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CONTINUING REGIONAL SOLAR ENERGY INFORMATION MINI-CENTER ACTIVITIES AND UPDATING THE SOLCOST PROGRAM

Quarterly Report

By
Loren J. Lantz
C. Byron Winn
James A. Leflar

October 1978

Work Performed Under Contract No. EG-77-C-02-4643

Solar Environmental Engineering Co., Inc.
Fort Collins, Colorado

U.S. Department of Energy

Solar Energy

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Prepared by

Mr. Loren J. Lantz
Dr. C. Byron Winn
Mr. James A. Leflar

Solar Environmental Engineering Co., Inc.
P. O. Box 1914
Fort Collins, Colorado 80522

October, 1978

PROGRESS SUMMARY

Following is the progress made during August, September, and October, 1978, for each task of the contract.

Task 1: Interactive SOLCOST Interface

Task 1 requires Solar Environmental Engineering Company (SEEC) to develop an informative and fully interactive front end to the SOLCOST program. It is to ensure that the users, through the interactive front end, can call the main SOLCOST program after the input data has been properly specified, and the user can easily alter input parameters and rerun the new data. The interactive instructions are to be in three forms: 1) detailed instructions, 2) long form questions, and 3) short prompters. To fulfill this requirement, SEEC acquired from Martin Marietta a listing and magnetic tape of the SOLCOIN program which is utilized on the CYBERNET. This program was reviewed thoroughly to, first of all, see how standard the code was, and if it was not standard, then could it easily be altered or adapted to ANSI Standard requirements. The code was observed to have certain portions written in a machine dependent language called COMPASS. This is a computer language peculiar only to CDC machines. It is one step above the bit level machine language. It is almost impossible to transfer it to other machines, such as a UNIVAC or IBM without first rewriting it into another language such as FORTRAN. Systems engineers at the central computing facility used by SEEC were contacted, and it was observed that these COMPASS subroutines were digging into the machine and actually altering functions of the machine which would not be possible on other computing facilities. This would make transformation of these subroutines very difficult without completely redefining the whole package. Consequently, SEEC decided to salvage what portions of the program it could; that is, subroutines which did functions such as list prompters, which give detailed instructions with default values, etc.; SEEC would then write the main program in ANSI Standard FORTRAN code.

The desired features of the SOLCOIN program furnished by Martin Marietta have been removed and are in the process of being transformed into ANSI Standard FORTRAN. The main program is in the process of being written, and the primary reading routine has been developed. The reading function in the SOLCOIN program furnished by Martin Marietta used machine dependent features. A subroutine has been developed by SEEC which utilizes machine independent language so that once the SOLCOIN program has been transformed and written in ANSI Standard, it will be portable to most any machine in the country which supports the ANSI Standards. The interactive interface should be completed shortly after the first of the year.

Task 2: Energy Conservation - Solar System Optimization

The main thing done in this area was to define the methodology. No coding has been done. A former SEEC employee, C. Dennis Barley, developed a methodology for optimizing the total environmental energy system. That is, pick the cost optimum solar system. This methodology was then encoded on magnetic cards for use by small handheld calculators and is available through SEEC. It has been observed that by total optimization of the system, alternatives to the possible choice of insulation might be different for a solar system than it would be for a non-solar system.

When choosing energy conserving subsystems on an economic basis, it becomes important to consider the whole system. For example, by using an active solar system, the effective cost of energy for heating is different than it would be if heat were supplied by conventional fuels only. This could alter the choice of insulation or windows or doors or infiltration limiting devices. Another possible feature that this type of optimization technique could do would be to pick a solar system as part of the total system which would be economically viable over the lifetime of the system, whereby if the house were defined completely before picking a solar system it may not be possible to choose an economically viable solar system. A mathematical development giving the equations which will be coded into the SOLCOST program is in Appendix A. The reader is referred there for more detailed information. This development is preliminary and may be altered depending upon the coding requirements as it is incorporated into the SOLCOST program. However, it is anticipated that the changes will be minor if there are any at all.

Task 3. Detailed Duct Sizing

The SOLCOST program presently has a subsystem selection subroutine unit which does ball park sizing of the collector ducts only. The design point for this subroutine is the maximum velocity in the duct that is to be allowed by the user or the designer. There is another important parameter that must be considered and that is the maximum allowable pressure drop. In certain situations, for example ducts with a large amount of internal insulation, resistances to flow may be sufficient enough to exceed this maximum allowable pressure drop. Also, the present duct design program does no sizing of collector manifolds. It also specifies only round ducts. It also does not specify the duct sizes for the distribution system to the house, nor does it indicate possible alternatives or the consequences of going off design. The reader is referred to Appendix B for a more complete description of what is actually involved in designing duct systems for air systems. This methodology has been defined in this appendix, and a preliminary subroutine has been written. However, it will not be incorporated into the SOLCOST program until after the final updated FORTRAN version is completed.

Task 4. Extension of the SOLCOST Data Base

The management system for the SOLCOST Data Base was developed in the previous portion of the contract. Also, a great deal of performance data, primarily for collectors, was received and incorporated into the Data Base. Efforts since the extension of the contract have been in the area of expanding the collector information and also further solicitation of information from manufacturers for pumps, heat exchangers, controls and storage devices. At the present time a mailing is underway to get information from the manufacturers of these devices, to incorporate it into the Data Base, and then to go back to the manufacturer to see if the included information is correct.

The other area concerned under the Data Base category is the coordination of the Data Base effort with the Solar Energy Research Institute (SERI) in Golden, Colorado. Recently, SEEC requested from SERI an updated list of manufacturers which they had accumulated in the process of the development of their overall solar data base. They were most cooperative in providing it. Efforts are also underway to possibly include the SOLCOST Data Base as a sub-data base of their large data collection system. There is some sort of a paradox in that SERI has little money to incorporate the SOLCOST Data Base, but they have a large manpower availability. At the same time, they have been mandated to put in a component data base within approximately the next year. It would cost less to incorporate the SOLCOST Data Base and its management system, which has already been developed, than it would be for SERI to develop its own. However, at the same time, they would have to rechannel their manpower efforts. Coordination to solve this problem is underway at the present time.

Task 5. Portable FORTRAN Version

Barely after the extension of the SOLCOST Mini-Center contract, SEEC obtained from Martin Marietta their developed SOLCOST FORTRAN version. This particular program has been reviewed thoroughly by the SEEC staff, and even though it is not in Standard FORTRAN, the non-Standard code is a small part of the program with one exception; the input routine. The input requirements for the present FORTRAN version are in the NAMELIST format. This particular input format is structured in such a way to make inputting of information easy. New values are written between dollar signs. This is completely different than the present version of SOLCOST on the CYBERNET network. Most computing systems support NAMELIST. However, it is not ANSI Standard.

The other area of concern is the inclusion of features in the CYBERNET SOLCOST program which are not in the FORTRAN version. There are two primary ones of interest. First is the inclusion of the sub-selection sizing routines which were developed by SEEC under a previous contract. Secondly, the plotting package which plots various parameters against time and payback and which is a user definable option at the present time. The software for doing the plotting will have to be developed by SEEC because the subroutines defined by Martin Marietta are very dependent on machine functions which are very non-Standard. It is anticipated that the ANSI Standard conversion of the FORTRAN code will be completed in January and then the inclusion of these omitted functions will be done shortly thereafter.

Task 6. Automatic Collector Parameter Subroutine

The reader is referred to Appendix C which gives a full-scale development and explanation of the automatic parameter updating subroutine functions. Coding of the equations given in this appendix is underway at the present time and will be completed shortly. It will be a simple matter to include this subroutine in the updated SOLCOST ANSI Standard FORTRAN version.

