a computer program that calculates the coefficients is shown on page 9.

The main program on page 9 reads in the values of n+1 pairs of points, sets up the simultaneous equations, calls the subroutine GAUSSY, shown on page 10, to solve the simultaneous equations, and prints out the results as shown on page 11.

The input data are arranged on cards as follows:

a. the first card provides the value of n, the degree of the equation, in an I2 field, right-justified.

b. the succeeding n+1 cards give the values of the x- and y-pairs of points, one pair to a card, each number in an F10.5 field.

The dimension statement should be set for n+1.

Polynomial representations--two independent variables

A frequently-encountered form of equation that is adequate

```
C
C        PØLYNØMIAL ØF DEGREE N,  GIVEN N+1 DATA PØINTS
C
      DIMENSIØN DEPVAR(3), VARIND(3), C(3,3), A(3), Y(3)
      READ (5,2) N
    2 FØRMAT (I2)
      NPLUS1 = N + 1
      READ(5,4) (VARIND(I), DEPVAR(I), I = 1, NPLUS1)
    4 FØRMAT (2F10.5)
C     PRINTING ØUT THE ØRIGINAL DATA
      WRITE (6,6) N
    6 FØRMAT(1H1,' ØRIGINAL DATA'//' N = ',I5,//' IND VAR',10X,'DEP VAR'
     1    //)
      WRITE (6,8) (VARIND(I), DEPVAR(I), I = 1, NPLUS1)
    8 FØRMAT (2F15.5)
      DØ 20  I = 1,NPLUS1
      C(I,1) = 1.
      DØ 18  J = 2, NPLUS1
   18 C(I,J) = C(I,J-1)*VARIND(I)
   20 CØNTINUE
      CALL GAUSSY (C, DEPVAR, A, NPLUS1)
C      PRINTØUT ØF CØEFFICIENTS
      DØ 30  I = 1,NPLUS1
      IMINUS =  I - 1
      WRITE (6,24) IMINUS, A(I)
   24 FØRMAT ('0A(',I1,')  =  ', E15.5)
   30 CØNTINUE
C     CHECK AGAINST ØRIGINAL PØINTS
      WRITE (6,36)
   36 FØRMAT('0USING CØEFFICIENTS TØ CALCULATE ØRIGINAL PØINTS'//)
      DØ 40 I = 1,NPLUS1
      Y(I) = A(1)
      DØ 38 J = 2,NPLUS1
   38 Y(I) = Y(I) + A(J)*VARIND(I)**(J-1)
      WRITE(6,39) VARIND(I), Y(I)
   39 FØRMAT(' X = ', E15.5, '  Y =  ',E15.5)
   40 CØNTINUE
      STØP
      END
```

```
      SUBRØUTINE GAUSSY (A, B, X, N)
C
C     SØLUTIØN ØF SIMULTANEØUS EQUATIØNS BY GAUSS ELIMINATIØN
C
      DIMENSIØN A(N,N), X(N), B(N)
C     BEGINNING ØF ELIMINATIØN PRØCESS
      DØ 28  K = 1,N
C     MØVING LARGEST CØEFFICIENT INTØ DIAGØNAL PØSITIØN
      AMAX = 0.
      DØ 4  I = K,N
      IF(ABS(A(I,K)) - ABS(AMAX)) 4, 4, 2
    2 AMAX = A(I,K)
      IMAX = I
    4 CØNTINUE
C     TESTING FØR INDEPENDENCE ØF EQUATIØNS
      IF(ABS(AMAX) - 0.1E-15) 10, 10, 14
   10 WRITE (6,12)
   12 FØRMAT ('0 EQUATIØNS ARE NØT INDEPENDENT')
      RETURN
C     EXCHANGING RØW IMAX AND RØW K
   14 BTEMP = B(K)
      B(K) = B(IMAX)
      B(IMAX) = BTEMP
      DØ 18  J = K,N
      ATEMP = A(K,J)
      A(K,J) = A(IMAX, J)
   18 A(IMAX,J) = ATEMP
C     SUBTRACTING A(I,K)/A(K,K) TIMES TERM IN FIRST EQ FRØM ØTHERS
      KPLUS = K + 1
      IF(K - N) 22, 28, 28
   22 DØ 24  I = KPLUS,N
      B(I) = B(I) - B(K)*A(I,K)/A(K,K)
      ACØN = A(I,K)
      DØ 24  J = K,N
   24 A(I,J) = A(I,J) - A(K,J)*ACØN/A(K,K)
   28 CØNTINUE
C     BACK SUBSTITUTIØN
      L = N
   32 SUM = 0.0
      IF(L - N) 34, 38, 38
   34 LPLUS = L + 1
      DØ 36  J = LPLUS, N
   36 SUM = SUM + A(L,J)*X(J)
   38 CØNTINUE
      X(L) = (B(L) - SUM)/A(L,L)
      IF(L - 1) 42, 42, 40
   40 L = L - 1
      GØ TØ 32
   42 RETURN
      END
```

```
ØRIGINAL DATA

N =      2

IND VAR              DEP VAR


        20.00000      1500.00000
        60.00000      3000.00000
       100.00000      4200.00000

A(0)   =        0.63750E 03

A(1)   =        0.45000E 02

A(2)   =       -0.93750E-01

USING CØEFFICIENTS TØ CALCULATE ØRIGINAL PØINTS


X =      0.20000E 02   Y =      0.15000E 04
X =      0.60000E 02   Y =      0.30000E 04
X =      0.10000E 03   Y =      0.42000E 04
```

to express the performance of many thermal components is an expression of one variable as a function of two others. If both the independent variables appear in the second degree, the complete form is

$$z = c_1 + c_2x + c_3x^2 + c_4y + c_5y^2 + c_6xy + c_7x^2y + c_8xy^2 + c_9x^2y^2 \qquad \text{(Eq. 4)}$$

A program that computes the values of c_1 through c_9 is shown on pages 12 and 13. The main program on pages 12 and 13 also employs the GAUSSY subroutine for the solution of (in this case) nine simultaneous equations. The output appears on page 14.

The input data appear on nine cards, each card containing three values--the dependent variable and the two independent variables--placed in 15-space F-fields.

```
C
C       DETERMINING CØEFFICIENTS ØF EQUATIØN FØR ØNE VARIABLE AS A
C        FUNCTIØN ØF TWØ ØTHER VARIABLES.   EQUATIØN IS ØF THE FØRM
C        DEPVAR = C1 + C2(VAR1) + C3(VAR1)(VAR1) + C4(VAR2)
C       + C5(VAR2)(VAR2) + C6(VAR1)(VAR2) + C7(VAR1)(VAR1)(VAR2)
C        +  C8(VAR1)(VAR2)(VAR2)  +  C9(VAR1)(VAR1)(VAR2)(VAR2)
C
C       INPUT FØRM ØN DATA CARD IN 15 SPACE FIELDS: DEPVAR, VAR1, VAR2
C
        DIMENSIØN DEPVAR(9), VAR1(9), VAR2(9), Z(9,9), C(9), Y(9)
        READ(5,300)  (DEPVAR(I), VAR1(I), VAR2(I), I = 1,9)
    300 FØRMAT(3F15.0)
        WRITE(6,302)
    302 FØRMAT(1H1,///,1X,' INPUT VALUES',/,6X, 'DEP VAR',9X,'VAR1',9X,
       1 'VAR2',//)
        WRITE(6,304) (DEPVAR(I), VAR1(I), VAR2(I),  I = 1,9)
    304 FØRMAT(3F15.4)
C
C        CALCULATING THE CØEFFICIENTS IN SIMULTANEØUS EQUATIØNS
        DØ 310  I = 1,9
        Z(I,1) = 1.
        Z(I,2) = VAR1(I)
        Z(I,3) = VAR1(I)**2
        Z(I,4) = VAR2(I)
        Z(I,5) = VAR2(I)**2
        Z(I,6) = VAR1(I)*VAR2(I)
        Z(I,7) = VAR1(I)*VAR1(I)*VAR2(I)
        Z(I,8) = VAR1(I)*VAR2(I)*VAR2(I)
        Z(I,9) = (VAR1(I)*VAR2(I))**2
    310 CØNTINUE
C
C        CALLING THE SIMULTANEØUS EQUATIØN SUBRØUTINE
C
        CALL GAUSSY(Z, DEPVAR, C, 9)
C
C       PRINTØUT ØF C-VALUES
        WRITE(6,135)
    135 FØRMAT (5X,//,' SØLUTIØNS FØR C IN THE FØLLØWING EQUATIØN' )
        WRITE (6,136)
    136 FØRMAT (' DEPVAR = C1 + C2(VAR1) + C3(VAR1)(VAR1) + C4(VAR2)')
        WRITE (6,137)
    137 FØRMAT(' +C5(VAR2)(VAR2) + C6(VAR1)(VAR2) + C7(VAR1)(VAR1)(VAR2)
        WRITE (6,138)
    138 FØRMAT(' + C8(VAR1)(VAR2)(VAR2) + C9(VAR1)(VAR1)(VAR2)(VAR2)')
        DØ 315 L = 1,9
        WRITE (6,175) L, C(L)
    175 FØRMAT(3H C(I2, 3H )= F20.6)
    315 CØNTINUE
        WRITE (6,316)
    316 FØRMAT (//, 5X, 'C-VALUES IN E FIELD')
        DØ 318  L = 1,9
        WRITE (6,317) L, C(L)
    317 FØRMAT (3H C(I2, 3H )= E20.6)
    318 CØNTINUE
C        CHECK AGAINST ØRIGINAL INPUT VALUES
        WRITE(6,320)
    320 FØRMAT('0CALCULATED VALUES ØF DEPENDENT VARIABLE USING CØMPUTED
       1CØEFFICIENTS')
        WRITE(6, 324)
```

```
324 FØRMAT(5X, ' DEPVAR', 9X, 'VAR1', 8X, 'VAR2')
    DØ 330 I = 1,9
    Y(I) = C(1) + C(2)*VAR1(I) + C(3)*VAR1(I)**2 + C(4)*VAR2(I)
   1  + C(5)*VAR2(I)**2 + C(6)*VAR1(I)*VAR2(I) + C(7)*VAR1(I)**2*VAR2(
   2  I) + C(8)*VAR1(I)*VAR2(I)*VAR2(I) + C(9)*(VAR1(I)*VAR2(I))**2
    WRITE(6,328)  Y(I), VAR1(I), VAR2(I)
328 FØRMAT(3F15.4)
330 CØNTINUE
    WRITE (6,329)
329 FØRMAT (1H1)
980  CØNTINUE
    STØP
    END
```

INPUT VALUES

DEP VAR	VAR1	VAR2
480000,0000	30,0000	80,0000
588000,0000	40,0000	80,0000
709000,0000	50,0000	80,0000
416000,0000	30,0000	100,0000
512000,0000	40,0000	100,0000
625000,0000	50,0000	100,0000
350000,0000	30,0000	120,0000
435000,0000	40,0000	120,0000
540000,0000	50,0000	120,0000

SOLUTIONS FOR C IN THE FOLLOWING EQUATION
DEPVAR = C1 + C2(VAR1) + C3(VAR1)(VAR1) + C4(VAR2)
+C5(VAR2)(VAR2) + C6(VAR1)(VAR2) + C7(VAR1)(VAR1)(VAR2)
+ C8(VAR1)(VAR2)(VAR2) + C9(VAR1)(VAR1)(VAR2)(VAR2)

```
C( 1 )=          140000,700000
C( 2 )=           21149,980000
C( 3 )=             -64,999830
C( 4 )=            2274,992000
C( 5 )=             -15,749970
C( 6 )=            -231,249800
C( 7 )=               2,124998
C( 8 )=               0,562500
C( 9 )=              -0,006250
```

C-VALUES IN E FIELD

```
C( 1 )=          0,140001E 06
C( 2 )=          0,211500E 05
C( 3 )=         -0,649998E 02
C( 4 )=          0,227499E 04
C( 5 )=         -0,137500E 02
C( 6 )=         -0,231250E 03
C( 7 )=          0,212500E 01
C( 8 )=          0,562500E 00
C( 9 )=         -0,625000E-02
```

CALCULATED VALUES OF DEPENDENT VARIABLE USING COMPUTED COEFFICIENTS

DEPVAR	VAR1	VAR2
480000,1000	30,0000	80,0000
587999,3000	40,0000	80,0000
708998,2000	50,0000	80,0000
416000,0000	30,0000	100,0000
511999,5000	40,0000	100,0000
624999,0000	50,0000	100,0000
350000,1000	30,0000	120,0000
434999,6000	40,0000	120,0000
539998,9000	50,0000	120,0000

Best-fit representation

There are many situations when it is desirable to find the equation providing the best fit to a large number of points in preference to fitting the equation exactly to a limited number of points. The original data points may be experimental or may be obtained from tables or may be read from graphs. The procedure for computing the best-fit coefficients is similar to the previous method except that the number of data points exceeds the number of coefficients.

While most computer centers have a library of subroutines for fitting polynomials for one variable or equations for more than one variable, a program is listed here as an additional example. This particular subroutine ORTHON is used with permission from the University of Minnesota Computing Center, Programmers: James Carlson and Richard L. Hotchkiss.

General Explanation of Program

The program allows the user to fit a polynomial with one or two independent variables. It also allows the user to examine several equation fits of the data in a given computer run by specifying a range of values for the number of terms in the polynomial equation. There are three types of input to the program for a given set of data points.

a. Card 1 contains program control information. All entries on card 1 are integers and must be right justified in the three-column fields.

Columns 1-3 is number of data points
Columns 4-6 is smallest number of terms to be considered in the polynomial

Columns 7 - 9 is the largest number of terms to be considered in the polynomial

Columns 10 - 12 is the number of independent variables, (1 or 2)

b. The next three cards contain problem description or title information and must be present. Any alphanumeric information on these cards will be printed at the top of the first page of program output.

c. The next one or more cards contain the data points in eight column fields across the card. For data points of two independent variables, place the data in order of X independent variable, Y independent variable and the dependent variable, three points per card for a total of nine F8.0 fields on the card. For data points having one independent variable, place the data points in order of X independent variable and dependent variable, five points per card for a total of ten F8.0 fields to a card. Each card, except the last one, must be completely filled. The data points may be entered in any order.

The program is designed to run multiple cases in a given submittal of the program. Simply stack the cases and for normal termination place a card with '99' in columns 2 - 3 after the last case.

Polynomial

This program (shown on the following pages) takes, as an example, psychrometric data from the 1972 ASHRAE Handbook of Fundamentals, Table 1, Chapter 32, and fits a polynomial for the wet-bulb temperature as a function of the enthalpy at saturation

($t = c_1 + c_2 h_s + c_3 h_s^2 + \ldots$) over the range of temperatures from 38 F to 90 F. In this program a best fit is produced with 3, 4, 5, 6, and 7 terms. The equation with 4 terms has a maximum deviation of 0.1 F and with 5 terms a maximum deviation of 0.01 F. The accuracy desired consistent with the computation time for an equation with more terms will determine the ultimate choice. Another criteria of best fit is the standard error which is printed at the bottom of the page.

Input data are arranged as follows:

a. First card states number of data points, the smallest number of terms to be considered, the largest number of terms to be considered and the number of independent variables. The four I3 fields are right-justified. In this example the card reads: -27--3--7--1.

b. The next three cards contain title information.

c. Succeeding data cards with 5 data points per card in F8.0 fields; DEP VAR and INDEP VAR make one data point. In this example there are 5 full data cards plus 2 data points on the 6th card.

<u>Polynomial representation with two dependent variables, multiple regression.</u>

The second example takes compressor capacity data in thousands of Btu/hr as a function of saturated suction temperature and saturated condensing temperature. The data points were obtained from a graph of the compressor performance of one manufacturer. The best fit results are shown for a 9-term equation similar to the exact fit on page 14.

This second example also calls the Subroutine ORTHON and Function DOTPR.

Input data are arranged as follows:

a. First card similar to polynomial program, but on this example -17--7--9.

