A how-to for passive thermal displacement ventilation

A DESIGN PROCEDURE FOR Displacement Ventilation

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Displacement-ventilation systems are commonly used in European countries. Different types of systems called "displacement ventilation" were described in our previous article published in *HPAC Engineering*.[1] In response to the reader interest expressed after publication, this article discusses the design procedure for the most commonly used type of system, termed "passive thermal displacement ventilation."

With passive thermal displacement ventilation, supply air is discharged directly into the occupied zone at low velocity near the floor level and at a slightly cooler temperature than the design room temperature. The air from the diffusers spreads along the floor, creating a relatively cool layer of fresh air near the floor. Heat sources within the room (people, process equipment, etc.) create thermal plumes of rising air that entrain this air and carry it up past the human-inhalation zone and eventually up near the ceiling. The warm, contaminated air forms a stratified region in the upper zone of the room, which is exhausted from high-level air returns. This stratification of contaminant levels makes it possible to provide higher-quality air in the occupant breathing zone without increases in system or operating cost. In 1997, the authors reviewed and analyzed the published data on the design and development of passive-thermal-displacement-ventilation systems.[2]

Two approaches are used for passive-thermal-displacement-ventilation design: The first is based on the analytical model, while the second relies on computational-fluid-dynamic (CFD) codes. The analytical approach is by far the most-used method of designing displacement systems. CFD codes can be useful in designing displacement ventilation for large rooms because there is sufficient data in this situation to support an analytical method. CFD also lends itself to large rooms since the dimensions are too large for full-scale measurements and because the design often is unconventional.

The use of CFD codes for practical three-dimensional computation requires expertise, experience, and computational power that usually is unavailable to typical designers. Besides, the prediction of velocities and temperatures in rooms with displacement ventilation using CFD codes generally is inaccurate.

This article presents design guidelines that resulted from the 1995 research project "Design Guide for Displacement Ventilation" conducted by International Air Technologies Inc. and sponsored by Philip Morris Management Corp.[2,3]

APPLICATION

Thermal comfort and indoor-air quality in most industrial and commercial spaces can be equally maintained by mixing type and displacement systems. Displacement ventilation usually is preferable where:

- Contaminants are released in combination with surplus heat.
- Contaminated air is warmer and/or lighter than the surrounding air.
- Supply air is cooler than the ambient air.
- Room height is more than 9.8 ft.
- Cooling load through the air supply does not exceed 12.7 Btuh per sq ft for commercial spaces and 25.4 Btuh per sq ft for industrial spaces with moderate activity levels when regular displacement-ventilation air diffusers are used. With

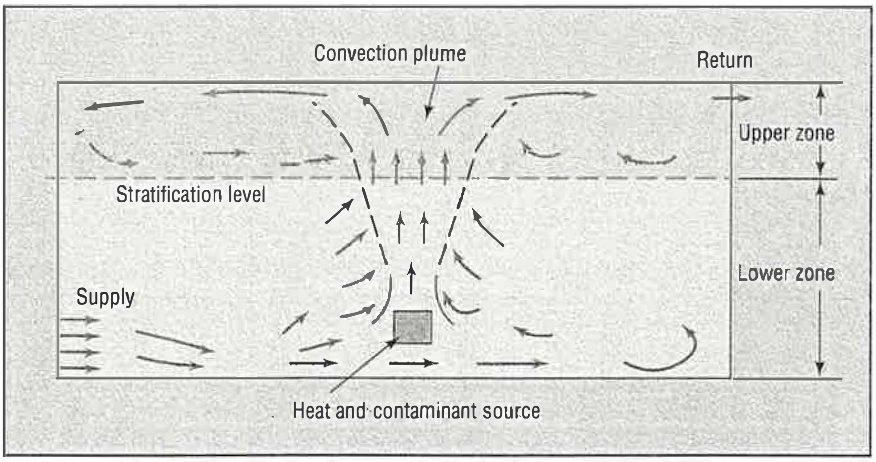

FIGURE 1. Schematic of a passive-thermal-displacement-ventilation system.

induction-type air diffusers, these loads can be increased to 19 Btuh per sq ft and 31.7 Btuh per sq ft, respectively.

- Mechanical disturbances are minor.
- There is room for air diffusers in the occupied zone.

DESIGN PRINCIPLES

Supply-air diffusers are located at or near the floor level, with the supply air introduced directly to the occupied zone. Returns are located at or close to the ceiling, through which the warm room air is exhausted from the room. The supply air is spread over the floor and rises as it is heated by the heat sources in the occupied zone.

Heat sources in the occupied zone release heat by convection and radiation. Convective heat warms the air adjacent to the source; this air moves upward in thermal plumes. Radiant heat warms up the colder surfaces of the room, which become secondary heat sources.

