Efficiency of a heated air curtain under cross-jet temperature gradients

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Abstract
The infiltration of unconditioned outdoor air to controlled indoor environments can have negative impacts on the energy performance and indoor environmental quality of buildings. Air curtains can be used to reduce infiltration through entrance doors where transit is frequent. The stability and performance of air curtains strongly depends on the combination of jet and environmental forces that act on them, including forces due to cross-jet density gradients and injection of buoyancy in the jet. This study addresses the performance of a heated air-curtain system by means of validated computational fluid dynamics (CFD) simulations. Two performance indicators are considered: (1) separation efficiency (related to mass transport), and (2) thermal efficiency (related to heat transport). The results indicate that based on the performance indicators, the use of a heated air curtain is not favorable over the use of an isothermal air curtain.

Key Innovations
- Heated and unheated air curtains are compared in terms of both thermal and separation performance.
- Performance is evaluated under common air-curtain operational settings for various representative environmental temperature conditions of cold and moderate climates.
- A suitable indicator for assessing thermal (energy) performance of heated air curtains is formulated.

Practical Implications
Despite the rather frequent usage of heated air curtains at the entrance of commercial, industrial and institutional buildings in cold and moderate climates, their implementation is less efficient than unheated air curtains from energy conservation and pollution control perspectives. Therefore, whenever air curtains are necessary, the employment of unheated air curtains for the restriction of heat losses and pollutant transport through building entrances is recommended.

Introduction
Air infiltration is usually associated with increased energy demand in buildings (Emmerich and Persily, 1998; Younes et al., 2012; Brinks et al., 2015). Moreover, problems concerning thermal comfort and indoor air quality can in many instances also be attributed to air infiltration (Liddament, 1986; Lstiburek et al., 2002). The infiltration of unconditioned air through access doors significantly contributes to these potential problems, especially in buildings where transit through these doors is frequent. Air curtains are often implemented at these locations in order to facilitate the transit of people, vehicles and materials while minimizing energy losses, decreasing the transport of outdoor air pollutants to the inside and reducing thermal discomfort.

The stability of an air curtain and its performance in separating two environments is greatly influenced by the combination of forces acting on it, which depend on jet parameters and surrounding environmental conditions. The former define the strength of the air curtain by the injection of momentum (which relates to the square of the bulk velocity of the jet) and buoyancy (which relates to the difference of jet temperature with respect to its surroundings) in the jet, whereas the latter induce lateral loads on the air curtain due to cross-jet gradients in density (temperature/species concentration) and/or pressure. Isothermal jet air curtains (only driven by momentum) are commonplace, however, for certain applications the use of air curtains with buoyant jets is fairly normal. Examples are heated air curtains at the entrance of buildings in cold or moderate climates and cooled air curtains in refrigerated compartments, both of which are subjected to temperature differences between the separated environments. In these air curtains an interaction occurs between forces due to cross-jet density differences, jet buoyancy and jet momentum. Despite the frequent occurrence of such scenarios, this interaction and its influence on air-curtain performance has not been studied frequently. This paper provides an analysis of the added effect of jet temperature and ambient temperature on the overall flow behavior and the efficiency of a heated air curtain at the entrance of a building. The analysis is performed numerically using validated computational fluid dynamics (CFD) simulations.
Methods
Validation