b. Succeeding data cards with 3 data points per card in F8.0 fields for each variable in the order, X, Y, and DEP variable. In this example there are five full data cards plus one more card with 2 data points.

```
     1PERCENT DIFF')
 113 FORMAT(' ',4E14.4,F10.2)
C
  20 REAC(5,7) NPTS,NTER1,NTER2,NIV
C
C             CHECK FCR END OF ALL CASES IN GIVEN JOB STREAM
C
     IF(NPTS-99) 25,205,205
  25 CONTINUE
C
C             READ IN THE THREE TITLE CARDS
C
     REAC(5,3) (ATITL(I),I=1,60)
     WRITE(6,4) (ATITL(I),I=1,60)
C
     WRITE(6,9)
C
     WRITE(6,10) NPTS,NTER1,NTER2,NIV
C
     IF(NIV.NE.2) GO TC 100
C
C        ECUATION FORM  CEP  =  C1 + C2*X + C3*X*X + C4*Y + C5*Y*Y +
C        C6*X*Y + C7*X*X*Y + C8*X*Y*Y + C9*X*X*Y*Y
C
C             INPUT DATA IN FIELDS OF F8.0. EACH DATA POINT CONSISTS CF
C             X IND VAR, Y IND VAR AND DEP VAR. THREE POINTS PER CARC
C
     REAC(5,11) (X(I),Y(I),CEP(I),I=1,NPTS)
     WRITE(6,6) (X(I),Y(I),CEP(I),I=1,NPTS)
     WRITE(6,8)
C
     CO 3C J=1,NPTS
C
     NX(J) = J
     XJ = X(J)
     YJ = Y(J)
     X2 = XJ*XJ
     Y2 = YJ*YJ
     A(J,1) = 1.0
     A(J,2) = XJ
     A(J,3) = X2
     A(J,4) = YJ
     A(J,5) = Y2
     A(J,6) = XJ*YJ
     A(J,7) = X2*YJ
     A(J,8) = XJ*Y2
     A(J,9) = X2*Y2
     W(J) = 1.0
  30 CONTINUE
C
     CO 50 NTER = NTER1,NTER2
     NTERM = NTER + 1
C
     CALL CRTHON(HYPERF,PH,65,NX,DEP,W,NPTS,0,NTER,COEF,SC,CEPC,CEL,
    11.E-9,MM)
```

```
C
C        THIS PROGRAM CALCULATES THE BEST FIT CCEFFICIENTS FOR
C        PCLYNOMIAL EQUATIONS OF 1 OR 2 INDEPENDENT VARIABLES FCR
C        SEVERAL CATA PCINTS
C
C         CIMENSIONS ARE ESTABLISHED FOR AS MANY AS 50 DATA PCINTS
C
C         NPTS    NUMBER OF DATA POINTS
C         NTER    NUMBER OF TERMS IN EQUATION
C         NTER1   THE PROGRAM ALLOWS THE USER TO STUDY SEVERAL EQUATICN
C                 FITS CF THE CATA IN A GIVEN RUN BY SPECIFYING A RANGE
C                 OF VALUES FCR THE NUMBER CF TERMS IN THE PCLYNCMIAL
C                 EQUATICN  NTER1 IS THE SMALLEST NUMBER CF TERMS TC BE
C                 CONSICERED
C         NTER2   THE LARGEST NUMBER CF TERMS TC BE CONSIDERED
C         NIV     NUMBER OF INCEPENDENT VARIABLES
C             =1  EQUATICN OF FCRM - CEP = FUNCTION(X)
C             =2  EQUATION OF FORM - DEP = FUNCTION(X,Y)
C
C         CEP     DEPENCENT VARIABLE
C         X       FIRST INDENCENT VARIABLE
C         Y       SECONC INDEPENDENT VARIABLE
C         CEPC    DEPENCENT VARIABLE CALCULATEC FROM FINAL SCLUTICN
C         CCEF    COEFFICIENTS FCR EQUATION
C         SC      STANDARD ERRORS OF COEFFICIENTS
C         CEL     DIFFERENCE BETWEEN ORIGINAL CATA PCINT AND CALC. VALUE
C
      REAL NX(50)
      COMMCN A
      EXTERNAL HYPERF,PCLY
      CIMENSION DEP(50),X(50),Y(50),CEPC(5C),CCEF(10),SC(11),W(50),
     1PH(65,50),DEL(50),A(50,11),ERROR(50),ATITL(60)
C
    3 FORMAT(20A4)
    4 FORMAT('1',20A4/(1X,20A4))
    6 FORMAT('0',3(5X,'X',7X,'Y',6X,'DEP',1X)//(1X,9F8.2))
    7 FCRMAT(4I3)
    8 FCRMAT('0 EQUATICN FORM - DEP = C1 + C2*X + C3*X*X + C4*Y + C5*Y*Y
     1 + C6*X*Y'/24X,'C7*X*X*Y + C8*X*Y*Y + C9*X*X*Y*Y')
    9 FORMAT('0 LISTING OF CRIGINAL INPUT CATA'//)
   10 FCRMAT('0 NUMBER CF PCINTS = ' I4/'  SMALLEST NUMBER OF TERMS TC B
     1E CONSIDERED = ',I4/'  LARGEST NUMBER OF TERMS TO BE CONSICEREC =
     2',I4/'  NUMBER OF INDEPENDENT VARIABLES = ',I4)
   11 FORMAT(9F8.C)
   12 FCRMAT('0' 7X,'XVAR',10X,'YVAR      CRIG CEP VAR  CALC CEP VAR   CI
     1FFERENCE  PERCENT CIFF'/)
   13 FORMAT(' ',5E14.4,F10.2)
   14 FCRMAT('0  TERM  COEFFICIENT')
   15 FORMAT(' ',I5,E18.8)
   16 FORMAT('0  THE NUMBER CF INDEPENDENT TERMS IS ',I2,'  AND THE STAN
     1CARD ERROR IS ',E12.4)
  106 FORMAT('0',5(5X,'X',6X,'DEP',1X)//(1X,10F8.2))
  108 FORMAT('C EQUATICN FORM - DEP = C1 + C2*X + C3*X*X + C4*X*X*X + C5
     1*X*X*X*X + ....')
  111 FORMAT(5(F8.0,F8.C))
  112 FORMAT('0         XVAR      ORIG DEP VAR  CALC DEP VAR    DIFFERENCE
```

```
C
      CO 4C I=1,NPTS
   40 ERRCR(I) = DEL(I)/DEP(I)*100.
C
      hRITE(6,12)
      WRITE(6,13) (X(I),Y(I),DEP(I),DEPC(I),DEL(I),ERROR(I),I=1,NPTS)
      WRITE(6,14)
      hRITE(6,15) (I,COEF(I),I=1,NTER)
      WRITE(6,16) MM,SC(NTERM)
   50 CONTINUE
C
      GC TC 20
C
  100 CCNTINUE
C
C         GENERATES POLYNOMIAL OF FCRM DEP=C1 + C2*X + C3*X*X + .....
C
C              INPUT DATA IN FIELDS CF F8.C. EACH DATA PCINT CCNSISTS OF
C              X IND VAR AND DEP VAR. FIVE DATA POINTS PER CARC
C
      REAC(5,111) (X(I),DEP(I),I=1,NPTS)
      WRITE(6,106) (X(I),DEP(I),I=1,NPTS)
      WRITE(6,108)
C
      CC 120 J=1,NPTS
  120 W(J) = 1.0
C
      CO 15C NTER=NTER1,NTER2
      NTERM = NTER + 1
C
      CALL CRTHCN(PCLY,PH,65,X,DEP,W,NPTS,0,NTER,COEF,SC,CEPC,CEL,1.E-9,
     1MM)
C
      CO 140 I=1,NPTS
  140 ERRCR(I) = DEL(I)/DEP(I)*100.0
C
      hRITE(6,112)
      hRITE(6,113) (X(I),DEP(I),DEPC(I),DEL(I),ERROR(I),I=1,NPTS)
      hRITE(6,14)
      hRITE(6,15) (I,CCEF(I),I=1,NTER)
      hRITE(6,16) MM,SC(NTERM)
  150 CCNTINUE
      GC TC 20
  205 CONTINUE
      STOP
      ENC
```

```
      SUBROUTINE ORTHON(FUN,PHI,IDA,X,Y,W,NBIG,M1,NSMALL,COE,SEOFC,
     1BACKSL,DELTAS,EPS,NCODE)
C
      DIMENSION SEOFC(1),DELTAS(1),BACKSL(1),X(1),Y(1),COE(1),W(1)
      DIMENSION PHI(1)
C
      IF(NBIG) 10,10,12
   10 WRITE(6,9010) NBIG
 9010 FORMAT('0 NUMBER OF POINTS = ',I4,' IS LE TO ZERO')
      GO TO 20
   12 IF(NSMALL) 14,14,16
   14 WRITE(6,9014) NSMALL
 9014 FORMAT('0 NUMBER OF TERMS = ',I4,'  IS LE TO ZERO')
      GO TO 20
   16 IF(NBIG-NSMALL)18,30,30
   18 WRITE(6,9018) NBIG,NSMALL
 9018 FORMAT('0 NUMBER OF POINTS = ',I4,' IS LT NUMBER OF TERMS = ',I4)
   20 WRITE(6,9020)
 9020 FORMAT('0 ERROR ORTHON')
      RETURN
   30 CONTINUE
      N1 = NBIG + NSMALL
      N2 = NSMALL + 1
      NSWITC = 1
      N5 = M1 + 1
      N4 =  NBIG + 1
      II = 1
      IF(N5-N2) 40,290,290
C
   40 DO 280 K=N5,N2
      N3 = K-1
      N3IDA = N3*IDA
      N4IDA = N3IDA + 1
C
      GO TO (50,60),NSWITC
   50 N7 = N3
      GO TO 65
   60 N7 = NCODE
   65 K1 = NBIG + K
C
      DO 70 I=N4,N1
      N3IDAI = N3IDA + I
   70 PHI(N3IDAI) = 0.0
C
      IF(K-N2) 80,100,100
   80 JJ = K
C
      DO 90 I=1,NBIG
      X1 = X(I)
      N3IDAI = N3IDA + I
   90 PHI(N3IDAI) = FUN(X1,JJ)
C
      N3IDAK = N3IDA + K1
```

```
      PHI(N3IDAK) = 1.
      GO TO 120
C
  100 II = NBIG + 1
C
      DO 110 I=1,NBIG
      NSIDI = NSMALL*IDA + I
  110 PHI(NSIDI) = Y(I)
C
  120 IF(N7) 150,150,130
C
  130 DO 140 J=1,N7
      K2 = NBIG + J
      JIDA = (J-1) * IDA
C
      C = DOTPRD(PHI(N4IDA),PHI(JIDA+1),W,NBIG)
C
      DO 140 I=1,K2
      N3IDAI = N3IDA + I
      JIDAI = JIDA + I
  140 PHI(N3IDAI) = PHI(N3IDAI) - C*PHI(JIDAI)
C
  150 IF(K-N2) 160,190,160
  160 C = DOTPRD(PHI(N4IDA),PHI(N4IDA),W,NBIG)
C
      IF(C-EPS) 300,300,170
  170 C = SQRT(C)
C
      DO 180 I=1,K1
      N3IDAI = N3IDA + I
  180 PHI(N3IDAI) = PHI(N3IDAI)/C
C
      IF(N7) 200,280,200
C
  190 K1 = N1
C
  200 DO 210 I = 1,K1
  210 DELTAS(I) = 0.0
C
      IF(N7) 240,240,220
C
  220 DO 230 J=1,N7
      K2 = NBIG + J
      JIDA = (J-1)*IDA
C
      C = DOTPRD(PHI(N4IDA),PHI(JIDA+1),W,NBIG)
C
      DO 230 I=1,K2
      JIDAI = JIDA + I
  230 DELTAS(I) = DELTAS(I) + C*PHI(JIDAI)
C
  240 DO 250 I=II,K1
      N3IDAI = N3IDA + I
  250 PHI(N3IDAI) = PHI(N3IDAI) - DELTAS(I)
C
      IF(K-N2) 260,280,280
C
```

```
  260 C = CCTPRD(PHI(N4IDA),PHI(N4IDA),W,NBIG)
C
      C = SQRT(C)
C
      CO 27C I=1,K1
      N3IDAI = N3IDA + I
  270 PHI(N3IDAI) = PHI(N3IDAI)/C
  280 CONTINUE
C
  290 GO TO (310,320),NSWITC
C
  300 NCCDE = K-1
      K = N2-1
      NSWITC = 2
      GO TC 280
  310 NCODE = NSMALL
  320 SUMS = 0.0
      JICA = NSMALL*IDA
      C = AMAXO(1,NBIG-NCCDE)
C
      CO 330 I=1,NCODE
      IJICAN = I + JIDA + NBIG
      CCE(I) = -PHI(IJICAN)
      SECFC(I) = 0.0
      CO 330 J = 1,NCODE
      JICANI = (J-1)*ICA + NBIG + I
      X1 = PHI(JICANI)
  330 SECFC(I) = SEOFC(I) + X1*X1
C
      CO 350 I=1,NBIG
      X1 = 0.C
      X2 = X(I)
      CO 340 JJ=1,NCODE
  340 X1 = X1 + FUN(X2,JJ) * COE(JJ)
C
      CELTAS(I) = Y(I) - X1
      SUMS = SUMS + DELTAS(I) * DELTAS(I)
  350 BACKSL(I) = X1
C
      CO 360 I = 1,NCODE
  360 SECFC(I) = SCRT(SEOFC(I)*SUMS/C)
C
      CELTAS(NBIG+1) = SUMS
      SECFC(NSMALL+1) = SQRT(SUMS/C)
      RETURN
      ENC
```

```
      FUNCTION DOTPRD(A,B,W,N)
C
C               FUNCTION LIMITED TO 500 POINTS
C
      DIMENSION A(500),B(500),W(500)
C
      SUM = 0.0
      DO 1 I=1,N
    1 SUM = SUM + A(I)*B(I)*W(I)
      DOTPRD = SUM
      RETURN
      END



      FUNCTION POLY(XX,JJ)
C
C               GENERATES POLYNOMIAL - DEP=C1+C2*X+C3*X*X+.....CN*X**N-1
C
      IF(JJ) 10,20,30
   10 POLY = 0.0
      RETURN
   20 POLY = 1.0
      RETURN
   30 POLY = XX**(JJ-1)
      RETURN
      END



      FUNCTION HYPERF(XX,JJ)
C
      DIMENSION A(50,10)
      COMMON A
C
      NN = XX
      HYPERF = A(NN,JJ)
      RETURN
      END
```

POLYNOMIAL FIT (ONE INDEPENDENT VARIABLE) OF PSYCHROMETRIC DATA -
WET BULB TEMPERATURE AS A FUNCTION OF ENTHALPY AT SATURATION
EQUATION FORM - T = C1 + C2*HS + C3*HS*HS + C4*HS*HS*HS + ...

LISTING OF ORIGINAL INPUT DATA

NUMBER OF POINTS = 27
SMALLEST NUMBER OF TERMS TO BE CONSIDERED = 3
LARGEST NUMBER OF TERMS TO BE CONSIDERED = 7
NUMBER OF INDEPENDENT VARIABLES = 1

X	DEP	X	DEP	X	DEP	X	DEP	X	DEP
14.32	38.00	15.23	40.00	16.17	42.00	17.15	44.00	18.16	46.00
19.21	48.00	20.30	50.00	21.44	52.00	22.61	54.00	23.84	56.00
25.12	58.00	26.46	60.00	27.85	62.00	29.31	64.00	30.83	66.00
32.42	68.00	34.09	70.00	35.83	72.00	37.66	74.00	39.57	76.00
41.58	78.00	43.69	80.00	45.90	82.00	48.22	84.00	50.66	86.00
53.23	88.00	55.93	90.00						

EQUATION FORM - DEP = C1 + C2*X + C3*X*X + C4*X*X*X + C5*X*X*X*X + ...

XVAR	ORIG DEP VAR	CALC DEP VAR	DIFFERENCE	PERCENT DIFF
0.1432E 02	0.3800E 02	0.3881E 02	-0.8119E 00	-2.14
0.1523E 02	0.4000E 02	0.4054E 02	-0.5443E 00	-1.36
0.1617E 02	0.4200E 02	0.4230E 02	-0.3041E 00	-0.72
0.1715E 02	0.4400E 02	0.4411E 02	-0.1069E 00	-0.24
0.1816E 02	0.4600E 02	0.4593E 02	0.6946E-01	0.15
0.1921E 02	0.4800E 02	0.4779E 02	0.2104E 00	0.44
0.2030E 02	0.5000E 02	0.4968E 02	0.3202E 00	0.64
0.2144E 02	0.5200E 02	0.5161E 02	0.3866E 00	0.74
0.2261E 02	0.5400E 02	0.5355E 02	0.4482E 00	0.83
0.2384E 02	0.5600E 02	0.5554E 02	0.4605E 00	0.82
0.2512E 02	0.5800E 02	0.5755E 02	0.4465E 00	0.77
0.2646E 02	0.6000E 02	0.5960E 02	0.3986E 00	0.66
0.2785E 02	0.6200E 02	0.6166E 02	0.3385E 00	0.55
0.2931E 02	0.6400E 02	0.6375E 02	0.2455E 00	0.38
0.3083E 02	0.6600E 02	0.6586E 02	0.1436E 00	0.22
0.3242E 02	0.6800E 02	0.6797E 02	0.2914E-01	0.04
0.3409E 02	0.7000E 02	0.7010E 02	-0.9898E-01	-0.14
0.3583E 02	0.7200E 02	0.7222E 02	-0.2153E 00	-0.30
0.3766E 02	0.7400E 02	0.7433E 02	-0.3298E 00	-0.45
0.3957E 02	0.7600E 02	0.7642E 02	-0.4151E 00	-0.55
0.4158E 02	0.7800E 02	0.7848E 02	-0.4754E 00	-0.61
0.4369E 02	0.8000E 02	0.8049E 02	-0.4902E 00	-0.61
0.4590E 02	0.8200E 02	0.8244E 02	-0.4379E 00	-0.53
0.4822E 02	0.8400E 02	0.8430E 02	-0.3036E 00	-0.36
0.5066E 02	0.8600E 02	0.8607E 02	-0.6796E-01	-0.08
0.5323E 02	0.8800E 02	0.8771E 02	0.2929E 00	0.33
0.5593E 02	0.9000E 02	0.8919E 02	0.8132E 00	0.90

TERM	COEFFICIENT
1	0.78367901E 01
2	0.24069185E 01
3	-0.17028797E-01

THE NUMBER OF INDEPENDENT TERMS IS 3 AND THE STANDARD ERROR IS 0.4167E 00

XVAR	ORIG DEP VAR	CALC DEP VAR	DIFFERENCE	PERCENT DIFF
0.1432E 02	0.3800E 02	0.3812E 02	-0.1181E 00	-0.31
0.1523E 02	0.4000E 02	0.4006E 02	-0.5794E-01	-0.14
0.1617E 02	0.4200E 02	0.4201E 02	-0.6149E-02	-0.01
0.1715E 02	0.4400E 02	0.4398E 02	0.2150E-01	0.05
0.1816E 02	0.4600E 02	0.4595E 02	0.5003E-01	0.11
0.1921E 02	0.4800E 02	0.4794E 02	0.6470E-01	0.13
0.2030E 02	0.5000E 02	0.4993E 02	0.7108E-01	0.14
0.2144E 02	0.5200E 02	0.5194E 02	0.5725E-01	0.11
0.2261E 02	0.5400E 02	0.5394E 02	0.6378E-01	0.12
0.2384E 02	0.5600E 02	0.5595E 02	0.4549E-01	0.08
0.2512E 02	0.5800E 02	0.5797E 02	0.2637E-01	0.05
0.2646E 02	0.6000E 02	0.6000E 02	-0.2136E-02	-0.00
0.2785E 02	0.6200E 02	0.6202E 02	-0.1817E-01	-0.03
0.2931E 02	0.6400E 02	0.6404E 02	-0.4355E-01	-0.07
0.3083E 02	0.6600E 02	0.6606E 02	-0.5695E-01	-0.09
0.3242E 02	0.6800E 02	0.6806E 02	-0.6497E-01	-0.10
0.3409E 02	0.7000E 02	0.7007E 02	-0.7274E-01	-0.10
0.3583E 02	0.7200E 02	0.7206E 02	-0.6197E-01	-0.09
0.3766E 02	0.7400E 02	0.7405E 02	-0.4988E-01	-0.07
0.3957E 02	0.7600E 02	0.7602E 02	-0.2089E-01	-0.03
0.4158E 02	0.7800E 02	0.7799E 02	0.7675E-02	0.01
0.4369E 02	0.8000E 02	0.7996E 02	0.3796E-01	0.05
0.4590E 02	0.8200E 02	0.8193E 02	0.6886E-01	0.08
0.4822E 02	0.8400E 02	0.8391E 02	0.8714E-01	0.10
0.5066E 02	0.8600E 02	0.8592E 02	0.7631E-01	0.09
0.5323E 02	0.8800E 02	0.8798E 02	0.1526E-01	0.02
0.5593E 02	0.9000E 02	0.9012E 02	-0.1155E 00	-0.13

TERM	COEFFICIENT
1	-0.29281813E 00
2	0.32565126E 01
3	-0.43804348E-01
4	0.25891233E-03

THE NUMBER OF INDEPENDENT TERMS IS 4 AND THE STANDARD ERROR IS 0.6415E-01

ERM	COEFFICIENT
1	-0.39182625E 01
2	0.37654705E 01
3	-0.68696260E-01
4	0.76523377E-03
5	-0.36467900E-05

THE NUMBER OF INDEPENDENT TERMS IS 5 AND THE STANDARD ERROR IS 0.6995E-02

TERM	COEFFICIENT
1	-0.49167938E 01
2	0.39413967E 01
3	-0.80401421E-01
4	0.11341241E-02
5	-0.91809115E-05
6	0.31787817E-07

THE NUMBER OF INDEPENDENT TERMS IS 6 AND THE STANDARD ERROR IS 0.3618E-02

XVAR	ORIG DEP VAR	CALC DEP VAR	DIFFERENCE	PERCENT DIFF
0.1432E 02	0.3800E 02	0.3800E 02	-0.9003E-03	-0.00
0.1523E 02	0.4000E 02	0.4000E 02	0.1373E-03	0.00
0.1617E 02	0.4200E 02	0.4200E 02	0.4822E-02	0.01
0.1715E 02	0.4400E 02	0.4400E 02	-0.3403E-02	-0.01
0.1816E 02	0.4600E 02	0.4600E 02	0.3815E-03	0.00
0.1921E 02	0.4800E 02	0.4800E 02	0.2899E-03	0.00
0.2030E 02	0.5000E 02	0.5000E 02	0.1221E-02	0.00
0.2144E 02	0.5200E 02	0.5201E 02	-0.9827E-02	-0.02
0.2261E 02	0.5400E 02	0.5399E 02	0.6348E-02	0.01
0.2384E 02	0.5600E 02	0.5600E 02	0.3342E-02	0.01
0.2512E 02	0.5800E 02	0.5800E 02	0.3448E-02	0.01
0.2646E 02	0.6000E 02	0.6000E 02	-0.3540E-02	-0.01
0.2785E 02	0.6200E 02	0.6200E 02	0.2045E-02	0.00
0.2931E 02	0.6400E 02	0.6400E 02	-0.3296E-02	-0.01
0.3083E 02	0.6600E 02	0.6600E 02	-0.5198E-03	-0.00
0.3242E 02	0.6800E 02	0.6800E 02	0.1877E-02	0.00
0.3409E 02	0.7000E 02	0.7000E 02	-0.2991E-02	-0.00
0.3583E 02	0.7200E 02	0.7200E 02	0.1938E-02	0.00
0.3766E 02	0.7400E 02	0.7400E 02	-0.1236E-02	-0.00
0.3957E 02	0.7600E 02	0.7600E 02	0.3693E-02	0.00
0.4158E 02	0.7800E 02	0.7800E 02	0.1236E-02	0.00
0.4369E 02	0.8000E 02	0.8000E 02	-0.2411E-02	-0.00
0.4590E 02	0.8200E 02	0.8200E 02	-0.1587E-02	-0.00
0.4822E 02	0.8400E 02	0.8400E 02	0.4272E-03	0.00
0.5066E 02	0.8600E 02	0.8600E 02	0.1389E-02	0.00
0.5323E 02	0.8800E 02	0.8800E 02	-0.5188E-03	-0.00
0.5593E 02	0.9000E 02	0.9000E 02	-0.7629E-04	-0.00