The air volume of the thermal plumes increases as the plumes rise because the plumes entrain ambient air. A stratification level develops where the air-flow rate in the plumes equals the supply-air-flow rate. Thus, two distinct zones are formed within the room: (1) the lower zone below the stratification level with no air recirculation from the upper zone and (2) the upper zone, where the upward thermal plume entrains only the recirculated air from the upper zone (Figure 1). Thermal plumes serve as natural channels by which convective heat and contaminants are transferred from the lower zone to the upper zone. The height of the lower zone depends on the supply-air-flow rate and characteristics of heat sources and their distribution across the floor area. Some researchers and practitioners recommend that displacement-ventilation systems be designed so the lower zone is higher than the occupants and the occupied zone can be ventilated effectively. Others allow the stratification level to be lower than the breathing-zone level, taking into account research data showing that fresh air reaches the inhalation zone from below with a plume created around the person's body.

Secondary heat sources are formed on room surfaces that are heated as a result of heat radiation from primary heat sources. Radiant heat exchange occurs between heated and unheated surfaces and results in the redistribution of heat flow. As a rule, the intensity of heating air with secondary sources is low, with no stable thermal plumes created above them. The heat from the secondary heat sources is transferred to the air of the lower zone. If the primary heat sources in the room are not intense enough or if their surface area is evenly distributed through the room (e.g., a heated floor or spectators sitting close to each other) or all sources of heat are located in the upper zone, no stable thermal plumes will form. The convective heat from these sources is assimilated by the supply air in the occupied zone. The supply air is heated while flooding the lower zone due to the convective heat and is forced into the upper zone by the cooler supply air.

Heat sources

The total heat load (W_o) is introduced by each source by convection (W_{conv}) and radiation (W_{rad}).[4] This relationship can be described by the following equation :

$$W_o = W_{conv} + W_{rad} = \psi W_o = (1-\psi)\ W_o \quad (1)$$

The total radiant component (W_{rad}) of the heat load introduced by each

TABLE 1. Coefficient ψ

	Source surface temperature, C										
ϵ	40	50	60	100	150	200	300	500	800	1000	1200
0.8	0.42	0.44	0.45	0.48	0.45	0.4	0.32	0.2	0.1	0.1	0
0.5	0.52	0.55	0.58	0.59	0.56	0.51	0.42	0.29	0.14	0.1	0.
0.2	0.73	0.76	0.77	0.78	0.76	0.73	0.65	0.59	0.3	0.2	0.14

TABLE 2. Coefficient $\varphi_{horizontal}$

Source location in the room	B/H			
	1	2	3	4
Along room axis	0.3	0.12	0.04	0
Between axis and wall	0.38	0.17	0.11	0.07
Close to wall	0.51	0.3	0.3	0.16

TABLE 3. Coefficient $\varphi_{vertical}$

Source location in the room	B/H			
	1	2	3	4
Along room axis	0.8	0.7	0.65	0.6
Between axis and wall	0.8	0.72	0.67	0.63
Close to wall	0.85	0.75	0.7	0.68

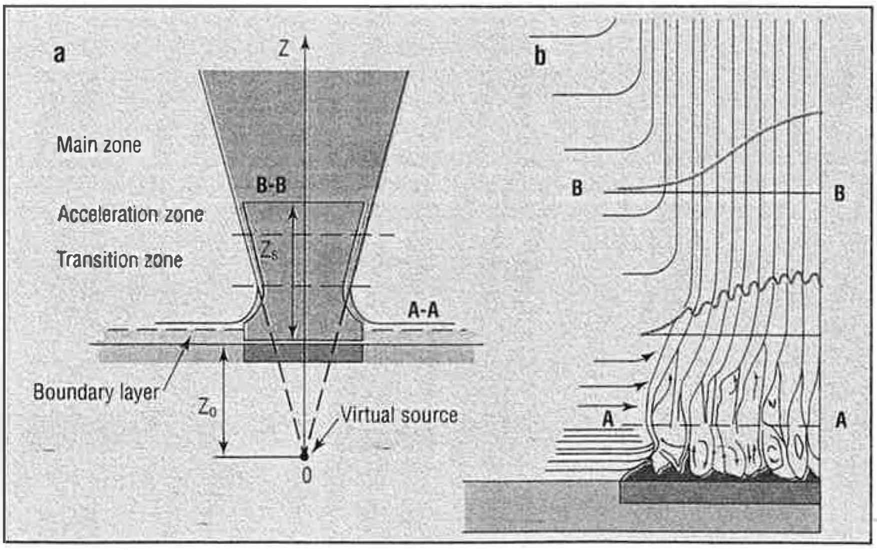

FIGURE 2. Convection flow above a heat source. Notes: Z_s, distance from the source surface to the virtual source; Z, distance from the source surface to the convection flow cross-section of interest; a, convection flow schematic; b, boundary layer. Reproduced courtesy of Elterman, 1980.

