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Theoretical and Experimental Simulation of Exhaust Hoods

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Synopsis

The paper presents a criterion to assess the performance of mechanical exhaust hoods for domestic kitchens and a procedure to experimentally test them; an analysis of the relevant parameters which affect their performance is made, the test results are shown, and finally these are compared with the results of a numerical fluid dynamic code.
Experiments were performed using the tracer gas technique, and attention has been drawn rather on the hood efficiency in the removal of pollutants than on the IAQ in the test room. It was first shown that the choice of the tracer gas has no influence on the results of the experiments. Then, the influence of the following parameters has been investigated: distance from hood to pollution source, thermal power dissipated under the source itself, extracted air flow rate, and hood-to-walls geometry. The repeatibility of the experimental apparatus was less than 10 %.
A simplified empirical expression based on non-dimensional parameters has been derived, which provides a precise estimate of hood efficiency.
Furthermore, a number of tests have been simulated by means of a CFD model. The results provide an interesting insight on the hood fluidynamic behaviour, yet the calculated efficiency is underestimated by 20 - 30 % respect to the measured values.

List of symbols

C = concentration
D = diameter
E = efficiency of the hood
g = gravity acceleration
h = height of the hot plate
H = distance between hot plate and base of the hood
q = tracer gas volume flow rate
Q = air volume flow rate
S = area
V = room volume
W = electrical power released by the hot plate
β = volume coefficient of expansion
ν = kinematic viscosity

1. Introduction

Kitchen hoods are used to remove and extract, directly from the source, pollutants of different kinds, usually while a certain amount of heat is released underneath.

Different indices have been adopted to describe their performance. For example, the French and Swedish standards NF E 51-704 and SS433 05 01 have adopted the so-called "collection efficiency", defined as:

$$E = 1 - C\,Q / [q\,(1 - \exp(-Q\,t/V))] \qquad (1)$$

where C is the concentration *in the room.*

A similar indicator, called "pollution index", more oriented to the definition of indoor air quality in the room, has been introduced by Wouters (1992).

On the other side, a "device-oriented" performance indicator could be simply defined as the ratio between the contaminant concentration at the exhaust of the hood and the maximum attainable contaminant concentration, i e, the value that would be reached if *all* the contaminant emitted by the source would be "captured" by the hood. This *hood efficiency* as a local extraction device is therefore given by:

$$E = C/C_{max} = C/(q/Q) \qquad (2)$$

where C is the concentration *in the exhaust*.

With this definition, one assumes that

(a) the size of the room has no influence whatever on the efficiency of the hood, and

(b) the phenomenon takes place in steady-state regime.

2. Description of the experimental apparatus

The experimental apparatus is located at the University of Basilicata (Potenza), where other ventilation facilities, as the Controlled Ventilation Chamber described by Fracastoro et al (1991), are located.

The hood under test is mounted on a steel frame and is placed above a table on which a circular electric hot plate (D = 0.17 m, h = 0.15 m) rests.

The dimensions and shape of the hood used in this first set of experiments are shown in Fig. 1. The hood is suspended to a metal framework; the exhaust duct is flexible and is connected to an iron plate duct (D = 80 mm, L = 1.20 m), sized (L = 15 x D) in order to ensure a sufficiently uniform distribution of air streams.

Three quantities have to be measured to derive the efficiency of the hood (see Eqn. 2): Q, q, and C. Furthermore, the parameters, as the released electrical power (W) and the distance (H) from the hot plate to the base of the hood, are to be known.

Air flow rate measurement and control

The air flow measurement and control systems (Fig. 2) are mounted on the first arm of a union tee, whose second arm is connected to a centrifugal fan having a nominal flow rate of 50 m^3/h, while the third arm is connected to the ambient air through a gate valve serving as a by-pass for the control of flow rate through the hood.