TERM	COEFFICIENT
1	-0.46608486E 01
2	0.38671288E 01
3	-0.75826645E-01
4	0.93751843E-03
5	-0.46246359E-05
6	-0.22383272E-07
7	0.25905589E-09

THE NUMBER OF INDEPENDENT TERMS IS 7 AND THE STANDARD ERROR IS 0.3666E-02

POLYNOMIAL FIT (TWO INDEPENDENT VARIABLES) OF COMPRESSOR CAPACITY IN K BTU/HR AS A FUNCTION OF SATURATED SUCTION TEMPERATURE AND SATURATED CONDENSING TEMPERATURE

LISTING OF ORIGINAL INPUT DATA

NUMBER OF POINTS = 17
SMALLEST NUMBER OF TERMS TO BE CONSIDERED = 9
LARGEST NUMBER OF TERMS TO BE CONSIDERED = 9
NUMBER OF INDEPENDENT VARIABLES = 2

X	Y	DEP	X	Y	DEP	X	Y	DEP
80.00	30.00	40.80	100.00	30.00	36.00	125.00	30.00	29.80
145.00	30.00	23.40	80.00	35.00	40.40	100.00	35.00	40.40
125.00	35.00	33.20	145.00	35.00	26.20	80.00	40.00	51.40
100.00	40.00	45.10	125.00	40.00	36.60	145.00	40.00	29.30
100.00	45.00	50.00	125.00	45.00	40.80	145.00	45.00	32.50
125.00	50.00	44.80	145.00	50.00	36.00			

EQUATION FORM - DEP = C1 + C2*X + C3*X*X + C4*Y + C5*Y*Y + C6*X*Y
C7*X*X*Y + C8*X*Y*Y + C9*X*X*Y*Y

XVAR	YVAR	ORIG DEP VAR	CALC DEP VAR	DIFFERENCE	PERCENT DIFF
0.8000E 02	0.3000E 02	0.4080E 02	0.4006E 02	0.7361E 00	1.80
0.1000E 03	0.3000E 02	0.3600E 02	0.3633E 02	-0.3296E 00	-0.92
0.1250E 03	0.3000E 02	0.2980E 02	0.2990E 02	-0.1036E 00	-0.35
0.1450E 03	0.3000E 02	0.2340E 02	0.2336E 02	0.4453E-01	0.19
0.8000E 02	0.3500E 02	0.4040E 02	0.4256E 02	-0.2157E 01	-5.34
0.1000E 03	0.3500E 02	0.4040E 02	0.3959E 02	0.8080E 00	2.00
0.1250E 03	0.3500E 02	0.3320E 02	0.3330E 02	-0.9834E-01	-0.30
0.1450E 03	0.3500E 02	0.2620E 02	0.2619E 02	0.7614E-02	0.03
0.8000E 02	0.4000E 02	0.5140E 02	0.4973E 02	0.1674E 01	3.26
0.1000E 03	0.4000E 02	0.4510E 02	0.4468E 02	0.4179E 00	0.93
0.1250E 03	0.4000E 02	0.3680E 02	0.3682E 02	-0.2008E-01	-0.05
0.1450E 03	0.4000E 02	0.2930E 02	0.2928E 02	0.1630E-01	0.06
0.1000E 03	0.4500E 02	0.5000E 02	0.5160E 02	-0.1601E 01	-3.20
0.1250E 03	0.4500E 02	0.4080E 02	0.4047E 02	0.3312E 00	0.81
0.1450E 03	0.4500E 02	0.3250E 02	0.3263E 02	-0.1282E 00	-0.39
0.1250E 03	0.5000E 02	0.4480E 02	0.4424E 02	0.5568E 00	1.24
0.1450E 03	0.5000E 02	0.3600E 02	0.3622E 02	-0.2236E 00	-0.62

TERM	COEFFICIENT
1	0.69064111E 03
2	-0.10172931E 02
3	0.37578840E-01
4	-0.39382843E 02
5	0.58593160E 00
6	0.60625738E 00
7	-0.22966426E-02
8	-0.88001341E-02
9	.0.33063625E-04

THE NUMBER OF INDEPENDENT TERMS IS 9 AND THE STANDARD ERROR IS 0.1225E 01

5. COMPONENT SIMULATION

The pattern followed for the presentation of most of the performance characteristics is as follows:

a. List of performance characteristics that are of importance in energy calculations for the component in question.

b. Form of the equation

c. Typical values

d. Example equation using typical values.

In part "a", if the flow rate of a component is important for energy calculations and this flow rate is a function of the temperature and pressure, the form would be written:

flow rate, lb/hr (kg/s) = f(temperature, °F or °C, pressure, psia or N/m^2)

The suggested form of the equation in part "b" is the simplest in the judgment of the writers of this document to adequately express the performance of the component. The authors are hopeful that users of this document will offer suggestions for either simplification or refinement of the proposals.

The inclusion of typical values of the variables does not constitute a recommendation of these values, because the values in many cases depend upon the size of the unit, and the efficiency of a given component depends upon its design and condition. The only purpose of the typical values and example equation is to indicate that an equation can be adapted to the data.

5a. Boilers and furnaces

The conversion efficiency of energy in a boiler or furnace is defined as the quotient of the energy transferred to the air, water, or other medium divided by the heating capability of the input energy (fuel or electricity). The accuracy of this conversion efficiency η often has a significant influence on the accuracy of the final energy calculation, so effort in obtaining detailed information about the specific boiler or furnace actually used in the installation is usually justified.

Three classifications apply: A. electric heaters; B. boilers equipped with modulating or shutoff capabilities for combustion air; and C. boilers where the combustion air is supplied by natural convection. In classification A there is normally a complete conversion of electric energy into heating of the air or water, so η = 100 percent. The major distinction between B and C is whether there are part-load losses due to excess combustion air at low or zero burner capacity. Boilers and furnaces of Classification B are capable of operating at a fairly constant conversion efficiency of between 70 and 85 percent.

The efficiencies of boilers in Classification C drop off appreciably at low capacities.

a. $\eta = f(L)$

where η = conversion efficiency, percent
L = percent of full load

b. $\eta = a_o + a_1 L + a_2 L^2$

c.

η	L
60	20
70	75
80	100

d. $\eta = 60.46 - 0.0773\ L + 0.00273\ L^2$

5b. Centrifugal and absorption water chillers

Centrifugal water chillers

Cooling capacity:

a. $Q = f(t_{CH}, t_{CD})$

where Q = cooling capacity, tons
t_{CH} = leaving chilled water temperature, °F
t_{CD} = entering condenser water temperature, °F

b. $$Q = c_1 + c_2 t_{CH} + c_3 t_{CH}^2 + c_4 t_{CD} + c_5 t_{CD}^2 + c_6 t_{CH} t_{CD} + c_7 t_{CH}^2 t_{CD} + c_8 t_{CH} t_{CD}^2 + c_9 t_{CH}^2 t_{CD}^2$$

c. Full load capacity expressed in tons (12,000 Btu/hr)

Nominal 400-ton unit with fixed components:
2-pass chiller and 2-pass condenser
0.0005 fouling factor for chiller and condenser
fixed water flow rates that produce approximately a 10°F rise in chiller and condenser water temperatures
chiller: 1,000 gpm and condenser: 1200 gpm

t_{CH} °F \ t_{CD} °F	80	82.5	85	87.5	90
40	420	412	400	392	380
42	428	420	408	400	388
44	437	429	417	409	393
46	445	437	425	413	401

d. $$Q = -6901.34 + 432.367\, t_{CH} - 6.0333\, t_{CH}^2 + 177.867\, t_{CD} - 1.10222\, t_{CD}^2 - 10.4133\, t_{CH} t_{CD} + 0.14667\, t_{CH}^2 t_{CD} + 0.063111\, t_{CH} t_{CD}^2 - 0.000888\, t_{CH}^2 t_{CD}^2$$

Each particular machine's performance will vary according to the selection of components, pass arrangements, fouling factors, water flow rates, and hence water temperature rises

The effects of variations in fouling factors and water temperature rises may be approximated by:

Fouling factor change from nominal 0.0005 value

$\dfrac{\text{evaporator actual fouling factor}}{0.0005} = N$ Correction: subtract $(2°)(N-1)$ from t_{CH}

$\dfrac{\text{condenser actual fouling factor}}{0.0005} = M$ Correction: add $(2.5\ °F)(M-1)$ to t_{CD}

Change in rise of water temperature from nominal 10°F value	Correction
Chiller temperature rise = 5 °F	Add 0.5°F to t_{CH}
Chiller temperature rise = 20 °F	Subtract 1°F from t_{CH}
Condenser temperature rise = 20 F	Add 0.5 °F to t_{CD}

Power requirement:

a. $P = f(L, t_{CH}, t_{CD},$ actual capacity)

where P = power, kW

L = percent load = (actual capacity/full capacity) x 100

t_{CH} = leaving chilled water temperature, °F

t_{CD} = entering condenser water temperature, °F

actual capacity = actual machine output under the particular operating conditions, tons

b. $P = (c_1 + c_2L + c_3L^2) \times A \times B \times$ (actual capacity)

where $A = c_5 + c_6\ t_{CH}$

$B = c_7 + c_8\ t_{CD} + c_9\ t_{CD}^2$

c. Power expressed in kW (3413 Btu/hr)

Nominal 400-ton unit with fixed components

(same unit as described in the previous capacity section)

Power at maximum rated capacity, kW

t_{CH}, °F \ t_{CD}, °F	80	82.5	85	87.5	90
40	352	355	355	360	362
42	354	356	357	362	364
44	355	358	358	364	362
46	355	358	359	361	364

d. Power at maximum rated capacity

$$P = 0.87821 \times A \times B \times Q$$

where $A = 1.3527 - 0.00833\ t_{CH}$

$B = 2.577616 - 0.050054\ t_{CD} + 0.00036949\ t_{CD}^2$

Q = cooling capacity in tons as shown previously under cooling capacity, section d.

Part-load operation:

The cooling capacity and power requirements described previously are applicable for full-load operation.

a. For part-load operation, the power consumption would be of the form described in the power requirement section.

b. $P = (c_1 + c_2\ L + c_3\ L^2) \times A \times B \times$ (actual capacity)

where $A = c_5 + c_6\ t_{CH}$

$B = c_7 + c_8\ t_{CD} + c_9\ t_{CD}^2$

c. Power expressed in kW (3413 Btu/hr)

Nominal 400-ton unit previously described

$Q_{actual} = (60 \text{ percent})(Q_{full})$

t_{CD}, °F / t_{CH}, °F	80	82.5	85	87.5	90
40	227	227	228	232	233
42	227	229	229	233	234
44	228	230	230	234	233
46	228	230	231	232	234

d. $P = (1.68984 - 0.0190203\ L + 0.00010904\ L^2) \times A \times B \times (\text{actual capacity}$

where $A = 1.3527 - 0.00833\ t_{CH}$

$B = 2.577516 - 0.050054\ t_{CD} + 0.00036949\ t_{CD}^2$

actual capacity, tons

Application:

Application of the method and equations is as follows:

If the power requirement is sought knowing the actual cooling capacity (Q_{actual}), the leaving chilled water temperature (t_{CH}) and the entering condenser water temperature (t_{CD})

Calculate Q_{full} from the equation in the cooling capacity section

$L = Q_{actual}/Q_{full}$ in percent

Calculate values for A and B and then P from the equations in the previous part-load section.

Absorption water chillers

a. $Q = f(t_{ch}, t_{cd}, p_{stm})$

where Q = cooling capacity, tons

t_{ch} = leaving chilled water temperature, °F

t_{cd} = entering condenser water temperature, °F

p_{stm} = supply steam pressure, psig

b. For a given supply steam pressure, the cooling capacity can be represented by an equation of the following form

$$Q = c_1 + c_2 t_{ch} + c_3 t_{ch}^2 + c_4 t_{cd} + c_5 t_{cd}^2 + c_6 t_{ch} t_{cd} + c_7 t_{ch}^2 t_{cd} + c_8 t_{ch} t_{cd}^2 + c_9 t_{ch}^2 t_{cd}^2$$

c. Full-load capacity for a nominal 520-ton unit with the following specifications:

2-pass absorber and 1-pass condenser, 3-pass evaporator, 0.0005 fouling factor for condenser, absorber and evaporator, fixed water flow rate that produces approximately a 10°F rise in condenser water temperature, condenser water flow rate of 1870 gpm, chilled water flow rate of 1200 gpm, and 12 psig steam supply pressure

t_{ch} \ t_{cd}	80	85	90
40	528	468	396
42	551	494	426
44	572	520	457
46	595	546	483

$$Q_{full} = -38064.00 + 1798.00\ t_{ch} - 21.25\ t_{ch}^2 + 858.59\ t_{cd} - 4.84\ t_{cd}^2 - 39.27\ t_{ch} t_{cd} + 0.046\ t_{ch}^2 t_{cd} + 0.22 t_{ch} t - 0.002\ t_{ch}^2\ t_{cd}^2$$

For chilled water flow rates other than the specified 1200 gpm, cooling capacity can be estimated by determining a capacity correction factor as follows:

$$\text{CORR} = 1.075 - 0.877\times10^{-4}\ (\text{flow rate, gpm}) + 0.206\times10^{-7}(\text{flow rate, gpm})^2$$

where

CORR = multiplying factor applicable for flow rates between 600 and 1500 gpm

Steam requirements

(a) $S = f(t_{ch}, t_{cd}, Q_{act}, Q_{nom})$

where S = steam consumption, lb/hr

t_{ch} = leaving chilled water temperature, °F

t_{cd} = entering condenser water temperature, °F

Q_{act} = actual machine output under particular operating conditions, tons

Q_{nom} = nominal machine capacity, tons

(b) $S = (Q_{full})(A)(B)$

where Q_{full} = machine capacity at full load

A = full-load steam rate, lb/(hr)(ton)

$= c_1 + c_2 L_n + c_3 L_n^2$

B = fraction of full-load steam input

$= (c_4 + c_5 L_f + c_6 L_f^2)/100$

$L_n = Q_{full}/Q_{nom}$

$L_f = 100\left(\frac{Q_{act}}{Q_{full}}\right)$

(c) Steam consumption in lb/hr at maximum rated capacity (i.e., $Q_{act} = Q_{full}$ and $L_f = 100$ percent)

Nominal 520-ton unit (same unit as described in the previous capacity section)

t_{ch} °F \ t_{cd} °F	80	85	90
40	9820	8925	7960
42	10138	9336	8392
44	10467	9724	8820
46	10829	10101	9225

(d) $S = (Q_{full})(A)$

where Q_{full} is Q from equation in (c) of previous section

and

$$A = 32.11 - 22.94\, L_n + 9.44\, L_n^{\,2}$$

Part-Load operation

The cooling capacity and power requirements described previously are applicable for full-load operation

(a) for part-load operation, the steam consumption would be expressed by the equations in (b) of the previous section, except that B takes on values other than unity

(b) $S = (Q_{full})(A)(B)$

where Q_{full} = capacity of machine at full load at given t_{ch} and t_{cd} as determined above

A = full load steam rate, lb/(hr)(ton)

$= c_1 + c_2L_n + c_3L_n^{\,2}$

B = fraction of full-load steam input

$= (c_4 + c_5L_n + c_6L_n^{\,2})/100$

If the entering condenser water temperature is held constant during part-load operation, then $B = f(L_f)$. If, however, the entering condenser water temperature is allowed to vary with outdoor conditions, then $B = f(L_f,$ t

(c) Part-load steam consumption in lb/hr

Nominal 520 ton unit at 60 percent of full load, thus L_f = 60 percent.

t_{ch} \ t_{cd}	80	85	90
40	5499	5391	5015
42	5677	5602	5287
44	5862	5834	5557
46	6064	6060	5812

(d) $S = (Q_{full})(A)(B)$

Q_{full} as before

$$A = 32.11 - 22.94\ L_n + 9.44\ L_n^2$$

At t_{cd} = 55 F

$$B = (4.66 + 0.62\ L_f + 0.0013\ L_f^2)/100$$

At t_{cd} = 65 F

$$B = (7.78 + 0.61\ L_f + 0.0015\ L_f^2)/100$$

At t_{cd} = 75 F

$$B = (9.33 + 0.65\ L_f + 0.0017\ L_f^2)/100$$

At t_{cd} = 85 F

$$B = (12.5 + 0.62\ L_f + 0.0025\ L_f^2)/100$$

At t_{cd} = 95 F

$$B = (16.89 + 0.68\ L_f + 0.0028\ L_f^2)/100$$

Application review:

If the steam consumption is sought knowing the actual cooling load, the leaving chilled water temperature, and the entering condenser water temperature,

1. Calculate Q_{full} from equation in cooling capacity section
2. Calculate L_f and L_n
3. Calculate A and B
4. Calculate S from equation in part-load section

5c. Heating coils

Steam coils

The equation sought is one from which the heat transfer rate of the coil can be computed knowing the air flow rate, entering temperature of the air, and steam temperature. The "effectiveness" of the heat exchanger is a useful tool in this assignment.

(a) $Q = f(G, t_{steam}, t_{ent\ air})$

where Q = rate of heat transfer, Btu/hr

t_{steam} = steam temperature, °F

$t_{ent\ air}$ = temperature of entering air, °F

G = rate of air flow, lb/hr

(b) $Q = (G)(c_a)(E)(t_{steam} - t_{ent\ air})$

where c_a = specific heat of air, Btu/(lb)(°F)

E = effectiveness, dimensionless

When one of the fluids remains at a constant temperature, as does the steam in this heat exchanger

$$E = 1 - e^{-(UA/Gc_a)} = 1 - \mathrm{EXP}(-UA/Gc_a)$$

The U-value (overall heat-transfer coefficient, in Btu/(hr)(ft^2)(°F)) is a function of G, and the term UA/Gc_a for a specific steam coil can be represented by an equation

$$UA/Gc_a = \frac{1}{C_1 G + C_2 G^{0.2}}$$

where C_1 and C_2 are constants unique to the particular coil.

(c) A certain steam coil having a face area of 8 ft^2, t_{steam} = 218 and $t_{ent\ air}$ = 40 F has the following performance:

face vel, fpm	G	air temp rise, °F	outlet air temp, °F	E
300	10800	56.2	96.2	0.315
700	252[illegible]0	37.2	77.2	0.209
1200	43200	26.6	66.6	0.150

5c. (cont.)

For convenience, let $G' = G/10{,}000$

The 300 and 1200 fpm face velocity conditions provide two linear equations to determine C_1 and C_2, thus

$$E = 1 - EXP\left(\frac{-1}{0.918G' + 1.63\ G'^{0.2}}\right)$$

Checking this value at the intermediate face velocity

$$E = 1 - EXP\left(\frac{-1}{0.918(2.52) + 1.630(2.52)^{0.2}}\right) = 0.2086$$

Hot water coils

A slightly more general expression for the effectiveness is needed for hot water coils, because the effectiveness is based on the lowest value of the product of the mass flow rate and specific heat--water or air.

$$E = \frac{\text{rate of heat transfer, Btu/hr}}{[(\text{mass rate of flow})(\text{specific heat})]_{min}(t_{ent\ H_2O} - t_{ent\ air})}$$

(c)

a certain hot-water coil has a face area of 8 ft^2, 60 F entering air, and 160 F entering water. The air-temperature rise and effectiveness are shown in the table

Water flow	300 fpm	700 fpm	1100 fpm
28 gpm	47.5 F, 0.475	32.0 F, 0.32	23.6 F, 0.236
56 gpm	50.1 F, 0.501	33.8 F, 0.338	25.0 F, 0.250
84 gpm	51.1 F, 0.511	34.5 F, 0.345	25.5 F, 0.255

In all the conditions expressed in the table the product of the mass flow rate and specific heat of air is less than that of the water, so the lowest wc_p is that of the air.

