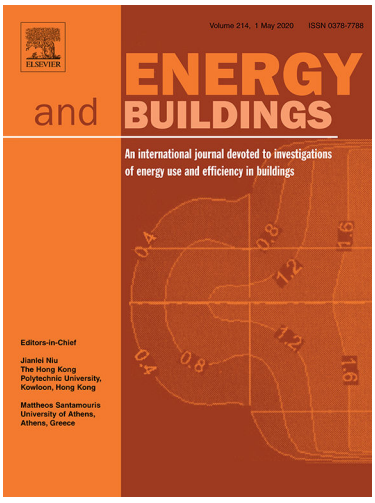
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A comparative study on convective heat transfer in indoor applications

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A COMPARATIVE STUDY ON CONVECTIVE HEAT TRANSFER IN INDOOR APPLICATIONS

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ABSTRACT

Analytical solutions may not always be applicable in the calculation of the turbulent flow convective heat transfer rate, unlike the radiative and the conductive ones. Therefore, experimental correlations on convective heat transfer coefficient have been developed in enclosures. Convective heat transfer in a cavity is classified as natural, forced, and mixed convection on the basis of the driving forces (buoyant or mechanical). In recent years, there has been an increasing interest in the mixed convection, particularly in cooled ceiling – displacement ventilation indoor applications. It seems this interest is tending to increase while contamination of viruses and energy saving are growing as health and environmental concerns worldwide. Hence, the reasons behind the interest in mixed convection applications have been investigated along the paper.

This comparative study seeks to explain the progress of convective heat transfer at indoor applications by reviewing mostly experimental correlation studies in time. The mixed convection has not been widely studied experimentally in indoor applications in comparison to natural and forced convection. Therefore, this study is devoted to indicate this gap in the literature on this issue and it includes all convection types with a wide and up-to-date review, descriptions, explanations, and comparisons, as well. Moreover, almost all empirical correlations on the topic are given in tables in detail. It can be concluded that general correlations for mixed convection applications is needed. Correlations related to radiant floor cooling applications are nearly non-existent. Additionally, more experimental studies are required for various split air conditioner cases. These gaps in the literature are unveiled and comparison of applications with various convection types have been made as a first comparative study in the literature.

Keywords: *Convective Heat Transfer Coefficient; Natural Convection; Mixed Convection, Forced Convection; Room Ventilation.*

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1. INTRODUCTION

Nowadays, instead of experiments, thermal studies for building applications are often carried out using thermal and fluid analysis tools to save time and cost. Although equations derived from analytical solution methods can be used for conductive and radiative heat transfer analyses, they are not practical for turbulent flow convective heat transfer calculations in cavities/enclosures. Therefore, obtaining empirical correlations on convective heat transfer coefficients plays a vital role in numerical heat transfer calculations. In this current work, foremost

experimental based studies are evaluated and tabulated. Also, ventilation types such as displacement ventilation and mixing ventilation are elaborated and thoroughly contrasted with each other from a thermal comfort, energy, heat transfer, and human welfare point of view.

Recently, researchers have shown an increasing interest in displacement ventilation and mixed convection on air conditioning applications. Within the content of the present paper, the reasons of the recent interest are clarified and comparative assessments based on the specific criteria have been conducted. It can be stated that the interest in the displacement ventilation applications in public creates an increased demand from researchers to perform indoor mixed convection experiments while contaminants in the air is a growing public health concern worldwide.

In recent years, a considerable literature has grown up around the theme of indoor convection heat transfer. However, there have been few review studies that give place to indoor mixed convection applications. In addition to significant studies on natural and forced convection correlations, the mixed convection studies are widely investigated in this work. Prior to summarizing, the following common terms are explained below.

As reviewing the developed standards for designing air conditioning systems while providing methods to determine the thermal output for HVAC systems by means of calculation and test methods, it can be concluded that the widely used ones are; EN1264, ISO 11855, EN 15377, REHVA (guidebook), and ASHRAE (handbook-system and equipment). It is known that the amount of heat removed/gained from a heated/cooled surface is comprised of the radiation and convection heat transfer. All aforementioned standards and guidebooks (apart from ASHRAE) propose calculation methods using total heat transfer coefficient. Thus, as the subject of this study is convective heat transfer in indoor applications, standards focusing on total heat transfer coefficients are not mentioned.

In order to determine the convective heat transfer coefficients (CHTC), the key point is assigning the appropriate reference temperature. In air convection studies, the reference temperature basically gives us the measurement point of the air such as; the center of the room, supply-air/room-inlet, exhaust-air/room outlet, local air, or the average of inlet and outlet air in order to determine the temperature differences between surface and the air in the convection heat transfer equations. The reference temperature should be given in all convection types (natural, forced, and mixed) to reuse, compare, and develop the correlations for other researchers. Generally, in natural convection the air temperature at the center of the room is selected as reference temperature. Particularly in forced and mixed convection cases, the reference temperature can directly affect convective correlations. Therefore, experiments have been carried out to investigate the most suitable reference temperature for proposed CHTC correlations [1–6].

The other parameter that should be given with CHTC correlations is cooled/heated/thermal-activated/investigated “surface”. That internal room surface (floor, ceiling, wall) represents a specified surface through which the aimed heat transfer occurs. The given CHTC correlations cannot be applicable for all surfaces in the room but only specified one. Furthermore, the surface must be well explained such as cooled/ heated for natural convection. For forced convection, airflow type (hot, cold), flow direction (parallel/perpendicular to the surface, upward/downward), diffusion type (radial/longitudinal/rectangle), diffuser location (at ceiling/floor/wall) should be given, as well. For mixed convection, all the information mentioned above should be stated for other researchers to apply CHTC correlations.

2. NATURAL CONVECTION

One of the most important heat transfer phenomena in the building heating/cooling applications is the natural convection that takes place through building’s heated or cooled surfaces. The driving mechanisms in natural convection in an enclosure is the temperature difference between the indoor air and internal surfaces that results in heat convection. In order to determine the heat transfer characteristics of a heated or cooled enclosure, the convective heat transfer has a crucial role with CHTCs and convective heat fluxes. As Peeters et al. [7] also state the convective heat transfer mechanisms need experimental results due to the complexity of its mechanisms in determining the heat transfer characteristics in an enclosure while the radiative and conductive heat transfer mechanisms occurring in the system can accurately be defined/calculated using mathematical models. Therefore, it can be clearly seen in the literature that there exists a variety of experimental studies conducted on this issue. In the early term, experimental studies used the horizontal or vertical plates to understand the convective heat transfer mechanisms that do not match the real building applications. As Awbi and Hatton [8] state, the number of

experimental studies conducted in the full-sized enclosures that represent the real thermal conditions has increased for three decades. As a result of these experimental studies, the authors propose correlations and coefficients to determine the convective heat fluxes and heat transfer coefficients. In the correlations of convective heat fluxes, the heat flux is given as the function of temperature difference and CHTC as indicated in the "Newton's law of cooling". On the other hand, in the given equations for the CHTCs there can be dimensionless parameters to identify them, such as, the Nusselt (Nu), Rayleigh (Ra), Prandtl (Pr) and Grashof (Gr) numbers.

Alamdari and Hammond [9] propose equations to determine the convective heat transfer for a building with natural ventilation. They define the CHTC as a function of temperature differences, surface length, and the type of fluid in the room. When it is defined as dimensionless numbers;

$$Nu = C\,Ra^{n} \tag{1}$$

In Eq. 1, they suggest the value of "n" as 1/4 for the Rayleigh number in the range of 10^4 <Ra <10^5, and in the case of Ra> 10^5, they determine the number "n" as 1/3. In the calculations, it is stated that when determining the physical properties of the air in the room (assuming that it does not change significantly), the considered temperature can be taken as the average value of the floor surface temperature and the room air temperature. As a result, Eq. 2 and Eq. 3 are proposed when determining the CHTC and Nusselt number.

$$h = \left[\left\{a\left(\frac{\Delta T}{L}\right)^{p}\right\}^{m} + \{b(\Delta T)^{q}\}^{m}\right]^{1/m} \tag{2}$$

$$Nu = \left[(A\,Ra^{p})^{m} + (B\,Ra^{q})^{m}\right]^{1/m} \tag{3}$$

Where the values of A, B, a, b, p, q, and m are given in Table 1 for vertical and horizontal surfaces.

Table 1 Coefficient values in the h and Nu equations

	A	**B**	**a**	**b**	**p**	**q**	**m**
Vertical surfaces	0.58	0.11	1.50	1.23	1/4	1/3	6
Horizontal surfaces	0.54	0.14	1.40	1.63	1/4	1/3	6

Therefore, the correlations derived to determine convection heat transfer coefficients for horizontal surfaces are stated in Eqs. 4 and 5.

$T_f > T_a$ – Floor heating;

$$h = \left[\left\{1{,}4\left(\frac{\Delta T}{L}\right)^{1/4}\right\}^{6} + \{1{,}63(\Delta T)^{1/3}\}^{6}\right]^{1/6} \tag{4}$$

$T_f < T_a$ – Floor cooling;

$$h = 0{,}6\left(\frac{\Delta T}{L}\right)^{1/5} \tag{5}$$

Awbi and Hatton [8] state that there are significant differences between the values of convection heat transfer coefficients in closed volumes when they are mostly determined by using small scale, horizontal, and vertical plates. It is noted that the CHTC should be determined under appropriate experimental conditions in order to accurately determine the thermal state in the rooms. For this purpose, two test rooms of different sizes have been built.

They conclude that the effect of the dimensions on the CHTC in the case of floor heating is more limited than in the case of wall heating. As a result of the experimental study, the authors propose new equations to define convective heat transfer characteristics of wall, floor, and ceiling heating systems via CHTCs.

