CHAPTER

TWO

UTC SYSTEM THEORY{ TC "UTC SYSTEM THEORY" \l 1 }

The UTC system model is solved several thousand times in a typical annual simulation. Several assumptions are necessary to reduce the theory to a set of equations that can be solved quickly. This chapter details the UTC system equations and the assumptions necessary to obtain them.

2.1 Energy Balances { TC "2.1 Energy Balances " \l 2 }

The first step in predicting the thermal performance of the UTC system is to calculate the outlet air temperature from the collector, T_{out}. There are four fundamental energy balance equations that are solved to find T_{out}.

The canopy is assumed to act as a perfect constant velocity header. This means that the air in the plenum only flows vertically, not horizontally, and the UTC system becomes two-dimensional, as shown in Figure 2.1.1. Ambient air is drawn through the holes in the UTC plate and into the plenum, the space between the collector plate and the wall. The air flowing through the collector into the plenum convects energy from the collector surface.

$$m_{out}\, c_p\, (T_{plen} - T_{amb}) = Q_{conv,col\text{-}air} \qquad (2.1.1)$$

The labelling convention that is used for heat flows is $Q_{mode,from\text{-}to}$. So $Q_{conv,col\text{-}air}$ is convection from the collector to the air. The air flow rate m_{out} is the outdoor air flow rate through the collector. The air temperature in the plenum is assumed to be uniform, which eliminates the need for a complicated CFD analysis.

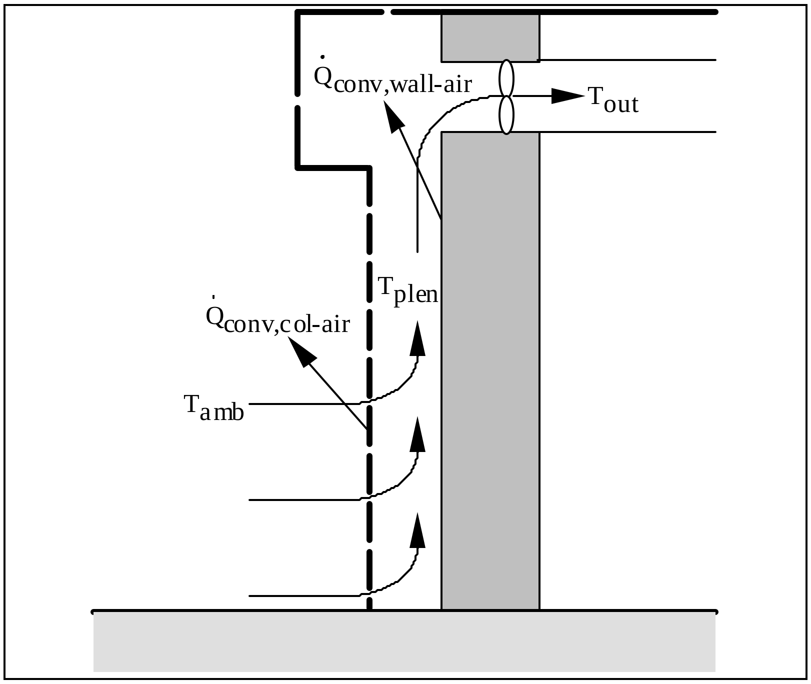

Figure 2.1.1. Energy balance on air.{ TC "Figure 2.1.1. Energy balance on air." \l 5 }

Also in Figure 2.1.1, the air convects energy from the outside wall surface as it travels up the plenum.

$$m_{out}\, c_p\, (T_{out} - T_{plen}) = Q_{conv,wall\text{-}air} \tag{2.1.2}$$

Figure 2.1.2 shows an energy balance on the outside wall surface.

$$Q_{cond,wall} = Q_{conv,wall\text{-}air} + Q_{rad,wall\text{-}col} \tag{2.1.3}$$

The outside surface of the wall gains energy by conduction through the wall from the inside, and it loses energy by convection to the air in the plenum and radiation to the back of the collector.

Although the specific heat is temperature dependent, constant air properties are assumed. This eliminates the need to continually recalculate the air properties at every stage in the collector system. Air properties at ambient temperature are used because the only point in the system where air properties are crucial to the calculations is as the air travels through the collector.

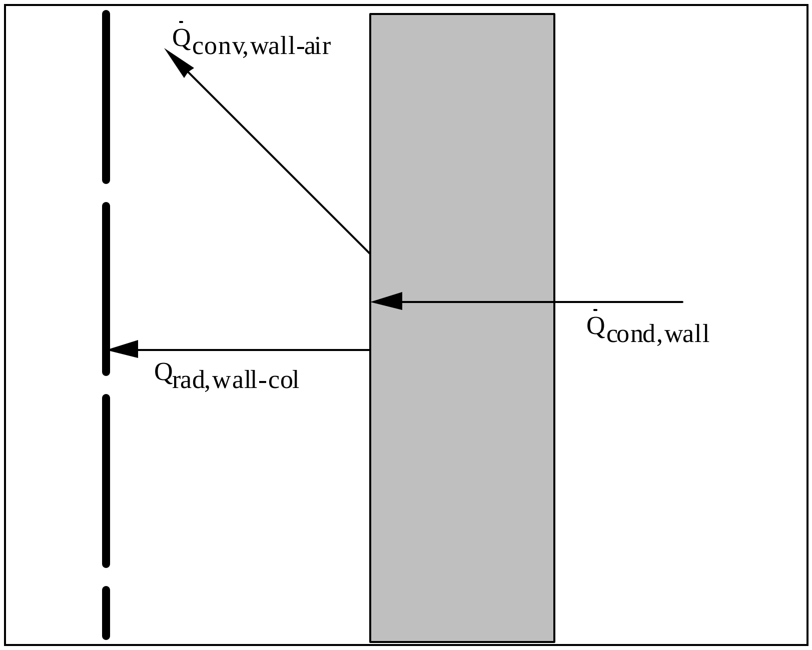

Figure 2.1.2. Energy balance on wall.{ TC "Figure 2.1.2. Energy balance on wall." \l 5 }

The energy balance on the collector plate is shown in Figure 2.1.3. The energy gained by the plate is the absorbed solar radiation and the infrared radiation from the wall behind the collector. The energy removed from the collector is the convection to the air flowing through the collector holes and the net radiation from the collector to the surroundings (sky and ground).

$$Q_{abs} + Q_{rad,wall-col} = Q_{conv,col-air} + Q_{rad,col-sur} \qquad (2.1.4)$$

The convection from the collector to the air includes convection from the front, hole, and back surface.