(d)

A polynomial equation for E based on WF (where WF is the water flow in gpm divided by 10) and AV (where AV is the air velocity divided by 100) may be written in the form

$$E = c_1 + c_2AV + c_3AV^2 + c_4WF + c_5\ WF^2 + c_6(AV)(WF) + c_7(AV)^2(WF) + c_8(AV)(WF)^2 + c_9(AV)^2(WF)^2$$

5c. (Cont.)

and $c_1 = 0.6032$ $c_2 = -0.0622$ $c_3 = 0.00249$ $c_4 = 0.01237$

$c_5 = 0.0000$ $c_6 = 0.001263$ $c_7 = -0.000165$

$c_8 = -0.000289$ $c_9 = 0.000025$

Then $$Q = E(G)(c_a)(t_{ent\ H_2O} - t_{ent\ air})$$

5d. Cooling and dehumidifying coils

The performance characteristics generally desired when modeling a specific cooling and dehumidifying coil is some combination of the sensible heat transfer and the dehumidifying rate. The variables on the air side are:

Q_T = total rate of heat transfer, Btu/hr
Q_S = sensible heat transfer, Btu/hr
Q_L = latent heat transfer, Btu/hr
G = rate of air flow, lb/hr
FPM = face velocity of air, fpm
EDB = dry-bulb temperature of entering air, °F
EWB = wet-bulb temperature of entering air, °F
LDB = dry-bulb temperature of leaving air, °F
LWB = wet-bulb temperature of leaving air, °F
h = enthalpy of air, Btu/lb
W = humidity ratio, lb/lb

For a direct expansion coil the variables are:

t_R = saturation temperature of evaporating refrigerant, °F
FA = face area, ft^2
NR = number of rows deep
FPI = fins per inch

The equations that apply are over the normal air conditioning range:

$$Q_T = G(h_L - h_E)$$
$$Q_S = G(0.245)(LDB - EDB)$$
$$Q_L = G(\Delta W)(1061)$$

and for a standard density

$$Q_T = 4.5(CFM)(h_E - h_L)$$

where h_L = enthalpy of leaving air, Btu/lb
h_E = enthalpy of entering air, Btu/lb
CFM = volume rate of flow, cfm

also

$$Q_S = 1.10(CFM)(EDB - LDB)$$
$$Q_L = 4840(CFM)(W_E - W_L)$$

where W_E = humidity ratio of entering air, lb/lb
W_L = humidity ratio of leaving air, lb/lb

Generally the data are presented as total heat transfer per unit area of the face of the coil. For a given coil the performance is

$$Q_T / FA = f(t_R, EWB, FPM)$$

The base rating for a given coil and face velocity, applicable to sensible heat ratios of 0.9 and less and for wetted surfaces only, BRDX

$$BRDX = c_1 + c_2(TR) + c_3(EWB) + c_4(TR)(EWB) + c_5(TR)^2 + c_6(EWB)^2$$

where

$c_1 = 0.11586173\ E{+}01$ $\quad c_4 = -0.57119459\ E{-}03$

$c_2 = -0.97877806\ E{-}02$ $\quad c_5 = -0.52896887\ E{-}04$

$c_3 = -0.26227126\ E{-}01$ $\quad c_6 = 0.91074735\ E{-}01$

The constants are applicable to the following performance data

Base rating of a Direct Expansion Coil
Tons/(Sq. Ft. of Face Area)

EWB, °F	Saturated Refrigerant Temperature, °F						
	30	35	40	45	50	55	60
57	1.31	1.07	0.83				
59	1.43	1.20	0.95				
61	1.56	1.32	1.08	0.83			
63	1.70	1.46	1.20	0.95			
65	1.85	1.60	1.34	1.08	0.83		
67	1.99	1.74	1.48	1.22	0.97		
69	2.18	1.90	1.64	1.37	1.11	0.83	
71	2.34	2.05	1.79	1.52	1.24	0.96	
73	2.51	2.22	1.94	1.67	1.40	1.11	0.81
75	2.70	2.40	2.12	1.84	1.55	1.26	0.96
77		2.58	2.30	2.01	1.72	1.42	1.11
79		2.77	2.49	2.19	1.89	1.59	1.28
81			2.68	2.38	2.07	1.77	1.46
83				2.57	2.26	1.94	1.62
85				2.80	2.46	2.15	1.83

The equation was tested for a best fit against the above data and it approximated the data with less than 1 percent maximum differen for 70 points of the 75 points. The other five points fell in the range between 1 and 2 percent difference from the published data.

A correction factor may be applied to the base rating to correct for the face velocity,

$$CF = c_1 + c_2(FPM) + c_3(FPM)^2$$

For a 3-row coil, the following corrections apply to a certain coil,

Face velocity, fpm	Correction Factor
200	0.51
250	0.57
300	0.62
350	0.68
400	0.73
450	0.77
500	0.81
550	0.85
600	0.88
650	0.91
700	0.93

Coefficients that fit the above data are

c_1 = 0.22399534 E+00

c_2 = 0.15857343 E-02

c_3 = -0.82051282 E-06

In the method presented some manufacturers of coils, after the total heat transfer is obtained, the dehumidification and the leaving conditions are obtained by using a wet-bulb depression ratio.

$$WBDF = \frac{LDB - LWB}{EDB - EWB}$$

where WBDF = f(FPM) for a given coil

For a certain coil

FPM	WBDF
200	0.070
300	0.108
400	0.138
500	0.164
600	0.187
700	0.207

$$WBDF = c_1 + c_2(FPM) + c_3(FPM)^2$$

where c_1 = -0.12742857 E-01

c_2 = 0.4605000 E-03

c_3 = -0.21071429 E-06

The equation fits the data to less than a 1.5 percent difference

Chilled Water Coils

For chilled water coils the water-side equations are as follows:

$$Q_T = M\, c_p(LW - EW)$$

where M = mass rate of flow of water, lb/hr
c_p = specific heat of water, Btu/(lb)(°F)
LW = leaving water temperature, °F
EW = entering water temperature, °F

For most chilledwater applications this equation can be simplified to

$$Q_T = 500(GPM)(LW - EW)$$

The heat transfer in this method is described as follows:

$$Q_T = (\text{Base Rating})(\text{Wetted Surface Factor})(\text{Log Mean Temperature Difference})$$

$$BRCW = f(FPS, FPM)$$

where FPS = water-side velocity, fps
FPM = air-side face velocity, fpm

The water velocity is determined by the water flow rate in gpm, the number of parallel circuits, and the inside diameter of the tubes

$$FPS = \frac{(GPM)(7.48\ ft^3/gal)}{(60\ min/hr)\ (0.7854)\ (ID)^2(NC)}$$

where ID = inside diamter, ft
NC = number of parallel circuits

The base rating is expressed in Btu/hr per square foot of face area per row of tubes deep per degree log mean temperature difference. The form suggested is

$$\frac{1}{BRCW} = c_1 + \frac{c_2}{FPM} + \frac{c_3}{FPS} + \frac{c_4}{(FPS)^2} + \frac{c_5}{(FPM)^3} + \frac{c_6}{(FPS)^2(FPM)^4}$$

where $BRCW$ = base rating

The following table of performance data has been fitted to the foregoing equation.

Base Rating for Chilled Water Coils

BRCW, Btu/hr per ft^2 of face area per degree LMTD								
Face velocity fpm	Water Velocity, fps							
	1.0	1.5	2.0	2.5	3.0	4.0	6.0	8.0
200		102.3	106.1	109.1	111.6	115.0	118.1	119.3
250	100.5	110.0	115.1	118.9	121.9	126.0	130.1	132.4
300	107.0	117.7	123.9	128.2	131.8	136.4	141.6	144.6
350	113.0	125.0	132.0	137.0	141.0	147.4	152.5	156.2
400	118.8	131.8	139.7	145.3	149.8	156.0	162.9	167.1
450	124.0	138.2	146.9	153.0	158.0	164.9	172.6	177.4
500	128.8	144.1	153.4	160.1	165.5	173.1	181.5	187.0
550	133.0	149.3	159.4	166.9	172.4	180.7	189.9	195.5
600	136.7	154.0	164.9	172.7	178.8	187.9	197.6	
650	139.8	158.0	170.0	178.2	184.4	194.3		
700	142.5	161.2	174.5	183.2	189.8	200.0		
750	144.2	163.8	178.2	187.6	194.3			
800	145.5	165.8	180.8	191.0	198.0			

The coefficients in the equation are

c_1 = 0.22074304 E+01 $\quad$ c_4 = -0.55238537 E+00

c_2 = 0.14178455 E+04 $\quad$ c_5 = -0.11071271 E+08

c_3 = 0.33925943 E+01 $\quad$ c_6 = -0.39288579 E+09

The unusual form of the equation was obtained by evaluating the effect of all terms out of approximately 30 terms in a polynomial for the most efficient form. The six terms chosen fit the data with all points except one showing less than one percent difference and 83 of the 92 points showing less than 1/2 of one percent difference between the tabular data points and that computed by the equation. Qualitative influences of the terms in the equation can be explained from basic heat-transfer relationships.

The wetted-surface factor can be expressed by the equation

$$\begin{aligned} WSF = {} & c_1 + c_2(PW) + c_3(PW)(BW) + c_4(PW)^2 + c_5(BW)^2(PW) \\ & + c_6(BW)(PW)^2 + c_7(BW)^3(PW) + c_8(BW)^3(PW)^2 + c_9(BW)^3(PW)^3 \end{aligned}$$

where PW = EDP - EWT

BW = EDB - EWT

The data from the following table:

Wetted Surface Factors

EDB - EWT	Entering dewpoint - entering water temperature									
	0	2.5	5.0	7.5	10.0	12.5	15.0	20.0	25.0	30.0
10	1.000	1.065	1.151	1.271	1.370					
20	1.000	1.038	1.085	1.147	1.225	1.330				
30	1.000	1.022	1.052	1.092	1.141	1.200	1.276			
40	1.000		1.033	1.057	1.087	1.123	1.174	1.309		
50	1.000		1.022		1.058		1.110	1.200	1.328	
60	1.000		1.012		1.038		1.070	1.122	1.207	1.330

Applicable coefficients are:

c_1 = 0.10007596 E +01

c_2 = 0.42652727 E-01

c_3 = -0.22742528 E-02

c_4 = 0.16232278 E-02

c_5 = 0.41384645 E-04

c_6 = -0.24564063 E-04

c_7 = -0.24092761 E-06

c_8 = -0.11066765 E-08

c_9 = 0.54302870 E-10

The balanced conditions for chilled water coils with this method requires an iteration process. One procedure that has been found effective is as follows:

Conditions given: entering water temperature
water flow rate, gpm
entering air conditions, EDB, EWB, CFM
coil data

Assume: two temperatures for the water leaving the coil, LW1 and LW2

Calculate for each LW

QTW1 = (500)(GPM)(LW1 - EW)

Enthalpy of leaving air $HL1 = HE - \frac{QT1}{4.5(CFM)}$

LWB1 = f(HL1) from psychrometric routine

Assume leaving relative humidity is 90 percent

LDB1 = f(LWB1, RH)

MLTD1 = (D1 - D2)/ln(D1/D2)

where $D1 = EDB - LW1$

$D2 = LDB1 - EW$

$$QTC1 = (BRCW)(NR)(FA)(WSF)(MLTD1)$$

$$DIFF1 = QTW1 - QTC1$$

$$DIFF2 = QTW2 - QTC2$$

A linear interpolation is then used to obtain a closer approximation to the leaving water temperature that yields no difference between QTW and QTC.

When chilled water performance data is published in some other form, such as with entering and leaving air conditions, it may be advisable to use the effectiveness concept for sensible and latent cooling capacity.

5e. Reciprocating compressors

Refrigerating capacity:

a. $Q = f(t_e, t_c)$

where Q = refrigerating capacity, Btu/hr
t_e = evaporating temperature, °F
t_c = condensing temperature, °F

b. $$Q = c_1 + c_2 t_e + c_3 t_e^2 + c_4 t_c + c_5 t_c^2 + c_6 t_e t_c + c_7 t_e^2 t_c + c_8 t_e t_c^2 + c_9 t_e^2 t_c^2$$

c.

	t_e = 30 F	t_e = 40 F	t_e = 50 F
t_c = 80 F	Q = 480,000	588,000	709,000
100 F	416,000	512,000	625,000
120 F	350,000	435,000	540,000

d. $$Q = 140{,}000 + 21{,}150\ t_e - 65\ t_e^2 + 2275\ t_c - 13.75\ t_c^2 - 231\ t_e t_c + 2.125\ t_e^2 t_c + 0.5625\ t_e t_c^2 - 0.00625\ t_e^2 t_c^2$$

Power requirement:

a. $P = f(t_e, t_c)$

where P = power, Btu/hr
t_e = evaporating temperature, °F
t_c = condensing temperature, °F

b. $$P = c_1 + c_2 t_e + c_3 t_e^2 + c_4 t_c + c_5 t_c^2 + c_6 t_e t_c + c_7 t_e^2 t_c + c_8 t_e t_c^2 + c_9 t_e^2 t_c^2$$

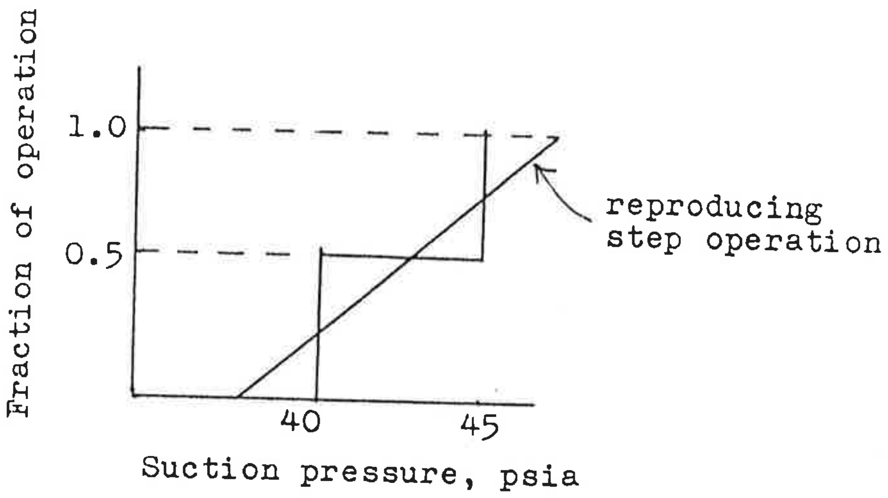
c.

	t_e = 30 F	t_e = 40 F	t_e = 50 F
t_c = 80 F	P = 75,000	75,700	74,500
t_c = 100 F	88,500	92,500	94,500
t_c + 120 F	101,000	109,000	115,000

d. $$P = 25{,}500 - 375\, t_e - 2.5\, t_e^2 + 712.5\, t_c - 3.125\, t_c^2 + 10.375\, t_e t_c - 0.1375\, t_e^2 t_c + 0.04375\, t_e t_c^2 + 0.000625\, t_e^2 t_c^2$$

Unloading of cylinders for capacity control

Many reciprocating compressors employ a capacity control whereby the suction pressure or the leaving water temperature, in the case of a package chiller, controls the unloading. If, for example, a four-cylinder compressor unloads two cylinders when the suction pressure drops to 50 psia and the compressor shuts off when the pressure drops to 45 psia, this step operation could be reproduced by a straight line as shown in the graph:



The ordinate is the fraction of full operation. The capacity and power calculated in d above are multiplied by this fraction.

On the four-cylinder compressor in question, a fraction of operation of 0.8 superficially would mean 3.2 cylinders in operation. Physically, it is possible for only 0, 2, or 4 cylinders to be in service, so the 0.8 operating fraction indicates that the compressor is cycling between two and four cylinders in operation--four cylinders 60 percent of the time and 2 cylinders 40 percent of the time.

5f. Water-cooled and air-cooled condensers

Water-cooled

a. $Q = f(\Delta t, \text{gpm})$

where Q = heat rejection rate, Btu/hr

Δt = (condensing temperature) - (outlet water temperature)

gpm = water flow rate, gpm

b. $Q = (\Delta t)\left[a_o + a_1(\text{gpm}) + a_2(\text{gpm})^2\right]$

c.

gpm	Q/Δt	
	clean tubes	0.001 scale factor
40	18	15
160	63	51
280	97	67

d. for clean tubes

$$Q = \Delta t\left[0.55557 + 0.4514(\text{gpm}) - 0.000382(\text{gpm})^2\right]$$

for 0.001 scale factor

$$Q = \Delta t\left[-1.444 + 0.43889(\text{gpm}) - 0.000694(\text{gpm})^2\right]$$

In certain cases it may be more convenient to express performance as a function of inlet water temperature rather than outlet temperature. Since

$$t_o = t_i + Q/(\text{gpm})(500)$$

$$\Delta t = (\text{condensing temperature}) - (t_i + Q/(\text{gpm})(500)$$

then

$$Q = \frac{(t_c - t_i)\left[a_o + a_1(gpm) + a_2(gpm)^2\right]}{1 - \dfrac{a_o + a_1(gpm) + a_2(gpm)^2}{500\ (gpm)}}$$

Air cooled

a. $Q = f(t_a, t_c)$

where Q = heat-rejection rate, Btu/hr

t_a = ambient air temperature, °F

t_c = condensing temperature, °F

b. $Q = C(t_c - t_a)$

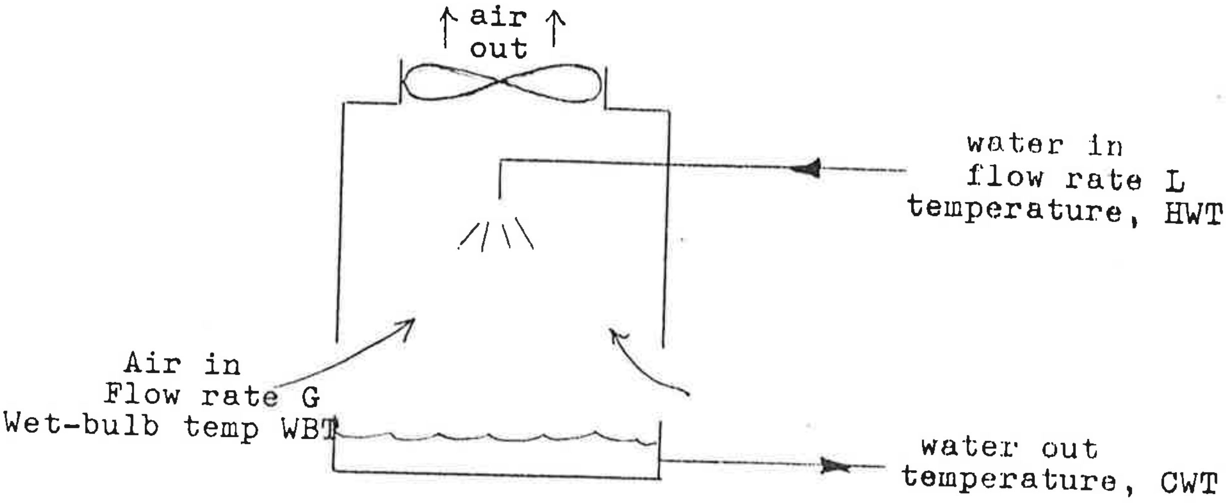
c. For a 10-degree F temperature difference,

$$Q = 149{,}000 \text{ Btu/hr}$$

d. $Q = (14{,}900)(t_c - t_a)$

5g. Cooling Towers

Given: A cooling tower with constant air flow rate G and constant water flow rate L



a. CWT = f(WBT, HWT)

where CWT = leaving cold water temperature, °F

WBT = entering air wet-bulb temperature, °F

HWT = entering hot water temperature, °F

b. $$CWT = c_1 + c_2(WBT) + c_3(WBT)^2 + c_4(HWT) + c_5(HWT)^2 + c_6(WBT)(HWT) + c_7(WBT)^2(HWT) + c_8(WBT)(HWT)^2 + c_9(WBT)^2(HWT)^2$$

c. Leaving cold water temperature, CWT

HWT, °F \ WBT, °F	66	72	78
100	78.0	81.0	84.2
115	79.5	84.0	88.0
130	80.6	86.5	91.3

d. $$CWT = 652.7856 - 15.819(WBT) + 0.103092(WBT)^2 - 7.568(HWT) + 0.016226(HWT)^2 + 0.205667(WBT)(HWT) - 0.001312(WBT)^2(HWT) - 0.000463(WBT)(HWT)^2 + 0.0000031(WBT)^2(HWT)^2$$

5h. Water-chilling evaporators

a. $Q = f(\Delta t, \text{gpm})$

where Q = heat transfer rate, Btu/hr

Δt = (temperature of leaving water) - (evaporating temperature)

gpm = water flow rate, gpm

b. $$Q = c_1 + c_2(\text{gpm}) + c_3(\text{gpm})^2 + c_4(\Delta t) + c_5(\Delta t)^2 + c_6(\text{gpm})(\Delta t) + c_7(\text{gpm})^2(\Delta t) + c_8(\text{gpm})(\Delta t)^2 + c_9(\text{gpm})^2(\Delta t)^2$$

c. Q in thousands of Btu/hr

	Δt, °F				
gpm	6	7	8	9	10
100	420	720	940	1180	1430
200	620	950	1300		
300	760	1170	1580		
400	870	1350			
500	950	1500			

d. $c_1 = -2449.$
$c_2 = 14.64$
$c_3 = -0.03921$
$c_4 = 659.3$
$c_5 = -35.91$
$c_6 = -4.411$
$c_7 = 0.01158$
$c_8 = 0.3940$
$c_9 = -0.0008896$

5i. Pumps

For a constant speed, the expression of head as a function of flow is

a. $H = f(Q)$

where H = head, ft of water

Q = flow, gpm

b. $H = a_o + a_1 Q + a_2 Q^2$

c.

