E3S Web of Conferences **356**, 04022 (2022) https://doi.org/10.1051/e3sconf/202235604022
ROOMVENT 2022

INFLUENCING FACTORS OF AIR VOLUME IN KITCHEN COOKING EXHAUST SHAFT SYSTEMS OF HIGH-RISE RESIDENTIAL BUILDINGS

Minghui Yu,* *Kailiang Huang,Guohui Feng,Xu Hou*
Shen'yang Jian'zhu University, China.

Abstract. This paper uses mathematical model theoretical calculation and experimental verification to study the pressure distribution inside the residential kitchen flue and the flow rate of each user through three aspects: temperature difference, opening rate and hood performance. Based on fluid network theory, the flow equation of oil smoke in the cooking exhaust shaft system is established. The accuracy of the mathematical model model was verified through experiments and applied to the calculation of the flue in the kitchen of a 30-story residential building in Shenyang. The results show that the static pressure decreases continuously with the increase of floor height, and the highest pressure appears at the lower level of the floor. The pressure keeps rising as the opening rate increases, i.e., the more users open, the higher the pressure at the bottom floor. Under the joint action of thermal pressure and flue resistance, the static pressure shows a low-high-low distribution form, with the temperature difference increases, the maximum static pressure in the flue moves to the bottom; the flow rate of users on each floor increases with the height, and decreases with the increasing rate of opening, under the influence of thermal pressure user flow distribution is high-low-high distribution.

Corresponding author: 952567057@qq.com

1 Introduction

In this paper, the cooking exhaust shaft system of a 30-story residential building in a cold region of China is selected as a case study to analyze the stereotyped products suitable for cold regions in the National Standard Drawing-Residential Exhaust Duct (I) (Drawing No. 07J916-1). A mathematical model of the flow of grease in the cooking exhaust shaft system was established and corrected for thermal pressure, along-range resistance and local resistance[1]. Fixed-section flue Due to the flow resistance of the high-rise residential kitchen exhaust system is too large the exhaust volume of the lower part of the system occupants can not reach the design value (300 ~ 500m3 / h), the system exhaust smoke uneven problem is serious, the bottom part of the middle is in a high static pressure state, static pressure in about 245pa while the upper part of the middle often exists because of the lack of pressure of the phenomenon of backflow of smoke, causing serious inconvenience to the life of the occupants.In fact, as long as $p_1 > p_0 + \rho g h$, there will be a backflow of flue gas[2]. When the exhaust hood is not on at a particular level, $p_0 = 0$ and $\rho g h$ is only about 45 Pa, which is a small pressure.The engineering is intended to solve the problem of stringing smoke and odor by setting up variable pressure plates and optimizing the flow guiding components, but in practice, backflow will definitely occur without installing check valves. At present, the internal flue gas flow pressure analysis is not yet sound, in the case of low static pressure value [3], continuous conversion of dynamic pressure and static pressure will instead make the pressure loss larger, resulting in too small ventilation volume. The temperature difference between the inside and outside of the flue studied in this paper is $\Delta T = 49$ ℃ , $\Delta T = -5$ ℃ , and the temperature difference has a greater impact on the flue gas flow characteristics. It often leads to an increase in energy consumption of the building. So how much smoke is discharged from each branch flue, and how to determine the cross-sectional area of the flue becomes the core problem of flue design. The analysis of the internal flow field of the flue with variable pressure structure and check valve is necessary to reduce the building energy consumption and to prevent smoke cascading.

2 Mathematical models

2.1 Flow equation of oil fumes in the cooking exhaust shaft system

The following is the analysis of the cooking exhaust shaft system in a N-storey residential building. the schematic diagram of a cooking exhaust shaft system, in which H is the height of the floor, and H m is the height of the end duct. Cooking exhaust shaft is connected by N exhaust ducts in the height of H.

The schematic diagram of the ventilation system. Where ‘i’ is the number of range hoods, qvi is the exhaust airflow of the ith range hood for kitchen, and qvci is the exhaust airflow in the exhaust shaft between the ith and (i+1)th range hood.

It is considered that the backdraft damper on the non-operating floor (the range hood is closed on this floor) is sealed, and there is no air leakage in the cooking exhaust shaft system. Due to the slight stack effect, the temperature difference has little impact on the exhaust airflow. Therefore, the buoyancy effect is not applicable in these equations. The continuity equation and the pressure drop equation for the flow of oil fumes in the cooking exhaust shaft system are as follows.

