双层 Low E 玻璃

CHAPTER 4
STEADY-STATE THERMAL ANALYSIS

4-1. Introduction

The mechanisms of conductive, convective, and radiative heat transfer affect thermal exchanges between a building and its ambient surroundings. Heat is transferred through walls, the ceiling and roof, doors, windows, and the floor. Walls may be entirely above ground, entirely below ground, or a combination.

Models to describe heat flow along these paths have been developed and are widely used. We will focus on steady-state models. Numerous time-dependent models can be found in the research literature, and some are being adopted in engineering practice. That is happening in the design of thermally massive commercial and industrial buildings and some residences, but has not yet been widely adopted for most agricultural buildings.

Because heat transfer basics have been presented in Chapter 3, this chapter will concentrate on examples to demonstrate calculations of steady state heat transfer through the components of a building's structure.

4-2. Heat Transfer Through Walls

Design and construction practices can make walls of buildings complex to analyze if heat transfer through them is considered in fine detail. Fortunately, various simplifying assumptions have been found adequate and provide sufficiently accurate calculations for most design purposes.

P399

Types. Many wall types are encountered in building practice. Appendix 4-1 shows several common designs. Perhaps the most common is the first shown in the appendix, a wood-framed wall with an inside sheathing to protect the wall, an outside sheathing also for strength and mechanical protection, and siding to protect the wall from weather.

间柱

The cavity between studs in a wall can be left empty, if no added insulation value is desired, or can be partially or totally filled with insulation. Insulation is commonly used even when energy savings are not a concern. If a wall is insulated, its inner surface will be warmer during cold weather and condensation on the inner surface will not be as severe a problem as for an uninsulated wall. In agricultural buildings, insulation may be limited to that provided by outside sheathing rather than added insulation in the wall cavity.

Insulation. Insulation can be in several forms. Mineral fiber (glass wool, mineral wool) is commonly used. The strands of such insulation suppress air circulation within the pore spaces so that heat moves primarily by conduction.

Air is a good insulator against thermal conduction.

When mineral fiber insulation is used, care must be taken to prevent moisture condensation within the wall cavity. If excessive free water forms, insulation will become compact and its insulation value lost forever. Care must be taken to prevent wind or an exhaust ventilation system from forcing outside air slowly through small leaks and through the stud cavity. That is, air infiltration must be avoided. If air infiltrates through mineral fiber insulation, insulation acts as an effective air filter but not as insulation.

Foamed insulation is also frequently used in agricultural buildings, either as foam boards or formed in place using foam. Foam has the advantage of a high insulation value. If wet conditions are expected, closed-cell foam is recommended to prevent moisture saturation and serious loss of insulation value. If foam is to be used where sunlight can strike it, it must be covered for sunlight slowly degrades foam by accelerating the oxidation process. When foam contains a gas other than air (to achieve a higher insulation value), the R-value used in design should be the "aged" value. This means the value after sufficient time has passed to permit a representative fraction of gas to leak out of the foam and be replaced by atmospheric air.

Loose fill such as mineral fiber, cellulose, and vermiculite can also be used as insulation. However, the moist conditions of agricultural buildings can cause rapid and severe damage to such insulations. When an insulation such as cellulose fiber is wetted, it compacts rapidly and drying will not restore it.

Sheathing. The inner sheathing on a wall acts primarily to protect the wall from mechanical damage (and water spray, etc.). The outer sheathing provides mechanical protection and strength, and more recently materials have been used which provide a significant degree of thermal insulation as well.

R-values. A wood-framed wall is a combination of series and parallel thermal circuits. We have seen in Chapter 3 how to combine series and parallel thermal circuits into an overall R-value. These concepts will now be applied to engineering design problems. The thermal bridging effect of framing can be a significant factor in reducing the effective R value of a building shell.

The parallel thermal circuit feature of wood-framed walls arises because heat can transfer either through the wall by way of the framing or through insulation. Framing comprises a significant fraction of a wall. Not only wall studs, but sills and plates at the floor and ceiling, bracing, framing for windows and doors, and other uses of wood in the wall can be as much as 20% of a wall's cross-section. A lightly framed wall can be as little as 12% wood. Typical values for the area over wood are: 15% for walls framed from 38 x 140 mm (2 x 6) lumber, and 20% for 38 x 90 mm (2 x 4) lumber. Wall studs of larger lumber can be spaced farther apart. These data can be made more exact by knowledge of (or specifying) the wall framing.

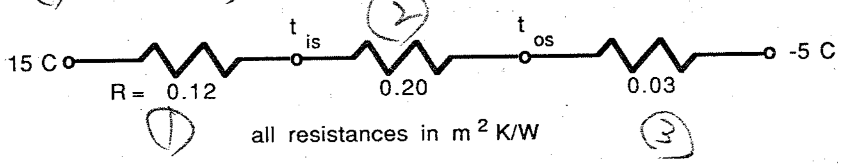
Example 4-1

Problem: The walls of a dairy barn have been constructed of (three oval core) concrete blocks 203.2 mm thick (8 in. blocks). Determine the R-value of the wall and estimate the temperatures of the inside and outside surfaces when it is 15 C indoors and - 5 C outdoors. (Note: Conversion from IP to SI units should not increase the number of significant digits. However, ASHRAE data in Appendix 3-2 will be our database, so we will use the SI dimensions as listed in that table.)

Solution: Appendix 3-2 contains R-values of building materials. Appendix 3-5 has surface resistances to be used in building heat loss calculations. The problem concerns only a series thermal circuit.

P388 In Appendix 3-2, concrete blocks with three oval cores, cast using sand and gravel aggregate, are in the section titled: Masonry Units. Conductance for a block 203.2 mm thick is listed as 5.11 W/m^2 K. (Note that light-weight aggregate and cinder blocks, with more air in pore spaces, have a conductance roughly 40% lower, but are not used as frequently because of their lower strength.) The thermal resistance of the block is the inverse of conductance and is given as 0.20 m^2 K/W.

P397 Surface resistance values (in Appendix 3-5) for a vertical wall apply. Surface emittance will be assumed to be 0.90, and winter conditions with a wind velocity of 6.7 m/s will be assumed. For these conditions, surface resistances of the inside and outside surfaces of the wall are 0.12 and 0.030 m^2 K/W, respectively.



The series thermal circuit is comprised of three resistances with resistance values of 0.12, 0.20 and 0.03 m^2 K/W. The three total a resistance of 0.35 m^2 K/W. The thermal circuit is shown above where t_{is} and t_{os} are the temperatures of the inside and outside surfaces. In steady-state heat transfer, temperature differences scale linearly with resistance differences, thus,

$$t_{is} = 15 - (0.12 / 0.35)(15 - (-5)) = 8.1 \text{ C, and}$$
$$t_{os} = 15 - ((0.12 + 0.20) / 0.35)(15 - (-5)) = -3.3 \text{ C.}$$

Example 4-2

Problem: A wood-framed wall is to be constructed with an inside sheathing of plywood, an outside sheathing of vegetable fiber board, and plywood siding. A question has arisen whether it will be necessary to specify full-thickness glass wool insulation (which means insulation to fill the wall cavity) or whether a lesser thickness will be sufficient. 間柱 保溫棉

The desired wall insulation value is 2 m^2 K/W. Framing members are 88.9 mm thick and 38.1 mm wide (2 x 4 wall studs). Inside sheathing is to be 15.88 mm thick Douglas fir plywood, outside sheathing is to be 12.70 mm thick regular density vegetable fiber board, and the siding is to be 11.11 mm thick hardboard. Winter design conditions are assumed. P.386

fir 縱樹 壁板 牆板

Solution: Framing has not been specified in detail, assume wood to fill 20% of the wall's cross section. The desired R-value of 2.0 m^2 K/W includes the effect of framing. Full thickness insulation will be assumed for now. Surface emittances are assumed to equal 0.90 (typical construction materials).

Each of the two parallel heat transfer paths must be analyzed. The R-values can be obtained from Appendix 3-2 for construction materials and Appendix 3-5 for surface coefficients. Values are as shown below.

Heat transfer paths through the insulation and framing are in parallel with each square meter of wall having 0.8 m^2 of insulated area and 0.2 m^2 of wood-framed area. The overall unit area R-value is, thus,

$$R_{overall} = 1.0 / (0.8 / 2.86 + 0.2 / 1.15) = 2.20 \text{ m}^2 \text{ K/W}.$$

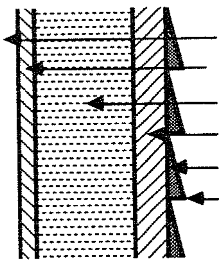
$\frac{1}{2.2} = \frac{0.8}{2.86} + \frac{0.2}{1.15}$ 2.0

The insulation value is slightly greater than desired. To bring the wall to an R-value of 2, the insulated part must have an R-value of

2.45

$$R_{insulated} = 0.8 / (1.0 / 2.0 - 0.2 / 1.15) = 2.45 \text{ m}^2 \text{ K/W}.$$

牆面 支架



outdoors	R-value, m^2 K/W between framing	over framing
inside surface	0.12 (a)	0.12 (a) P397
inside sheathing	0.14 (a)	0.14 (a) P386
wall cavity	2.29 (b)	0.58 (c)
outside sheathing	0.23 (a)	0.23 (a) P386
siding	0.05 (a)	0.05 (a) P389
outside surface	0.03 (a)	0.03 (a) P397
TOTAL	2.86	1.15

2.86 ↓ 0.41 2.45 2.29 − 0.41 1.88

Notes:
(a) obtained directly from data tables
(b) obtained directly from data table for 88.9 mm thickness → P.386
(c) assume Dougles fir will be the framing lumber, and its R-Value will be an average of data in Appendix 3-2 (6.5 mK/W)(0.0889 m) = 0.58 m^2 K/W

why not 8.68

To reduce the insulated part of the wall to an R-value of 2.45 m^2 K/W, the R-value within the cavity must be reduced from 2.29 to 1.88 m^2 K/W. This would equal the R-value of the insulation plus any resulting airspace created if the insulation were not full thickness.

Blanket insulation is not available in a continuous gradation of thicknesses. The next step below full thickness (88.9 mm, or 3.5 in.) is 50.8 mm (2 in.). This would leave an airspace 38.1 mm thick. The R-value of blanket insulation 50.8 mm thick is approximately 1.31 m^2 K/W. The airspace must, thus, have the equivalent of $1.88 - 1.31 = 0.57$ m^2 K/W insulation value.

