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Original Article

Research on the relationship between the centerline velocity, aspect ratio and exhaust airflow rate for a slot and a rectangular capture hood in an local exhaust ventilation system

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Abstract: When using a local exhaust hood to remove harmful substances from the production process, the exhaust airflow rate must be calculated according to the capturing velocity specified by the relevant regulations. The Numano and American Conference of Governmental Industrial Hygienists (ACGIH) equations are used in Japan and the US, respectively, for estimating the exhaust airflow rate of slot hoods. However, these equations differ from each other, and when using these equations to calculate the exhaust airflow rate of the capture hood, whether using Japan's equation or ACGIH, the hood type (slot or rectangular hood) should be distinguished at first. Therefore, this study performs experiments and a computational fluid dynamics (CFD) simulation to investigate the relationship between the centerline velocity and the aspect ratio for five types of capture hoods. The results showed good agreement between simulated and experimental centerline velocities when the distance from the hood face. A dimensionless velocity was introduced and a significant difference in the relationship between the centerline velocity and the distance from the hood face with different aspect ratios was found. A unified equation was obtained that can express the relationship between exhaust airflow rate and centerline velocity regardless of the aspect ratio of the hood face of the free-standing capture hood.

Key words: Slot capture hood, Centerline velocity, Computational fluid dynamics (CFD), Aspect ratio, Dimensionless, Local exhaust ventilation

Introduction

Local exhaust ventilation (LEV) systems have been widely studied in the field of industrial hygiene as an effective means to prevent the exposure of workers to harmful gases[1, 2] as well as dust and vapors[3–6]. The capture hood without an enclosure is one of the most widely used types of the LEV, and it is usually installed near an emission source. In general, the effectiveness of the capture hood increases with increasing airflow levels and therefore with increasing capturing velocities. The capturing velocity is a minimum hood induced air velocity necessary to capture and convey contaminants into the hood and stipulated by the relevant regulations for the various production pro-

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cesses[7]. Therefore, the capturing velocity is an important factor in the design of a capture hood.

For the free-standing capture hood, the centerline velocity is the most convenient direction to characterize the rate of change of inflow velocity with distance. In the 1930s, Dalla Valle first studied capture hoods[8]. He found that the velocity distribution in front of a capture hood is affected by the shape and size of the hood face and accordingly proposed empirical equations for centerline velocity of the hood shape, exhaust airflow rate (Q), centerline distance of the source from the hood face (X), and hood area (A). These equations were applied to plain and flanged hoods with circular and rectangular hoods, respectively. Following his recommendation, the method of using centerline velocities to evaluate the effectiveness of capture hood became widely adopted by other researchers and has remained a design standard in the Industrial Ventilation Manuals since 1951[7].

In the 1940s, Silverman studied circular and slot hoods in detail[9]. A hood face with an aspect ratio (R_a) (the ratio of the length to the width of the hood face) $L/H_{slot} > 5$ was classified as a slot hood. An equation applicable to plain and flanged slot hoods was developed. These equations have since become the design standard in the Industrial Ventilation manual. Later, Fletcher developed equations for the centerline velocity in flangeless and flanged hoods using the aspect ratio[10]. His research showed that the centerline velocity obtained from the conventional equation does not agree with the experimental value for a large aspect ratio; further, the centerline velocity decreases with the distance from the hood face and decreases more rapidly with increasing aspect ratio.

In Japan, Numano studied the equation to calculate the exhaust airflow rate of capture hoods, and the equations were widely used in the design of LEVs in Japan[11]. Equation (1) is the equation for calculating the plain slot hood used in Japan, where Q (m^3/min) is the exhaust airflow rate of the capture hood, Vc (m/s) is the control velocity, defined as the minimum velocity necessary to suck contaminants at the dispersal limit point into the hood face, X (m) is the distance from the hood face, L (m) is the length of the hood face, respectively.

$$Q = 60 \times 5.0 \times L \times X \times V_C \tag{1}$$

Equation (2) is the equation for calculating the exhaust airflow rate of the plain circular hood and plain rectangular hood, Equation (3) is for plain slot hood recommended by the American Conference of Governmental Industrial Hygienists (ACGIH)[7], where V_X is the capturing velocity, A (m^2) is the area of the hood face, Q, X, and L are the same as Japan's equations.

$$Q = 60 \times V_X \times (10X^2 + A) \tag{2}$$

$$Q = 60 \times 3.7 \times L \times X \times V_X \tag{3}$$

However, when using these equations to calculate the exhaust airflow rate of the capture hood, the hood type should be distinguished (slot or rectangular capture hood) at first whether using Japan's equations or ACGIH.

Chen Jianwu analyzed the centerline velocity change rule of the desktop capture hood by using the relationship between the centerline velocity (V), the centerline velocity of the hood face (V_0), the distance from the hood face (X), and the equivalent diameter of the hood face (d)[12]. He obtained a better change rule from this relationship of centerline velocity by using dimensionless processing which can be used without considering the aspect ratio of the desktop capture hood.

The purpose of this paper is to study the relationship between the centerline velocity, aspect ratio, and exhaust airflow rate of the free-standing rectangular and slot capture hood, to obtain a relatively simple and general expression to provide the basic design proposal for the free-standing capture hood.

Subjects and Methods

Experiment

Experiments were conducted in a laboratory free from a cross-draft. Figure 1 shows the local exhaust system used in this experiment. The duct and exhaust fan were connected to the exhaust hood via a reducer. Measured the total pressure and static pressure from the compound pitot tube inserted into the exhaust duct with a differential pressure gauge to obtain velocity pressure and then obtained the velocity in the duct. The output current from the differential pressure gauge was inputted to the inverter to control the exhaust fan to ensure that the velocity in the duct corresponded to exhaust airflow rates.

