Effectiveness and optimum jet velocity for a plane jet air curtain used to restrict cold room infiltration.

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Abstract

The effectiveness of a 1.0 m wide air curtain fitted over a 1.36 m wide entrance to a cold store has been evaluated. It was shown that careful setting up of the air curtain (adjusting the jet velocity and angle) was needed, this achieved an effectiveness of 0.77 compared to the initial value of only 0.31 as set by the installer. An analytical model to predict the optimum jet velocity was compared to measured data. It is important to choose the correct safety factor (an increase in the jet velocity) for this model, as an effectiveness of between 0.37 and 0.70 could be produced using the range of safety factors found in the literature. A 2-D computational fluid dynamics (CFD) model predicted a maximum effectiveness of 0.84 and showed how the effectiveness of the curtain was related to the shape of the jet and how this varied with jet velocity and door open duration.

Key words: Cold store; Entrance; Computational fluid dynamics (CFD), Modelling, Air curtain, Turbulent jet

1. Introduction

Air infiltration can account for more than half the total heat load for refrigerated stores. It is also the main source of frost on evaporators [1] and can lead to accidents caused by ice [2].

The traditional, and most common, equipment for reducing infiltration is the transparent PVC strip curtain. However, Ligtenburg and Wijjfels [3] claim that ‘they are generally considered as unsafe, not particularly efficient, unhygienic and requiring much maintenance and it is possible that they maybe banned in the future.’ Vestibules (air locks) and flexible, fast-opening doors, often in combination with each other, are other methods employed to reduce infiltration. Vestibules restrict access, are difficult to fit to existing sites and can be bulky. Flexible, fast-opening doors have heavy maintenance requirements and reduce vision for forklift truck operators.

Air curtains reduce infiltration without taking up as much space as vestibules and without impeding traffic. Their origin dates back to a patent applied for by Van Kennel in 1904 [15] and they have been popular for around 50 years. Air curtains consist of a fan unit that produces a jet of air forming a barrier to heat, moisture, dust, odours, insects etc. In the case of cold store air curtains, the fan unit is usually above the door blowing a jet vertically down. Some air curtains recirculate their air via a return duct but it is simpler and more common not to do so.

Measured values of the static effectiveness of air curtains as found installed on cold stores (before improvements were made) have been between -0.44 and 0.78 [4]. An effectiveness of 1.0 means that infiltration is totally eliminated, 0 means that there is no effect on the infiltration and a negative value indicates that the infiltration is worse.

The basic theory for natural convection of fluids at different densities through openings was expressed more than 70 years ago [5]. Since then many authors have created improved models [6, 7, 8, 9, 10, 11]. Most air curtains are essentially plane turbulent jets and the physics of these jets are also well documented [12, 13, 14].

Solving the equations for natural convection through openings and also for turbulent jets allows the interaction between the air curtain and infiltration through the entrance to be evaluated. Hayes and Stoecker [15, 16] created analytical equations to predict heating and cooling loads across non-recirculatory air curtains. This theory deviated from measurements by about 15% for air curtain height (H) to air jet thickness (b) ratios of up to 84. Guyonnaud et al. [17] demonstrated differences between the Hayes and Stoecker, and Lajos and Preszler [18] model. They showed that knowing the height of the air curtain, jet thickness and jet velocity was not sufficient to simply describe the fluid mechanics of the air curtain. The convection of jet vortices and the height of the impinging zone, take part in the air curtain sealing mechanism.

Ge and Tassou [19] used finite difference models with good success to predict the performance of air curtains for vertical refrigerated display cabinets. Due to the large computing power required by the finite difference models, the authors then based simplified correlation models on the predictions. Foster et al. [20, 21] investigated the use of computational fluid dynamics (CFD) to predict flow through an unprotected cold store entrance.

The authors of this paper have found only one publication [3] where finite difference models have been used to predict the effectiveness of an air curtain for a cold store entrance.

The aim of this study was to measure the effectiveness of a commercially available air curtain at different jet velocities and find the optimum jet velocity to give the maximum effectiveness. The analytical model of Hayes and Stoecker [16] was used to predict the optimum jet velocity for this air curtain and a CFD model was used to provide a deeper understanding of the problem.

The air curtain used in the study had a very thin jet compared to the height of the door. The ratio of door opening height to air curtain jet thickness was 107, which was higher than any found in the literature.

2. Materials and methods

2.1 Test room and air curtain

Experimental studies were carried out on a cold storage room with internal dimensions 4.8 m x 5.8 m x 3.8 m (Figure 1) used in previous studies by Foster et al. [20, 21]. The room had an entrance with dimensions 1.36 m wide x 3.2 m high, which was closed off by a sliding door. The thickness of the door-frame was 0.16 m.

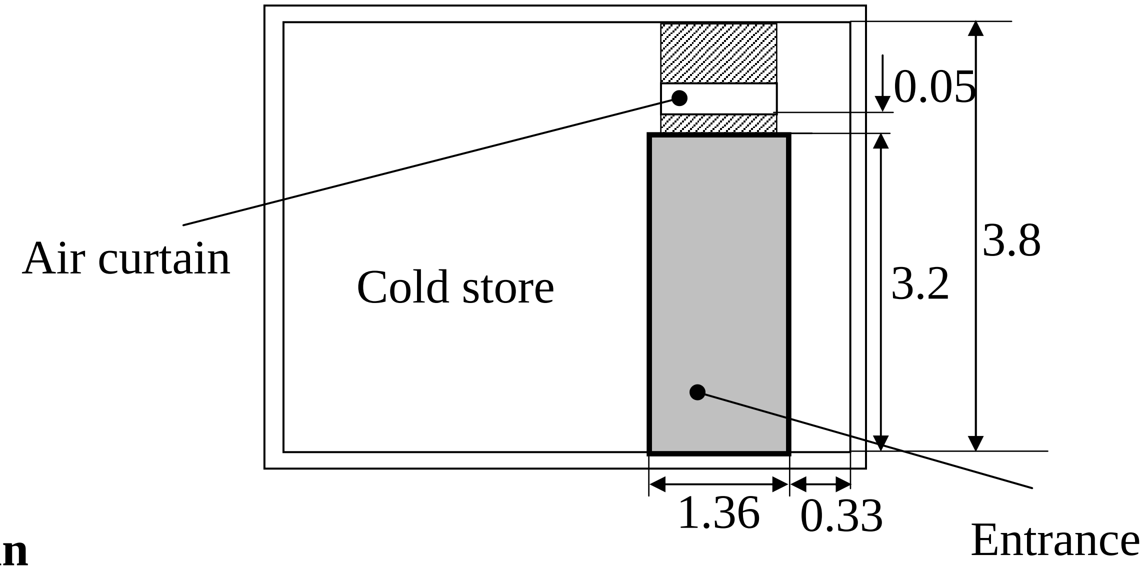
A 1.0 m long air curtain with a 30 mm slot (Model TS-40, Thermoscreens Ltd, Hants., UK) was fitted centrally above the door on the outside of the cold store. The air curtain was fitted and set up by a contractor and had controls to enable adjustment of the jet velocity and angle.