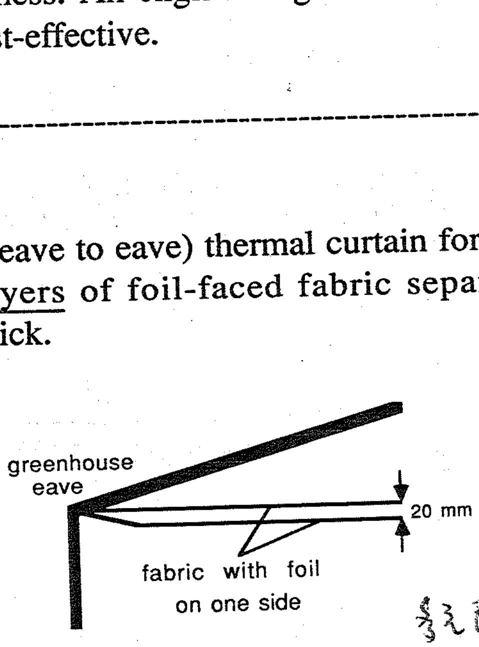
P398 Appendix 3-6 contains data for R-values of airspaces. The airspace is vertical with horizontal heat flow. The temperature difference across the airspace will likely be between 5.6 and 16.7 K, and the mean temperature will be approximately 10 C in most buildings. Data in the appendix show it is possible to attain an airspace R-value of 0.56 only if the temperature difference across the space is low and the effective emittance of the cavity is very low.

To attain an effective emittance of 0.03 or 0.05 requires one surface forming the airspace to have a surface emittance of nearly zero. This is not practical. The best one can expect is to use foil-backed insulation. The foil has an effective emittance of at least 0.20, making the effective emittance of the cavity approximately 0.20. This does not achieve the required insulation value for the wall.

In conclusion, one would specify full thickness insulation; this meets the required R-value of the wall. To specify less would make the wall slightly less insulated than desired. Of course, the outside sheathing could be changed to obtain more insulation value and permit the wall insulation to be reduced to, perhaps, 50.8 mm thickness. An engineering economic analysis could reveal which would be most cost-effective.

Example 4-3

Problem: A horizontal (eave to eave) thermal curtain for a greenhouse is to be designed using two layers of foil-faced fabric separated by an airspace approximately 20 mm thick.



The fabric in the curtain is a loose-weave polyester skrim faced with aluminum foil on one side. The fabric itself has negligible thickness. The surface emittance of the foil side of the fabric is approximately 0.20 and the other side has an effective emittance of approximately 0.60. The two fabric layers can be oriented so

1. each layer has its foil side up,
2. each layer has its foil side down,
3. the top layer has the foil face up and the bottom layer has the foil face down, and
4. the bottom layer has the foil face up and the top layer has the foil face down.

Which of the four orientations provides the largest R-value?

Solution: The fabric was stated to have negligible thickness, thus, all the thermal resistance of the thermal curtain is provided by surface resistances (top and bottom) and the resistance of the airspace between the layers.

Appendices 3-5 and 3-6 contain data for surface resistances and airspace resistances, respectively. A horizontal surface (Appendix 3-5) with heat flow upward (the function of a thermal curtain in a greenhouse is to conserve heating energy) has a surface coefficient which is a function of surface emittance. Data for surface conductance as a function of emittance form a straight line when graphed. P397

The equation for the line is

$$h_s = 4.02 + 5.82\varepsilon \qquad (4\text{-}1)$$

For surface emittances, ε, of 0.20 and 0.60, surface conductances, h_s, are 5.18 and 7.51 $W/m^2\ K$, respectively, making surface resistances 0.19 and 0.13 $m^2\ K/W$, respectively. 软料书. 取倒数,

The airspace can be formed with each surface facing into the space having an emittance of 0.2 or 0.6, or it can be formed with one surface having an emittance of 0.2 and the other 0.6. Resulting effective emittances (see Equation 3-74) of the airspace are shown in the table below.

P398. Appendix 3-6 will be used to obtain airspace R-values. The mean temperature of the airspace will be assumed to be approximately 10 C and the temperature difference across the space will be assumed to be 5.6 K. The airspace is 20 mm thick or approximately 19.1 mm. The airspace is horizontal and heat flow is up.

Interpolation must be used to obtain values (with R-value units of $m^2\ K/W$) and are as follows:

Effective emittance = 0.11:

$$R = 0.39 + (0.11 - 0.05)(0.30 - 0.39)/(0.20 - 0.05)$$
$$= 0.35\ m^2\ K/W$$

	ε, side one	ε, side two	effective emittance
1.	0.20	0.20	0.11
2.	0.60	0.60	0.43
3.	0.20	0.60	0.18

Effective emittance = 0.43:

$$R = 0.30 + (0.43 - 0.20)(0.20 - 0.30)/(0.50 - 0.20)$$
$$= 0.22 \text{ m}^2 \text{ K/W}$$

Effective emittance = 0.18:

$$R = 0.39 + (0.18 - 0.05)(0.30 - 0.39)/(0.20 - 0.05)$$
$$= 0.31 \text{ m}^2 \text{ K/W}$$

The four possible orientations of the layers and the resulting R-values are found in the table below.

	Orientation of foil face		R-values			
	top layer	bottom layer	top surface	bottom surface	air space	total
3.	up	up	0.19	0.13	0.31	0.63
4.	down	down	0.13	0.19	0.31	0.63
2.	up	down	0.19	0.19	0.22	0.60
1.	down	up	0.13	0.13	0.35	0.61

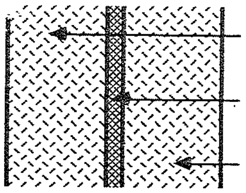
Differences among the four possibilities are not great and would not be expected to be so. However, the best arrangement is to have both layers facing the same direction. Whether they face up or down does not appear to make a difference based on what has been calculated. However, field experience shows it best if both layers have foil sides up. Why? Think about it. Hint: The data for surface coefficients are based on the assumption the surrounding mean radiant temperature equals the air temperature. In a greenhouse, glazing temperature will be significantly lower than greenhouse air temperature during cold weather.

Note: The above example is based on the assumption the surface characteristics of the thermal curtain material do not degrade (as would happen with dust accumulation). Such degradation may overwhelm any small advantages as were calculated.

Example 4-4

Problem: A refrigerated apple storage building is to be designed. The walls will be constructed as a composite of 101.1 mm thick concrete blocks as an inner wall, the same as an outer wall, and polyurethane foam boards will be used between the inner and outer walls to provide insulation. The desired R-value of the wall is 2.5 m^2 K/W. Design the wall using steady-state analysis to meet this goal. (In reality, the wall's mass may cause transient analysis to be used in practice.)

Solution: The R-value contributions of the surfaces and concrete blocks cannot



be changed. The thickness of foam boards is the one variable over which there is control in the design.

Winter conditions will be assumed and the concrete blocks will be specified in the design (by you, the design engineer) to be three oval core blocks, sand, and gravel aggregate. Component R-values are available in Appendix 3-2 and surface resistances are in Appendix 3-5. Only a series thermal circuit is involved for the concrete block walls and foam boards form a homogeneous plane.

see P.388 P397

The concrete blocks have R-values of 0.12 m^2 K/W apiece. The inside surface will be assumed to have a surface emittance of 0.90. The inside and outside wall surface resistances are thereby 0.12 and 0.03 m^2 K/W, respectively. The series thermal circuit consists of the foam boards, two concrete blocks, the inside surface, and the outside surface. The contributions of the blocks and surfaces add to

$$0.12 + 0.12 + 0.12 + 0.03 = 0.39 \text{ m}^2 \text{ K/W}$$

Note how the thermal resistance of the inside surface is as great as that of each block. Masonry has very little thermal resistance.

An insulation value of 2.5 m^2 K/W is desired; 2.5 - 0.39 = 2.11 m^2 K/W must be provided by the insulation. Appendix 3-2 lists the R-value of unfaced cellular polyurethane which has been expanded (foamed) using R-ll refrigerant; the value is 43.38 m^2 K/W per meter thickness. Note R-ll does not refer to the insulation value of the foam board. Refrigerant R-ll can be used to create the foam and provides better insulation value than would air. P387.

The thickness of foam board which is required is 49 mm (2.11 m^2 K/W)/(43.38 mK/W). As the engineer responsible for the design, you would choose and specify a standard thickness of unfaced polyurethane board with at least this much R-value.

Note in this calculation no thermal resistance was assigned to the areas of contact between the foam boards and concrete blocks. A mastic would likely be used to hold the foam board onto one of the walls during construction and contact would be continuous with both concrete block walls after the wall is completed. The exact construction practice would dictate the extent of contact resistance; good practices would result in very small values. Thus, it is conservative design to ignore contact resistances in most situations. 樹脂 規定

Finally, note that polyurethane foam has a thermal resistance greater than has polystyrene. However, it is also likely to be more expensive. In a complete engineering design, the cost of each to attain the desired R-value could be determined, and the least cost design chosen.

4-3. Heat Transfer Through Ceilings

Ceiling construction is generally simpler than walls. A sheathing forms the ceiling, and insulation (if any) is placed above the sheathing. In modern agricultural buildings, attic spaces are generally unused and sheathing (attic flooring) is not placed above the insulation.

Less framing lumber is used in a ceiling. Generally, the ceiling is attached to the underside of roof trusses. The fraction of framing is generally assumed to be 0.1 when trusses are spaced at 0.4 m and 0.07 when spaced at 0.6 m.

The effect of the roof is frequently assumed to be negligible when calculating heat loss though a ceiling. Standard engineering practice calls for attics to be well ventilated to prevent moisture accumulation. This is especially important in barns where the animal housing space is frequently very humid during cold weather. If the attic is well ventilated, attic temperature will be near outdoor air temperature. When heat passes through the ceiling into the attic air, it is carried to the outdoors by ventilation rather than conduction through the roof. In this case, the roof effect is insignificant.

During hot weather, much of the heat transmission down through a ceiling results from thermal radiation emitted by the under side of the roof and absorbed by the upper side of the ceiling. It has been estimated that as much as three-quarters of the heat gain through a ceiling is due to this radiation effect. Data and procedures to estimate this effect may be found in the *ASHRAE Handbook of Fundamentals.*

Data to calculate the thermal resistance of ceilings is found in Appendix 3-2, and the procedure to calculate resistances is identical to calculations for walls, as in Section 4-2. Note how the direction of heat flow affects the calculated R-value of a ceiling.

4-4. Heat Transfer Through Glazings

4-4.1. General. The technical term for a light-transmitting opening in a building is fenestration. Greenhouses, obviously, are almost entirely light transmitting. Barns, such as for dairy cows and swine, usually have some windows to provide light during the day. Other agricultural buildings such as poultry houses and refrigerated horticultural storages have no windows.

