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Hood Entry Coefficients of Compound Exhaust Hoods

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A traditional method for assessing the flow rate in ventilation systems is based on multiple readings of velocity or velocity pressure (VP) (usually 10 or 20 points) taken in ductwork sections located away from fittings (> seven × diameters of straight duct). This study seeks to eliminate the need for a multiple-point evaluation and replace it with a simplified method that requires only a single measurement of hood static pressure (SP_h) taken at a more accessible location (< three × diameters of straight duct from the hood entry). The SP_h method is widely used for the assessment of flow rate in simple hoods. However, industrial applications quite often use compound hoods that are regularly of the slot/plenum type. For these hoods, a “compound coefficient of entry” has not been published, which makes the use of the hood static pressure method unfeasible. This study proposes a model for the computation of a “compound coefficient of entry” and validates the use of this model to assess flow rate in two systems of well-defined geometry (multi-slotted/plenum and single-slotted/tapered or “fish-tail” types). When using a conservative value of the slot loss factor (1.78), the proposed model yielded an estimate of the volumetric flow rate within 10% of that provided by a more comprehensive method of assessment. The simplicity of the hood static pressure method makes it very desirable, even in the upper range of experimental error found in this study.

Keywords compound hoods, entry coefficients, entry loss factors, industrial ventilation, local exhaust, slotted/plenum hoods

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INTRODUCTION

Local exhaust ventilation is a versatile and effective method used for controlling exposures to airborne contaminants. This method is based on the creation of air velocity at a pre-selected location to capture and air transport contaminants in a desired fashion. Air velocity is related to volumetric flow rate. Empirical relations between frontal centerline velocities and volumetric flow rate for hoods of a regular configuration have been developed and are widely used for sizing ventilation

needs.^(1,2) Conversely, the assessment of actual volumetric flow rates in operating systems gives, in some cases, a reasonable diagnosis of the ability of these systems to capture and control air contaminants.

The volumetric flow rate is seldom measured directly. Rather, it is calculated from air velocity readings (or their estimates) in a section of regular geometry. This method can be laborious since it requires a number of velocity readings for a reliable determination.⁽³⁾ In addition, velocity measurements are usually taken in ducts under conditions of “well-developed flow” to limit the effects of eddy currents and reverse flow. Well-developed flow conditions in ducts are expected at locations relatively distant from fittings or other flow disturbances. A minimum distance of seven diameters of straight duct run is usually recommended for velocity or velocity pressure measurements.⁽³⁾ This distance could be reduced when two perpendicular traverses are taken (less than 5% deviation from the results that would have been obtained under well-developed flow conditions).⁽⁴⁾ Obviously, these locations are not always available or accessible.

A simpler method for assessing volumetric flow rates is based on the measurement of the static pressure of a hood. This method is facilitated by the fact that static pressure in ducts becomes fully developed much faster than velocity (< 3 diameters of straight duct run downstream of fittings).⁽⁵⁾ In addition, when fully developed, static pressure is the same at any point of a section perpendicular to flow direction. Therefore, a single measurement (regularly obtained at a wall opening) is enough for a reasonable estimate of the static pressure value.

Static pressure measured downstream and at a close proximity (2.5 to 3.0 diameters of straight run) of a hood can be used to calculate the actual volumetric flow rate of an operating system only when the hood’s coefficient of entry is known. Coefficients of entry for simple hoods have been published and are frequently used to calculate hood flow rates and hood entry losses.⁽³⁾ Industrial applications quite often use compound hoods of the slot/plenum type. For these types of hoods, a “compound coefficient of entry” is not available. This deficiency hinders the conversion of hood static pressure into an equivalent flow rate for performance monitoring. This study proposes a model for the computation of compound coefficients of entry and validates the use of this model in systems of well-defined geometry.

