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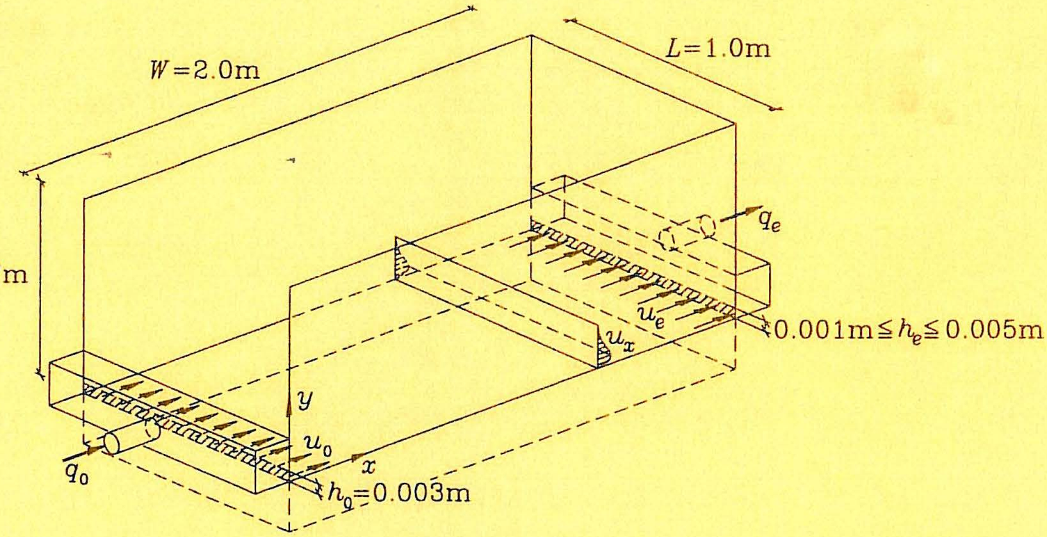

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Removal of Airborne Contaminants from a Surface Tank by a Push-Pull System

by

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ABSTRACT

Open surface tanks are used in many industrial processes, and local exhaust systems are often designed to capture and remove toxic fumes diffused from materials in the tanks prior to their escape into the workplace environment. The push-pull system seems to be the most efficient local exhaust system, but proper design is required to ensure health and safety of the workers and, furthermore, it is very desirable from an energy conservation point of view to determine an optimum and an efficient design of push-pull hoods which can exhaust all contaminants with a minimum quantity of volume flow.

The paper describes and discusses different design methods and compares designed values with results from a measurement series of push-pull system efficiency.

The measurements showed that the balance between supply and exhaust air flow rates and the level of supply air momentum were very critical for a well-working push-pull ventilation system with high exhaust efficiency.

Different design methods were compared with the measured situation. The ACGIH design method resulted in a well-working and very safe system with regard to exhaust efficiency, but also in a relatively high energy consumption and liquid evaporation rates. The Skistad design method also resulted in a well-working system but allowed an optimum and an efficient design of push-pull systems from both the workers' safety and an energy conservation point of view.

INTRODUCTION

Open surface tanks are used in many industrial processes and are quite common for processes such as pickling, etching, and plating. If proper industrial ventilation for removal of contaminants from the working environment is not provided, workers in the vicinity of these processes might experience irritation due to toxic fumes diffused from materials in the open surface tanks. For this purpose local exhaust systems are designed to capture and remove process emission prior to its escape into the workplace environment.

In the case of open surface tanks the push-pull system seems to be the most efficient system, but proper design of the push-pull hoods is required to ensure health and safety of the workers as well as to maintain the integrity of the associated products. In addition, it is very desirable, from an energy conservation point of view to determine an optimum and an efficient design of push-pull hoods which can exhaust all contaminants with a minimum quantity of volume flow. This type of design will reduce heat loss/gain to the space, the evaporation of tank liquid as well as minimise the fan power required.

There exist several design methods and the differences in system design resulting from these methods, the uncertainty on which method gives the best results as well as examples of inefficient systems initiated this work. The paper describes and discusses different design methods and recommendations and conclusions are based on a measurement series of push-pull system efficiency.