Floor heating;

$$Nu = 0.269\,(Gr)^{0.308} \tag{6}$$

$$h = \frac{2.175}{D^{0.076}}\,(\Delta T)^{0.308} \tag{7}$$

where, "D" is the hydraulic diameter and is determined by Eq. 8.

$$D = \frac{4\ x\ Area}{Perimeter} \tag{8}$$

In the study, Eqs. 9 - 10 are proposed to determine the CHTC for heating from ceiling or cooling from floor, while Eqs. 11 - 12 are for heating from wall.

$$Nu = 2.376\ (Gr)^{0.133} \tag{9}$$

$$h = \frac{0.704}{D^{0.601}} (\Delta T)^{0.133} \tag{10}$$

$$Nu = 0.289\ (Gr)^{0.293} \tag{11}$$

$$h = \frac{1.823}{D^{0.121}} (\Delta T)^{0.293} \tag{12}$$

Min et al. [10] conduct experimental studies for determination of CHTCs over surfaces in enclosed volumes. In the study, correlations are obtained in the case of no ventilation occurs in the room, i.e. the natural convection mechanism is dominant. Min et al. [10] propose Eqs. 13 - 14 to determine CHTC for floor heating and cooling, respectively:

Floor heating:

$$h = 2.13 \frac{\Delta T^{0.31}}{p^{0.08}} \tag{13}$$

Floor cooling:

$$h = 0.134\ \Delta T^{0.25} \tag{14}$$

An experimental study that simulates the realistic thermal conditions of a floor heating system with wall cooling is performed by Açıkgöz et al.[11]. As a result of the study, the authors propose a correlation in the determination of CHTC for such floor heating systems. The proposed correlation is shown in Eq. 15.

$$h = 1.79\ (T_f - T_a)^{0.28} \tag{15}$$

Karadağ [12] conducts a numerical study to define the heat transfer characteristics of a ceiling cooling system. Numerical simulations are done for a variety of cases including different surface emissivities, room dimensions, and surface temperatures. Consequently, Karadağ [12] suggests a new correlation for the determination of CHTC to use in the ceiling cooling applications. The suggested correlation is given in Eq. 16.

$$h = 3.1\ (T_a - T_c)^{0.22} \tag{16}$$

Khalifa [13] reviews the studies on CHTCs over the surfaces of two and three-dimensional structures. In the study, it is stated that there are noteworthy differences between the correlations used in the literature. Correlations developed for three-dimensional volumes are summarized in Table 2.

Table 2 Natural convective heat transfer coefficient equations

Reference	Correlation	Applied surface
Li et al. [14]	$h\ =\ 3.08\ (\Delta T)^{0.25}$	Floor heating
Schlapmann [15]	$h = \left(0.33\frac{H}{L} + 0.75\right)(T_r - T_{wall})^{1/3}$	Floor heating
Khalifa and Marshall [16]	$h\ =\ 1.983\ (\Delta T)^{0.25}$	Floor heating
Khalifa and Marshall [17]	$h =\ 1.983\ (\Delta T)^{0.25}$	Wall / Floor heating

Table 3 Correlations used to estimate the natural convective heat transfer coefficient

Reference	Surface	Correlation		Remarks
Ashrae [18]	Vertical plate	$h = 1.33\left(\frac{\Delta T}{H}\right)^{1/4}$	$10^5 < Ra < 10^9$	
		$h = 1.26(\Delta T)^{1/3}$	$Ra < 10^9$	
	Horizontal Plate	$h = 0.53\frac{(\Delta T)^{1/6}}{L^{1/2}}$	$1 < Ra < 2002$	Downward cooled plate Upward facing heated plate
		$h = 1.53\frac{(\Delta T)^{1/4}}{L^{1/4}}$	$200 < Ra < 104$	
		$h = 1.4\frac{(\Delta T)^{1/4}}{L^{1/4}}$	$2.2x10^4 < Ra < 10^6$	
		$h = 1.8(\Delta T^{1/3})$	$8x10^6 < Ra < 1.5x10^9$	
		$h = 0.7\frac{(\Delta T)^{1/4}}{L^{1/4}}$	$10^5 < Ra < 10^{10}$	Downward facing heated plate or upward cooled plate
Ashrae [19]	Enclosed volume	$h = 2.42\ \lvert T_t - T_h\rvert^{0.31} / D^{0.08}$		Floor heating
		$h = 0.20\ \lvert T_t - T_h\rvert^{0.25} / D^{0.25}$		Ceiling heating
		$h = 2.13\lvert T_t - T_h\rvert^{0.31}$		Floor heating - Simplified
		$h = 0.134\lvert T_t - T_h\rvert^{0.25}$		Ceiling heating - Simplified
Min [10]	Enclosed volume	$h = 2.123\frac{(\Delta T)^{0.31}}{P^{0.08}}$		Floor heating / Ceiling cooling
		$h = 0.233\frac{(\Delta T)^{0.25}}{P^{0.25}}$		Ceiling heating / Floor cooling

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Table 3 Correlations used to estimate the natural convective heat transfer coefficient (Cont.)

Reference	Surface	Correlation	Remarks
Alamdari and Hammond [9]	Floor	$h = \left[\left[1.4\left(\frac{\Delta T}{H}\right)^{1/4}\right]^6 + \left[1.63(\Delta T)^{1/3}\right]^6\right]^{1/6}$	$T_s > T_a$
	Ceiling	$h = 0.6\left(\frac{\Delta T}{P}\right)^{1/5}$	$T_a > T_s$
Açıkgöz [20]	Wall	$h = 1.5\left[\frac{H}{L}\right]^{0.09}(\Delta T)^{0.47}$	Wall heating
Churchill and Chu [21]		$h = \frac{0.0175}{H} + 1.31\left(\frac{\Delta T}{H}\right)^{1/4}$	Vertical plate
		$h = \frac{0.0275}{H} + \left(0.825 + 7.01(\Delta T)^{\frac{1}{6}}H^{\frac{3}{6}}\right)^2$	
		$h = \frac{0.0175}{H} + 0.28\left(\frac{\Delta T}{H}\right)^{1/4}$	
		$h = \frac{0.0275}{H} + (0.825 + 7.08(\Delta T)^{\frac{1}{6}}H^{\frac{3}{6}})^2$	
Awbi and Hatton [8]	Enclosed volume	$h = \frac{2.175}{D^{0.076}}(\Delta T)^{0.308}$ $Nu = 0.269(Gr)^{0.308}$	Floor heating
		$h = \frac{0.704}{D^{0.601}}(\Delta T)^{0.133}$ $Nu = 2.376(Gr)^{0.133}$	Ceiling heating
Khalifa and Marshall [17]		$h = 2.30(\Delta T)^{0.24}$	Wall heating
		$h = 2.92(\Delta T)^{0.25}$	Wall heating (opposed wall)
		$h = 8.07(\Delta T)^{0.11}$	Wall heating (partial)

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		$h = 2.12(\Delta T)^{0.33}$	Ceiling cooling

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Also, correlations have been proposed by Ashrae [18] to determine CHTCs in enclosed volumes. In the proposed correlations, the effect of airflow in the room has been neglected. Correlations developed by various researchers and Ashrae [18] are summarized in Table 3.

In the literature, there exists a group of studies concerning both convective heat fluxes and heat transfer coefficients. While some of these experimental studies recommend CHTCs as a value or within an interval, some of them define the heat fluxes to be occurring in heating/cooling systems. However, according to a study performed by Hu and Niu [22], even though there exist lots of experimental and numerical studies concerning the determination of heating/cooling capacities of HVAC systems, the load estimation problems are still continuing. Thus, in order to eliminate such problematic issues related to determination of heating/cooling loads of HVAC systems, further studies are needed.

Hanibuchi and Hokoi [23] experimentally investigate the heat transfer characteristics of a 6 m x 4 m x 2.45 m test chamber that is heated from floor. In the study, they examine surface temperatures, indoor air temperature distribution, and air velocity distribution. It is concluded that, a homogeneous temperature distribution is observed in the test room, which is heated from floor, except for the areas close to windows. When convection and radiative heat transfer rates are compared, it is seen that convection heat transfer accounts for almost half of the total heat transfer.

Karakoyun et al. [24] examine the effect of uninsulated walls, simulates the effect of solar radiation, on cooling loads of floor cooling systems. Walls of the enclosure are used as heat gain sources separately. One of the outputs of study is proposing a correlation that can be used in defining convective heat flux of floor cooling system. The suggested correlation is given in Eq. 17.

$$q_c = 0.67\ (T_a - T_f)^{0.90} \tag{17}$$

Evren et al. [25] determine CHTCs for floor heating systems using a test facility that is constructed according to the Ashrae 138 standard. It is stated that heat transfer coefficients are important design parameters in determination of heating and cooling loads of air conditioning systems. In the study, radiative and CHTCs for floor heating system are determined experimentally and proposed as 5.4 W/m^{2}K and 3.7 W/m^{2}K, respectively.

Olesen et al. [1] experimentally investigate the heat transfer that occurs through radiant cooled floor system using a 6 m x 4 m x 2.8 m test chamber. It is concluded that the temperature difference between radiant floor surface and indoor air temperature, and a CHTC of 1.0 W/m^{2}K can be used in calculation of convective heat transfer rate over a radiant floor cooling system.

Cholewa et al. [3] state that the basic parameters in designing of radiant heating/cooling systems and analysing thermal comfort are heat transfer coefficients. It is emphasized that heat transfer coefficients should be determined properly. Thus, experimental studies are carried out using a test room constructed with the dimensions of 1.56 m x 1.56 m x 2.21 m. As a result of the study, it is deduced that obtained heat transfer coefficients are lower than the ones commonly used in the literature and CHTC over floor cooling system is proposed as 0.1 W/m^{2}K.

The CHTCs are summarized in Table 4 that involves various heating and cooling applications in enclosures. The recommended CHTCs are in the range of 2.2 – 3.7 W/m^{2}K, 0.1 – 1.0 W/m^{2}K, 0.0 – 1.2 W/m^{2}K, 2.04 – 4.4 W/m^{2}K, and 2.32 – 3.5 W/m^{2}K for floor heating, floor cooling, ceiling heating, ceiling cooling, and wall heating respectively. It can be clearly seen that CHTCs demonstrate different values even for the same heating/cooling applications, as Shinoda et al. [26] also indicate. This variety of CHTCs may result in higher energy consumption and thermal discomfort.

As a result of the literature survey on natural convection over surfaces and in enclosures, the following deductions can be stated;

- Experimental studies on the determination of heat transfer characteristics of test rooms that are close to real dimensions are insufficient.
- There are still problems in the estimation of heating/cooling loads, which are important in terms of designing, sizing, and thermal comfort of HVAC systems.
- There are significant differences between heat transfer coefficients in scrutinized studies.
- Correlations for determining the heat fluxes that occur over floor cooling systems are almost non-existent.