Task 7. The Solar Index

Ever since the summer of 1978, SEEC has been supporting DOE in its effort to provide the Solar Index to as many people as possible. At present, the city list has been expanded to about 20 and is primarily concentrated on the East coast. The second largest time zone density is the Central region, the third is the Mountain States region, and the fourth is the West coast. Interest in the Index has recently been growing, and it is getting to the point where it must be automated if it is to continue to grow. If the present growth rate continues, it won't be much longer before SEEC will reach its saturation point and will start having to turn down cities just because of the increased work load. This is undesirable, for if that is to take place, it may limit interest in the Index effort. SEEC has submitted to the Department of Energy a follow-on proposal to accomplish the full-scale automation and expansion of the Index effort. The reader is referred to the proposal for details.

Task 8. Operation of the SOLCOST Mini-Center

The software list used by the SOLCOST Mini-Center is getting quite lengthy. At the present time we have two version of MITAS. One is to support the SOLCOST program, and the second version of MITAS is to support the heat loads version which was developed to support the SOLCOST version. These two versions of MITAS have common support libraries, but the lengths of these programs are quite long, requiring a great amount of file space. Also, the computer time requirements for generation of an executable binary program are high. It is hoped that once the final FORTRAN version is completed a lot of this supporting computer code can be eliminated. This will lessen a lot of the manpower requirements to support the SOLCOST program.

Interest in execution of the SOLCOST program has been increasing steadily over the last month. The Mini-Center is beginning to receive SOLCOST inputs using the new space heating handbook form. However, execution of the programs by the Mini-Center demand is still light. After conferring with Martin Marietta, it became obvious that the primary use of SOLCOST is over the networks. This further accentuates the need for the portable FORTRAN version so that many of the users who currently use it through the networks or through the Mini-Center can use the code in their own house. This will increase confidence in the program's ability and also shorten the time from input to answer. This alone should substantially increase the number of users of the SOLCOST program.

In coordination with Martin Marietta, it has been decided that SEEC will eventually take over maintenance and operation of the SOLCOST program on the computer networks. When the new FORTRAN version is completed, it will be installed on the networks along with the new portable front end (SOLCOIN) program. It is expected that this changeover will take place in the spring of 1979.

Appendix A

COMBINED OPTIMIZATION OF SPACE HEATING LOADS FOR WINDOWS, INSULATION, DOORS, ETC. AND THE SOLAR COLLECTOR AREA

Introduction

When load variables such as windows and insulation types are included in the optimization of a conventional or solar space heating system, the overall solar cost is lower than that resulting from optimization of a heat supply system for a fixed load system. This appendix presents a simple, straightforward algorithm for optimizing the trade off between construction and fuel costs in a conventional heating system or between construction,fuel and collector costs in active solar systems. The methodology which follows was developed by C. Dennis Barley, a former SEEC employee, and was implemented on small handheld calculators for easy use by the engineering and/or design community. When Mr. Barley developed this procedure, he assumed continuity and uniformity in some of the parameters which are not continuously defined in the SOLCOST program; one of these assumptions is that the fuel cost per unit is constant. SOLCOST allows for the user to define the fuel costs in a piecewise linear fashion. That is, the average fuel cost may be different for different loads. Also inherent in Mr. Barley's development is the assumption that non-modular collector areas could be considered to be optimum. These assumptions have been removed to make the following development more general. A cost was paid since this development is more general. An iterative procedure must be used if the cost of fuel is not constant, or the collector area must be an integer multiple of the module size.

The design heat load (UA) of a building may be defined by equation A-1. This equation adds up all of the conductances through the enclosure shell increments, that is, walls, windows and/or doors and then includes parameters for infiltration losses through cracks in the walls and around windows and doors, etc. This UA value is used to calculate the design load of a building and can be easily modified using ASHRAE techniques to represent the average UA value. For those segments of the external shell which have associated infiltration rates with them, the U_i can be transformed to an effective U' value as shown in Eqn. A-2.

$$UA = \underbrace{\sum_{i=1}^{N} U_i A_i}_{\text{conduction}} + \underbrace{\rho \, C_p \sum_{j=1}^{M} V_j}_{\text{infiltration}} \qquad \text{(A-1)}$$

U_i = conductance of ith element

A_i = area of ith element

ρ = density of air

C_p = specific heat of air

V_j = leakage rate of jth leak

This effective U_i' value would remain the same as shown in Eqn. A-3 for segments of the shell such as walls.

$$U_i' = U_i + \frac{V_i \, \rho \, C_p}{A} \qquad \text{(A-2)}$$

$$U_i' = U_i \qquad \text{(A-3)}$$

The individual U_i's and V_j's to be considered for the optimization process are the heat load contributing elements of the shell. That is, should R6, R7, R9, or R16 and so on be considered, or should single-pane, double-pane, or triple-pane windows be considered and similarly for doors; a foam core, solid wood, no core, etc. A-4 shows the resultant UA as modified to incorporate those sections of the shell which have associated infiltration rates.

$$UA = \underbrace{\sum_{i=1}^{N} U'_i A_i}_{\text{conduction}} + \underbrace{\rho C_p \sum_{j=1}^{M} V_j}_{\text{infiltration}} \qquad \text{(A-4)}$$

The cost of building this shell is given by A-5; this particular cost is not the total cost of the external portions of the building. It is the cost only of those candidate pieces which are to be considered in the optimization process. A-6 gives the total cost for the thermal system in present worth form.

$$C_e = \underbrace{\sum_{i=1}^{N} A_i C_{u_i}}_{\text{surface costs}} + \underbrace{\sum_{j=1}^{M} C_{v_j}}_{\text{infiltration costs}} \qquad \text{(A-5)}$$

A_i = area of ith segment

C_{u_i} = cost of ith shell portion for conduction

C_{v_j} = cost of jth shell portion for infiltration

$$C_T = (C_e + \underbrace{C_h + LC_L}_{\text{cost of furnace}})E_1 + \underbrace{L\ C_f}_{\text{fuel costs}} E_4 \qquad \text{(A-6)}$$

C_a = space heating system fixed costs

C_L = load dependent costs

L = heating load

C_f = cost of fuel per unit

E_1 and E_4 are multipliers to bring all of these costs to present worth. The inputs to generate these economic multipliers are discount rate, inflation rates, etc., and are currently in the SOLCOST program. No further explanation will be given as to how to generate them in this development. By combining A-5 and A-6, equation A-7 is generated. It has been reorganized to indicate the various contributions to total thermal system cost expressed in present worth.

$$C_T = \underbrace{\sum_{i=1}^{N} [A_i(U_i' \, DD(C_L E_1 + C_f E_4) + C_{u_i} E_1)]}_{\text{conduction and shell costs}}$$

$$+ \underbrace{\sum_{j=1}^{M} [U_j \, \rho \, C_p \, DD(C_L E_1 + C_f E_4) + C_{v_j} E_1)]}_{\text{infiltration costs}} \qquad \text{(A-7)}$$

$$+ \underbrace{C_h E_1}_{\text{auxiliary fixed costs}}$$

DD = number of degree days for heating

To optimize the choice of insulation, windows, or doors and also the choice of infiltration lessening features, it is desired to minimize C_t as shown mathematically in A-8. That is, select the individual u's and v's from each subsection choice or select the door and select the appropriate insulation and select the appropriate windows that will minimize the function C_T. This is further simplified if the user is willing to assume a fixed fuel cost; i.e., the fuel cost per unit does not vary as the fuel requirements are increased. A simplification results. The individual minimization selection of the insulation can be done independently of the cost minimization to select the doors and so on and so forth and finally the minimization to select the infiltration conserving features.

$$\operatorname*{Min}_{u,v}(C_T);\quad u = 1, 2, 3, \ldots,\quad v = 1, 2, 3, \ldots$$

$$\text{if } C_f = \text{Constant}$$

$$\begin{aligned}\operatorname*{Min}_{U,V}(C_T) = {} & \operatorname*{Min}_{u}(C_T, \text{windows}) \\ & + \operatorname*{Min}_{u}(C_T, \text{doors}) \\ & + \operatorname*{Min}_{u}(C_T, \text{insolation}) \\ & + \\ & \;\,\vdots \\ & + \operatorname*{Min}_{v}(C_T, \text{infiltration})\end{aligned} \tag{A-8}$$

If the fuel cost per unit is not a constant and is a function of the amount consumed, this separation is no longer possible. An iterative procedure can be contrived, however, that allows one to assume a fixed fuel rate, do the minimizations separately for the windows, doors, and so on; then calculate the effective fuel cost based on the total fuel requirements, and repeat the whole procedure again and then keep repeating it until the load or the total amount of required energy converges. Since in the neighborhood of fuel utilization, fuel costs are fairly constant, this iterative procedure should converge rapidly (See Figure A-1).