PUSH-PULL VENTILATION

A push-pull system consists of a push nozzle and an exhaust hood each running in the full length of a tank. Air is blown from the nozzle creating a plane jet which travels the width of the tank and will evenly sweep the entire tank liquid surface. Control is accomplished primarily by the push jet, while the principle function of the exhaust hood is to receive and remove the contaminant laden push air jet. Ambient air is entrained in the push jet which results in a jet flow rate at the exhaust hood several times greater than the push nozzle flow rate. If the push nozzle is located near the tank edge the jet will stick to the liquid surface and form a wall jet. If there is a certain distance between the nozzle and the liquid surface there will exist a recirculating flow below the air jet, see figure 1. However, entrainment air will only come from the surroundings in both cases.

The jet velocity decays with distance from the nozzle and the entrainment of air in a jet is directly proportional to its momentum, which can be related to the product of the nozzle supply air flow and the nozzle exit velocity. While large entrainment is desirable the exhaust volumetric flow rate must be sufficient, otherwise contaminated air will spill over and enter the workplace.

The advantage of a push-pull system is the fact that a push jet will maintain velocity over large distances, whereas the velocity in front of an exhaust hood decays very rapidly as the distance from the hood increases. Push-pull systems can therefore carry contaminants over relatively long distances into the exhaust hood, thus providing control where it otherwise may be difficult or impossible. The disadvantages are that large objects disrupt the air jet when they are lowered into the tank or removed from it. The air jet increases the turbulence at the liquid surface and causes an increase in the evaporation of tank contents and the heat loss from hot baths. For some solvents the evaporation loss is a real economic problem.

Push-pull systems have in many cases not been too effective because of poor air supply design and a poor balance between the supply air rate and the exhaust rate. If the supply air flow is too great compared with the exhaust rate the supply air is deflected into the workplace. If the supply air rate is too low the exhaust slot works as a pull-only system and does not get the benefit of the air supply.

Momentum, velocity decay and volume flow rate in a plane wall jet can be expressed by:

$$I_o = \rho_o \cdot h_o \cdot u_o^2 \tag{1}$$

$$u_x = C \cdot \sqrt{\frac{I_o}{x \cdot \rho_o}} \tag{2}$$

$$q_x = 0.147 \cdot u_x \cdot x \tag{3}$$

$$I_x = 0.109 \rho_x \cdot u_x^2 \cdot x \tag{4}$$

where	I	Momentum	(N/m)
	h_o	Slot height	(m)
	u	Velocity	(m/s)
	ρ	Density	(kg/m^3)
	x	Distance from supply opening	(m)
	C	Constant	
	q	Volume flow rate	(m^3/s·m)
	o, x	Subscripts referring to supply opening and a certain horizontal distance from supply opening, respectively	

EVAPORATION

The push-pull ventilation system increases velocity and turbulence at the liquid surface and thus evaporation and heat loss from hot baths. With evaporation of water as an example a theoretical calculation method for estimation of evaporation from a liquid surface is described in the following in the case of push-pull ventilation.

Following from Fick's law of diffusion and the perfect-gas equation the mass flux per unit area caused by evaporation can be found from equation (5), (Holman 1989)

$$\frac{\dot{m}_w}{A} = h_D \frac{M_w}{R_o}\left(\frac{P_w}{T_w} - \frac{P_\infty}{T_\infty}\right) \tag{5}$$

where	$\dot{m}_w$	Mass flux per unit time	(kg/s)
	A	Water surface area	(m^2)
	h_D	Mass-transfer coefficient	(m/s)
	M_w	Molecular weight of water ($M_w = 0.01802$)	(kg/mol)
	R_o	Universal gas constant ($R_O = 8315$)	(J/mol K)
	P_w	Partial pressure of water vapour at water surface	(Pa)
	P_∞	Partial pressure of water vapour in air	(Pa)
	T_w	Absolute water temperature	(K)
	T_∞	Absolute air temperature	(K)

In the case of forced turbulent flow along a flat plate ($5 \cdot 10^5 < \mathrm{Re}_x < 10^7$) the mass-transfer coefficient can be found from equation (6), (Holman, 1989).

$$h_D = 0.0296 u_\infty \mathrm{Re}_x^{-1/5} \mathrm{Sc}^{-2/3} \tag{6}$$

where	u_∞	Free-stream velocity	(m/s)
	Re_x	Local Reynolds number ($\mathrm{Re}_x = u_\infty x/\nu$)	
	Sc	Schmidt number ($\mathrm{Sc} = \nu/D = 0.6$)	
	ν	Kinematic viscosity of air ($\nu = 15.69 \cdot 10^{-6}$, $T = 300$ K)	(m^2/s)
	D	Diffusion coefficient for water in air ($D = 2.56 \cdot 10^{-5}$)	(m^2/s)

Equation (6) applies to conditions with a forced convection boundary layer with a constant free stream velocity. In the case of push-pull ventilation the conditions are a wall jet with a decaying maximum velocity and very low velocities outside the jet area.