H	Q
104.5	40
95.1	160
75.0	240

c. $H = 102.1 + 0.0946\ Q - 0.000865\ Q^2$

For a constant speed, power as a funxtion of flow

a. $SHP = f(Q)$

where SHP = shaft power, hp

b. $SHP = a_o + a_1 Q + a_2 Q^2$

c.

SHP	Q
2.55	40
4.90	160
5.90	240

d. $SHP = 1.54 + 0.0267\ Q - 0.000354\ Q^2$

5j. Fans

Flow rate as a function of fan speed and static pressure

a. $Q = f(H, N)$

where Q = air-flow rate, cfm

H = static pressure, inches of water

N = rotative speed, rpm

b. $$Q = c_1 + c_2H + c_3H^2 + c_4N + c_5N^2 + c_6HN + c_7H^2N + c_8HN^2 + c_9H^2N^2$$

c.

Q	H	N
12,460	0.25	189
12,460	1.00	315
12,460	1.50	407
26,990	0.25	292
26,990	1.00	387
26,990	1.50	455
41,530	0.25	416
41,530	1.00	486
41,530	1.50	526

d.

c_1 = -46965.
c_2 = 158290.
c_3 = -201605.
c_4 = 283.92
c_5 = -0.1757
c_6 = -540.54
c_7 = 683.986
c_8 = 0.42619
c_9 = -0.57177

Power as a function of flow rate and static pressure

a. $P = f(Q, H)$

where P = power, hp

b. $$P = c_1 + c_2Q + c_3Q^2 + c_4H + c_5H^2 + c_6QH + c_7Q^2H + c_8QH^2 + c_9Q^2H^2$$

5j. Fans (Cont'd)

c.

P	Q/100,000	H
2.34	0.2699	0.25
3.35	0.2699	0.50
5.44	0.2699	1.00
4.37	0.3531	0.25
5.63	0.3531	0.50
8.24	0.3531	1.00
6.86	0.4153	0.25
8.33	0.4153	0.50
11.32	0.4153	1.00

d.

$c_1 = 5.314327$

$c_2 = -42.31355$

$c_3 = 102.3999$

$c_4 = 2.935346$

$c_5 = -1.506405$

$c_6 = -2.515906$

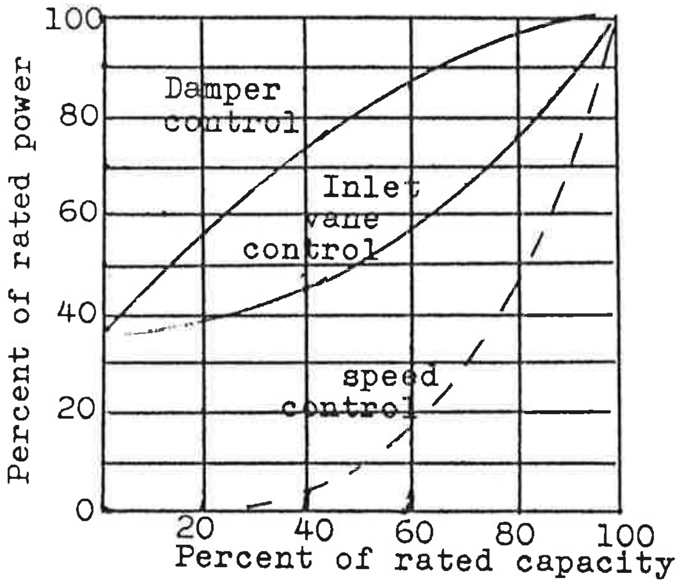
$c_7 = 22.56411$

$c_8 = 10.57784$

$c_9 = -15.94999$

Part-load operation of fans

Typical power requirements at part-load operation for three different methods of capacity control are as shown in the figure below. The "percent of rated horsepower" can be represented as a second or third-degree function of the "percent of rated capacity."



5k. Condensing units

Refrigerating capacity as a function of evaporating and ambient temperatures for an air-cooled unit

a. $Q = f(t_e, t_a)$

where Q = refrigerating capacity, Btu/hr

t_e = evaporating temperature, °F

t_a = ambient temperature, °F

b. $$Q = c_1 + c_2 t_e + c_3 t_e^2 + c_4 t_a + c_5 t_a^2 + c_6 t_e t_a + c_7 t_e^2 t_a + c_8 t_e t_a^2 + c_9 t_e^2 t_a^2$$

c.

t_a, °F \ t_e, °F	40	45	50
75	Q = 65,500	73,500	85,500
95	53,000	60,000	67,700
105	48,200	55,000	62,000

d.

c_1 = 3,504,500
c_2 = -155,850
c_3 = 1810.
c_4 = -73,336.
c_5 = 380.
c_6 = 3295.
c_7 = -37.73
c_8 = -17.156
c_9 = 0.19555

5k. Condensing units (Cont'd)

Power requirement as a function of evaporating and ambient temperatures for an air-cooled unit

a. $PF = f(t_e, t_a)$

where PF = performance factor, Btu/kWh

b. $PF = c_1 + c_2 t_e + c_3 t_e^2 + c_4 t_a + c_5 t_a^2 + c_6 t_e t_a + c_7 t_e^2 t_a + c_8 t_e t_a^2 + c_9 t_e^2 t_a^2$

c.

t_e, °F / t_a, °F	40	45	50
75	PF = 10,100	11,100	12,400
90	7,900	8,600	9,600
105	6,600	7,100	7,700

d.

c_1 = -94,600.
c_2 = 5390.
c_3 = -54.0
c_4 = 2616.67
c_5 = -15.78
c_6 = -136.0
c_7 = 1.467
c_8 = 0.8000
c_9 = -0.008888

5-1. Engines

Fuel consumption

a. $FCR = f(PL)$

where FCR = fuel consumption rate, lb/hr

PL = percent of full load

b. $FCR = a_1 + a_2(PL/100)$

c.

PL	FCR
100	320
50	180

d. $FCR = 40 + 280(PL/100)$

Recoverable heat rate

a. $HRR = f(PL)$

where HRR = heat recovery rate, Btu/hr

b. $HRR = a_1 + a_2 PL + a_3 (PL)^2$

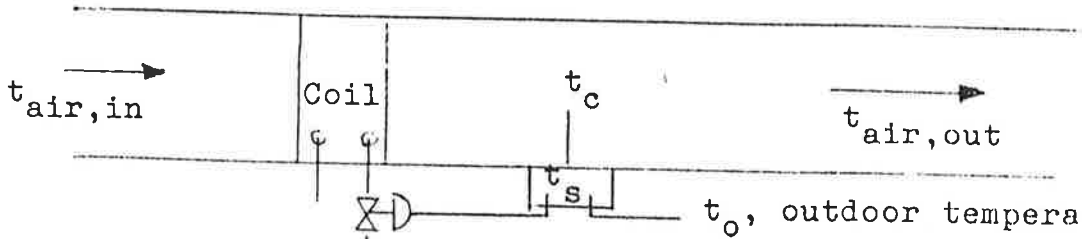
c.

HRR mBtu/hr	PL percent
1500	20
3000	60
4200	100

d. $HRR = 637.5 + 45\ PL - 0.0937\ PL^2$

5m. Controls

I. Water coil, discharge temperature controlling throttling valve



a. $t_c = f(t_s, R, \Delta t_o, x, x_m, TR)$

where t_c = discharge temperature, °F

t_s = set point temperature, °F

R = outdoor reset ratio

Δt_o = change of outdoor temperature from reference temperature

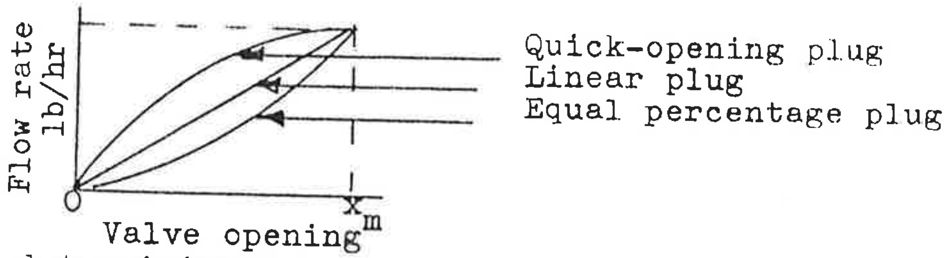
x = valve opening, inch

x_m = maximum valve opening, inch

TR = throttling range

b. $t_c = t_s + R(\Delta t_o) - (\frac{x}{x_m})\ TR$

c. With a constant supply and return water pressure, the flow rate through the valve, depending upon the type of plug, is shown in the sketch



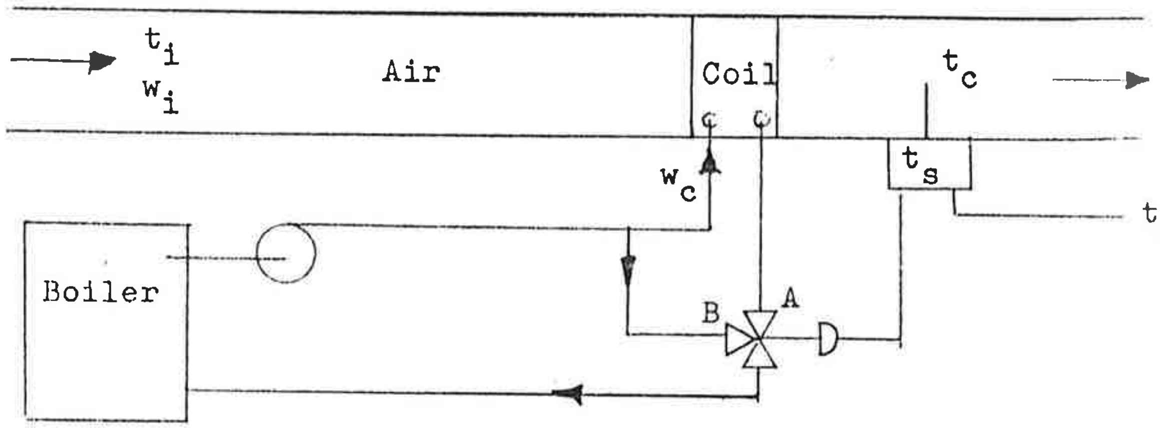
d. Procedure in determining t_c:

1. Determine flow rate from coil routing,
2. From appropriate graph in above sketch, determine x,
3. Substitute into equation to calculate t_c.

If changes in flow rate result in appreciable differences in supply pressure (indicating a revised valve flow graph), one or more additional iterations may be needed.

5m. Controls (Cont'd)

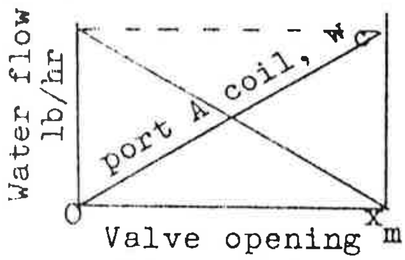
II. Water coil, discharge temperature controlling mixing valve



a. $w_c = f(t_c,\ t_i,\ t_w,\ \text{and}\ w_i)$ from coil routine (Eq. 1)

where w_c = water flow rate to coil, lb/hr
w_i = air flow rate, lb/hr
t_i = entering air temperature
t_w = entering water temperature

b. Valve characteristic, assuming linear plug and a constant supply and return pressure as shown in sketch:



$$w_c = a_o + a_1\left(\frac{x}{x_m}\right) \quad \text{(Eq. 2)}$$

c. $t_c = f(t_s,\ R,\ \Delta t_o,\ x,\ x_m,\ TR)$ (Eq. 3)

d. All quantities are known except t_c, w_c and x.

To solve:
1. Assume t_c and solve for w_c in Eq. 1.
2. Solve for x/x_m in Eq. 2.
3. Solve for t_c in Eq. 3.

Continue iteration until convergence

If supply pressure changes, Eq. 2 must be revised after each iteration.

5n. Heat transfer to pipes and ducts

a. $t_{fl} + f(t_{fe}, t_s, U, P, L, M, c_p)$

where t_{fl} = leaving temperature of fluid, °F

t_{fe} = entering temperature of fluid, °F

t_s = ambient temperature

U = overall heat transfer coefficient through duct or pipe, including insulation, Btu/(hr)(ft^2)(°F)

P = perimeter, ft

L = length, ft

M = mass flow rate of fluid, lb/hr

c_p = specific heat of fluid, Btu/(lb)(°F)

b. $t_{fl} = (t_{fe} - t_s)\, e^{-(UPL/Mc_p)} + t_s$

c. If t_{fe} = 120 F; t_s = 70 F; U = 1.5: P = 3 ft, L = 40 ft; M = 9,000 lb/hr; c_p = 0.24

d. $t_{fl} = (120 - 70)\, e^{-(240/2160)} + 70$

t_{fl} = 114.7 F

5-o. Part load performance of unitary air conditioners (3 to 5 ton)

a. $PF = f(t_a)$ for a given temperature of the conditioned space

where PF = performance factor, Btu/kWh

t_a = ambient temperature

b. $PF = c_1 + c_2 t_a + c_3 t_a^2$

c.

t_a, °F	PF, Btu/kWh
80	9,400
95	7,600
110	6,200

d. $PF = 32510 - 420.8\ t_a + 1.661\ t_a^2$

PART II. SYSTEM SIMULATION

6. REFINED VS. STREAMLINED PROCEDURES FOR SIMULATING SYSTEMS

6-1. Introduction. The system simulation section of the energy computation program receives information about the thermal loads (heating and cooling, sensible and latent) of each of the individual spaces in the building and converts this information to energy requirements of the equipment. For most buildings it is imperative that the loads on the individual spaces be used rather than a net load on the entire building, because most systems are sensitive not only to the net load, but also to how that load is distributed. The frequency of the computation is normally hourly for whatever days, months, or years that represent typical periods for energy calculations.

The ultimate test of a system simulation program is how well the calculation matches the way the equipment performs. A faithful system simulation, then, abides by all the rules that the equipment itself observes: mass balances, energy balances, equations of state of working substances, and performance characteristics of individual components or subsystems. Such a description of the task of system simulation makes the simulation assignment appear straightforward, but in actual systems there are usually supplementary processes in operation in addition to the basic ones, for example, heat transfer to or from ducts and pipes, that frustrate attempts to achieve a simple simulation. Furthermore, the operating characteristics of a component shown in the manufacturer's catalog may not prevail for the component when it is installed in a system because of the influence of the interconnecting elements. Examples of such influences are the alterations of performance characteristics of fans, heating or cooling coils, or temperature sensing elements of a control due to non-uniform or non-ideal air-flow profiles.

Despite the complications and the realization that the energy computation will be subject to some errors, a program must be developed that is not prohibitive in demands on the time of the user or the computer. The task of the program developer is to incorporate the mechanisms that have greatest influence on the energy requirements and omit those of lesser importance.

6-2. Departures from previous editions. The first two editions of this system simulation booklet treated in considerable detail one particular system (a four-zone dual-duct system whose coils were supplied with chilled and hot water, respectively). The instructions were provided with equations and decisions in FORTRAN-type form so that it was a short step from the material in the booklet to an operating program. The format of the presentation in this current booklet is quite different for several reasons. Computer programming capabilities in the air conditioning industry have so advanced during the past five years that the FORTRAN form of presentation is superfluous, and indeed skilled programmers might devise methods of translating the recommended logic into a program that is superior to one proposed by the authors of this booklet.

One reaction of the users of the earlier editions of the system simulation booklet was that they would prefer some guidance on a wider variety of systems in addition to the one chosen. A further reaction of users was that they would like recommendations for streamlining energy calculations in order to make them more practical during the design stage. The initial objective of the ASHRAE Task Group on Energy Requirements was to develop the most accurate procedures possible and leave to the judgment of the practitioner how to revise and simplify. Responses from the field continued to ask for recommendations for simplifications,and these requests have influenced the presentation in this booklet.

The form of the system simulation section of this edition, then, covers a number of systems encountered in current practice, discusses in words and equations rather than in programming logic the simulation procedures, and suggests modifications of refined simulation procedures that reduce programming effort and computer running time with only a modest sacrifice in precision.

6-3. The accuracy-complexity compromise. One of the major reasons for complexity and long running time in system simulation programs is the need to iterate during calculations because of the existence of simultaneous equations. The ability to proceed through a series of calculations in sequence requires much less computer time than if it is necessary to iterate through calculation loops until convergence is achieved. On the other hand, a precise representation of many air conditioning systems necessarily incorporates simultaneous equations. The ASHRAE Task Group on Energy Requirements sponsored a study* of simplification possibilities and a summary of the conclusions of that study follows in the next several sections. The topics include: (1) treatment of part-load operation of a vapor-compression water chiller, (2) incorporating the effect of varying the condensing temperature on a chiller, (3) simulating of wild cooling coils, (4) effect of throttling range of controls in dual-duct and terminal reheat systems, and (5) simplification of humidity calculations.

6-4. Part-load operation of centrifugal water chillers. Two types of control valves are in common use for regulating the flow of water to cooling coils--the three-way valve as shown in Fig. 6-1a and a two-way valve shown in Fig. 6-1b. The three-way valve maintains an approximately constant flow rate of water through the chiller, and as the refrigeration load decreases, the temperature of the return water drops. With the two-way valve in Fig. 6-1b, reduction in refrigeration load reduces the flow rate of water. The behavior of the return water temperature is a function of the heat-transfer characteristics of the coil and control.