METHODS

Model

The hood static pressure for simple hoods is calculated by the following expression:

$$SP_h = VP_d + F * VP_d \quad (1)$$

where SP_h is the static pressure of the hood, VP_d is the velocity pressure in the duct, and F is the hood or duct entry loss factor (dimensionless). Pressure components in Eq. 1 are regularly expressed in units of an equivalent column of fluid (regularly, water).

The coefficient of entry, C_e , represents the ratio of the actual flow rate through the hood to the ideal flow rate if there were no hood losses.⁽³⁾ C_e is calculated by the following expression:

$$C_e = \sqrt{\frac{VP_d}{SP_h}} \quad (2)$$

By replacing Eq. 1 in Eq. 2, the relation between C_e and F is obtained:

$$C_e = \sqrt{\frac{1}{1 + F}} \quad (3)$$

At standard temperature and pressure, the volumetric flow rate Q (m³/sec) is given by the following equation:

$$Q = 12.76 * A_d * \sqrt{VP_d} \quad (4)$$

where A_d is the cross-section area of the duct in m², and VP_d is the velocity pressure of the air flowing in the duct in cm of water.

A new expression of Q as a function of static pressure of the hood can be derived by replacing Eq. 2 in Eq. 4:

$$Q = 12.76 * A_d * C_e * \sqrt{SP_h} \quad (5)$$

where SP_h is expressed in cm of water.

A compound hood contains design features and accessories with a characteristic pressure drop that are used to distribute, regulate, or otherwise direct flow (e.g., rectangular entries, slotted panels, area transition pieces, and others, each of which is called a “singularity” throughout this study). For a compound hood with “n” sequential singularities and with a maximum velocity attained in the connecting duct, the hood static pressure becomes:

$$SP_h = VP_d + F_1 * VP_1 + F_2 * VP_2 + \dots + F_n * VP_d \quad (6)$$

where F_i are the loss factors for each of the singularities present in the hood. The last singularity in the series corresponds to the hood-duct connection. The contribution to the hood static pressure of the hood-duct connection is customarily calculated by the product of F_n and the duct velocity pressure, VP_d .

Equation 6 can be rewritten into the following equivalent expression:

$$SP_h = VP_d + (F_1 * \frac{VP_1}{VP_d} + F_2 * \frac{VP_2}{VP_d} + \dots + F_n) * VP_d \quad (7)$$

The application of the principle of flow continuity allows the rewriting of Eq. 7 into the following:

$$SP_h = VP_d + \left(F_1 * \left(\frac{A_d}{A_1} \right)^2 + F_2 * \left(\frac{A_d}{A_2} \right)^2 + \dots + F_n \right) * VP_d \quad (8)$$

where A_i represent the flow cross-section areas of the different singularities. A comparison between Eq. 8 and Eq. 1 yields an expression of the loss factor for a compound hood, which, in this study, is called the compound loss factor (F_c):

$$F_c = \left[\sum_{i=1}^{n-1} F_i * \left[\frac{A_d}{A_i} \right]^2 + F_n \right] \quad (9)$$