In Equation 2.1.4, it is assumed that there are no convection losses from the collector to the surroundings. This assumption means that there is no natural convection loss from the collector to the surroundings and no forced convection loss due to wind. These approximations have been validated analytically if the air flow rate per unit of collector area, or approach velocity, is greater than 0.02 m/s and the collector area is large enough that edge loss is negligible [Kutscher, 1992].

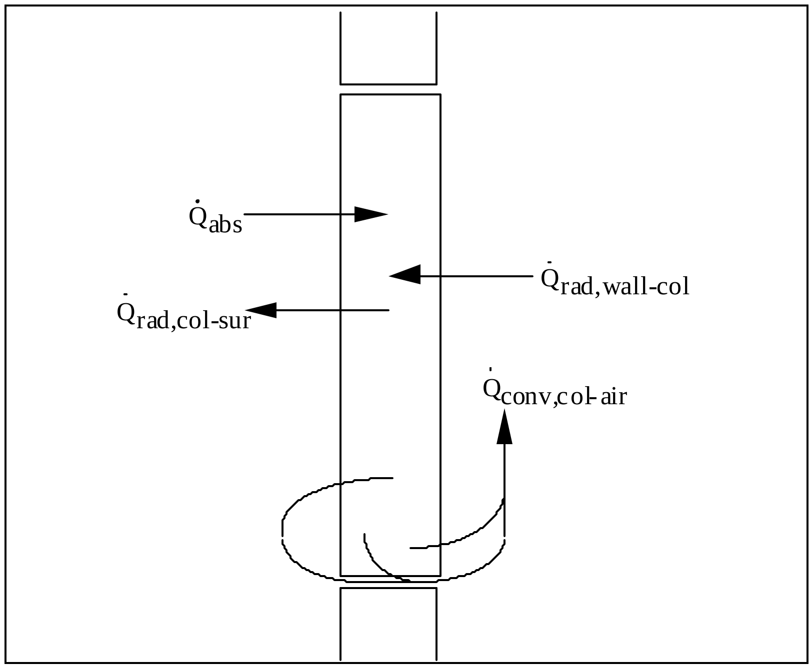

Figure 2.1.3. Energy balance on collector plate.**{** TC "Figure 2.1.3. Energy balance on collector plate." \l 5 **}**

2.2 Rate Equations { TC "2.2 Rate Equations" \l 2 }

The rate equations for the energy flows are necessary to solve the energy balance equations in Section 2.1.

For convection from the collector to the air, an empirical heat transfer correlation is used [Kutscher, 1992].

$$Nu_D = 2.75 \ (P / D)^{-1.2} \ Re_D^{0.43} \qquad (2.2.1)$$

P is the distance between hole centers, called hole pitch, and D is the hole diameter. This correlation determines the Nusselt number based on hole diameter that is used to find $h_{conv,col-air}$. Convection occurs on the front surface, the hole surface, and the back surface of the collector plate. Although convection on the front surface comprises the majority of the heat transfer, all three are included in Kutscher's correlation. The heat exchanger effectiveness of the collector is calculated by Equation 2.2.2.

$$\varepsilon_{HX} = 1 - \exp((h_{conv,col-air} A_s) / (m_{out} c_p)) \quad (2.2.2)$$

This effectiveness is used in the relation between the plenum air temperature and the collector temperature.

$$\varepsilon_{HX} = (T_{plen} - T_{amb}) / (T_{col} - T_{amb}) \quad (2.2.3)$$

Equation 2.2.3 is effectively a rate equation for $Q_{conv,col-air}$.

The collector plate temperature is assumed to be uniform. Thermographs of operating UTC plates show that the plate temperature is fairly uniform [Enermodal, 1994].
This has been found to be a good assumption given relatively uniform flow through the collector [Kutscher, 1992]. To ensure uniform flow through the collector, the pressure drop across the collector must be at least 25 Pa.

The heat transfer correlations used for convection from the wall to the air are flat-plate correlations for parallel flow [Incropera and DeWitt, 1990]. Equation 2.2.4 is a mixed flow correlation, used for $Re_{ht} > 5 \times 10^5$.

$$Nu_{ht} = (0.037\, Re_{ht}^{0.8} - 871)\, Pr^{1/3} \quad (2.2.4)$$

For $Re_{ht} < 5 \times 10^5$, the following laminar flow correlation is used.

$$Nu_{ht} = 0.664\; Re_{ht}^{0.5}\; Pr^{1/3} \quad (2.2.5)$$

The average plenum velocity is used in these correlations. Since the velocity varies from zero at the bottom of the plenum to its maximum at the top, the average is just half of the maximum. These correlations are used to find $h_{conv,wall-air}$, used in the following rate equation.

$$Q_{conv,wall-air} = h_{conv,wall-air} A (T_{wall} - T_{plen}) \quad (2.2.6)$$

The temperature of the outside wall surface is also assumed to be uniform.

The flat-plate correlations are admittedly suspect since the flow in the plenum has not been studied extensively. However the experimental work has not been done to provide a better correlation. Previous work by Rhee and Edwards [1981] yields a correlation for a channel with asymmetric suction and heating. However, the suction is out of the channel, not into the channel. Kuroda and Nishioka [1989] report on a channel with injection through a perforated wall, but the parameters of the perforated wall (e.g. porosity) are not applicable to UTC plates. Since experimental work is beyond the scope of this thesis, these flat-plate correlations must be used. This approximation causes the UTC model to under predict the convection from the wall to the air, as shown in Section 4.1.