Validation is performed using dedicated experimental data for a generic impinging jet flow. Khayrullina et al. (2017) performed 2D particle image velocimetry measurements of the velocity field of isodense turbulent impinging jets in a water tank under dynamic (i.e., jet Reynolds number $Re_{jet}$) and geometric (i.e., jet height-to-width ratios $h_{jet}/W_{jet}$) conditions that are characteristic of air-curtain jets. In the present study, the experimental data corresponding to $Re_{jet} = 13,500$ and $h_{jet}/W_{jet} = 45$ is used for validation. The experimental setup (Figure 1) is replicated in a 3D computational domain (Figure 2) where appropriate boundary conditions are applied to produce the correct impinging jet flow: at the inlet boundary a velocity magnitude $V = 0.14$ m/s, turbulence intensity $TI = 15\%$ and turbulence length scale $l = 0.426$ mm are imposed, yielding a jet inlet velocity $V_0 = 1.7$ m/s; at the outlet boundary a zero static gauge pressure is enforced; wall boundaries are smooth and non-slip; lastly, symmetry boundaries are used to shorten the domain and reduce computational costs. Following best practices, a fully structured computational grid consisting of 1,209,630 hexahedral cells is generated (see Figure 2), of which the resolution is determined through a grid-sensitivity analysis. For this analysis, the grid is coarsened and refined with an overall factor $\sqrt{2}$ (1.123 per coordinate direction), resulting in grids with 838,782 cells (basic), 1,209,630 cells (medium) and 1,667,632 cells (fine). The results showed that the medium grid attains a reasonably low grid convergence index value of 0.8%, and thus this grid is used for the simulations presented in the remainder of this section. CFD simulations are performed using steady Reynolds-averaged Navier-Stokes (RANS) equations in conjunction with the standard $k-\varepsilon$ turbulence model (Jones and Launder, 1972) and scalable wall functions (i.e., standard wall functions as per Launder and Spalding (1972) with an imposed lower limit on the dimensionless wall distance). The SIMPLE algorithm is used for pressure-velocity coupling and purely second-order schemes are implemented for the discretization of the flow equations. A comparison of

![Figure 1: Schematic of experimental setup.](image1)

![Figure 2: Domain dimensions, computational grid and boundaries of the validation case indicated in xy plane.](image2)
experimental and simulation results, in terms of the mean velocity magnitude is presented in Figure 3. The predicted values of the dimensionless mean velocity magnitude \(|V|/V_0\) with \(V_0\) the mean jet inlet velocity) show a very good agreement with the measurement results (NRMSE = 3.2% for the centerline and 6.7% for the crossline), therefore, the settings and parameters are used in the case study. Detailed information on the experimental campaign can be found in Khayrullina et al. (2017), whereas details on the CFD simulations are given in Alanis Ruiz et al. (2018).

**Case study**

Two cases of an air-curtain system at the entrance of a conditioned building are evaluated and compared over a reasonable range of cross-jet air temperature differences (\(5^\circ C \leq \Delta T_{\text{crossjet}} \leq 25^\circ C\); where \(\Delta T_{\text{crossjet}} = T_{\text{in}} - T_{\text{out}}\) is the air temperature difference between indoor and outdoor environments) that are representative of temperature conditions in cold and moderate climates. These cases refer to a heated (non-isothermal, i.e. \(T_{\text{jet}} \neq T_{\text{in}}\)) air curtain and an isothermal air curtain (\(T_{\text{jet}} = T_{\text{in}}\)).

The 3D computational domain and relevant dimensions are shown in Figure 4. The computational grid is carefully constructed from hexahedral cells (~0.9 million) with a grid-resolution based on a grid-sensitivity analysis. The grid sensitivity analysis is performed using three consecutive grids that are nearly uniformly refined with an overall factor 1.5 (1.145 per coordinate direction): basic grid with 598,680 cells, medium (selected) grid with 896,750 cells and fine grid with 1,334,034 cells. In Figure 5, the numerical solutions computed with the three grids are compared in terms of the mean dimensionless velocity.
magnitude along the air-curtain jet centerline for a situation without temperature gradients.

At the indoor side a pressure boundary with zero static gauge pressure and constant backflow air temperature ($T_{in} = 21^\circ C$) is applied, whereas at the outdoor side (pressure boundary with zero static gauge pressure) air temperature varies according to $T_{out}$ of each case. Moreover, in the transversal direction (not visible in Figure 4), symmetry boundary conditions delimit the domain to a depth of 500 mm. A mixture consisting of “distinct” air species—yet each with the same physical properties—is introduced in the model with the aim of calculating mass transport. Air is injected downwards into the domain through a nozzle (width $W_{jet} = 60$ mm) by a heated (non-isothermal) air-curtain device with uniform velocity $V_{jet} = 6.7$ m/s, turbulence intensity $TI = 6\%$ and temperature $T_{jet} = 30^\circ C$, corresponding to a momentum flux $M_{jet} = \rho W_{jet} V_{jet}^2 = 3.3$ kg/s$^2$ and a buoyancy flux $B_{jet} = W_{jet} V_{jet}^2 g(T_{jet} - T_{in}) = 0.12$ m$^3$/s$^3$, both per unit curtain length, where $\rho$ is the density of air in the jet and $\beta$ the thermal expansion coefficient. The same set of parameters applies to the isothermal case, however for this case the jet temperature is equal to the indoor air temperature and thus the buoyancy flux is zero.

