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STUDY OF AIR CURTAINS USED TO RESTRICT INFILTRATION INTO REFRIGERATED ROOMS

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ABSTRACT

This study has two goals considering the use of air curtains to restrict infiltration of air into refrigerated rooms. First, the optimal parameters of the air curtain device are determined, including the jet velocity and the jet nozzle width. Next, an expression to estimate the heat transfer rate through the air curtain is proposed. A computational fluid dynamics (CFD) model provides numerical results. This model and its boundary conditions are validated through experimental measurements and by comparison with results from scientific literature.

INTRODUCTION

One of the major sources of heat gain in refrigerated storage rooms is the infiltration of warm ambient air through doorways. Air infiltration can also be a source of ice or mist forming. Commonly used methods of reducing infiltration are PVC strip curtains and fast- sliding doors. Sometimes a vestibule or air lock is used, often in combination with these curtains or doors. Another possible solution, which allows an easier passage of traffic, is the use of an air curtain.

An air curtain or ACD (air curtain device) consists of one or several fans that blow a planar jet of air across the opening. The jet disturbs the free air movement through the doorway, caused by natural convection. In this way, the air curtain is able to reduce the amount of mass and heat transported through the doorway. This principle is called aerodynamic sealing. Besides cold stores, air curtains are also used for heating applications, such as heated buildings and furnaces, or to prevent the circulation of smoke, dust and odours.

There is a wide variety of commercially available air curtain devices. Vertically downwards blowing air curtains are the most common , especially in the case of cold stores, but also upwards blowing and horizontal jets are used. Some ACD's are provided with a system for the recirculation of the blown air, but mostly the air is simply drawn from the indoor or outdoor surroundings. There may be some provision for the heating or even cooling of the air to increase comfort.

NOMENCLATURE

A	[m²]	Area of the doorway
b_0	[m]	Jet nozzle width
c_p	[J/kgK]	Specific heat of air
D_m	[-]	Deflection modulus
g	[m/s²]	Gravitational acceleration
h	[m]	Height of the doorway and the air curtain
I_t	[-]	Turbulence intensity
k	[m²/s²]	Turbulent kinetic energy
L_ε	[m]	Length scale for turbulent dissipation
Nu	[-]	Nusselt number
Δp	[Pa]	Pressure difference across the opening
Δp_a	[Pa]	Auxiliary pressure
Δp_s	[Pa]	Stack pressure
Pr	[-]	Prandtl number
q	[W]	Heat transfer rate through the opening
Q	[m³/s]	Transfer rate of air through the opening
R^2	[-]	Coefficient of regress ion
Re	[-]	Reynolds number
T	[K]	Air temperature
ΔT	[K]	Temperature difference between the two areas
v	[m/s]	Jet velocity
w	[m]	Width of the doorway and the air curtain

Greek symbols

α	[°]	Jet angle
ε	[m²/s³]	Turbulent dissipation rate
η	[-]	Effectiveness
λ	[W/mK]	Thermal conductivity of air
μ	[Pa s]	Dynamic viscosity of air
ρ	[kg/m³]	Density of air

Subscripts

0	At jet nozzle
f	At the end of the jet (impingement)
min	Minimum value to ensure stability
ref	Reference value
c	Cold air
w	Warm air

Some air curtains consist of two or three parallel jets. This is very often the case when used to restrict infiltration into refrigerated display cabinets.

In this study a CFD model is validated with experimental measurements. A parametric study performed with this CFD model provides the optimal parameters for the ACD such as the outlet velocity v_0 and the jet nozzle width b_0. The model will also allow to assess the amount of heat that is transferred through the air curtain into the cold room.

Properties of an air jet

The first fundamental research of an air curtain was performed by Hayes & Stoecker [1,2]. They considered the case of a vertically downwards blowing air curtain without recirculation, mounted at the doorway of an airtight room. They showed how the air jet is affected by the pressure difference Δp across the doorway, which exists of two components. The first component is a consequence of the operation of the air curtain itself. Air is taken from one space but is blown into both spaces, which causes a pressure build- up. This pressure, which is called the auxiliary pressure Δp_a, causes a deflection of the jet directed towards the space from which the air is taken. The second component of Δp is due to the temperature difference ΔT across the door. It is called the stack pressure Δp_s and varies linearly from the top to the bottom of the opening. Using expressions for Δp_a and Δp_s, Hayes & Stoecker were able to integrate momentum equations for a control volume at the jet centreline, which provides analytical equations to determine the shape of the air jet.

