NUMERICAL STUDY OF THE OPTIMAL NOZZLE WIDTH AND JET ORIENTATION IN A DOWNWARD BLOWING AIR CURTAIN

J.C. Gonçalves[1,2], J.J. Costa[2], A.R. Figueiredo[2], A.M.G. Lopes[2]

1: CI&DETS, ESAV, Polytechnic Institute of Viseu, Quinta da Alagoa, 3500-606 Viseu, Portugal.
Phone: +351-232-446-600 Fax: +351-232-426-536
E-mail: jgoncalves@esav.ipv.pt - URL: http://www.esav.ipv.pt

2: ADAI,LAETA, Department of Mechanical Engineering, University of Coimbra, 3030-788 Coimbra, Portugal

E-mail: jose.costa@dem.uc.pt, rui.figueiredo@dem.uc.pt, antonio.gameiro@dem.uc.pt - URL: http://www.uc.pt/fctuc/dem

Abstract: *This paper presents a numerical study of the performance of an air curtain device (ACD) installed on top of the access door of a refrigerated room, aiming to quantify the influence of some parameters (nozzle width, initial orientation and discharge velocity of the jet) towards the maximization of the sealing effect. For this purpose, a numerical model was developed to simulate the turbulent non-isothermal 3D airflow generated by the ACD jet, focusing on the period after the door is opened. The situation when the ACD is switched off is taken as the reference for the performance evaluation. The results allow identifying an optimum jet nozzle width corresponding to maximum sealing efficiency and lower jet airflow rate (lower energy consumption in the air curtain operation). For the present conditions (ACD installed outside the refrigerated room), the sealing efficiency is highest with the ACD oriented vertically and is practically unaffected if the jet discharge is directed towards the warmer space. On the contrary, a jet discharged towards the cooler space is very detrimental to the ACD sealing effect.*

Keywords: CFD, air curtain, nozzle width, jet discharge orientation, sealing efficiency.

1. INTRODUCTION

In refrigerating stores, the access doors usually must remain open during long periods either for routine or for loading/unloading operations. Due to the indoor-outdoor temperature difference, the infiltration of warm air into the refrigerated space can lead to important energy waste and detrimental effects on the indoor conditions. Different solutions can be adopted to minimize these effects; aerodynamic sealing by an air curtain devices (ACD) is one of the most interesting, as it can preserve the indoor conditions, while allowing free access for people and machines [1]. Air curtains can also be found in many other applications, such as for sealing the access zones to air-conditioned commercial or industrial spaces [2], preserving the indoor air quality in confined areas [3], sealing the access sections of industrial chemical-treatment furnaces [4], sealing vertical or horizontal display cabinets [5], etc.

Due to their diverse applications, air curtain devices can present different configurations, featuring: air jet direction (vertical or horizontal); return grill for immediate air recirculation; injection nozzles (single or twin-jet); fan type; inclusion of heaters and/or cooling air equipment, etc. In applications of sealing the access doorway of refrigerated rooms, a most common configuration consists of an ACD installed above the door, blowing vertically downwards (cf. Figure 1(b)).

The first air curtain reference reports to Theophilus Van Kemmel who, in 1904, patented a sealing equipment by air stream. However, the first references to studies on the heat transfer across air curtains and to proposals of non-dimensional technical approaches appeared only in the 1960's [6-9]. Hayes and Stoecker [8,9] developed an analytical model to assist in the design of air curtains. This model allows the calculation of the minimum jet velocity required to provide an unbroken curtain. However, since this is the velocity at the borderline of stability, a safety factor should be considered.

Apart from the experimental methodologies, approaches based on computational fluid dynamics (CFD) have been increasingly used in the study of air curtains [5,10,11], taking advantage of its great versatility.

A number of research works can be found on the study of the influence of the ACD parameters (location, angle and initial velocity of the jet), in order to maximize its sealing efficiency [10,13], but few have addressed the jet nozzle characteristics. Moreover, the results of such studies can only be related with each specific configuration, and a dimensionless analysis is required.

The present paper describes a parametric study conducted on a full-scale geometry, aiming to establish the dependency of some parameters of the ACD jet, namely the nozzle width b, the jet discharge angle α_j and the initial jet velocity V_j, towards the maximization of the sealing efficiency.

2. CFD MODEL

The numerical model used in the present study was developed and validated as described in [10], and the necessary adaptations were implemented for full-scale geometries.