For a solar system, the thermal cost in present worth (C_{T_s}) is shown in A-9.

$$C_{T_s} = \underbrace{(C_b + A_c\,C_a)\,E_1}_{\text{solar system costs}} + \underbrace{A_c\,C_o\,E_2}_{\substack{\text{solar system}\\ \text{operating costs}\\ \text{(area dependent)}}}$$

$$+ \underbrace{C_m\,E_3}_{\substack{\text{solar system}\\ \text{maintenance}\\ \text{costs}}} + \underbrace{L\,C_f\,E_4(1-F)}_{\substack{\text{auxiliary fuel}\\ \text{costs}}} + \underbrace{C_e\,E_1}_{\text{shell costs}} \qquad \text{(A-9)}$$

$$+ \underbrace{(C_h + L\,C_L)E_1}_{\substack{\text{auxiliary}\\ \text{installation}\\ \text{costs}}} \;;\quad L = UA(DD)$$

C_b = solar system fixed costs

C_c = solar system variable costs

A_c = collector area

C_o = solar system area dependent operating costs

C_m = solar system maintenance costs

F = solar load fraction

L is assumed to be a function of the UA and the number of degree days (DD). To minimize this function, that is, to choose the individual u's, v's and the collector area, becomes a more detailed task because of the coupling of the active solar system to the thermal envelope. However, there are simplifying porcedures.

If the first partial of A-9 is taken with respect to the collector area (see A-11), one can observe that it is independent of the thermal envelope cost and areas. C_e does not appear in this equation. However, L (the total load) does. Mr. Barley demonstrated in his relative areas work that in the neighborhood of a collector area, the partial of F with respect to A_c is approximately equal to a constant over the area of the collector (see A-12). If this is plugged into A-11, the optimum collector area (A_{opt}) is approximately equal to a constant multiplied by the load.

$$\underset{U,V,A_c}{\text{Min}} (C_{T_s}) \tag{A-10}$$

$$\frac{\delta C_{T_s}}{\delta A_c} = C_a E_1 + C_o E_2 - L C_f E_4 \frac{\delta F}{\delta A} \tag{A-11}$$

$$\frac{\delta F}{\delta A_c} \cong \frac{\text{Constant}}{A_c} \tag{A-12}$$

$$A_{opt} \cong \frac{C L C_f E_4}{C_a E_1 + C_o E_2} \tag{A-13}$$

As is shown in A-14, this reduces to: optimum area divided by load is a constant;

$$\frac{A_{opt}}{L} \cong \text{Constant} \tag{A-14}$$

a very valuable reduction. If the optimum area is calculated for one particular load, then it is observed that the load has changed slightly (let's say it is increased by 10%), then to get the new optimum collector area, the old area only need be multiplied by an appropriate scaling factor (in this case 1.1). If one is considering the module size in the optimum process, only candidate areas are to be considered which are integer multiples of the modular size, then it becomes necessary to check the module based collector areas in the neighborhood of the calculated optimum. The two adjacent candidate areas must be checked to see which one is optimum. It cannot be construed that either one would be optimal just because the calculated optimal area is closer to it. This reduction allows the user to, first of all, check the optimal load fraction, which is presently done automatically in the SOLCOST program and then to use the fraction derived and to minimize A-9. Once the optimal load fraction is known, the appropriate insulation, windows, and doors and infiltration conserving features can be easily determined. Once the optimal thermal shell features have been discerned, the new load can be calculated and then the optimal collector area can be scaled because the load fraction will remain virtually constant. However, this is only true if the cost of fuel is to remain a constant. If it is not to remain a constant, then the whole process becomes iterative as illustrated in Figure A-1.

This iterative procedure will save a lot of computer time because the alternative would be to, in a brute force manner, consider all of the possible combinations and then depict the one with the best economic advantage. To use the SOLCOST program and to evaluate all possible combinations would require possibly several hundred candidate sets to be considered. For example, if nine candidate collector areas are to be considered along with five candidate insulation values, three candidate windows, and three candidate doors, this would lead to 405 possible system combinations. Using the iterative procedures as described, many fewer system evaluations will be evaluated, saving a lot of time and money.