2.2 Measurements

Measurements of the jet velocity were made across the width of the air curtain duct with a hot wire anemometer (Model 1650, Thermo Systems Inc, Bristol, UK) (accuracy ±2%) whilst the door was fully open. Turbulence levels were not measured as Guyonnaud et al. [2]

showed that the initial turbulence intensity in the range of 0 to 20% had no effect on the air curtain

Vertical elevation



Plan

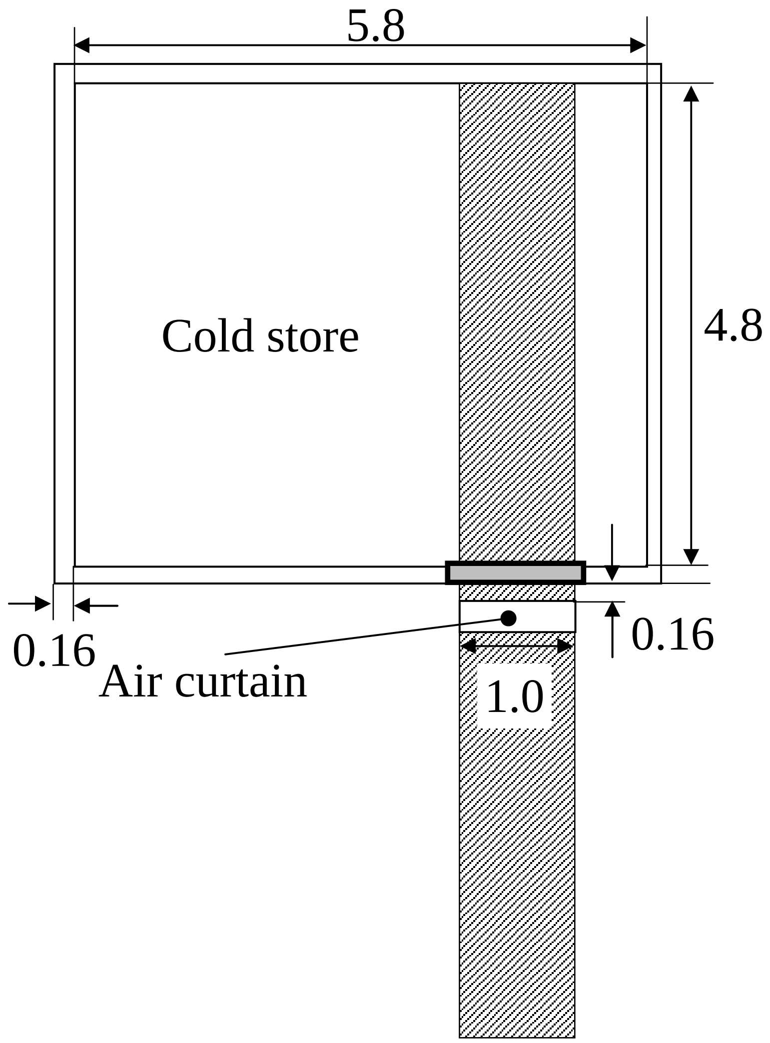


Figure 1. Vertical elevation (top) and plan view (bottom) of cold store and air curtain.

Hatched area is domain of 2-D CFD model. Dimensions in m.

performance and that commercial air curtains were in the range of 10 to 20% turbulence intensity.

Air exchange was calculated by the measured decay of an elevated CO_2 concentration in the room over time. CO_2 was released into the room and mixed using the evaporator fans to give a concentration of approximately 0.5% (5 000 ppm). All CO_2 concentrations were measured with a CO_2 infra-red analyser (Model ABPA-210, Horiba Ltd, Kyoto, Japan) (accuracy 5% of full scale).

Immediately prior to each door-opening test, the evaporator fans were switched off to allow the air movement to settle for 30 s. The door was fully opened for 10 or 30 s and then closed. The door took a total of 6±1 s to open and the same to close. All trials were carried out with an initial cold room temperature of -20°C. The air temperature outside of the cold room was between 20 and 26°C.

The concentrations of CO_2 measured immediately before and after the door opening were used to calculate the infiltration as shown in Equation 1.

$$Q = V.\ln\left(\frac{C_2}{C_1}\right) \qquad (1)$$

A door switch was fitted to start the air curtain as the door started to open and stop it just after the door closed. Experiments were carried out for different air jet velocities, ranging from the minimum to the maximum attainable air jet velocity and with air curtain switched off. Each test was replicated three times. A vertical jet angle was used for all jet velocities, and in addition the jet angle was adjusted to 10° and 20° from the vertical pointing away from the cold store, at the maximum velocity.

From these measurements the effectiveness, E, of the air curtain was measured at each condition, defined in Equation 2.

$$E = \frac{Q_b - Q_a}{Q_b} \qquad (2)$$

2.3 Analytical model

Hayes and Stoecker [16] have presented an analytical model to assist in the design of air curtains (Equation 3). Their model allows the calculation of the 'deflection modulus', which is the ratio of air curtain momentum to transverse forces, caused by temperature difference either side of the curtain (stack effect).

$$D_m = \frac{b.u^2}{g.H^2.\left(\frac{T_o}{T_c} - \frac{T_o}{T_w}\right)} = \frac{\rho_o b.u^2}{g.H^2.(\rho_c - \rho_w)} \qquad (3)$$

They also presented a chart showing the minimum outlet momentum required to maintain an unbroken curtain (Figure 2).