Common glazing materials are glass or plastic sheets. Double-walled rigid plastic sheets and air inflated double polyethylene sheets are frequently used for greenhouse glazing. Windows in barns are usually glass. Double glass for insulation can be used to control condensation on the inside surface, if desired. In barns, the windows need not be openable for the ventilation system should be designed to provide all necessary fresh air. In fact, opened windows usually defeat the ability of a well designed ventilation system to provide good air distribution.

Glazings act both to admit solar radiation and to transmit heat between the inside and outside of the building. Procedures to calculate solar transmittance are described in more detail in Chapter 5.

4-4.2. R-Values. Overall coefficients of heat transfer have been determined for several common glazing materials. Appendix 4-2 contains glazing data for windows and greenhouse glazings.

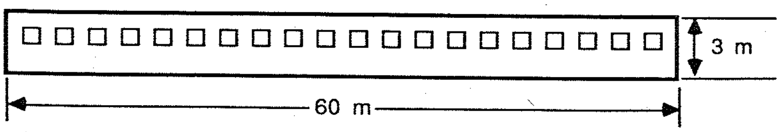
P 400

It should be noted the data for windows apply only to conditions listed in footnotes of the appendix. The conditions are relatively extreme and, thus, should be used to determine peak heating or cooling loads. If used to determine season-long heating or cooling needs, the needs will be overestimated. The data include the effects of both convection and radiation from surfaces and radiation effects grow nonlinearly with increasing temperature differences.

It should also be noted that heat loss coefficients for glazings typically include the effects of inside and outside surface resistances. Single glazings have little thermal resistance in the glazing itself; almost all the resistance is provided by the surfaces.

Example 4-5

Problem: The wood-framed wall described in Example 4-2 is to be constructed with the full thickness of insulation and have an average R-value of 2.20 m^2K/W. The wall will be 3 m high and 60 m long. Double glazed windows of 3 mm glass separated by 6 mm airspaces will be installed. The windows will be in wood frames and will be 1.0 m wide and 0.6 m high. There will be 20 of the windows in the wall. What will be the expected R-value of the windows and when the windows are installed what will be the average R-value of the wall with its windows? Assume cold weather conditions.



Solution: From Appendix 4-2, for the windows as described, the unit area thermal resistance is 0.30 m^2 K/W. Wood frames during winter provide an adjustment factor between 1.00 and 1.11. We do not have exact data from the window manufacturer, so assume an average adjustment factor of 1.05. This brings the unit area thermal resistance to 0.32 m^2 K/W.

Total window area is

$$A_{windows} = 20(1.0 \times 0.6) = 12 \text{ m}^2,$$

and total wall area, less windows, is

$$A_{wall} = (3 \times 60) - 12 = 168 \text{ m}^2.$$

Heat loss through the windows and wall forms a parallel thermal circuit and Equation 3-28 applies. The sum of wall and window area is 180 m^2, and

$$180 / R_{average} = 12 / 0.32 + 168 / 2.20$$

from which the average unit area thermal resistance, $R_{average}$, is calculated as 1.58 m^2 K/W.

This is a significant decrease of the wall's insulation value even though the window area is relatively small. This effect is one of the problems with windows when energy conservation is a goal. Even double glazing or storm windows do not compare well with an insulated wall section in terms of R-value.

It would be a useful exercise to repeat this calculation assuming only single glazed windows in aluminum frames are to be used.

---

ex: $\frac{180}{R_{avg}} = \frac{12}{0.16 \cdot \left(\frac{0.914}{2}\right)} + \frac{168}{2.2}$, $R_{avg} = 1.162 \ \frac{m^2 K}{W}$

4-5. Heat Transfer Through Doors

Doors are often a significant factor in calculating heat loss from agricultural buildings, although they may comprise only a small fraction of total wall area. Flush doors are used for permitting people to enter and leave and overhead (usually panel) doors permit entry of equipment and animals. Unit area thermal resistances of doors are listed in Appendix 4-3. The data include surface resistances and interpolation and moderate extrapolation are reasonably accurate for conditions different from those listed in the footnotes of the appendix.

p401

Example 4-6

Problem: The wall described in Example 4-5 is to be modified to include an overhead door 2.5 m high and 3 m wide and a flush door 2.5 m high and 1 m wide. The overhead door will be a panel door with 11 mm thick panels, 35 mm thick frame, with no glazing, and the flush door will be a 35 mm thick solidcore door with no glazing. Calculate the effect of the doors on the average R-value of the wall for winter conditions with windows and doors included. No storm doors are to be installed.

Solution: Appendix 4-3 contains the data needed to solve this problem, a further example of parallel heat transmission.

The unit area thermal resistance of the flush door and panel door are 0.45 and 0.31 m^2 K/W, respectively. The respective areas are 2.5 and 7.5 m^2. Adding the doors reduces the framed wall area to 168 m^2 - 10 m^2 = 158 m^2. Windows are not affected. The average R-value can be calculated using Equation 3-28.

window wall flush door panel door

$$
\begin{aligned}
180 / R_{average} &= 12 / 0.32 + 158 / 2.20 + 2.5 / 0.45 + 7.5 / 0.31 \\
&= 37.5 + 71.8 + 5.6 + 24.2 \\
&= 139.1 \text{ W/K.}
\end{aligned}
$$

2.86

The average unit area thermal resistance, $R_{average}$, is 1.29 m^2 K/W.

This shows the potentially significant effect of doors. The average R-value of the wall has changed from 1.58 to 1.29 m^2 K/W – an 18% decrease. What was originally an insulated wall (Example 4-2) with an R-value of 2.86 through the insulation, reduced to an R-value of 2.20 by the effect of the framing, reduced again to an R-value of 1.58 because of the windows, and then dropped to 1.29 when doors were added. The insulation value of the wall when averaged with its windows and doors is only 45% of the insulated section. If energy conservation were a goal in the design, it would be logical to search first for means to increase the averaged R-value by changing the windows and doors rather than by choosing the more expensive approach of redesigning the wall to add more insulation.

framing ↓ 2.20 windows ↓ 1.58 ↓ doors ↓ 1.29

The heat loss factor calculated above for the wall (139.1 W/K) is a sum of four contributions. The relative magnitude of each contribution is an indication of its importance. In this example, windows contribute 27% of the heat transmission though the wall (37.5/139.1), the wall contributes 52%, and the flush and panel doors contribute 4% and 17%, respectively.

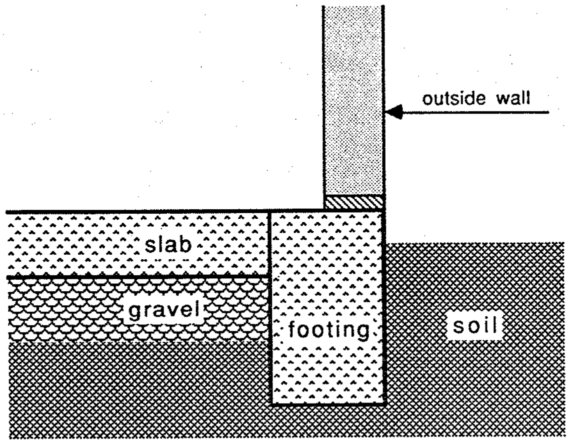
4-6. Heat Transfer Through Floors on Grade

寬度大的厚平板

Many agricultural buildings are built on concrete slabs and have no basement. The perimeter of the slab is likely to have a concrete foundation wall extending down to a concrete footing below frost line. Insulation may or may not be a part of the perimeter.

Heat transmission through the perimeter is likely to be an important factor in barns, especially those with well insulated walls, but will be less significant in greenhouses. Perimeter heat loss is also likely to be significant in horticultural product storages where above-ground walls and the ceiling are well insulated.

Thermal comfort, however, may be a factor in some barns and greenhouses. Small animals, such as calves and pigs, penned along an outside wall may suffer significantly from chilling when lying on a cold floor. Plants along the outside wall in a greenhouse, if on the floor rather than on benches or in ground beds, may suffer from low root temperatures during cold weather. When this factor is expected to exist, perimeter insulation may be recommended.



An unheated floor slab has been found to lose heat primarily from its perimeter rather than from the interior part of the building. If a building on a slab is heated, soil under the interior parts of the building is warmed by heat loss from the building. The thermal conduction path to cold soil surrounding the building is long and the heat does not transfer quickly away from the central regions. In contrast, heat transmitted through the floor near the perimeter can move readily to the cold soil surrounding the building. In terms of heat conduction, the outer two meters of perimeter are the only parts of the floor which are important.

Experimental evidence indicates heat loss from perimeters of buildings is proportional to the length of the perimeter and the temperature difference between air inside the building and out. If steady-state conditions are assumed, heat loss from the perimeter can be estimated by

$$q_{floor} = FP(t_{inside} - t_{outdoors}), \tag{4-2}$$

where q_{floor} is heat loss in W; P is the building perimeter, m; t is air temperature; and F is an experimentally determined perimeter heat loss factor, W/m K.

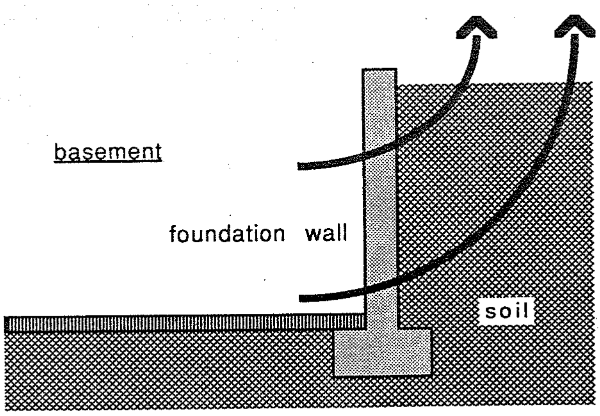
Values of F for an uninsulated and unheated slab floor on grade range between 1.4 and 1.6 W/m K, depending on the severity of the winter. The factor is reduced to between 0.8 and 0.9 W/m K when insulation of R = 0.95 m^2 K/W is installed from under the edge of the slab down to the top of the footing (applied to the outside surface of the foundation wall, or the inside surface before the slab is poured, see Figure 4-1 for the first case). Within the ranges of the perimeter factor, the large values apply when winters are less severe and air temperature differences are smaller.

More complete data for perimeter heat losses in agricultural buildings should be based on highly detailed computer analysis; finite element analysis works well.