The fortunate behavior of the two systems in Fig. 6-1 is that

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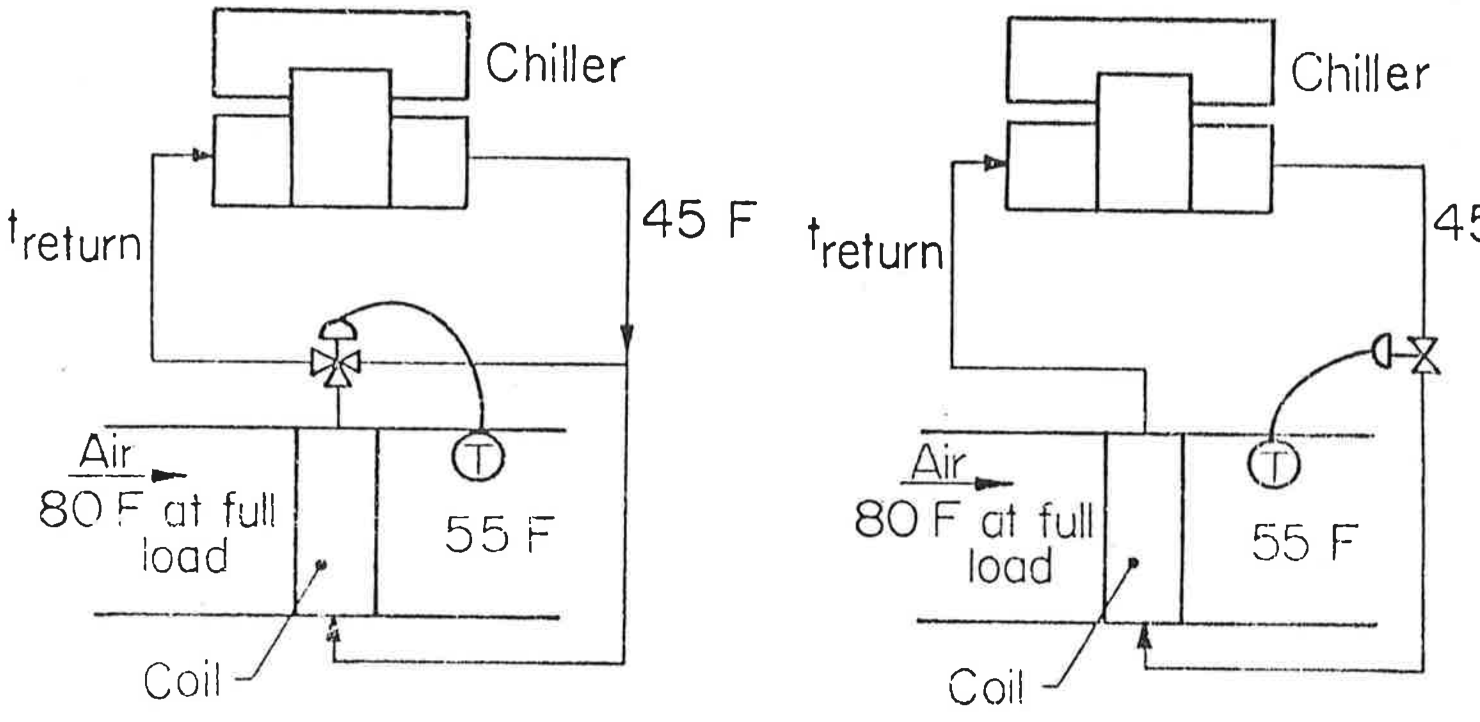
* Stoecker, W. F., R. P. Fenske, C. C. Wilkin, "The Accuracy-Complexity Compromise in Simulating Systems for Energy Calculations," ASHRAE Trans., v. 79, part 2, pp. 152-158, 1973.



(a) three-way valve (b) two-way valve

Fig. 6-1. Two types of capacity control.

the energy per unit refrigeration (kw/ton) is approximately identical in the two modes of control. The part-load computation described in Section 5b of this booklet is applicable, then, to both systems. For a given inlet condenser water temperature the kw/ton is dependent upon both the outlet chilled water temperature (which in many systems remains constant) and the fraction of full load as computed according to the form illustrated in Section 5b.

6-5. Adjustment of kw/ton with a change in condensing water temperature. The kw/ton of a centrifugal compressor water chiller increases with an increase in condensing temperature.

In the water chiller for which data are shown in Section 5b, the kw/ton at a 40 F leaving chilled water temperature increases from 0.838 kw/ton with an 80 F temperature of entering condenser water to 0.95 kw/ton with a 90 F entering temperature. It is clear that changes in the level of temperature of the condenser cooling water should be considered, but an accurate calculation of the subsystem shown in Fig. 6-2 requires an iterative loop that alternates between

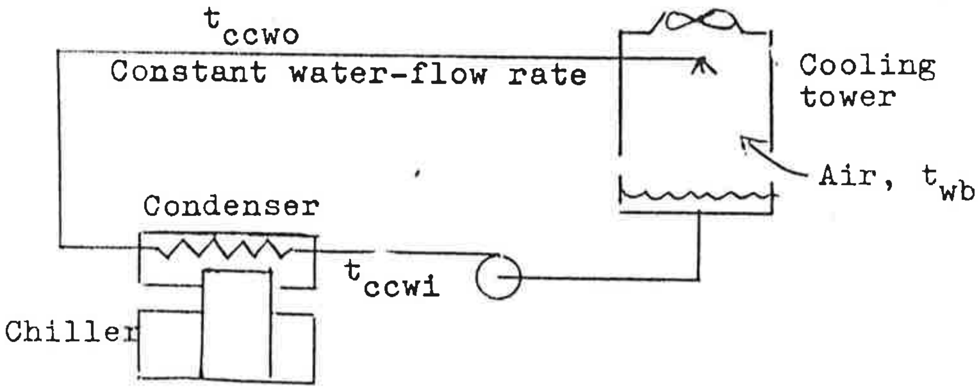


Fig. 6-2. Condenser and cooling tower subsystem.

the cooling tower performance of Section 5g and the centrifugal chiller, Section 5b. The cooling tower characteristics indicate that the entering temperature of condenser cooling water, t_{ccwi}, is a function of the leaving temperature of the cooling water, t_{ccwo}, for a given wet-bulb temperature t_{wb}. Within the condenser, however, the heat rejection rate controls the difference between t_{ccwo} and t_{ccwi}, and this rejection rate is a mild function of t_{ccwi}. As the wet-bulb temperature decreases t_{ccwi} drops, even with a constant condenser heat load, but very likely the air-conditioning load is also dropping with a reduction in the wet-bulb temperature. Unless a control system exists that maintain t_{ccwi} constant (a practice that requires more chiller energy than necessary) the kw/ton should be reduced as the wet-bulb temperature drops. A crude rule of thumb is that the temperature of the condenser cooling water drops 1 °F for each 1 °F drop in wet-bulb temperature. It is probably preferable to use such an approximation in comparison to assuming t_{ccwi} remains constant.

6-6. Wild cooling coil. Some designers choose to operate the cooling coil of a dual-duct system with full flow of chilled water at a constant supply temperature. The rationale for this decision is that during mild weather when high humidity might be a problem, the cooling coil delivers low-temperature air with a large ratio of latent-to-total heat removal. An appreciable error can result in the calculated cooling capacity of the coil if a constant outlet air temperature is assumed, rather than computing the actual outlet air conditions. In one case explored, at an air flow rate 50 percent of design, the actual cooling capacity of the coil was of the order of 20 percent higher than if a controller had regulated the outlet temperature at a constant value.

The procedures in Section 5d for a cooling and dehumidifying coil are recommended.

6-7. Throttling range of controls in dual-duct and terminal reheat systems. A deviation of the controlled variable from the set point is a fundamental requirement for motivating a controller. The temperature of a fluid being heated, for example, must progressively drop in order for the thermostat and heat controller to provide a higher rate of heat flow. The difference in temperatur of the controlled fluid between the zero load and full load conditio is called the "Throttling range." Throttling ranges in space thermostats may be of the order of 1 to 2 °F, but a 5 °F throttling range may be typical in duct air temperature controllers.

The throttling range has special influence in the dual-duct (or multizone) and the terminal reheat systems. In the dual-duct system the air temperature in the hot duct at part load rises higher than its value at full load requiring more heating energy and an associated increase in cooling energy over the amounts that would be required if the hot-duct air temperature remained constant. A corresponding effect occurs at part-load operation of the cooling coil.

In the terminal reheat system, the throttling range in the controller of the cooling coil causes a reduction in temperature of cool air that flows to the reheat coils. The lower-than-necessary temperature of the cool air causes additional cooling energy requirements and corresponding increases in reheat coil energy.

Neglecting the influence of throttling ranges may result in underestimates of both heating and cooling energy of the order of 5 to 15 percent at part loads, but generally very little error occurs near full-load operation. A highly-refined calculation requires introduction of the control characteristics in a manner similar to that explained in Section 5m. Such a procedure is somewhat costly in computer time, so Chapter 7 proposes a compromise procedure that acknowledges throttling ranges, but in an approximate manner.

6-8. Avoiding iterations for space humidities. To compute the latent heat removal at a cooling coil (Section 5d) the humidity ratio of the air entering the coil must be known, but this variable is dependent upon the humidities in the conditioned spaces and in the return air. Calculation of the latent heat removal by the cooling coil can correctly be performed by iterating around the air circuit, but iterations are to be avoided, if possible.

We propose the following procedure that is illustrated by the regions on the psychrometric chart, Fig. 6-3. Of interest is the

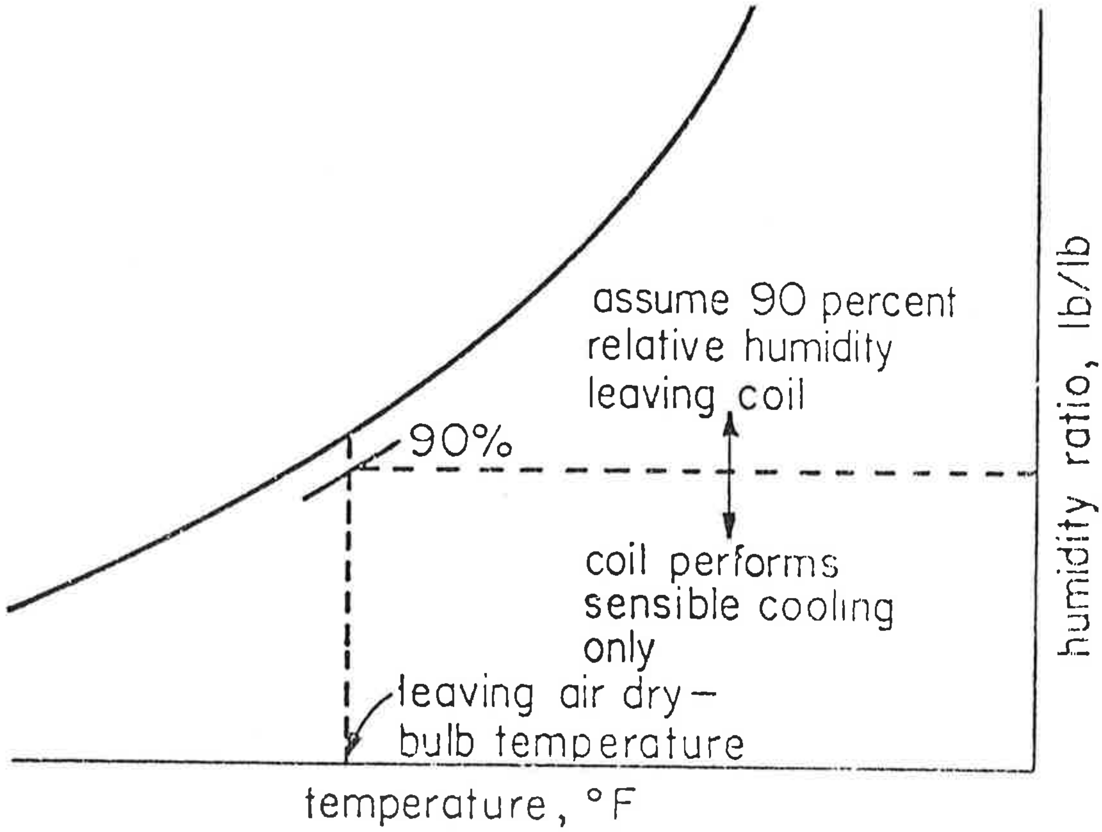


Fig. 6-3. Computing humidity of outlet air from a coil as a function of entering air conditions.

region where the inlet air temperature is greater than the outlet air temperature. This region is in turn divided into two parts. If the humidity ratio of the entering air is higher than that corresponding to 90 percent relative humidity at the leaving air temperature of the coil, the outlet condition is assigned a relative humidity of 90 percent. If the entering humidity ratio is lower than the above condition, then the coil shall be considered to perform sensible cooling only.

The basis for the above proposal is that there would normally be only a small error introduced by an inaccuracy in predicting the outlet relative humidity so long as the coil performs some dehumidification. Examination of catalog data from coil manufacturers shows that the outlet relative humidity may be as high as 98 percent, but this value occurs with a deep coil (10 rows, for example) with low air face velocity (330 fpm) and high humidity ratio of entering air. A shallow coil (3 rows) with a high air face velocity (600 rpm) may have outlet air humidities of about 85 percent. The error in cooling energy near a full load condition at these extremes might be of the order of 5 percent but under some conditions the error is on the high side and other times too low, so there is a compensating effect with respect to the cumulative energy requirements.

7. SIMULATION OF SELECTED SYSTEMS AND SUBSYSTEMS

7-1. Introduction. This chapter presents recommended procedures for simulating the following systems: terminal reheat, dual duct or multizone, variable air volume, air-water induction, decentralized heat pump, internal source heat pump, and combined air system and perimeter radiation. In addition, procedures are presented for simulating the outdoor air control that is a component of many air systems. The procedures are applicable to the conventional arrangements shown for these various systems. There are dozens of variations of these basic systems, and often a variation (for example, in a control sequence) requires a modification of the simulation procedure. The responsibility still rests with the developer of the simulation program to match the simulation to the actual system. It is intended, however, that the patterns explained in this chapter will be applicable to the majority of simulation assignments and also provide the program developer with a notion of the degree of detail that the Task Group on Energy Requirements envisions as appropriate for most energy analyses.

The structure of most of the system presentations in the following chapter includes (1) system characteristics that must be known, (2) information that must be fed from the Load Program, (3) calculation procedure that results in values of heating, cooling and shaft work energy. There are occasional references to the Load Calculation Booklet, which is a companion document to this system simulation booklet, for which the specific reference is:

"Procedure for Determining Heating and Cooling Loads for Computerized Energy Calculations--Algorithms for Building Heat Transfer Subroutines, M. Lokmanhekim, ed; compiled and published by the ASHRAE Task Group on Energy Requirements for Heating and Cooling, 1973.

Variables used in the equations are defined appropriate to each system or subsystem, but the symbols general to all are:

<u>Nomenclature</u>

c_p = specific heat of humid air = 0.245 Btu/(lb)(°F)

h = enthalpy, Btu/lb

q = rate of heat transfer, Btu/hr

t = temperature, °F

W = humidity ratio, lb of H_2O per lb of dry air

w = mass rate of flow, lb/hr

<u>7-2. Outdoor air control.</u> This subsystem is common to many air-type thermal distribution systems and is often equipped so that outdoor air may be used for cooling. The minimum supply of outdoor ventilation air must always be provided.

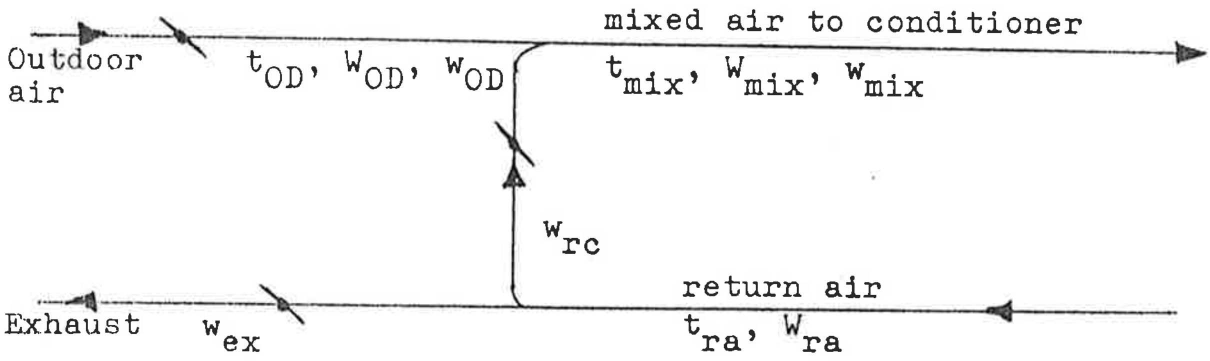


Fig. 7-1. Outdoor air control.

<u>If dampers are fixed and set so that minimum outdoor air is always provided</u>

System characteristics needed:

$w_{OD,min}$, value of minimum flow rate of outdoor ventilation air
w_{mix}, flow rate of supply air to conditioner and system

From load program

t_{OD}, temperature of outdoor air

From simulation of rest of system

t_{ra}, temperature of return air

W_{ra}, humidity ratio of return air

Procedure

$$t_{mix} = \left[t_{ra}(w_{mix} - w_{OD,min}) + t_{OD}w_{OD,min}\right] / w_{mix}$$

$$W_{mix} = \left[W_{ra}(w_{mix} - w_{OD,min}) + W_{OD}w_{OD,min}\right] / w_{mix}$$

<u>If outdoor air cooling is used</u>

System characteristics needed:

$w_{OD,min}$, value of minimum flow rate of outdoor ventilation air

w_{mix}, flow rate of supply air to conditioner and system

$t_{OD,co}$, changeover temperature from minimum ventilation air to 100 percent outdoor air (often same as return air temperature)

$t_{mix,set}$ desired temperature of mixed air

From load program

t_{OD}, W_{OD}

From simulation of rest of system

t_{ra}, W_{ra}

Procedure

If $t_{OD} \geq t_{OD,co}$

$$t_{mix} = \left[t_{ra}(w_{mix} - w_{OD,min}) + t_{OD}w_{OD,min}\right] / w_{mix}$$

$$W_{mix} = \left[W_{ra}(w_{mix} - w_{OD,min}) + W_{OD}w_{OD,min}\right] / w_{mix}$$

If $t_{mix,set} \leq t_{OD} < t_{OD,co}$

100 percent outdoor air, thus

$$t_{mix} = t_{OD}$$

$$W_{mix} = W_{OD}$$

If $t_x \leq t_{OD} < t_{mix,set}$

where t_x is the outdoor temperature at which the outdoor air damper has closed to its minimum flow position, thus

$$t_x = \left[w_{mix} t_{mix,set} - (w_{mix} - w_{OD,min}) t_{ra}\right] / w_{OD,min}$$

then

$$t_{mix} = t_{mix,set}$$

$$w_{OD} = \left(\frac{t_{ra} - t_{mix,set}}{t_{ra} - t_{OD}} \right) w_{mix}$$

$$W_{mix} = \left[W_{OD} w_{OD} + W_{ra}(w_{mix} - w_{OD})\right] / w_{mix}$$

If $t_{OD} < t_x$

$$t_{mix} = \left[t_{ra}(w_{mix} - w_{OD,min}) + t_{OD} w_{OD,min}\right] / w_{mix}$$

$$W_{mix} = \left[W_{ra}(w_{min} - w_{OD,min}) + W_{OD} w_{OD,min}\right] / w_{mix}$$

7-3. Terminal reheat system.

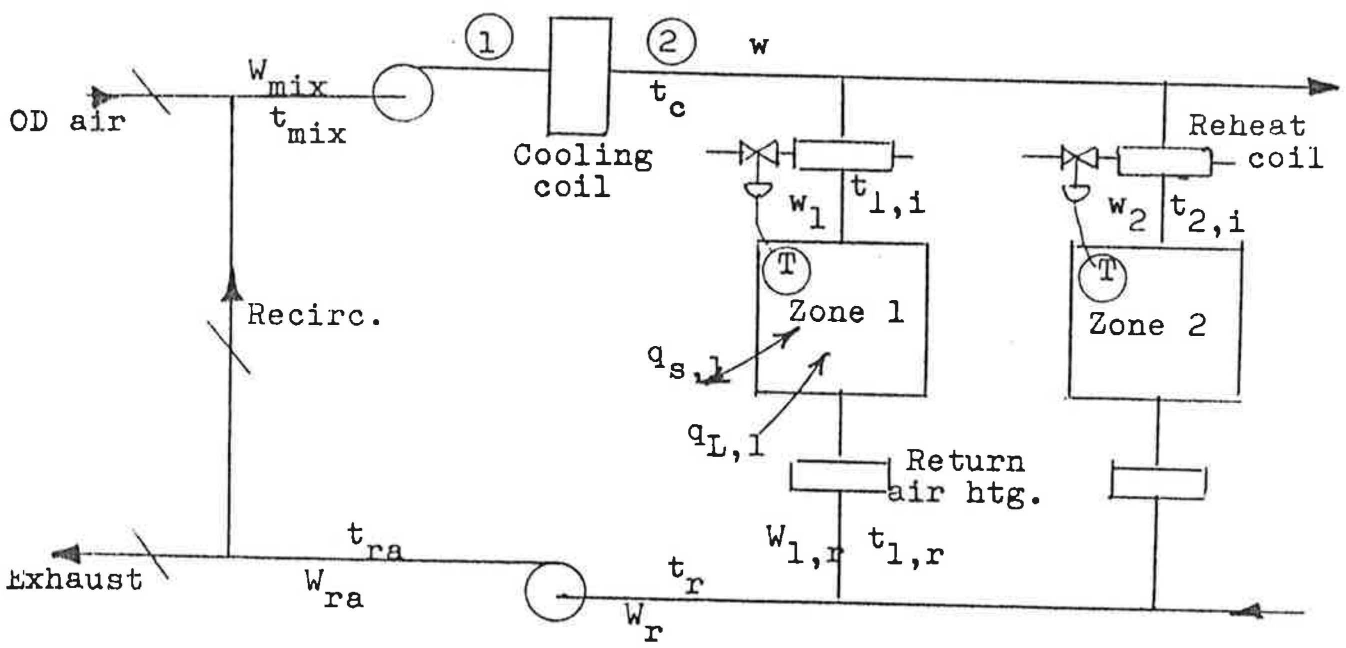


Fig. 7-2. Terminal reheat system.