The selection of the exhaust airflow rate in this study was calculated by the Eqs.(1) and Eqs.(3) for a slot capture hood with a 0.5 length face. The capturing velocity (V_X) adopted 1 m/s for the case of the average motion of the source dispersion recommended by the Industrial Ventilation Manual and the distance from the hood opening of the capture point was chosen as 0.2 m[7]. So, the exhaust airflow rate was set to 22.2 and 30 m^3/min. Additionally, 15, 20, and 25 m^3/min were also used.

The centerline velocities were measured at different dis-

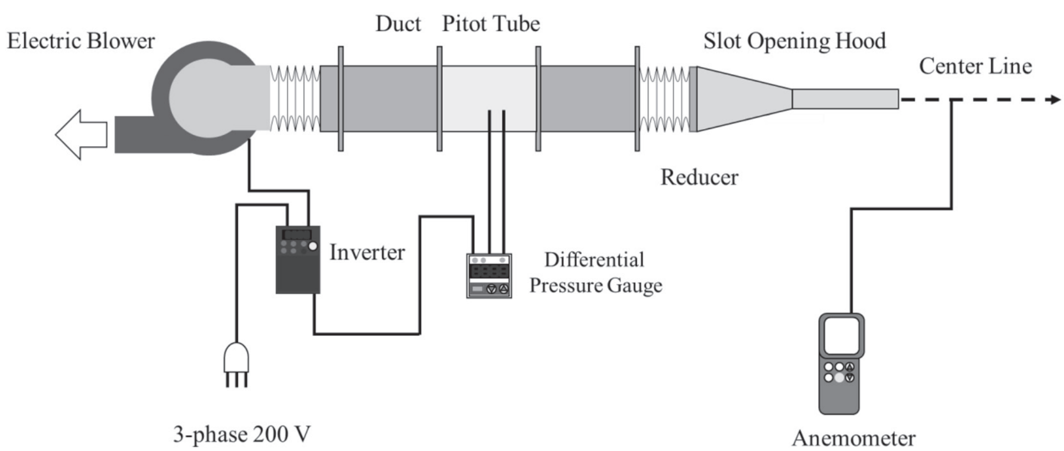

Fig. 1. Schematic diagram of the local exhaust ventilation.

Table 1. Summary of the hood type and measurement methods

Hood type	Hood size L (m) × H (m)	Aspect ratio L/H	Distance from hood opening X (m)	Exhaust airflow rate Q (m^3/min)	Measurement time of each point (min)	Sampling interval (s)	Position of measurement point with respect to opening surface (m)
Slot hood	0.5 × 0.05	10	0–0.2	15, 20, 22.2, 25, 30	1	1	0, 0.01, 0.02, 0.03, 0.04, 0.05, 0.06, 0.08, 0.1, 0.12, 0.14, 0.16, 0.18, 0.2
	0.5 × 0.075	6.67					
Rectangular hood	0.5 × 0.1	5	0–0.55				0, 0.01, 0.02, 0.03, 0.04, 0.05, 0.06, 0.08, 0.1, 0.12, 0.14, 0.16, 0.18, 0.2, 0.25, 0.3, 0.35, 0.4, 0.45, 0.5, 0.55
	0.5 × 0.15	3.33					
	0.5 × 0.5	1					

L: length; H: height.

tances from the hood face for five different capture hoods with different aspect ratios, including rectangular and slot types. The centerline velocity from the center of the hood face along the direction perpendicular to the opening surface was measured using an anemometer MODEL6543 (KANOMAX, Osaka, Japan).

In this study, the centerline velocities of the rectangular opening capture hood were measured at each point within the range of 0.55 m from the hood face. For the slot capture hood, the centerline velocities from the hood face to the point 0.2 m away from the hood face were measured. The distribution of measurement points is shown in Table 1. Each point was measured for 1 min at 1-s intervals, and the average value was used as the result of each measurement point. The experiments were conducted with different exhaust airflow rates for each hood.

The relationship between the distance attenuation of the centerline velocity and the aspect ratio of the hood face was investigated by measuring the centerline velocity with different hood faces at different exhaust airflow rates. These relationships were compared with the theoretical values using the calculating equation.

CFD

CFD has become a powerful tool for analyzing the air velocity, temperature, and distribution of various spatial pollutants in ventilation systems. In addition, compared with actual experiments with complex setup operations and equipment costs, the CFD method has the characteristics of low cost, high speed, and high reproducibility of actual scenes, and had been widely used in the environmental prediction of flow fields in recent years[13, 14]. In this study, FLUENT 18.1 (ANSYS, Canonsburg, PA, USA), a CFD software package, was used to predict the effect of the aspect ratio of five types of capture hoods on the exhaust airflow rate and the centerline velocity, and the result was compared with the experimental results to ensure that it can be used accurately. The selection of measurement points in the simulation was the same as in the experiment.

The Navier–Stokes equations were solved using the finite volume method. The flow was considered incompressible and isothermal, with constant air properties. In this study, a hood opening of the same size as the laboratory experiment was selected for analysis. A 3 m (L) × 3 m (W) × 3 m (H) room was chosen for CFD simulation. The capture hood was located 0.75 m above the ground. A duct with an opening of 0.22 m × 0.22 m connected the hood and the exhaust opening on the sidewall, as shown in Figs. 2 and 3. No other flow obstacles and workers were present in the room.

The standard wall function was applied to the hood, ceiling, floor, and wall connected to the duct. The boundary condition of the inlet velocity was determined on the exhaust opening of the duct connected to the sidewall. The inlet velocities were calculated from the exhaust airflow rate used in the experiment. Except for the back wall, floor, and roof, the pressure outlet boundary conditions of the room walls were all set to zero.