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source into the space can be divided between the upper ($W_{rad\,up}$) and the lower or occupied ($W_{rad\,low}$) zones of this space:

$$W_{rad} = W_{rad\,low} + W_{rad\,up} = \varphi (1-\psi)W_o + (1-\varphi)(1-\psi)W_o \quad (2)$$

The total convective component of the heat load introduced by each source:

$$W_{conv} = W_{conv\,low} + W_{conv\,up} = \beta\psi W_o + (1-\beta)\psi W_o \quad (3)$$

where:

ψ, φ and β are non-dimensional coefficients.

ψ = the portion of the convective component of the total heat load released into the space.

φ = the portion of the radiant component of the total radiant heat load in the low zone.

β = the portion of the convective component of the total convective heat load in the low zone.

Coefficients ψ, φ, and β vary within a range of 0 to 1. The value of coefficient ψ depends on the heated surface temperature, while emitance (ϵ) and can be estimated from Table 1.

The value of coefficient φ depends on the source location in the ventilated room (e.g., in the center, close to the wall, etc.) and the source dimensions relative to the room size. Coefficient φ values for horizontal ($\varphi_{horizontal}$) and vertical ($\varphi_{vertical}$) surfaces of small sources (less than 1/10 of the room size) can be estimated using tables 2 and 3.

In tables 2 and 3, $\varphi_{horizontal}$ and $\varphi_{vertical}$ are related to horizontal and vertical surfaces.

Examples of coefficients ψ and φ for some typical heat sources are:

- A sitting or standing person, $\psi = 0.57$; $\varphi = 0.63$.
- Machining equipment, $\psi = 0.5$; $\varphi = 0.6$.

β coefficient values depend on the air-supply method (e.g., $\beta = 0$ with displacement and natural ventilation; $\beta = 1$ with convective plumes dissipating within the occupied zone due to interaction with supply jets, air flows created by moving objects, etc.).

Thermal plumes

Empirical, analytical, and CFD are the commonly used approaches for evaluating air-flow rates in thermal plumes created above people, lights, hot surfaces of process equipment, and other objects with a surface temperature greater than the room air temperature. Information on air-flow rates in thermal plumes is essential for designing displacement-ventilation systems.

Many design procedures for displacement ventilation recommend calculating air-flow rates in thermal plumes using equations derived for so-called "point" and "linear" sources. In reality, heat sources are seldom a point, a line, or a plane vertical surface. The most common approach to accounting for the real source dimensions is to use a virtual source from which the air-flow rates are calculated[4,5,6,7,8] (Figure 2). The virtual origin is located along the plume axis at a distance (Z_0) on the other side of the real source surface. The adjustment of the point-source model to the realistic sources using the virtual-source method gives a reasonable estimate of the air-flow rate in thermal plumes. The weak part of this method is how to estimate the location of the virtual located point source.

The method of a "maximum case" and a "minimum case" provides a tool for such estimation (Figure 3).[8] According to the "maximum case," the real source is replaced by the point source so the border of the plume above the point source passes through the top edge of the real source (e.g., cylinder). The "minimum case" is when the diameter of *vena contracta* of the plume is about 80 percent of the upper surface diameter and is located approximately ⅓ of that diameter above the source. The spreading angle

