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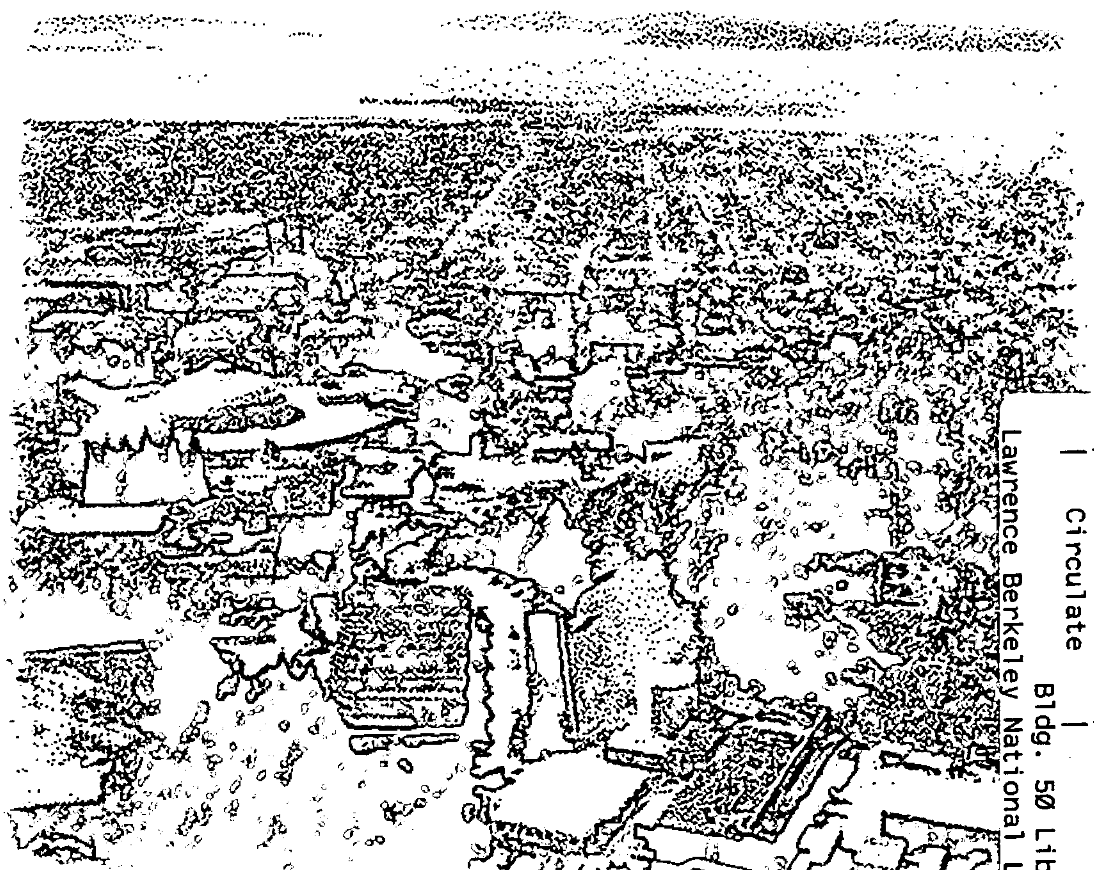

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Simulation Model
Air-to-Air Plate Heat Exchanger

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Simulation Model

Air-to-Air Plate Heat Exchanger

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Note:

This model will be a part of the HVAC component and system library for the SPARK simulation program.

The library is currently under development.

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Air-to-Air Plate Heat Exchanger

Abstract

A simple simulation model of an air-to-air plate heat exchanger is presented. The model belongs to a collection of simulation models that allows the efficient computer simulation of heating, ventilation, and air-conditioning (HVAC) systems. The main emphasis of the models is to shorten computation time and to use only input data that are known in the design process of an HVAC system. The target of the models is to describe the behavior of HVAC components in the part-load operation mode, which is becoming increasingly important in energy efficient HVAC systems. The models are intended to be used for yearly energy calculations or load calculations with time steps of about 10 minutes or larger. Short- time dynamic effects, which are of interest for different aspects of control theory, are neglected.