As noted above, simulations were conducted for five different opening hoods. For this purpose, an unstructured mesh with 275,248–281,256 cells was modeled. In this study, the mesh in areas where there was a large gradient in the flow vibration was refined. A coarser cell design was applied to the entire geometry. Fine cells were created for the high-sensitivity zone, and medium cell size for the surrounding sensitivity zone. The standard k-epsilon turbulence model was chosen to model the turbulence because it performed well for indoor airflows and was a simple and applicable solution for the flow under consideration. The SIMPLE algorithm was used to couple the pressure and velocity, and the second-order upwind scheme for momentum, turbulence, and species was chosen.

In the validation study, the lower residuals were suggested to achieve convergent solutions. To ensure complete convergence of the residual equations, the absolute value of the residuals of all equations was set to 10^{-5}. The absolute value of the residuals was applied for the steady-state case.

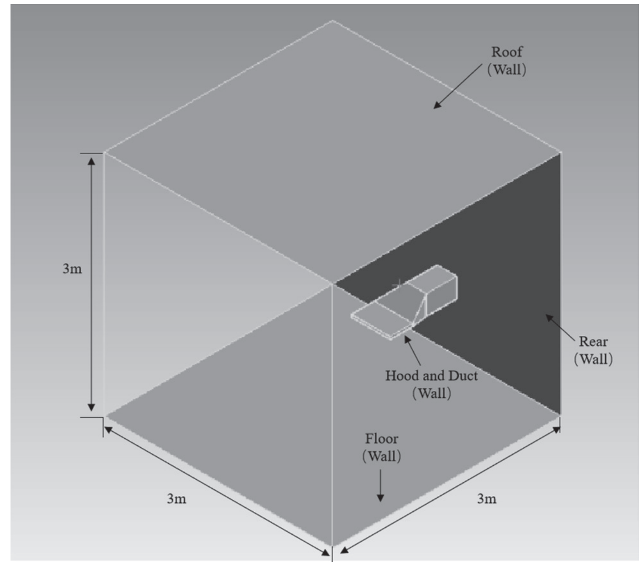

Fig. 2. Simplified computational fluid dynamics (CFD) model of the lab with Local Exhaust Ventilation (a).

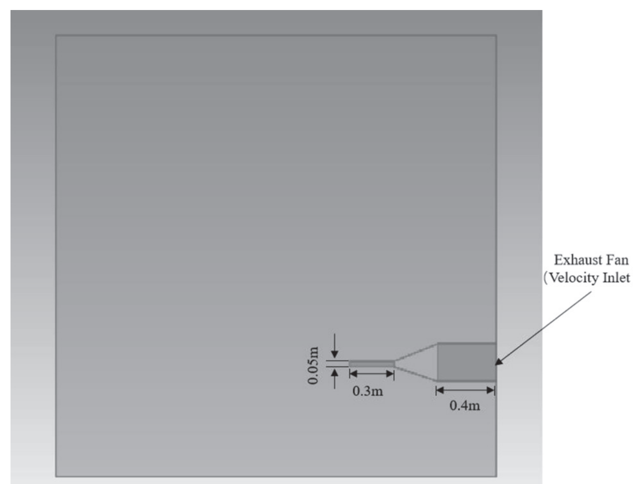

Fig. 3. Simplified computational fluid dynamics (CFD) model of the lab with Local Exhaust Ventilation (b).

Results

Centerline velocities

Figure 4 shows the relationship between the centerline velocity (V) and the distance from the hood face obtained (X) from experiments, CFD analysis, and the ACGIH equation for a free-standing rectangular hood with an aspect ratio (R_a) of 3.33 when the exhaust airflow rate was 20 m^3/min. The results of the CFD analysis and the experiments were in good agreement with those obtained from the ACGIH equation. It can also be found that starting from a position of 0.2 m from the hood face, as the distance from the hood face increases, the experimental results are more consistent with the results obtained from the ACGIH equation and CFD analysis. This shows that the rectangular hood equation is more consistent with the actual situation, which was verified through CFD analysis and experimental methods. Similar results were obtained for other exhaust airflow rates (Q).

Figure 5 shows the relationship between V and X for a hood with a Ra of 5. This is the boundary between the rectangular hood and the slot hood. As a result, the centerline velocities obtained from the ACGIH equation were consistent with the experimental and CFD analysis results at a distance greater than 0.02 m in front of the hood face, especially the experimental and CFD analysis results at distances greater than 0.2 m. However, within 0.02 m of the hood face, the experimental and CFD analysis results

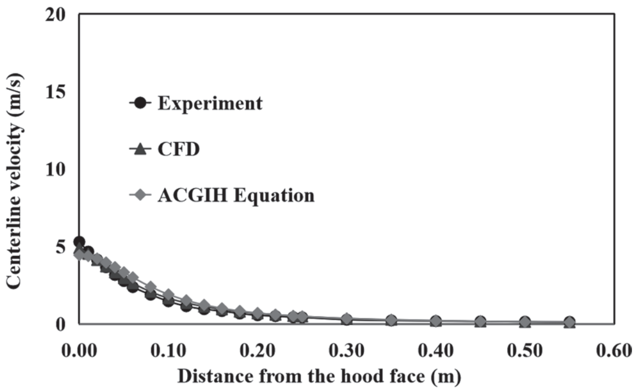

Fig. 4. Relationship between the distance from the hood face (X) and the centerline velocity (V). Rectangular hood face size: 0.5 m (L) × 0.15 m (H), aspect ratio (R_a): 3.33, exhaust airflow rate (Q): 20 m^3/min. CFD: computational fluid dynamics; ACGIH: American Conference of Governmental Industrial Hygienist.