3D steady RANS equations are adopted and supplemented with the conservation equations of energy and species. In conformity with the validation study, the standard $k$-$\varepsilon$ model consists of using a criterion to provide closure to the equations. Standard wall functions are used for near-wall treatment and the SIMPLE algorithm is employed for the coupling of pressure and velocity. Second-order discretization schemes are used throughout. The commercial CFD code ANSYS Fluent 18.2 is used.

Definitions

Two distinct performance indicators are defined in this paper to evaluate the performance of the air-curtain system: (1) the separation efficiency ($\eta_s$) which concerns mass transfer, and hence pertains to the dispersion of external pollutants into the controlled indoor environment; and (2) the thermal efficiency ($\eta_t$) which concerns heat transfer, and hence pertains to the energy losses that take place in the system. In both instances the indicator is represented by the ratio between the corresponding transported quantity across the door/opening when the air curtain is in operation and when it is not. The first indicator is obtained by adopting the definition of the ‘adapted’ separation efficiency, as presented in Alanis Ruiz et al. (2021), and formulating it in terms of mass transport of infiltrated outdoor air species—this can be thought of as the quantification of polluted air transfer to a conditioned environment.

$$\eta_s = 1 - \frac{\dot{m}_{in}}{\dot{m}_{in,ref}}$$

where $\dot{m}_{in}$ is the mass flow rate of infiltrated outdoor air and the subscript $ref$ relates to the reference scenario (i.e., when the air curtain is not in operation). Here the mass flow rate of infiltrated air is calculated as:

$$\dot{m}_{in} = \int x_{out}(\vec{v} \cdot \vec{n}) dS_{in}$$

with $x_{out}$ the mass fraction of outdoor air, $\rho$ the average air density, $dS_{in}$ a surface element at the indoor side where the mass flow rate is computed, and $\vec{v} \cdot \vec{n}$ the dot product between the local velocity vector $\vec{v}$ and the outward normal vector $\vec{n}$ of the measurement surface. The definition of the second indicator is formulated from an overall energy balance of the air-curtain system, where relevant contributions to the heat losses of the indoor side of the system are taken into account.

$$\eta_t = 1 - \frac{\dot{q}_{in} + \dot{q}_{ex} + \dot{q}_{ACloss}}{\dot{q}_{in} + \dot{q}_{ex}}$$

where $\dot{q}_{in}$ and $\dot{q}_{ex}$ are the heat flow rates associated to the infiltration of cold outdoor air and the exfiltration of conditioned indoor air, respectively, and $\dot{q}_{ACloss}$ is the heat flow rate related to the losses of heated air from the air curtain to the outdoor environment. The heat flow rates can be expressed as:
\[ \dot{q}_{\text{in}} = m_{\text{in}} c_p |T_{\text{out}} - T_{\text{in}}| \]  
(4)

\[ \dot{q}_{\text{ex}} = m_{\text{ex}} c_p |T_{\text{out}} - T_{\text{in}}| \]

\[ = c_p |T_{\text{out}} - T_{\text{in}}| \int x_{\text{in}} \rho (\vec{v} \cdot \vec{n}) dS_{\text{out}} \]

(5)

\[ \dot{q}_{\text{AC losses}} = c_p |T_{\text{jet}} - T_{\text{out}}| \int x_{\text{jet}} \rho (\vec{v} \cdot \vec{n}) dS_{\text{out}} \]

(6)

In these expressions, \( c_p \) is the specific heat capacity of air at constant pressure, \( x_{\text{in}} \) the mass fraction of indoor air, \( x_{\text{jet}} \) the mass fraction of air that is ejected by the air-curtain device and \( dS_{\text{out}} \) a surface element at the outdoor side where the flow rate is computed.