Table 4 Natural convective heat transfer coefficients with boundary conditions

Reference	Convective heat transfer coefficient (W/m^2K)	Boundary conditions	Applied surface
Olesen et al. [1]	1.0	T_a= 22.6˚C - 30.7˚C T_s= 19.5˚C - 25.8˚C	Floor cooling
Feustel et al. [27]	3.5-6.5	T_a= 21.4˚C - 26.7˚C $T_s \simeq$15˚C	Floor heating
Açıkgöz and Kıncay [28]	2.7	T_a= 24.54˚C – 26.78 ˚C $T_{s,\ heated\ wall}$= 27.14˚C – 29.58˚C $T_{s,\ cooled\ wall}$= 18.85˚C – 22.60˚C	Wall cooling
Miriel et al. [29]	3.0	T_{op}= 23.0˚C – 26.0˚C T_s= 17.0˚C – 19.0˚C	Ceiling cooling
	1.2	T_{op}= 20.0˚C – 24.0˚C T_s= 25.9˚C – 35.7˚C	Ceiling heating
Cholewa et al. [3]	2.2-3.5	T_a= 22.4˚C – 34.1˚C T_s= 25.0˚C – 42.2˚C	Floor heating
	0.1	T_a= 17.3˚C – 21.4˚C T_s= 11.5˚C – 17.4˚C	Floor cooling
Cholewa et al. [4]	0.0	T_a= 19.55˚C – 23.95˚C T_s= 20.87˚C – 31.86˚C	Ceiling heating
	2.04-3.65	T_a= 18.40˚C – 21.00˚C T_s= 16.74˚C – 20.20˚C	Ceiling cooling
Causone et al. [2]	4.4	T_a= 22.5˚C – 26.5˚C T_s= 18.7˚C – 22.8˚C	Ceiling cooling
	0.3	T_a= 20.3˚C – 22.7˚C T_s= 26.1˚C – 32.8˚C	Ceiling heating
Koca et al. [30]	3.16	T_a= 17.05˚C – 20.34˚C T_s= 22.31˚C – 28.27˚C	Wall heating – Case 1
	2.32	T_a= 19.15˚C – 22.80˚C T_s= 23.35˚C – 29.03˚C	Wall heating – Case 2
	2.65	T_a= 21.59˚C – 26.51˚C T_s= 24.80˚C – 31.84˚C	Wall heating – Case 3
Karakoyun et al. [24]	0.6	T_a= 21.98˚C – 36.05˚C T_s= 12.80˚C – 21.01˚C	Floor cooling
Evren et al [25]	3.7	T_a= 17.5˚C – 30.6˚C T_s= 22.4˚C – 35.7˚C	Floor heating
Açıkgöz et al. [11]	3.3	T_a= 19.25˚C – 29.26˚C T_s= 25.75˚C – 38.25˚C	Floor heating
Açıkgöz [20]	3.5	$T_{s,\ heated\ wall}$= 20.0˚C – 40.0˚C $T_{s,\ cooled\ wall}$= 5.0˚C – 15.0˚C	Wall heating
Andres-Chicote et al. [31]	4.2	T_a= 22.4˚C – 26.3˚C T_s= 15.6˚C – 21.2˚C	Ceiling cooling
Koca and Cetin [32]	0.82	T_a= 21.7˚C – 32.9˚C	Ceiling heating
	2.44	T_a= 20.0˚C – 27.2˚C T_s= 26.4˚C – 41.3˚C	Wall heating

3. FORCED CONVECTION

3.1 Flow in Enclosure

Fluid flow can be classified in two types that are internal flow and external flow. If the fluid flows unboundedly *over* a flat or rounded surface, it can be defined as *external* flow. Contrarily, the flow fully bounded *in* a solid surface such as a pipe or duct is *internal* flow. After the previous classification, the flow within an enclosure such as in a room or in a cavity can be determined as internal flow. However, the flow in an enclosure is not guided as in a pipe or a duct. In addition, the flow shows a variety of behavior according to geometry and size of enclosure, locations and direction of the flow inlet and outlet, flow rates, and either the inside and outside surface boundary conditions to be adiabatic, isothermal, or non-isothermal. Some of the cases about the location and the direction of a flow inlet and outlet in a room are shown in Figure 1. The cases are explained as follows:

(a) the inlet and outlet located on the ceiling, direction of flow is perpendicular to ceiling,
(b) the inlet on the floor and the outlet on the ceiling, direction of flow is perpendicular to indicated surfaces,
(c) the inlet and outlet located on one vertical surface, direction of flow is perpendicular to indicated surface,
(d) the inlet and outlet applied on a wall type diffuser such a fan coil, this system can also be expressed as a closed system,
(e) the inlet on the wall and the outlet on the ceiling, direction of flow is perpendicular to indicated surfaces,
(f) the inlet on the diffuser parallel to vertical surface and outlet flow is in a diagonal way.

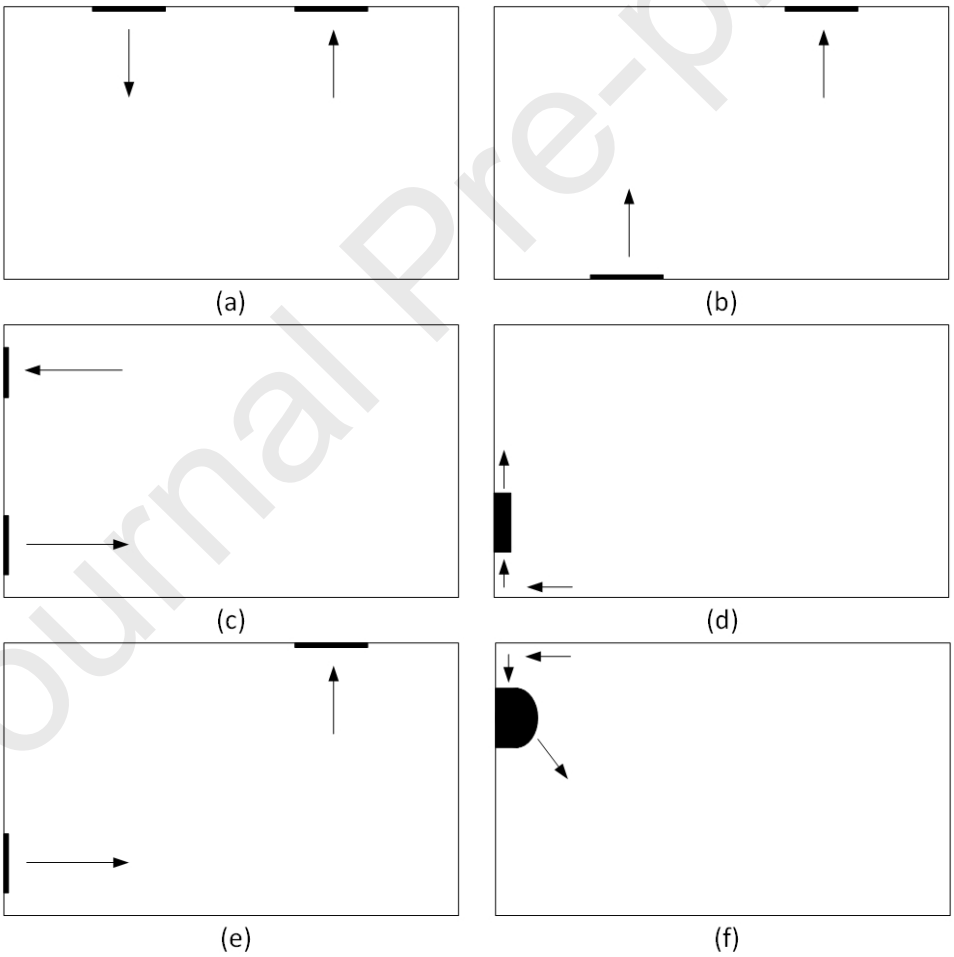


Figure 1. Cases on the thermal unit location and the direction of flow

Some of the terms used in the heat transfer process should be revised while dealing with indoor applications. The scalar magnitude and the direction of velocity can be varied on inner surfaces of the enclosure

(walls, ceiling, and floor), whereas air mass flow rate is constant due to conservation of mass. Therefore, applying an enclosure velocity, given in Eq. 18, based on the mass flow rate can be more convenient [33].

$$U_e = \frac{\dot{V}_e}{\left(\sqrt[3]{V_e}\right)^2} = \frac{\dot{m}/\rho}{{L_e}^2} \tag{18}$$

where:

$$L_e = \sqrt[3]{V_e}$$

The cubic root of the enclosure volume (V_e) is used as an enclosure length (L_e). Based on the jet's mass flow rate, an enclosure Reynold number, to be used in Fisher's CHTC correlation [33], can be expressed in Eq. 19 given below:

$$Re_e = \frac{U_e L_e}{\nu} = \frac{\left(\frac{\dot{m}/\rho}{L_e}\right)}{\nu} \tag{19}$$

Enclosure Archimedes number, to be used in Fisher's CHTC correlation [33], is given in Eq. 20 as follows:

$$Ar_e = \frac{g\beta L_e \Delta T}{{U_e}^2} \tag{20}$$

where the temperature difference in Eq. 20 is shown below:

$$\Delta T = T_{in} - T_{out}$$

3.2. Experimental studies on the forced convection heat transfer coefficient

Fisher in his thesis [33] performs numerous experiments which contain forced convection in an isothermal room and mixed convection in a non-isothermal room within ACH number ranging from 3 to 12. The result of the Nusselt number correlation is transferred to the convection coefficient correlation as given in Table 5 for forced convection case with two diffuser types. The isothermal room experiments are performed at three different air inlet temperatures that are 10°C, 15°C, and 20°C while the inside room surface temperature is maintained at 30°C. The study concludes that surface convection affected by radial ceiling jet is dependent on the volumetric flow rate rather than inlet diffuser velocity. The enclosure Nusselt number is proportional to $Re^{0.8}$ for wall and floor, and $Re^{0.8}/Ar_e$ for the ceiling.

Night purging can be explained as a passive HVAC process which benefits cool night air in order to remove heat from building in the summer season. In addition to the advantage of receiving fresh air, heated building during daytime can be cooled at night without using the significant amount of energy. An experimental study is carried out by Spitler et al. [34] to simulate an office building using a night purging process. To achieve night purging, the experiments have been conducted with a high ventilation rate (15<ACH<100). Before producing correlations for the heat transfer coefficient, the paper focuses on choosing the suitable reference temperature out of the four suggestions. The room outlet temperature is chosen as the reference temperature instead of "bulk air temperature", "air temperature adjacent to a surface", and "local air temperature as a function of height" due to resulting in lowest error in convection calculation. As a and b are empirically derived constants, the convection correlations are formatted as "$h = a + bJ^{0.5}$" for each case, given in Table 5. The dimensionless jet momentum (J) is given in Eq. 21:

$$J = \frac{\dot{m} U_n}{\rho g V} \tag{21}$$

Fisher and Pederson [5] prefer the room inlet temperature as the reference temperature rather than the outlet temperature used by Spitler et al. [34]. Fisher and Pederson claim that Spitler's convection coefficient and his outlet reference temperature give unacceptable uncertainty at low air flow rates.

Fisher and Pederson [5] carry out room ventilation experiments to provide convective coefficient to be used in thermal energy simulation software. The experiments are performed in an iso-thermal room with a radial ceiling jet of temperatures between 10°C and 25°C. In spite of the applied ventilation flow rates with the range of 3-12 ACH, the study claims that the convective correlations in Table 5 can also be applicable up to 100 ACH because their results include the data set of Spitler et al. [34]. Observation of the experiments leads the researcher to choose the enclosure air change rate (ACH) as the correlation parameter for the CHTC. The study also claims that for the cooling case, when the new film correlations are implemented in the energy simulation software, the calculation error of cooling load is maintained around 10%.