Computer analysis has shown the integrity of insulation around a perimeter is critical in determining the effectiveness of the insulation. Thermal bridges (Figure 4-2) formed by concrete in the slab or foundation wall seriously detract from energy conservation efforts. If, for example, the slab is in physical contact with the foundation wall, the conduction of heat through that bridge may reduce heat savings by 20 to 30%, compared to construction where the joint between the slab and the foundation is insulated.

4-7. Heat Transfer Through Basement Walls and Floors

Heat loss from heated basements is complicated by the two- and three-dimensional nature of the conduction heat transfer process. Heat lost from the upper parts of a basement wall has a much shorter path length to the outside air than does heat lost from near the floor. Heat loss from the floor is influenced more by soil temperature than by outdoor air temperature, except indirectly through the long-term influence of air temperature on ground temperature.



To permit estimates of basement heat loss, empirical factors have been determined to be used in calculations. The factors are used in a heat loss equation

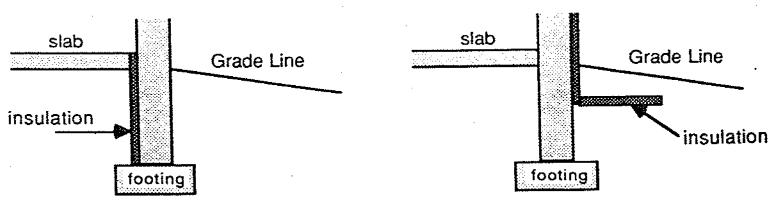


Figure 4-1. Examples of perimeter insulation placement.

$$q_{ground} = U'(t_{inside} - t_{ground}) \qquad (4\text{-}3)$$

where q is heat loss in W, t is temperature in C, and U' is an integrated conductance over the area of heat loss, having units of W/K. Note that Equation 4-3 is written to express heat loss in terms of ground temperature rather than outdoor air temperature.

Basements are not deep enough to be surrounded by soil at constant temperature. Essentially constant temperatures are not reached until 10 m below ground surface, thus, t_{ground} must be estimated and chosen to represent the heating season.

土壤年均温 = 空气年均温

Yearly average soil temperature approximately equals yearly average air temperature. Winter soil temperature is offset from yearly average temperature by a decrement. Figure 4-3 can be used to determine the magnitude of that decrement. The decrement, subtracted from the yearly average, is the heating season soil temperature.

Yearly average soil temperature in a region frequently is the same as the temperature of water from deep wells in the region. For example, locate the region of Central New York State on the map in Figure 4-3. It is just southeast of Lake Ontario at approximately 75° west longitude and 43° north latitude. The average yearly temperature in the region is approximately 10 C. The map in Figure 4-3 shows the decrement which represents winter soil temperatures is 10 C. Thus, the winter design soil temperature is 0 C, which is used as t_{ground} in Equation 4-3.

Table 4-1 provides data to estimate the value of U′ for both basement walls and floors. The data for walls is both for uninsulated construction and for three levels of possible insulation. The floor is assumed to be not insulated.

Example 4-7

含水箱, 工作区

Problem: A packing shed for a fruit farm is to have a basement for storage. The basement is to be 8.5 m by 12 m, and the basement floor will be 1.8 m below grade. The shed is to be located in the state of Washington at 120° west

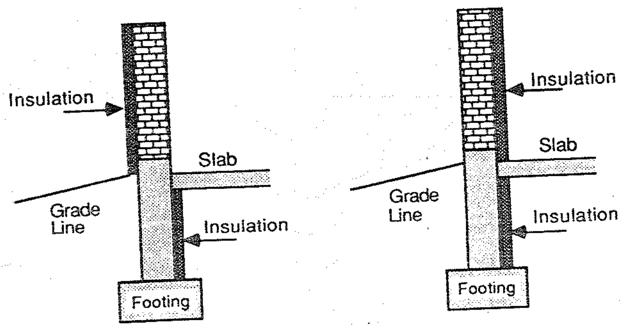


Figure 4-2. Examples of insulation added to slab floors and footings which form thermal bridges.

longitude and 45° north latitude. The average yearly temperature is 12 C. The basement will be maintained at 15 C and the walls will not be insulated. Estimate the rate at which heat will be lost from the basement by conduction through the walls and floor to the soil.

Solution: Heat loss through the walls and floor can be estimated using the data in Table 4-1. Conductances for walls between 0 and 1.8 m can be averaged to obtain the average conductance per unit length of wall perimeter. The average of the six values is

$$U_{average} = (2.33 + 1.26 + 0.88 + 0.67 + 0.54 + 0.45)/6 = 1.02\ \mathrm{W/m^2K}$$
$$\cancel{= 1.02\ \mathrm{W/m^2K} \times 1.8\ \mathrm{m^2/m} = 1.84\ \mathrm{W/mK}}.$$

The perimeter of the basement is 41 m, thus, the wall's contribution to the total conductance, U′ to be used in Equation 4-3, is

$$U'_{wall} = (\overset{1.02 \times 1.8\ \mathrm{W/mK}}{\cancel{1.84\ \mathrm{W/mK}}})(41\ \mathrm{m}) = 75\ \mathrm{W/K}.$$

The shortest width of the basement is 8.5 m, thus, for a depth below grade of 1.8 m the unit area conductance is 0.14 $\mathrm{W/m^2K}$. The floor area is 102 $\mathrm{m^2}$ and the floor's contribution to the total conductance is

$$U'_{floor} = (0.14\ \mathrm{W/m^2K})\ (102\ \mathrm{m^2}) = 14\ \mathrm{W/K}.$$

Therefore, the total conductance, $U'_{basement}$, is 75 + 14 = 89 W/K.

From the map in Figure 4-3, the decrement of ground temperature below

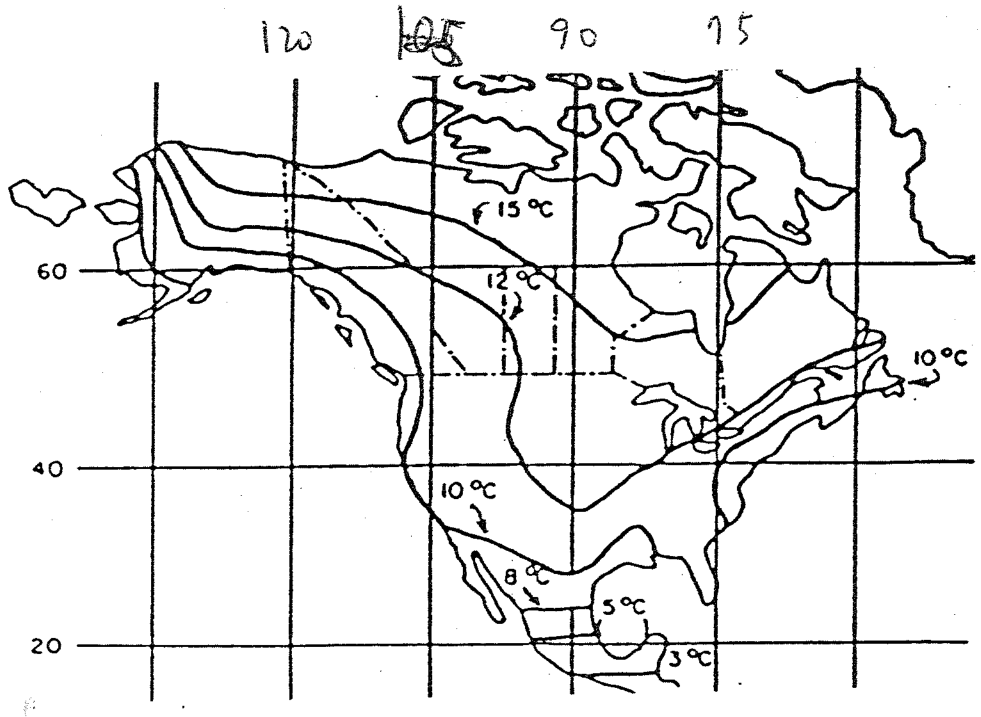


Figure 4-3. Isotherms of the decrement of soil temperature below average yearly soil temperature during the heating season. Adapted from the 1989 *ASHRAE Handbook of Fundamentals*.

average yearly temperature is approximately 10 C, thus, ground temperature is estimated to be 2 C during the heating season.

The rate of heat loss to the ground, computed using Equation 4-3, is

$$q_{ground} = (89\ W/K)(15\ C - 2\ C) = 1157\ W.$$

The heat loss from the part of the basement wall above grade must be added to obtain the total basement heat loss, of course.

4-8. Program RVALUE

The total conductance of a building's shell, ΣUA, summed over the walls, ceiling, windows and doors, is important in determining thermal behavior of the building. Sections of this chapter have shown how to obtain the components needed to calculate this sum.

Good design should include investigating many possible options and selecting the best based on overall economic considerations and other factors. Many options exist when designing a building to meet a thermal standard. Can the ceiling be heavily insulated (which is less expensive to do than heavily insulating walls), and the walls lightly insulated and achieve the same thermal standard? Can insulative outer sheathing be substituted for insulation within a wall cavity and achieve the same thermal standard at a lower cost? Or vice-

Table 4-1. Heat loss conductance for basements x

Through walls

Depth below grade	Conductance, W/m²K, for wall insulated by: uninsulated	R = 0.73	1.47	2.20 m²K/W
0 - 0.3 m	2.33	0.86	0.53	0.38
0.3 - 0.6	1.26	0.66	0.45	0.36
0.6 - 0.9	0.88	0.53	0.38	0.30
0.9 - 1.2	0.67	0.45	0.34	0.27
1.2 - 1.5	0.54	0.39	0.30	0.25
1.5 - 1.8	0.45	0.34	0.27	0.23
1.8 - 2.1	0.39	0.30	0.25	0.21

Through the floor, conductance in W/m²K

Depth of floor below grade	Minimum width of basement, m: 6.0	7.3	8.5	9.7
1.5 m	0.18	0.16	0.15	0.13
1.8	0.17	0.15	0.14	0.12
2.1	0.16	0.15	0.13	0.12

Notes:
a. Assumes thermal conductivity of the soil is 1.38 W/mK.
b. Uninsulated wall assumed to be masonry.
c. Data adapted from the *ASHRAE Handbook of Fundamentals*.

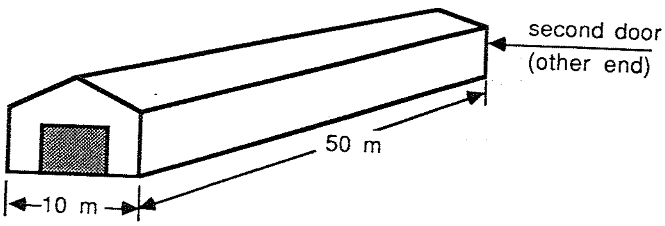
versa? If window area is significant, can the cost of double glazing be avoided by adding more or different insulation to the walls or ceiling and still have the same overall building U value? Or, can windows be double glazed and walls insulated less for lower total cost and the same overall building thermal characteristics? These are only some of the questions a designer might ask while deciding the best design for a specific application.