2.1. Physical domain

The geometry of the numerical model consists of two adjacent rooms, with similar dimensions 6×6×4 m^3, connected by a door with height 2 m and width 1.8 m (Figure 1(a)). The ACD is installed on top in the outside. One of the rooms represents the indoor refrigerated space that is to be kept sealed, while the other one represents the outdoor environment.

The air curtain device is represented by a parallelepipedic solid (0.4 m wide, 0.3 m high and 2 m long). The discharge nozzle on the bottom face has a width of 10 cm and the vertical air intake section is 15 cm high. Usually, the access doors of refrigerated chambers are suspended from rails and are opened/closed by way of a horizontal sliding mechanism. Therefore, the ACD was positioned 10 cm apart from the lintel of the door. A length of 1.9 m was considered for the discharge nozzle to ensure that the air curtain covers the whole width of the doorway.

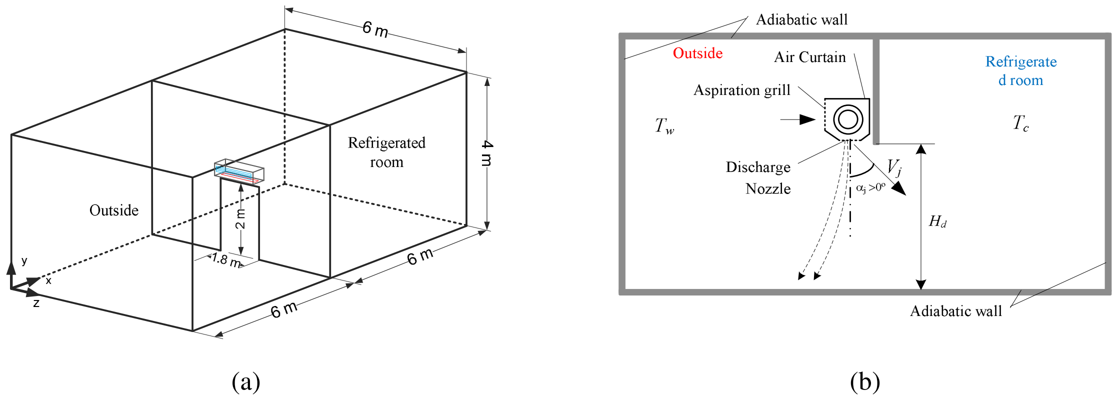


Figure 1. (a) Sketch of the 3D geometry and dimensions of the calculation domain; (b) sketch of a cross section at the doorway middle plane, showing some important parameters and boundary conditions.

2.2. Numerical method and solution procedure

A 3D numerical model was implemented using the commercial code ANSYS CFX®. For symmetry reasons, only half of the physical domain was simulated. The methodology consists on the numerical simulation of the transient, 3D turbulent airflow originated by the curtain jet. The calculations are based on the solution of the Reynolds averaged equations for the conservation of mass, momentum and energy. The turbulence effects were modeled using the k-ω SST (Shear Stress Transport) model. The advection and transient terms were discretized with the High Resolution and the Second-Order Backward Euler schemes, respectively. The iterative calculation procedure was assumed close enough to convergence when all normalized residuals became lower than 10^{-4}.

An unstructured mesh was adopted for the discretization of the spatial domain, ensuring a better refinement in regions where higher gradients were expected (close to the doorway plane). For example, b=10 cm a total of 584 861 elements were used in the simulated domain. A computer Core(TM) Duo CPU E8500 @ 3.16 GHz 64 bit was used, which typically took about 56 hours of CPU time to simulate 180 s of physical phenomena.

2.3. Initial and boundary conditions

The envelope of the calculation domain was taken as impermeable and adiabatic, air changes being allowed only through the doorway between the two rooms. As initial conditions, the air was assumed stagnant all over the domain, at uniform temperatures of 5°C and 30°C, for the refrigerated space and for the outside environment, respectively. All surfaces were considered as adiabatic, smooth and non-slipping. The airflow field inside the air curtain device was not simulated. The downward jet exit section corresponds to the airflow inlet section of the calculation domain, where both temperature and velocity are assigned. On the other hand, the air return (intake) frame of the ACD is the only flow exit (from the domain), with specified relative pressure. During the calculations, the average temperature estimated in the return section is assigned to the downward injected airflow, thus ensuring the global energy and mass conservation. A turbulence intensity of 5 % was adopted for the jet.