By using the local maximum wall jet velocity as the free stream velocity and by substituting equation (2) into equation (6) and integrating the equation over the length of the water surface the average water evaporation rate becomes:

$$\frac{\bar{\dot{m}}_w}{A} = 0.074 \cdot C^{4/5} \nu^{1/5} Sc^{-2/3} \left(\frac{I_o}{\rho_o} \right)^{2/5} \frac{M_w}{R_o} \left(\frac{P_w}{T_w} - \frac{P_\infty}{T_\infty} \right) L^{-3/5} \tag{7}$$

LABORATORY SET-UP

The test set-up in the laboratory consists of a vessel filled with water with the dimensions L × W = 1 m × 2 m, see figure 2. The vessel is placed on top of a precision balance with an accuracy of ± 0.001 kg. Supply and exhaust of the push-pull system are located at each short end of the vessel. The supply consists of a supply plenum with the dimensions 0.2 m × 0.2 m with a horizontal slot with a height of h_o = 0.003 m. The exhaust consists of a plenum with the same dimensions with an adjustable horizontal slot where the height can be changed between 0.001 m < h_e < 0.010 m. Both slots are placed close to the water surface. To ensure two-dimensional flow conditions and to avoid disturbances from the surroundings walls are placed at the two other edges of the vessel. A 0.5 m high flange is mounted above the exhaust.

Supply and exhaust air flows could be varied and measured independently. Supply and exhaust air temperatures as well as air temperature above the water surface were measured with thermocouples. Water temperature and heat supply to the water were measured as well as evaporation rate by the vessel weight loss. The humidity was measured in the supply, the exhaust and in the surrounding air above the vessel. The system exhaust efficiency was found as the relation between the increase of water content in the exhaust air and the measured evaporated amount of water by the precision balance.

MEASUREMENT RESULTS

The measurement results include push-pull system efficiency as a function of push jet momentum and exhaust air flow, and average evaporation rate as a function of push jet momentum and vapour pressure difference.

Push-Pull System

Figure 3 shows the exhaust efficiency of the push pull system in a case with a constant exhaust air flow rate of q_e = 400 m^3/h, and supply air flow rates varying from q_o = 0 m^3/h to q_o = 60 m^3/h corresponding to a variation of the supply air momentum from 0 - 0.11 N/m. The push jet flow rate depends on the supply air momentum. The calculated push jet flow rate by equation (3) is shown for three different distances from the inlet. The optimum efficiency was found when the exhaust air flow rate was equal to the air flow rate of the push jet 1.6 m from the inlet (x = 0.8 W). The results show that the exhaust slot works as a pull-only system if the supply air momentum is too low. There is no benefit of the air supply and the efficiency is too low. If the supply air momentum is too high the push jet air flow rate will be higher than the exhaust flow rate, and supply air is deflected into the workplace, which also will result in low efficiencies. In this case a minimum supply air momentum of 0.028 N/m is necessary to benefit by air supply. This corresponds to a maximum velocity of the push jet of 0.4 m/s 1.6 m (x = 0.8 W) from the inlet slot.

Figure 4 shows the exhaust efficiency of the push pull system in cases with different supply air momentum and exhaust air flow rates. The exhaust efficiency in figure 4 is shown as a function of the relative exhaust air flow which is the exhaust air flow rate divided by the calculated push jet flow at the distance x = 0.8 W. The results show that the exhaust efficiency becomes close to one when the exhaust flow rate exceeds the push jet flow rate. As it is seen in figure 5 the supply air momentum values in figure 4 correspond to supply air flow rates between 30-70 m^3/h, and the minimum exhaust air flow rates for optimum exhaust efficiency will be between 440-1050 m^3/h. The choice of supply air momentum depends on the necessary velocity level in the push air jet. For conditions with calm surroundings ($u_\infty < 0.15$ m/s) the experiments showed that a velocity level of u_x = 0.4 m/s in the push jet was satisfactory, but for situations with stronger disturbances higher velocity levels will be needed.