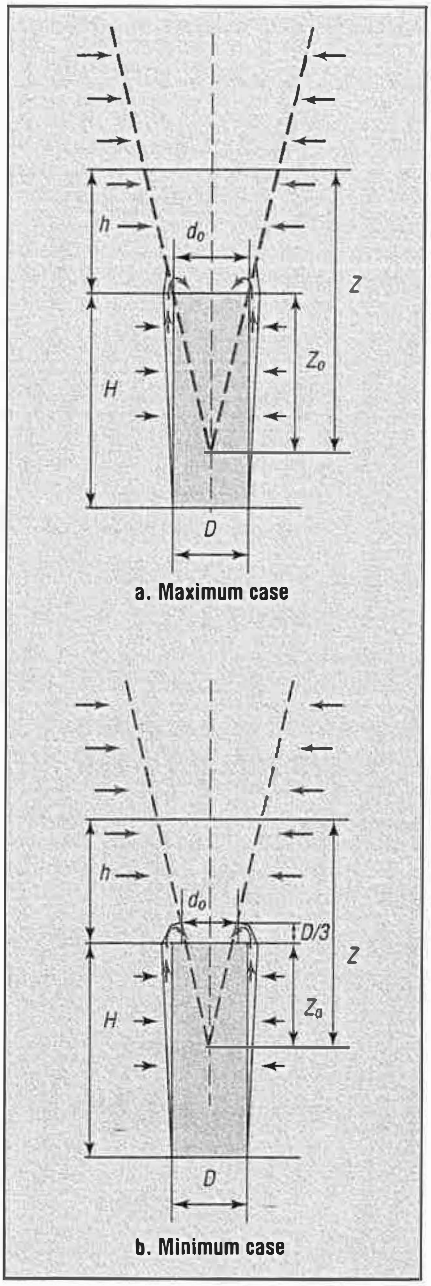

FIGURE 3. Convection flow above a vertical cylinder. Reproduced courtesy of Skistad, 1994.

TABLE 4. *Air-flow rates in convective plumes above typical heat sources in restaurants (@Δt/H = 0)*

No.	Source characteristic	Source size, m			Source strength, W	Height above the bottom of source, m								
		A	B	H		0.1	0.2	0.5	0.75	1.0	1.5	2.0	2.5	3.0
1	Person sitting	Ø = 0.3		1.3	70	2	4	11	18	25	42	62	87	117
2	Person standing	Ø = 0.3		1.8	105	2	4	12	19	38	67	81	95	118
3	TV	0.66	0.66	0.66	300	7	14	39	63	89	137	180	229	278
4	Cash register	0.46	0.46	0.3	500	6	13	36	51	70	121	188	264	332
5	Bottle box	0.1	0.76	0.71	770	8	15	33	52	74	132	212	320	448
6	Mug chiller	1.2	0.76	0.71	830	11	22	53	84	124	181	248	335	448
6	Ice cream cab.	0.46	0.6	0.9	360	5	12	25	38	54	92	142	210	275
8	Hanging lamp	Ø = 0.15		0.1	100	13	15	23	30	40	72	117	164	204

TABLE 5. *Air-flow rates in convective plumes above typical heat sources in restaurants (@Δt/H = 0.5 C/m)*

No.	Source characteristic	Source size, m			Source strength, W	Height above the bottom of source, m								
		A	B	H		0.1	0.2	0.5	0.75	1.0	1.5	2.0	2.5	3.0
1	Person sitting	Ø = 0.3		1.3	70	2	4	11	15	18	26	32	42	26
2	Person standing	Ø = 0.3		1.8	105	2	4	11	13	18	28	36	39	42
3	TV	0.66	0.66	0.66	300	5	10	31	47	61	85	114	142	161
4	Cash register	0.46	0.46	0.3	500	6	13	36	51	69	83	106	134	168
5	Bottle box	0.1	0.76	0.71	770	8	15	33	48	70	104	162	200	241
6	Mug chiller	1.2	0.76	0.71	830	11	22	53	84	124	169	190	195	215
6	Ice cream cab.	0.46	0.6	0.9	360	5	11	25	38	54	72	92	111	129
8	Hanging lamp	Ø = 0.15		0.1	100	7	9	14	22	28	42	50	54	39

TABLE 6. *Air-flow rates in convective plumes above typical heat sources in restaurants (@Δt/H = 1.0 C/m)*

No.	Source characteristic	Source size, m			Source strength, W	Height above the bottom of source, m								
		A	B	H		0.1	0.2	0.5	0.75	1.0	1.5	2.0	2.5	3.0
1	Person sitting	Ø = 0.3		1.3	70	2	4	11	15	18	21	20	20	26
2	Person standing	Ø = 0.3		1.8	105	2	4	11	14	17	20	22	22	13
3	TV	0.66	0.66	0.66	300	5	10	31	45	57	71	85	94	102
4	Cash register	0.46	0.46	0.3	500	6	13	36	51	52	70	84	111	124
5	Bottle box	0.1	0.76	0.71	770	8	15	33	48	65	100	127	149	166
6	Mug chiller	1.2	0.76	0.71	830	11	22	53	84	104	121	148	153	156
6	Ice cream cab.	0.46	0.6	0.9	360	5	11	25	35	47	61	73	77	74
8	Hanging lamp	Ø = 0.15		0.1	100	7	9	13	17	23	32	32	9	-

TABLE 7. *Air-flow rates in convective plumes above typical heat sources in restaurants (@Δt/H = 2.0 C/m)*