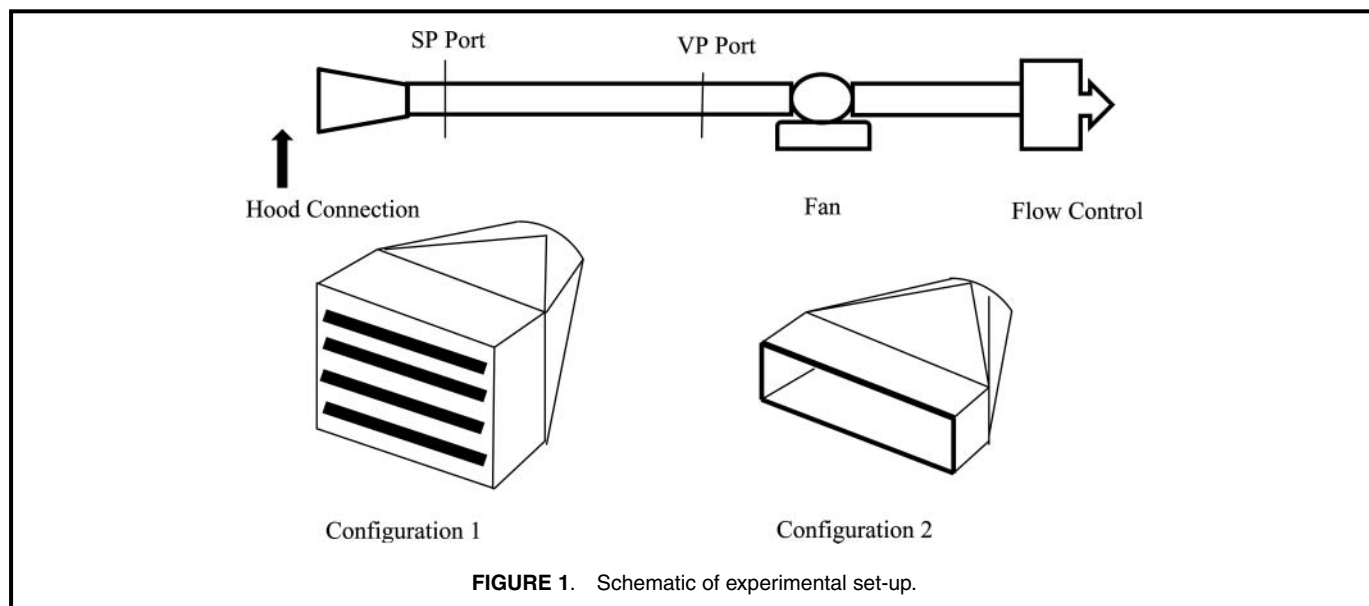


FIGURE 1. Schematic of experimental set-up.

TABLE I. Main Characteristics of Laboratory-Sized System

Components	Configuration 1	Configuration 2
Slot length (cm)	60.96	76.20
Slot width (cm)	2.54	7.62
Connecting duct diameter (cm)	20.32	20.32
Plenum cross-section area (W × H; cm ²)	76.20 × 76.20	76.20 × 7.62
Plenum depth (cm)	76.20	76.20
Slot ratio (W/L)	0.041	0.100
Number of slots	4	1
Hood taper angle (degrees)	30	30
Hood opening loss factor	1.78	0.93
Tapered transition loss factor	0.16	0.16
Duct/slot area ratio	0.52	0.559

This compound loss factor can then be entered in Eq. 3 to calculate an equivalent compound coefficient of entry (C_{ece}):

$$C_{ece} = \sqrt{\frac{1}{1 + F_c}} \quad (10)$$

Finally, the equivalent compound coefficient of entry C_{ece} given by Eq. 10 can be used in Eq. 5 for the conversion of hood static pressure into equivalent volumetric flow rate in compound hoods.

Validation of the Model

Experimental data were obtained in a laboratory-sized, local exhaust system equipped with interchangeable hoods of two different geometries (Figure 1). The hoods were made of 22-gauge galvanized sheet steel with welded joints and seams. Metal structures were built to support the hoods and reduce the mechanical stress placed on flanged connections. One hood was a multi-slotted/plenum type (Configuration 1); the other was a single-slotted/tapered or “fish-tail” type (Configuration 2). Main dimensions and characteristics of the

hoods are given in Table I. A centrifugal fan (Greenheck model SFD-9-158) provided the required operational flow rate and static pressure. A damper with a set of manually actuated, opposing blades was used to change volumetric flow rates.

Readings of static pressure were obtained on a hole drilled 2.5 diameters from the hood connection. Readings of velocity pressure or air velocity were obtained in a section located 11.0 diameters from the hood connection. The downstream section had two holes drilled 90° apart, allowing for two determinations of 10-point VP or velocity, perpendicular traverses. Velocity pressure readings were obtained when air velocities exceeded 5.0 m/s. Velocity replaced velocity pressure readings for lower velocities (<5.0 m/s). Velocity pressure was measured with a Pitot tube and an inclined fluid manometer (Series 424, Durablock®; Dwyer Instruments, Michigan City, Ind.), velocity was measured with a thermo-anemometer (CompuFlow Model 8586; Alnor Products, TSI Inc., Shoreview, Minn.), and static pressure was measured with a swinging vane anemometer (Velometer 6000AP, Alnor).