The part-load behavior is expressed in terms of the nominal condition and the dimensionless variation of the heat transfer with change of mass flow and temperature. The effectiveness-NTU relations are used to parametrize the convective heat transfer at nominal conditions and to compute the part-load condition. If the heat transfer coefficients on the two exchanger sides are not equal (i.e. due to partial bypassing of air), their ratio can be easily calculated and set as a parameter. The model is static and uses explicit equations only.

The explicit model formulation ensures short computation time and numerical stability, which allows using the model with sophisticated engineering methods like automatic system optimization. This paper fully outlines the algorithm description and its simplifications. It is not tailored for any particular simulation program to ensure easy implementation in any simulation program.

Introduction

Many computer simulation models are based on input data that are hard to obtain by the HVAC system designer. The models are usually developed for research work rather than for system design and most of the models are rather complex. Only a few have been broken down into the most important laws that describe their physical behavior accurately enough for system design.

The available models for air-to-air plate heat exchangers usually do not take the dependence of the convective heat transfer coefficient on the air mass flow and temperature into account ([Brandemuehl et.al. 93], [TRNSYS 96]). However, this dependence should not be neglected if the air flow over the heat exchanger varies, which is the case in HVAC systems with variable volume flow (VAV systems) or in systems where the heat exchanger is installed in series with a mixing box.

The model that has been developed describes the steady-state part load behavior using a dimensionless variation of the heat transfers based on nominal conditions. First, the dependence of the convective heat transfer coefficient on the mass flow variation and temperature variation is taken into account. Second, the nominal heat transfer coefficients are calculated based on the nominal boundary conditions (inlet mass flows and temperatures and supply air outlet temperature).

An iteration is only required during the model initialization if the model is used as a cross flow heat exchanger with both streams unmixed. For all other flow configurations, no iteration is required. The numerical solution has to be done only once during the whole simulation. Convergence of the numerical solution is guaranteed.

General Description

We describe a model for the static behavior of a plate air-to-air heat exchanger without condensation. The main purpose of this model is to calculate the yearly energy consumption of an HVAC system. The algorithm is based on only those data that are known in the design of an HVAC system. No geometrical data for the heat exchanger are required. The model allows different mass flows on each side of the heat exchanger and a variation of the mass flow over time. The ratio between the heat transfer, $(h \cdot A)$, on both sides of the exchanger can be set as a parameter. The ratio can easily be calculated if both cross sections of the exchanger are equal as a function of the mass flow and temperature only. If the cross sections are not equal on both sides, e.g., due to different air flows, then good heat exchanger design attempts to achieve a ratio of the $(h \cdot A)$ values equal to one. This is usually done by increasing the contact surface on one side.

Condensation is not taken into account since - due to the small number of hours per year that condensation occurs - the latent energy gain has a negligible impact on the yearly energy consumption in most HVAC systems [Juettemann 84]. However, for special cases (e.g., indoor swimming pools, some industrial processes and tropical climates), this simplification leads to an under-estimate of the amount of recoverable energy. Care has also been taken if condensation occurs when the supply entry temperature is below the freezing point. In this case, special precautions have to be taken (e.g., preheating or partially bypassing the supply air).

Simplifications

- static model
- no condensation
- fouling neglected
- thermal resistance of the heat exchanger material neglected
- no leakage air flow
- no heat loss to the environment

Abbreviation

Variables

ε	exchanger heat transfer effectiveness
ϑ	Celsius temperature
ρ	density
A	area
C	capacity rate
c	specific heat capacity
h	convective heat transfer coefficient
m	mass flow
NTU	number of exchanger heat transfer units
p	absolute pressure
R	ideal gas constant
r	ratio of heat transfer
T	absolute temperature
U	heat transfer coefficient
V	velocity
Q	heat transfer rate
Z	capacity rate ratio

Subscripts

0	nominal (design) point
1	air stream 1
2	air stream 2
avg	average
in	inlet
max	maximum
min	minimum
out	outlet
p	constant pressure
r	room temperature

Mathematical Description

A cross flow heat exchanger is shown schematically in Fig. 1. The numerical indices in the following equations refer to Fig. 1.