No.	Source characteristic	Source size, m			Source strength, W	Height above the bottom of source, m								
		A	B	H		0.1	0.2	0.5	0.75	1.0	1.5	2.0	2.5	3.0
1	Person sitting	Ø = 0.3		1.3	70	1	3	7	9	12	14	9	-	-
2	Person standing	Ø = 0.3		1.8	105	1	3	6	10	14	16	17	5	-
3	TV	0.66	0.66	0.66	300	6	11	35	46	54	55	52	47	7
4	Cash register	0.46	0.46	0.3	500	6	13	36	51	68	66	45	7	-
5	Bottle box	0.1	0.76	0.71	770	8	15	31	41	59	90	107	99	69
6	Mug chiller	1.2	0.76	0.71	830	11	22	53	67	100	99	78	55	25
6	Ice cream cab.	0.46	0.6	0.9	360	5	10	23	30	39	50	49	38	34
8	Hanging lamp	Ø = 0.15		0.1	100	7	9	13	16	21	18	-	-	-

of the plume is set to 25. For low-temperature sources, the "maximum case" is recommended, whereas the "minimum case" best fits the measurements for larger, high-temperature sources.[8]

Also, in most design procedures, air-flow rates in thermal plumes are calculated without consideration of their interaction with each other or with different surfaces. The effects of confinement by surrounding walls and temperature stratification along the room height often are overlooked. The results of numerous research studies indicate that these factors have a significant effect on thermal-plume characteristics.

For example, the driving force of the plume is the temperature difference between the plume and the room air. When this difference diminishes, the plumes will disintegrate and spread horizontally in the room.

The analysis of these effects and recommended design-equation examples are given in Chapter 7 of the "Industrial Ventilation Design Guidebook," which will be published in 2001 by Academic Press. The influence of the temperature gradient ($\Delta t/H$ = air temperature difference between the floor and the ceiling levels over the room height) on air-flow rate in the thermal plume is illustrated in tables 4, 5, 6, and 7 for heat sources typical for dining areas in restaurants.[9]

INPUT DATA

The following data are required for system design:

- Room size: L_r - length in meters; B_r - width in meters (floor area, $A_{room} = L_r \times Br$); H_{room} - room height in meters.
- Location and number of people.
- Type of human activity.
- Location, number, and specification of process equipment and other heat and contaminant sources.
- Requirements to the space environment (occupied zone air): t_{oz}, T_{oz}- occupied zone temperature (°C, °K); V_{oz} - maximum occupied zone velocity (close to floor level) (m/s); C_{oz} - contaminant concentration in the breathing zone (mg/m^3).
- Cooling load due to heat losses/gains through the building envelope (W_{ext} , W/m^2).
- Minimum air-flow rate to be supplied into the space.

DESIGN PROCEDURE

There are four primary assumptions for a displacement-ventilation-design procedure for rooms with primarily heat-removal requirements. They are:

- Temperature stratification is a linear function (there is no step stratification as there is with a contaminant-concentration distribution), $\Delta t = (t_{exh} - t_{floor})/H_{room}$. Heat balances and radiant and turbulent heat exchange are calculated for two zones: the lower zone limited by the height of the occupied zone and the upper zone above the occupied zone.
- Occupied-zone temperature is the air temperature at the height of h_{oz}—that is, 3.6 ft for spaces with a predominant seating activity and 5.9 ft for spaces with a standing activity. Occupied-zone temperature at these heights is considered to be the same throughout the occupied-zone area outside the direct influence of the supply-air flow.
- The temperature difference between the head level (h_{oz} = 1.1 m or 1.8 m) and the ankle level (h_{floor} = 0.1 m) is limited for the comfort reason by 2-3 C. This results in restriction of temperature gradient ($\Delta t/H$) along the room height by 2 to 2.5 C per m with a seating activity and 1.2 to 1.8 C with a standing activity.
- Heat-removal-coefficient evaluation is based on a standard model.[10]

Calculation goals

Calculations are conducted to obtain the following information:

- Heat-removal-efficiency coefficient (Kt) value.
- Supply-air-flow rate for heat-removal purpose, G_{ot}.
- Supply-air temperature, t_o.
- Exhaust-air temperature t_{exh}.
- Vertical temperature gradient ($\Delta t/H$).

Design algorithm

Step 1: List all heat sources in the room.

Step2: Calculate the average convective heat component (ψ) using the data for the individual heat sources:

$$\psi = \frac{\Sigma(W_i \times \Psi_i)}{\Sigma W_i} \tag{4}$$

Step 3: Calculate the averaged radiant heat component into the occupied zone (φ) using the data from Table 1.

$$\varphi = \frac{\Sigma(W_{rad\,i} \times \varphi_i)}{\Sigma W_{rad\,i}} = \frac{\Sigma[\varphi_i \times (1-\Psi_i) \times W_i]}{\Sigma[W_i \times (1-\Psi_i)]} \tag{5}$$

Step 4: Calculate the heat-removal coefficient (K_{to}) used as the base for the iteration process:

$$K_{to} = \frac{1}{\varphi(1-\Psi)} \tag{6}$$

Step 5: Select the supply-air-temperature difference, $\Delta t_o = t_{oz}-t_o$ (approximately 3 C), based on the air diffuser's performance data, the type of human activity, and the distance between the air diffuser and the nearest person.