Continuity equation of the oil fumes flow in the cooking exhaust shaft system,

When i = 1

$$q_{vc}1 = q_v1, \text{ that is, } A_1 \times V_1 = A_{b,1} \times v_{b,1} \quad (1)$$

When $2 \leqslant i \leqslant N$ (N is the total number of the operating range hoods),

$$q_{vc}i = q_{vc}(i-1) + q_v i,$$
$$A_i \times v_i = A_{i-1} \times v_{i-1} + A_{b,i} \times v_{b,i} \quad (2)$$

Where $A_i$is the section area of the ith exhaust duct, m^2; $v_i$is the average air velocity in the ith exhaust duct, m/s; $A_{b,i}$ is the section area of the ith branch duct, m^2; and $v_{b,i}$ is the average air velocity in the ith branch duct, m/s;

Pressure drop equation of the oil fumes flow in the cooking exhaust shaft system

When $1 \leqslant i \leqslant N-1$(N is the total number of the operating range hoods)

$$p_{j,i} + p_{d,i} + NH_o(\rho_n - \rho_w)g = p_{j,i+1} + p_{d,i+1} + p_{l,i} + p_{m,i} + iH_o(\rho_n - \rho_w)g \quad (3)$$

When i = N,

$$p_{j,i} + p_{d,i} = p_{o,j} + p_{o,d} + p_{lo,i} + p_{mo,i} \quad (4)$$

Where $p_{j,i}$ is the static pressure of the ith exhaust duct, Pa; $p_{d,i}$is the dynamic pressure of the ith exhaust duct, Pa; $p_{l,i}$ is the friction loss of the ith exhaust duct, Pa; $p_{m,i}$ is the local loss of the ith exhaust duct, Pa; $p_{o,j}$ is the static pressure at the outlet of the exhaust shaft, which is the outdoor atmospheric pressure ,0 Pa; $p_{lo,i}$ is the friction loss of the Nth duct to the roof exit, Pa; and $p_{mo,i}$ is the local loss of the roof hood, Pa.

Pressure drop equation of the oil fumes flow in the branch duct

$$p_{b,i} = p_{j,i} + p_{d,i} + p_{b,l,i} + p_{b,m,i} \quad (5)$$

Where $p_{b,i}$ is the total pressure of the ith range hood, determined according to the performance curve of the range hood, Pa;$p_{b,l,i}$is the friction loss of the ith branch duct, Pa; and $p_{b,m,i}$ is the local loss of the ith branch duct, Pa.

2.2 Calculation process

For the exhaust system with n hoods running at the same time, the flow rate, full pressure, flue gas pressure, and flue gas flow rate of each layer of hoods are unknown. In this calculation, first set the hood performance parameters (speed, static pressure), use the hood pressure and speed as the initial value of each layer, and the flue outlet pressure is 0 as the final value, and use matlab to write equations for each layer to calculate.The parameters of the mathematical model are entered in **Table 1.**

Table 1. Input parameters in the mathematical model.

Item	**Parameter**
Total height of the exhaust shaft	84000mm
Exhaust shaft cross-sectional area	$0.2704m^2$
Diameter of the branch duct	180mm
Height of the branch duct	1100mm
Wall roughness of the exhaust shaft	K=1.5
Loss coefficient of the backdraft damper	2.7
Loss coefficient of the roof hood	0.9

2.3 Calculation of relevant parameters in the flow equations

In the case where R and Re values are determined, it is convenient and accurate to use the Alitssouri formula to calculate the along-travel drag coefficient [5].

(1)Friction loss of the ith exhaust duct

$$p_{l,i} = \lambda_i \times \frac{H}{d_c} \times \frac{\rho}{2} \times {v_i}^2 \tag{6}$$

Where λ_i is the friction loss coefficient of the ith exhaust duct; H is the duct height, m; and d_c is the equivalent diameter of the exhaust duct, m.

Friction loss coefficient is calculated according to the following formula:

$$\lambda_i = 0.11 \times \left(\frac{K_\varepsilon}{d_c} + \frac{68}{Re_i}\right)^{0.25} \tag{7}$$

Where K_ε is the absolute roughness of the exhaust duct wall, mm; and Re_i is the Re of the ith exhaust duct.

(2)Local loss of the ith exhaust duct:

$$p_{m,i} = \xi_i \times \frac{\rho}{2} \times (v_i + v_{\hbar,i})^2 \tag{8}$$

Where ξ_i is the local resistance coefficient of the right-angle tee.