The following rate equations are used with the energy balances on the outside wall surface and collector plate.

$$Q_{cond,wall} = U_{cond,wall} A (T_{room} - T_{wall}) \quad (2.2.7)$$

$$Q_{rad,wall-col} = \sigma_{sb} A (T_{wall}^4 - T_{col}^4) / (1/\varepsilon_{wall} + 1/\varepsilon_{col} - 1) \quad (2.2.8)$$

$$Q_{abs} = \alpha_{col} I_T A_s \quad (2.2.9)$$

$$Q_{rad,col-sur} = \varepsilon_{col} \sigma_{sb} A_s (T_{col}^4 - T_{sur}^4) \quad (2.2.10)$$

The temperature of the surroundings is a combination of the radiative ground and sky temperatures.

$$T_{sur}^4 = 0.5 (T_{gnd}^4 + T_{sky}^4) \quad (2.2.11)$$

Equation 2.2.11 is only valid for a vertical UTC plate. The ground temperature is assumed to equal the ambient temperature. To estimate the blackbody sky temperature, the algorithm developed by Martin and Berdahl [1984] is used (see Appendix B).

2.3 UTC System Operation{ TC "2.3 UTC System Operation" \l 2 }

Ambient air is heated by the collector to the outlet temperature, T_{out}. The useful energy gained is the sum of convection from the collector and from the outside wall surface.

$$Q_u = Q_{conv,col-air} + Q_{conv,wall-air} \quad (2.3.1)$$

The outlet air from the collector is mixed with recirculated air from the building. This air, at T_{mix}, is heated to the necessary supply temperature to meet the heating load, as shown in Figure 2.3.1.

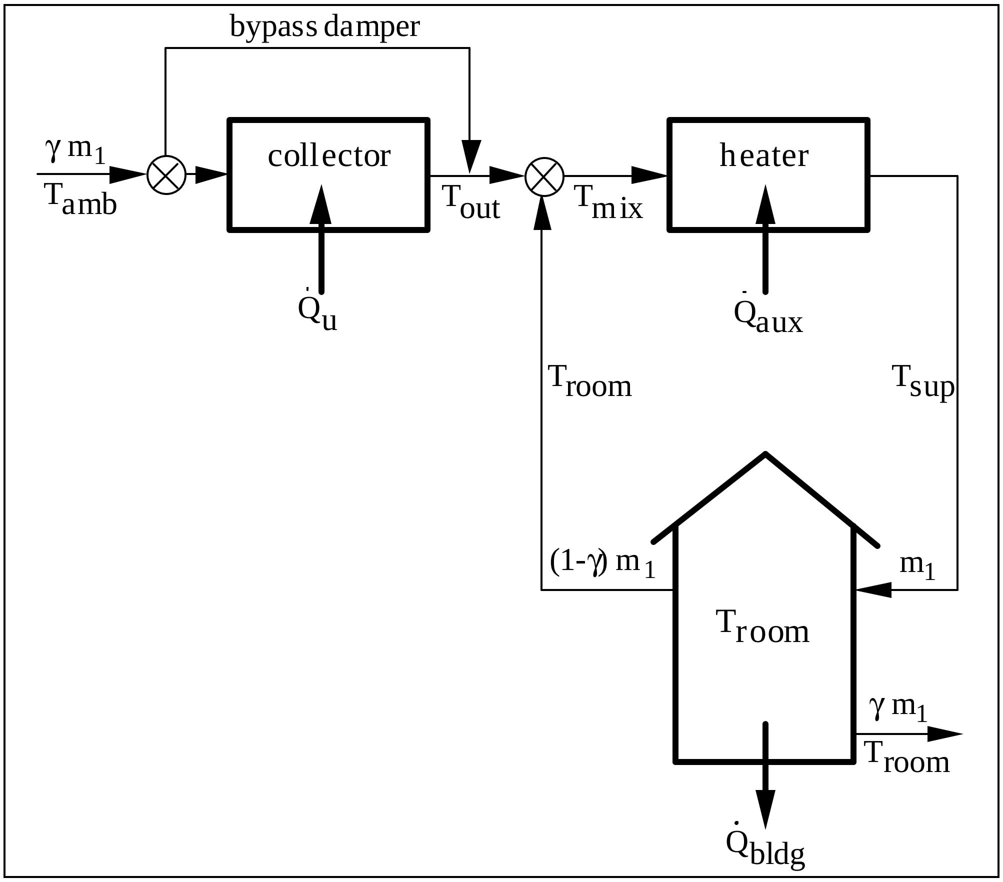

Figure 2.3.1. Basic overview of the UTC system model.**{** TC "Figure 2.3.1. Basic overview of the UTC system model." \l 5 **}**

The summer bypass damper is opened when the ambient temperature exceeds a pre-set level. The bypass damper is built into the collector plate directly in front of the building air intake. When the bypass damper is open, ambient air is drawn directly into the building without being heated by the collector. The plenum is sealed off from the building air intake so that natural convection currents do not bring heated air into the building.

The supply air flow rate to the building, m_1, is fixed by the fan size. Constant air volume fans are used in UTC systems [Hollick, 1995]. The recirculation damper varies γ, the fraction of the supply air that is drawn from the outside through the collector.

$$m_{out} = \gamma \, m_1 \tag{2.3.2}$$

The outdoor air flow rate must be within a certain range. The minimum required outdoor air flow rate is set by ventilation standards. Obviously, the maximum outdoor air flow rate is equal to the supply air flow rate ($\gamma = 1$). The recirculation damper varies γ such that the auxiliary energy is minimized.

An energy balance on the entire system yields the auxiliary energy required to meet the heating load.

$$Q_{aux} = m_{out} c_p (T_{room} - T_{amb}) + Q_{bldg} - Q_u$$
$$= Q_{load} - Q_u \qquad (2.3.3)$$

Under a given set of operating conditions, Q_{load} increases as m_{out} increases, as can be seen in Figures 2.3.2-4a. The y-intercept of the Q_{load} curve is Q_{bldg}, which consists of the building skin loss and the internal gain due to people, lights, etc.

$$Q_{bldg} = UA (T_{room} - T_{amb}) - \text{Gains} \qquad (2.3.4)$$

As expected from Equation 2.3.3, the slope of the Q_{load} curve decreases as the ambient temperature increases from 7 C in Figure 2.3.2a to 11 C in Figure 2.3.4a.