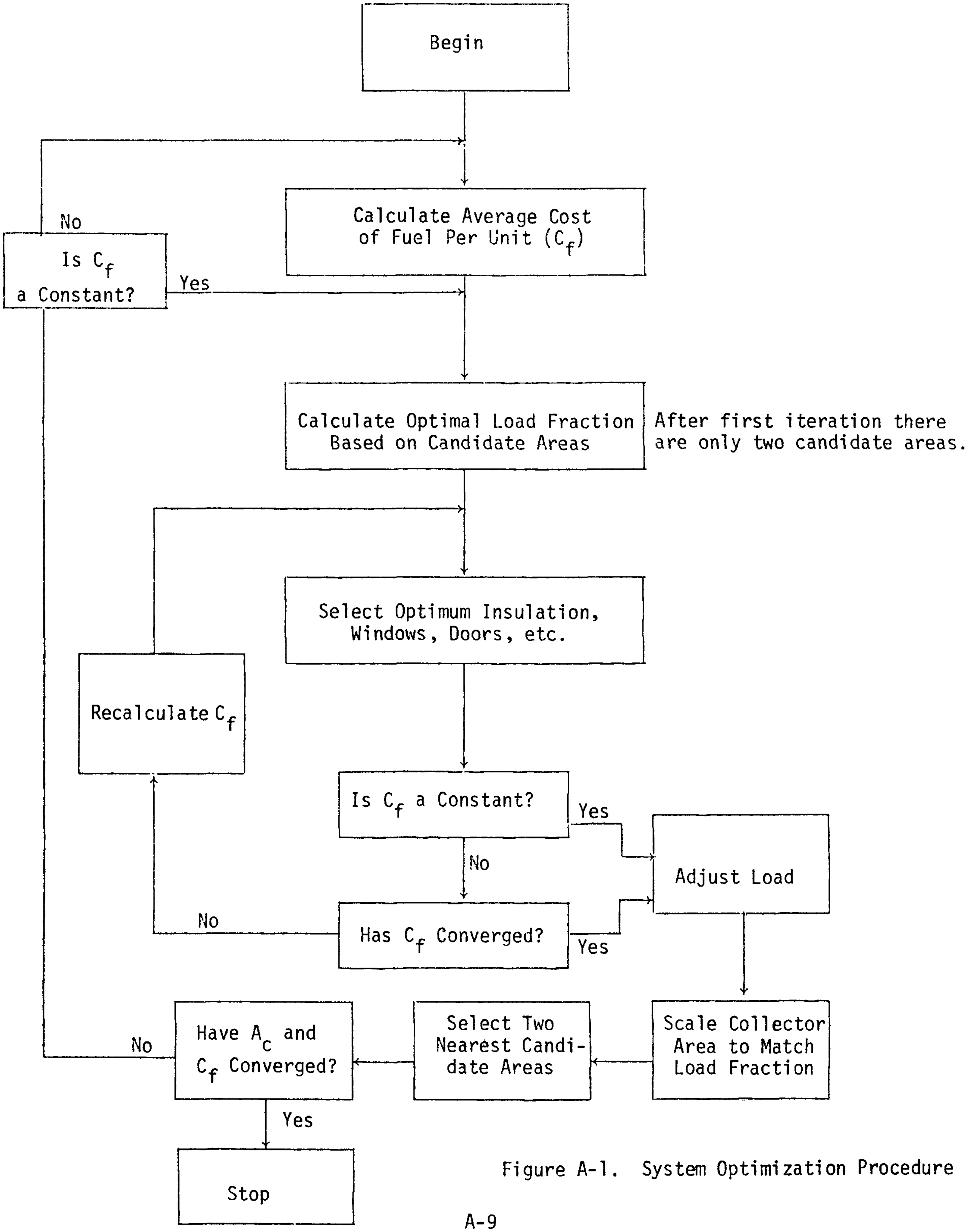

Figure A-1. System Optimization Procedure

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Appendix B

BASIC AIR ENGINEERING CONCEPTS

"Standard air" refers to dry air at a pressure of 14.7 psi, a temperature of 70°F, and a density of .075 lbm/ft^3. Standard air is used as a basis for rating air equipment,but non-standard air is most often encountered in real air systems. For instance, the average air pressure in Denver, Colorado is only 12.2 pounds per square inch (psi), air temperatures in home heating systems often are as high as 140°F, and air usually contains a certain amount of water vapor which also affects its density. All of these conditions affect the way in which air behaves in heating systems and must be accounted for in system design. To work with air at non-standard conditions, a few simple engineering concepts should be understood, which follow.

The air density ratio (ADR) is the ratio of the density of air at non-standard conditions (ρ) to that at standard conditions (ρ_{STD}) where the density is .075 lbm/ft^3. For example, air with a density of .062 lbm/ft^3 has an air density ratio given by Eqn. B.1.

$$\begin{aligned} ADR &= \rho/\rho_{STD} \\ &= .062/.075 \qquad (B.1) \\ &= 0.827 \end{aligned}$$

Since air temperature and pressure have the most significant effect on air density and atmospheric pressure varies closely with altitude, the air density ratio may also be expressed in terms of the air temperature and site altitude as given by Eqn. B.2

$$ADR = \frac{530}{(450+T)\ \exp\ (\frac{ALT}{27000})} \qquad (B.2)$$

where T is the air temperature in °F and ALT is the location altitude in feet. For example, the air density ratio for air at 100°F in Fort Collins, Colorado, which is located at an altitude of 5000 feet is computed as

$$ADR = \frac{530}{(460+100)\ \exp\ (\frac{5000}{27000})} = 0.786$$

The air density under these circumstances can then be found by Eqn. B.1.

$$\rho = (ADR)(\rho_{STD}) = (0.786)(.075) = .059\ \mathrm{lbm/ft^3}$$

The amount of water vapor in the air has a relatively minor effect on air density and may be neglected for design purposes.

SCFM is a measure of air flow in cubic feet per minute expressed in terms of standard air, that is, air having a density of .075 lbm/ft^3. Air flow can also be expressed in terms of its actual density if air is at non-standard conditions. ACFM is air flow in cubic feet per minute at the actual air density. These air flows are related by the ADR as shown in Eqn. B.3.

$$ACFM = SCFM/ADR \qquad (B.3)$$

For example, suppose 1000 actual cubic feet per minute of non-standard air having a density of .062 lbm/ft^3 (ADR = 0.827) is flowing in a duct. The flow rate can be described as 1000 ACFM or, in terms of standard air,

$$\begin{aligned} \text{SCFM} &= (\text{ACFM})(\text{ADR}) \\ &= (1000)(0.827) \\ &= 827 \text{ SCFM} \end{aligned}$$

Both of the terms 1000 ACFM and 827 SCFM describe the same air flow when the air density is 0.062 lbm/ft^3 (ADR = 0.827).

Air velocity in a duct may also be described in standard air velocity or actual air velocity, although actual air velocity is almost always used. For rectangular duct with dimensions A and B inches, the actual air velocity in ft/min is

$$V = \frac{(\text{ACFM})\text{x}(144)}{(\text{AxB})} \tag{B.4}$$

For round duct, with diameter D inches, the actual air velocity is

$$V = \frac{(183.35)(\text{ACFM})}{D^2} \tag{B.5}$$

(For standard air velocities, replace ACFM by SCFM in Equations B.4 and B.5.) To continue with the previous example, suppose that an air flow of 1000 ACFM is in a 12"x14" duct. The actual air velocity is then

$$V = \frac{(1000)\text{x}(144)}{(12\text{x}14)} = 857 \text{ fpm}$$

If the same air flow were carried in a 12" diameter round duct, the actual air velocity would be

$$V = \frac{(183.35)\text{x}(1000)}{12^2} = 1273 \text{ fpm}$$

A static pressure or head loss occurs whenever air flows through a duct, collector, furnace, or any other device. This pressure or head loss is due partly to the friction which occurs between the air and the surface of the device and partly due to air turbulence. Head or pressure is usually measured in inches of water, expressed as inches W.C. (water column) or inches W.G. (water gauge).

For every rectangular duct with dimensions A and B there is a round duct with diameter D which will have the same head loss when the same volume of air flows through each. These "equivalent" round and rectangular duct dimensions are related by Eqn. B.6.

$$D_E = 1.30 \frac{(AB)^{.625}}{(A+B)^{0.25}} \tag{B.6}$$

It should be noted that although these ducts have equal friction or head loss at equal air flow rates, the velocity of the air in each is different, the velocity in the rectangular duct always being less than the velocity in the round one.

The head loss through a duct can be caluclated by Eqn. B.7.

$$HL = \frac{(.027)(I.F.)(L)}{(D_E)^{1.23}(ADR)} \left(\frac{V_{RE}}{1000}\right)^{1.845} \tag{B.7}$$

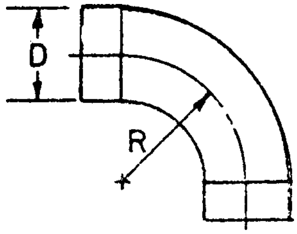
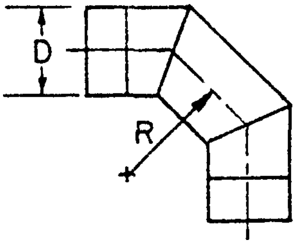
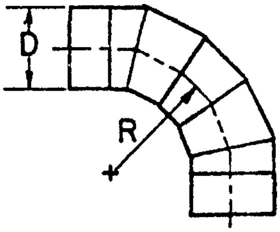
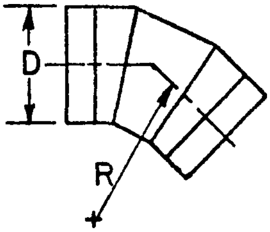
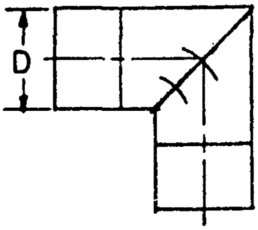
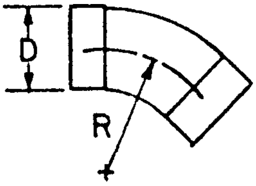
where HL is the head loss, inches W.G.; L is the equivalent duct length, ft.; ADR is the air density ratio of the air in the duct; D_E is the equivalent round duct diameter, inches; V_{RE} is the standard air velocity of air in an equivalent round duct, ft/min; and I.F. is the insulation friction factor. Some of these terms are explained in more detail below.