Results of a previous study that evaluated the effects of shape, size, and air velocity on entry loss factors of suction hoods⁽⁵⁾ were also used for the validation of the model proposed here. The description of hood characteristics (entry configurations and tapered transitions) allowed the selection of applicable loss factors. Loss factors and the reported area ratios were used to calculate a compound loss factor for each hood (Eq. 9). Calculated and experimental loss factors were then compared.

RESULTS

The compound loss factor and the equivalent compound coefficient of entry for Configurations 1 and 2 were calculated by using Eqs. 9 and 10 of the model, respectively. Loss factors were selected according to the geometrical configuration of the hoods (Table I) and obtained from the ACGIH® *Industrial Ventilation* manual.⁽³⁾ The calculated coefficient of entry was then used to calculate the volumetric flow rate according to Eq. 5. The calculated volumetric flow rate was compared to the volumetric flow rate obtained from two 10-point velocity pressure or velocity traverses (the reference method). The difference between the two estimates of flow rate (model-predicted minus reference) expressed as a percent

TABLE II. Summary of Model Validation Results, Configuration 1

Slotted/Plenum Hood	Damper Opening Setting			
	Fully Open	$\frac{3}{4}$ Open	$\frac{1}{2}$ Open	$\frac{3}{4}$ Closed
Average duct velocity (m/s)	18.47	14.69	6.09	2.54
Hood static pressure (cm. water)	−3.56	−2.21	−0.46	−0.08
Flow rate (reference method, L/s)	598.85	476.57	197.46	82.40
Flow rate (predictive model, L/s)	608.10	479.40	218.04	89.01
Agreement in flow rate estimate (%)	1.54	0.59	10.42	8.02

TABLE III. Summary of Model Validation Results, Configuration 2

Fish-Tail Hood	Damper Opening Setting			
	Fully Open	$\frac{3}{4}$ Open	$\frac{1}{2}$ Open	$\frac{3}{4}$ Closed
Average duct velocity (m/s)	16.86	13.21	6.09	2.40
Hood static pressure (cm. water)	-2.44	-1.55	-0.38	-0.05
Flow rate (reference method, L/s)	546.61	428.48	211.01	77.73
Flow rate (predictive model, L/s)	536.84	427.92	212.19	77.49
Agreement in flow rate estimate (%)	-1.79	-0.13	0.56	-0.30

of the reference flow rate was considered as an indication of agreement between the proposed model and the reference method (Tables II and III).

The equivalent compound coefficient of entry (C_{ece}) calculated from Eqs. 9 and 10 for Configurations 1 and 2 were, respectively, 0.779 and 0.830. These values were contrasted with the experimental value of the coefficient of entry calculated from flow rate (based on VP or velocity readings), actual hood static pressure, and duct area data (Eq. 5). The experimental coefficient of entry of Configuration 1 was 0.742 ± 0.034 and Configuration 2 was 0.834 ± 0.008 . At a level of confidence of 95%, the results of the t-test showed that the experimental and model-calculated values of the coefficient of entry were not different (Table IV).

McLoone et. al.⁽⁵⁾ calculated the values of the actual hood loss factor from hood static pressure readings at 1.0, 2.5, 5.0, and 20.0 diameters from the hood connection. The compound loss factors applicable to hood configurations described in McLoone's study were also predicted by using Eq. 9 of the model proposed in this study. Typical loss factors of hood components (singularities) were obtained from the *Industrial Ventilation Manual*.⁽³⁾ The respective flow areas of these singularities were given by McLoone. The loss factor values published at 2.5 D for slotted and rectangular hoods were then compared to the values of the compound loss factor predicted by the computational model. This comparison is

shown graphically in Figures 2 and 3, where the solid line represents a perfect agreement between the model prediction and the experimental values.