There is a limit to the amount of useful energy that can be gained from the collector because there is a limited amount of solar energy absorbed by the collector as well as a limited amount of wall loss available to be recaptured. Q_u asymptotically approaches that limit as m_{out} increases. The UTC system is controlled to minimize Q_{aux}, the difference between Q_{load} and Q_u, as shown in Figures 2.3.2-4b. Therefore, the UTC system should be operated at the outdoor air flow rate at which Q_{load} and Q_u intersect.

As shown in Figures 2.3.2-4c, the supply temperature necessary to meet the heating load is not a function of m_{out}.

$$T_{sup} = T_{room} + Q_{bldg} / (m_1 c_p) \qquad (2.3.5)$$

T_{mix} is found by a simple energy balance on the recirculation damper.

$$T_{mix} = \gamma T_{out} + (1 - \gamma) T_{room} \qquad (2.3.6)$$

T_{mix} and T_{sup} intersect at the same value of m_{out} at which Q_{aux} is zero.

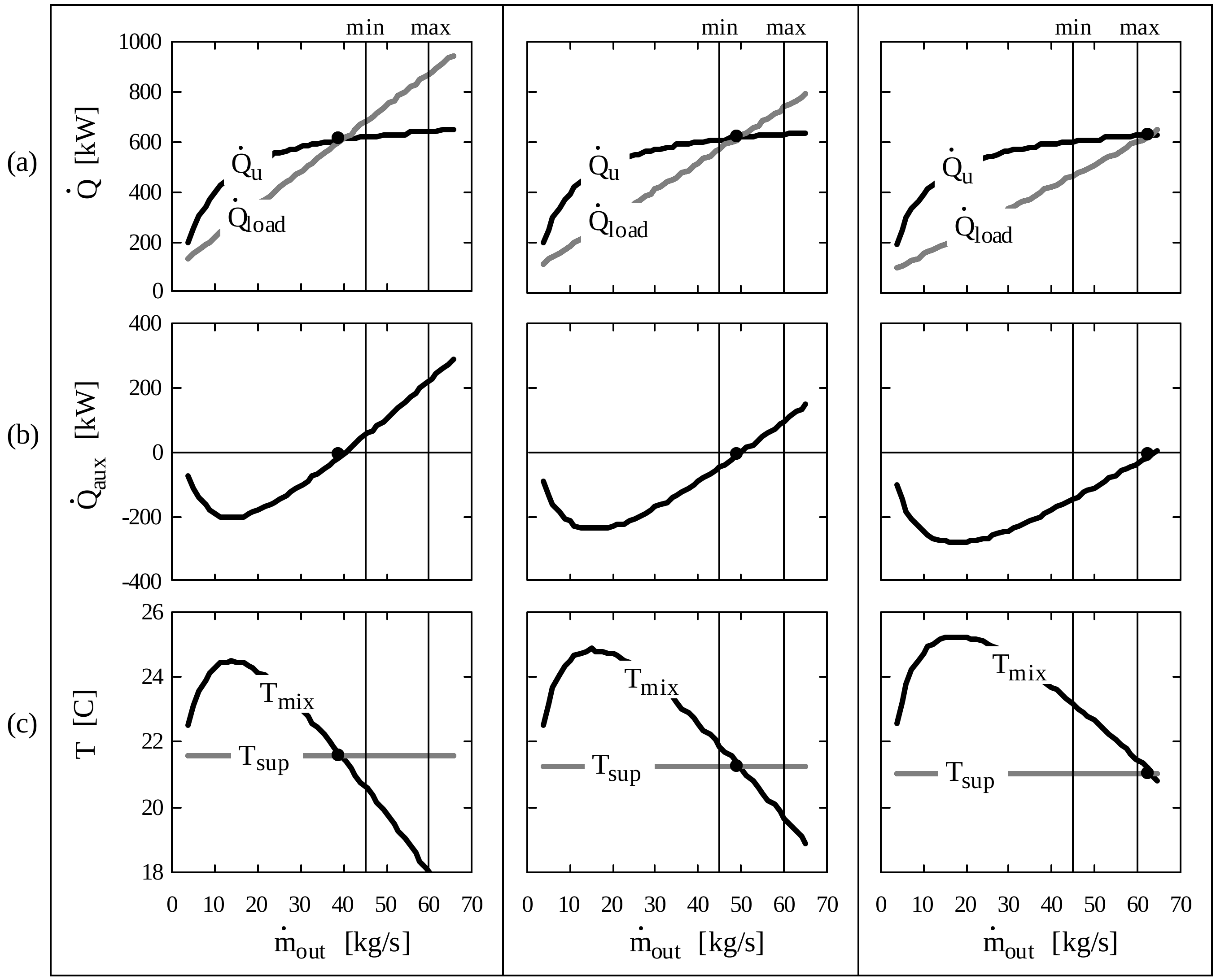

Figure 2.3.2. UTC system operation at T_{amb} = 7 C.{ TC "Figure 2.3.2. UTC system operation at T_{amb} = 7 C." \l 5 }

Figure 2.3.3. UTC system operation at T_{amb} = 9 C.{ TC "Figure 2.3.3. UTC system operation at T_{amb} = 9 C." \l 5 }

Figure 2.3.4. UTC system operation at T_{amb} = 11 C.{ TC "Figure 2.3.4. UTC system operation at T_{amb} = 11 C." \l 5 }

In Figure 2.3.3, Q_{aux} is zero within the allowable range for m_{out}. Therefore, the UTC system should be operated at this outdoor air flow rate. If less outdoor air is used, T_{mix} is too hot. Also, bringing in less outdoor air does not take advantage of the free heat available to bring in outdoor air which improves the indoor air quality. If more outdoor air is used, Q_u increases and indoor air quality improves even more, but auxiliary energy is required.