From the chart and equation it is possible to calculate the minimum jet velocity to provide an unbroken curtain. However, because this is the velocity at the borderline of stability, a safety factor is usually applied to this. They showed that heat transfer through the curtain is roughly proportional to jet velocity and therefore it is not beneficial to have too high a safety factor. The literature suggests a range of safety factors between 1.3 and 2.0 to use in this model [16].

2.4 CFD model

Although CFD is generally a three dimensional modelling tool, preliminary work had shown that creating a 3-D model to encompass the cold store and its surroundings whilst still having a small enough mesh to accurately predict the narrow (30 mm) air curtain jet required a prohibitively large amount of computer resources.

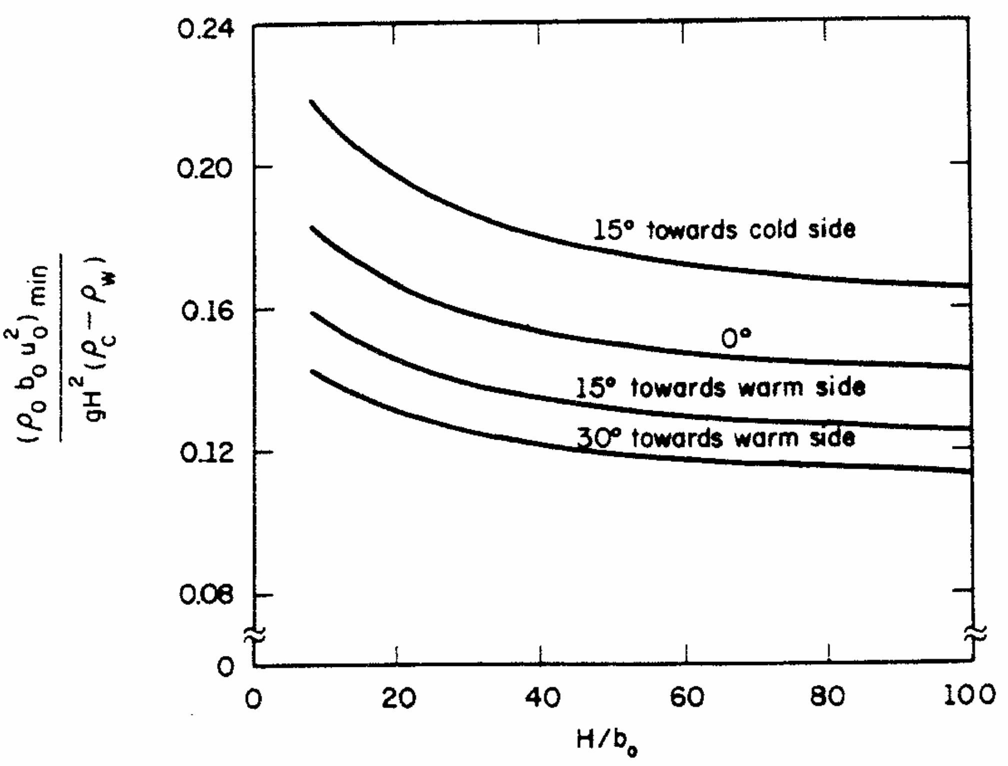


Figure 2. Minimum outlet momentum required to maintain an unbroken curtain.

A 2-D model was created using CFX 5.7.1 (ANSYS Inc., Canonsburg, USA), a general purpose CFD code. The domain of the 2-D model only covered the width of the air curtain (1.0 m) as shown by the hatched region in Figure 1. The 2-D model was created using 3-D finite volume code and by fixing only one numerical mesh cell over the width of the domain and applying symmetrical boundary conditions to the faces at either side.

The domain of the model contained the volume inside the cold store for the width of the air curtain (initially at –20°C) and a much larger volume outside the cold store (initially at +20°C), to provide a source of warm air for exchange. The volume outside the cold store was 8 times larger than the volume inside the cold store for the 'air curtain not operating' condition (jet velocity of 0) and 4 times larger for all the ‘air curtain operating’ conditions. This gave a good compromise between accuracy and numerical speed.

The jet from the air curtain was modelled as a constant velocity inlet into the domain. The velocity was varied from 0 to 18 $m.s^{-1}$. The temperature of the jet was set at 20°C, which was the initial temperature of the ambient. The level of turbulence was set to 10%, which corresponds to a high level of turbulence. An opening boundary condition was set at the top side of the air curtain, this opening removed an identical mass of air that entered from the inlet. All other boundaries not previously described, were adiabatic walls.

The numerical mesh was at its finest (8 mm) at the entrance of the door and around the air curtain nozzle. This was in order to accurately resolve the large shear created by the air jet at the nozzle exit. The total number of grid nodes for the problem was 42 000.

To enable the effects of turbulence to be predicted without implicitly modelling it, which requires a large amount of computer resources, the k-ε turbulence model was used. This two-equation model offers a good compromise between numerical effort and computational accuracy. The hardware used to run the model was a Viglen Genie PC with an Intel Pentium 4 processor running at 1.6 GHz with 1 Gb RAM.

The CFD model was used to predict the effect of cold spillage from the cold store to the surroundings and the effectiveness of the air curtain at differing jet velocities. Infiltration was calculated for different door opening durations (from 1.5 s up to 10.5 s in 1 s increments) from the predicted average temperature inside the cold store. The change in cold store temperature follows the same pattern of logarithmic decay as that in Equation 1, as shown in Equation 4.

$$Q = V.\ln\left(\frac{\Delta T}{T_w - \overline{T}_c}\right) \qquad (4)$$

From this, a dimensionless air exchange parameter, I, was defined as shown in Equation 5. An air exchange of 1.0 corresponds to the whole volume of the cold store exchanging with the outside.

$$I = \frac{Q}{V} \tag{5}$$

3. Results and discussion

3.1 Initial measurements

Trials were carried out on the air curtain, as installed and set up by the contractor, who set it to its lowest velocity and pointing vertically downwards (0° to the vertical). The jet velocity of the air curtain across the width of the duct averaged 10.5 $m.s^{-1}$. The measured effectiveness of the air curtain was 0.2 for a door opening of 10 s duration and 0.31 for 30 s.