The air flow rate was measured by a mass flow meter (error less than 2 % on the reading) based on a venturi and a hot wire anemometer, provided by a digital display. The flow rate is expressed in normal litres per minute, and to obtain the correct volume flow rate a correction coefficient, function of air temperature and ambient pressure, has to be applied.

Tracer gas flow rate measurement and control

The tracer gas source is placed above the electric hot plate. The gas flows out of a glass tube (D = 6 mm, L = 0.15 m) connected to a rotameter by means of a plastic Rilsan tube. The rotameter has a range of 2.5 to 8.5 nl/h, with an error of 3 % full-scale. The flow rate is referred to air in standard conditions, therefore, in order to use it with gases different from air, a correction coefficient equal to the square root of the ratio between the molecular mass of the air and the gas has to be applied. In practice, since the value indicated by the rotameter was not considered too reliable, it was only used to check the stability of the flow during the test; the measurement of tracer gas flow rate was actually performed using "inversely" the procedure outlined by Svensson (1983) to measure the air flow rate in a duct. This procedure provides the volume flow rate in a duct (Q) as a function of the volume flow rate (q) of a tracer gas injected in the duct and of the concentration (C) which establishes downstream, where the tracer gas is assumed to be perfectly mixed, that is

$$Q = q/C \qquad (3)$$

The preceding Eqn. 3 was solved for q, being Q known from above, and the concentration measured as follows.

Tracer gas concentration measurement

The measurement of gas concentration in the exhaust of the hood is performed using two double-cell infrared gas analyzers for CH_4, SF_6, and N_2O, all three in the range 0-200 ppm, and an error of 2 % full scale. The analyzers are connected to a Data Acquisition and Control Unit, in its turn connected to a PC 286 through a GPIB interface.

Hot plate power measurements

The power released by the electric hot plate was measured indirectly through the eletrical resistance of the hot plate and the input voltage. The relative error was 0.5 % for the resistance and 1 % for the voltage.

The resulting probable and maximum relative errors are reported in Table 1.

Table 1 - Relative error of measured quantities.

Measured quantity	Probable error	Maximum error
Air flow rate	2.0 %	2.0 %
Tracer gas flow rate	2.8 %	4.0 %
Tracer gas concentration	2.0 %	2.0 %
Hot plate electrical power	1.4 %	2.5 %
Hood efficiency	2.8 %	6.8 %

3. Testing methodology and experimental results

Preliminary campaign: influence of the tracer gas type

At first, the question of the influence of the different tracer gas on the test results was faced. Experiments have shown that, when *no heat* is released from the hot plate, the influence of the tracer gas becomes very large. In fact, since there is no sufficient energy to break the link between molecules, heavy tracer gases tend to sink down and do not follow the air across the hood. Measurements turn out to be scarcely repeatible, and in practice the tracer gas technique cannot be applied. On the other hand, when a sufficient convective drag is present, experiments have shown that the type of gas does not influence the results of the tests.

Therefore, the experiments were always performed with non-zero values of the hot plate power. Experiments were performed varying the height of the hood above the hot plate and the exhaust air flow rate. The values assumed during the experiments by the variable parameters are shown in Table 2. The tracer gas volume flow rate and the hot plate power were kept constant respectively at 4 l/h and 1040 W.

Table 2 - Parameter values in the preliminary measurement campaign.

Case	Tracer gas	Height (a)	Air flow rate (b)
1	N_2O	0.50 m	500 l/min
2	SF_6	0.75 m	750 l/min
3	CH_4	1.00 m	

The results of the 18 tests performed combining the different parameters are summarized in Table 3. The influence of tracer gas is inversely proportional to the efficiency of the hood, as could be expected from the considerations above. In any case, the difference from gas to gas is always below $\pm$ 5.5 % of the average - that is, close to the measurement error itself - and does not show any systematic variation. It may then be concluded that in these conditions the influence of the tracer gas is negligible.

30 m³/h

Table 3 - Hood efficiency, in %, as a function of tracer gas type.