Novoselac et al. [35] conduct an experimental study in order to investigate displacement ventilation in a full-scale office room. Although correlated forced convection for the floor surface is given in Table 5, the study mainly focuses on the combined correlations for the floor which are both affected by natural and forced convection. The experimental results show that most of the convection heat transfer occurs at the floor surface due to displacement diffusers located at the floor level. As Fisher and Pederson [5] state that $ACH^{0.8}$ is the best fitting parameter for forced convection at all surfaces, Novoselac et al. [35] use the same parameter with a constant value of 0.48 in their forced correlation which is compatible with experimental values within the ACH range of 3.5 to 10.

Goldstein and Novoselac [36] study convective correlations in an office-sized room which includes a variety of window geometries and a ceiling diffuser. The ceiling slot diffuser is examined with two distances from the windows which are 10 cm and 23 cm. The experiments are performed in the range between 1.5 and 10 ACH but $\dot{V}/L$ equals volumetric flow rate per unit length of an external wall used instead of air change per hour (ACH). It is stated that correlations are mostly affected by forced convection, and correlations are sensible when supply (room inlet) temperature is used as reference temperature. Correlations from these studies are given in Table 5 for wall, floor, and window under various configurations. The configurations consist of window with/without blind and window occupied by the upside/downside/full of external wall.

The majority of the significant experimental studies in the forced convection literature have been reviewed. The empirical CHTC correlations are given with the required explanation in Table 5. In conclusion, the following comments can be inferred:

- Forced CHTCs are highly sensitive to ventilation rate as expected, but the location and direction of inlet/outlet, diffuser type, and diffuser section area should also be considered as selective indoor ventilation parameters before applying the correlations.
- More experimental studies might be required not only for specific cases but also for some common cases such as wall-mounted, ceiling cassette, and ceiling suspended air conditioners.

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Table 5 Forced convective heat transfer correlations

Reference	Surface	Correlation	Remarks	Reference Temperature
Fisher [33]	Wall	$h = (-0.65 + 0.009Re_e^{0.8})/L$ $h = (-0.36 + 0.0065Re_e^{0.8})/L$	Radial ceiling jet $for\ 600 < Re_e < 20\ x\ 10^3$ Free horizontal jet $for\ 500 < Re_e < 2500$	Room inlet temperature
	Floor	$h = (0.52 + 0.0058Re_e^{0.8})/L$ $h = (2.29 + 0.0084Re_e^{0.8})/L$	Radial ceiling jet $for\ 600 < Re_e < 20\ x\ 10^3$ Free horizontal jet $for\ 500 < Re_e < 2500$	
	Ceiling	$h = (-0.54 + 0.024Re_e^{0.8})/L$ $h = (0.21 + 29.05\frac{Re_e^{0.8}}{Ar_e})/L$	Radial ceiling jet $for\ 600 < Re_e < 20\ x\ 10^3$ Free horizontal jet $for\ 500 < Re_e < 2500$	
Spitler et al. [34]	Wall	$h = 4.2 + 81.3J^{0.5}$ $h = 1.6 + 92.7J^{0.5}$	High ventilation rate (15<ACH<100) Ceiling air inlet $for\ 0.001 < J < 0.03$ Sidewall air inlet $for\ 0.002 < J < 0.011$	Room outlet temperature
	Floor	$h = 3.5 + 46.8J^{0.5}$ $h = 3.2 + 44.0J^{0.5}$	High ventilation rate (15<ACH<100) Ceiling air inlet $for\ 0.001 < J < 0.03$ Sidewall air inlet $for\ 0.002 < J < 0.011\ and\ Ar < 0.3$	
	Ceiling	$h = 11.4 + 209.7J^{0.5}$ $h = 0.6 + 59.4J^{0.5}$	High ventilation rate (10<ACH<110) Ceiling air inlet $for\ 0.001 < J < 0.03$ Sidewall air inlet $for\ 0.002 < J < 0.011\ and\ Ar < 0.3$	
Novoselac et al. [35]	Floor	$h = 0.48ACH^{0.8}$	Displacement ventilation 2.5<ACH<10, $Gr/Re^2 < 1$	Room inlet temperature
Goldstein and Novoselac [36]	Wall	$h = 0.063(\dot{V}/L)^{0.8}$	The external wall *below* the window	Room inlet temperature
	Wall	$h = 0.093(\dot{V}/L)^{0.8}$	The external wall *above* the window	
	Floor	$h = 0.048(\dot{V}/L)^{0.8}$	All window configuration	
	Half window	$h = 0.117(\dot{V}/L)^{0.8}$	The window at the *upside* of the external wall, no blinds	
	Half window	$h = 0.093(\dot{V}/L)^{0.8}$	The window at the *downside* of the external wall, *no blinds*	
	Half window	$h = 0.083(\dot{V}/L)^{0.8}$	The window at the downside of the external wall, *with blinds*	
	Full window	$h = 0.103(\dot{V}/L)^{0.8}$	No blinds	
	Full window	$h = 0.063(\dot{V}/L)^{0.8}$	With blinds	
Fisher and Pedersen [5]	Wall	$h = 0.19ACH^{0.8}$	Radial ceiling jet Ventilation rate (3<ACH<100)	Room inlet temperature
	Floor	$h = 0.13ACH^{0.8}$		
	Ceiling	$h = 0.49ACH^{0.8}$		

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4. MIXED CONVECTION

4.1. Ventilation Types

Indoor ventilation is mainly used for improving air quality and air conditioning purposes such as heating, cooling, dehumidification, and supplying minimum fresh air.

In residential buildings, doors and windows are commonly used for natural ventilation. However, mechanical ventilation is preferred instead of natural type due to difficulties in controlling the temperature, humidity, and air exchange rate.

Mixing or displacement systems are preferred for indoor mechanical ventilation thanks to their advantages. Although there are many differences between both systems, the air exchange rate can be noted as the most distinguishing feature. The mixing ventilation system has a relatively high air exchange rate which leads air and airborne particles uniformly scatter throughout the room, whereas, low air exchange rate is used in a displacement ventilation system with suitable air inlet/outlet location as shown in Figure 2.

From left side of Figure 3, it can be seen that cold supply air rises because of the buoyancy effect as air temperature increases. This slow air rise results in a vertical temperature gradient and a non-uniform air distribution which can be seen as advantages for achieving purposes that are improving indoor air quality by exhausting contaminated air from the room and energy-saving by creating air conditioned partial comfort zones especially in terminals, hangars, conference halls, etc.

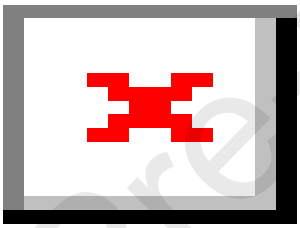


Figure 2. Displacement and mixing flows in an HVAC zone
[from Behne [37] with permission from Elsevier]

4.1.1. Comparison of the displacement and mixing ventilation

As a traditional ventilation system, mixing ventilation (MV) has been used not only for air exchange but also for air conditioning (heating/cooling/dehumidification). Displacement ventilation (DV) has been favored in Scandinavian countries for decades but it is now being globally popular because of its advantages of energy efficiency and indoor air quality. As it can be seen from Figure 3, although mixing ventilation spreads heat, smoke, odor, respiratory droplets which may carry viruses such as influenza, SARS, and COVID-19, displacement ventilation decontaminates them by means of low air velocity and vertical air movement. DV has been advised to be applied in sports centers, hospitals, and public buildings [38–40]. Both ventilation types are compared according to some HVAC criteria and presented in Table 6. The response time can be defined as the time period between setting the demanded temperature and reaching that temperature at the reference point. As expected, the response time is shorter at the mixing ventilation because of higher ventilation rate.

Although DV has many advantages, low thermal capacity is the crucial disadvantage of DV, seen in Table 7 - 8. Low air exchange rates of DV do not allow reaching the required room temperatures in buildings which are under the effect of high thermal loads. For that reason, DV is not preferred to be used as stand-alone, and requires an auxiliary heating/cooling system such as a cooled ceiling or a heated floor. While a room is heated/cooled by both forced air and radiant surface, the heat convection process in the room is classified as mixed convection. The widespread indoor mixed convection cooling case involves cooled ceiling along with displacement ventilation, commonly abbreviated as CC/DV, while radiant cooling applications bring about condensation risk on the cooled surface. Due to this handicap, radiant cooling applications usage are limited in highly arid regions. Using DV with radiant cooling makes perfect integration in the HVAC industry and tackles problems such as condensation, high

thermal load, and slow response time associated with radiant cooling. For these reasons, particularly CC/DV applications are taking more attention day by day in the research area and they have already come into use in terminal buildings at airports, railways, and bus stations.

Another benefit of DV applications is energy efficiency. As it is indicated before, in mixing ventilation applications airflow moves across the HVAC zone, therefore, the whole HVAC zone remains at the same temperature. Contrary to MVs, DV applications do not have to heat/cool the entire HVAC zone but only the occupied zone. As can be seen from Figure 3, the heat, emitted by thermal sources such as lightning, people, machines, creates thermal plumes caused by the buoyancy effect and upward ventilation direction. The heat is accumulated in the vicinity of the ceiling of the room. The accumulated heat discharges through the exhaust grill without affecting the occupied zone of the room. Therefore, most of the cooling thermal load resulting from heat sources and the roof do not have to be met by HVAC cooling power, except for radiation heat transfer. This energy-saving process allows the cooling power to be used mostly at the targeting volume (the occupied zone) instead of the whole HVAC zone. In addition to energy-saving potential, the vertical temperature gradient feature of DV also allows setting a desired temperature value at an intended vertical direction such as ankle level and head level of seated and standing occupants, respectively. This practice is generally used at high ceiling HVAC zones such as workshops, terminals, malls for energy saving. For these reasons, displacement ventilation (DV) is commonly recognized as an energy-efficient application [41–47].