The equivalent length of a duct is the actual measured length plus an equivalent length added for each elbow in the duct. This is because elbows cause added head loss in an air duct which is most easily dealt with by considering each elbow to be a piece of straight duct with a length longer than its actual length. Equivalent lengths of round and rectangular duct elbows are given in Tables B.1 and B.2 respectively. For example, suppose a rectangular duct of width (W) 18" and depth (D) 6" contains an elbow with a radius of 1.25 D or 7.5". For this geometry, and a W/D ratio of 18/6 = 3, the first figure in Table B.2 indicates that the length/depth (L/D) ratio is 8. That is, the equivalent length of the elbow is 8 times the depth of the duct, in this case 8 x 6" = 48" or 4 feet. This length (4') is added to the true length of the elbow to compute the head loss.

The insulation friction factor (I.F.) has a value of (1.0) if thermal insulation is not used or if it is applied to the outside of a duct and the inside of the duct is smooth steel. If, however, thermal insulation is applied on the inside of a duct, its roughness causes the air head loss in the duct to be increased and I.F. is greater than one. This increase in friction is usually expressed as a factor by the insulation manufacturer. Some thermal duct liners increase the head loss by only 5% (i.e., I.F. = 1.05) but others may increase it as much as 40% (I.F. = 1.40). If thermal insulation is to be applied to the inside of a duct, the insulation friction factor (I.F.) should be obtained from the insulation manufacturer before the duct system is designed so that the ducts will not have an excessive head loss after installation.

Table B.1

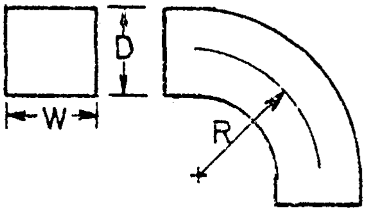
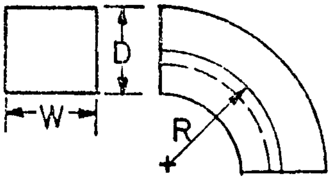
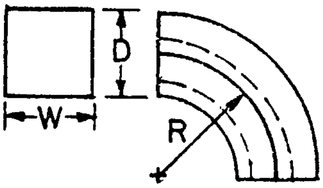
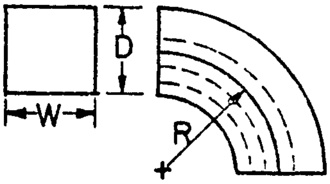
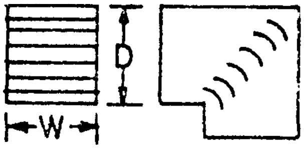
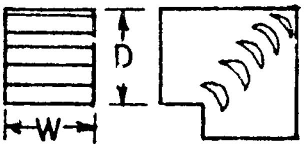
Equivalent Lengths of Round Duct Elbows

	ELEMENT	L/D RATIO
90° SMOOTH ELBOW R/D = 1.5		6
90° 3-PIECE ELBOW R/D = 1.5		17
90° 5-PIECE ELBOW R/D = 1.5		12
45° 3-PIECE ELBOW R/D = 1.5		6
90° MITER ELBOW R/D = .5		22 Vaned 65 Not Vaned
45° SMOOTH ELBOW R/D = 1.5		4.5

L is in Ft.
D is in Ft.

$L = D x \left(\frac{L}{D}\right) \text{ratio}$

Table B.2
Equivalent Lengths of Rectangular Duct Elbows

DESCRIPTION	ILLUSTRATION	W/D	L/D RATIO
RADIUS ELBOW (No Vanes) R = 1.25D		0.5	5
		1.0	7
		3.0	8
		6.0	12
RADIUS ELBOW 1 Vane R = 0.75D		0.5	8
		1.0	10
		3.0	14
		6.0	18
RADIUS ELBOW 2 Vanes R = 0.75D		0.5	7
		1.0	8
		3.0	10
		6.0	12
RADIUS ELBOW 3 Vanes R = 0.75D		0.5	7
		1.0	7
		3.0	8
		6.0	10
RECTANGULAR SQUARE ELBOW SINGLE THICKNESS TURNING VANES		0.5	8
		1.0	10
		3.0	12
		6.0	13
RECTANGULAR SQUARE ELBOW DOUBLE THICKNESS TURNING VANES		0.5	6
		1.0	8
		3.0	9
		6.0	10

L is in Ft.
D is in Ft.

$$L = D \times \left(\frac{L}{D}\right) \text{ Ratio}$$

There are two important design constraints for air distribution systems. These are air velocity and the friction rate. Too high an air velocity in a duct causes noise problems while too large a friction rate causes the blower motor to be excessively large and expensive to operate. The designer is required to set design limits on both the air velocity and friction rate in a duct system corresponding to the use of the system. For instance, residential duct systems will have different design requirements than industrial ones.

Residential solar and air distribution systems usually are specified with a velocity limit of 1000 ft/min and an air friction rate of .08 inches W.G. per 100 feet of equivalent duct length. These recommended standards are based on extensive experience with residential solar and air distribution systems and should not be changed considerably in order to achieve good system performance. Smaller duct sizes are specified if either the velocity or friction factor design limits are increased. However, the system may be noisy in the first case and expensive to operate in the latter. Presently the largest single problem with air solar heating systems is undersized ducts. Using the design limits of 1000 ft/min for velocity and .08" W.G./100' as a friction factor will ensure properly sized ducts for the solar system.