3. RESULTS

3.1. Sealing efficiency

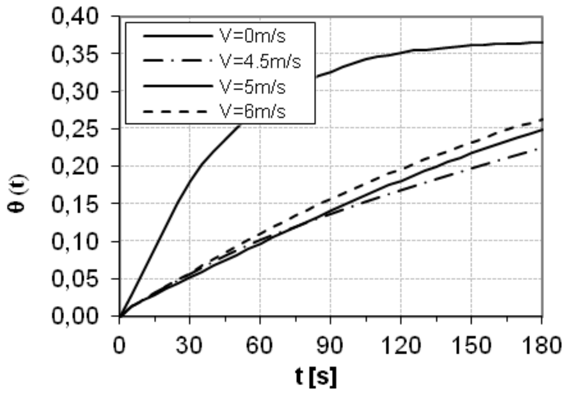
Considering only sensible heat transfer, the amount of energy gained by the cold room (responsible for the increase of the air temperature) from the instant the door is opened up to a generic instant *t*, can be represented by the normalized temperature increase $\theta(t)$, defined as:

$$\theta(t)=\frac{T_c(t)-T_{c,i}}{\Delta T_i}=\frac{T_c(t)-T_{c,i}}{T_{w,i}-T_{c,i}}, \tag{1}$$

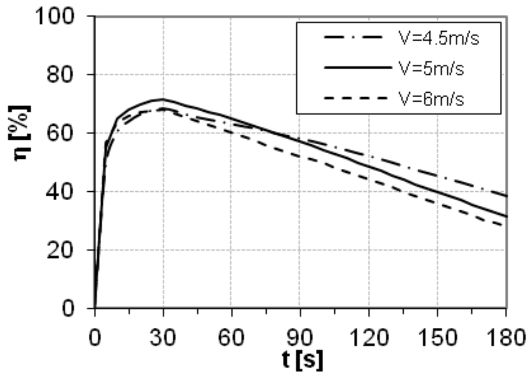
where $T_c(t)$ stands for the volume-averaged air temperature in the cold room at instant *t*, and $T_{c,i}$ and $T_{w,i}$ are the initial temperatures of the cold and warm room, respectively.
The sealing efficiency of the air curtain can be defined as the ratio between the reduction of the energy leakage through the doorway due to the air curtain operation ($\theta_0(t)-\theta(t)$) and the energy leakage $\theta_0(t)$ with the ACD turned off ($V_j = 0$).

$$\eta(t)=\frac{\theta_0(t)-\theta(t)}{\theta_0(t)}=1-\frac{\theta(t)}{\theta_0(t)}=1-\frac{T_c(t)-T_{c,i}}{\left(T_c(t)-T_{c,i}\right)_0}. \tag{2}$$

Initially, different values of the discharge velocity V_j of the jet were tested in order to determine the optimal velocity corresponding to the maximum sealing effect. Figure 2(a) shows the time evolution of the thermal energy gained through the doorway by the refrigerated space, from the instant the door is opened ($t = 0$).



(a)



(b)

Figure 2. Time evolution of (a) of the dimensionless temperature increase θ, and (b) the sealing efficiency, for different jet discharge velocities. Efficiency peaks: 68.1 % (V_j = 4.5 m/s); 71.6 % (V_j = 5 m/s) and 67.9 % (V_j = 6 m/s). [H_d = 2 m, ΔT = 25 ºC, b = 10 cm, α_j = 0º].

It is observed that, when the air curtain is switched off (V_j = 0 m/s), the energy flow rate through the doorway is approximately constant in the early 30 s after opening the door and then decreases gradually until about 120 s. With the ACD turned on, the heat flow rate across the doorway is almost constant.

For all different jet velocities, Figure 2(b) shows a fast decrease in the sealing efficiency after $t = 30$ s. This is due to the fact that the heat flow rate in the reference case ($V_j = 0$) also decreases quickly after this period. The highest peak of sealing efficiency (71.6 %) was obtained for $V_j = 5$ m/s.

3.2. Jet discharge nozzle width

To study the influence of the ACD nozzle width b on the sealing efficiency, calculations were conducted considering b values in the range of 5 cm to 12.5 cm. The results presented in Table 1 show that, as b increases, the maximum peak of sealing efficiency (in bold) is achieved for lower initial jet velocity values.

Table 1. Peaks of the sealing efficiency (%) for different values of the discharge nozzle thickness and of the jet discharge velocity ($\Delta T = 25$ºC, $H_d = 2$ m).