The effort involved in calculating ΣUA for a building is significant, and precludes repeated calculations to answer the "what if"questions so important to good design. In addition, a lack of time to complete a wide selection of designs works against obtaining an intuitive feeling for the importance of various factors in determining building thermal behavior. Numerous thermal analysis programs for buildings have been written for small computers and are becoming commercially available. They can be invaluable tools for saving time in thermal analysis. If less time is required for analysis, more time is available for design, which should be the main focus of engineering.

Program RVALUE is an executable file provided as an example of the utility of a building thermal analysis program. It is not a general purpose program intended for general building design; it is too limited. However, RVALUE can be used to obtain some understanding of how parts of a building interrelate thermally to yield an overall thermal resistance, R, and an overall thermal conductance, ΣUA. Instructions for using the program are included within the menus provided the user.

Example 4-8

Problem: A building (10 by 50 m, with 3 m high sidewalls) is to be built. Windows will be single glass, normal emittance. The building will have two, 2.5 by 4 m, 44 mm thick, panel doors with 11 mm thick panels. The ceiling will be sheathed on the underside with 15.88 mm plywood and insulated with 150 mm of mineral fiber blanket. The walls will have 12.7 mm plywood as inside sheathing, 88.9 mm of mineral fiber blanket as the cavity insulation, 19.8 mm regular density vegetable fiber board as outside sheathing, and 15.88 mm plywood as siding. Wall framing is estimated to constitute 20% of the wall area.



Develop a graph to show the effect of glazing area on the overall R-value of the building. Graph the building R-value as a function of glazing area for a glazing area ranging from 0% to 30% of the gross wall area. Assume winter conditions.

Solution: Program RVALUE may be used to accumulate data for the required graph. Door data (the door area is 20 m^2), window data, ceiling data, and wall data are entered as requested, and the window data is repeatedly changed to obtain the overall R-value as a function of glazing area from 0 m^2 to 30% of the gross wall area, which is

$$\text{gross wall area} = 3 \text{ m} \times 120 \text{ m} = 360 \text{ m}^2.$$

The R-value data is:

Glass Area, %	Glass Area, m^2	Building R-value
0	0	2.41 m^2K/W
5	18	1.86
10	36	1.51
15	54	1.28
20	72	1.10
25	90	0.97
30	108	0.87

Note: In using RVALUE, one must be careful not to enter impossible conditions. For example, in this problem, to enter a window area of 100 m^2 and a door area of 300 m^2 (by mistake) results in a net wall area of - 40 m^2, clearly impossible.

The data for building R-value is graphed in Figure 4-4. For comparison, the extreme condition of the walls being entirely glazed (100% glazing) results in an overall R-Value for the building of 0.37 m^2K/W. The graph in Figure 4-4 shows the relatively rapid change of the building's R-value as glazing area is added. When the glazing area changes from 0% to 5% of the wall area (5% is a modest amount of glazing), the overall building R-value drops from 2.41 m^2 K/W to 1.86 m^2 K/W. This is a 23% decrease with a corresponding 30% increase of heat loss.

It would be a useful exercise to repeat this example, using RVALUE, to ensure its use is understood.

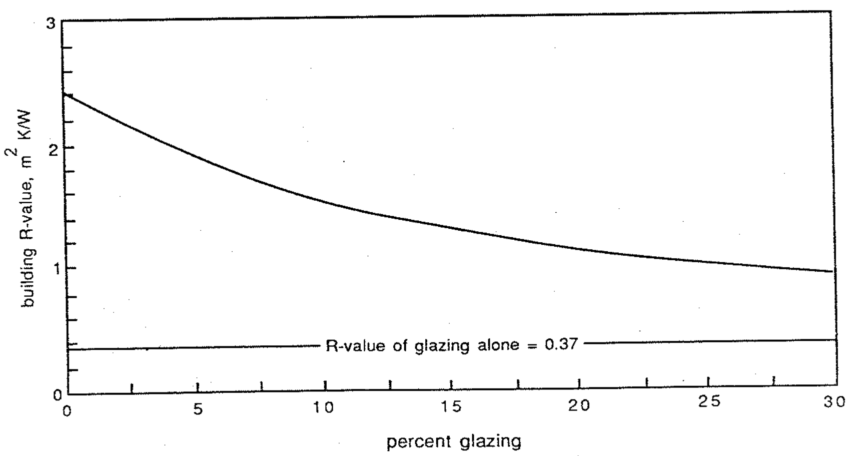


Figure 4-4. Average R-Value of the structural cover of the building described in Example 4-8 as a function of the percentage of the wall devoted to glazing.

4-9. Sol-Air Temperature

All the analyses of heat gains and losses from buildings have, to this point, assumed thermal exchange is between regions at the temperature of the outside air and the temperature of the inside air. This may be a valid assumption at night, or when it is cloudy, but during times of significant solar insolation, solar heating is an additional factor which can significantly affect the magnitudes of heat exchange. A concept which permits consideration of solar effects and long-wave thermal reradiation, without introducing the nonlinearities of radiative heat transfer, is the sol-air temperature. In preview, the sol-air temperature, in the absence of solar heating and long-wave exchange, acts as an equivalent air temperature which would cause heat to be exchanged in the same magnitude that it is exchanged when actual air temperature, thermal radiation, and solar heating are considered.

Consider a building surface irradiated by solar insolation, and exchanging thermal energy with the ambient air by convection, and with the radiant surroundings by long-wave thermal radiation. An energy balance on the surface contains, as fluxes on the upper surface,

$$h(t_a - t_w) + \alpha I + \varepsilon_w \sigma (T_{sky}^4 - T_w^4). \tag{4-4}$$

Simplify Equation 4-4 by treating long-wave exchange as a single term, $\varepsilon\Delta R$, the difference between thermal radiation leaving the surface, and that striking it, to modify the thermal gains terms to

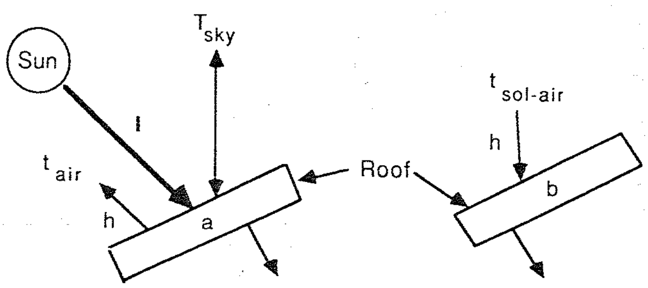
$$h(t_a - t_w) + \alpha I - \varepsilon\Delta R. \tag{4-5}$$

Next hypothesize a situation where there is no solar irradiation or long-wave thermal exchange, only convective exchange between the surface and air at a temperature, t_{sa}, which produces exactly the same net heat exchange with the surface. The gains in the thermal energy balance for this situation are only convective,

$$h(t_{sa} - t_w). \tag{4-6}$$

Heat exchanges in the two cases are the same, thus, the gains on the upper surface must be equivalent. Equating the two gains yields

$$t_{sa} = t_a + (\alpha I - \varepsilon\Delta R) / h. \tag{4-7}$$



Same heat conduction through the roof

If this equivalent temperature, the "sol-air temperature", is known for each building surface, thermal exchange between the building and its surroundings can be calculated using only linear equations. Fortunately, data exists to estimate the sol-air temperature.

Methods to determine solar gain will be covered later, in Section 5-3. For now, assume a value of I can be determined for each surface. Means to determine the reradiation term are less obvious. It has been observed (Parmelee and Aubele, 1952) that ΔR is approximately 0 for vertical building surfaces. Frequently, the sky is colder than a vertical surface, but the ground is warmer because of solar heating and the net effect is to balance the loss and gains. The same researchers found ΔR to be approximately 60 W/m^2 (loss from the surface) for a horizontal surface facing a clear sky. Later researchers (Lokmanhekim, 1971, and Kusuda, 1976) modified ΔR further to account for cloudiness,

$$t_{sa} = t_a + (\alpha I - 6\varepsilon \cos\phi(10 - \Omega))/ h, \qquad (4\text{-}8)$$

where Ω is the cloudiness factor, ranging from 0 for a clear sky, to 10 for complete cloud cover. The angle ϕ is the tilt angle of the surface under consideration, being 0° for a horizontal surface and 90° for a vertical surface.

Example 4-9

Problem: A building roof having a 1:3 slope is irradiated with solar heat at a rate of 500 W/m^2. Air temperature is 28 C and the sky is clear. The roof's solar absorptance is 0.8, and its thermal emittance is 0.95. The surface convective coefficient is approximately 30 W/m^2 K (R_{os} = 0.03 m^2 K/W). Calculate the sol-air temperature which applies and compare it to the actual air temperature.

Solution: Equation 4-8 can be used to calculate the sol-air temperature directly. The roof slope is 1:3, or 18.43°, and

$$\begin{aligned} t_{sa} = {} & 28\text{ C} + [(0.80)\ (500\text{ W/m}^2) \\ & - (6\text{ W/m}^2)\ (0.95)\ (\cos(18.43°))\ (10 - 0)]/30\text{ W/m}^2\text{ K} \\ = {} & 39.5\text{ C} \end{aligned}$$

This is a significant increase above the actual air temperature. In effect, the solar (primarily) effect is to raise the effective outdoor temperature by more than 11 C. The solar effect is more than 13° (400 Wm^{-2} / 30 Wm^{-2} K^{-1}), while reradiation is nearly a 2° reduction, making the net effect 11.5 C. Of course, this example is for a building surface receiving a high amount of solar radiation. However, on any building surface irradiated by the sun, the net flow of heat is likely to be into the building (in the steady state) instead of out of the building, as we have assumed for example in Section 4-2.

加入 · 防破坏 分析 · 补充数据

4-10. Heat Exchangers

Engineers frequently must design equipment to transfer thermal energy from one fluid stream to another. Heat exchangers are used for this task. Standard design techniques are available which can be applied to most design problems and which are well suited to computerized analysis as a tool for design.

Heat exchangers find wide application in the food processing industry to heat and cool food products. In environmental control, heat exchangers are used, for example, to prewarm fresh air by recovering heat from ventilation air being exhausted from a building. Heat exchangers can be used to transfer heat from steam or water (heated in a boiler) to air in air circulators such as fan-jet units.