Step 6: Calculate the preliminary value of the supply-air-flow rate (G_{ot}, kg/h) required for heat removal, using Kt = 0.5 K_{to} for the first iteration:

$$G_{ot} = \frac{\Sigma W_i}{C_p \Delta t_o K_t} \tag{7}$$

Step 7: Calculate the adjusted heat-removal coefficient, K_t *:

$$K_t^* = \frac{1 + \dfrac{K\,\alpha_{rad} A_{room} + \alpha_{turb} A_{room}}{C_p G_o}}{\varphi(1-\Psi) + \dfrac{K\,\alpha_{rad} A_{room} + \alpha_{turb} A_{room}}{C_p G_o}} \tag{8}$$

where:

$$\alpha_{turb} = \frac{3600\, C_p\ Y\ A_o}{\left(1 + 3.3 \dfrac{g}{T_{o.z.}} \dfrac{\Delta t_o (K_t - 1)}{(H_{room} - h_{o.z.})} \left(\dfrac{dz}{dV}\right)^2\right)^{1.5} (H_{room} - h_{o.z.})} \tag{9}$$

$$K = 1 + 0.16\frac{G_o}{1+2\frac{H_{room}}{\sqrt{A_o}}} \times \left(1-\varphi-\Lambda\varphi\right) - 2.87 \times \frac{1+\Lambda}{1+2\frac{H_{room}}{\sqrt{A_{room}}}} \qquad (10)$$

$$\Lambda = 0.46\left[\frac{\left(1-\varphi\right)\left(1-\psi\right) - 17.85\frac{1-1/K_t}{G_o/A_{room}}}{\varphi\left(1-\psi\right) + 17.85\frac{1-1/K_t}{G_o/A_{room}}}\right]^{1/4} \qquad (11)$$

where:

α_{rad} and α_{turb} = radiant and turbulent heat flux from the upper zone to the lower zone; W/m^2 K; A_o = turbulent exchange coefficient, m^2/s; Cp = specific heat of air at constant pressure, kJ/(kg K); γ = specific weight of air, kg/m^3; dV/dz = velocity gradient, 1/s.

For a realistic room, A_o is between 0.3 and 0.4 m^2/s, dV/dz is between 0.075 and 0.1 1/s, T_{oz} is 293K, γ is 1.2 kg/m^3, g is 9.8 m/s^2, Cp is 1kJ/(kg K) (specific heat of air at constant pressure), α_{rad} is 17.85 W/m^2, and K is radiant heat flux from the upper zone to the lower zone. Thus, Equation 9 can be rewritten as follows:

$$\alpha_{turb} = \frac{1500}{\left(1 + 20\frac{\Delta t_o\left(K_t - 1\right)}{\left(H_{room} - h_{o.z.}\right)}\right)^{1.5}\left(H_{room} - h_{o.z.}\right)} \qquad (12)$$

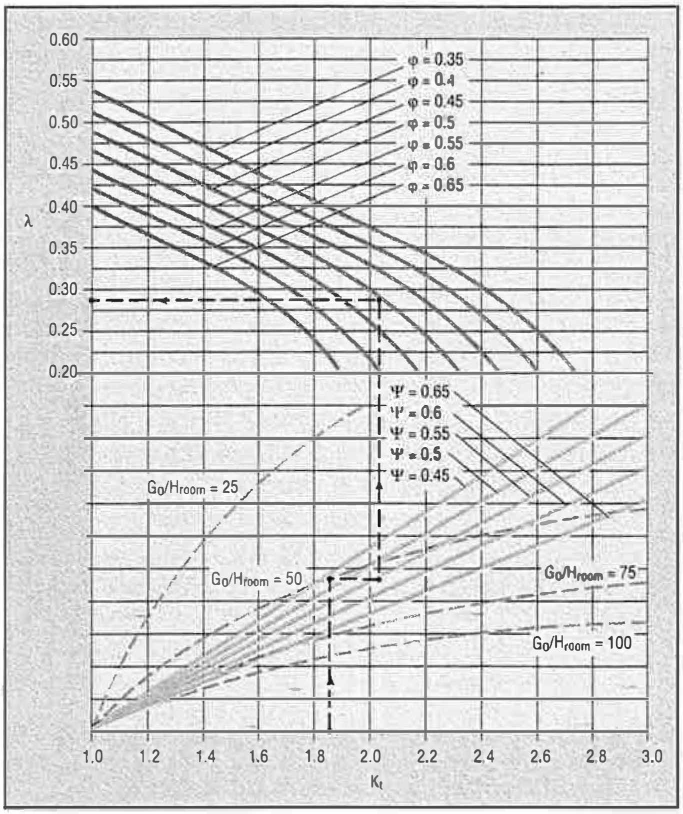

FIGURE 4. Graph for parameter λ evaluation.