Figure 6 demonstrates the implementation of the aforementioned expressions for the air-curtain system in the present case study.

Results and discussion

Figure 7 shows the air temperature distributions resulting from the isothermal and non-isothermal (heated) air-curtain cases under three cross-jet temperature differences. Two consistent patterns can be observed. The first one is related to the reduced sealing capability of the heated air curtain compared to the isothermal air curtain,
especially at greater cross-jet temperature difference. The second is in relation to the dispersion of heat from the air-curtain jet to the outdoor environment at lower cross-jet temperature differences, in which considerable heat losses are evident, however, the losses are more noticeable for the heated jet case (Fig. 7b,d,f). Clearly, the temperature distributions reflect a situation where the heated air curtain allows exfiltration of warmer jet air at lower cross-jet temperature differences while admits more infiltration of cold outdoor air at higher cross-jet temperature differences; both events are undesirable.

Figure 8 shows the evolution of the thermal and separation efficiencies with increasing cross-jet temperature differences for both air-curtain cases, which reinforces and quantifies previous observations. Indeed the separation efficiency (Figure 8a) of the heated device decreases due to the opposing force to the jet momentum caused by positive buoyancy, which results in higher infiltration for every cross-jet temperature difference. The reduced thermal efficiencies (Figure 8b) found for the heated air curtain at lower cross-jet temperature differences are caused by the same reason, whereas reduced thermal efficiencies at higher cross-jet temperature differences are triggered by losses of heated air from the air curtain.

**Limitations and future work**

Although performance is quantified and demonstrated for two distinct cases of an air curtain with and without a heated jet at a building entrance, while representative, these are hypothetical cases consisting of simplified geometries and static environmental boundary conditions (i.e., constant pressure and steady temperature differences), and thus they do not aim to portray a specific air-curtain system. Dynamic and non-uniform conditions often occur, including for example fluctuating wind pressure loads, density changes due to transient temperature or concentration variations, varying geometric obstructions and movement of people or materials. In a similar manner, jet parameters that control the air discharge by an air-curtain device may also be dynamic and incorporate features that alter the issued jet in time and space. Another limitation to consider is that the validation of the numerical model is performed using impinging jet data under constant density conditions and without cross-jet gradients, while, typically, air curtain applications involve some sort of cross-jet gradients, and sometimes jet density variations (as in one of the cases presented in this study).

Addressing these aspects, ongoing work includes detailed numerical and experimental research on the behavior of air curtains subjected to thermal interactions (i.e., combined cross-jet density differences and jet buoyancy) and cross-jet pressure effects. The emphasis is put into in-depth analysis of performance and jet dynamics of heated, cooled and isothermal air curtains under transient conditions that are more realistic for building and industrial applications. Furthermore, the generation of extensive experimental data for the validation of numerical and analytical models for different types of air-curtain systems, including non-isothermal air curtains, is targeted.

**Conclusions**

In this study the performance of a heated air-curtain system is evaluated in terms of heat and mass transport by the indicators defined as thermal and separation efficiency, respectively. Results are compared against those from an air-curtain system without heating (isothermal). The results indicate that from energy conservation and pollution control perspectives, the use of a heated air curtain is not favorable over the use of an isothermal air curtain, since in every instance the heated air curtain yields additional heat losses and infiltration. Nevertheless, from a thermal comfort standpoint, it could be that the adoption of heated air curtains is beneficial to counteract potential factors of thermal discomfort at building entrances such as cold air drafts. The former is an aspect that will be considered in future investigations.
Acknowledgement

The FWO is acknowledged for the financial support of the first author and of the present research (project FWO G085618N). This work was sponsored by NWO Exacte Wetenschappen (Physical Sciences) for the use of supercomputer facilities, with financial support from the Netherlands Organization for Scientific Research (NWO). Finally, the authors gratefully acknowledge the partnership with ANSYS CFD.

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