3.2 Optimisation

Figure 3 shows the effectiveness of the air curtain over a door open duration of 30 s and for different air jet velocities at an angle of 0° to the vertical. The effectiveness of the air curtain was increased from a minimum of 0.31 with a jet velocity of 10.5 $m.s^{-1}$ to a maximum of 0.71 with a jet velocity of 18 $m.s^{-1}$, the highest jet velocity obtainable.

Figure 4 shows the effectiveness of the air curtain for different jet angles to the vertical. Increasing the angle to 10° gave the best effectiveness, increasing the effectiveness at 18 $m.s^{-1}$ to 0.77. Increasing the angle further, to 20° slightly reduced its effectiveness.

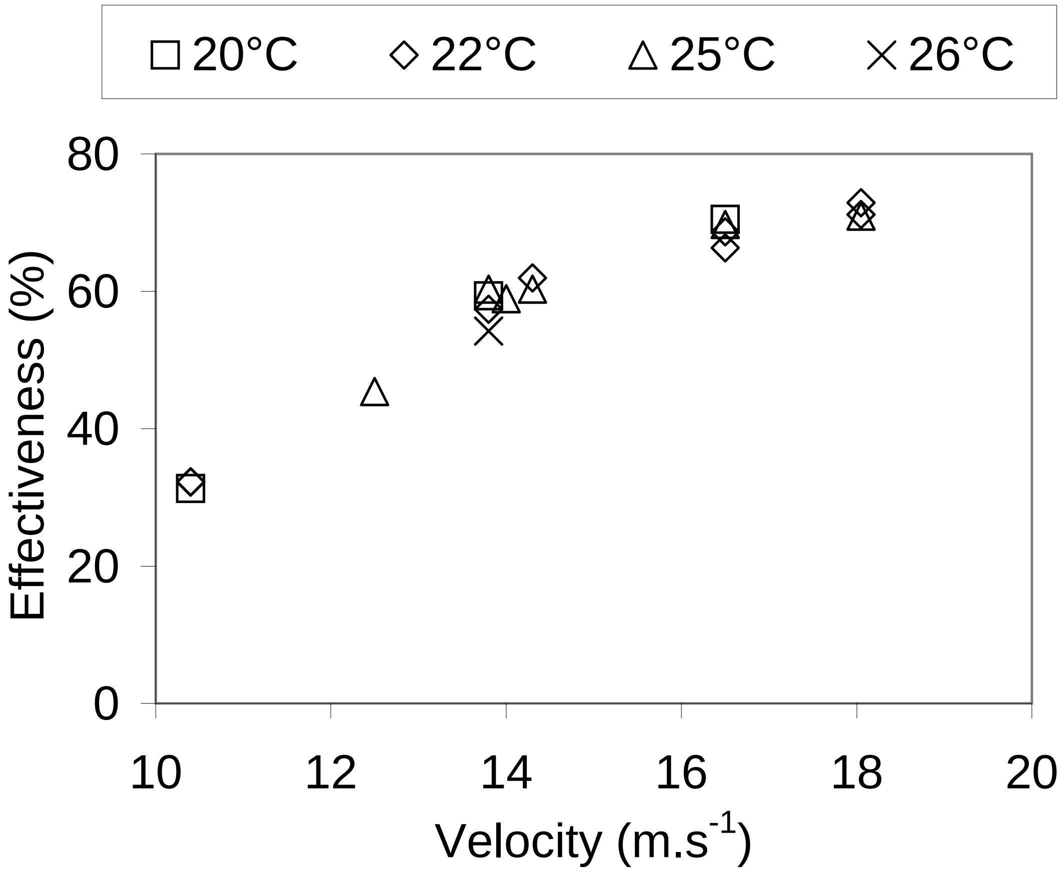


Figure 3. Effectiveness of the air curtain for different air curtain jet velocities at a door open time of 30s and jet angle of 0°. The measurements were taken at different ambient temperatures (shown in the legend).

3.3 Analytical predictions

From equation 3 the deflection modulus was calculated as 0.24 for the air curtain with a jet velocity of 10.5 m.s^{-1}. By slight extrapolation of the curve for an angle of 0° in Figure 2 it was calculated that the minimum discharge momentum for an air curtain with a door height to jet thickness ratio of 107 is approximately 0.15. According to the analytical solution, the jet should be powerful enough to reach the floor.

To calculate the minimum velocity required to achieve a continuous air curtain across the entrance given a known deflection modulus, equation 3 can be re-arranged to give.

$$u_m = \sqrt{D_m . g . \frac{H^2}{b} . \left(\frac{T_o}{T_c} - \frac{T_o}{T_w} \right)} \qquad (6)$$

Applying equation 6 a minimum velocity of 8.3 m.s^{-1} was calculated. This is the minimum recommended velocity; in practice a safety factor of between 1.3 and 2.0 is commonly used. Using this range of safety factors, a number of velocities can be chosen from 11 to 17 m.s^{-1}. Using these velocities in Figure 3 results in an effectiveness of the air curtain varying from 0.37 to 0.70. A higher safety factor of 2.2 yields the best effectiveness in these tests.

From Figure 3 it can be seen that increasing the angle of the air jet up to 20^{o} (away from the cold room) should reduce the outlet momentum required to maintain an unbroken curtain. However, experimentally an angle of 10° was found to be the optimum (Figure 4).

3.4 CFD predictions

Figure 5 shows the predicted air exchange of the cold store against door open duration for different air curtain jet velocities at a jet angle of 0˚. The most effective jet velocity (lowest exchange) was dependant on the door open duration. For a door open duration of more than 13 s a jet velocity of 10.4 m.s^{-1} was the most effective. For a door open duration of between 4 and 12 s a jet velocity of 12.5 m.s^{-1} was the most effective. For the first couple of seconds of door open duration, the jets are establishing themselves. At 23 s door open duration the 8 m.s^{-1} jet becomes more effective than the 18 m.s^{-1} jet. Vector plots of the 8 m.s^{-1} jet show that it is initially drawn into the room and does not reach the floor and as door open duration time increase the jet attaches to the floor. This can be explained by the reduction in driving force caused by the air exchange deflecting the jet to varying degrees.