COLLECTOR MANIFOLD DESIGN

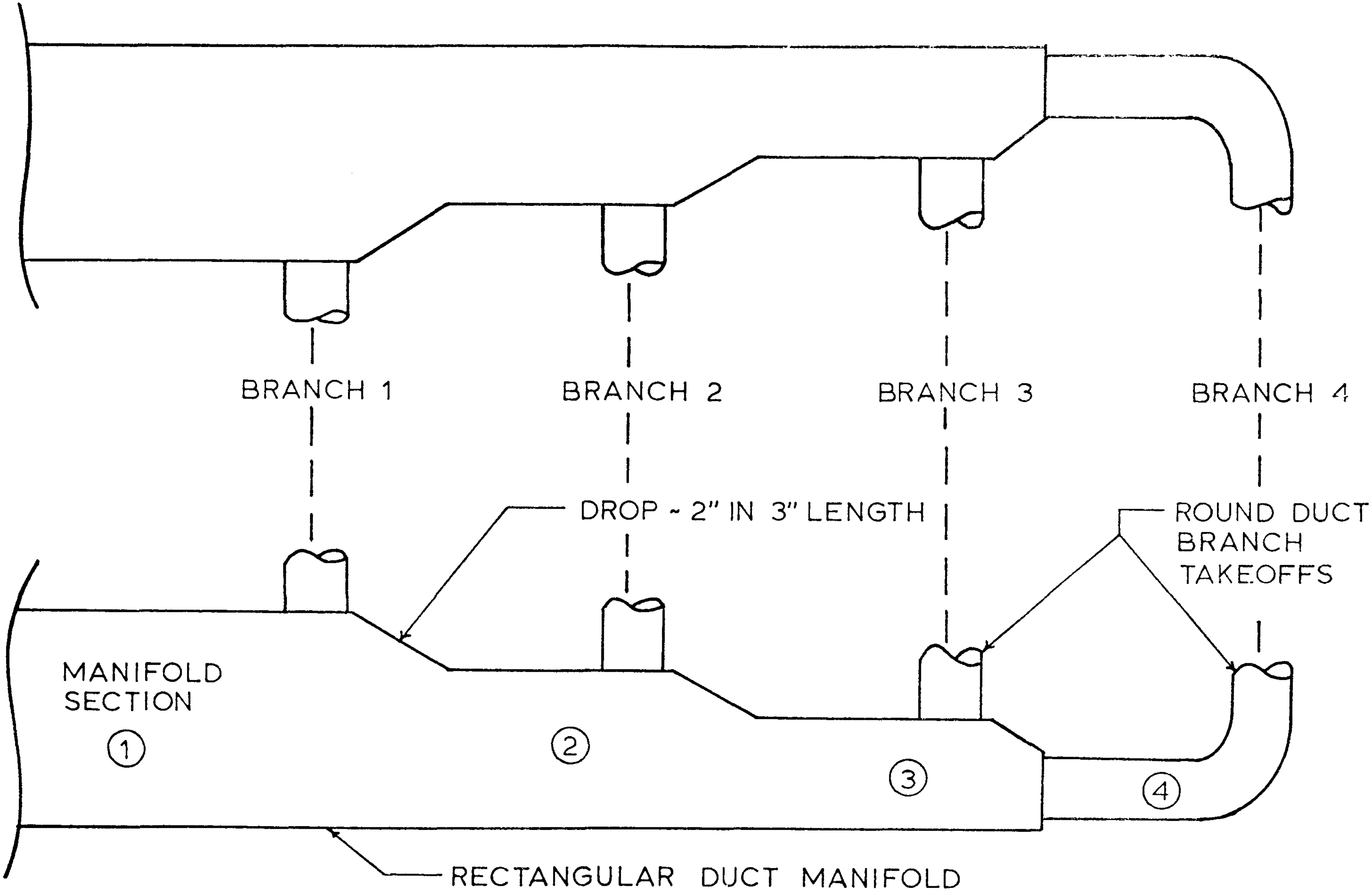
An array of collectors must have supply and return manifolds which are properly sized to assure even distribution of air to all of the collectors. If manifolds are improperly designed, some collectors in the array may receive so little air that they can contribute little or no heat to the system and the cost of purchasing and installing them will have been wasted. In this case other collectors will have higher than their design air flow which improves their performance but also greatly increases the pressure drop across the array causing more electrical energy to be consumed when the system is operating. High air flow near the collector temperature sensor can also cause erratic control problems for the solar system.

Some collectors are internally manifolded to a certain degree. At least one air collector manufacturer makes a collector which is completely internally manifolded for a 30 collector array, so that only one connection each for the supply and return ducts is needed in the center of the array. Other manufacturers internally manifold four collectors so that one connection each for the supply and return ducts is required for every four panels. And, still other collectors have no internal manifolding at all and a supply and return duct connection is required for each collector. A collector branch will be defined as a set of collectors which are internally manifolded together so that only one supply and one return connection is required for that branch. If no internal manifolding is built into a collector system, then each collector is a branch.

A typical collector manifold is shown in Figure B.1. Note that in this figure the supply and return ducts are located on the same side of the collector array. This is usually the most convenient arrangement, because both ducts can be located in one chase through the building. However, if a particular building plan is more suited to having the supply and return ducts entering and leaving the collector array on opposite sides, that arrangement may also be used.

Figure B.1 shows a solar system manifold with four branches. If the collectors used with this system are designed with 6 collectors internally manifolded, then this manifold is good for a 24 collector panel array. If no internal manifolding is built into the collectors, then the four branch manifold system shown in Figure B.1 is suitable for only four collector panels. The manifolds are typically built with rectangular duct and have only one dimension reduced at a branch takeoff. Branch takeoffs connecting the manifold to the collector are usually round duct of the appropriate size. High pressure take-off fittings should be used and caulked with a good silicone caulk to ensure against leaks. Duct tape should not be used near collectors as it will deteriorate quickly and allow leaks to occur after the system is operated for a time.

Sizing the manifold sections is as easy as sizing any duct in the solar system. The design air flow rate through each section of the manifold is calculated based on even air distribution to the collectors,



TYPICAL SOLAR COLLECTOR AIR DISTRIBUTION MANIFOLD

Figure B.1

HEAD LOSS IN SOLAR SYSTEM COMPONENTS

Once the major components in a solar system have been sized, the head (or pressure) loss through each needs to be computed in order to choose a blower, operating speed, and motor which will supply the proper static pressure for the system. The following components are grouped for convenience of head loss calculations.

1. Collector Systems Ducts - all ducts associated with the solar heating of air.
2. Collector and collector manifold
3. Rock Box
4. Filters, dampers, duct coils, etc., any restriction to air flow in a duct system.

Collector System Ducts

The head loss in collector system ducts is a simple matter to compute if the head loss per 100' of duct was recorded when the duct was sized. In this case the head loss per 100' need only be multiplied by the total equivalent length of that duct and divided by 100. For instance, if a collector return duct was selected which has a head loss of .078" W.G./100', an actual duct length of 35' with 3 elbows, and an additional 12' of equivalent length due to the elbows, the total equivalent length is 35+12=47 feet and the head loss through the duct is

$$\frac{.078'' \text{ W.G.}}{100} \times 47' = .037'' \text{ W.G.}$$

Collector and Collector Distribution Manifolds

The head loss of a single collector panel is usually given by the collector manufacturer for standard air conditions. This nominal head loss needs to be multiplied by the number of collector panels which will be connected in series to compute the total standard air head loss, if not already done so by the manufacturer. For instance, if a collector is rated at a 0.12" W.G. head loss and air flows through 2 panels in series, then the nominal head loss of the collector is 2.0.12" W.G. = 0.24" W.G. Altitude and temperature effects must also be accounted for as well as the head loss in the collector distribution manifolds. Altitude and temperature effects are easily dealt with as will be seen, but a complete analysis of collector manifold pressure changes is very difficult due to the numerous branching dynamic losses, static regains and acceleration losses. Numerous investigations of collector and manifold pressure losses, however, have yielded the following expression for the static head loss across the complete solar collector array and manifold distribution system $(HL)_c$:

$$HL_c = 1.25(HL)_{NOM}/ADR \qquad (B.8)$$

where $(HL)_{NOM}$ is the nominal head loss across the number of collectors in series as specified by the manufacturer. For instance, suppose 24 collector panels are installed in an array such that every 2 panels are connected in series. The manufacturer specifies a head loss of 0.12" W.G. per panel or 2x0.12" W.G. = 0.24" W.G. for two panels in series. The collectors are located at an altitude of 4500 feet and the average design temperature for solar heated air is 120°F.

The ADR in this case is 0.774. The static pressure drop across the entire collector and collector distribution manifold system can now be caluclated.

$$HL_c = \frac{(1.25)(0.24)}{0.774} = 0.388'' \text{ W.G.}$$

Rock Box

The pressure drop across a rock box has been carefully studied and a somewhat complex equation has been found to predict its value. It is not given, but it was used to create the following example.

Example

A rock box is to be used for heat storage in a solar system using 260 ft^2 of collector with a design air flow of 520 SCFM. The required volume of a rock box in cubic feet is usually taken as one half of the collector size in ft^3 so this rock box has a volume of ½x260=130 ft^3. The box height is chosen to be 5' so that it fits easily into a 7½' basement with room for top and bottom manifolds. The cross sectional area of the box is therefore 130/5=26 ft . This is accommodated by making the box length and width 5'-1", since this is approximately 5.1'x5.1'=26 ft . The rock to be used in the box varies in diameter from 3/4" to 1½" and the system is located 2000 feet above sea level. The head loss across the rock box was calculated to be 0.225' W.G. for a flow rate of 520 SCFM.