4-10.1. Overall Heat Transfer. The concept of series thermal circuits has already been discussed in Section 3-2.4. In heat exchangers, heat is transferred from one stream of fluid to another stream, with transfer occurring across a wall separating the two fluid streams. The series thermal circuit contains two convective resistances and one conductive resistance.

The thermal circuit is as shown in Figure 4-5, and unit area thermal flux, q'', can be expressed as

$$q'' = \frac{t_b - t_a}{1/h_a + x/k + 1/h_b}, \quad \approx U \cdot \Delta t \qquad (4\text{-}9)$$

where k is the thermal conductivity of the wall material and x is its thickness.

However, a plane wall with uniform temperatures on its two sides is not a practical concern with heat exchangers. The goal is to transfer heat, thus, the two fluid streams continuously vary in temperature. The total heat transfer can be expressed in terms of an overall heat transfer coefficient, U, a total area for heat exchange, A, and some average temperature difference, Δt_m,

$$q = UA\Delta t_m, \qquad (4\text{-}10)$$

The determination of Δt_m depends on the type of heat exchanger being designed. For example, consider the double pipe exchangers shown in Figure 4-6. With parallel flow heat exchangers, the temperatures of the two fluid streams approach a single, intermediate value and Δt_m varies from the initial difference

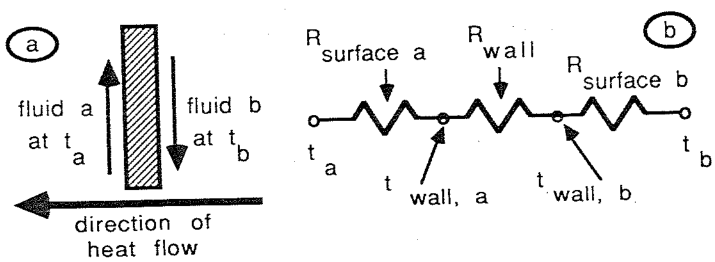


Figure 4-5. Thermal circuit for heat exchange: (a) fluid flow past the heat exchanger wall and (b) the series thermal circuit showing resistances (R) and temperatures (t).

between the entering streams to no difference if the exchanger is large enough to be completely effective. With the counter flow exchanger, the temperature difference between the streams may or may not change along the length of the exchanger but the difference is not known a priori at any point along the exchanger.

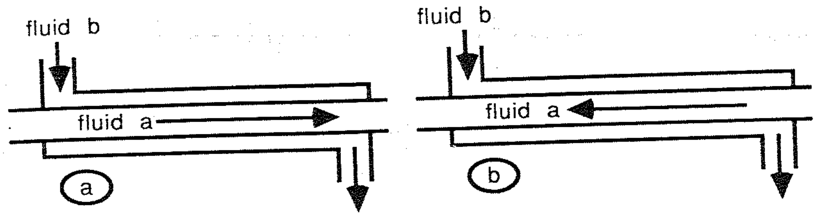


Figure 4-6. Double pipe heat exchangers: (a) parallel flow and (b) counter flow.

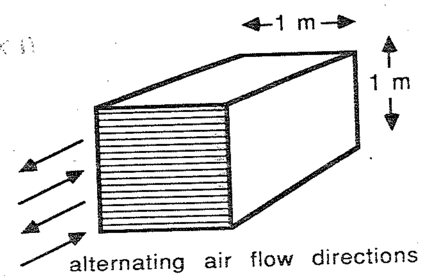
As a practical matter, counter flow heat exchangers are more typically used for environmental control applications. Temperature changes on each side of the exchanger can be greater than would be the case for parallel flow exchangers. Parallel flow heat exchangers are useful when a rapid initial temperature change is desired in one or both of the fluid streams.

4-10.2. Overall Heat Transfer Coefficient. When commercially designed heat exchangers are to be used, the manufacturer's data should be available to determine the overall UA value of the exchanger, perhaps as a function of fluid properties and flow rates, or perhaps as a relatively constant value. Such data is obtained by experiment and is the most accurate as long as the exchanger is used as tested. However, if a heat exchanger is to be designed and constructed for local use, heat transfer analysis procedures covered previously can be used to estimate the UA value of the exchanger.

Example 4-10

sheet steel mild

Problem: A heat exchanger is to be designed and locally fabricated to prewarm ventilation air for a calf nursery. The exchanger will be built of parallel plates of 0.5 mm thick galvanized sheet metal. Dimensions are as shown and the airflow rate in each channel is to be approximately 0.25 m^3/s. Determine the average unit area thermal conductance (U) for the exchanger.



Solution: The unit area thermal conductance can be determined using

$$U = 1/(1/h_1 + x/k + 1/h_2),$$

where x = 0.5 mm and k = 45.3 W/mK (from Appendix 3-1 for mild steel); and the convective coefficients, h_1 and h_2, must be determined. Each is a forced convective heat transfer coefficient, which is discussed in Section 3-3.2.

Equation 3-44 applies, see P.71

$$h = cG^{0.8} D^{-0.2},$$

where G and D are as defined in Equations 3-45 and 3-46. From the sketch of the exchanger and the dimensions given, each airflow channel is approximately 0.125 m by 1 m, and

$$D = 4(0.125 \text{ m}^2)/(2.25 \text{ m}) = 0.222 \text{ m}.$$

$\frac{4 \cdot \text{面積}}{\text{周長}}$ = 水力直徑

To determine G, air density is required. Operating conditions are not known, but we can assume if the unit is used to temper outdoor air, air temperature will be between outdoor air temperature and room air temperature. Temperatures will vary from day to day and within the exchanger but often will be below freezing. As an approximation, assume air density to be approximately 1.10 kg/m^3 (cool air somewhat above sea level).

Density variations within the exchanger will be ignored and it will be assumed the convective coefficient remains constant from place to place within the exchanger. Variations should be small and insignificant compared to other unknowns such as variation of local air velocity. Airflow nonuniformity from top to bottom in each channel, and from channel to channel, will likely contribute a greater degree of uncertainty than air density differences. Such practical matters lead to the need to confirm overall U values by experimental means. However, heat transfer analysis can be used as a first approximation to evaluate designs. The mass rate airflow, G, is thereby

$$G = \underbrace{(1.10 \text{ kg/m}^3)}_{\rho}\underbrace{(0.25 \text{ m}^3/\text{s})}_{V}/\underbrace{(0.125 \text{ m}^2)}_{A} = 2.2 \text{ kg/m}^2 \text{ s}.$$

P.72.

The coefficient c required to determine h is estimated (Table 3-3) to be approximately 3.10 at temperatures at which the exchanger would be expected to operate. The convective heat transfer coefficient is, thus,

$$h = (3.10)(2.2 \text{ kg/m}^2 \text{ s})^{0.8}(0.222 \text{ m})^{-0.2} = 7.87 \text{ W/m}^2 \text{ K}.$$

This value applies to each side of the exchanger, thus, the overall unit area thermal conductance is

$$\begin{aligned} U &= 1/[(1 / 7.87 \text{ W/m}^2 \text{ K}) + (0.0005 \text{ m} / 45.3 \text{ W/mK}) + (1 / 7.87 \text{ W/m}^2 \text{ K})] \\ &= 3.94 \text{ W/m}^2 \text{ K}. \end{aligned}$$

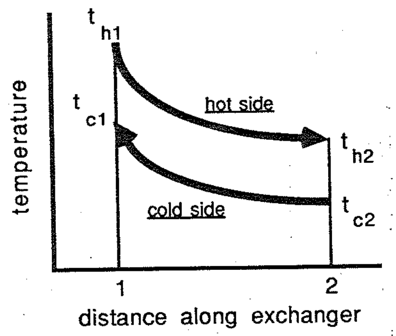
P355

4-10.3. Logarithmic Mean Temperature Difference. The average temperature difference from one end of the heat exchanger to the other is needed to determine the total heat exchanged. Heat transfer texts show how differential calculus can be applied to derive equations for the types of heat exchangers in single pass counter flow and parallel flow exchangers, Figure 4-6. Texts such as Holman (1981) present correction factors to determine the average temperature difference for many other types of heat exchangers.

The average, or mean, temperature difference is called the "log mean temperature difference" (LMTD) defined as the temperature difference at one end of the exchanger, less the temperature difference at the other end of the exchanger, divided by the natural logarithm of the ratio of the two temperature differences. In symbol form,

$$\Delta t_m = \frac{(t_{h2} - t_{c2}) - (t_{h1} - t_{c1})}{\ln[(t_{h2} - t_{c2})/(t_{h1} - t_{c1})]}. \qquad (4\text{-}11)$$

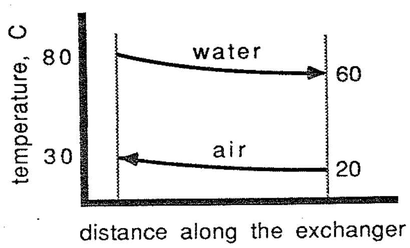
The entering and exiting fluid stream temperatures must be known to determine the LMTD. The concept is, thus, most useful in determining design parameters to achieve certain heat exchange goals such as shown in Example 4-11.



Example 4-11

Problem: A heat exchanger is used to transfer heat from hot water to recirculated air within a greenhouse. Hot water arrives at the exchanger at 80 C, and leaves at 60 C, and flows at a rate of 0.05 kg/s. Air flows through the exchanger at a rate of 0.4165 kg/s, and is heated from 20 C to 30 C. The measured unit area thermal conductance of the heat exchanger is 25 W/m^2 K (this high rate is achieved using fins on the air side of the exchange surface).

What area of heat exchange is needed to achieve these conditions?



Solution: The specific heats of air and water are 1006 and 4190 J/kgK, respectively (approximately). The rate of heat exchange can be found from

$$q = mc_p\Delta t$$

m T_{in} T_{out}

$$= (0.05 \text{ kg/s})(4190 \text{ J/kgK})(20 \text{ K}) \text{ on the water side, or}$$
$$= (0.4165 \text{ kg/s})(1006 \text{ J/kgK})(10\text{K}) \text{ on the air side, thus,}$$
$$= 4{,}190 \text{ J/s.}$$

The LMTD is

$$\text{LMTD} = \frac{(80 - 30) - (60 - 20)}{\ln(80 - 30)/(60 - 20)]} = 44.81 \text{ K.}$$

Note the LMTD is intermediate between the 50 K difference at one end of the heat exchanger and the 40 K difference at the other end, but is not the simple average. Equation 4-10 can be restated as

$$q = UA(\text{LMTD}),$$

thus,

$$A = q/[U(\text{LMTD})]$$
$$= (4{,}190 \text{ J/s})/(25 \text{ W/m}^2\text{K})(44.81 \text{ K})$$
$$= 3.74 \text{ m}^2.$$

4-10.4. Heat Exchanger Effectiveness and the NTU Method. As used in Example 4-11, the LMTD approach to heat exchanger design is useful when all fluid flow temperatures are known and the heat exchange area, or thermal conductance, is to be determined. When inlet or outlet fluid temperatures are to be determined for a given exchanger, the LMTD method becomes cumbersome.