The stability of the air jet is determined by two effects. The destabilizing factor is the stack pressure caused by the temperature difference. A higher ΔT tends to deflect the jet. The stabilizing factor is the outlet momentum of the jet. A numerical parameter that assesses the stability of the jet, is the *deflection modulus D_m*. The deflection modulus is the ratio of the outlet momentum of the jet (the stabilizing factor) to the temperature difference (the destabilizing factor).

$$D_m = \frac{\rho_0 b_0 v_0^2}{g h^2 (\rho_c - \rho_w)} \quad (1)$$

Hayes & Stoecker provide an expression for the minimal theoretical outlet momentum needed to ensure that the air jet reaches the opposite side of the opening and to avoid so- called breakthrough of the air curtain. The sign of (3) dependents on the use of indoor or outdoor air.

$$D_{m,\min} = \frac{-\sin\alpha_f - \sin\alpha_0 + 2 - 2\sqrt{(1-\sin\alpha_f)(1-\sin\alpha_0)}}{2(\sin\alpha_f - \sin\alpha_0)^2} \quad (2)$$

$$\sin\alpha_f = \pm 2.4\sqrt{\frac{b_0}{h}}\left(1 - 2.56\frac{b_0}{h}\right) \quad (3)$$

When the assumption of an airtight room is not valid, the expression for Δp has to be extended with some extra correction terms, as proposed by Sirén [3,4]. When the air curtain is used at an outdoor opening, an extra term is needed to account for the influence of the wind. Another correction has to be made for the imbalance in ventilation flows. This imbalance can have several causes. First, the mechanical ventilation, used to provide the room with fresh air, could be poorly balanced. Another possibility is the extraction of smoke or dust in industrial buildings. The most important reason however is the leakage of air by natural convection.

Heat transfer through an air curtain

The most important property regarding air curtains for refrigerated rooms is of course the heat transfer rate q through the doorway, which can be seen as the sum of two components. The first term is called the advective heat transfer, which is the thermal energy carried away by the air flow crossing the doorway. The second component is the diffusive heat transfer, including molecular and turbulent diffusion mechanisms.

To express how effective an air curtain is in reducing the heat transfer through a doorway, Ashrae [5] suggests the use of an effectiveness η, defined as follows:

$$q = q_{ref}(1-\eta) \quad (4)$$

The actual heat transfer rate q is compared to a reference value q_{ref}, which is the heat transfer rate through the doorway without an air curtain in operation. η = 1 designates a perfectly isolated door, while η = 0 is the situation of an open door without an air curtain in use. Common values of the effectiveness are 0.85 to 0.95 for PVC strip doors and fast- fold doors, while the values for air curtains range from very low to more than 0.7 [5]. Downing [6] uses a similar definition of the effectiveness of air curtains, but with mass transfer instead of heat transfer. His experimental results for η range from - 1.58 to 0.85. Note that a negative value of η

signifies that the air curtain increases the heat transfer compared to q_{ref}.

EXPERIMENTAL METHOD

A series of experiments was performed on a vertically downwards blowing air curtain in a supermarket. The ACD is installed at a door that separates a refrigerated room from the rest of the store. The device draws its air from the warm side of the store. The width of the door and jet is 2 m. The size of the jet nozzle b_0 is 93 mm and the jet is blown straight down. Some other geometrical properties are given in figures 1 and 2.

The air curtain can be set to 4 different levels. The corresponding outlet velocities were measured using a hot-wire anemometer. Averaged along the door width, the outlet velocities range from 2.25 m/s for the first level to 4.98 m/s for level 4.

The temperature of the refrigerated room is kept constant at a temperature of about 8 °C by a cooling unit. Air is drawn from the center of the room, is cooled down, and then re-enters through perforated walls around the room. During the measurements, the temperature in the warm area of the store was about 17 °C.

To be able to visualize the temperatures and the shape of the jet, a whole-field technique was carried out, as suggested by Neto [7]. A paper screen was positioned in the doorway and its temperatures were registered by an infrared thermographic camera. The temperatures of the screen were about the same as those of the surrounding air, due to the low thermal mass of the screen.