The auxiliary energy may not equal zero within the allowable range for m_{out}. At low ambient temperatures, the condition for zero auxiliary may be below the minimum required outdoor air flow rate, as shown in Figure 2.3.2. To minimize Q_{aux}, the UTC system brings in the minimum amount of outside air necessary to meet ventilation standards.

At high ambient temperatures, the intersection of Q_{load} and Q_u may be at an outdoor air flow rate above the maximum possible, as shown in Figure 2.3.4. At the maximum flow rate, T_{mix} is greater than the desired T_{sup}, and the summer bypass damper should be opened. However, in actual UTC system installations, the summer bypass is only opened when the ambient temperature rises above the summer bypass set temperature. So, in this model, the UTC system operates in the same way, and the bypass damper is not opened. The UTC system operates at the maximum outdoor air flow rate, and the building becomes slightly over heated. Optimally, the summer bypass set temperature is adjusted so that over heating does not occur.

2.4 Reduced Wall Loss{ TC "2.4 Reduced Wall Loss" \l 2 }

The reduced wall loss is calculated by taking the difference of the potential conduction and the actual conduction through the south wall. The potential conduction through the wall is calculated for a temperature difference of (T_{room} - T_{solair}) across the wall. With the UTC plate over the south wall, the actual temperature difference becomes (T_{room} - T_{plen}), which may reduce the wall loss.

The conduction through the wall covered by the UTC plate is given by Equation 2.4.1.

$$Q_{cond,wall} = U_{cond,wall} A (T_{room} - T_{plen}) \tag{2.4.1}$$

The conduction through the wall if the UTC plate were not there is calculated using the sol-air temperature.

$$Q_{pot} = U_{cond,wall} A (T_{room} - T_{solair}) \tag{2.4.2}$$

The sol-air temperature is given by Equation 2.4.3 [ASHRAE, 1993].

$$T_{solair} = T_{amb} + \alpha_{wall} I_T / h_{film} \tag{2.4.3}$$

The average film coefficient for the air against the original wall, h_{film}, is assumed to be 15 W/m^2-C [Enermodal, 1994]. So the reduced conduction through the wall due the UTC plate is given by Equation 2.4.4.

$$\begin{aligned} Q_{red,wall} &= Q_{pot} - Q_{cond,wall} \\ &= U_{cond,wall} A (T_{plen} - T_{solair}) \end{aligned} \tag{2.4.4}$$

The reduced wall loss is a small component of the total energy saved by the UTC system. A simple numerical example shows the magnitude of the reduced wall loss. Values for the parameters needed in the calculation are given in Table 2.4.1.

Table 2.4.1. Sample UTC system parameters for reduced wall loss calculation.**{** TC "Table 2.4.1. Sample UTC system parameters for reduced wall loss calculation." \l 7 **}**

Parameter	Value
$U_{cond,wall}$	0.568 W/m^2-C
collector area, A	100 m^2
ambient air temperature rise, T_{amb}	0 C
plenum air temperature, T_{plen}	10 C
wall absorptivity, α_{wall}	0.4
solar radiation, I_T	700 W/m^2
air density, ρ	1.2 kg/m^3
approach velocity, V	0.035 m/s
air specific heat, c_p	1006 J/kg-C

The sol-air temperature is calculated from Equation 2.4.3.

$$T_{solair} = 0 - (0.4)\,(700)\,/\,(15) = 18.7 \text{ C} \tag{2.4.5}$$

The reduced wall loss is calculated from Equation 2.4.4.

$$Q_{red,wall} = (0.568)\,(100)\,[10 - 18.7] = -492 \text{ W} \tag{2.4.6}$$

In this example, the wall loss actually increases. The wall loss increases when the sol-air temperature is higher than the plenum air temperature. In effect, the UTC plate increases the conduction through the wall by shading the wall.

The majority of the useful energy gained by the air in the UTC system is convection from the collector to the air, given by Equation 2.1.1. The mass flow rate through the collector is necessary for this energy balance.

$$m_{out} = \rho\, V\, A = (1.2)\,(0.035)\,(100) = 4.2 \text{ kg/s} \tag{2.4.7}$$

Evaluating Equation 2.1.1 yields the convection from the collector to the air.

$$Q_{conv,col\text{-}air} = (4.2)\,(1006)\,(10) = 42{,}250 \text{ W} \tag{2.4.8}$$

So the magnitude of the reduced wall loss is on the order of 1% of the useful energy gained by the air in the UTC system. The reduced wall loss is more significant during the night and on cloudy days.

2.5 Energy Savings{ TC "2.5 Energy Savings" \l 2 }

There are three energy savings mechanisms for a UTC system: active solar gain, recaptured wall loss, and reduced wall loss. However, the energy savings of the UTC system is not simply the sum of these three components. Fundamentally, the energy savings is the reduction in the heat required from a traditional system, which translates into a reduction of the heating bill. The heat required from an auxiliary unit of a UTC system is less than the heat required from a traditional heating system.

$$Q_{save} = Q_{trad} - Q_{aux} \tag{2.5.1}$$

For the UTC system, the auxiliary energy is the difference between the load on the building and the useful energy gained by the UTC system.

$$Q_{aux} = Q_{load} - Q_u$$
$$= Q_{load} - (Q_{conv,col\text{-}air} + Q_{conv,wall\text{-}air}) \qquad (2.5.2)$$

The load on the building depends on the amount of outdoor air drawn into the building.

$$Q_{load} = \gamma\, m_1\, c_p\, (T_{room} - T_{amb}) + Q_{bldg}$$
$$= \gamma\, m_1\, c_p\, (T_{room} - T_{amb}) + Q_{skin} - Gains - Q_{red,wall} \qquad (2.5.3)$$

The heat supplied by a traditional system is the load on the building (without the reduced wall loss) with the minimum outdoor air flow because a traditional system never brings in more outdoor air than the minimum required.

$$Q_{trad} = \gamma_{min}\, m_1\, c_p\, (T_{room} - T_{amb}) + Q_{skin} - Gains \qquad (2.5.4)$$