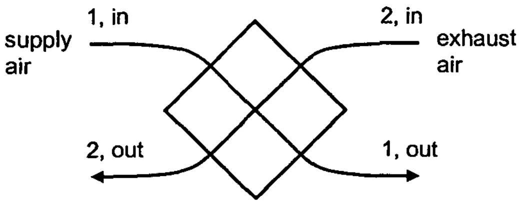

Fig. 1: Cross flow exchanger

Exchanger Heat Transfer Effectiveness

The dimensionless exchanger heat transfer effectiveness, ε, is defined as the actual heat transfer divided by the maximum possible heat transfer [ASHRAE 85]:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}}$$

Eq. 1

Assuming that there is no leakage flow, no heat loss and no phase change, the enthalpy differences across the supply and exhaust air streams are equal. Hence, the heat transfer can be expressed as

$$\dot{Q} = \dot{C}_1 \left(\vartheta_{1,out} - \vartheta_{1,in}\right) = \dot{C}_2 \left(\vartheta_{2,in} - \vartheta_{2,out}\right)$$

Eq. 2

where C is defined as the capacity flow:

$$\dot{C} = \dot{m}\, c_p$$

Eq. 3

The maximum heat exchange is given by the product of the lower capacity flow and the inlet temperature difference, i.e.,

$$\dot{Q}_{\max} = \dot{C}_{\min} \left|\vartheta_{2,in} - \vartheta_{1,in}\right|$$

Eq. 4

where C_{min} is the lower capacity rate.

Substituting Eq. 2 and Eq. 4 into Eq. 1, the exchanger heat transfer effectiveness can be computed as

$$\varepsilon = \frac{\dot{C}_1 \left(\vartheta_{1,in} - \vartheta_{1,out}\right)}{\dot{C}_{\min} \left(\vartheta_{1,in} - \vartheta_{2,in}\right)}$$

Eq. 5

and, analogously,

$$\varepsilon = \frac{\dot{C}_2 \left(\vartheta_{2,in} - \vartheta_{2,out}\right)}{\dot{C}_{\min} \left(\vartheta_{2,in} - \vartheta_{1,in}\right)}$$

Eq. 6

where C_{min} is the lower capacity rate:

$$\dot{C}_{\min} = \min\left(\dot{C}_1, \dot{C}_2\right)$$

Eq. 7

Number of Exchanger Heat Transfer Units

The effectiveness can also be calculated as a function of the *NTU* value, the capacity rate ratio and the heat exchanger configuration (counter-, cross- or parallel flow, mixed or unmixed)

$$\varepsilon = f(NTU, Z, \mathit{flow\ arrangement})$$

Eq. 8

with the dimensionless capacity rate ratio

$$Z = \frac{\dot{C}_{min}}{\dot{C}_{max}}$$

Eq. 9

and the dimensionless number of transfer units

$$NTU = \frac{U_{avg}\ A}{\dot{C}_{min}}$$

Eq. 10

Table 1 shows the different equations according to heat exchanger configuration and capacity rate ratio.

counter flow	$\varepsilon(Z \neq 1) = \frac{1 - e^{-NTU\,(1-Z)}}{1 - Z\ e^{-NTU\,(1-Z)}}$ $\lim_{Z\to 1}\left[\frac{1 - e^{-NTU\,(1-Z)}}{1 - Z\ e^{-NTU\,(1-Z)}}\right] = \frac{1}{1 + NTU^{-1}}$ Eq. 11 Possible range: $0 \le \varepsilon \le 1$	$NTU(Z \neq 1) = \frac{1}{Z-1}\ln\left(\frac{1-\varepsilon}{1-\varepsilon\ Z}\right)$ $\lim_{Z\to 1}\left[\frac{1}{Z-1}\ln\left(\frac{1-\varepsilon}{1-\varepsilon\ Z}\right)\right] = \frac{\varepsilon}{1-\varepsilon}$ Eq. 12
parallel flow	$\varepsilon = \frac{1 - e^{-NTU\,(1+Z)}}{1+Z}$ Eq. 13 Possible range: $0 \le \varepsilon \le \frac{1}{1+Z}$	$NTU = -\frac{\ln(-\varepsilon - \varepsilon\ Z + 1)}{Z+1}$ Eq. 14
cross flow, both streams unmixed	$\varepsilon = 1 - \exp\left(\frac{e^{-NTU\ Z\ \eta} - 1}{Z\ \eta}\right)$ *with* $\eta = NTU^{-0.22}$ Eq. 15 Possible range: $0 \le \varepsilon \le 1$	$NTU = f(\varepsilon, NTU, Z)$ Eq. 16 must be solved numerically. However, the solution is unique (see Fig. 2).
for all configurations	$\lim_{Z\to 0}\varepsilon = 1 - e^{-NTU}$ Eq. 17	