Suppose the rock box is part of a two blower system as shown in Figure B.2 and the air flow through the house is 750 SCFM. (The collector design flow rate is 520 SCFM which is the flow rate through the rock box when the collectors are heating storage. When storage is heating the house, in this case, the rock box design flow rate is 750 SCFM.) The pressure drop for a rock box can be reevaluated by using the new air flow rate.

The head loss across the rock box when heating the house with an air flow of 750 SCFM was determined to be 0.410" W.G.

Filters, Dampers, Duct Coils, etc.

Any device placed in a duct system will have a pressure drop specified by its manufacturer. These head losses are specified for a design air flow rate or range of flow rates and are usually given for air at standard conditions. To account for air flows other than that given by the manufacturers as well as non-standard air conditions, use the analysis procedure previously shown and then interpolate between the manufacturers data points.

HEAD LOSS IN RESIDENTIAL AIR DISTRIBUTION SYSTEM

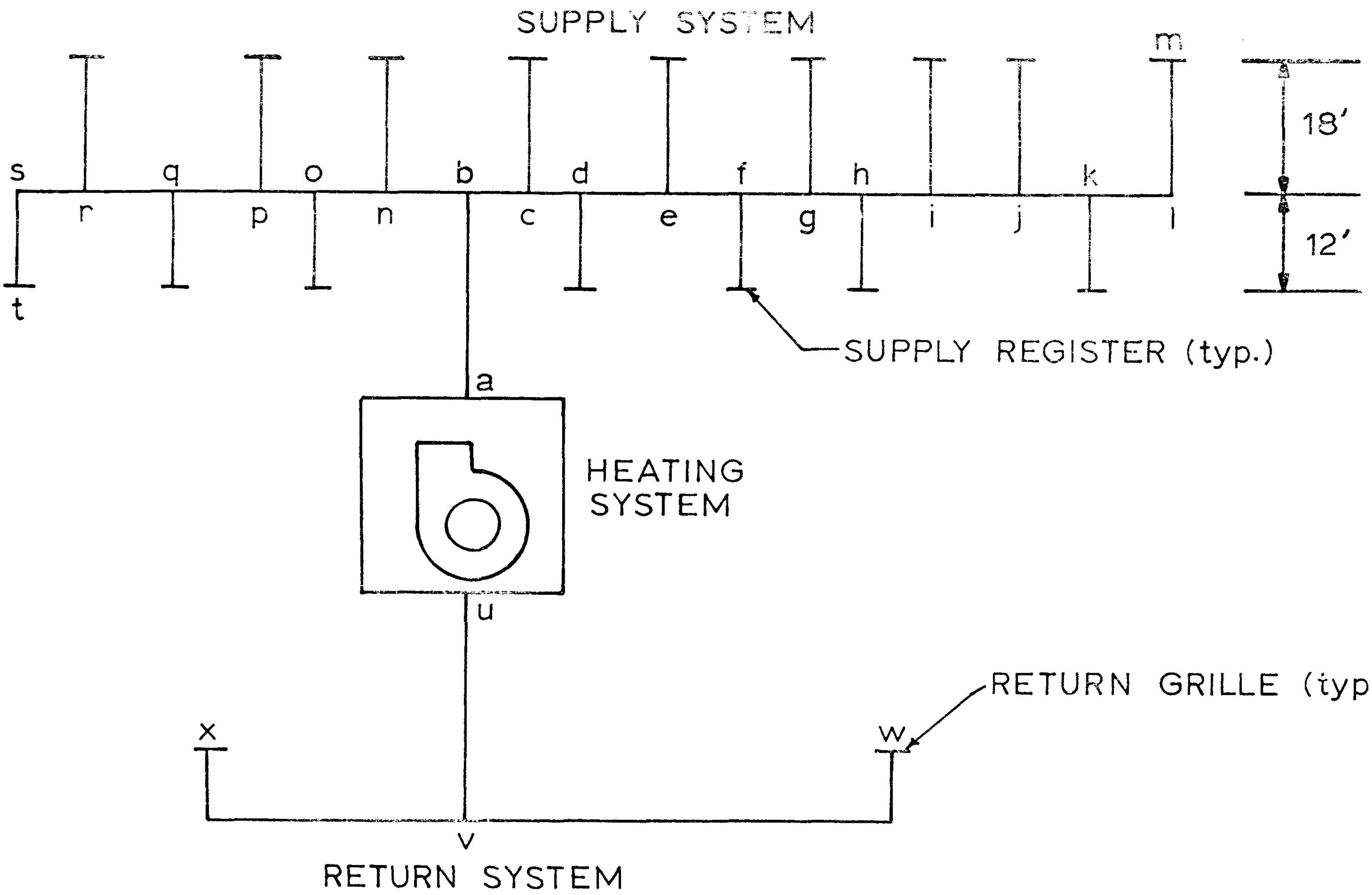
One "component" of a solar heating system which has not yet been dealt with is the house air distribution system. The head loss of the distribution system at the design flow rate sometimes must be known in order to properly identify the solar system blower requirements.

If the house distribution air flow is similar to that of the collector and the distribution system static head loss is nearly equal to or less than that of the collector system, a single blower system is adequate if no other air conditioning is needed. In any other case a two-blower series or parallel arrangement should be used. In the series situation, the second blower must be capable of moving air through the house distribution system and the rock box simultaneously. In the parallel situation, the second blower need only supply the proper flow rate and head for the air distribution system. This feature of the parallel system simplifies the design since the house air distribution system head loss need not be computed. The air distribution system can be designed and installed as any heating system without solar would be, matching the duct size to the furnace output. Most heating system contractors are fully capable of doing this without computation.

No strict engineering techniques are generally employed in the design of a residential air distribution system. Rather, most installers size the ducts in a residential system by certain "rules of thumb" and by their past experience. In such a case the installer may be able to supply a reasonable estimate of the air distribution system head loss based on his design technique and past experience. Often, such an estimate is adequate for the solar system designer. If a more certain analysis is required, however, the distribution system head loss must be calculated.

For example, the residential air distribution system shown in Figure B.2 was taken from a set of blueprints for a residential structure. The solar system designer wishes to calculate the head loss of the distribution system at a design flow rate of 1200 SCFM. The house is to be located 1500 feet above sea level and the design air temperature is taken as 120°F.

The only part of the system which needs to be analyzed is that part which has the largest head loss. The rest of the system will be balanced by partially closing the register dampers so that too much air does not pass through the shortest part of the distribution system. This is easily done by using the duct design methodology previously illustrated.



RESIDENTIAL AIR DISTRIBUTION SYSTEM (EXAMPLE)

Figure B.2

Appendix C

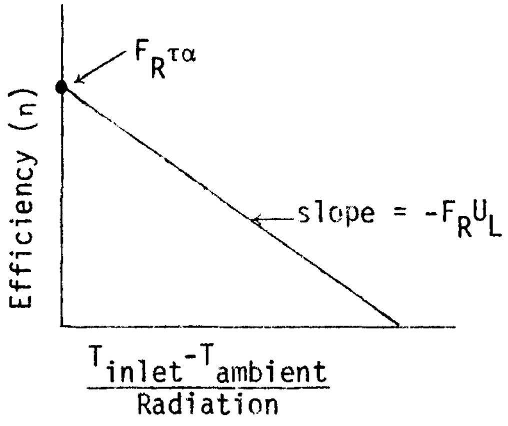
COLLECTOR PARAMETER MODIFICATION

General Description

There are five different situations of importance that can arise which could require redefinition of the collector efficiency curve. They are:

1) The efficiency parameters have been defined in terms of different variables than the ones required by SOLCOST which needs $F_R\tau\alpha$ and F_RU_L.
2) The flow rate has been varied from the conditions under which the $F_R\tau\alpha$ and F_RU_L were empirically determined.
3) Two or more collectors with different efficiency parameters are connected in parallel. This can occur by using different flow rates in the same type of collector.
4) Two or more collectors are coupled in series. They may be the same or different collectors.
5) The final parameter change of importance is caused by putting a heat exchanger between the collectors and storage.