Test no.	Height (m)	Flow rate (l/min)	Tracer gas CH_4	N_2O	SF_6	Std dev (%)
a1b1	0.50	500	82.7	82.0	81.9	0.44 %
a1b2	0.50	750	90.4	91.1	92.5	0.96 %
a2b1	0.75	500	49.4	45.8	47.7	3.08 %
a2b2	0.75	750	60.3	60.7	60.4	0.28 %
a3b1	1.00	500	20.2	17.7	19.5	5.53 %
a3b2	1.00	750	53.5	54.2	55.6	1.60 %

Measurement campaign

Once that it was realized that the type of tracer gas had no apparent influence on the results of the tests, the actual measurement campaign was carried out using only SF_6 as a tracer gas at a flow rate of 4 l/h. SF_6 was chosen because its flow rate showed the greatest stability.

The following parameters have been varied (see Table 4):
(a) height of the hood above the table (H)
(b) exhaust air flow rate (Q)
(c) geometry of the hood (c1: isolated, c2: with vertical back panel)
(d) electric power released by the hot plate (W)

Table 4 - Parameter values in the measurement campaign.

Case	Height (a)	Flow rate (b)	Geometry (c)	Power (d)
1	0.50 m	500 l/min	c1	320 W
2	0.75 m	750 l/min	c2	730 W
3	1.00 m	1000 l/min		1040 W

In this measurement campaign every test was repeated 10 times for each combination of the parameters in order to evaluate the repeatibility of the tests and the reliability of the experimental apparatus. The results of the tests, expressed in terms of efficiency (average and standard deviation) are shown in Table 5 (see Table 4 to identify the test case).

As expected, the hood efficiency increases with:
- decreasing distance from the contaminant source (H),
- increasing exhaust flow rate (Q),
- decreasing "free" surface around the hood ($S_{l,f}$), and
- increasing power released by the hot plate (W)

These four quantities may be adimensionalized introducing the following parameters:

$$H^* = H/D_{hyd,h}$$

$$S^* = S_{l,f} / S_h$$

$$Re = Q\, D_{hyd,h} / (A_h \nu) \qquad \text{(Reynolds number of the hood)}$$

$$Gr = g\, \beta\, (T-T_a)\, (D_{hp})^3 / \nu^2 \qquad \text{(Grashof number of the hot plate)}$$

$D_{hyd,h}$
$S_{l,f}$
S_h

not proven that dimensionless approach has sense (niet alle parameters vermeld)

Table 5. Hood efficiency for different values of the parameters.

Case	c1d1	c1d2	c1d3	c2d1	c2d2	c2d3
a1b1	81.5±2.7	83.6±2.5	84.8±2.2	83.2±2.2	84.7±1.9	85.9±1.7
a1b2	89.2±3.8	91.8±3.4	92.5±2.8	91.6±3.4	92.9±2.8	93.0±2.5
a1b3	94.5±4.6	95.2±3.9	95.7±3.2	96.1±3.1	96.4±2.7	96.4±2.5
a2b1	47.5±4.2	52.8±4.0	55.1±3.7	52.5±3.0	56.3±2.3	57.6±1.9
a2b2	61.9±3.1	65.7±7.1	68.3±5.9	68.4±4.2	71.2±3.3	72.6±2.7
a2b3	75.8±6.3	78.5±5.7	80.1±5.4	80.7±4.9	82.2±3.6	82.9±2.8
a3b1	18.3±3.1	28.2±4.1	32.4±3.7	30.3±3.1	37.8±2.5	41.1±2.2
a3b2	41.3±8.8	47.8±6.6	50.6±5.3	48.0±6.4	53.3±4.6	56.5±4.0
a3b3	61.9±6.3	65.7±5.8	68.8±5.1	69.4±7.1	71.1±4.9	72.8±4.2