Table 6. Comparison of displacement and mixing ventilation

Features	**Mixing Vent.**	**Displacement Vent.**
Temperature distribution	Uniform	Vertical gradient
Air distribution	Mixing (Homogeneous)	Displacement (Vertical movement)
Air exchange rate	High	Low
Energy efficiency [48–51]	Low	High
Response time [52]	Fast	Slow
Initial investment cost [37]	Low	High
Contamination [37,50,53]	High	Low
Indoor air quality (IAQ) [37,49,50,54–56]	Low	High
Thermal load capacity [37]	High	Low
Thermal Comfort [49,50,57]	Low	High

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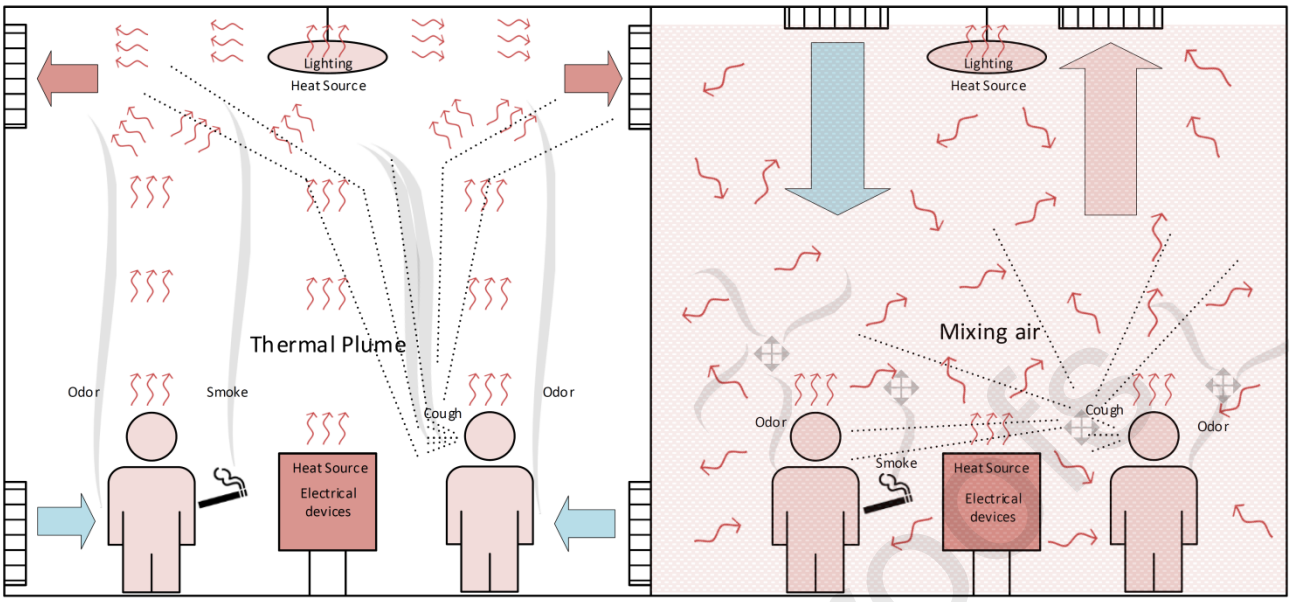


Figure 3. Visual comparison of displacement (at the left) and mixing ventilation (at the right) for room cooling

Table 7 Advantages and disadvantages of mixing ventilation systems

Advantages	**Disadvantages**
Uniform temperature distribution	Contamination (spreading biological and chemical particles, odor, smell, smoke, etc. all around the closed space by mixing air)
High thermal load capacity	
Fast responding time in control of climatization	

Table 8 Advantages and disadvantages of displacement ventilation systems

Advantages	**Disadvantages**
Improved indoor air quality	Low thermal load
Energy efficient	
Reduction of intrusive sound levels	
High thermal comfort	

4.2. Evaluation of mixed convection heat transfer coefficient

Nowadays, building thermal analyses are mainly carried out by the help of thermal analysis programs instead of doing experiments due to saving time and cost. In order to predict thermal loads, these programs need analytical equations for conduction and radiation, and empirical correlations for convection. The key element of the simulation analysis is assigning a best-fitting empirical correlation to approximate the output as much as possible. Related studies in the literature have suggested that flow, governed by both buoyant and mechanical forces, is defined as "mixed flow". Mixed flow should not be solved by using individual natural or forced convection correlations. Therefore, these cases consisting of mixed flow can be solved by their specific correlations identified as "mixed convection correlations" (also named "blended" or "combined" convection correlations in the literature). It is a well-known knowledge in the literature that there are three main heat transfer mechanisms in enclosures as natural, forced, and mixed convections. A forced convection case can have both mechanical and buoyant forces due to temperature difference, but the dominant force is mechanical one, and buoyant force can be negligible because of the overwhelming role of the forced convection. On the other hand, in mixed convection, buoyant force is not negligible for the case of surface heating/cooling applications, and mechanical force is not dominant because of the low ACH value which is preferred as an advantage of displacement ventilation. In this

section, mixed convection correlations are comprehensively evaluated. In addition, all known empirically validated correlations for mixed CHTC are tabulated in detail.

Thermal and fluid analyses play a significant role in designing mechanical systems for buildings. In a thermal building analyzing software, it is expected that the result of computer simulation should be close to the real conditions as much as possible. In order to get close to real cases, defining accurate thermal boundary conditions as heat transfer coefficients is one of the most important factors in the calculation process.

Although calculations of conductive and radiative heat transfer in numerical models are well known and accurately modeled, the convection part of the total heat transfer phenomena could be the weakest link of the building heat transfer numerical models. The difficulties in calculating convective heat transfer are mostly caused by determining accurate CHTCs which are related to the complexity of fluid behavior in the enclosed space. To tackle these difficulties, various fit-for-purpose CHTC correlations have been developed empirically in the literature. Although there are several review studies in the literature on the convective heat transfer, this work specifically focuses on experimentally validated studies at building applications.

Beausoleil-Morrison [58] perform an adaptive simulation study on CHTCs at internal building surfaces. In his study, an adaptive convective algorithm is developed with a basis of 28 CHTCs to improve the modeling of building internal surface convection. In addition, the study classified the convective regimes according to their driving force and cause of driving force. The simplified classification is given in Table 9 [58]. In Table 9, the main two convection types (natural and forced) are classified into the five flow regimes from A to E. Regime A represents a case in which there is no high heat source such as a radiator, stove and only natural convection occurs on the heated or cooled surface. Therefore, it can be noticed that the convection heat transfer in Regime A is expected to be relatively lower than that in Regime B which has thermal sources with high-temperature surfaces. These studies [2–4, 8, 9, 11, 12, 31, 59] can be given as examples to Regime A defined in Table 9. Similarly, a study [17] carried on the natural convection affected by relatively higher buoyant force can be given as examples to Regime B. Regime C and Regime D representing forced convection affected by a mechanical force. A room heated/cooled by an air supply is classified as Regime C [34,36]. Similar to Regime C, a room heated/cooled with no air supply or discharges but cooling/heating via circulation fan, diffuser, fan coil, etc. are classified as Regime D [17,60]. Regime E represents a mixed flow regime affected by both buoyant and mechanical forces. Lastly, a thermal application in which a room heated/cooled by air supply and low radiant panel can be identified as Regime E. Studies [6,33,68,35,61–67] in the literature can be given as examples to Regime E defined in Table 9.

Table 9 Convective regimes at the building internal surface [58]

Regime	Convection type	Cause of driving force
A	Natural flow (Buoyant force)	The temperature difference between surface and air, caused by heated or cooled surface (by solar ray, hydronic panels, external environment or envelope)
B	Natural flow (Buoyant force)	Thermal source (a heating device such as a radiator, stove) in the room
C	Forced flow (mechanical force)	HVAC system (supplying heated or cooled air to room)
D	Forced flow (mechanical force)	Heating or cooling devices with a circulation fan (no air supply or discharge)
E	Mixed flow (Buoyant and mechanical forced)	Both A and C regimes applying simultaneously (HVAC, and heating or cooled surfaces such as floor, ceiling, and walls)

4.3. Experimental studies on the mixed convection heat transfer coefficient

In another study carried out by Beausoleil-Morrison [58], a new approach is developed for calculating mixed CHTC which is built on Alamdari and Hammond's [9] natural convection study and Fisher's [33] forced convection study. It is noted that the new calculation model is applicable for rooms ventilated by a ceiling diffuser and the convection phenomenon is governed by both buoyant and mechanical forces. The developed mixed CHTC

correlations are given in Table 10 [64]. The developed algorithm dynamically controls the convection model by selecting appropriate equations. Users are asked to determine appropriate equations depending on these questions as follows: "Which flow regime from A to E matches to real condition?", "where is the diffuser located?", "where is the heating device located?", "what causes the temperature difference occurred?". According to the responses to the above mentioned questions, the user continues with the selection of either "floor heating" or "heated wall panel".

Awbi and Hatton [68] carry out experimental and numerical studies on prediction of CHTCs. They use a variable-speed fan-fed nozzle for air-jet adjustment on a heated surface. A small sized office room is used in the experiments, and different fan positions are run in the center of the wall, ceiling, and floor. The developed empirical correlations pertaining to mixed CHTCs for wall, floor and ceiling are given in Table 11.

Novoselac et al. [35] report that DV is a relatively new system and the traditional CHTCs for floor surfaces do not provide adequate accuracy to simulate real DV heat transfer cases in building thermal analyses simulation programs. They conclude that convection correlations for the floor are more accurate when they are expressed as a function of volume flow rate as well as the temperature difference between the surface and local air (ΔT). Therefore, they express their new CHTC correlations for floor surfaces as a function ACH. According to the experimental data, the existing CHTCs for vertical surfaces (walls) and ceiling with DV diffuser can be successfully applied. The new and existing CHTCs are given in Table 11.

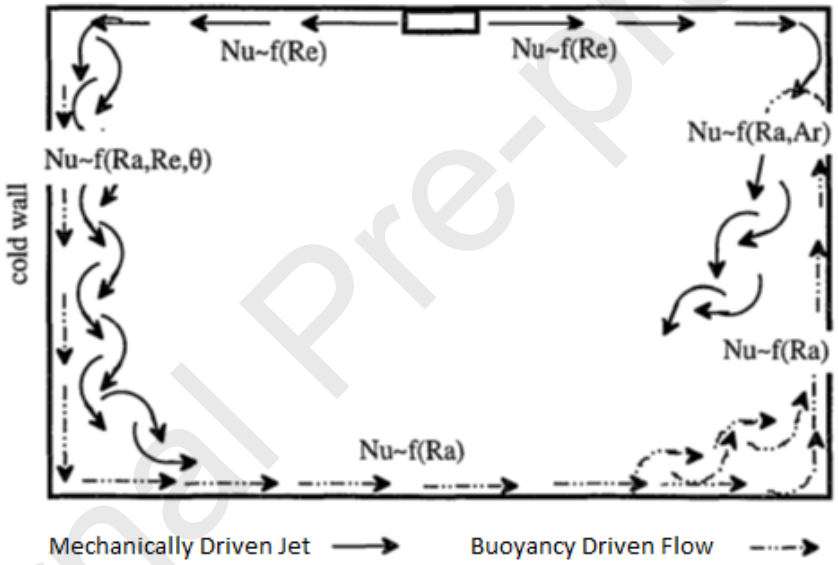


Figure 4. Mechanically and buoyant driven flows in a room cooled by a radiant wall and a ceiling diffuser [from Fisher's thesis [33] with general permission note by the copyright holder]

As it is mentioned, Fisher's thesis [33] contains forced convection experiments in an isothermal room. However, the study is mainly focused on mixed convection correlations at the inner surfaces of a non-isothermal room. First of all, Fisher determines the flows over the inner surfaces are either mechanically driven, buoyancy-driven, or both with their directions, as shown in Figure 4. It is claimed that the Nusselt number at the ceiling is related to Re number because of the mechanically driven flow caused by the radial ceiling diffuser. At the cold wall, the hot walls, and the floor, the Nusselt numbers are related to f (Ra, Re, θ), f (Ra, Re, Ar), f (Ra, Re, Ar), respectively. The dimensionless temperature (θ) is given two times in two different equations (Eqs.22 - 23) in the study as follows:

$$\theta = \frac{T_{out} - T_{in}}{T_{in}} \tag{22}$$

$$\theta = \frac{T_{out} - T_{in}}{T_{hotwalls} - T_{coldwall}} \quad (23)$$

The Nusselt number correlations are converted to the CHTC correlations form, and given for each surface with the operating range in Table 11. The non-isothermal experiments are performed with six surfaces which consist of one cold and five hot surfaces. The cold wall temperature is ranged between 5°C and 20°C while air inlet temperatures of 10°C, 15°C, and 20 °C are applied with ACHs of 0, 3, 6, 9 and 12. Uncertainties in CHTC of floor, celling and wall are given with two reference temperatures. Uncertainties in CHTC with inlet reference are better than those for the selection of outlet reference regarding with all surfaces and ACH values.