Average Evaporation rate

Evaporation from the liquid surface will depend on the velocity and turbulence level of the push air jet. Figure 6 shows the measured average evaporation rate as a function of supply air momentum and vapour pressure difference. It shows that both parameters have a large impact on the evaporation rate. As the vapour pressure difference, which greatly depends on the liquid temperature, is often process dependent the only way to reduce evaporation is to decrease the velocity level. The predicted evaporation rate by equation (7) is shown as lines in figure 6. The correspondence between measured and predicted evaporation rates is very good. In the two cases with the lowest supply air momentum the Reynolds number is below the acceptable range for equation (7), and also the momentum levels are below the limit (I_o = 0.028) for a well-working push-pull system found for this configuration.

COMPARISON OF DESIGN METHODS

Several design methods for push-pull systems are available and it can be difficult to estimate the differences in system design which are a result of these methods. In the following three different design methods are described, and a comparison is made based on the measurement case configuration.

The starting point of the design method described by Skistad (1995) is a recommended minimum velocity level in the push jet along the vessel. Skistad recommends that the maximum velocity in the push jet does not come below u_x = 0.7 m/s and states that the minimum velocity level will occur at 70-80 % of the vessel width. In order to be sure that the exhaust flow is higher than the air flow in the jet, the exhaust flow is designed to be 30-40 % higher than the push jet flow calculated by equation (3) at the distance x = 0.8 W from the inlet. The necessary supply air momentum is calculated by equation (4) at the distance x = 0.8 W, and by recommending a supply air velocity between 5-10 m/s the slot height can be estimated by equation (1) and hereby the supply air flow rate can be found. The minimum velocity of u_x = 0.7 m/s in the push jet will in the measurement case occur 1.6 m from the inlet. Table 1 shows the calculated design values for air flows and slot height for the measurement case configuration.

ACGIH (1995) recommends slot heights between 0.003 - 0.006 m (1/8"- 1/4"). The necessary supply flow rate can be calculated from equation (8) and the exhaust flow rate from equation (9), (ACGIH, 1995 and Hughes, 1986). The designed values in the measurement case configuration are also shown in table 1.

$$q_o = 0.675\sqrt{h_o} \qquad (8)$$

$$q_e = 0.381W \qquad (9)$$

Sørensen (1996) recommends a supply air velocity of u_o = 5-10 m/s and that the supply air flow rate can be calculated from equation (10). The exhaust flow rate is 4-5 times the supply air flow rate. The designed values in the measurement case configuration are shown in table 1.

$$q_o = 0.04W^{0.57}\left(\frac{h_o}{0.005}\right)^{½} \qquad (10)$$

Table 1. Design values for a push-pull system calculated by three different design methods.

	Skistad 1995	**ACGIH 1995**	**Sørensen 1996**
q_o (m^3/h·m)	36 - 63	133 - 188 (8)[1]	254 - 508 (10)[1]
u_o (m/s)	5 - 10	8.7 - 12.3	5 - 10[2]
h_o (m)	0.001 - 0.003 (1)[1]	0.003 - 0.006[2]	0.007 - 0.028
q_e (m^3/h·m)	800 (3)[1]	2743 (9)[1]	1142 - 2285
$I_o = I_x$(N/m)	0.103 (4)[1]	0.549	0.850
u_x (m/s)	0.7[2]	1.6	2.0
q_x (m^3/h·m)	593	1355	1693

All three design methods are based on constant supply air momentum, which in the Skistad method can be changed according to a desired minimum velocity value, while it is constant in the two other methods. The measurement results for optimum efficiency of the push pull system fit very well the design values of the Skistad method, both with regard to supply and exhaust air flow rates and slot height.

The ACGIH method demands a much higher supply air momentum which results in a higher supply air flow rate and higher push jet velocities. The demand on the exhaust air flow rate is twice the push jet air flow rate. This gives a well-working system with a high degree of security, but also a high energy consumption for fans and air heating.

The method described by Sørensen results in even higher supply air momentum and push jet velocities. However, the demands on the exhaust flow rate are not high enough and will result in a system where parts of the push air jet will be deflected into the workplace and the exhaust efficiencies will be poor. Only a case with a supply air velocity of u_o = 5 m/s will result in a well-working push-pull system, but with a high energy consumption.

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[1] Recommended equation to be used

[2] Starting point for design

The evaporation rate is proportional to the supply air momentum, see equation (7) and, therefore, the different design methods will also result in different evaporation rates. In fact the evaporation rate with a push-pull system designed by the ACGIH method will be twice the evaporation rate for a system designed by the Skistad method.