U 已 A 為 未知

Environmental control design frequently involves using a commercially available heat exchanger and designing fluid temperatures and flow rates to achieve certain goals, rather than designing the exchanger itself. The NTU method is more useful for such design problems.

The effectiveness, ε, of a heat exchanger is defined as

$$\varepsilon = \frac{\text{actual rate of heat transfer}}{\text{maximum possible rate of heat transfer}}. \tag{4-12}$$

The actual rate of heat transfer can be determined by calculating either the rate heat is lost from the hot fluid or the rate heat is gained by the cold fluid.

$$q_{actual} = m_h c_h \Delta t_h, \text{ or} \tag{4-13a}$$
$$= m_c c_c \Delta t_c. \tag{4-13b}$$

where m is mass flow rate, c is specific heat, Δt is temperature change; the subscripts h and c refer to the hot and cold sides of the exchanger, respectively.

The maximum possible rate of heat exchange will be achieved when fluid on one side of the exchanger attains the entering temperature of the other fluid stream. The stream in which this is possible is the one with the minimum value of the (mc) product, because an energy balance shows the heat gained by the cold side must equal the heat lost by the hot side. Thus,

$$q_{max} = (mc)_{min}(t_{h,\,inlet} - t_{c,\,inlet}). \tag{4-14}$$

When entering fluid temperatures, mass flow rates and specific heats are known, the maximum possible rate of heat exchange can be determined and if the effectiveness can be predicted, the actual rate of heat transfer can be found.

Equations to calculate heat exchanger effectiveness can be found in heat transfer texts (such as Holman, 1981). For simple parallel and counter flow heat exchangers, effectiveness values can be determined from

$$\text{parallel flow: } \varepsilon = \frac{1 - \exp[-\,NTU(1 + C)]}{1 + C} \tag{4-15}$$

$$\text{counter flow: } \varepsilon = \frac{1 - \exp[-\,NTU(1 - C)]}{1 - C\exp[-\,NTU(1 - C)]} \tag{4-16}$$

$$\text{counter flow, } C = 1\text{: } \varepsilon = \frac{NTU}{1 + NTU} \tag{4-17}$$

$$\text{all exchangers for } C = 0\text{: } \varepsilon = 1 - \exp(-\,NTU) \tag{4-18}$$

eq 4-15 4-16/皆可用.

$$\text{where } NTU = UA/(mc)_{min}, \text{ and} \tag{4-19}$$

$$C = (mc)_{min}/(mc)_{max}. \tag{4-20}$$

The grouping of terms, $UA/(mc)_{min}$, is termed the "Number of Transfer Units", or NTU. This term arises because the grouping can be interpreted to indicate the relative size of the heat exchanger. Equations to determine NTU are:

ln(x)

$$\text{parallel flow: } NTU = \frac{-\ln[1 - (1 + C)\varepsilon]}{1 + C} \tag{4-21}$$

x<0 不可行

$$\text{counter flow: } NTU = (C - 1)^{-1}\ \ln\left(\frac{\varepsilon - 1}{\varepsilon C - 1}\right) \tag{4-22}$$

C=1

$$\text{counter flow, } C = 1\text{: } NTU = \varepsilon/(1 - \varepsilon) \tag{4-23}$$

(須代入此式)

$$\text{all exchangers, } C = 0\text{: } NTU = -\ln(1 - \varepsilon) \tag{4-24}$$

皆可用 4-21 與 4-22

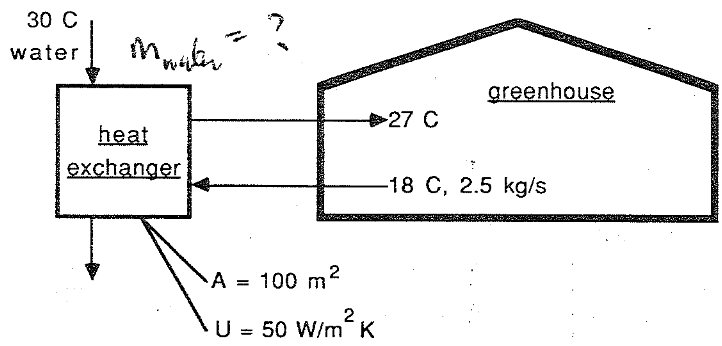
Heat transfer texts provide graphs to determine ε and NTU, but equations are more useful for computerized analyses.

Applications of the NTU method will be seen through two examples. The basic method of solution is based on trial and error.

Example 4-12

Problem: Warm water from the cooling condenser of a power generating station is to provide heat for a large greenhouse range. The warm water flows through counter flow heat exchangers (many exchangers operating in parallel) and greenhouse air is heated by being circulated through the exchangers. Warm water is available at 30 C and greenhouse air is to be heated from 18 C to 27 C. Air flows through the exchanger at a mass flow rate of 2.5 kg/s. The heat exchanger surface area is 100 m^2 and the average unit area thermal conductance (U) is 50 W/m^2 K.

At what rate must water be pumped through the exchanger to provide this amount of air heating?



Solution: The limiting side of the heat exchanger is not known at this point. The trial and error approach of the NTU method assumes one of the sides is limiting, completes the analysis, and then checks to determine whether the assumption was correct. The check is whether the heat gained and lost by the fluid streams is the same as the heat transferred across the heat exchange surface.

Begin by assuming the air side is limiting. (It is easier to start using the air side because both m and c are known for the air side.)

$$(mc)_{min} = (2.5 \text{ kg/s})\,(1006 \text{ J/kg K}) = 2515 \text{ J/s K}.$$

Air enters at a temperature of 18 C and exits at 27 C, thus, the heat gain is

$$q_{air} = (2515 \text{ J/s K})(27 \text{ C} - 18 \text{ C}) = 22{,}635 \text{ W}.$$

The Number of Transfer Units can be calculated,

$$NTU = UA\,/\,(mc)_{min} = (100 \text{ m}^2)\,(50 \text{ W/m}^2 \text{ K})/(2515 \text{ J/s K}) = 1.99,$$

and the effectiveness is

$$\varepsilon = \frac{\text{actual temperature change}}{\text{maximum possible temperature change}}$$
$$= (27\ C - 18\ C)/(30\ C - 18\ C) = 0.75,$$

assuming the specific heat is not a function of temperature and the best that can be done is to heat the air to the entering water temperature (from 18 C to 30 C).

Now the question is: Are the NTU and ε values compatible for a value of 0 < C < 1? If so, the original assumption of air as the limiting side is correct (C < 1). If C < 1 the minimum is less than the maximum, as it should be.

Equation 4-16 expresses effectiveness of a counter flow heat exchanger, and everything is known but C. Unfortunately, the equation is nonlinear and a solution for C is not straightforward. One method to approach a solution for C is by trial and error, assuming values for C, solving the equation, and iterating until the value of effectiveness from the equation approximates the known value of ε = 0.75. This is an ad hoc method of numerical solution which could be readily implemented on a computer. Sample results of iterations for NTU = 1.99 are:

C	ε	
0.5	0.773	(already close to 0.75)
0.4	0.793	(wrong direction)
0.6	0.753	(looking better)
0.65	0.742	(too far)
0.62	0.748	(still too far)
0.61	0.750	(good enough for me)

So what has been learned? The initial assumption was correct. Air is the limiting side and C = 0.61. From the definition of C,

$$(mc)_{max} = (mc)_{min} / C = 2515\ J/s\ K / 0.61 = 4123\ J/s\ K.$$

In the temperature range from 18 C to 27 C the specific heat of water is approximately 4180 J/kg K, thus,

$$m_{water} = (4123\ J/s\ K)/(4180\ J/kg\ K) = 0.986\ kg/s.$$

The density of water in the temperature range is approximately 0.998 kg/L, thus, the pumping rate should be approximately 1 L/s.

The exit temperature of water can also be determined. The rate of heat exchange is 22,635 W, water flows at 0.986 kg/s with a specific heat of approximately 4180 J/kg K, thus, its temperature change must be

$$\Delta t_{water} = (22{,}635\ J/s)/(0.986\ kg/s)(4180\ J/kg\ K)$$
$$= 5.5\ K.$$

Water exits the exchanger at 30 C - 5.5 K = 24.5 C.

Example 4-13

Problem: Reconsider Example 4-12 and determine the required water pumping rate when water enters the heat exchanger at a temperature of 35 C instead of 30 C.

Solution: The same approach as before can be used; air can be assumed to be the limiting side. The NTU is unchanged and equals 1.99. The effectiveness is now

$$\varepsilon = (27\text{ C} - 18\text{ C})/(35\text{ C} - 18\text{ C}) = 0.53.$$

The same iterative approach may be used as before,

C	ε	(as before)
0.5	0.773	
0.7	0.731	
0.9	0.688	
0.99	0.668	(not even close)

To reach ε = 0.53 would require C > 1, which is clearly impossible. What is the conclusion? Air must not be the limiting side. It may not be intuitively obvious that warmer water suddenly causes the water side to be limiting, but the equations are sufficiently complicated that intuition should not be trusted.

So what can be done? A solution is possible using a trial and error solution for mc_{water}. A beginning point is known, mc_{water} must be less than 2515 J/sK (the mc of the air side) for the air side is not limiting.

Values of mc_{water} can be assumed, and the effectiveness determined both by Equation 4-12 and Equation 4-16. When the two agree, a solution has been attained.

mc_{water}	C	NTU = UA/C_{min}	Δt_{water}	$\varepsilon = \Delta t_{water}/17$	ε by Eq. 4-16
			= 22635/(mc)water		
2000	0.795	2.500	11.32 K	0.666	0.766
1500	0.596	3.333	15.09	0.888	0.876
1600	0.636	3.125	14.15	0.832	0.853
1550	0.616	3.226	14.60	0.859	0.865
1530	0.608	3.268	14.79	0.870	0.869
1533	0.610	3.262	14.77	0.869	0.868
1534	0.610	3.259	14.76	0.868	0.868

Values in the table are determined from the following:

$C = mc_{water}/2515$ J/s K
NTU $= (50 \text{ W/m}^2\text{ K})(100 \text{ m}^2)/mc_{water}$
$\Delta t_{water} = (22{,}635 \text{ J/s}) / mc_{water}$
$\varepsilon = \Delta t_{water} / 17$ is based on a maximum possible water temperature change of 17 K (from 35 C to 18 C).