Substituting Equations 2.5.2-4 into Equation 2.5.1 yields the energy savings as a function of the amount of outdoor air drawn into the building.

$$Q_{save} = Q_{conv,col\text{-}air} + Q_{conv,wall\text{-}air} + Q_{red,wall}$$
$$- (\gamma - \gamma_{min})\, m_1\, c_p\, (T_{room} - T_{amb}) \qquad (2.5.5)$$

When the UTC system is operating at the minimum outdoor air flow rate, the energy savings is indeed equal to the sum of the three energy savings components from the UTC system. The recirculation damper on the UTC system is controlled in such a way that the outdoor air flow is above the minimum only when no auxiliary energy is required (see Section 2.3). Under these conditions, the useful energy gained equals the building load.

$$Q_{load} = Q_u \qquad (2.5.6)$$

Equation 2.5.5 can be simplified.

$$Q_{save} = Q_{load} + Q_{red,wall} - (\gamma - \gamma_{min})\, m_1\, c_p\, (T_{room} - T_{amb})$$
$$= Q_{skin} - Gains + \gamma_{min}\, m_1\, c_p\, (T_{room} - T_{amb})$$
$$= Q_{trad} \qquad (2.5.7)$$

So the energy savings never exceeds the heating requirements of the building with a traditional system. Bringing more outside air into a building increases the amount of energy gained from the UTC system and improves the indoor air quality, but it does not save more energy. The actual energy savings can be substantially less than the sum of the three UTC savings components if the UTC system operates above the minimum outdoor air flow rate.

The energy savings is dependent on the building balance temperature, the ambient temperature at which there is no heating load on the building. The balance temperature is calculated from Equation 2.5.8 [ASHRAE, 1993].

$$T_{bal} = T_{room} - Gains / (UA + (\gamma_{min}\ m_1 + \beta\ m_2)\ c_p) \qquad (2.5.8)$$

At the balance temperature, the heating load with a traditional heating unit, Q_{trad}, is equal to zero. Since the energy savings rate cannot exceed Q_{trad}, there is no energy savings when the ambient temperature is higher than the building balance temperature, even if the summer bypass damper is closed. Therefore, UTC systems on buildings with low balance temperatures save less energy than those on buildings with high balance temperatures.

2.6 Minimum Approach Velocity and Maximum Collector Area{ TC "2.6 Minimum Approach Velocity and Maximum Collector Area" \l 2 }

Obviously, the collector area cannot be larger than the available south wall area. However, there are other constraints on the maximum area of the collector. In Section 2.1, an important assumption made in the energy balance on the collector plate is that there is no convection loss to the surroundings. This assumption is only valid if the approach velocity is above V_{min} = 0.02 m/s [Kutscher, 1992]. Therefore, for a building with a given minimum outdoor air requirement, there is a maximum allowable collector area to operate at maximum efficiency.

$$A_{max} = (\gamma_{min}\ m_1) / (\rho\ V_{min}) \qquad (2.6.1)$$

If the collector area is larger than this maximum, there may be significant convection losses from the collector when the approach velocity is below 0.02 m/s.

The following equation is used to calculate the pressure drop across the collector [Kutscher, 1992].

$$\Delta P_{col} = \zeta\ \rho\ V^2 / 2 \tag{2.6.2}$$

The non-dimensional pressure drop, ζ, is calculated from an empirical correlation [Kutscher, 1992].

$$\zeta = 6.82\ (\ (1-\sigma) / \sigma\ )^2\ (Re_D)^{-0.236} \tag{2.6.3}$$

The porosity for a UTC plate with an equilateral triangular hole pattern is given by Equation 2.6.4 [Kutscher, 1992].

$$\sigma = 0.907\ (D / P)^2 \tag{2.6.4}$$

The pressure drop across the collector plate must be at least 25 Pa to ensure a uniform flow distribution over the collector. A uniform flow distribution results in a uniform temperature distribution over the collector, which optimizes the performance of the collector. If the temperature distribution is not uniform, the hot spots on the collector increase the radiation loss to the surroundings, thereby reducing the efficiency of the UTC system. Depending on the porosity of the collector plate, the pressure drop across the plate may be less than 25 Pa for an approach velocity of 0.02 m/s. In this case, the minimum approach velocity has to be greater than 0.02 m/s. Increasing the minimum allowable approach velocity decreases the maximum collector area.

2.7 Complete System Air Flow{ TC "2.7 Complete System Air Flow" \l 2 }

Depending on the outdoor and supply air flow rates of a building, a secondary air supply unit may be necessary. For example, a building may have a small maximum collector area because of limited south wall area. If this building has a high outdoor air requirement that must be met, the approach velocity could be large if all of the outdoor air is brought through the collector. However, above an approach velocity of 0.05 m/s the useful energy gained from the UTC system does not increase very much. And for high approach velocities, the fan power required by the UTC system may become a concern (fan power is discussed further in Section 2.9). Therefore, it may be desirable to bring only a fraction of the outdoor air through the collector and bring the rest through a secondary air supply unit.