The measurement showed that the height of the exhaust was not very important. In the measurements it was very small, between 0.001 - 0.005 m, and therefore much smaller than the push air jet width. It was only important that the amount of exhausted air was higher than in the push air jet.

CONCLUSION

The measurement series showed that the balance between supply and exhaust air flow rates was very critical for a well-working push-pull ventilation system with high exhaust efficiency. When the exhaust flow rate was too small compared with the push jet flow rate the push jet air was deflected into the workplace and resulted in low exhaust efficiency. The optimum efficiency was achieved when the exhaust air flow rate was higher than the push jet air flow rate, and a further increase only resulted in unnecessary energy consumption.

Secondly, the measurement showed that supply air momentum was very important as a certain momentum was needed for the push air jet to be able to carry contaminants to the exhaust. When the supply air momentum was too low there was no benefit of the air supply, and the exhaust slot worked as a pull-only system. When the supply air momentum was higher than needed it only resulted in an unnecessary high evaporation rate, higher exhaust air flow rates and higher energy consumption

Different design methods were compared in the measured situation. The ACGIH design method resulted in a well-working and very safe system with regard to exhaust efficiency but due to high flow rates also in a relatively high energy consumption and liquid evaporation rates. The Skistad design method also resulted in a well-working system but in smaller flow rates and thereby lower energy consumption due to the possibility of designing the system based on expected flow conditions in the surroundings and exact vessel configuration. Therefore, Skistad's method allows an optimum and efficient design of push-pull systems from both the workers' safety and an energy conservation point of view.

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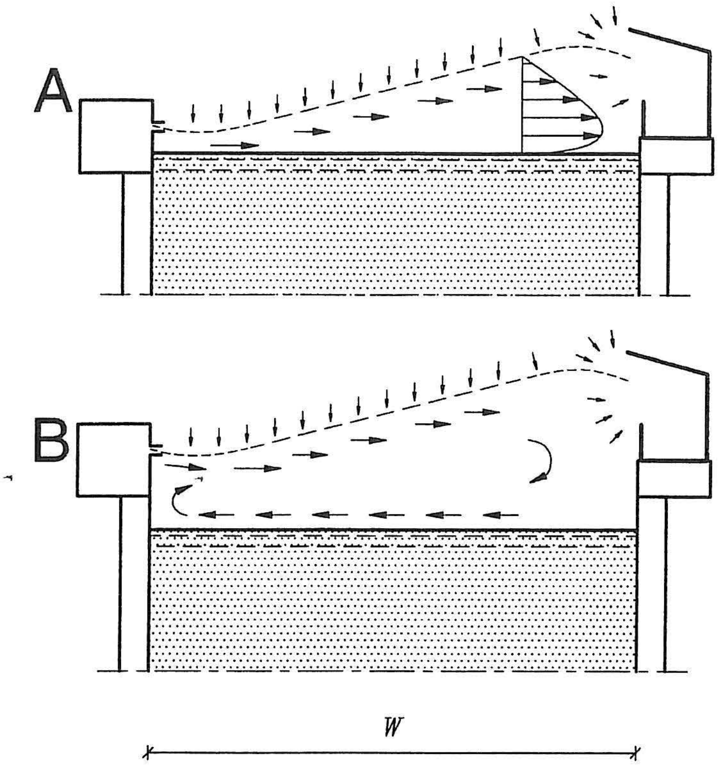

Figure 1. Outline of push-pull ventilation system principle. A) case with a wall jet along the liquid surface. B) case with recirculating flow below an air curtain.