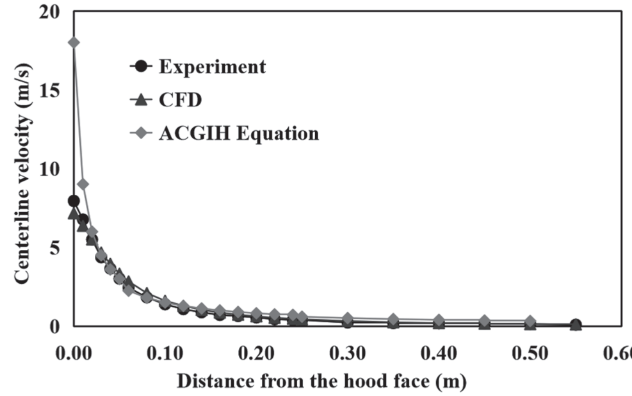

Fig. 5. Relationship between the distance from the hood face (X) and the centerline velocity (V). Slot hood face size: 0.5 m (L) × 0.1 m (H), aspect ratio (R_a): 5, exhaust airflow rate (Q): 20 m^3/min. CFD: computational fluid dynamics; ACGIH: American Conference of Governmental Industrial Hygienist.

were significantly different from those obtained using the ACGIH equation.

Figure 6 shows the relationship between V and X from the experiments, CFD analysis, ACGIH equation, for a free-standing slot hood with a Ra of 6.67. As a result, the centerline velocities obtained using the ACGIH equation were consistent with the experimental and CFD analysis results at a distance (X) greater than 0.02 m in front of the hood face. However, within 0.02 m of the hood face, the experimental and CFD analysis results were significantly different from those obtained using the ACGIH equation. Similar results were also obtained for the four other hoods with different aspect ratios. In other words, the experimental and CFD analysis results showed that when the aspect ratio was different, the relationship between V and X was also different. This further illustrates that the aspect ratio needs to be considered when calculating the centerline velocity.

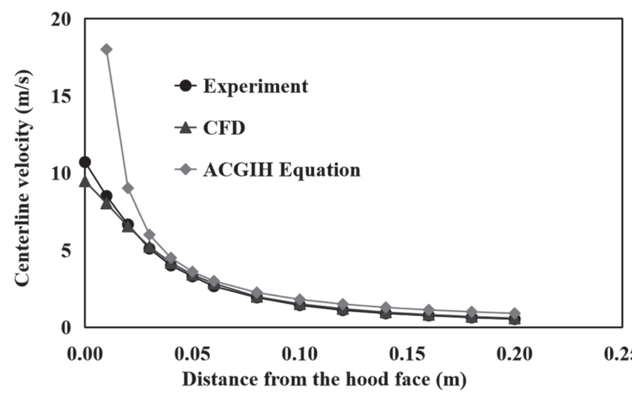

Fig. 6. Relationship between the distance from the hood face (X) and the centerline velocity (V). Rectangular hood face size: 0.5 m (L) × 0.075 m (H), aspect ratio (R_a): 6.67, exhaust airflow rate (Q): 20 m^3/min. CFD: computational fluid dynamics; ACGIH: American Conference of Governmental Industrial Hygienist.

Dimensionless velocities

Based on the relationship between the centerline velocity and the distance from the hood face with different sizes of hoods, in this study, the results within a distance from the hood face of 0.2 m were used to investigate the relationship between the centerline velocity (V) and the distance from the hood face (X) with different aspect ratios (R_a) of the hood. Taking the distance from the hood opening as the X-axis and taking the ratios of the centerline velocity of the hood face (V_0) with the centerline velocity results (V) as the Y-axis, Figs. 7, 8, 9, 10, and 11 plot the relationship between X and V_0/V for five different faces of the capture hood at all exhaust airflow rates, respectively.

From Fig. 7 to Fig. 11, it can be seen that the relationship between V_0/V and X was very close for all five Q when the hood aspect ratio (R_a) was 1, 3.33, 5, and 6.67. When the aspect ratio of the hood face was 10, although the approximate relationship between X and V_0/V for five exhaust airflow rates in the range of 0.15–0.2 m from the hood face is not better than for other aspect ratio (R_a), it can be seen to be roughly close. Therefore, in this study, the relationship between X and V_0/V obtained from all exhaust airflow rates of each aspect ratio (R_a) was treated by the dimensionless processing of V and took the approximate average value of the approximate relationship between X and V_0/V. To facilitate the following discus-

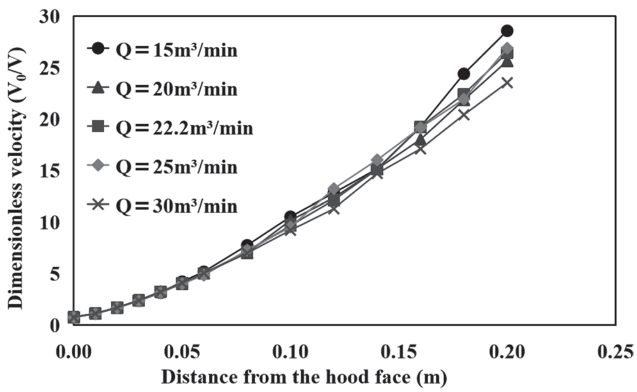

Fig. 7. Relationship between the distance from the hood face (X) and dimensionless velocity (V_0/V) of each exhaust airflow rate (Q). Slot hood face size: 0.5 m (L) × 0.05 m (H), aspect ratio (R_a): 10.