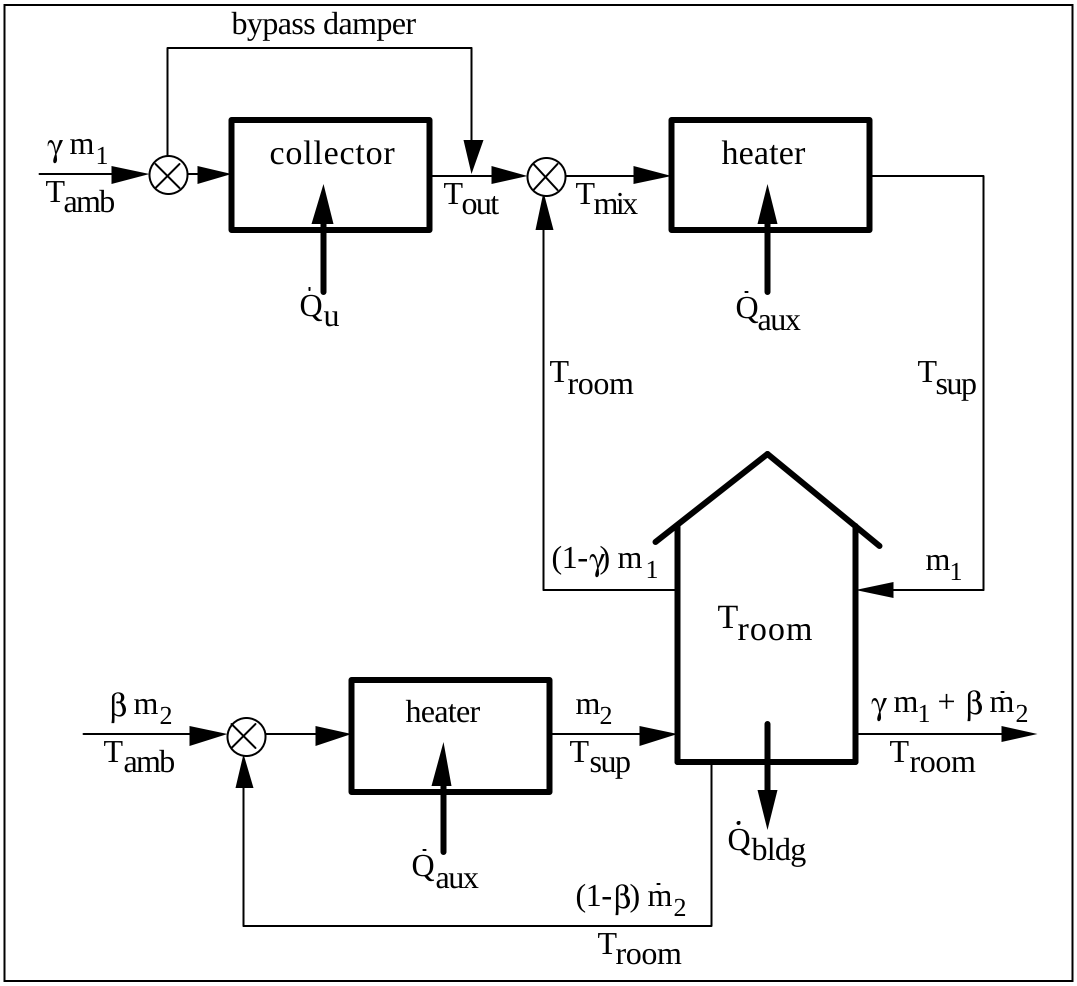

Figure 2.7.1. Complete overview of the UTC system model.{ TC "Figure 2.7.1. Complete overview of the UTC system model." \l 5 }

The model incorporates a traditional air supply unit that is not connected to the UTC system, as shown in Figure 2.7.1. The minimum outdoor and supply air flow rates in the UTC system are chosen, and any additional outdoor and supply air comes from the secondary unit. The addition of the secondary unit changes Equation 2.3.5; the supply air temperature is calculated by Equation 2.7.1.

$$T_{sup} = T_{room} + Q_{bldg} / ((m_1 + m_2) c_p) \qquad (2.7.1)$$

Once the supply air temperature is calculated, the addition of the traditional unit does not affect the operation of the UTC system and is included only to complete the model.

2.8 Approach Velocity as a Function of Collector Area{ TC "2.8 Approach Velocity as a Function of Collector Area" \l 2 }

The approach velocity and air flow rate through the collector are related by the collector area.

$$\text{Air Flow Rate} = \gamma\ m_1 / \rho = V * A \qquad (2.8.1)$$

Figure 2.8.1 shows the approach velocity as a function of collector area for a sample UTC system. Above a certain collector area, the air flow rate through the collector at γ_{min} provides all of the required outdoor air. In Figure 2.8.1, the air flow rate through the collector meets the outdoor air requirement above 900 m^2. The maximum collector area is determined as discussed in Section 2.6. In Figure 2.8.1, the maximum collector area is 2250 m^2 because the minimum approach velocity should not be below 0.02 m/s.

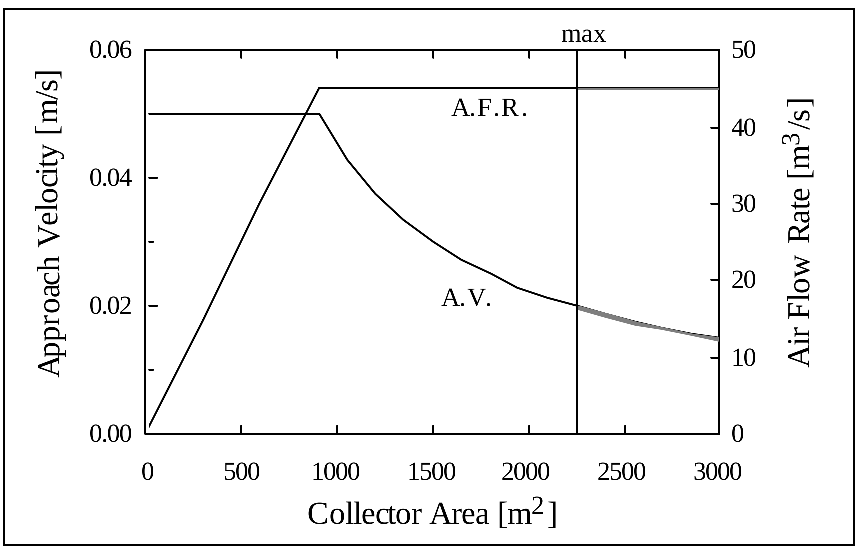

Figure 2.8.1. Approach velocity and air flow rate at γ_{min}.{ TC "Figure 2.8.1. Approach velocity and air flow rate at γ_{min}." \l 5 }

As seen in Figure 2.8.1, the approach velocity at γ_{min} for small collectors is 0.05 m/s. This value is chosen because the efficiency of the UTC does not increase substantially for an approach velocity above 0.05 m/s [Kutscher, 1992], and the fan power requirements may become a concern. If a smaller approach velocity is used, the efficiency of the UTC system is less [Kutscher, 1992]. For small collectors, the minimum outdoor air requirement of the building is not met by the air flow rate through the collector, so the secondary air supply unit provides the rest of the outdoor air, as described in Section 2.7.