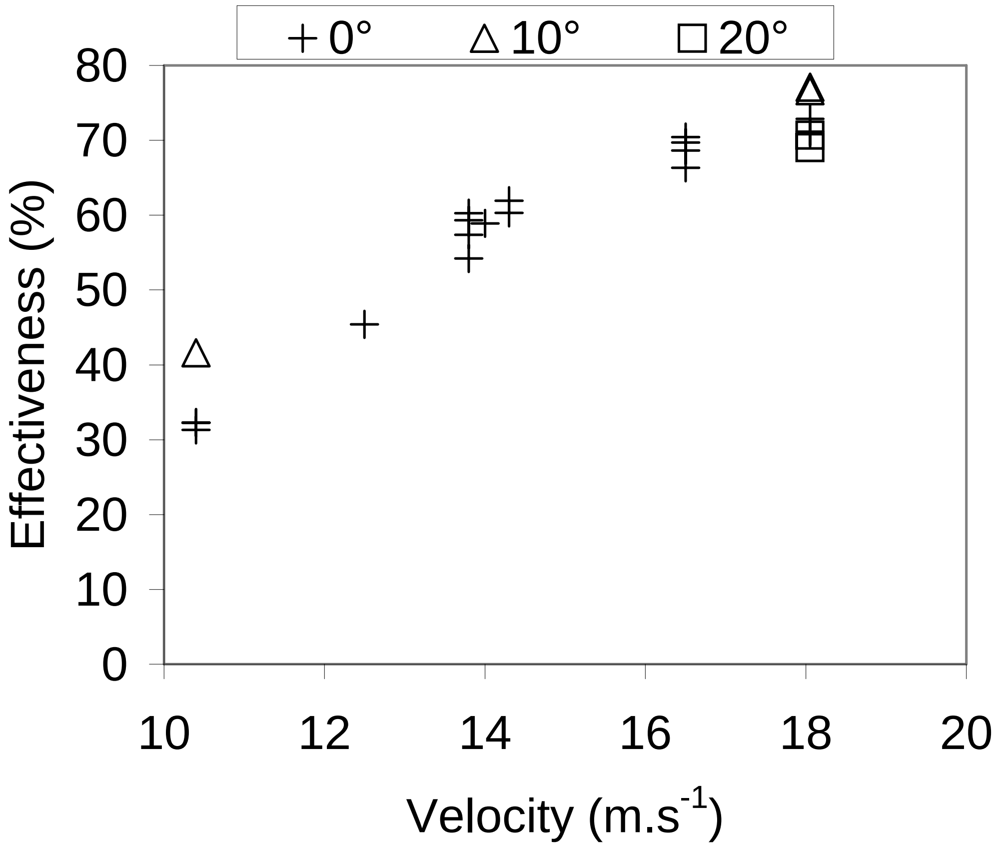


Figure 4. Effectiveness of the air curtain for different air curtain jet velocities at a door open time of 30s. The measurements were taken at different jet angles (shown in the legend).

The air exchange for each jet velocity at 30 s door open duration has been divided by the exchange for a jet velocity of zero to produce a predicted effectiveness (Figure 6). The predicted effectiveness follows the same profile as that measured by Longdil et al [22], where there is an optimum velocity where effectiveness is a maximum.

The measured effectiveness for different jet velocities (jet angle of 0°) for a door open duration of 30 s is also plotted on Figure 6. The data is incomplete in that it only shows velocities lower than the optimum velocity and therefore the maximum effectiveness was not found.

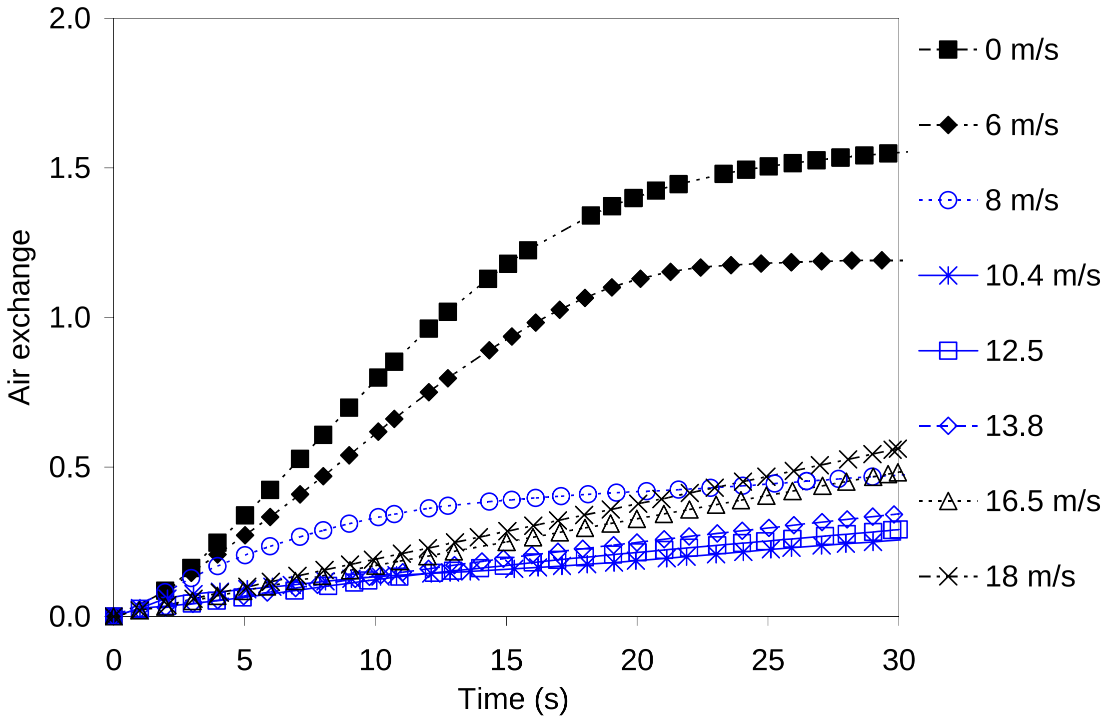


Figure 5. CFD predicted air exchange through the entrance at different door open durations and jet velocities at a jet angle of 0˚.