V_j [m/s]	b [cm]				
	5	7.5	8.75	10	12.5
4.5		-	-	68.1	**71.8**
5	12.8	62.6	**71.8**	**71.6**	69.4
5.5	-	-	70.7	-	-
6	-	**70.7**	-	67.9	-
7	52.8	66.7	-	62.4	-
8	60.3	62.3	-	-	-
9	**63.9**	57.8	-	51.0	-

Depending on the manufacturer and the ACD model, a large variety of dimensions for the jet discharge nozzle can be adopted. For sealing refrigerated rooms, thin air curtain jets are commonly used, but this requires larger discharge velocities in order to keep the initial momentum of the jet. In air-conditioned spaces, where thermal comfort conditions are required, jet discharge velocities should be low, in order to avoid human discomfort. Quite often the adoption of thicker air curtains for this situation is suggested [8,9].

Table 2. Optimum discharge conditions for different nozzle widths ($\Delta T = 25$ ºC, $H_d = 2$ m).

b [cm]	$V_{j,opt}$ [m/s]	Discharged flow rate [m^3/s]	η_{max} [%]	$\rho_j b V_j^2$
5	9	0.855	63.9	4.70
7.5	6	0.855	70.7	3.13
8.75	**5**	**0.831**	**71.8**	**2.54**
10	5	0.950	71.6	2.90
12.5	4.5	1.069	71.8	2.94

Except for Longdill and Wyborn [12], who suggest a range of $17 < H_d/b < 45$, no other recommendation was found regarding the magnitude of this parameter.

Derived from Table 1, Table 2 displays, for each b value, the jet velocity, discharged airflow rate, efficiency and momentum, for the situation of maximum sealing efficiency. According to the results, the nozzle thickness $b = 8.75$ cm represents the most favourable case, with lower jet airflow rate and, consequently, lower energy costs in the ACD operation. A wider discharge nozzle would not improve the sealing efficiency and would require an increase of the fan speed to guarantee a larger airflow rate. Moreover, this is also the configuration corresponding to a lower initial momentum of the jet, keeping at the same time a high sealing performance. Thus, one may conclude that, at least for the simulated conditions (ΔT = 25 ºC, H_d = 2 m), this is the optimum width of the discharge nozzle. It should also be emphasized that, although a good sealing effect is observed for almost all discharge nozzle widths, the results seem to indicate that narrow discharge nozzles lead to lower sealing efficiencies.

3.3. Jet discharge orientation

Hayes and Stoecker [10,11] recommend directing the jet towards the heated space, taking an angle of 15° to 30° with the vertical, to achieve better jet stability and effectiveness. Simulations were carried out keeping H_d = 2 m, b = 10 cm, V_j = 5 m/s, and varying the initial orientation angle α_j from −15° to +15°. α_j is considered positive when the jet is directed towards the cold room, as shown in Figure 1(b). The numerical results plotted in Figure 3 indicate that the sealing performance seems to be unaffected when the jet is directed towards the heated space (α_j < 0°). On the other hand, if the jet is discharged towards the cooled space (α_j > 0°), the sealing effect is severely reduced. These results are in agreement with those obtained by Jaramillo *et al.* [11]. In fact, directing the jet towards the heated space significantly increases its stability, while keeping a reduced sensitivity to external disturbances such as moving people or objects, external winds or even due to the cooling system (fans). Figure 3 also indicates that, under the present conditions, the highest sealing efficiency (71.6 %) is observed when the jet is vertically oriented.

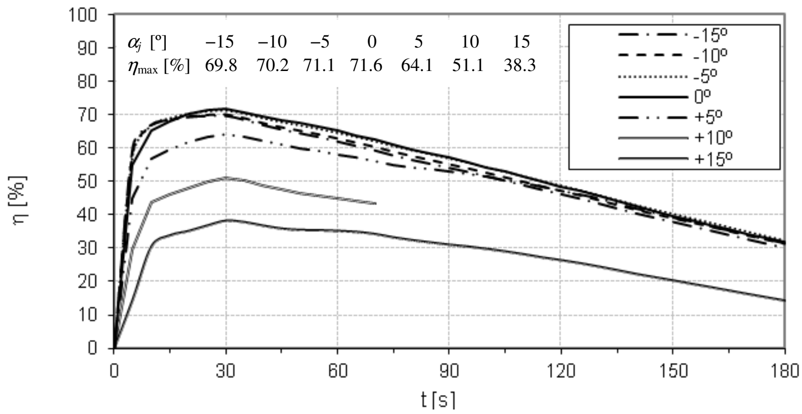


Figure 3. Time evolution of sealing efficiency for different jet discharge orientation angles (H_d = 2 m, b = 10 cm, V_j = 5 m/s).