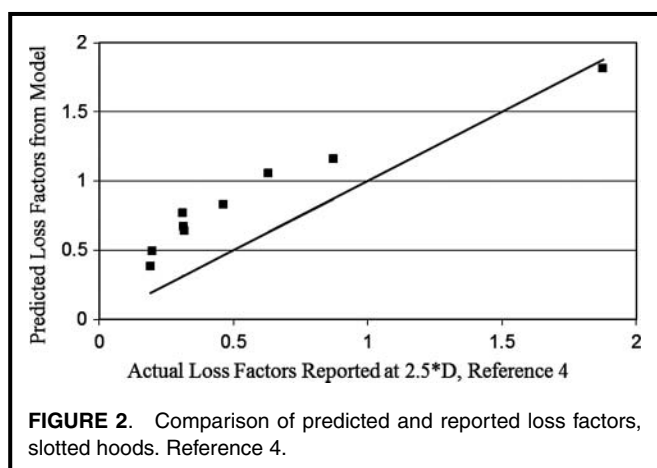
DISCUSSION

The evaluation of the results of the slotted/plenum hood (multiple and relatively narrow slots) indicates that the loss factors considered (slot loss factor 1.78 and taper loss factor 0.16) and the model proposed are suitable for this application (statistically, the calculated and experimental values of the compound coefficient of entry were not statistically different). Error and bias obtained in the results at different flow rates were possibly caused by the experimental methods used for the evaluation of velocity and velocity pressure, which have a variable accuracy across the spectrum of air velocities considered. This finding is apparent when observing the trend of the results; the disagreement between predictions became greater as the duct velocity decreased. It is well known that the accuracy of Pitot tubes' and thermo-anemometers' measurement of velocity pressure and air speed decreases as the air speed decreases.^(3,6,7)

The compound coefficient of entry of the fish-tail hood was in better agreement with the experimental results when a loss factor corresponding to a plain entry (loss entry factor 0.93) was selected. The configuration of the fish-tail does not have

TABLE IV. Statistical Comparison of Coefficient of Entry Estimates

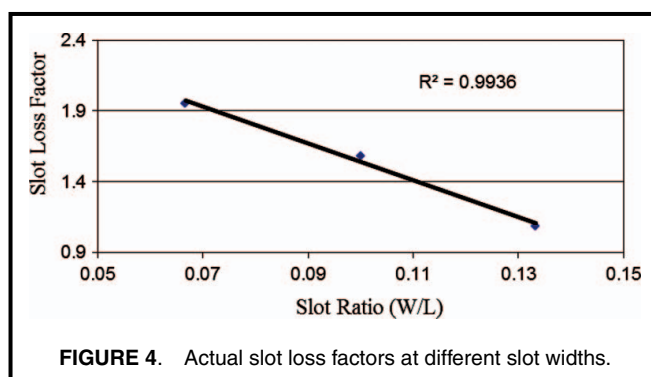
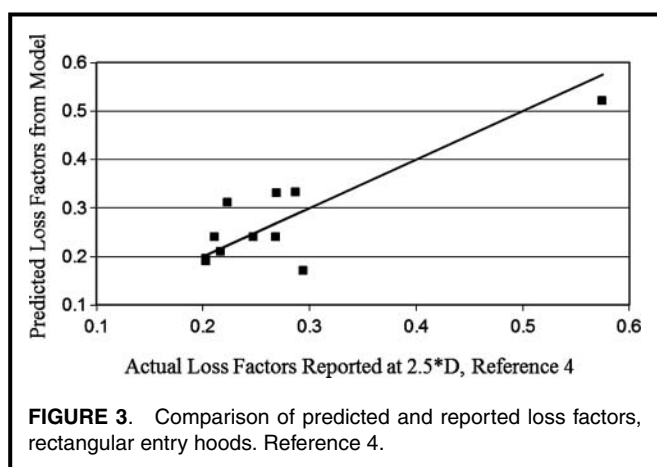
	Flow	Configuration 1	Configuration 2
C_e Calculated from experimental SP_h and VP_d	Fully Open	0.767	0.846
	$\frac{3}{4}$ Open	0.774	0.831
	$\frac{1}{2}$ Open	0.705	0.826
	$\frac{1}{4}$ Open	0.721	0.833
Average (C_e)		0.742	0.834
Standard deviation (C_e)		0.034	0.008
C_{ec} Calculated from model when applied to experimental design		0.779	0.830
t-Student statistic (calculated)		-2.178	0.838
t-Student statistic (3, 95%)		3.182	3.182
Decision (H_0 : C_e is equal C_{ec})		Accept H_0	Accept H_0