Due to the limited number of parameters actually varied in the tests performed, the last two parameters may be replaced, without any loss of accuracy, simply by the following:

$$Q^* = Q/Q_o \quad \text{with, for example, } Q_o = 100 \text{ m}^3/\text{h}$$

$$W^* = W/W_o \quad \text{with, for example, } W_o = 1000 \text{ W}$$

A regression analysis has shown that the hood efficiency may be expressed as a function of the nondimensional parameters above:

$$E = 1 - \exp(A\, H^{*a}\, S^{*b}\, Q^{*c}\, W^{*d}) \qquad (4)$$

with

$A = -2.527$
$a = -1.544$
$b = -0.346$
$c = 1.165$
$d = 0.160$

$$E = 1 - e^{\frac{1}{e^{2.53} \cdot e^{H}}}$$

A comparison between the values calculated by means of Eqn 4 and the experimental data is shown in Fig. 3. A correlation coefficient $r^2 = 0.96$ was found.

4. Theoretical approach and simulation results

The theoretical simulations were carried on at Turin Polytechnic using the Creare Inc. FLUENT package, chosen after several recently performed validations. A description of this model may be found in Cafaro et al. (1992), to which the interested reader is sent back.

For this specific problem the boundary conditions at the *inlet cells* were assumed at the tracer gas immission and air extraction duct in terms of distribution of air velocity and turbulent intensity value, and at the ideal lateral surface from the table to the hood base, in terms of fixed pressure conditions. At the *hood walls* the gradient of the main variable and the heat flux are assumed zero. At the hot plate surface a specific thermal flux was imposed.

To the domain of calculation a non-uniform three-dimensional cartesian grid made of 30x25x33 cells has been superposed. Power Law discretization scheme has been adopted. Simulations have been performed on a workstation HP 720 Apollo. A good convergence was achieved with about 3,000 iterations. The running time was about 24 hours.

At the moment only cases a2b1c1d2, a2b2c1d2, and a2b3c1d2 (see Table 5) were simulated. The results of the simulations are compared to the experimental results in Fig. 4. It can be observed that, although the quantitative agreement is rather poor, the qualitative trend is the same, and the relative error decreases with increasing efficiency (from about 32% to 18%).

Since this was only the first set of simulations no particular analyses were carried out in order to investigate configuration, boundary conditions and grid influences. Therefore it is not possible to state if the systematical under-estimation is due to the particular working conditions or to the approximations made in building the model.

Figures 5 and 6 respectively show the velocity and concentration fields in a vertical section of the hood. An accurate analysis of the simulation results points out some numerical features that can negatively influence the prediction accuracy: in particular, the flow field is perturbed by the "step-wise" surfaces that simulate the actually smooth hood walls. These fictitious steps create some local recirculations, and consequently some leakage out of the hood, that does not exist in the actual apparatus. Moreover the thermal field presents some local over-heating near the hot plate, due to a locally coarse grid.

5. Conclusions

The results illustrated in this paper represent a first approach in this field of study. In order to perform a more accurate analysis and achieve a more extensive data base, a further investigation both in the experimental tests and theoretical predictions is required. In any case, the experimental results obtained so long and the adopted efficiency parameter have shown good repeatibility and ability in focusing the behaviour of the hood.

From the analysis of the physical phenomena an empirical correlation between the influencing parameters has been obtained. This equation fits, with good agreement, the experimental data.

On the other hand, the numerical predictions have shown some limitations in the capability of providing satisfactory quantitative results. Some modifications in the model may surely improve its performance; however, also the present simulations can provide some interesting information about the behaviour of the apparatus, namely, a fine description of the three-dimensional flow fields, concentration fields and thermal fields can be obtained.