The right-angle tee is a confluence tee.The local resistance coefficient is:

$$\xi_i = 1.55 \times \frac{v_{b,i} \times A_{b,i}}{v_i \times A_i} - \left(\frac{v_{b,i} \times A_{b,i}}{v_i \times A_i}\right)^2 \tag{9}$$

(3)Friction loss of the ith exhaust branch duct：

$$P_{b,l,i} = \lambda_{b,i} \times \frac{l}{d_{b,c}} \times \frac{\rho}{2} \times v_{b,i}^2 \tag{10}$$

Where $v_{b,i}$ is the friction loss coefficient of the ith exhaust branch duct; l is the length of the branch duct, m; and $d_{b,c}$ is the equivalent diameter of the branch duct, m.

Friction loss coefficient $\lambda_{b,i}$ is calculated according to the following formula:

$$\lambda_{b,i} = 0.11 \times \left(\frac{K_b}{d_{b,c}} + \frac{68}{Re_{b,i}}\right)^{0.25} \tag{11}$$

Where K_b is the absolute roughness of the branch duct wall, mm; and $Re_{b,i}$ is the Re of the ith branch duct.

(4)Local loss of the ith branch duct:

$$P_{b,m,i} = (\xi_i + \xi_b) \times \frac{\rho}{2} \times v_{b,i}^2 \tag{12}$$

Where ξ_b is the local loss coefficient of the backdraft damper, obtained from the actual test.

The right-angle tee is a confluence tee.The local resistance coefficient is:

$$\xi_i = 1.55 \times \frac{v_{b,i} \times A_{b,i}}{v_i \times A_i} - \left(\frac{v_{b,i} \times A_{b,i}}{v_i \times A_i}\right)^2$$

(5)Local loss of the roof hood:

$$p_{mo,i} = \xi_f \times \frac{\rho}{2} \times v_N^2 \tag{13}$$

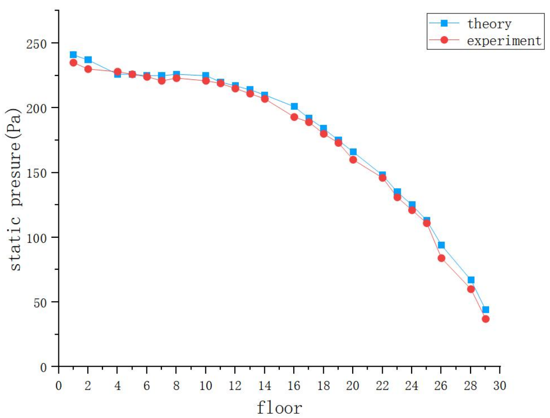
Where ξ_f is the loss coefficient of the roof hood, determined according to the actual roof hood; and v_N is the air velocity in the Nth exhaust duct, m/s.

2.4 Comparing the numerical model with the experimental

Table 2. Test instruments.

Instruments name	**Performance**
Pitot tube	Pitot tube coefficient K=1.00
Digital pressure gauge	Level 0.5
Digital anemometer	Accuracy: ± 3% of reading or ± 0.015m/s, larger value

A comparison of the numerical model and experimental data for the static pressure in the kitchen flue is shown in Figure1 This figure shows that the theoretical values are in good agreement with the measured values. Except for the 3rd,8th,16th,20th and 26th floors. The reasons for the errors are due to the fact that the performance of the exhaust hood models is not the same in each household; air leakage in the flue system due to the installation of the households in the test; the lack of preheating before the experiment and the existence of fluctuations in the temperature inside the flue, etc. However, the deviation between the numerical model and the static pressure test of the exhaust shaft is less than 10%, and the model is still considered credible.



https://doi.org/10.1051/e3sconf/202235604022

Fig. 1. comparation between the numerical model and the experiment.

3 Results and discussion

In severe cold regions fume emission is strongly influenced by temperature, especially in winter. This section discusses the analysis of the static pressure and velocity distribution of the central exhaust system for the most unfavorable temperature differences 49℃, 22℃ and -5℃ in summer and winter with the data calculated from the model. The temperature difference at four operating conditions. 30%, 66.7%, 76.7 % and 90% of the fume opening rate is also explored as to which operating condition has a greater effect.

3.1 Static pressure distribution in central exhaust shaft

With the coupling of hood operation and heat pressure driven ventilation, the static pressure distribution in the central exhaust shaft is shown in Figure2. It can be noted that the airflow velocity in the kitchen gradually increases as the floor rises, with users on the bottom floor tending to obtain the lowest kitchen airflow velocity due to the highest resistance level. As the temperature difference increases, the maximum static pressure in the flue gradually moves in the middle and lower levels, and the static pressure shows a low-high-low distribution.