Tab. 1: Equations for the exchanger heat transfer effectiveness ε and its inverse for NTU for different heat exchanger configurations (Eq. 11, see [Kays, London 84], Eq. 13 see [Holman 76], Eq. 15 see [Incropera, DeWitt 90])

As shown later, we calculate our model at the nominal point, NTU_0, as a function of ε_0, which is calculated from Eq. 5. If the user enters wrong input values for Eq. 5, the logarithms in the equations for evaluating NTU_0 as a function of ε_0 could become undefined. However, to detect these wrong inputs, ε_0 can easily be checked against the bounds listed in Table 1 before proceeding with the NTU_0 calculations.

Eq. 15 is exact only for Z = 1, but can be used for $0 < Z \leq 1$ as an excellent approximation [Incropera, DeWitt 90].

To guarantee convergence of Eq. 16, it can be rewritten in the form

$$1-\exp\left(\frac{e^{-NTU^{0.78}Z}-1}{Z\ NTU^{-0.22}}\right)-\varepsilon=0$$

Eq. 18

and solved for NTU using an algorithm such as Regula Falsi or Bisection. The efficiency of the algorithm is not critical since Eq. 18 has to be solved only once during the whole simulation.

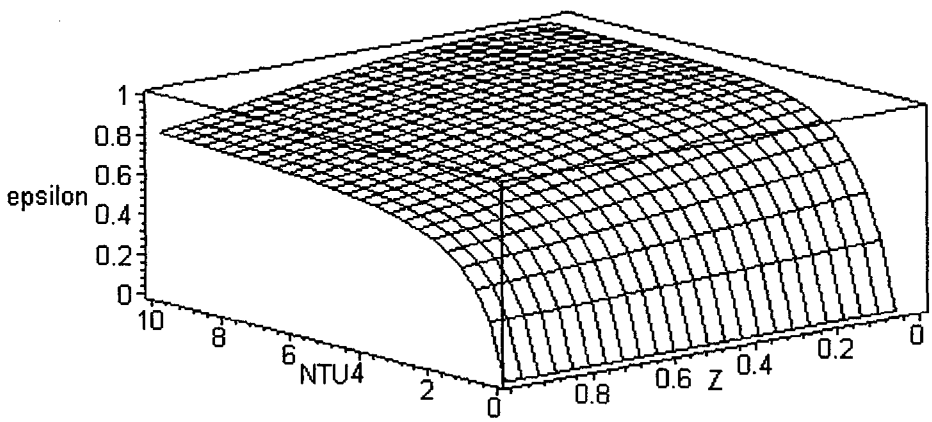

Fig. 2: Exchanger heat transfer effectiveness ε for cross flow heat exchangers, both streams unmixed (Eq. 15)

Outlet Temperatures

Once the effectiveness, ε, is known, the outlet temperature of fluid 1 can be calculated using

$$\boxed{\vartheta_{1,out} = \vartheta_{1,in} + \varepsilon \frac{\dot{C}_{\min}}{\dot{C}_1}\left(\vartheta_{2,in} - \vartheta_{1,in}\right)}$$

Eq. 19

The heat transferred then becomes

$$\boxed{\dot{Q} = \dot{C}_1 \left(\vartheta_{1,out} - \vartheta_{1,in}\right)}$$

Eq. 20

and the outlet temperature of fluid 2 is

$$\boxed{\vartheta_{2,out} = \vartheta_{2,in} - \frac{\dot{Q}}{\dot{C}_2}}$$

Eq. 21

Heat Transfer

The number of transfer units, *NTU*, depends on the product of the heat exchanger area and the overall coefficient of heat transfer from fluid to fluid.