System characteristics needed:

$w_1, \ldots w_n$ flow rate to each zone. These are constant.

t_c setting

$t_{1,o} \cdots t_{n,o}$ settings of space thermostats

supply fan air power and return fan air power

magnitude of return air heat (for example for lights) for each zone

operating schedule specifying which hours and which days the system operates

From load program

$q_{s,1} \cdots q_{s,n}$ sensible heat gains or losses from each zone (the q_s's are positive if they indicate heat added to the space)

$q_{L,1} \cdots q_{L,n}$ latent heat gain in each zone

Procedure

Conditions at **point 2 leaving the cooling coil**

$t_2 = t_c$

W_2 is the humidity ratio computed according to Section 6-8. (after the first hour of calculation, use W_{mix} from previous hour to determine region on psychrometric chart, Fig. 6-3).

h_2 = enthalpy computed using PSY in Load Calculation Booklet based on t_2 and W_2

Entering temperature to zones

$$t_{1,i} = t_{1,o} - \frac{q_{s,1}}{w_1 c_p}$$

similarly for other zones

Zone return air temperature and humidity ratio

$$t_{1,r} = t_{1,o} + \frac{\text{return air heating, Btu/hr}}{w_1 c_p}$$

$$W_{1,r} = W_2 + \frac{q_{L,1} \text{ Btu/hr}}{w_1 \text{ (1060 Btu/lb)}}$$

similarly for other zones

Temperature and humidity ratio of combined return air

$$t_r = \frac{t_{1,r}w_1 + \dots + t_{n,r}w_n}{w_1 + \dots + w_n}$$

$$W_r = W_{ra} = \frac{W_{1,r}w_1 + \dots + W_{n,r}w_n}{w_1 + \dots + w_n}$$

Return air temperature to conditioner

$$t_{ra} = t_r + \frac{\text{return fan air power, Btu/hr}}{(w_1 + \dots + w_n)\ c_p}$$

Mix temperature and humidity ratio

(See outdoor air control, Section 7-2)

Temperature after supply fan

(This heating effect should be introduced after the coil for a draw-through arrangement)

$$t_1 = t_{mix} + \frac{\text{supply fan air power, Btu/hr}}{(w_1 + \dots + w_n)\ c_p}$$

$$W_1 = W_{mix}$$

Enthalpy after supply fan

h_1 from PSY in Load Calculation Booklet using t_1 and W_1

Energy summaries

Heating energy of reheat coils

$$q_{rh,1} = w_1\ c_p(t_{1,i} - t_c)$$

similarly for other zones, then sum the q_{rh}'s

Cooling energy

$$q_{cool} = (h_1 - h_2)(w_1 + \dots + w_n)$$

Electric energy: supply air fan and return air fan

<u>If throttling range of cooling coil control considered</u>

Additional system information needed: relationship for t_c as a function of the cooling load, q_{cool}, Fig. 7-3

$$t_c = a + b(q_{cool})$$

where <u>a</u> and <u>b</u> are constants

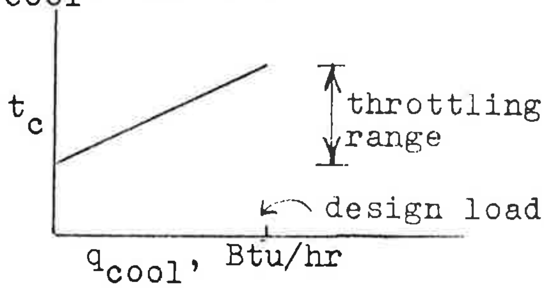


Fig. 7-3. Throttling range of cooling coil

Procedure:

After first hour of calculation, use q_{cool} from previous hour to compute t_c for present hour.

7-4. Dual-duct or multizone system

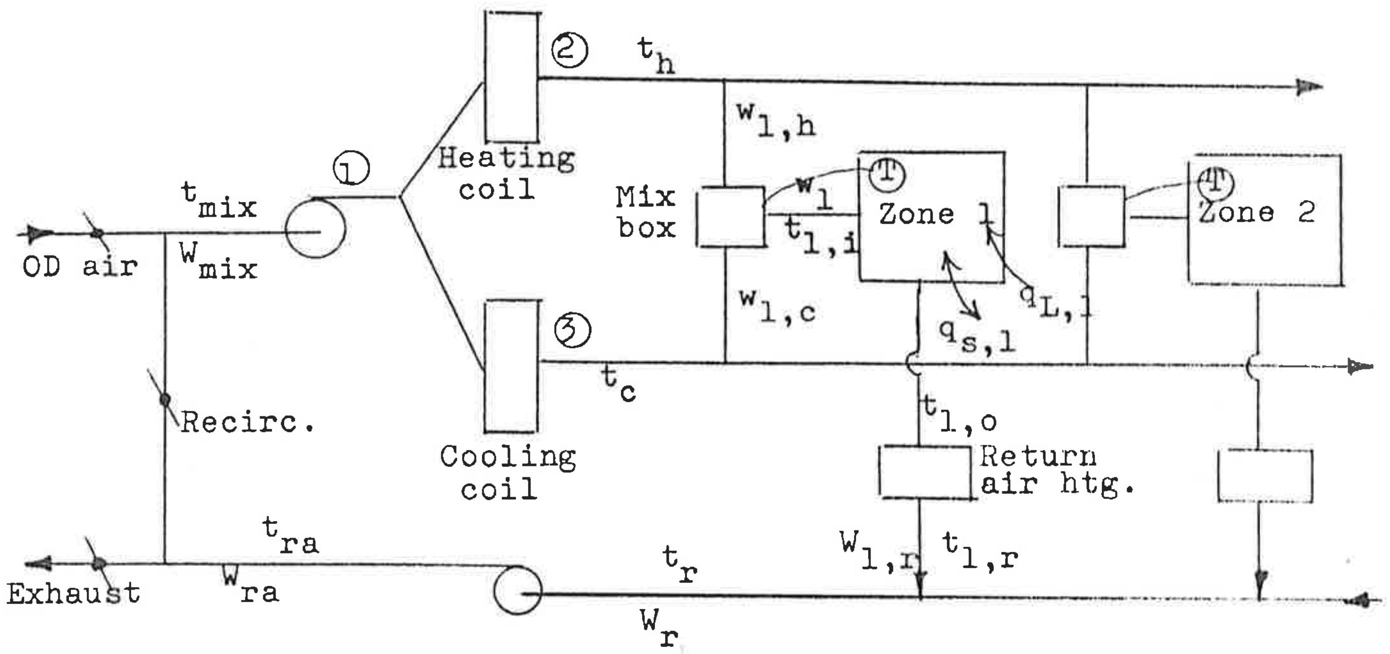


Fig. 7-4. Dual-duct system

System characteristics needed

w_1, ... w_n air flow rates to each zone, lb/hr (these values remain constant)

t_c and t_h setting. (Some systems program t_h from the outdoor temperature)

space thermostat settings $t_{1,o}$, ... $t_{n,o}$

magnitude of return air heating for each zone

supply fan air power and return fan air power

operating schedule specifying which hours and which days the system operates

From load program

$q_{s,1}$ ... $q_{s,n}$ sensible heat gains or losses from each zone (the q's are positive if they indicate heat added to the space)

$q_{L,1}$ ... $q_{L,n}$ latent heat gain in each zone

Procedure

Conditions at 2

$$t_2 = t_h$$

$W_2 = W_{mix}$ from calculation of previous hour

Conditions at 3

$$t_3 = t_c$$

W_3 = humidity ratio computed according to Section 6-8 (after first hour of calculation use W_{mix} from previous hour to determine region on psychrometric chart, Fig. 6-3).

h_3 = enthalpy from PSY in Load Calculation Booklet using t_3 and W_3

Entering temperature to zones, for example entering Zone 1, $t_{1,i}$

$$t_{1,i} = t_{1,o} - \frac{q_{s,1}}{w_1 c_p}$$

similarly for other zones

Mass flow rates of warm and cool air to zones, for example Zone

$$w_{1,h} = w_1\left(\frac{t_{1,i} - t_3}{t_2 - t_3}\right)$$

$$w_{1,c} = w_1 - w_{1,h}$$

similarly for other zones

Zone return air temperature and humidity ratio

$$t_{1,r} = t_{1,o} + \frac{\text{return air heating, Btu/hr}}{w_1 c_p}$$

$$W_{1,r} = \left(W_2 w_{1,h} + W_3 w_{1,c} + \frac{q_{L,1}\ \text{Btu/hr}}{1060}\right) / w_1$$

similarly for other zones

Temperature and humidity ratio of combined return air

$$t_r = \frac{t_{1,r}w_1 + \ldots + t_{n,r}w_n}{w_1 + \ldots + w_n}$$

$$W_r = W_{ra} = \frac{W_{1,r}w_1 + \ldots + W_{n,r}w_n}{w_1 + \ldots \quad w_n}$$

similarly for other zones

Return air temperature to conditioner

$$t_{ra} = t_r + \frac{\text{return air fan power, Btu/hr}}{(w_1 + \dots + w_n)c_p}$$

Mix temperature and humidity ratio
(See outdoor air control, Section 7-2)

Temperature and humidity ratio and enthalpy after supply fan

$$t_1 = t_{mix} + \frac{\text{supply fan air power}}{(w_1 + \dots + w_n)c_p}$$

$$W_1 = W_{mix}$$

h_1 from PSY in Load Calculation Booklet using t_1 and W_1

Energy summaries

Heating energy of heating coil

$$q_h = (w_{1,h} + \dots + w_{n,h})(t_2 - t_1)c_p$$

Cooling energy at cooling coil

$$q_{cool} = (w_{1,c} + \dots + w_{n,c})(h_3 - h_1)$$

Electric energy

energy for supply-air fan and return-air fan

<u>If throttling ranges of controls of cooling coil and heating coil are considered</u>.

Additional system information needed:

instead of $t_2 = t_h$

$t_2 = t_h + a_1 - b_1 q_h$ as in Fig. 7-5

where a_1 and b_1 are constants

t_h is either fixed or programmed on the basis of outdoor temperature

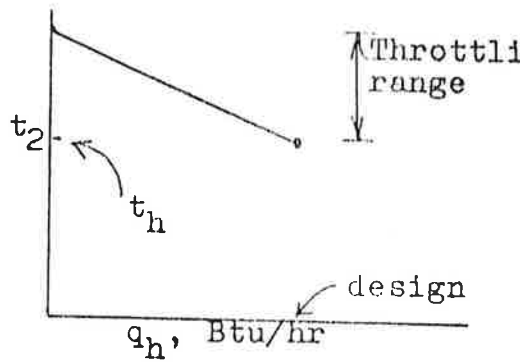


Fig. 7-5. Throttling range of heating coil.

For the cooling coil,

$t_3 = a_2 + b_2(q_{cool})$

corresponding to Fig. 7-3.

<u>Procedure:</u> after the first hour of calculation, use q_{cool} and q_h from previous hour to compute t_2 and t_3 for present hour.

7-5. Variable-air-volume system.

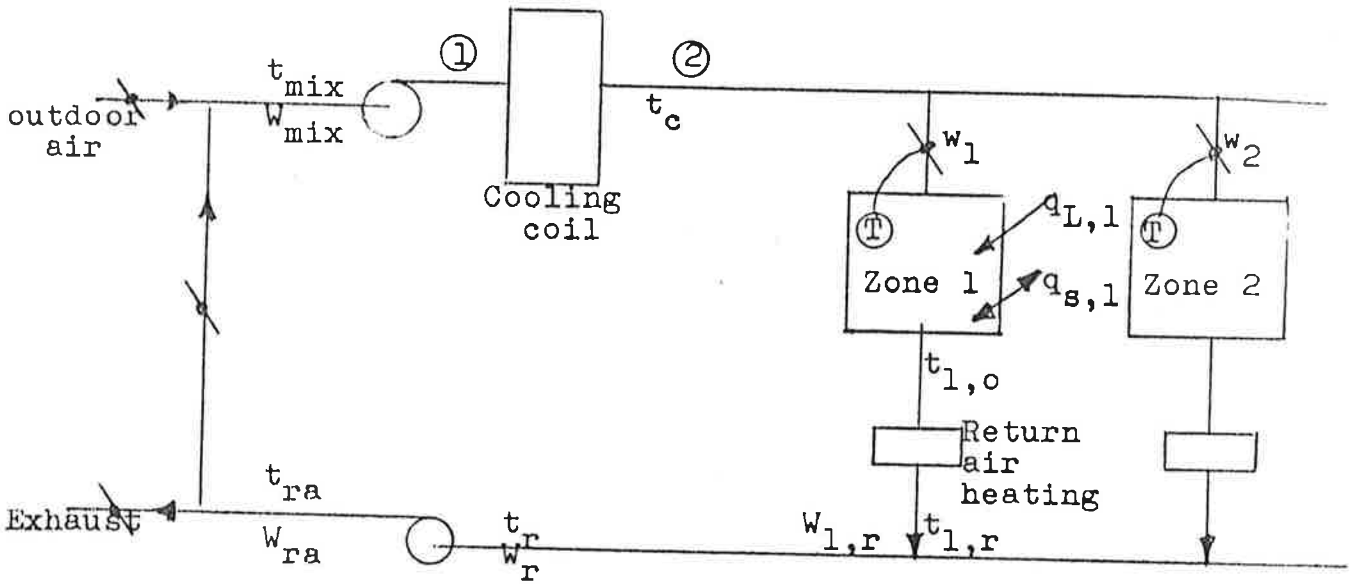


Fig. 7-6. Variable-volume system--cooling capability only.

System characteristics needed:

t_c air temperature leaving cooling coil

$t_{1,o} \cdots t_{n,o}$ settings of space thermostats

magnitude of return air heating for each zone

supply fan power as a function of flow rate and static pressure

From load program

$q_{s,1} \cdots q_{s,n}$ sensible heat gains or losses from each zone (only heat gains can be accommodated here)

$q_{L,1} \cdots q_{L,n}$ latent heat gain in each zone

Procedure

Temperature and humidity ratio at 2.

$t_2 = t_c$

W_2 is the humidity ratio computed according to Section 6-8 (after the first hour of calculation use W_{mix} from the previous hour to determine region on the psychrometric chart, Fig. 6-3.

h_2 = enthalpy from PSY in Load Calculation Booklet using t_2 and W_2.

Flow rates to zones

$$w_1 = \frac{q_{s,1}}{c_p(t_{1,o} - t_c)}$$

similarly for other zones

Zone return air temperature and humidity ratio

$$t_{1,r} = t_{1,o} + \frac{\text{return air heating, Btu/hr}}{w_1\ c_p}$$

$$W_{1,r} = W_2 + \frac{q_{L,1}\ \text{Btu/hr}}{1060\ w_1}$$

similarly for other zones

Temperature and humidity ratio of combined return air

$$t_r = \frac{t_{1,r}w_1 + \ldots + t_{n,r}w_n}{w_1 + \ldots + w_n}$$

$$W_r = W_{ra} = \frac{W_{1,r}w_1 + \ldots + W_{n,r}w_n}{w_1 + \ldots + w_n}$$

Return air temperature to conditioner

$$t_{ra} = t_r + \frac{\text{return air fan power}}{(w_1 + \ldots + w_n)c_p}$$

Mix temperature and humidity ratio

See outdoor air control, Section 7-2.

Temperature and humidity ratio after the supply fan

$$t_1 = t_{mix} + \frac{\text{supply fan air power, Btu/hr}}{(w_1 + \ldots + w_n)c_p}$$

$$W_1 = W_{mix}$$

h_1 from PSY in Load Calculation Booklet, using t_1 and W_1

Power for supply-air fan

The power required by the motor driving the supply fan in a VAV system may be appreciable, particularly in a high-pressure system. Over a range of operating conditions of the system, this power may vary appreciably. Figure 7-7 shows the power required by the fan of one manufacturer under two methods of control. In the "throttling only" mode the static pressure increases as the dampers at the zones reduce the flow rate of air.

When inlet-vane regulation is provided to maintain a constant static pressure, the power required by the fan drops off noticeably as the flow decreases. Since the

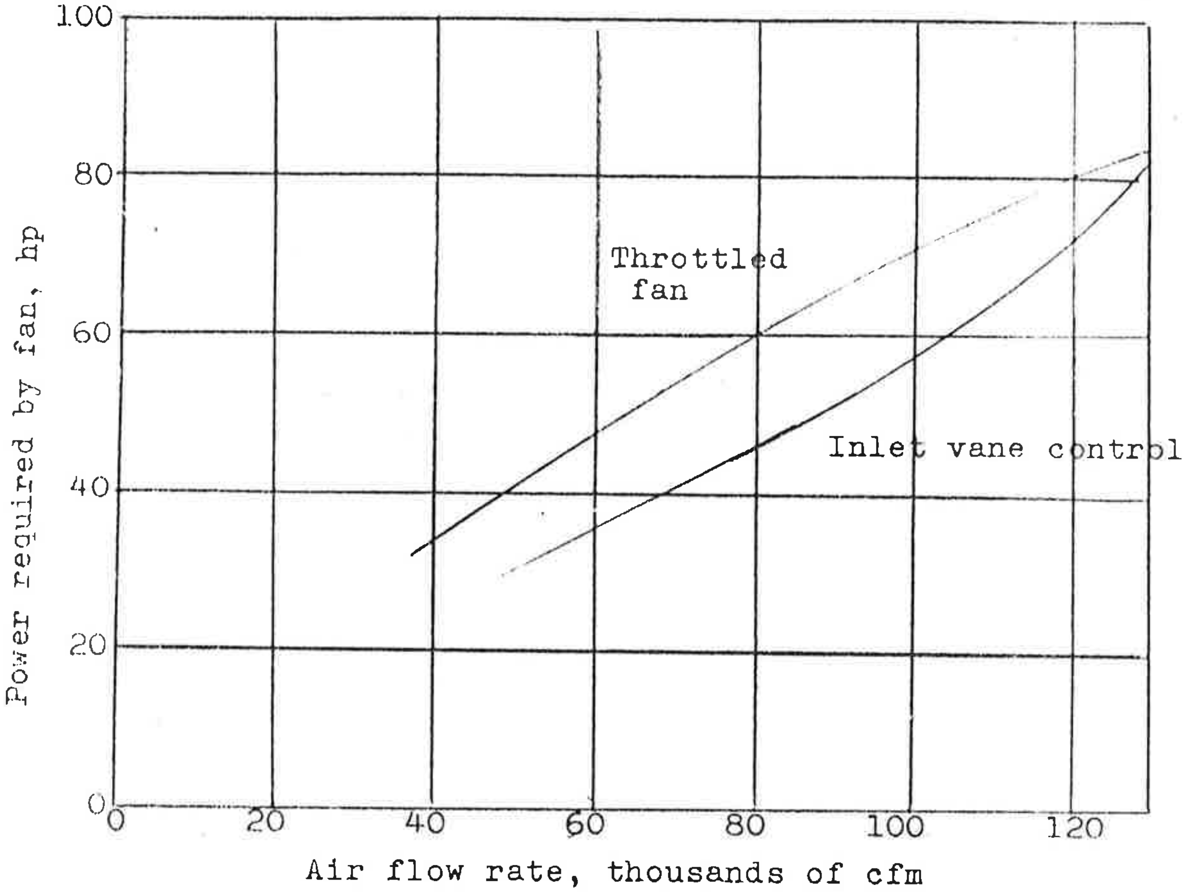


Fig. 7-7. Power required by a fan in a variable-volume system. (The inlet vane control maintains constant discharge static pressure).

fan is a high consumer of energy, calculation of its power requirement at part-load will enhance the accuracy of the energy calculation. An approximation that is far superior to assuming a constant power requirement on a variable-volume fan would be to assume a linear variation in power as a function of air-flow rate

Energy summaries

Cooling energy

$$q_{cool} = (w_1 + \dots + w_n)(h_2 - h_1)$$

Electric energies

supply-air fan motor
return-air fan motor

7-6. Air-water induction units.