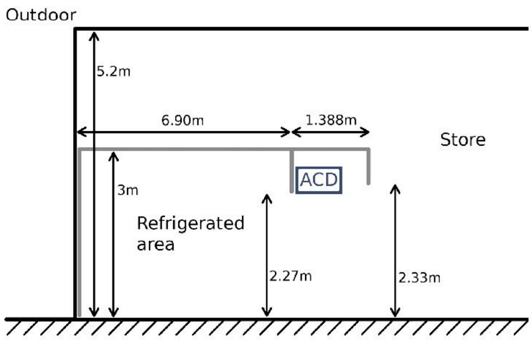

Figure 1 Geometrical properties of the test site

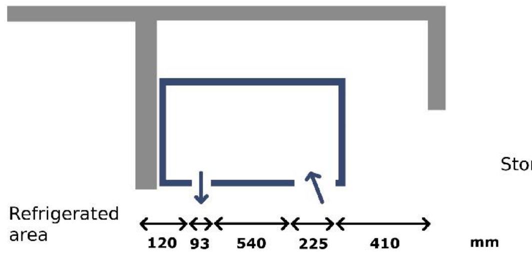

Figure 2 Geometrical properties of the air curtain device
used at the test site

In case of an air curtain, all heat transfer is associated with mass transfer. So the heat transfer rate q (W) can be calculated from the air transfer rate Q (m^3/s) using the following expression:

$$q = \rho c_p Q \Delta T \tag{5}$$

In order to measure the air transfer rate, concentration decay tests were performed using CO_2 as a tracer gas.

NUMERICAL METHOD

A numerical study was performed using the CFD software *Fluent*®. Foster [8] showed that the behaviour of air curtains can be strongly affected by three-dimensional effects. In this study however, only 2-D simulations were performed, in order to investigate the general behaviour of air curtains at the centre of the door.

Most simulations are assumed steady-state, except for the investigation of some unstable regimes. A pressure based solver is used, as well as the k-ε turbulence model. For the fluid physical properties, the Boussinesq and Sutherland approximations are applied. The discretized equations are solved using the 2nd order upwind scheme.

Studied configurations

First, a set of simulations formerly performed by Costa [9] was repeated, to check whether the used models, assumptions and boundary conditions are acceptable.

The geometry and the temperatures are identical to those used by Costa, as shown in figure 3, and will be called geometry A. The outlet velocities of the downwards blowing air curtain range from 0 to 9 m/s and the jet nozzle width b_0 is 0.045 m. The number of cells in the grid is 14300, which is a lot more than Costa's grid (1600 cells), but still requires little calculation time.

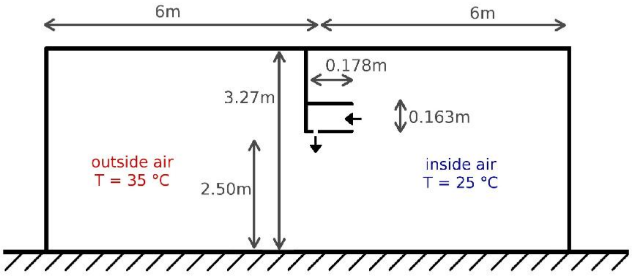

Figure 3 CFD Geometry A, also used by Costa [9].

After this set of simulations, the geometry from the test site was simulated. This will be called geometry B. The properties are shown in figure 4, as well as figures 1 and 2. The total length of the store is 20 m and the jet nozzle is 0.093 m wide. After several grid- dependence tests using the Richardson extrapolation method [10], a grid of 154840 cells was chosen. Reproductions of the experiments and a parametric study were performed using geometry B.

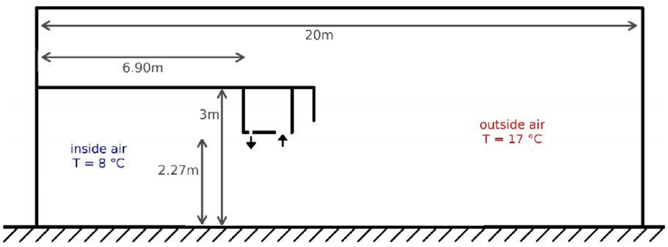

Figure 4 CFD Geometry B, corresponding to the test site

Boundary Conditions

In CFD simulations of an air curtain, the ACD is typically no part of the mesh. In that case, the jet nozzle is modelled as a velocity inlet while the air return grille is an outlet of the mesh. The disadvantage of this approach, however, is that the temperature of the blown air has to be specified. But this temperature is equal to the temperature of the return air, which is a priori unknown. So instead, the volume taken by the air curtain device is included in the mesh and the velocity in the cells at the jet nozzle is specified. A uniform velocity profile is imposed to the cells at the nozzle, as well as values for k and ε calculated from the following equations:

$$k_0 = \frac{3}{2} I_{t0}^2 \, v_0^2 = 0.00375 v_0^2 \qquad (6)$$

$$\varepsilon_0 = \frac{k_0^{3/2}}{L_\varepsilon} = \frac{2k_0^{3/2}}{b_0} \qquad (7)$$

The length scale for turbulent dissipation L_ε is chosen as half of the nozzle width $b_0/2$. The turbulence intensity I_t is set at 0.05. This value is not critical, since Guyonnaud [11] verified that the turbulence intensity does not affect the air curtain performance, as long as it is in the range of 0 to 0.20. This is always the case for a commercial air curtain.