$(U_{avg} \cdot A)$ is calculated as

$$\left(U_{avg}\ A\right) = \frac{1}{\left(\frac{1}{h\ A}\right)_1 + \left(\frac{1}{U\ A}\right)_{metal} + \left(\frac{1}{h\ A}\right)_2}$$

Eq. 22

where indices 1 and 2 refer to the two air sides and A is the corresponding area that the heat transfer coefficients are based on.

The thermal resistance of the two convective heat exchanges denominate the overall heat transfer:

$$\left(\frac{1}{U\ A}\right)_{metal} << \left(\frac{1}{h\ A}\right)_1 + \left(\frac{1}{h\ A}\right)_2$$

Eq. 23

Hence, Eq. 22 can be approximated by

$$\left(U_{avg}\ A\right) \approx \frac{1}{\left(\frac{1}{h\ A}\right)_1 + \left(\frac{1}{h\ A}\right)_2}$$

Eq. 24

The ratio, r, of the convective heat transfer coefficients at nominal conditions can be defined as

Definition:

$$r = \frac{(h\ A)_{1,0}}{(h\ A)_{2,0}}$$

Eq. 25

Using Eq. 24 and Eq. 25, the $(h \cdot A)_0$ values at nominal conditions can be written as

$$(h\ A)_{1,0} = (r+1)\left(U_{avg}\ A\right)_0$$

Eq. 26

and

$$(h\ A)_{2,0} = \frac{(r+1)}{r}\left(U_{avg}\ A\right)_0$$

Eq. 27

Eq. 8 can be solved for the number of transfer units at the nominal operation point:

$$NTU_0 = f(\varepsilon_0, Z, flow)$$

Eq. 28

where ε_0 is known from the boundary condition (Eq. 5) at the nominal operation point. (See Table 1 for the corresponding formula of Eq. 28.)

Once NTU_0 is known, $(U_{avg} \cdot A)_0$ can be expressed by using Eq. 10:

$$\boxed{\left(U_{avg}\ A\right)_0 = NTU_0\ \dot{C}_{min,0}}$$

Eq. 29

The number of transfer units, NTU, depends on the convective heat transfer coefficients, h_i, which are a function of the air velocity. McAdams [McAdams 54] gives an approximation for the convective heat transfer for forced convection. The approximation is based on the measurements of Jurges at room temperature and atmospheric pressure [Jurges 24]. The approximation for the convective heat transfer coefficient in [W/(m²·K)] is

$$h = 7.2\ V_r^{0.78} \quad \text{for } V = 5 \text{ to } 30\ [m/s]$$

Eq. 30

where V is the air velocity along the plate. In HVAC applications, the air velocity is less than 30 m/s. However - particularly in variable volume systems - the air velocity can drop below 5 m/s. In this model, Eq. 30 is nonetheless applied for all air velocities. Fig. 3 shows the error of applying Eq. 30 rather than the linear approximation proposed in [ASHRAE 85].

It can be seen that the error of using Eq. 30 over the whole range of the air velocity lies in an acceptable band for usual HVAC simulations.

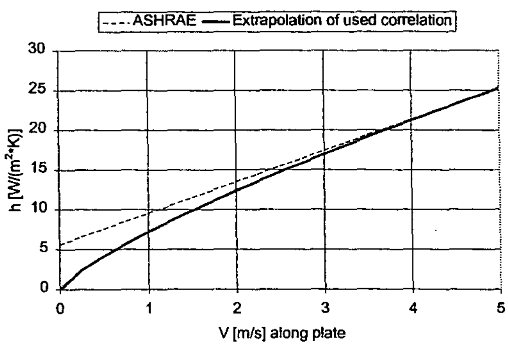

Fig. 3: Comparison between proposed correlation for h according to [ASHRAE 85] and extrapolated correlation.

Since mass velocity is the fundamental variable in forced-convection equations, Schack recommends that the velocity should be corrected according to the ideal gas law for temperatures away from room temperature (70°F, 21.1°C) [McAdams 1954].