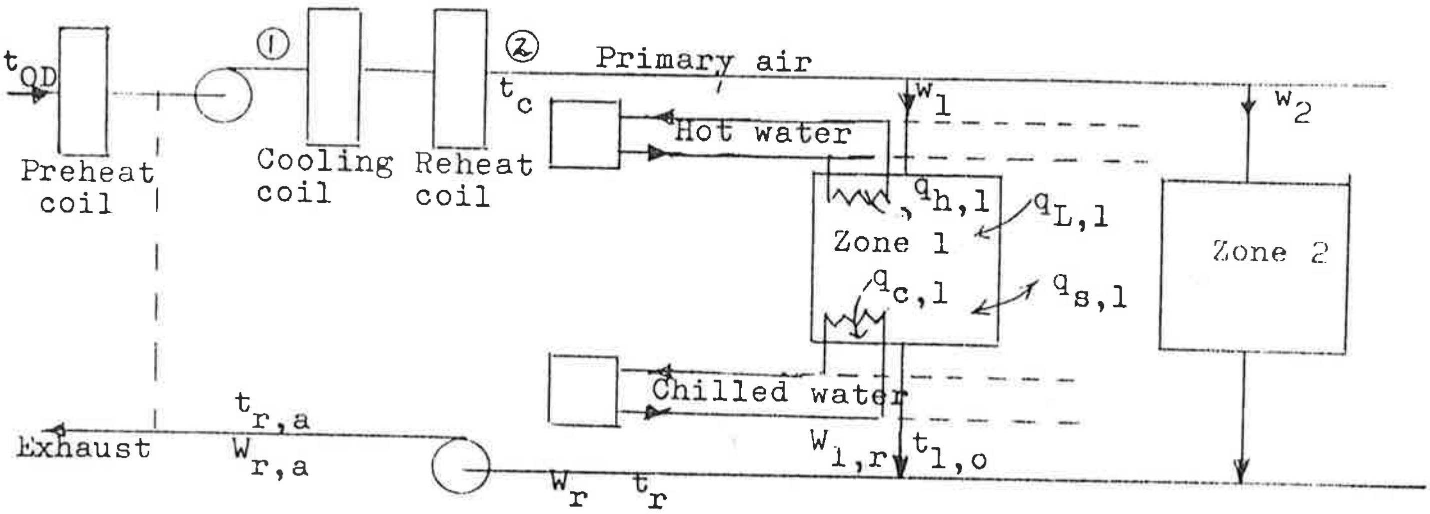


Fig. 7-8. Air-water induction system.

System operation on which simulation procedure is predicated:

Each zone receives a specified flow rate of primary air supplied at a temperature t_c that is programmed according to the outdoor temperature (see page 4.6, ASHRAE Handbook and Product Directory, 1973 Systems.) The primary air at the induction unit aspirates secondary air from the room over a cooling or heating coil. Figure 7-8 shows a 4-pipe system with separate heating and cooling coils in the induction unit. If instead a two-pipe system is used, it is assumed that the conditions of the primary air combined with the available temperature of water can accommodate the varied heating and cooling loads prevailing in the zones. These systems are often designed so that the cooling coil in the induction unit performs only sensible cooling and no dehumidification. This mode of operation will be assumed in the procedure below resulting in all of the latent heat being removed by the primary air. The air-water system is often designed so the primary air is 100 percent outside air. The simulation procedure assumes 100 percent outside air, but if such is not the case, the methods of Section 7-2 may be applied.

System characteristics needed

$w_1 \ldots w_n$, the rates of primary air flow to each zone

$t_{1,o} \ldots t_{n,o}$ the settings of the space thermostats

schedule of t_c as a function of outdoor temperature

supply fan air power and return fan air power

From load program

$q_{s,1} \ldots q_{s,n}$ sensible heat gains or losses from each zone (a positive sign indicates a heat gain)

$q_{L,1} \ldots q_{L,n}$ latent heat gain in each zone

outdoor conditions t_{OD}, W_{OD}

<u>Procedure:</u>

Temperature leaving central conditioner

t_c of primary air from t_c vs t_{OD} program

Heating ($q_{h,1}$) or cooling ($q_{c,1}$) required by water coil in Zone 1

If $w_1 c_p(t_{1,o} - t_c) \leq q_{s,1}$

then water coil cooling capacity is

$$q_{c,1} = q_{s,1} - w_1 c_p(t_{1,o} - t_c)$$

and $q_{h,1} = 0$

If $w_1 c_p(t_{1,o} - t_c) > q_{s,1}$

then water coil heating capacity is

$$q_{h,1} = - q_{s,1} + w_1 c_p(t_{1,o} - t_c)$$

and $q_{c,1} = 0$

Energy summary

Heating energies

zone coils $q_{h,1} + \cdots + q_{h,n}$

preheat and reheat coils

$$(w_1 + \cdots + w_n)(t_c - t_{OD})c_p$$

Cooling energy

zone coils $q_{c,1} + \dots + q_{c,n}$

primary air cooling coil

If $t_c < t_{OD}$, $q_{c,pc} = (w_1 + \dots + w_n)(h_2 - h_{OD})$

where h_{OD} is calculated from PSY in Load calculation Booklet using t_{OD} and W_{OD}

and h_2 is computed from PSY using t_c and W
(W_2 computed from Sec. 6-8)

Electric energy

supply-air fan and return-air fan

7-7. Decentralized heat pumps. The flow diagram of this system, Fig. 7-9, shows a water circulating loop supplying or

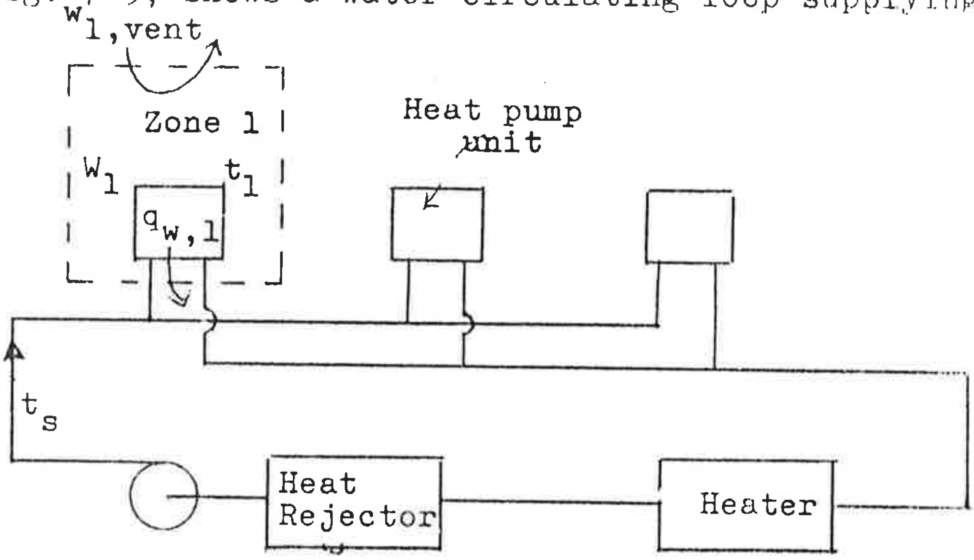


Fig. 7-9. Decentralized heat pump system.

absorbing heat from individual heat pump units in the zones. If the loop water supplies a net quantity of heat to the heat pumps, the supply temperature t_s drops to a low value t_{low} (75 °F, for example) at which point the heater in the water loop comes into service. When there is a net rejection of heat to the water loop, t_s rises to a high value t_{high} (90 °F, for example) at which point a heat rejector comes into service. Because a net value of zero heat rejection or absorption by the water loop is unlikely, the simulation procedure is based on t_s being either at t_{low} or t_{high}.

Numerous versions of the terminal unit are available, including heating-and-cooling capabilities of the vapor compression unit; cooling only with the compression unit combined with electric heating; and with-or-without ventilation air capabilities. The simulation procedure below presupposes a heating/cooling heat pump with ventilation capabilities.

System information needed

For each heat pump unit, the a-values and the b-values in the tables below

Heating operation

Inlet water temperature	Heating capacity Btu/hr	Power required kW	Heat abosrbed from loop water, Btu/hr
t_{low}	a_{11}	a_{12}	a_{13}
t_{high}	a_{21}	a_{22}	a_{23}

Cooling operation

Inlet water temperature	Air wet bulb temp, °F	Cooling capacity Btu/hr	Power required kW	Heat rejected to water Btu/hr
t_{low}	60	b_{11}	b_{12}	b_{13}
	68	b_{21}	b_{22}	b_{23}
	75	b_{31}	b_{32}	b_{33}
t_{high}	60	b_{41}	b_{42}	b_{43}
	68	b_{51}	b_{52}	b_{53}
	75	b_{61}	b_{62}	b_{63}

Flow rate of ventilation air to each decentralized unit $w_{1,vent}$ lb/hr

Settings of space temperatures t_1, t_2, etc.

From load program

$q_{s,1} \cdots q_{s,n}$ sensible heat gain (or loss), exclusive of ventilation air load, Btu/hr

$q_{L,1} \cdots q_{L,n}$ latent heat gain, exclusive of ventilation air load, Btu/hr

t_{OD} outdoor air temperature

W_{OD} outdoor air humidity ratio

h_{OD} outdoor air enthalpy

Procedure

Establish t_s as either t_{low} or t_{high}, based on whether net rejection or absorption, respectively, of heat from water loop during previous hour of computation

Determine whether heat pump in heating or cooling mode

Heating mode if $q_{s,1} \geqslant w_{1,vent}(t_1 - t_{OD})(c_p)$

$$F_1 = \text{fraction of operating time} = \frac{q_{s,1} + w_{1,vent}(t_1 - t_{OD}}{a_{11} \text{ or } a_{21}, \text{ whichever applicable}}$$

P_1 = power required by heat pump unit, kW

$= (F_1)(a_{12}$ or a_{22}, whichever is applicable)

$q_{w,1}$ = heat transferred to loop water

$= (-F_1)(a_{13}$ or a_{23}, whichever is applicable)

similarly for other zones

Cooling mode if $q_{s,1} < w_{1,vent}(t_1 - t_{OD})$

W_1 = humidity ratio in zone

$$= \frac{(w_{1,vent})W_{OD} + \dfrac{q_{L,1}}{1060}}{w_{1,vent}}$$

WBT_1 = wet bulb temperature in zone from PSY in Load Calculation Booklet using t_1 and W_1

h_1 = enthalpy in space from PSY using t_1 and W_1

$q_{c,1}$ = cooling capacity of unit by interpolation based on WBT_1 between b_{11} and b_{31} or between b_{41} and b_{61}, whichever is applicable

F_1 = fraction of operating time

$$= \frac{q_{s,1} + q_{L,1} + w_{1,vent}(h_{OD} - h_1)}{q_{c,1}}$$

P_1 = power required by heat pump unit

= (F_1)(value from interpolation based on WBT_1 between b_{12} and b_{32} or between b_{42} and b_{62}, whichever is applicable)

$q_{w,1}$ = heat transferred <u>to</u> water in loop

= (F_1)(value from interpolation based on WBT_1 between b_{13} and b_{33} or b_{43} and b_{63}, whichever is applicable)

Determination of loop water temperature

If $q_{net} > 0$ then $t_s = t_{high}$

where $q_{net} = q_{w,1} + q_{w,2} + \cdots q_{w,n}$

If $q_{net} < 0$ then $t_s = t_{low}$

use this value of t_s in next hour calculation

Energy summaries

Heat pump energy

$P_1 + \cdots + P_n$

Loop heating energy

q_{net} Btu/hr if $q_{net} < 0$

Auxiliaries

7-8. Internal-source heat pump. The central component of this system, illustrated in Fig. 7-10, is the heat pump whose purpose is to transfer heat from cooling assignments to locations where heating is needed. If more heat is available at the condenser of

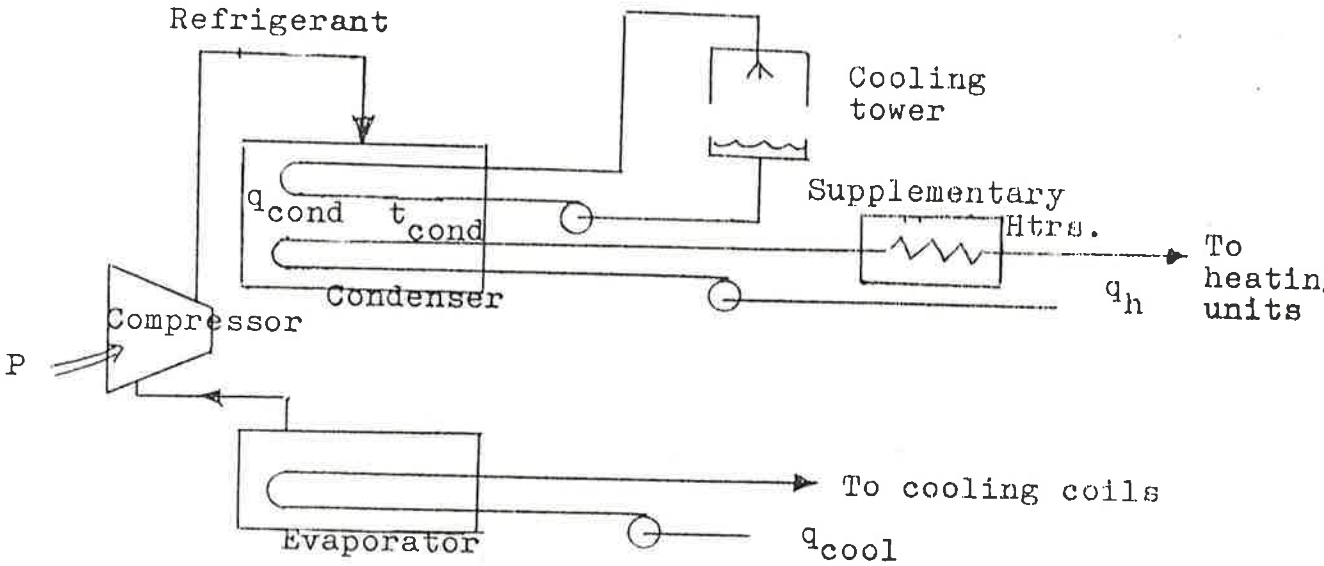


Fig. 7-10. Internal-source heat pump.

the heat pump than needed in the heating assignments, the excess heat is rejected to a cooling tower.

System information needed

The compressor power requirement P and the heat rejection rate at the condenser q_{cond} as a function of the cooling capacity q_{cool} is needed at least for two levels of condensing temperature. The high level of condensing temperature is that typical of operation when the cooling tower must be in service. The low level of condensing temperature should be typical of conditions when all of the heat rejection from the condenser is needed for the heating assignments. Thus,

at high condensing temperature $\quad P_1$ kW $= f_1(q_{cool}$ Btu/hr)

and $\quad q_{1,cond}$ Btu/hr $= f_2(q_{cool}$ Btu/hr)

at low condensing temperature $\quad P_2$ kW $= f_3(q_{cool}$ Btu/hr)

and $\quad q_{2,cond}$ Btu/hr $= f_4(q_{cool}$ Btu/hr)

Information needed from companion system simulation

q_{cool} and q_h are values fed from a companion system simulation of applicable system(s) such as dual-duct, perimeter radiation, induction units, etc.

Determination of mode of operation

Compute $q_{1,cond}$ and $q_{2,cond}$

If $q_{2,cond} > q_h$ the cooling tower should be in operation and the high condensing temperature applies

then $P = P_1$

If $q_{2,cond} < q_h$ the supplementary heater should be in operation and the low condensing temperature applies

then $P = P_2$

Energy summary

Heat pump energy, kW = P

Supplementary heater energy (if operating) $= q_h - q_{2,cond}$

Auxiliaries

Possible refinement

Compute heat pump power, P, using equations in Section 5b at the applicable chilled water and condenser water temperatures.

7-9. Combined air system and perimeter convectors. Especially in cold climates many buildings have their exterior zones equipped with a combination of an air system (dual-duct, variable volume, or terminal reheat) and with perimeter convectors supplied with hot water, as shown in Fig. 7-11. The temperature of the hot water

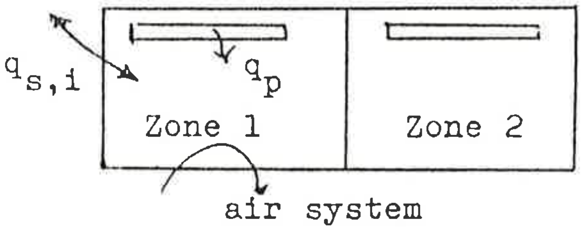


Fig. 7-11 Combination of air system and perimeter convectors.

is often programmed according to the outdoor temperature, Fig. 7-12, so that the output of the perimeter units, q_p Btu/hr, shown in Fig. 7-13, increases with a reduced outdoor temperature.

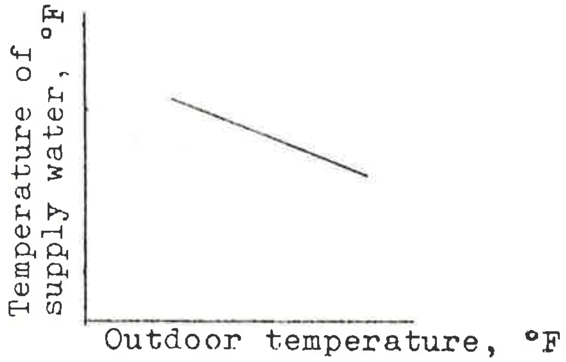


Fig. 7-12. Hot water temperature supplied to perimeter units

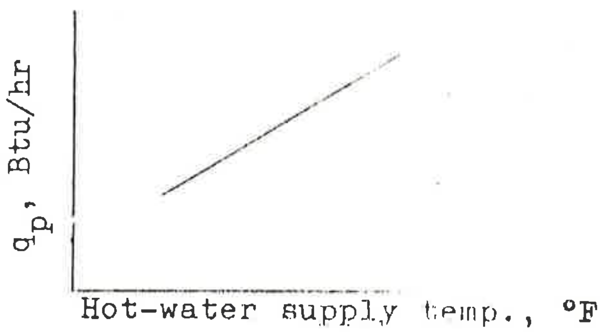


Fig. 7-13. Thermal output of convectors

The simulation of the air system proceeds as outlined previously in this chapter except that the sensible heat load on a zone is increased by the value q_p.

<u>7-10. Conclusion.</u> Magnitudes of the heating energy requirement and the cooling energy requirement are the output information for many of the simulations illustrated in this chapter. When the system includes a central heating (hot water or steam) plant, the thermal load must then be translated to a basic energy requirement (gas, oil, electricity) through application of the appropriate conversion efficiency. The cooling energy requirement must be translated into an electric energy requirement (for vapor compression refrigeration) or into steam/hot water requirements and ultimately fuel requirements (for an absorption chilling plant).