Before showing how the efficiency parameters can be easily adjusted to reflect the above mentioned perturbations, the SOLCOST collector efficiency methodology must be defined so that the user can understand how the parameters are modified. The SOLCOST program assumes a linear efficiency curve for the collector system. The curve has two parameters. The first is $F_R\tau\alpha$ which is the curve intercept when $(T_{inlet}-T_{ambient})/$Radiation is equal to zero. See the following figure.



The second parameter is F_RU_L which is the negative of the slope of the efficiency line. More specifically the efficiency (η) is defined:

$$\eta = F_R\tau\alpha - F_RU_L\left(\frac{T_{inlet}-T_{ambient}}{Radiation}\right).$$

In reality this efficiency representation is only valid for one collector and under precisely the same conditions that the $F_R\tau\alpha$ and F_RU_L were derived empirically. If the panel flow rate ($\dot{m}$) or the specific heat of the fluid (c_p) are changed, if two or more collectors are placed in series, if two or more disimilar collectors are placed in parallel, or if a heat exchanger is included in the collection loop, these parameters must be adjusted. The new parameters can be easily estimated using basic heat transfer principles and are stated below

Definition of F_R as a function of inlet temperature

The efficiency of a collector can be alternatively defined as a function of T_{fluid} instead of T_{inlet}. To further complicate the issue the fluid temperature can be approximated by an arithmetic average of T_{outlet} and T_{inlet} or $(T_{inlet} + T_{outlet})/2$. If the fluid temperature is defined to be an average of the inlet and outlet temperatures then the F_RU_L (T_{inlet}) and $F_R\tau\alpha$ (T_{inlet}) can be defined as follows:

$$F_R\tau\alpha\ (T_{inlet}) = \frac{F_R\tau\alpha\,(T_{fluid})}{1 + \frac{F_RU_L\,(T_{fluid})\,A_c}{2\dot{m}\,c_p}}$$

$$F_RU_L\ (T_{inlet}) = \frac{F_RU_L\,(T_{fluid})}{1 + \frac{F_RU_L\,(T_{fluid})\,A_c}{2\dot{m}\,c_p}}$$

where A_c is the area of the collector plate, $\dot{m}$ is the collector panel total flow rate and c_p is the specific heat of the fluid. The flow rate and specific heat area the same as used to empirically define $F_R(T_{fluid})$. These must be provided by the manufacturer and must be the ones used when the panel was tested. If efforst have been taken to obtain the bulk fluid temperature then the $F_R\tau\alpha$ and F_RU_L ahve an alternative definition. That is:

$$F_R\tau\alpha = F'\tau\alpha\left(\frac{\dot{m}\,c_p}{F'U_L\,A_c}\right)\left(1-\exp\left(-\frac{F'U_L\,A_c}{\dot{m}\,c_p}\right)\right)$$

$$F_RU_L = \frac{\dot{m}\,c_p}{A_c}\left(1-\exp\left(-\frac{F'U_L\,A_c}{\dot{m}\,c_p}\right)\right)$$

where $\dot{m}$, c_p, and A_c are as defined above. F' is the collector efficiency factor; it is the effective transmittance-absorptance product and U_L is the overall loss coefficient. For flow rates that are commonly used in collectors, either method of correction should give approximately the same answer. Care must be taken to ensure that the units check.

Adjust $F_R\tau\alpha$ and F_RU_L for Different Flow

To redefine the $F_R\tau\alpha$ and F_RU_L products to reflect different flow capacitance rates is a simple process. Two things can happen to cause the flow capacitance rate ($\dot{m} c_p$) to change. They are: the flow rate ($\dot{m}$) could change or the fluid could be altered or changed to cause the specific heat (c_p) to change.

The following equation has superscript numbers which reflect the state of conditions; i.e., state 1 and state 2. The required relationship is as follows:

$$\frac{F_R\tau\alpha_1}{F_R\tau\alpha_2} = \frac{F_RU_{L_1}}{F_RU_{L_2}} = \frac{A_cF_RU_{L_1}/\dot{m}c_{p_1}}{1 - \left(1 - \frac{F_RU_{L_1}A_c}{\dot{m}c_p}\right)\frac{\dot{m}c_{p_1}}{\dot{m}c_{p_2}}} \cdot \frac{\dot{m}c_{p_1}}{\dot{m}c_{p_2}}$$

Parallel Collectors of Different Efficiencies

Occasionally collectors of different efficiencies are connected in parallel. They could be the same collector but with different flow rates. The following relationship will transform the efficiency parameters:

$$F_R\tau\alpha = \frac{\sum_{i=1}^{n} A_{c_i} F_R\tau\alpha_i}{\sum_{i=1}^{n} A_{c_i}}$$

$$F_RU_L = \frac{\sum_{i=1}^{n} A_{c_i} F_RU_{L_i}}{\sum_{i=1}^{n} A_{c_i}}$$

where i designates the ith parameter for n parallel collectors.

Series Collector Combination

When two or more collectors of the same or different type are connected in series the resultant efficiency parameters will most likely be different than the ones for any one collector. This comes about by the fact that the inlet temperatures to the individual collectors are different. The resultant

$F_R\tau\alpha$ and $F_R U_L$ are:

$$F_R\tau\alpha = \frac{\dot{m}\,c_p}{A_{c(total)}} \sum_{i=1}^{n} \{\frac{F_R\tau\alpha_i}{F_R U_{L_i}} [1-\exp(-\frac{A_{c_i} F'U_{L_i}}{\dot{m}\,c_p})] \cdot \exp[-\sum_{j=i+1}^{n} \frac{A_{c_j} F'U_{L_j}}{\dot{m}\,c_p})]\}$$

$$F_R U_L = \frac{\dot{m}\,c_p}{A_{c(total)}} [1 - \exp(-\sum_{i=1}^{n} \frac{A_{c_i} F'U_{L_i}}{\dot{m}\,c_p})]$$

where $A_{c(total)}$ is the total series collector area, $\dot{m}\,c_p$ is the capacitance flow rate through each of n collectors and A_{c_i} is the i^{th} collector area. The sum of the individual A_{c_i} must equal $A_{c(total)}$. A helpful equation to remember for calculating $F'U_L$ is:

$$F'U_{L_i} = -\frac{\dot{m}\,c_p}{A_{c_i}} \ln(1-\frac{F_R U_{L_i} A_{c_i}}{\dot{m}\,c_p})$$

Modification Due to Heat Exchangers

$F_R'\tau\alpha$ and $F_R' U_L$ are terms used as effective efficiency parameters whenever heat exchangers are included between the collection and storage systems. The inclusion of a heat exchanger in general degrades the overall collection efficiency by increasing the collector inlet temperature. In general the designer picks the heat exchanger depending on how much of a collection efficiency penalty he is willing to withstand. This is done by defining the ratio F_R'/F_R, which must be less than one.

$$\frac{F_R'}{F_R} = \frac{1}{1+y(x-1)}$$

$$y = \frac{A_c F_R U_L}{\dot{m}\,c_{p(collector)}}$$

$$x = \frac{\dot{m}\,c_{p(collector)}}{E\,\min(\dot{m}\,c_{p(collector)}, \dot{m}\,c_{p(storage)})}$$

where E is the heat exchanger effectiveness which, with the flow rates, defines the heat exchanger. If the F'_R/F_R ratio is picked by the designer and the flow rates are established, the heat exchanger has been defined. The exchanger can be selected from this information. Likewise, the F'_R/F_R ratio can be defined by knowing the flow rates and the heat exchanger effectiveness.

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