The predicted optimum velocity (10.5 m.s^{-1}) was much lower than that measured (>18 m.s^{-1}). The predicted maximum effectiveness (0.84) was also higher than measured (0.71), however, if it was possible that a higher effectiveness would have been measured if a higher jet velocity was possible. The differences between measured and predicted can be attributed to the fact that the air curtain did not cover the full width of the door (1.0 m wide air curtain, 1.36 m wide door), which makes the air curtain less effective than predicted by the 2-D models (which assume the air curtain is the full width of the door), as air would infiltrate into the room at the edges of the entrance.

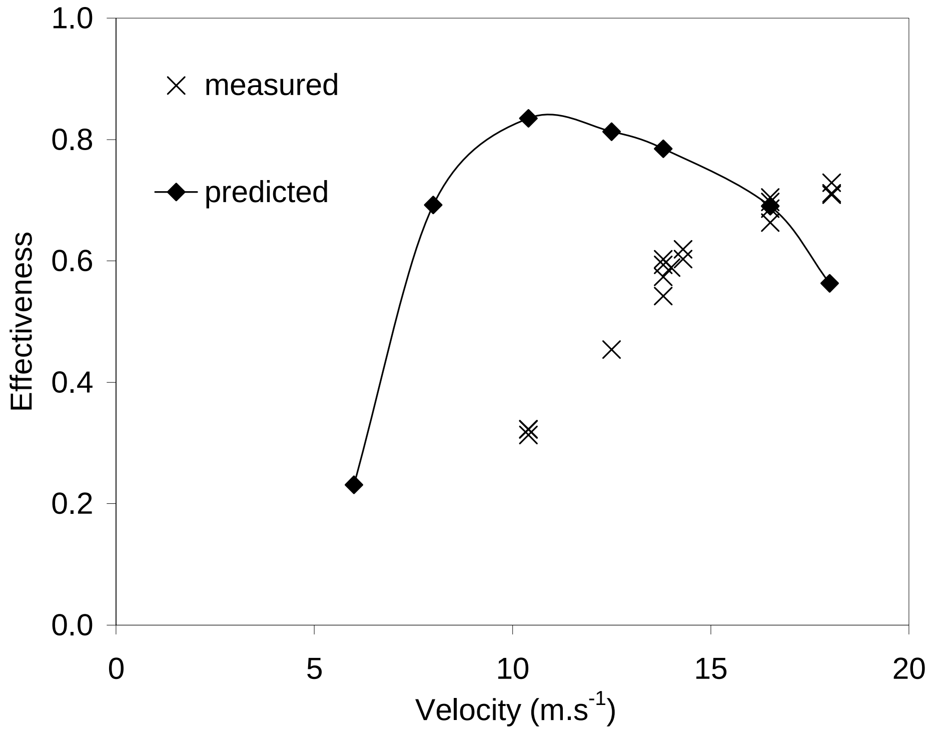


Figure 6. Predicted and measured effectiveness of the air curtain at different jet velocities

Figures 7, 8 and 9 show predicted velocity vectors and temperature contours for air jet velocities of 6, 10.4 and 18 $m.s^{-1}$, these correspond to lower than optimum, optimum and above optimum velocity, respectively. For a jet velocity of 6 $m.s^{-1}$, (Figure 7) the air curtain is bent into the cold store such that it does not reach the floor and therefore does not seal the entrance. For a jet velocity of 10.4 $m.s^{-1}$, (Figure 8) the air curtain is vertical providing optimum sealing. For the jet velocity of 18 $m.s^{-1}$, (Figure 9) the increased entrainment caused by the high velocity air curtain causes a pressure build up inside the cold store which forces the air curtain outwards reducing its effectiveness.