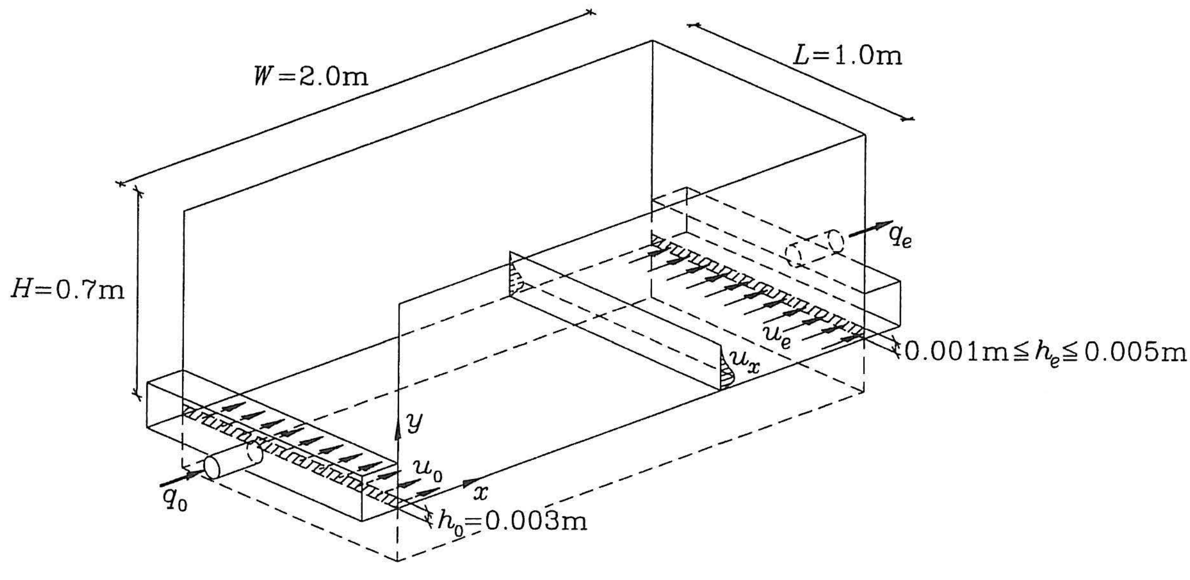

Figure 2. Sketch of test set-up for push-pull exhaust system measurements.

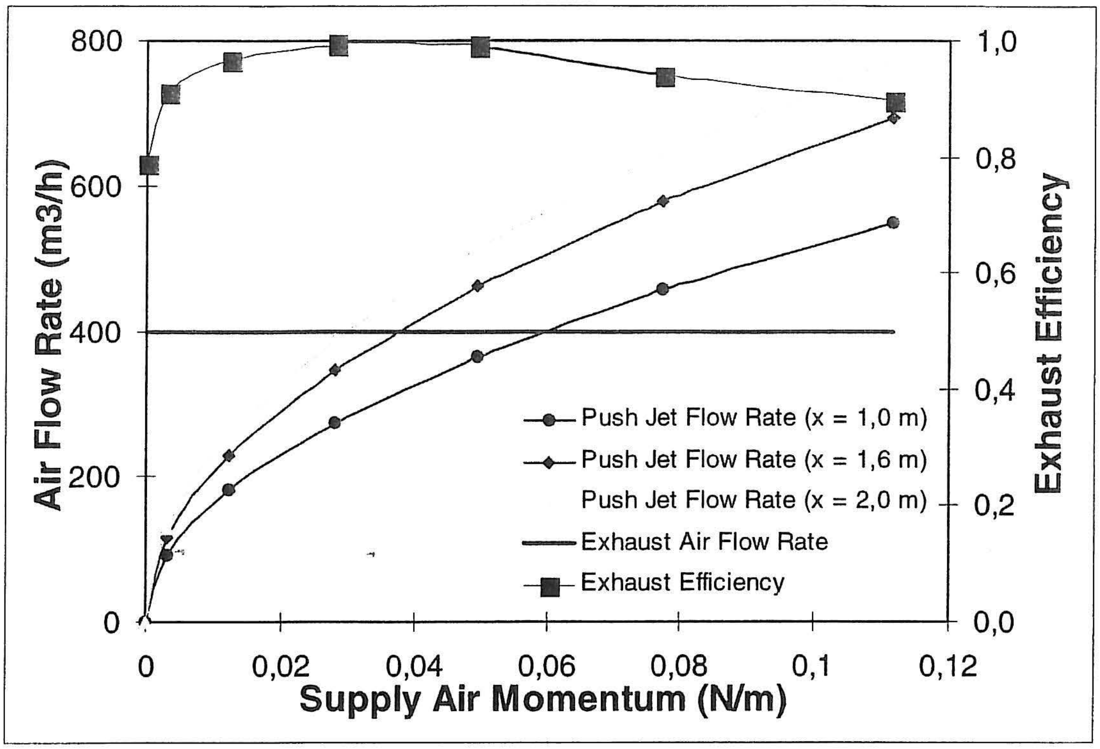

Figure 3. Exhaust efficiency at a constant exhaust flow rate of q_e = 400 m^3/h as a function of supply air momentum.