The effectiveness value calculated in two ways agrees when mc_{water} is 1534 J/s K, which is less than mc_{air}, thus, water is indeed the limiting side.

With mc_{water} = 1534 J/s K and the specific heat of water = 4180 J/kg K, the mass rate flow of water must be

$$m_{water} = (1534 \text{ J/s K})/(4180 \text{ J/kg K}) = 0.367 \text{ kg/s},$$

and with a water density of 0.998 kg/L the flow rate of water should be (0.367 kg/s)/(0.998 kg/L) = 0.368 L/s. The exiting water will be cooled by 14.76 K (already calculated in the table). The exiting water temperature will thus be 35 C - 14.76 K = 20.24 C, which is more than 4 K colder than the exiting water temperature in Example 4-12 where entering water temperature was 5 K colder.

A careful comparison of Examples 4-12 and 4-13 can help in gaining an understanding of the action of heat exchangers. In these two examples, when the water enters at a higher temperature, heat is transferred more quickly at the beginning and less heat must be transferred at the end, thus, a colder water exit temperature can be tolerated. The greater temperature change in the water permits a significantly lower water flow rate, which is important for design. One could conclude that when reject heat sources are used for heating, slight source temperature increases can lead to significant changes of operation.

4-10.5. Program XCHANGER. As is obvious from Examples 4-12 and 4-13, designing heat exchangers can be a tedious exercise if done using hand methods. Program XCHANGER is provided as a tool useful for exploring heat exchanger designs. The program uses the NTU method and can calculate any of four temperatures: the warm side entering or exiting temperature, or the cold side entering or exiting temperature.

Examples 4-12 and 4-13 demonstrated how the NTU method can be used to find required mass flow rates and exiting temperatures of the flow streams. The capability of calculating required entering temperatures is included in XCHANGER through iterative searching for an entering temperature which satisfies an energy balance (in the manner demonstrated in Examples 4-12 and 4-13). The program also determines when a solution is not possible – which occurs more frequently than one might intuitively expect!

可使用 DOSBOX 來 run dos 版程式

Example 4-14

Problem: Reconsider Example 4-13. The heat exchanger manufacturer markets units in many sizes, ranging from a heat exchange surface area of 20 m^2 to an area of 200 m^2 in increments of 20 m^2. The choice of 100 m^2 in Examples 4-12 and 4-13 was somewhat arbitrary. In this example, explore the effects of using exchangers of different areas for the design conditions of Example 4-13.

Solution: Program XCHANGER was used to determine all data provided below. The program was run repeatedly, changing the UA value for each iteration.

Convergence is very sensitive to the UA value. At UA = 1895 W/K there is not convergence, while at UA = 1896 W/K there is a solution for a water flow rate of 1308 kg/s, which is clearly not suitable.

Counter flow

Area, m^2	Limiting	UA, W/K	m_{water}	t_{exit}^{water}	NTU	ε
20	solution does not converge					
40	cold side	2000	3.4837	33.446	0.80	0.5294
60	warm side	3000	0.5474	25.107	1.31	0.5819
80	warm side	4000	0.4110	21.826	2.33	0.7749
100	warm side	5000	0.3669	20.243	3.26	0.8681
120	warm side	6000	0.3468	19.386	4.14	0.9185
140	warm side	7000	0.3361	18.887	4.98	0.9479
160	warm side	8000	0.3298	18.581	5.80	0.9658
180	warm side	9000	0.3260	18.387	6.61	0.9772
200	warm side	10000	0.3235	18.262	7.39	0.9846

Data such as calculated in this example could be used to select the most appropriate size of exchanger. For example, an area of 60 m^2 results in need for much less water than would be required by an exchanger with an area of 40 m^2, yet increasing the area to 80 m^2 does not result in a comparable further savings. If water supply and pumping costs are a major consideration, a larger area might be cost-effective. If not, the exchanger with 60 m^2 of heat exchange area could be best. However, even if water flow is a large concern, an exchanger area larger than 100 m^2 will not provide significant savings in the need for water.

Note that all the exchangers except the first are acceptable in that they solve the design problem of providing heat to the greenhouse. However, some are more cost-effective than others and the financial aspects of engineering design are always important.

Parallel flow

60	cold	3000	0.979	29.482	1.19	0.5294
100	cold	5000	0.7144	27.48	1.99	0.5294
140	cold	7000	0.6831	[illegible]	2.78	0.5294

SYMBOLS

- A area, m^2
- C ratio of mc products, see Equation 4-15
- c coefficient, see Equation 3-44
- c_c, c_h specific heat, kJ/kg K
- D hydraulic diameter, m
- F perimeter heat loss factor, W/m K
- G unit area mass flow rate, kg/m^2 s
- h convective heat transfer coefficient, W/m^2 K
- I solar irradiation, W/m^2
- k thermal conductivity, W/m K
- m mass flow rate, kg/s
- P perimeter, m
- q heat transferred, W
- R unit area thermal resistance, m^2 K/W
- ΔR net long-wave thermal radiation exchange, W/m^2
- t temperature, C
- T absolute temperature, K
- U unit area or unit length thermal conductance, W/m^2 K or W/m K
- x spacial variable, m
- α solar absorptance
- ε heat exchanger effectiveness
- ε surface emittance for radiation heat exchange
- ϕ surface tilt angle
- Ω degree of cloudiness, from 0 (none) to 10 (total)

EXERCISES

1. You are designing a building to be 14 m wide, 60 m long, and 3 m high. It will have a well ventilated attic space.

   The ceiling is hung from the roof trusses and will be sheathed on the lower side with 19.05 mm plywood and insulated with 88.9 mm of mineral fiber insulation above the sheathing. The walls are framed using 38.1 mm x 88.9 mm lumber, sheathed on the inside with 15.88 mm thick high density particleboard, insulated with full thickness cellulosic insulation, sheathed on the outside with 25.4 mm cellular polyurethane (surfaced) and sided with 9.53 mm lapped plywood. There are two panel doors in the building, each 2.5 m x 3.5 m, and each panel door is 44 mm thick with 11 mm thick panels. Single glazed windows (no storm windows) comprise 4% of the gross wall area.

   The owners decided a window area of 4% of gross wall area will not permit enough light to enter, yet they do not want simply to increase window area at the expense of energy conservation. Calculate the window area which would provide the same overall R-value of the shell of the building if double glass ("thermopane") with 3 mm thick glass and a 6 mm thick airspace were used in place of single glazed windows.

2. Determine the effective unit area R-value of a framed wall made of 2 x 6s (38.1 mm x 139.7 mm) which are 24 in. (610 mm) on center. The wall has 88.9 mm of batt insulation (mineral fiber) on the inside (warm) wall side of the wall cavity. The inside wall sheathing is 12.7 mm plywood and the outside of the wall is sided with 15.88 mm plywood (there is no outside sheathing under the siding). Include effects of framing on the R-value. Assume framing is 18% of the gross wall area.

3. Consider the barn wall in Example 4-5 constructed as outlined in Exercise 2 above. The windows will be single glazed with wood frames. Determine the average unit area R-value of the wall. Assume window area is 8% of the gross wall area.

4. Continue with the wall described in Exercise 3 above. Add a panel door (44 mm thick with 11 mm panels) which is 2.8 m high and 4.2 m wide and determine the unit area R-value of the wall. Assume doors are equivalent to 3% of the gross wall area.

5. A wall has been built which has an average unit area R-value of 1.8 m^2 K/W. The wall is built on an unheated slab floor insulated along the edge by an insulation with R = 0.95 m^2 K/W. The wall is 2.8 m high. Of all the heat lost through the wall and perimeter, what percent is through the wall and what percent is through the perimeter?

6. For the wall described in Exercise 5 above, by how much would heat loss through the wall and perimeter increase if the perimeter were not insulated?

7. The heat exchanger described in Example 4-10 is to be built with a total heat exchange area of 50 m^2. If outdoor air is - 5 C and indoor air is 18 C, what will be the rate (watts) of heat exchange between the two airstreams? What will be the temperatures of the two airstreams as they exit the exchanger? Assume there is no condensation within the exchanger.

8. A forced air heating unit (counter flow heat exchanger) is to be used for a greenhouse. The unit, which has a UA value of 800 W/K, receives water from a boiler at 85 C. During night the heater must warm the greenhouse air from 16 C to 32 C (at a mass airflow rate of 2.3 kg/s). What water flow rate is required to do this?

During the day, greenhouse temperature rises from 16 C to 23 C and air enters the exchanger at 23 C. What will be the exit temperature of the greenhouse air? What will be the heat delivery rate (watts) during the night and w

REFERENCES

ASHRAE. 1976. Procedures for determining heating and cooling loads for computerizing energy calculations. American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, GA. 182 pp.

ASHRAE. 1989. Handbook of Fundamentals. American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, GA.

ASHRAE. 1985. Environmental Control Principles, a textbook supplement to the ASHRAE Handbook 1985 Fundamentals Volume. American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, GA.

Esmay, M.L. and J.E. Dixon. 1986. Environmental Control for Agricultural Buildings. The AVI Publishing Co., Westport, CT. 287 pp.

Holman, J.P. 1981. Heat Transfer. McGraw-Hill Book Co., New York. 570 pp.

Kusuda, T. 1976. The computer program for heating and cooling loads in buildings. National Bureau of Standards Load Determination Program. U.S. Department of Commerce/National Bureau of Standards. BSS No. 69. Washington D.C.

Lokmanhekim, M., editor. 1971. Algorithms for building heat transfer subroutines. American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, GA.

Mackay, C.O. and L.T. Wright. 1946. The sol-air thermometer – A new instrument. Transactions ASHVE 52:271.

Midwest Plan Service. 1978. Professional design supplement. Midwest Plan Service, Ames, IA. 218 pp.

Midwest Plan Service. 1983. Structures and Environment Handbook. Midwest Plan Service, Ames, IA.

Parmelee, G.V. and W.W. Aubele. 1952. Radiant energy emission of atmosphere and ground. Transactions ASHVE 58:85-106.

Stoecker, W.F. and J.W. Jones. 1982. Refrigeration and Air Conditioning. McGraw-Hill Book Company, New York. 443 pp.

Threlkeld, J.L. 1970. Thermal Environmental Engineering. Prentice-Hall, Englewood Cliffs, NJ. 493 pp.

Timmons, M.B. and L.D. Albright. 1978. Wind directional dependence of sol-air temperatures. Transactions of the ASAE 21 (4) :742-746.