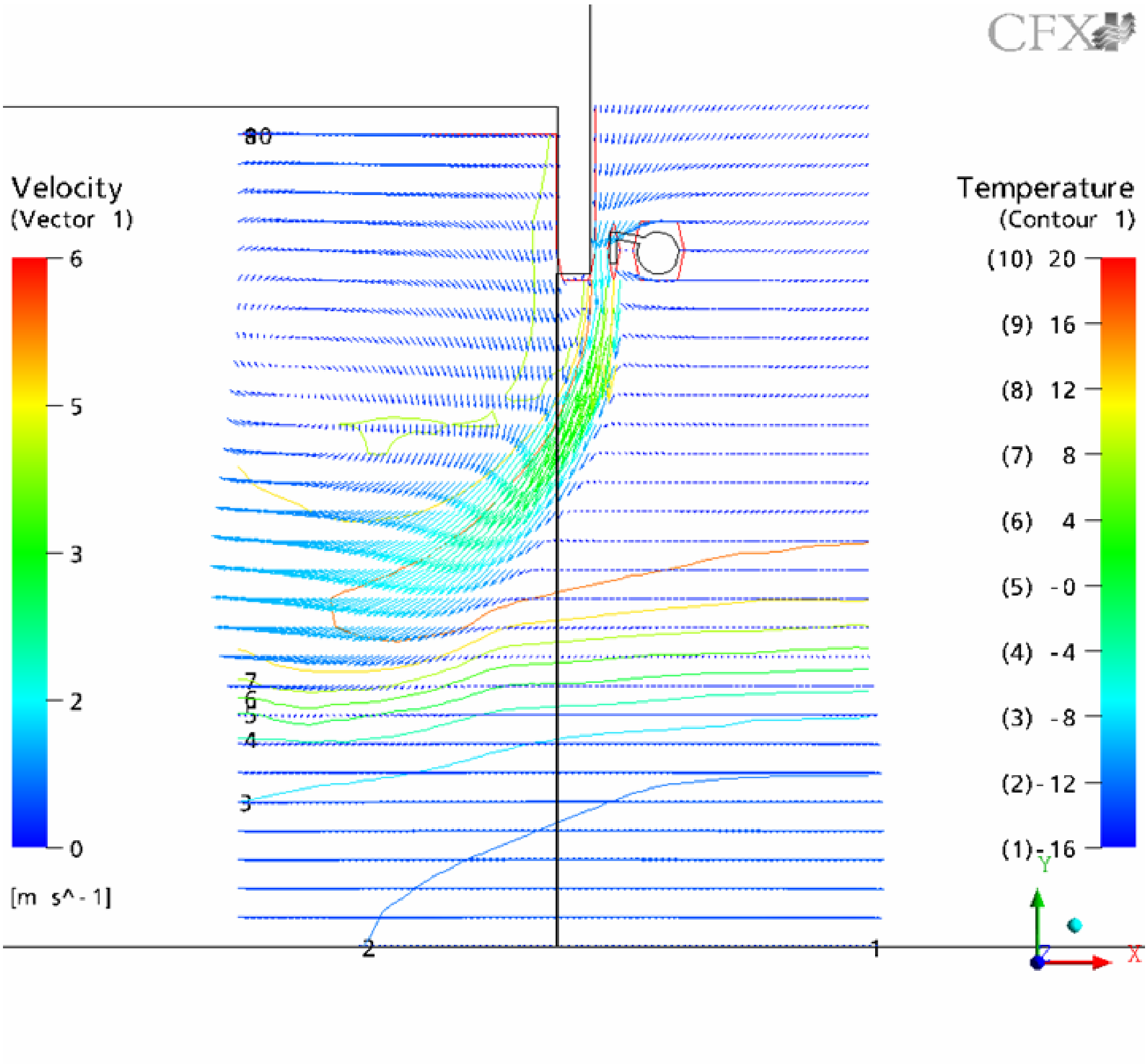


Figure 7. CFD predicted velocity vectors and temperature contours for a vertical section through the entrance with a jet velocity of 6 m.s^{-1}.

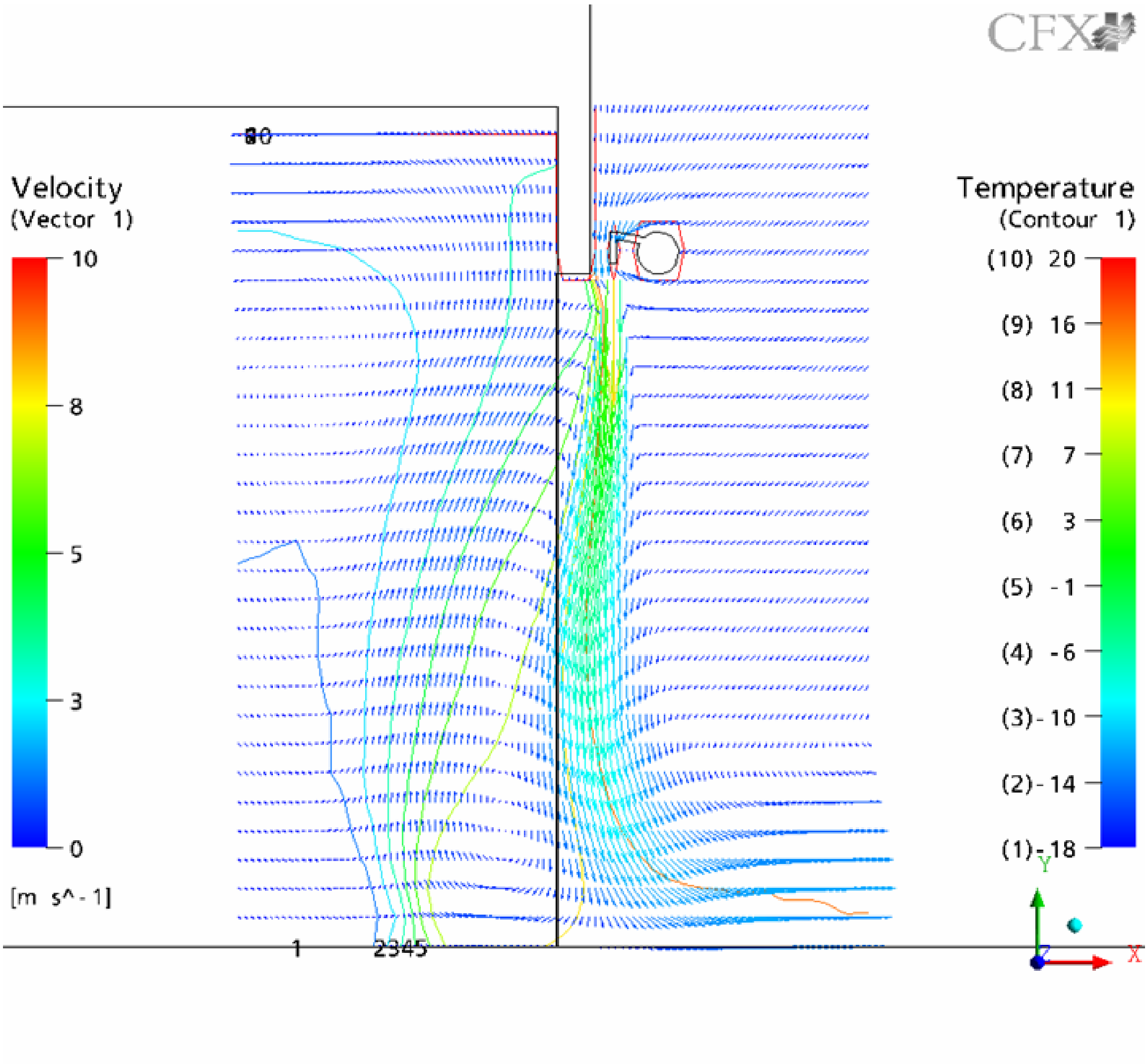


Figure 8. CFD predicted velocity vectors and temperature contours for a vertical section through the entrance with a jet velocity of 10.4 m.s^{-1}.