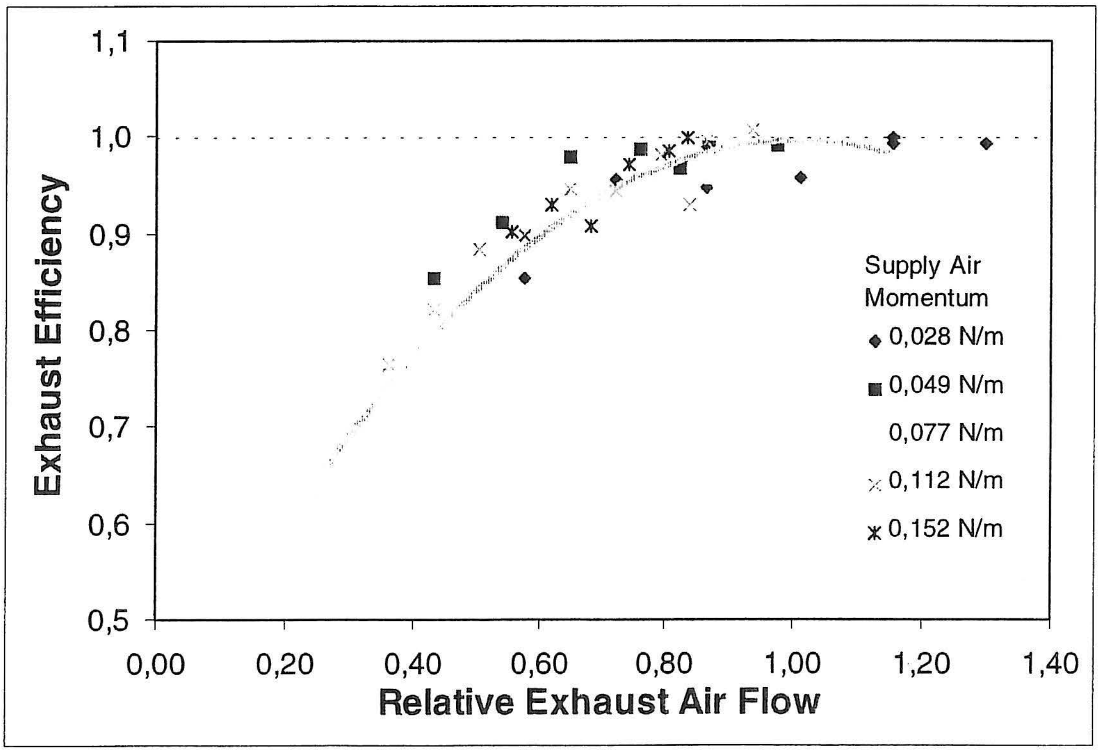

Figure 4. Exhaust efficiency as a function of the relative exhaust air flow, defined as the exhaust air flow rate divided by the push air jet flow rate.