Obyn and van Moeseke [62] investigate the internal surfaces' CHTCs in office buildings. In this study, equations of CHTC in the experimental and numerical literature are categorized based on four expressions of CHTC. The categories are summarized and proposed as follows: as a constant value; as a function of temperature difference; as a function of the air exchange rate (ACH), or type of airflow in the room (laminar or turbulent); and "mixed expressions". Although the category based on the convection mode (natural, mixed, forced) [68] and the CHTC expressions [62] are not corresponding to each other, the expressions consider all convection modes. The various analytical and empirical expressions of CHTC are evaluated in terms of their impact on estimating thermal loads. The assessment is carried out with the test office room modeled in TRNSYS 17. A small sized office room is used as a TRNSYS model office based on the case studies [2,35,68]. The authors conclude that except for heavy thermal structure such as lights, using constant CHTC value can be acceptable in many cases instead of the temperature-dependent expressions.

An experimental study analyses the influence of jet trajectory on CHT in an air-based and radiant cooled room conducted by Dréau et. al. [61]. The study aimed both to investigate the energy effectiveness of the two terminals (radiant wall and active chilled beam) and the convective flow in the room at steady-state and dynamic conditions (with solar radiation in the test room). The radiant activated surfaces are mainly under the effect of natural convection. It is concluded that when the temperature difference is below 3 K, the cooled wall is influenced by natural convection. On the other hand, forced convection is observed when the local Richardson number is lower than 4 for the temperature differences are upper than 3 K. The other surfaces (ceiling, the other walls, and floor) are dominated by forced convection, the existing correlations cannot make good matching with measured convective fluxes (especially on the floor where both inlet flow and the down-draught from the cooled wall). New convection correlations [61] have been developed and given in Table 11. The correlations are developed by means of experiments performed at a specific range of parameters such as Ar_{ratio}= 0-6000 Ks^2/m^6, ACH = 3-15, and T_{inlet}=17-27°C.

$$Ar_{ratio} = \left(T_{out} - T_{in}\right)/\dot{V}^2 \quad (24)$$

Apart from previous studies, Neiswanger et al. [69] investigate mixed CHTC with a different approach. Instead of the commonly used method of the coefficient determination, the authors state that the mixed CHTC consists of natural convection (h_{nc}) and forced convection coefficients (h_{fc}), using an exponent, n of 3.2, and it can be expressed as:

$$h = (h_{nc}^n + h_{fc}^n)^{1/n} \quad (25)$$

If flow has low heat flux from surfaces and high supplied air flow rate, flow is dominated by forced convection and the CHTC is assumed to be $\overline{h} \approx \overline{h}_{fc}$. Conversely, if flow has high heat flux from surfaces and low supplied air flow rate, flow is dominated by natural convection and the CHTC is assumed to be $\overline{h} \approx \overline{h}_{nc}$. Between these extreme cases, overall CHTC can be expressed from a superposition of two convection types, given in Eq. 25. Detailed equations with the other parameters used to calculate mixed CHTC are given in Table 11. It is realized that the reference temperature is not mentioned in the study.

Fernández-Hernández et al. [70] investigate their HVAC system consisting of a rotational diffuser positioned at the center of the radiant floor panel, as shown in Figure 5. The test room is cooled from the radiant floor fed by a heat pump and the rotational diffuser fed by an air handling unit, simultaneously. The study aims to

create correlations based on data obtained from experimental and simulation results. The required main equations to calculate the experimental CHTC correlations are given in this study referring to the previous studies [4,8,35,71]. The energy balance on the radiant floor surface is given as follows:

$$Q_{floor} = Q_c + Q_r \tag{26}$$

Q_{floor} is the conducted heat flux from the floor surface to the circulated water and it is equal to the energy received by the water, as given in Eqs. 26 - 27. The heat fluxes from air and other surfaces to the floor surface are caused by radiation and convection, as given in Eqs. 28 - 29. In the study, the Nusselt number's correlation is given in Eq.30 to calculate CHTC.

$$Q_{floor} = \dot{m}_w Cp_w (T_{w,0} - T_{w,1}) \tag{27}$$

$$Q_c = h_c A_s (T_{air} - T_s) \tag{28}$$

$$Q_r = h_r A_s (T_{sup} - T_{air}) \tag{29}$$

$$Nu = \frac{h_c D_h}{k_a} \tag{30}$$

where $k_a$is the thermal conductivity of air at the supply temperature, and $D_h$is the hydraulic diameter of the diffuser.

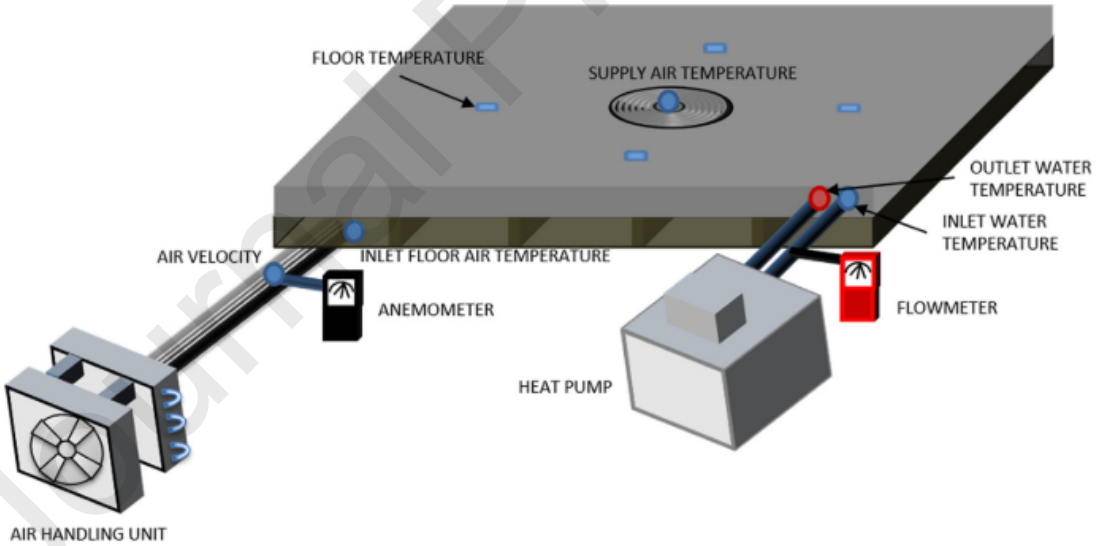


Figure 5. A rotational diffuser coupled with a radiant floor cooling system [from Fernández-Hernández et al. [70] with permission from Elsevier]

One of the latest experimental study on natural and mixed CHTC has been carried out by Venko et al. [72]. In their enclosure, the mixed convection system is established by a longitudinal ceiling slot which supplies air flow in direction of downward parallel to the cooled wall, and two longitudinal ceiling exhaust slots are at the center of celling and above the opposite wall. In their work, local natural and mixed CHTC correlations are developed to use in the calculation of energy consumption in an office room equipped with the thermal activated cooled wall by TRNSYS simulation model. The experiments are performed in a thermal test compartment (L × W × H = 4 m × 3 m × 2.5 m). CHTC values versus distance from diffuser (h-x) are plotted with two air velocity

values (U_{air}=2 m/s and 4 m/s) and two temperature difference values ($\Delta T \approx 4K$ and 9K). The developed correlation with boundary conditions is given in Table 11.

There are few experimental studies on the correlation of mixed convection heat transfer coefficient. The empirical correlations are summed up with the required explanations in Table 11. The mixed convection applications can be considered as one of the latest trend in comparison to natural and forced convection ones. Therefore, the mixed convection applications are comprehensively explained in this work in order to take readers' attention. More energy-saver, healthier, and comfortable HVAC applications require displacement ventilation rather than mixing ventilation. The reason for this trend is tried to be explained with the comparison of these two ventilation systems.

From the investigation, the following conclusions can be stated:

- By mixed convection's nature, making a comparison of the empirical mixed coefficients is almost not possible because every case is unique. Therefore, each study uses its particular parameters to obtain its correlation
- Most of the mixed convection studies investigate mixed convection applications in terms of thermal comfort, energy performance, and air quality. However, more studies on the mixed convection correlations are still needed for the heat transfer calculations, determining system capacity, energy consumption, insulation thickness, etc.
- Current building designing standards for volume dimensioning are specified for the thermal comfort in conventional cases in which building zones are heated, cooled, ventilated by forced convection. Thermal comfort studies have concluded that mixing convection-DV is superior with respect to aforementioned advantages to forced convection from macro zones (terminals, conference, theatre, meeting rooms, libraries, temples, ships, airplane etc.) to micro zones (office rooms). Particularly in small zones under high thermal load, the size of the cooled zone should be considerably large for avoiding draught risk. In a small room cooled by forced convection applications, either the air temperature or the air velocity must be higher to meet the high thermal load. Therefore, the distance between a person and a supply diffuser ought to be increased. However, new building designing standards, which allow smaller sizes, can be specified for buildings with mixed convection-DV applications by increasing number of studies on the issue.