To simplify the K_t coefficient evaluation, the following procedure can be used:

- Obtain λ using the graph in Figure 4.
- Calculate: $K = F_1 - F_2$, where F_1 and F_2 can be obtained using graphs in Figure 5.

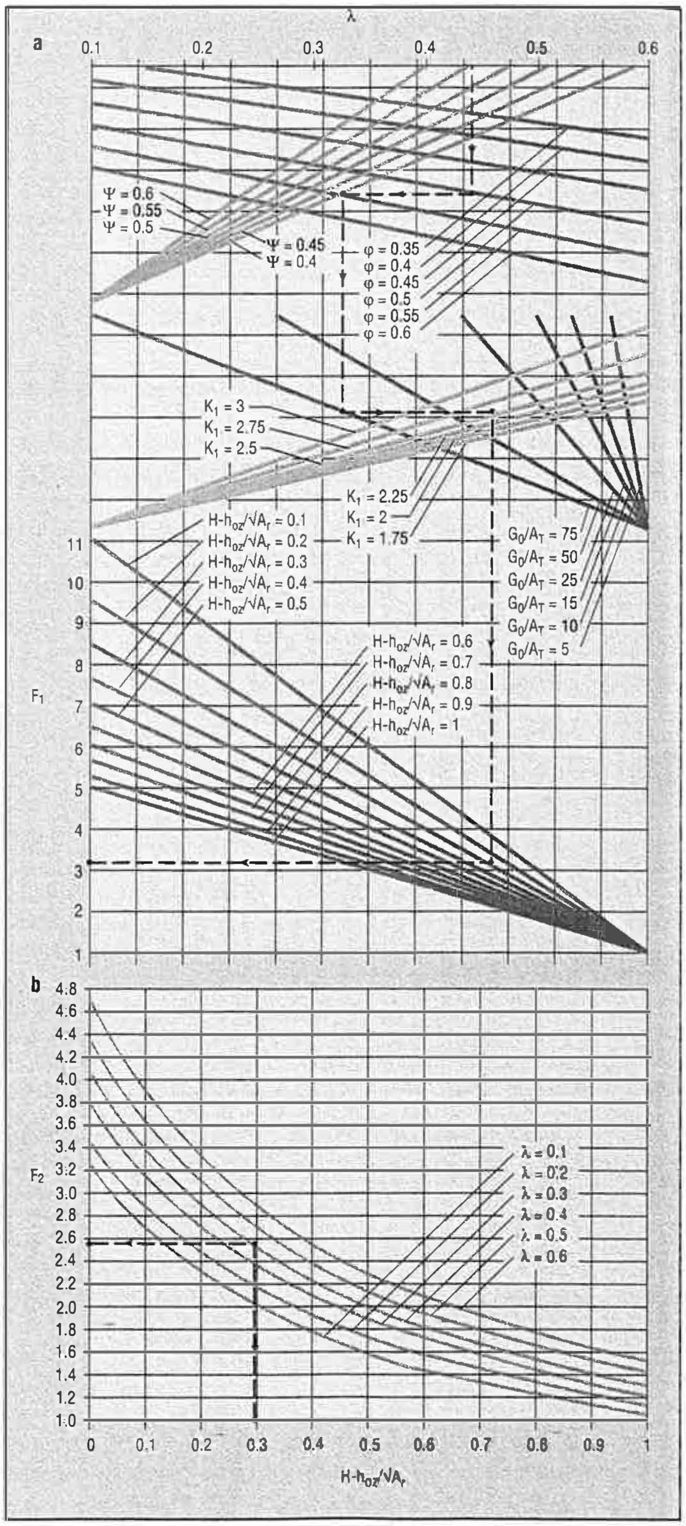

*FIGURE 5. Supporting graphs for **K = F1-F2** evaluation: a-F1, b-F2.*

- Obtain α_{turb} using graphs in Figure 6.
- Obtain K_t using graphs in Figure 7.

Step 8: Compare K_t *, calculated using Equation 8 or graphs in figures 4 through 7, with K_t calculated as 0.5 K_{to}.