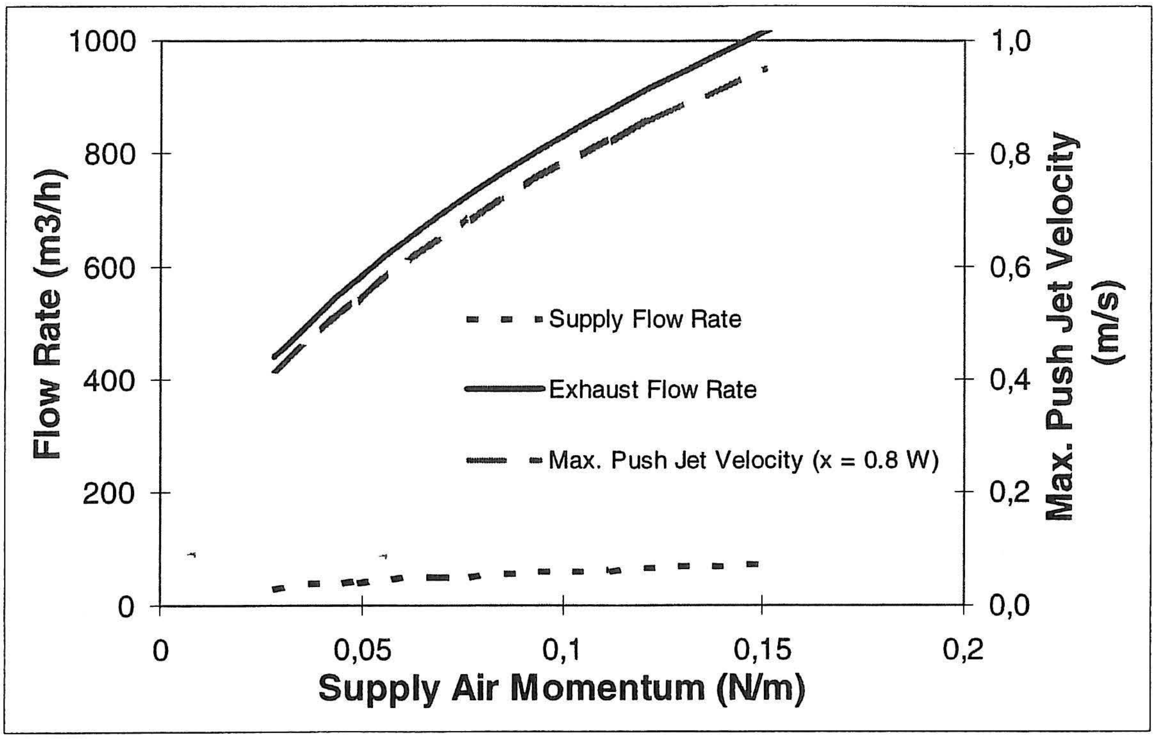

Figure 5. Measured optimum exhaust air flow rates as a function of supply air momentum. Corresponding supply air flow rate and push air jet velocity.

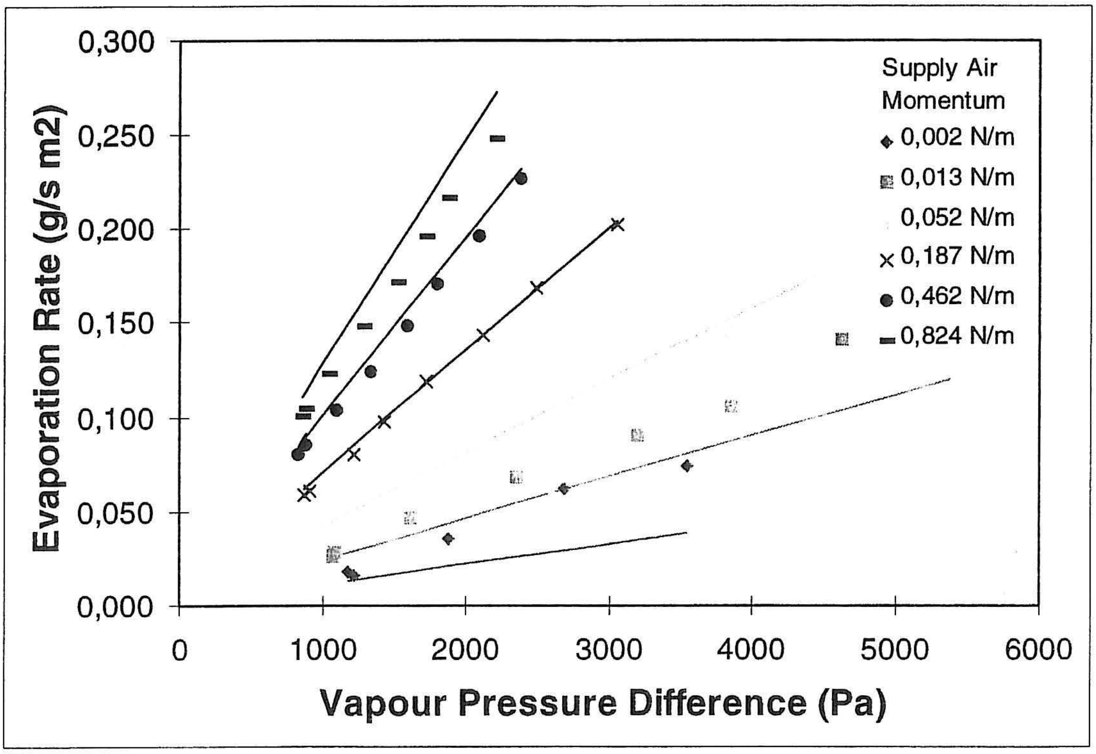

Figure 6. Measured average evaporation rates as a function of supply air momentum and vapour pressure difference (points). Calculated average evaporation rates by equation (7) (lines).

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