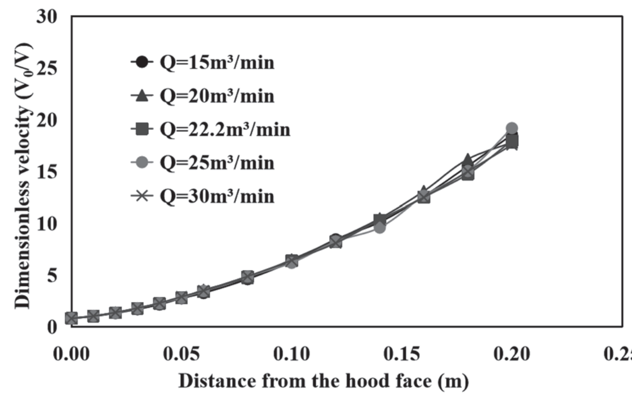

Fig. 8. Relationship between the distance from the hood face (X) and dimensionless velocity (V_0/V) of each exhaust airflow rate (Q). Slot hood face size: 0.5 m (L) × 0.075 m (H), aspect ratio (R_a): 6.67.

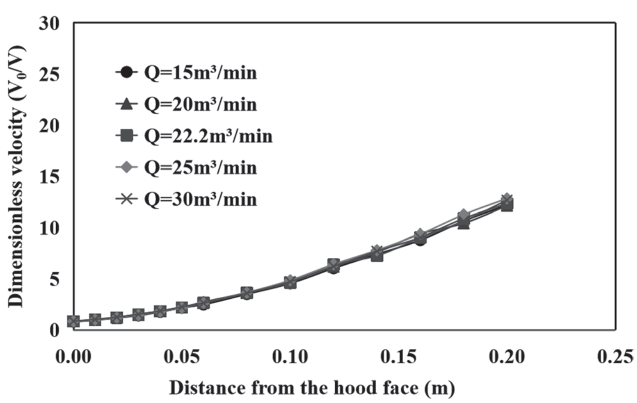

Fig. 9. Relationship between the distance from the hood face (X) and dimensionless velocity (V_0/V) of each exhaust airflow rate (Q). Slot hood face size: 0.5 m (L) × 0.1 m (H), aspect ratio (R_a): 5.

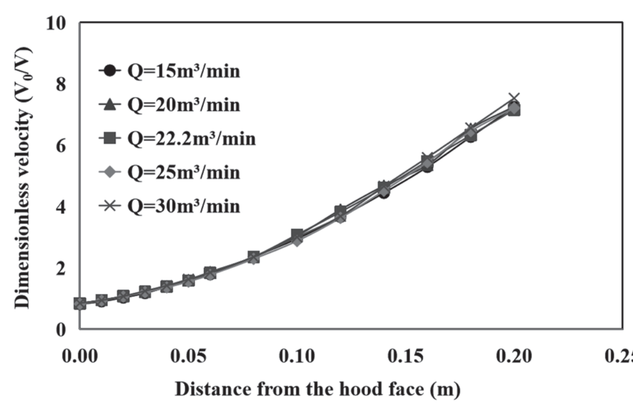

Fig. 10. Relationship between the distance from the hood face (X) and dimensionless velocity (V_0/V) of each exhaust airflow rate (Q). Slot hood face size: 0.5 m (L) × 0.15 m (H), aspect ratio (R_a): 3.33.

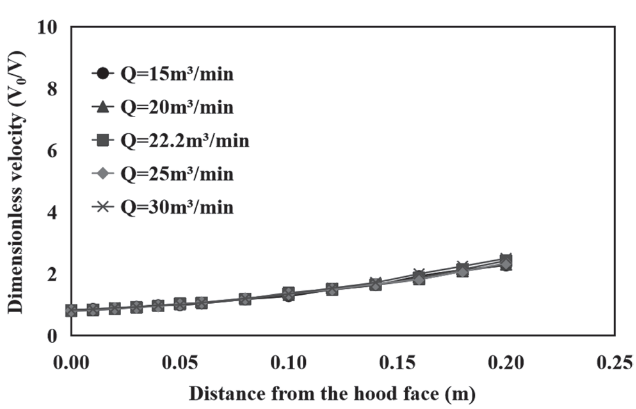

Fig. 11. Relationship between the distance from the hood face (X) and dimensionless velocity (V_0/V) of each exhaust airflow rate (Q). Slot hood face size: 0.5 m (L) × 0.5 m (H), aspect ratio (R_a): 1.

sion, the average value of V_0/V was defined as V_A, as shown in Eqs.(4). Where V_0 is the centerline velocity on the hood face, V is the centerline velocity.

$$V_A = V_0/V \tag{4}$$

Figure 12 shows the experimental results of the relationship between V_A and X for five different R_a of the capture hood. The results show that the relationship between V_A and X can be approximated as a quadratic polynomial equation. These results also revealed a difference in the relationship between V_A and X concerning R_a of the hood. These findings show that V has a strong relationship with R_a by using the dimensionless treatment method. A similar relationship between V_A and X was obtained using the CFD analysis.

The quadratic polynomial equation from the relationship between V_A and X of five different aspect ratios

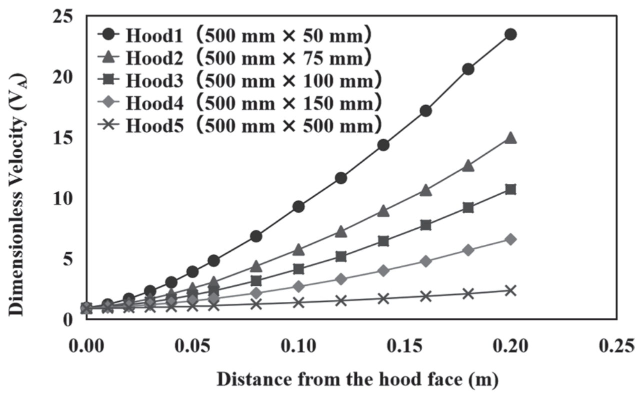

Fig. 12. Relationship between the distance from the hood face (X) and dimensionless velocity (V_A) of experimental results.