In many real situations, the jet is poorly regulated in terms of initial orientation (Downing and Meffert [13]), and the sealing efficiency is consequently reduced. In fact, it was verified that orienting the jet under a small angle (5° and V_j = 5 m/s) towards the cold side results in a worse sealing effect than orienting the jet towards the warm side (15°), even for a lower jet discharge velocity (V_j = 4 m/s). However, as reported by [11], if the ACD is installed inside the colder space, directing the jet towards the warm space increases the ACD sealing effect.

4. CONCLUSIONS

Using a numerical model, a parametric study was conducted considering a full-scale geometry of a cold store, in order to investigate the influences of the nozzle width and of the jet orientation on the optimum jet discharge velocity of a vertical air curtain device. The advantages of using air curtains with wider nozzles were confirmed, allowing lower jet discharge velocities (with particular interest in thermal comfort applications) and providing a better sealing effect.

The commercial ACD devices currently in use have usually thin discharge nozzles, below the optimal value. Thus, it is advisable that the ACD manufacturers increase the discharge nozzle width, while maintaining air curtain power to provide sufficient jet discharge velocity.

It was also confirmed that, if the ACD is installed outside the refrigerated room, the sealing effect is severely reduced when the air jet is directed towards to the cooled space ($\alpha_j > 0°$). Directing the jet discharge towards the warm space does not seem to affect the sealing efficiency.

REFERENCES

[1] J.J. Costa, L.A. Oliveira and M.C.G Silva. "Energy savings by aerodynamic sealing with a downward-blowing plane air curtain - A numerical approach", Energy and Buildings, 38(10), pp. 1182-1193, 2006..

[2] L.P.C. Neto, M.C.G. Silva and J.J. Costa. "On the use of infrared thermography in studies with air curtain devices", Energy and Buildings, 38(10), pp. 1194-1199, 2006.

[3] J.P. Rydock, T. Hestad, H. Haugen and J.E. Skaert. "An isothermal air curtain for isolation of smoking areas in restaurants". Proceedings of 7th International Conference on Air Distribution in Rooms, RoomVent 2000, Reading (England), vol. 1, pp. 663–668, 2000.

[4] L.A. Oliveira, J.J. Costa, M.G. Carvalho, H.J. Gerhardt and C. Kramer. "On Aerodynamic Sealing for Industrial Applications". Journal of Wind Engineering and Industrial Aerodynamics, 37, p. 255-268, 1991.

[5] P. D'agaro, G. Cortella and G. Croce. "Two- and three-dimensional CFD applied to vertical display cabinets simulation", International Journal of Refrigeration, 29(2), pp.178-190, 2006.

[6] K. Takahashi and M. Inoh. "Some measurements on air curtain efficiency for cold rooms", Proc. 11th International Congress of Refrigeration, Vol. II, pp. 1035-1039, Munich, 1963.

[7] G. Hetsroni, C.W. Hall, and A.M. Dhanak. "Heat transfer properties of an air curtain", Proceedings ASRAE Transaction, pp. 328-334, 1963.

[8] F.C. Hayes and W.F. Stoecker. "Heat transfer characteristics of the air curtain", ASHRAE Transactions, 75(2), pp. 153-167, 1969.

[9] F.C. Hayes and W.F. Stoecker, "Design data for air curtains", ASHRAE Transactions, 75(2), pp. 168-180, 1969.

[10] J.C. Gonçalves, J.J. Costa, A.R. Figueiredo and A.M.G. Lopes. "Study of the aerodynamic sealing of a cold store – Experimental and numerical Approaches", Energy and Buildings 55 (2012) 779–789, 2012.

[11] J.E. Jaramillo, C.D. Pérez-Segarra, A. Oliva and C. Oliet. "Analysis of the Dynamic Behaviour of Refrigerated Space Using Air Curtains". Numerical Heat Transfer, Part A, 55(6), p. 553–573, 2009.

[12] G.R. Longdill and L.G. Wyborn. "Performance of air curtains in single storey cold stores", Proc. XV International Congress of Refrigeration, pp. 77-88, Venice, 1979.

[13] C.C. Downing and W.A. Meffert. "Effectiveness of cold-storage door infiltration protective devices", ASHRAE Transactions, 99(2), pp. 356-366, 1993.