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Table 10 Adaptive correlations of mixed convection heat transfer

Reference	Surface	Correlation	Remarks
Beausoleil-Morrison [64]	Wall	$h = \left[\left[\left(1.5\left(\frac{\Delta T}{H}\right)^{1/4}\right)^6 + \left(1.23\Delta T^{1/3}\right)^6\right]^{3/6} + \left[\left(\frac{T_s - T_{sup}}{\Delta T}\right)\left(-0.199 + 0.19ACH^{0.8}\right)\right]^3\right]^{1/3}$	Assisting forces
	Wall	$h = max\left\{\begin{array}{l}\left[\left[\left(1.5\left(\frac{\Delta T}{H}\right)^{1/4}\right)^6 + \left(1.23\Delta T^{1/3}\right)^6\right]^{3/6} - \left[\left(\frac{T_s - T_{\sup}}{\Delta T}\right)\left(-0.199 + 0.19ACH^{0.8}\right)\right]^3\right]^{1/3} \\ 80\%\left[\left(1.5\left(\frac{\Delta T}{H}\right)^{1/4}\right)^6 + \left(1.23\Delta T^{1/3}\right)^6\right]^{1/6} \\ 80\%\left[\left(\frac{T_s - T_{\sup}}{\Delta T}\right)\left(-0.199 + 0.19ACH^{0.8}\right)\right]\end{array}\right.$	Opposing forces
	Floor	$h = \left[\left[\left(1.4\left(\frac{\Delta T}{D_h}\right)^{1/4}\right)^6 + \left(1.63\Delta T^{1/3}\right)^6\right]^{3/6} + \left[\left(\frac{T_s - T_{sup}}{\Delta T}\right)\left(0.159 + 0.116ACH^{0.8}\right)\right]^3\right]^{1/3}$	Buoyant
	Floor	$h = \left[\left[0.6\left(\frac{\Delta T}{D_h{}^2}\right)^{1/5}\right]^3 + \left[\left(\frac{T_s - T_{sup}}{\Delta T}\right)\left(0.159 + 0.116ACH^{0.8}\right)\right]^3\right]^{1/3}$	Stably stratified
	Ceiling	$h = \left[\left[\left(1.4\left(\frac{\Delta T}{D_h}\right)^{1/4}\right)^6 + \left(1.63\Delta T^{1/3}\right)^6\right]^{3/6} + \left[\left(\frac{T_s - T_{sup}}{\Delta T}\right)\left(-0.166 + 0.484ACH^{0.8}\right)\right]^3\right]^{1/3}$	Buoyant
	Ceiling	$h = \left[\left[0.6\left(\frac{\Delta T}{D_h{}^2}\right)^{1/5}\right]^3 + \left[\left(\frac{T_s - T_{sup}}{\Delta T}\right)\left(-0.166 + 0.484ACH^{0.8}\right)\right]^3\right]^{1/3}$	Stably stratified
Reference temperature: Mean air temperature, ΔT: Surface to the air temperature difference			

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Table 11 Mixed convective heat transfer coefficient correlations

Reference	Surface	Correlation	Remarks	Reference Temperature
Awbi and Hatton [7,68]	Wall	$h = \left[\left(1.823\frac{\Delta T^{0.293}}{{D_h}^{0.121}}\right)^{3.2} + \left(3.79 W^{1.536} {U_n}^{0.873}\right)^{3.2}\right]^{1/3.2}$	Diffuser over a heated wall	Local air temp.
	Floor	$h = \left[\left(2.175\frac{\Delta T^{0.308}}{{D_h}^{0.076}}\right)^{3.2} + \left(4.248 W^{0.575} {U_n}^{0.557}\right)^{3.2}\right]^{1/3.2}$	Diffuser over a heated floor	
	Ceiling	$h = \left[\left(0.704\frac{\Delta T^{0.133}}{{D_h}^{0.601}}\right)^{3.2} + \left(1.35 W^{0.074} {U_n}^{0.772}\right)^{3.2}\right]^{1/3.2}$	Diffuser over a heated ceiling	
Novoselac [35]	Floor (T_s>T_{air})	$h = \left[\left(2.175\frac{\Delta T^{0.308}}{{D_h}^{0.076}}\right)^{6} + \left(\frac{\lvert T_s - T_{\sup}\rvert}{T_s - T_{air}} 0.48 ACH^{0.8}\right)^{6}\right]^{1/6}$	Floor with displacement ventilation	Local air temp.
	Floor (T_s<T_{air})	$h = \left[\left(0.704\frac{\Delta T^{0.133}}{{D_h}^{0.601}}\right)^{6} + \left(\frac{\lvert T_s - T_{\sup}\rvert}{T_s - T_{air}} 0.48 ACH^{0.8}\right)^{6}\right]^{1/6}$		
	Wall	$h = \left[\left(1.823\frac{\Delta T^{0.293}}{{D_h}^{0.121}}\right)^{3} + \left(1.31 ACH^{0.8}\right)^{3}\right]^{1/3}$	High aspiration diffusers	
	Floor (T_s>T_{air})	$h = \left[\left(2.175\frac{\Delta T^{0.308}}{{D_h}^{0.076}}\right)^{3} + \left(0.96 ACH^{0.8}\right)^{3}\right]^{1/3}$		
	Floor (T_s<T_{air})	$h = \left[\left(0.704\frac{\Delta T^{0.133}}{{D_h}^{0.601}}\right)^{3} + \left(0.96 ACH^{0.8}\right)^{3}\right]^{1/3}$		

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Reference	Surface	Correlation	Remarks	Reference Temperature
	Ceiling ($T_s<T_{air}$)	$h = \left[\left(0.704\frac{\Delta T^{0.133}}{D_h{}^{0.601}}\right)^3 + \left(1.19ACH^{0.8}\right)^3\right]^{1/3}$		
	Ceiling ($T_s>T_{air}$)	$h = \left[\left(2.12\Delta T^{0.33}\right)^3 + \left(1.19ACH^{0.8}\right)^3\right]^{1/3}$		

Table 11 Mixed convective heat transfer coefficient correlations (Cont.)

Reference	Surface	Correlation	Remarks	Reference Temperature
Fisher [33]	Cold Wall	$h = (-1.1 + 0.0084\theta^{0.15}(Re_e Ra_e)^{0.25})/L$	$for\ 580 < Re_e < 2.5\ x\ 10^3$ $3.5\ x\ 10^{10} < Ra_e < 1\ x\ 10^{11}$ $4.2\ x\ 10^3 < Ar_e < 1\ x\ 10^5$ $5\ x\ 10^{-2} < \theta < 0.9$ $T_{cold\ wall} < T_{inlet\ air} < T_{hot\ wall}$	Average room inlet and outlet temp.
	Opposing Hot Wall	$h = (0.34 + 0.16(\frac{Ra_e}{Ar_e})^{0.25})/L$		
	End Walls	$h = (-0.58 + 0.47Re_e{}^{-0.15}(\frac{Ra_e}{Ar_e})^{0.25})/L$		
	Floor	$h = (-8.6 + Ar_e{}^{-0.15}(\frac{Ra_e}{Re_e})^{0.25})/L$		
	Ceiling	$h = (-1.2 + 0.042Re_e^{0.8})/L$		
Dreau [61]	Ceiling	$h = \left(-1.5\times10^{-5}\frac{T_{\text{outlet}} - T_{\text{inlet}}}{\dot{V}^2} + 0.19\right)\text{ACH}^{0.8}$	Cooling conditions with the inlet located in the middle of the ceiling.	Inlet temp.
	Walls	$h = 0.15\text{ACH}^{0.8}$		
	Floor	$h = \left(4.1\times10^{-6}\frac{T_{\text{outlet}} - T_{\text{inlet}}}{\dot{V}^2} + 0.14\right)\text{ACH}^{0.8}$		
Neiswanger et al. [69]	Wall (heated)	$h_{nc} = \left(\frac{5k}{4x_m}\right)[F(Pr)Ra^*_{xm}]^{0.2}$	Natural convection (High heat flux, low flow rate) x_m: overall height of the heated wall	-

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		where $F(\mathrm{Pr}) = Pr/(4 + 9Pr^{1/2} + 10Pr)$		
		$h_{fc} = 0.5\left(\frac{k}{z_m}\right)Pr^{0.6}Re_{zm}^{0.6}$	Forced convection (Low heat flux, high flow rate) z_m: overall length of the heated wall	
		$h = (h_{nc}^{3.2} + h_{fc}^{3.2})^{1/3.2}$	Mixed convection	
		$\Gamma = \frac{h_{nc}}{h_{fc}} = \frac{0.4(x_m/z_m)Pr^{0.6}Re_{zm}^{0.6}}{[F(Pr)Ra_{xm}^*]^{0.2}}$	$\Gamma < 0.57$: nc dominated regime $\Gamma > 1.74$: fc dominated regime $0.57 \le \Gamma \le 1.74$ mixed convection dominated regime	

Table 11 Mixed convective heat transfer coefficient correlations (Cont.)

Reference	**Surface**	**Correlation**	**Remarks**	**Reference Temperature**
Fernández-Hernández et al. [70]	Floor	For $Re \le 16000$ $h = \left(-41.567 + 4.750Re^{0.3}{\theta_1}^{-0.02}{\theta_2}^{-0.02}\right)^{k_a}/_{D}$ or $h \cong \left(-41.567 + 4.750Re^{0.3}\right)^{k_a}/_{D}$ For $Re \ge 16000$ $h = (17.168 + 4.750Re^{1.26}{\theta_1}^{-0.32}{\theta_2}^{0.46})^{k_a}/_{D_h}$	a rotational diffuser coupled with a radiant floor cooling Re: Reynold number of the diffuser $\theta_1 = \frac{T_{air} - T_{sup}}{T_{air} - T_s}$ $\theta_2 = \frac{T_{sup} - 273K}{T_{air} - T_s}$	-
Venko et al. [72]	Wall (cooled)	$h(x) = (35.404)\sqrt{U_{air}}/(\Delta T^{1.630}x^{0.308}) + 2.704\sqrt{U_{air}}/x^{0.614}$ where $\Delta T = T_{supply_air} - T_{cooled\ wall}$	longitudinal ceiling slot with width w = 10 mm is positioned on the top of TACW; • H_{TACW} = 2.5 m; • 0 m < x < 2.5 m; • 1.0 m/s ≤ U_{air} ≤ 4.0 m/s; • −10 K ≤ ΔT_{air} ≤ −2 K; • reference temp. in the room center at height 1.1 m: 21.5 K ≤ T_f ≤ 27.5 K	Room center

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5. COMPARISON OF THE CONVECTION TYPES IN INDOORS

In Table 12, commonly used HVAC applications with the convection types (natural, mixed, and forced convection) are compared to each other with respect to a number of aspects considering for the same thermal condition. The second column represents natural convection occurring in the room by heated/cooled radiant surfaces as also explained and classified as *Regime A* in Table 9. At the third column, natural convection created by a radiator classified in *Regime B,* and both of natural convection cases are assumed that there is no ventilation system. These systems are mostly used in domestic purposes or small apartments. Because there is no ventilation system, the air circulation cannot be controlled properly but it can be adjusted by the occupants. Instantaneous wind speed and its direction and behavior of occupants affects ACH in Regime A and B. For these reasons, air quality of natural convection applications is determined as low or medium compared to others. In this applications, windows and doors are not made airtight or an air duct is drilled due to satisfy vital oxygen requirement and ensure moisture balance. Therefore, cold/hot air uncontrollably passes through the room, and leads to degradation of energy efficiency, as well. The other drawbacks of natural convection applications are relative humidity to increase to undesired levels as well as condensation risk over cooled surfaces. Since there is no ventilation system in the natural convection cooling applications, relatively hot air leaves its humid as condensed water on the cooled surfaces. Therefore, such applications are not practical in living environments as seen in Table 12. These disadvantages lead users to utilize the mixed convection applications at indoor cooling processes.