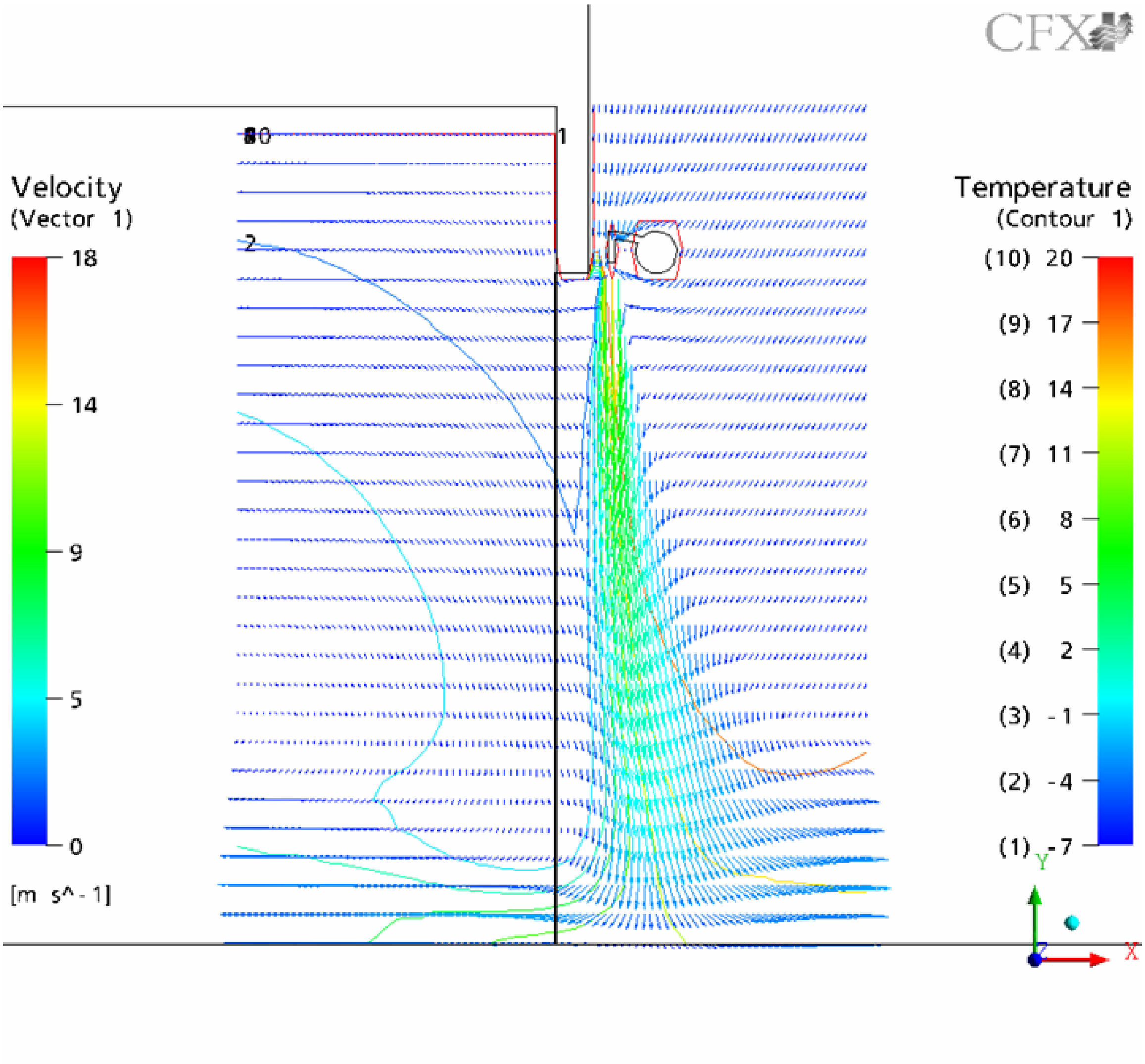


Figure 9. CFD predicted velocity vectors and temperature contours for a vertical section through the entrance with a jet velocity of 18 m.s^{-1}.

4. Conclusions

The effectiveness of an air curtain fitted over a 1.36 m wide entrance to a cold store has been measured and predicted using both 2-D analytical and CFD models. The air curtain was selected and fitted by the air curtain manufacturer and fitting contractor.

This paper has shown the importance of setting up an air curtain to give optimal effectiveness. It has been reported by Downing and Meffert [4] that air curtains are often not installed at their optimum. This paper has re-enforced those statements by demonstrating data from controlled experiments.

The analytical model gives a guide to the jet velocity; however, we have shown that with the specific air curtain used, the effectiveness can vary greatly, depending on which velocity factor was chosen. The analytical model is unable to give any guide on the effectiveness of the air curtain at the different jet velocities.

The 2-D CFD model was unable to accurately predict the optimum jet velocity. However, it predicted a value equivalent to that predicted by the analytical model with a low safety factor. The CFD model showed how the shape of the air curtain as it travelled down the entrance varied with different jet velocities and how these different jet velocities affected the effectiveness of the curtain.

The best effectiveness predicted by the CFD model was higher than that measured. However, the limited maximum jet velocity of the fitted air curtain limited the measured results. The best predicted values of effectiveness were found to be at the upper end of the range of measured values found in the literature [4]. Further work reported in a subsequent paper, reduced the experimental door width to 1.0 m so that the air curtain covered the full width of the door; the effect of this was to increase the effectiveness from 0.71 to 0.79. This brought the maximum measured effectiveness closer to the predicted (0.84).

2-D models such as the analytical model and the CFD model used in this study will most likely predict higher effectivenesses than measured values as end effects will allow air to leak around the edge of the air curtain, causing increased infiltration. As air curtains are sold in incremental widths it is possible that doors that fall in-between these widths will have air curtains that do not cover the full width of the door. Higher jet velocities will to some extent negate this problem and this is a probable explanation for the reduced effectiveness and increased jet velocity of the real air curtain compared to the 2-D CFD model. This perhaps also explains why such a large range of safety factors exists for the analytical model.

A 3-D model would probably provide greater accuracy as the infiltration at the edges of the door would be more accurately modelled. The authors intend to investigate this, as further advances in computing power allow.

5. Acknowledgements

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6. Nomenclature

b	thickness of air jet, m
C	concentration of CO_2 in the room, %
D_m	deflection modulus, dimensionless
E	Effectiveness, %
g	acceleration due to gravity, 9.81 $m.s^{-2}$
H	height of entrance, m
I	Air change, dimensionless
Q	Infiltration, m^3
T	Temperature of the air jet, cold room and ambient, °C
$\overline{T}$	Predicted average temperature, °C
ΔT	Initial temperature difference between cold store and ambient, °C
u	jet velocity, $m.s^{-1}$
V	volume of air within the room, m^3

Greek

ρ	Density, $kg.m^{-3}$

Subscripts

a	with air curtain
b	without air curtain

c	cold room
o	supply air jet
w	ambient
1	initial
2	final

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