The reason for this is because in the middle and lower floors, the number of floors is reduced when the flue gas thermal pressure in the flue although there is an increase, but the flue gas from the interior through the flue longer distance, the flue gas resistance loss increases, when the resistance loss increment is greater than the increment of thermal pressure, the flow rate decreases with the floor. At the same time, the flow of flue gas decreases flow rate decreases, the resistance loss decreases layer by layer, so the flow of smoke decreases layer by layer with the number of layers. If the amount of smoke loss in the area is less than the amount of heat pressure increase, the flow of exhaust hood will appear with the floor down to increase the phenomenon.

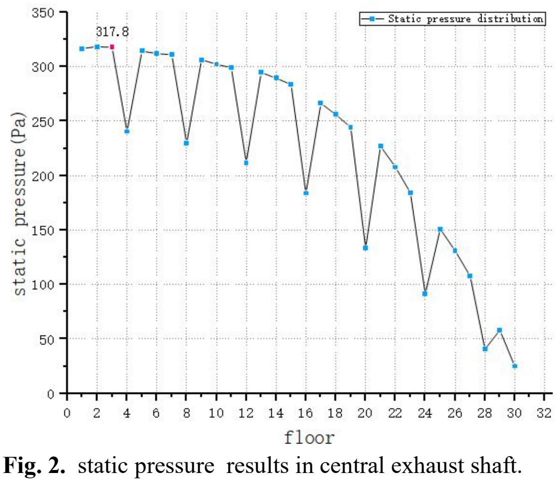


Fig. 2. static pressure results in central exhaust shaft.

Figure3 illustrate the effect of temperature difference on the static pressure distribution in the central exhaust shaft at four operating rates. According to the principle of heat-pressure driven ventilation, the higher the temperature difference, the higher the density difference. The static pressure is proportional to the density difference and height difference between the inlet and outlet of the central exhaust shaft. At the same operating rate, when the outdoor temperature increases, the static pressure decreases accordingly. In the summer time, as the temperature difference between indoor and outdoor is negative, the thermal pressure effect should be ignored, gravity plays a certain obstructive role, and the flow rate in the flue is in a decreasing trend. At the same time, in the case of low hood operation rate, the static pressure hardly changes with the increase in floor height and remains at no higher than 45pa correspondingly the mechanical ventilation momentum is small, which is not conducive to smoke exhaust.

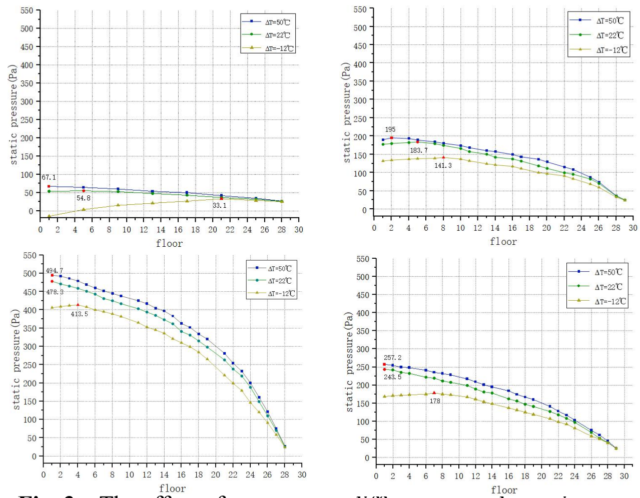


Fig. 3. The effect of temperature differences on the static pressure distribution in central exhaust shaft:30%, 66.7%, 76.7 % and 90%

3.2 Exhaust flow distribution in central exhaust shaft

As can be noted in Figure 4, the smoke flow in the branch pipe gradually increases as the floor rises, with users on the lower floors tending to receive a smaller smoke exhaust flow due to the highest resistance levels. With the three temperature differences, the smoke exhaust flow rate for users increases as the floor height increases. The law of change from the bottom to the top floor shows a decrease, increase and then decrease, and this law is verified with the results analyzed above. By comparing the flow rate of the branch pipe with different temperature difference for the same opening rate (100% operation rate as an example), Figure4 shows that the maximum flow rate in the branch pipe is 537 m^3/h at a temperature difference of Δ T=49 ℃ , which occurs on the 28th floor. At the temperature difference of Δ T=22℃, the maximum flow rate in the branch pipe is 531m^3/h, which occurs in the 28th floor. At a temperature difference of Δ T=-5 ℃ , the maximum flow rate in the branch pipe is 524m3/h on the 28th floor. It can be concluded that there is not only

https://doi.org/10.1051/e3sconf/202235604022

a linear relationship between the exhaust flow rate and the floor height, but also when the temperature difference increases, the overall exhaust flow rate will slightly increase, and the increase of exhaust flow rate is more obvious compared to the middle floor users and the upper floor users. For individual residents, the higher flow rate in the branch flue is beneficial to the removal of smoke.