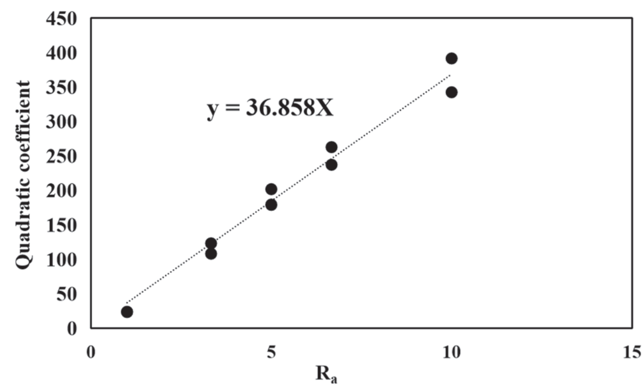

Fig. 13. Relationship between the quadratic coefficient of approximate equation and aspect ratio.

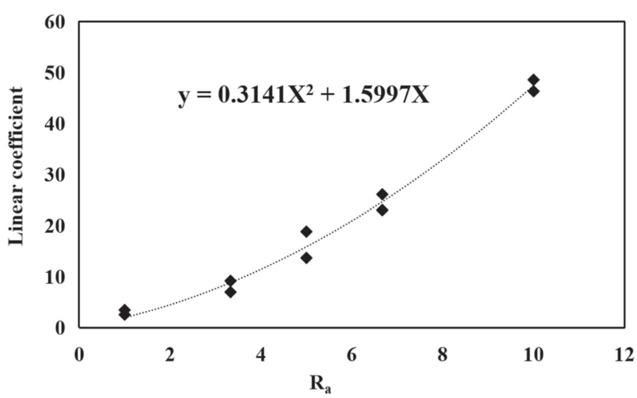

Fig. 14. Relationship between linear coefficient of approximate equation and aspect ratio.

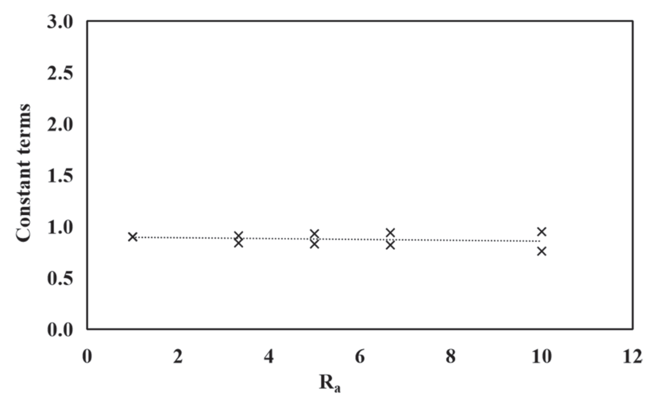

Fig. 15. Relationship between constant term of approximate equation and aspect ratio.

Table 2. The quadratic polynomial trend-line equation from the relationship between V_A and X

	Aspect ratio (L/H)	Quadratic polynomial equation of V_A	R^2
Experiment	10	V_A=391.38X^2+48.55X+0.76	0.9998
	6.67	V_A=262.45X^2+26.08X+0.82	0.9998
	5	V_A=201.35X^2+18.81X+0.83	0.9993
	3.33	V_A=123.13X^2+9.15X+0.84	0.9999
	1	V_A=23.26X^2+3.43X+0.9	0.9993
CFD	10	V_A=342.25X^2+46.30X+0.95	0.9989
	6.67	V_A=236.86X^2+23.01X+0.94	0.9994
	5	V_A=179.01X^2+13.65X+0.93	0.9996
	3.33	V_A=108.05X^2+6.95X+0.91	0.9999
	1	V_A=23.72X^2+2.52X+0.9	0.9997

L: length; H: height; CFD: computational fluid dynamics.

obtained from experimental, simulation is summarized in Table 2. The approximate equations for the five aspect ratios obtained from CFD and experiments can be expressed as highly correlated quadratic polynomial approximation equations. Taking R_a as the X-axis and taking the quadratic coefficient, the linear coefficient, and the constant of the approximate equation as the Y-axis, the scatter diagram of the experiment and CFD result is drawn as Fig. 13 to Fig. 15. It is known from Fig. 14 to Fig. 15. that as R_a increases, the quadratic coefficients tend to increase linearly and the linear coefficients increase quadratically, whereas the constant term changes little regardless of the aspect ratio.

It can also be seen that both CFD and experimental results show that the coefficient of V_A with R_a has good regularity, which can use to estimate the influence of the variation of the R_a of the hood on V, and the trend line fitting is made by Excel. Using the approximate equations of the relationship between quadratic coefficient and R_a and linear coefficient and aspect ratio to express V_A can be approximately expressed as Eqs.(5). Since the constant terms do not change with the aspect ratio, the average value is 0.88.

$$V_A = 36.86R_a X^2 + (0.31R_a^2 + 1.60R_a)X + 0.88 \qquad (5)$$

Where R_a is the aspect ratio of the hood, X is the distance from the hood face. Based on the centerline velocity change rule of slot and capture hood proposed in this paper, the velocity of the hood opening (V_0) can be calculated by Eqs.(1) and expressed as the following Eqs.(6).

$$V_0 = \frac{Q}{A} \qquad (6)$$

Substituting V_0/V in Eqs.(4) into V_A in Eqs.(5), and then substituting Q/A in Eqs.(6) into V_0, an equation that calculates the Q and contains only Ra, V and A can be obtained, as shown in Eqs.(7).

$$Q = 60AV[(36.86R_a X^2 + (0.31R_a^2 + 1.60R_a)X + 0.88)] \qquad (7)$$

The newly derived equation is named the W-TKM equation, where Q is the exhaust airflow rate of the hood; A, the area of the hood face; V, the centerline velocity; X, the distance from the hood face; and R_a, the aspect ratio.