2.9 Pressure Drop and Fan Power{ TC "2.9 Pressure Drop and Fan Power" \l 2 }

The total pressure drop of the UTC system is due to the pressure drop through the plenum as well as the pressure drop across the collector plate. The total pressure drop is used to calculate the fan power required by the UTC system. This calculated fan power is not the total fan power needed to supply the air to the building; it is just the additional fan power needed because the UTC system is added to the building.

Fan power calculations are not included in the energy savings calculation because the energy savings may reduce gas or oil consumption, and fan power requires electricity. So fan power is kept separate from energy savings to do the economic analysis. However, fan power calculations are not included in the economic analysis in Chapters 5 and 6 because the required fan power is insignificant relative to the energy savings, as shown later in this section. Also, the following calculations are not exact; they are simply order-of-magnitude calculations to find the approximate fan power requirements of a UTC system. To calculate the additional fan power more accurately would require experimental work that is beyond the scope of this thesis.

The pressure drop across the collector can be calculated from Equations 2.6.2-4. There are three components of the plenum pressure drop: friction, buoyancy, and acceleration. The frictional pressure drop is calculated from Equation 2.9.4 [ASHRAE, 1993].

$$\Delta P_{fric} = f\ (ht / D_h)\ \rho\ (V_{plen,avg})^2 / 2 \qquad (2.9.4)$$

Since the plenum velocity is zero at the bottom of the plenum, the average plenum velocity is just half of the maximum plenum velocity at the top. The hydraulic diameter is given by Equation 2.9.5 [ASHRAE, 1993].

$$D_h = 4\, A_c / P_c \tag{2.9.5}$$

The buoyancy pressure term is due to air in the plenum being warmer and lighter than ambient air [Dymond and Kutscher, 1995].

$$\Delta P_{buoy} = \Delta\rho\; g\, ht \tag{2.9.6}$$

The buoyancy force pushes air up the plenum, acting in the opposite direction of the frictional force. The acceleration pressure drop is based on the Bernoulli effect [Dymond and Kutscher, 1995]. Since the plenum velocity at the bottom of the plenum is zero, this pressure drop can be simplified to Equation 2.9.7.

$$\Delta P_{acc} = \rho\, (V_{plen,\, max})^2 / 2 \tag{2.9.7}$$

The total pressure drop of a UTC system a combination of these components.

$$\Delta P = \Delta P_{col} + \Delta P_{fric} - \Delta P_{buoy} + \Delta P_{acc} \tag{2.9.8}$$

A sample UTC system might have the parameters in Table 2.9.1. The maximum plenum velocity is calculated.

$$V_{plen,max} = (0.035 \text{ m/s})\, (12.8 \text{ m}) / (0.10 \text{ m}) = 4.48 \text{ m/s} \tag{2.9.9}$$

The average plenum velocity is half of the maximum plenum velocity.

$$V_{plen,avg} = (4.48 \text{ m/s}) / 2 = 2.24 \text{ m/s} \tag{2.9.10}$$

The cross-sectional area of the plenum is $A_c = 2.85 \text{ m}^2$, and the perimeter is $P_c = 57.2$ m. Therefore, the hydraulic diameter is found.

$$D_h = 4\, (2.85 \text{ m}^2) / (57.2 \text{ m}) = 0.20 \text{ m} \tag{2.9.11}$$

These values are used to calculate the frictional pressure drop.

$$\Delta P_{fric} = (0.05)\, (12.8 \text{ m} / 0.20 \text{ m})\, (1.0 \text{ kg/m}^3)\, (2.24 \text{ m/s})^2 / 2$$

$$= 8.0 \text{ Pa} \tag{2.9.12}$$

Table 2.9.1. Sample UTC system parameters for pressure drop calculation.{ TC "Table 2.9.1. Sample UTC system parameters for pressure drop calculation." \l 7 }

Parameter	Value
collector area, A	365.0 m^2
collector height, ht	12.8 m
collector length	28.5 m
plenum depth	0.10 m
approach velocity, V	0.035 m/s
air density, ρ	1.0 kg/m^3
air density difference, Δρ	0.05 kg/m^3
plenum friction factor, f	0.05
collector pressure drop, ΔP	25 Pa

An air density difference of 0.05 kg/m^3 between the plenum air and ambient air corresponds to a temperature difference of about 10 C (see Appendix D). The buoyancy pressure drop is found.

$$\Delta P_{buoy} = (0.05\ kg/m^3)\ (9.8\ m/s^2)\ (12.8\ m) = 6.3\ Pa \qquad (2.9.13)$$

The acceleration pressure drop is also calculated.

$$\Delta P_{acc} = (1.0\ kg/m^3)\ (4.48\ m/s)^2\ /\ 2 = 10.0\ Pa \qquad (2.9.14)$$

Therefore the total pressure drop of the UTC is approximated.

$$\Delta P = 25.0\ Pa + 8.0\ Pa - 6.3\ Pa + 10.0\ Pa = 36.7\ Pa \qquad (2.9.15)$$

Now, the fan mass flow rate is calculated.

$$m_{out} = \rho\ V\ A$$

$$= (1.0\ kg/m^3)\ (0.035\ m/s)\ (365.0\ m^2) = 12.8\ kg/s \qquad (2.9.16)$$

The additional fan power required by the UTC system is found with the pressure drop and mass flow rate.

$$\text{Fan Power} = m_{out}\, \Delta P / \rho$$

$$= (12.8 \text{ kg/s})\ (36.7 \text{ Pa}) / (1.0 \text{ kg/m}^3) = 470 \text{ W} \qquad (2.9.17)$$

For this same system, a TRNSYS simulation yields average energy savings of over 24 kW during UTC system operation. The additional fan power required by the UTC system is less than 2% of the energy saved. The additional cost of fan power is negligible compared to energy cost savings, even considering that fans are powered by electricity while air is often heated with cheaper natural gas.