The desired temperatures are specified to the left and right walls, as well as to some columns of cells adjacent to those walls. All other walls, including the roof and the ceiling are treated as adiabatic.

RESULTS

Flow patterns

Simulations show that three different flow regimes can occur, as can be seen in figure 5.

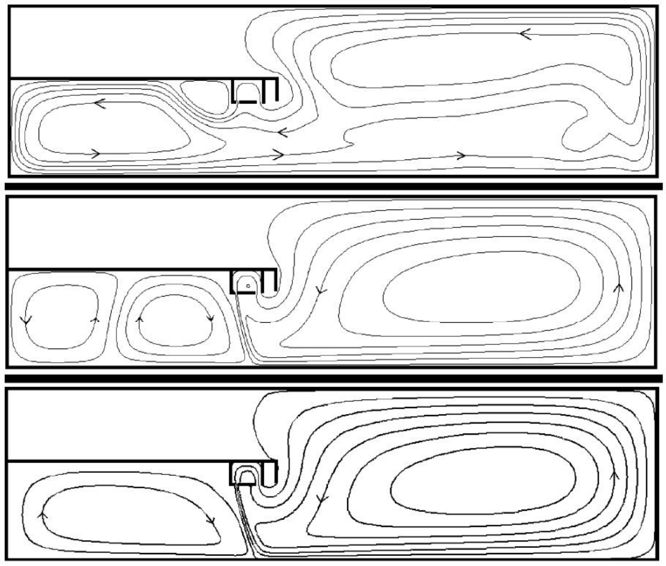

Figure 5 Streamline patterns (kg/s) for the 3 regimes: natural convection, mixed convection and forced convection.
Simulations of geometry B.

The first regime is found at low outlet velocities and is similar to the case where no air curtain is installed. In this case, buoyancy forces are dominant causing one circulation cell flowing around both rooms. This bulk transport of air is accompanied by a large amount of heat transfer. The third regime is found at high velocities and is dominated by the momentum of the air jet. In that case, two separate circulation cells (one in each space) are induced by the air jet. The air curtain is stable and the heat transfer only takes place due to turbulent mixture of the two cells at the door opening. At intermediate velocities (figure 5 middle) a mixture between the two other regimes is observed. In one area (the warm room in case of downwards blowing ACD) the air curtain momentum and the buoyancy forces cooperate to form a single circulation cell. In the other room, those two forces are opposite and two separate cells are created. These conclusions are similar to those found by Costa [9].

Temperature boundary condition

An important factor in the CFD simulations is the boundary condition for setting the temperatures. As mentioned before, a fixed temperature is set at columns of cells adjacent to the lateral walls. When the width of the area with fixed temperatures is varied, a serious variation can occur in the results of the simulation. As an example, figure 6 shows 3 different regions of geometry A to which temperatures can be assigned.

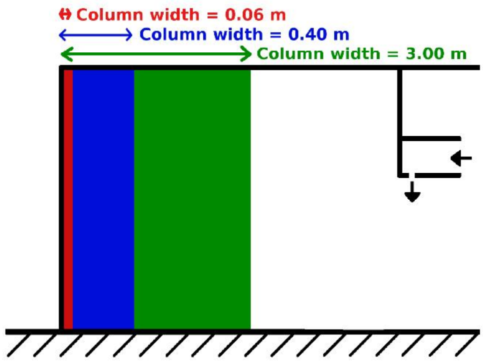

Figure 6 Three different column widths to which the temperature can be set as a boundary condition. Simulations of geometry A.

The amount of cells of which the temperature is fixed, affects the temperature distribution in an important way. In the case 3 (3 m), all air from the recirculation cell is cooled down or warmed up to room temperature before it reaches the air curtain, whereas only a part of the recirculated air takes on the correct temperature in the other cases (0.06 and 0.40 m). This variation in ΔT also affects the heat transfer through the air curtain. The heat transfer rates are 1120 W for 0.06m fixed temperature cells, 1695 W for 0.40m fixed temperature cells and 1900 W for 3m fixed temperature cells.