Discussion

Validation of new equation

The following example shows the calculating result of the velocity of a slot capture hood with R_a=10 (the hood opening face: L=0.5 m, W=0.05 m) when the exhaust airflow rate (Q) is 15 m^3/min and the distance from the opening face (X) is 0.2 m.

$$15 = 60 \times 0.5 \times 0.05 \times V \times [36.86 \times 10 \times 0.2^2 + (0.31 \times 10^2 + 1.60 \times 10) \times 0.2 + 0.88]$$

$$V = \frac{15}{60 \times 0.5 \times 0.05 \times [36.86 \times 10 \times 0.2^2 + (0.31 \times 10^2 + 1.60 \times 10) \times 0.2 + 0.88]}$$

$$V = 0.4 \text{ m/s}$$

Simultaneously, the centerline velocity of the other 4 exhaust airflow rates of X=0.2 m is calculated and compared with the results obtained from the experiment, CFD, and ACGIH equation, as shown in Table 3. The calculation results of the new equations were significantly closer to the experiment and CFD than ACGIH equations.

The results of comparing the centerline velocities of different distances from the hood face under the same exhaust airflow rate are shown in Fig. 16. The graph shows the centerline velocity (V) results of a slot hood with an aspect ratio of 10 as obtained from the experiments, CFD analysis, ACGIH equation, and W-TKM equation for Q=22.2 m^3/min. The closer to the position of the hood face, the result of the centerline velocity calculated by the W-TKM equation is almost consistent with the CFD analysis and experimental results. However, the ACGIH equation results were significantly different from the CFD analysis and experimental results near the hood face.

Figure 17 shows a comparison of the centerline velocities (V_X) of a rectangular hood with an aspect ratio (R_a) of 3.33 at Q=20 m^3/min. The results obtained using the W-TKM equation were more closely with the CFD analysis and experimental results than obtained using the ACGIH equation. The comparison results confirm that the new equation leads to better results than the conventional equations. Simultaneously, the validity of the new equation has also been verified.

The results of the exhaust airflow rate (Q) calculated by the new equation are compared with experiment and CFD analysis, as shown in Fig. 18. The graph shows the relationship of the exhaust airflow rate (Q) with the distance from the hood face (X) of a slot opening hood with aspect ratio (R_a) is 10, when the centerline velocity is 1 m/s. It can be seen that the results of the exhaust airflow rate (Q) calculated by the new equation were very close to the experiment and CFD analysis.

Figure 19 shows the results of centerline velocities

Table 3. The results of centerline velocity (V), the distance from the hood opening (X): 0.2 m, the aspect ratio: 10

	The exhaust airflow rate (m^3/min)				
	15	20	22.2	25	30
Experiment	0.35	0.52	0.56	0.62	0.85
CFD	0.43	0.57	0.63	0.71	0.85
W-TKM equation	0.40	0.53	0.59	0.67	0.80
ACGIH equation	0.68	0.90	1.00	1.13	1.35

CFD: computational fluid dynamics; ACGIH: American Conference of Governmental Industrial Hygienist.

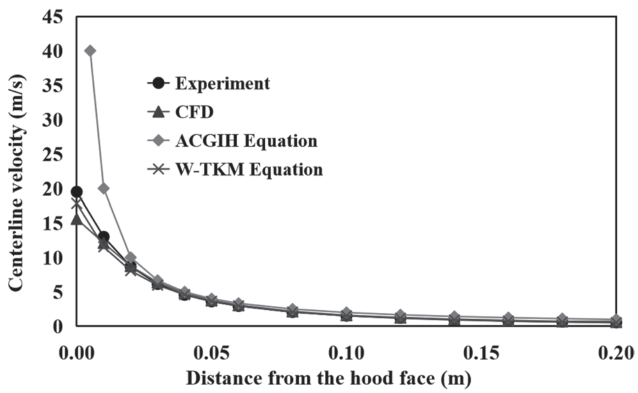

Fig. 16. Relationship between the distance from hood face (X) and the centerline velocity (V). Slot hood face size: 0.5 m (L) × 0.05 m (H), aspect ratio (R_a): 10, exhaust airflow rate (Q): 22.2 m^3/min. CFD: computational fluid dynamics; ACGIH: American Conference of Governmental Industrial Hygienist.

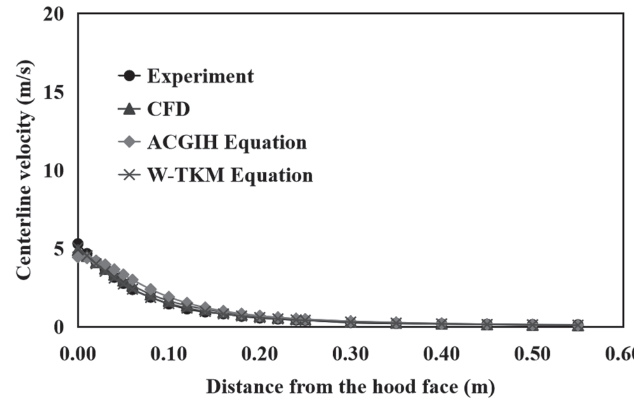

Fig. 17. Relationship between the distance from hood face (X) and the centerline velocity (V). Rectangular hood face size: 0.5 m (L) × 0.15 m (H), aspect ratio (R_a): 3.33, exhaust airflow rate (Q): 20 m^3/min. CFD: computational fluid dynamics; ACGIH: American Conference of Governmental Industrial Hygienist.