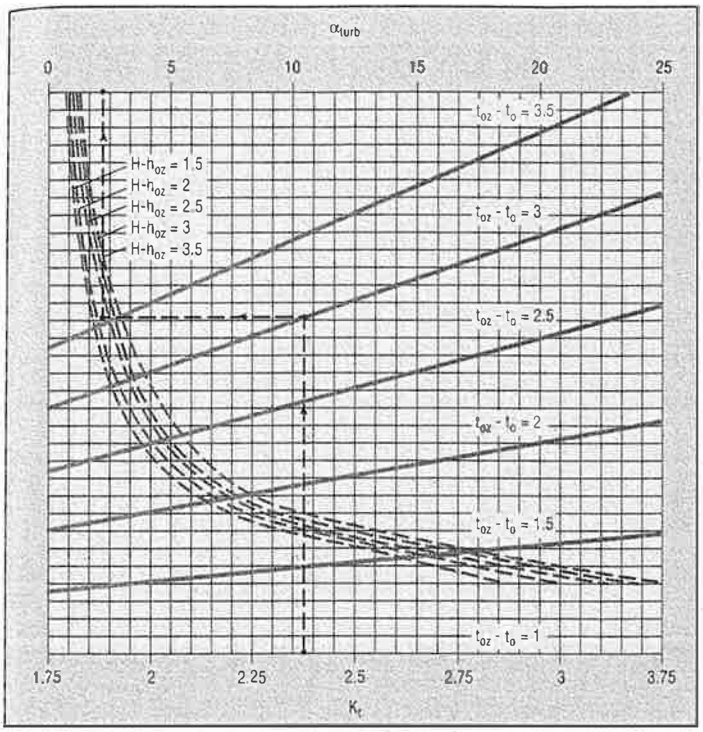

FIGURE 6. Graph for parameter α_{turb} evaluation.

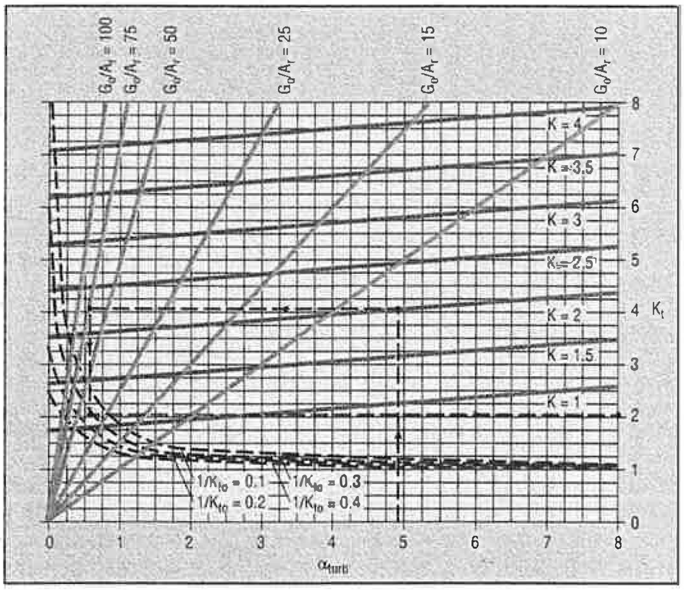

FIGURE 7. Graph for heat-removal-coefficient K_t evaluation.

If $(K_t^* - K_t)/K_t^*$ is less than 0.1, proceed starting with Step 8. If $(K_t^* - K_t)/K_t^*$ is greater than 0.1, assume $K_t = K_t^*$ and repeat the calculations in Step 8.

Step 9: Calculate the exhausted-air temperature ($t_{exh} = t_o + K_t \Delta t_o$).

Step 10: Calculate the supply-air temperature (t_o), given the occupied zone temperature t_{oz}

$$t_o = t_{oz} - \Delta t_o \qquad (13)$$

Step 11: Calculate the temperature gradient ($\Delta t/H$) along the room height:

$$\Delta t / H = \frac{t_{exh} - t_{o.z.}}{H_{room} - h_{o.z.}} = \frac{\Delta t_o (K_t - 1)}{H_{room} - h_{o.z.}} \qquad (14)$$

If $\Delta t/H$ is greater than the prescribed one to achieve thermal comfort, decrease Δt_o and repeat calculations starting with Step 6.

Step 12: Calculate supply-air-flow rate (G_o) required for heat removal, with the final values of K_t and Δt_o:

$$G_o = \frac{\Sigma W_i}{C_p \Delta t_o K_t} \qquad (15)$$

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In the second part of this two-part series, to appear in a future issue of HPAC Engineering, *the authors will detail the displacement-ventilation-design procedure for rooms with heat- and contaminant-removal requirements and discuss air-diffuser selection and location considerations. Also, they will include a case study demonstrating the proper use of the design procedure presented in this series.*