baffles perpendicular to the flow and has a plenum that guides the flow into a tapered transition resulting in less turbulence than that expected for a typical slotted plenum configuration.

The use of proper loss factors greatly contributes to the accuracy of the model. Unfortunately, entry loss factors recommended for small openings are not always available or in agreement. Hemeon⁽⁸⁾ recommended an entry loss factor of 2.78 for a sharp-edged round orifice after considering that the velocity at the vena contracta was about 1.66 times the velocity of the orifice. For the same configuration, ACGIH recommended an entry loss factor of 1.78 (pressure loss calculated in both cases as the loss factor times the velocity pressure of the orifice). For a rectangular narrow opening (slot), Hemeon recommended a loss factor of 1.3. According to the ACGIH, “the slot entry coefficient, F_s , would have a value typically in the range of 1.00 to 1.78.”⁽³⁾ Obviously this range is too broad for an accurate prediction of the entry loss.

McLoone et al.⁽⁵⁾ found that the area ratio (hood face area divided by duct section area) affects entry loss factors. They indicated that the loss factors increased for area ratios below 5 and remained stable for area ratios above 5. A changing value of the slot loss factor is also indicated in Figure 2. The slot loss factor assumed (1.78) gave a close estimate only for the narrowest slot



width they considered (4.22 cm; farthest point to the right of the plot). For other slot widths (5.2 to 15.3 cm), a loss factor of 1.78 was in excess of the experimental values reported. Figure 3 shows that for a rectangular opening, the agreement between experimental and predicted values is better. The experimental variability in this case is more of a random type. The cloud of experimental points around the prediction was possibly an indication of poor precision of the experimental method used rather than a lack of agreement between model and practice.

The hypothesis that the slot width would influence the value of the slot loss factor was demonstrated by a second experiment conducted in this research. The experimental configuration used for this demonstration consisted of a rectangular plenum (76.20 cm × 45.72 cm) with two front panels, placed perpendicular to the direction of airflow, that were adjusted vertically up and down to increase or decrease the width of a single slot. Three slot widths were evaluated (5.08, 7.62, and 10.16 cm) while the length of the slot was maintained unchanged (76.20 cm). The slot loss factor was calculated by using Eq. 8 and the experimental values of hood static pressure, average duct velocity pressure, and duct/slot area ratio for the different slot widths under consideration. The experimental method and instrumentation used for the evaluation of hood static pressure and duct velocity pressure were the same as those used in the validation of the computation model. Preliminary results of this evaluation are summarized in Figure 4 and show that a single value of the slot loss factor at different slot ratios (slot width/ slot length) is not recommended. Efforts should be devoted to refine these data and explore the effect of slot ratio and slot velocity on the slot loss factor value.

CONCLUSIONS AND RECOMMENDATIONS

The method proposed for the calculation of compound loss factors and equivalent compound coefficients of entry yielded an estimate of volumetric flow rate within 10% of that expected for a more comprehensive method of evaluation. The accuracy of the model prediction can be improved if a more accurate value of the slot loss factor is used. Instead of a single value of the slot loss factor, a transitional value seems to be more advisable for design and evaluation applications.

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