So it is important to impose the right boundary condition in order to obtain a realistic simulation result. In reality, the air reaching the air curtain is generally at room temperature, so the wide area gives the best prediction of the real situation. Proof of this is provided by simulations of geometry B in comparison with measured heat transfer rates, as demonstrated in table 1. The wide column option predicts the measured heat transfer very accurately, while the narrow column simulations give a considerable underestimation.

Table 1 Heat transfer rates q for simulations of geometry B, for both narrow and wide areas with fixed temperature, compared to experimental results. Note that ΔT was not always equal for different velocities.

Outlet velocity v_0 (m/s)	Measured q (W)	q (W) predicted by CFD column width = 0.5 m	q (W) predicted by CFD column width = 5.0 m
0	5548	2600	5440
2.90	1375	832	1524
3.90	2262	1186	2364
4.98	2443	1162	2530

Moreover, the narrow column predicts a mixed convection flow pattern, while the wide area gives a forced convection regime for outlet velocities of 2.90 m/s and more. Thermo graphic images confirm that in reality the forced convection pattern occurs and a wide area is the best choice. All further results in this study are based on simulations using wide columns as temperature boundary condition.

Optimal air curtain parameters

The most important parameter to adjust is the outlet velocity of the jet. Several authors [8,9,12] found that the heat transfer is high for low velocities (where the jet is unstable, such as the first flow pattern from figure 5), then drops to a minimal value for higher velocities, and subsequently increases linearly with increasing velocity. The same result was found here, as can be seen in figure 7.

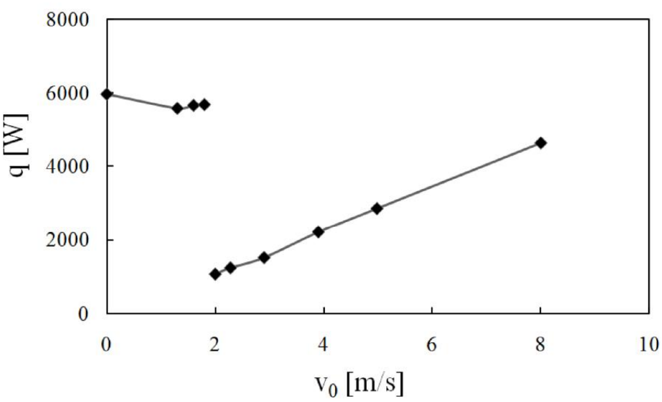

Figure 7 Heat transfer rate is a linear function of outlet velocity. Simulations of geometry B, door width w = 2 m.

The explanation for the increasing heat transfer is found in the higher amount of mixing for higher jet velocities. So it is obvious that the minimal velocity causing a stable jet leads to the optimal condition for the air curtain (2 m/s in the example of figure 8). Simulations show that the analytical formula by Hayes & Stoecker (expressions (1), (2) and (3)) is quite accurate in predicting the optimal jet velocity for a given jet nozzle width. Naturally, a suitable safety factor needs to be taken into account, to compensate for external effects such as traffic through the doorway, three dimensional effects, the influence of wind or an imbalance in ventilation flows.

Heat transfer through the air curtain

A more general representation of the heat transfer is a Nusselt number Nu. This number is defined by expression (8), where the convective heat transfer coefficient h_{conv} should not be confused with the height h. Using (9), the Nusselt number can be correlated to the heat transfer rate resulting in (10). The notations A and w represent the area and width of the doorway, while λ is the thermal conductivity of air.

$$Nu = \frac{h_{conv}\, h}{\lambda} \tag{8}$$

$$q = h_{conv}\, A\, \Delta T = h_{conv}\, w\, h\, \Delta T \tag{9}$$

$$Nu = \frac{q}{w} \frac{1}{\lambda\, \Delta T} \tag{10}$$

In order to obtain an expression for the heat transfer through the air curtain, a series of simulations was performed using geometry B and some geometrical variations of it. The range of used parameters is given in table 2.

Parameter	Minimum value	Maximum value
Jet outlet velocity v_0	0 m/s	8 m/s
Jet nozzle width b_0	0.047 m	0.130 m
Jet outlet angle α_0	-20°	20 °
Height of the doorway h	1.14 m	2.27 m
Temperature T	273 K	298 K
Temperature difference ΔT	9 K	25 K

Table 2 Range of the parameters used in simulations of geometry B.