Low temperature heating and high temperature cooling radiant systems are generally classified as energy saver systems because of their ability to be supplied by low-temperature heat sources such as heat pumps, solar collectors, and geothermal wells. However, in Table 12, they are assessed relatively medium when compared to the mixed convection system because of the energy loss caused by uncontrolled air exchange through the windows opening. Although noise resulted from HVAC application is one of the main issues, it is mostly ignored in the workplace but not in houses, particularly night time. This could be the main reason why natural convection heating applications are still widely preferred at houses but the workplaces.

Thermal comfort for heating and cooling cases are assessed as high in Table 12 for 'Mixed convection – DV'. However, in some cases, the vertical air temperature difference between head and ankles could reach up to 5 K [57] with respect to ACH values. In that case, the thermal comfort could be assessed as medium because thermal comfort standards in EN ISO 7730 [73] and ASHRAE 55 [74] recommend the temperature differences lower than 3 K.

The initial investment cost of each system is compared in Table 12. Mixed convection-DV is determined to have higher investment cost requirements than the forced and natural convection applications because mixed convection applications need components and infrastructure of both natural and forced convection applications. However, particularly in large volumes such as terminals, malls, etc., both investment and operating cost can be lowered by diminishing the capacity of the air conditioner and the size of ventilation equipment because some of the thermal requirement is satisfied by large radiant surfaces. The remained thermal load can be reduced by targeted volume to be air conditioned instead of the whole volume, more details and wide explanation are given in section 4.1.1.

Table 12 demonstrates the comparison of aforementioned convection types with respect to indicated aspects. DV is used in the mixed convection and MV is encountered in the forced convection applications.

Table 12 Comparison of the commonly used HVAC applications with the convection types

Assessment subject	**Natural convection - radiant surface**	**Natural convection - radiator**	**Mixed convection - DV**	**Forced convection - MV**
Load supply capacity	Low	Medium	Medium	High
Thermal comfort - air quality	Low	Low	High	Medium
Thermal comfort – heating case	Medium	Medium	High	Medium
Thermal comfort – cooling case	N/A	N/A	High	Medium
Contamination	Medium	Medium	Low	High
Noise	Low	Low	Medium	High
Response time	Slow	Medium	Medium	Fast
Energy saving	Medium	Low	High	Low
Initial investment cost	Medium	Low	High	Medium

Table 13 CHTCs with respect to convection types

Convection types	Conditions	Reference source	Applied surface		h (CHTC)		
Natural	Floor heating	Cholewa et al. [3]	Floor		2.2–3.5		
		Min et al. [10]			3.0–4.8		
		Awbi and Hatton [8]			2.7–4.3		
		ASHRAE [18]			3.0–4.8		
		Acikgoz et al. [11]			2.3–4.1		
	Floor cooling	Min et al. [10]			~0.2		
		Awbi and Hatton [8]			~0.5		
		Karakoyun et al. [24]			~0.6		
Mixed	Fan (internal air movement in test chamber)	Awbi and Hatton [68]	Heated wall		4.0-4.5		
			Heated ceiling		1.0-2.0		
			Heated floor		4.6-5.0		
	Radial ceiling diffuser (3-12 ACH), cooling inlet air 15°C, cold wall 5°C, five hot surfaces 30°C	Fisher [33]	Ceiling		1.3-4.9		
			Hot wall		1.2-2.4		
			Floor		1.8-3.7		
	Ceiling diffuser and radiant cooled wall, 3.6 ACH, CHT is dominated by natural convection	Dreau [61]	Wall		2.5-3		
	Longitudinal ceiling slot cooling, local CHTC at x= 1 m (x: distance from ceiling) air speed 1-4 m/s	Venko et al. [72]	Cooled wall		5.0-12.5		
Forced	Radial ceiling diffuser (3-12 ACH) – cooling (15°C inlet air)	Fisher [33]	Ceiling		1.0-4.0		
			Wall		0.3-1.2		
			Floor		0.4-1.0		
	Free Horizontal Jet (3-12 ACH) – cooling (15°C inlet air)	Fisher [33]	Ceiling		0.1-0.6		
			Wall		0.3-1.2		
			Floor		1.1-2.0		
	Night purging	Spitler [34]	Surface	Air inlet	ASHRAE	15 ACH	100 ACH
			Ceiling	Ceiling	0.95	16.8	47.4
			Vertical walls	Ceiling	3.08	6.3	18.1
			Floor	Ceiling	4.04	4.7	11.5
			Ceiling	Side wall	0.95	1.4	5.8
			Vertical walls	Side wall	3.08	2.8	9.7
			Floor	Side wall	4.04	3.8	7.0
	Ceiling slot diffusers surface-room ΔT:1-4°C	Goldstein and Novoselac [36]	Window		3.5-11		
			Floor		3.5-4.5		

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Several examples of CHTCs from the literature are given in Table 13. These CHTCs can be used for comparison purpose if the case is similar to the referenced studies. Some of the mixed and forced convection CHTCs could be seen to be very close to those of the natural convection heating cases.

In Table 13, at the first part, the calculated natural CHTCs using various correlations for the similar thermal condition within the temperature difference range of 2°C to 14°C are given. CHTC values of Acikgoz's [11] experimental study is slightly lower than those values of Min et al. [10] and Awbi and Hatton [8]. These differences may arise from different experimental conditions such as geometries of test chambers, unavoidable experimental errors, and uncertainties.

CHTCs obtained under forced and mixed convection conditions are taken from the literature, and presented in Table 13. There are limited number of comparisons among forced CHTC studies which share the similar operating conditions such as Fisher [33] and Spitler [34]. Because comparison of CHTCs for forced convection cases is more difficult than natural convection due to increasing complexity of phenomenon and number of parameters. For forced convection, ACH or air speed at inlet, applied surface, and air inlet flow direction could be the main parameters which should be considered before any comparison. Similar difficulties are also valid for mixed convection with extra considerations. In mixed convection, thermally activated surfaces in an enclosure are also considerable issues along with forced convection parameters.

6. CONCLUSION

In this study, heat convection types (natural, forced, and mixed) are explained with their cause of driving forces in the building applications. A great number of correlations on three types of CHTCs in the literature are given in this study. After analysis of well-known convection types (natural, forced), highly detailed explanations for mixed convection heat transfer applications are provided with many reasons that why the usage of mixed convection applications have become widespread recently. Definitions, explanations, and assessments corresponding to displacement ventilation and its conventional alternative mixing ventilation are mentioned. Finally, indoor applications of the heat convection types are compared with each other. Based on related studies in the literature, at first, displacement and mixing ventilation types are assessed, then the four commonly used convection heat transfer applications are contrasted.

After investigating the majority of the significant experimental study in the literature, the following comments can be deduced:

a) For the natural convection, there are noteworthy differences among the heat transfer coefficients. Correlations on radiant floor cooling applications are nearly non-existent.
b) As it is expected, forced convection heat transfer correlations are highly responsive to air exchange rate but positions of inlet/outlet, diffuser type should also be considered as selective indoor ventilation parameters before applying the given correlations. More experimental studies might be required not only for specific cases but also for some general cases such as split air conditioner's model of wall-mounted, ceiling-cassette, ceiling-suspended, window, and floor standing.
c) Mixed convection can also be identified as combination of natural and forced convections. That makes the mixed convection more complex in terms of parameters and applied cases. Each case study has a correlation which can only be used for examined case. For natural convection studies, cooled ceiling and a heated floor are general cases and many correlations can be easily compared. For forced convection studies, cases could be more various according to inlet and exit diffuser types (radiant/slot, grill, round, and half round), diffuser location, and jet direction (parallel or perpendicular to ceiling, wall, and floor). Mixed convection studies have combination of these cases and contains more variables. The complexity of mixed convection requires more cases to be carried out. Therefore, unlike natural convection, mixed convection correlations are not comparable until the literature provides coefficient studies that share similar cases.
d) Most of the mixed convection studies focus on thermal comfort, energy performance, and air quality. However, more computational and experimental investigations on the mixed CHTC correlations are still required for detailed building thermal load calculations, selection of appropriate equipment capacities, building energy performance classifications, and insulation thickness.
e) Building zone dimension standards are determined according to thermal comfort studies which contain conventional forced convection applications. Thus, more mixed convection studies on

thermal comfort and heat transfer characteristics should be conducted for living environments air conditioned by mixed convection system to make standards more up to date and compatible with those systems.

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NOMENCLATURE

A	Area (m^2)
ACH	Air changes per hour (h^{-1})
CC	Cooled ceiling
CHT	Convective heat transfer
CHTC	Convective heat transfer coefficient
DV	Displacement ventilation
MV	Mixing ventilation
D_h	Hydraulic diameter of room surface (4 × area over wetted perimeter) (m)
H	Height of vertical surface or length of horizontal surface (m)
h, h_c	Convective heat transfer coefficient (CHTC) ($W\ m^{-2}\ K^{-1}$)
Q	Heat transfer (W)
T	Temperature (K, °C)
V	Volume (m^3)
$\dot{V}$	Volumetric flow rates ($m^3\ s^{-1}$)
$\dot{m}$	Mass flow rates ($kg\ s^{-1}$)
U	Velocity ($m\ s^{-1}$)
W	Width of nozzle opening (m)
w	Water
ν	Kinematic viscosity of the fluid ($m^2\ s^{-1}$)
θ	Dimensionless temperature
g	Gravitational acceleration
L	Length (m)
Δ	Difference
Ar	Archimedes number
Re	Reynolds number
Pr	Prandtl
k	Fluid thermal conductivity ($W\ m^{-1}\ K^{-1}$)
x	Distance from diffuser

Subscripts

e	enclosure
c	convective
f	reference
hs	heat sink
in	inlet
n	nozzle
op	operative
out	outlet
r	radiative
s	surface
sup	supply
w	water
fc	forced convection
nc	natural convection

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