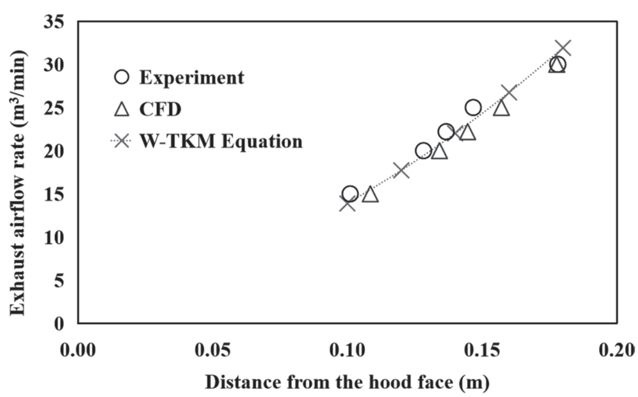

Fig. 18. Relationship between the distance from hood face (X) and the exhaust airflow rate. Slot hood face size: 0.5 m (L) × 0.05 m (H), aspect ratio (R_a): 10, the centerline velocity (V): 1 m/s. CFD: computational fluid dynamics.

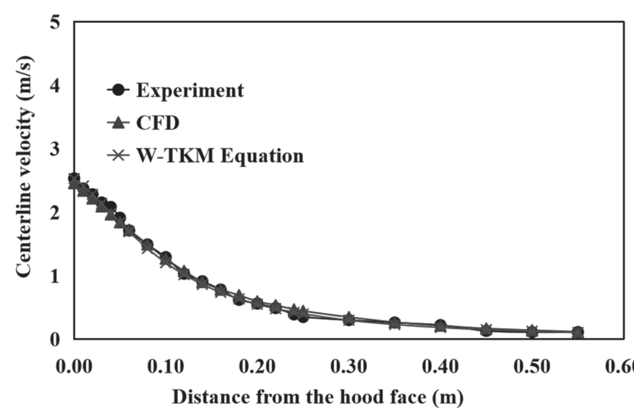

Fig. 19. Relationship between the distance from hood face (X) and the centerline velocity (V). Rectangular hood face size: 0.5 m (L) × 0.3 m, aspect ratio (R_a): 1.67, exhaust airflow rate (Q): 20 m^3/min. CFD: computational fluid dynamics.

obtained by experiment, CFD, and the new equation for a rectangular capture hood with an opening face of 0.5 m (L) × 0.3 m (H) when the exhaust airflow rate was 20 m^3/min. It can be found that the results of the new equation were in good agreement with the experimental and CFD analysis results.

Another condition, Q=35 m^3/min, was verified by using a slot capture hood with an opening face of 0.5 m (L) × 0.05 m (H). The results obtained by experiment, CFD, and the new equation was shown in Fig. 20. It can be clearly seen that the results obtained by the new calculation equation were in good agreement with the experimental results not only at positions far from the hood face (0.2 m–0.55 m), but also at positions close to the hood face.

From these comparisons, the new calculation equation

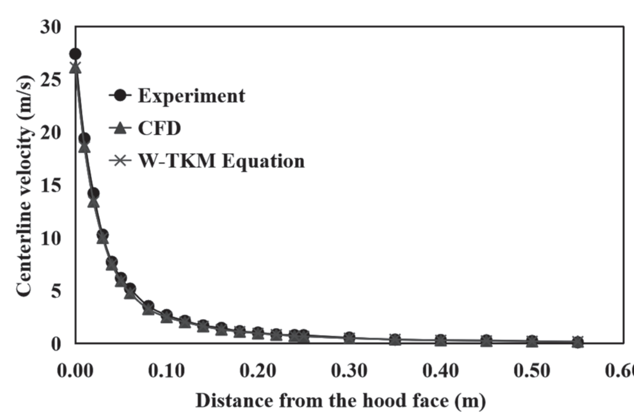

Fig. 20. Relationship between the distance from hood face (X) and the centerline velocity (V). Slot hood face size: 0.5 m (L) × 0.05 m, aspect ratio (R_a): 10, exhaust airflow rate (Q): 35 m^3/min. CFD: computational fluid dynamics.

leads to better results than the conventional approach and can accurately evaluate the exhaust airflow rate (Q) of both the slot opening hood and the rectangular hood when designing a local exhaust ventilation system.

It is important to know the velocity field induced within the suction action area when designing a local exhaust hood. Using the velocity along the hood face centerline to characterize the flow is a common practice. This simplification is easy to use. Combined with experimental and CFD results, it can be seen that the calculation equation obtained by studying the relationship between the centerline velocity, exhaust airflow rate, and aspect ratio (R_a) of the capture hood in this study is more accurate than the previous equation (ACGIH Equation). And it has good versatility for exhaust hoods with different aspect ratios. This provides a useful proposal when designing a free-standing capture hood.

Conclusions

In the capturing slot hood of an LEV system, the centerline velocities (V_X) calculated using the ACGIH equation is different from the experimental and CFD analysis results near the hood face. A challenge in the study area was the fact that the equations for rectangular and slot hoods differed. To solve this problem, this study conducted experiments and CFD simulation method to analyze the centerline velocities (V) near the hood face and introduce the non-dimensional method to treat the centerline velocity. By this method, a good correlation was found between the aspect ratio and centerline velocity. Consequently, a simple quadratic function was successfully created for a rectangular and slot hood, and a simple form of the conventional equations was derived.

By using this new equation, when estimating the relationship between exhaust airflow rate and centerline velocity, the step of distinguishing the aspect ratios of different capture hoods in the calculation can be omitted. Subsequently, an LEV system can be designed to optimize the exhaust airflow rate.

It is well known that a flange can be attached around a hood face to improve the capture efficiency of the capture hood. However, this paper only investigated non-flanged hood. This remains to be further clarified in future studies. Such as flanged capture hood or flanged desktop capture hood.

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