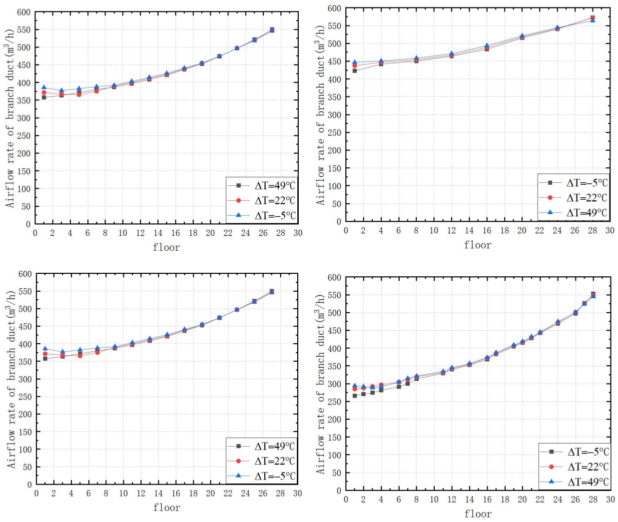


Fig. 4. Effect of temperature difference on branch pipe flow distribution: 30%, 66.7%, 76.7 % and 90%

4 Conclusions

In this paper, field tests and theoretical calculations of the fluid pipe network system were conducted in a 30-story residential kitchen in Shenyang, a severe cold region. The effects of the thermal pressure inside the kitchen flue and the working rate of the range hood on the airflow characteristics inside the flue and for each user were analyzed. Based on the studies in this paper, the main conclusions are summarized as follows. Under operating conditions, the static pressure in the central exhaust well decreases with increasing floor height. With the temperature change from 49 ℃ to -5 ℃ , the position of the maximum static pressure gradually moves downward in a low-high-low distribution. The maximum value occurs in the middle and low layers, which is most unfavorable for the central exhaust system. Therefore, a hood with a higher total pressure should be used in the middle and low floors to meet the exhaust needs. For residents living in the upper floors, the pressure in the flue is less than in the lower and middle floors. In this case, it is recommended to use a range hood with a lower total pressure. Not only can the exhaust flow balance be achieved in the kitchens of high-rise residences, but it is also more energy efficient to a certain extent. Under the joint influence of factors such as thermal pressure and resistance distribution, the rule of change of the exhaust volume of each layer of users is first reduced and then increased and then reduced; In the working condition where the exhaust hood is on at 50%, the overall smoke emission is improved by 32.5% after considering the effect of thermal pressure. Therefore, the role of thermal pressure cannot be ignored in the scientific study of the problem. The maximum exhaust air volume under the working condition of 14.2% of the hood opening rate is up to 577m^3/h, and the maximum exhaust air volume appears in the upper part of the system; with the increase of the hood opening rate the exhaust air volume of each floor gradually decreases, and the air volume under the working condition of 100% of the opening rate is less than 300m^3/h for 64% of the users. The theoretical calculation method of the exhaust air volume of users on each floor in high-rise residential buildings proposed in this paper is also applicable to other working conditions such as high-rise residential buildings with different heights, different flue cross-sectional areas and different indoor and outdoor temperatures. This method can be used to calculate the exhaust air volume of users under various working conditions and establish a database to provide a basis for practical engineering design.

References

1. L.Wang, Z.Y. Xing, Z. Ristovski, Role of Chinese cooking emissions on ambient air quality and human health, Science of the Total Environment, 589, 173-181(2017)
2. Y.J. Zhao, C. Chen, B. Zhao, Emission characteristics of PM2.5-bound chemicals from residential Chinese cooking, Building and Environment,149, 623-629(2019)
3. C.M. Chang, C.M. Lai, P.C. Chou, Y.Y. Li, The influence of an architectural design alternative on indoor air environment in conventional kitchens in Taiwan, Build.Environ, 355, 79 (2000)
4. L.C. Tseng, C.C. Chen, Effect of flow characteristics on ultrafine partic leemissions from range hoods, Ann. Occup. Hyg.,57,920 (2013) .
5. J. Gao, C.S. Cao, X. Zhang, Z.W. Luo, Volume-based size distribution of accumulation and coarse particles (PM0.1-10) from cooking fume during oil heating, Build. Environ, 59, 575(2013)
6. K. Straif, R. Baan, Y. Grosse, et al., Carcinogenicity of household solid fuel combustion and of high-temperature frying, Lancet Oncol, 7, 977(2006)