Hayes & Stoecker suggested that the dimensionless group Nu/RePr is independent of D_m for stable air curtains that reach the opposite wall. The Reynolds number Re can be calculated from (11) where μ is the dynamic viscosity of air.

$$\mathrm{Re} = \frac{\rho_0 b_0 v_0}{\mu} \tag{11}$$

Simulation results of this study confirm this suggestion, as can be seen in figure 8, where all relevant parameters have been varied except for the jet discharge angle α_0. Because of this conclusion, there are only two dependent parameters left, namely the h/b_0 ratio and α_0. From figure 9 it is seen that there is a linear dependency between Nu/RePr and h/b_0 for $\alpha_0 = 0$.

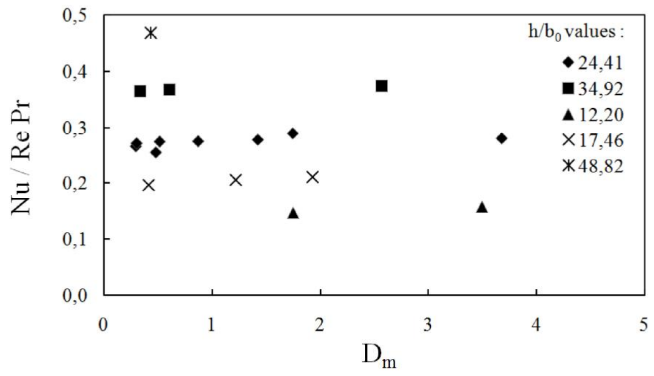

Figure 8 The dimensionless group Nu/RePr is independent of the deflection modulus. Simulations of geometry B for $\alpha_0 = 0$.

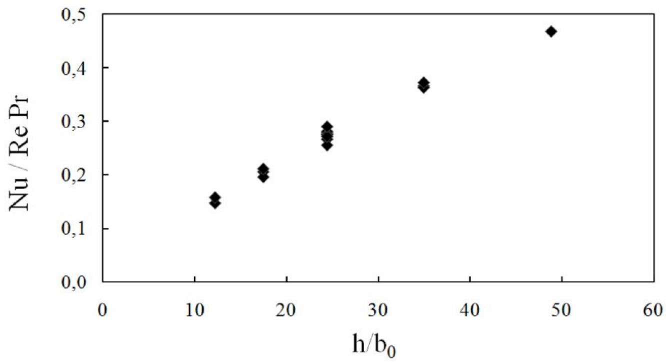

Figure 9 The dimensionless group Nu/RePr varies linearly with h/b_0. Simulations of geometry B for $\alpha_0 = 0$.

Linear regression of this data yields expression (12), with a coefficient of regression $R^2 = 0.9843$. This expression predicts all simulations to an error of less than 9%. Expression (12) is valid for downwards blowing air curtains with $\alpha_0 = 0$ and within the range of parameters given by table 2, but only when the jet outlet momentum is high enough to ensure a stable air curtain.

$$\frac{Nu}{\mathrm{Re}\,\mathrm{Pr}} = 0.008896 \frac{h}{b_0} + 0.05292 \tag{12}$$

CONCLUSION

A numerical model for a downwards blowing air curtain was created and verified through experiments, including thermo graphic imaging and tracer gas decay tests. Important to take into account in this model is the boundary condition for setting the temperatures of the surrounding rooms. Setting a fixed temperature in a number of cells gives satisfactorily results, provided that the area to which the temperature is set, is large enough to make sure that the entire air flow of the circulation cell takes on the appropriate temperature.

The maximum effectiveness of an air curtain occurs when the outlet momentum is just large enough to ensure that the air jet is stable and reaches the opposite side. The analytical expressions by Hayes & Stoecker (1), (2) and (3) are fairly accurate in predicting this minimal required jet momentum, although a safety factor needs to be taken into account. When operating at this optimal condition, an air curtain is able to accomplish an effectiveness of up to about 80%. This means it reduces the heat exchange to 20% of the corresponding value without an air curtain.

To estimate the actual heat exchange rate through a downwards blowing air curtain, a linear equation for the dimensionless group Nu/RePr is suggested (expression (12)), only as a function of the h/b_0 ratio. This equation was derived for the range of parameters shown in table 2 and a jet discharge $\alpha_0 = 0$. It is only valid when the outlet momentum is sufficient to ensure a stable operation of the air curtain.

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