The velocity along the plate can be expressed by

$$V = \frac{\dot{m}}{A_{flow}\ \rho}$$

Eq. 31

where A_{flow} is the cross sectional area and ρ is the air density, which, for an ideal gas, is

$$\rho = \frac{p}{R\ T}$$

Eq. 32

with

$$T = \vartheta + 273.15$$

Eq. 33

Substituting Eq. 31 and Eq. 32 into Eq. 30 gives the convective heat transfer coefficient as

$$h = 7.2 \left(\frac{\dot{m}\ R\ T}{A_{flow}\ p} \right)^{0.78}$$

Eq. 34

Using Eq. 34, the $(h \cdot A)_i$ value can also be expressed as

$$\left(h\ A\right)_{i,0} = A_i\ 7.2 \left(\frac{\dot{m}\ R\ T}{A_{flow}\ p} \right)_{i,0}^{0.78}$$

Eq. 35

and

$$\left(h\ A\right)_i = A_i\ 7.2 \left(\frac{\dot{m}\ R\ T}{A_{flow}\ p} \right)_i^{0.78}$$

Eq. 36

respectively.

Dividing Eq. 36 and Eq. 35 gives

$$\frac{(h\,A)_i}{(h\,A)_{i,0}}=\left(\frac{(\dot{m}\,T)_i}{(\dot{m}\,T)_{i,0}}\right)^{0.78}$$

Eq. 37

After inserting Eq. 37 (applied for i = 1 and 2, respectively) into Eq. 24 and using Eq. 26 and Eq. 27 to express the convective heat transfer in terms of its ratio at the nominal conditions and the overall heat transfer at the nominal condition, the overall heat transfer at a given operation point can be calculated by

$$\boxed{(U_{avg}\,A)=\frac{(U_{avg}\,A)_0\,(1+r)}{\left(\left(\frac{(\dot{m}\,T)_{1,in,0}}{(\dot{m}\,T)_{1,in}}\right)^{0.78}+r\left(\frac{(\dot{m}\,T)_{2,in,0}}{(\dot{m}\,T)_{2,in}}\right)^{0.78}\right)}}$$

Eq. 38

In Eq. 38 the density correction is done by using the inlet temperatures as an approximation to the mean air temperatures along the heat exchanger.

We are now left with determining the ratio, r, of the $(h{\cdot}A)_0$ values at the nominal conditions.

One of the main goals of good heat exchanger design is to ensure that

$$\left(\frac{1}{h\,A}\right)_{1,0}\approx\left(\frac{1}{h\,A}\right)_{2,0}$$

Eq. 39

This equality can be assumed to be obtained for plate heat exchangers with the same cross sections for fluid 1 and 2 when operated with same fluid velocities. In this case the ratio, r, equals 1.

However, it is often the case that the mass flows are not equal, due to partial bypassing of exhaust air. Another situation where the ratio, r, is not equal to 1 is if the air temperature and hence the velocities (if equality of the mass flows is assumed) are not similar on both sides of the heat exchanger. In both cases, the ratio, r, can be calculated as follows:
If the cross sections of the heat exchanger are the same on both sides, then the $(h{\cdot}A)_0$ values are equal if and only if the mass flows and air temperatures are the same.

Let $m^*_{i,0}$ and $T^*_{i,0}$ denote the mass flows and temperatures of the heat exchanger. It holds that

$$\dot{m}^*_{1,0}=\dot{m}^*_{2,0}$$
$$T^*_{1,0}=T^*_{2,0}$$

Eq. 40

and consequently

$$(h\,A)^*_{1,0}=(h\,A)^*_{2,0}$$

Eq. 41

Let $m_{i,0}$ and $T_{i,0}$ denote the effective mass flows and temperatures of the heat exchanger and $(h{\cdot}A)_{i,0}$ the corresponding $(h{\cdot}A)$ values. Using Eq. 37, we can write

$$\frac{(h\,A)_{i,0}}{(h\,A)^*_{i,0}}=\left(\frac{(\dot{m}\,T)_{i,0}}{(\dot{m}\,T)^*_{i,0}}\right)^{0.78}$$

Eq. 42

Inserting Eq. 42, solved for $(h{\cdot}A)_{i,0}$, into the definition (Eq. 25) of the ratio, r, and using Eq. 40 and Eq. 41, we get

$$\boxed{r=\left(\frac{(\dot{m}\,T)_{1,0}}{(\dot{m}\,T)_{2,0}}\right)^{0.78}}$$

Eq. 43

At this point we know all the variables needed to calculate the outlet conditions of the heat exchanger.

Interface

The interface to the model is separated into input and output variables. However, some simulation languages ([SPARK 97], [NMF 96]) do not require this separation (i.e., the user determines which variables are inputs and which are outputs).

Parameter

No.	Variable	Description
1	r	ratio between air-side and water-side convective heat transfer coefficient (Eq. 43)
2	-	flow arrangement: 1: counter flow 2: parallel flow 3: cross flow, both flows unmixed
3	$m_{1,0}$	supply air mass flow at nominal operating point
4	$\vartheta_{1,in,0}$	supply air inlet temperature at nominal operating point
5	$\vartheta_{1,out,0}$	supply air outlet temperature at nominal operating point
6	$m_{2,0}$	exhaust air mass flow at nominal operating point
7	$\vartheta_{2,in,0}$	exhaust air inlet temperature at nominal operating point

Input

No.	Variable	Description
1	m_1	supply air mass flow
2	$\vartheta_{1,in}$	supply air inlet temperature
3	m_1	exhaust air mass flow
4	$\vartheta_{1,in}$	exhaust air inlet temperature

Initial value

none

Output

No.	Variable	Description
1	m_1	supply air mass flow
2	$\vartheta_{1,out}$	supply air outlet temperature
3	m_2	exhaust air mass flow
4	$\vartheta_{2,out}$	exhaust air outlet temperature
5	ε	exchanger heat transfer effectiveness
6	Q	heat transfer rate

Algorithm

This section shows conceptually how the formulas are used in a sequential algorithm.

Initalization (executed only once during the simulation)

$\varepsilon_0 \leftarrow C_{1,0},\ C_{2,0},\ \vartheta_{1,in,0},\ \vartheta_{2,in,0},\ \vartheta_{1,out,0}$ (Eq. 5)

$NTU_0 \leftarrow \varepsilon_0,\ Z_0,$ flow arrangement (Eq. 12, Eq. 14 or Eq. 16)

$(U_{avg} \cdot A)_0 \leftarrow NTU_0,\ C_{min}$ (Eq. 29)

store $(U_{avg} \cdot A)_0$

Each call

$(U_{avg} \cdot A) \leftarrow m_1,\ m_2,\ T_{1,in},\ T_{2,in},\ m_{1,0},\ m_{2,0},\ T_{1,in,0},\ T_{2,in,0},\ (U_{avg} \cdot A)_0,\ r$ (Eq. 38)

$NTU \leftarrow (U_{avg} \cdot A),\ C_{min}$ (Eq. 10)

$\varepsilon \leftarrow NTU,\ Z,$ flow arrangement (Eq. 11, Eq. 13 or Eq. 15)

$\vartheta_{1,out} \leftarrow \varepsilon,\ \vartheta_{1,in},\ \vartheta_{2,in},\ C_1,\ C_{min}$ (Eq. 19)

$Q \leftarrow C_1,\ \vartheta_{1,in},\ \vartheta_{1,out}$ (Eq. 20)

$\vartheta_{2,out} \leftarrow Q,\ C_2,\ \vartheta_{2,in}$ (Eq. 21)

Conclusion

Using the ε-NTU relation and the easy obtainable ratio of the $(h \cdot A)$ values of both air streams at the nominal operating point, the $(h \cdot A)$ values at nominal conditions can be determined.

Describing the convective heat transfer coefficient as a power function of the air velocity allows the part-load behavior to be calculated and eliminates the need for geometrical data for the heat exchanger. With this approach, the outlet conditions can be expressed as an explicit function of only the inlet mass flows and temperatures. Furthermore, the model requires only data that are known during the design process of an HVAC system, i.e., the inlet mass flows and temperatures, the supply air outlet temperature at the nominal operating point, and the exchanger flow arrangement.

The explicit model formulation ensures short computation time and eliminates possible convergence problems.

Acknowledgments

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