References

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$$E = 1 - \frac{1}{\alpha \cdot e^{\frac{Q^{1.17} \cdot W^{0.16}}{H^{1.5} \cdot S^{0,35}}}}$$

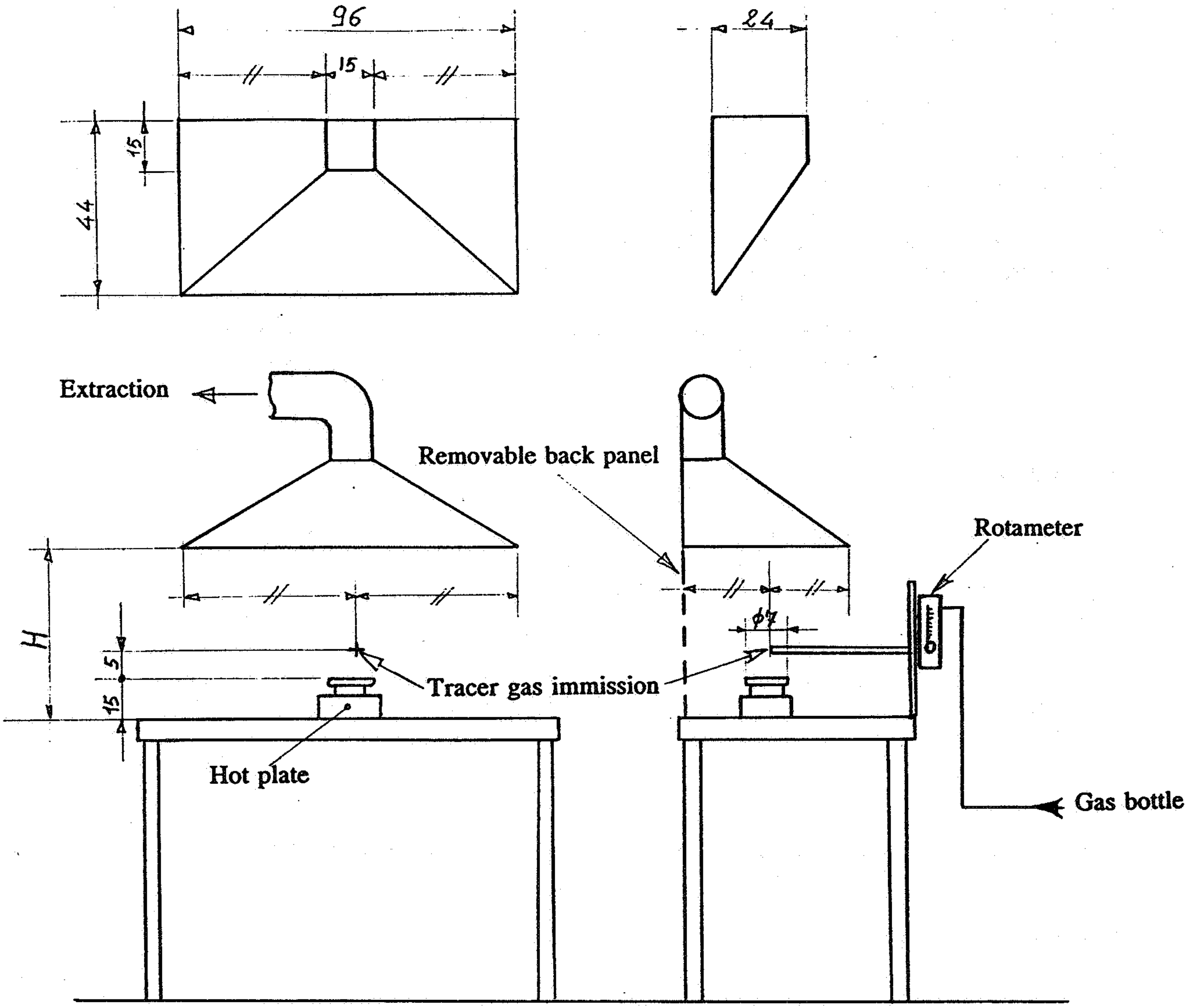


Fig. 1 - Layout of the experimental apparatus.

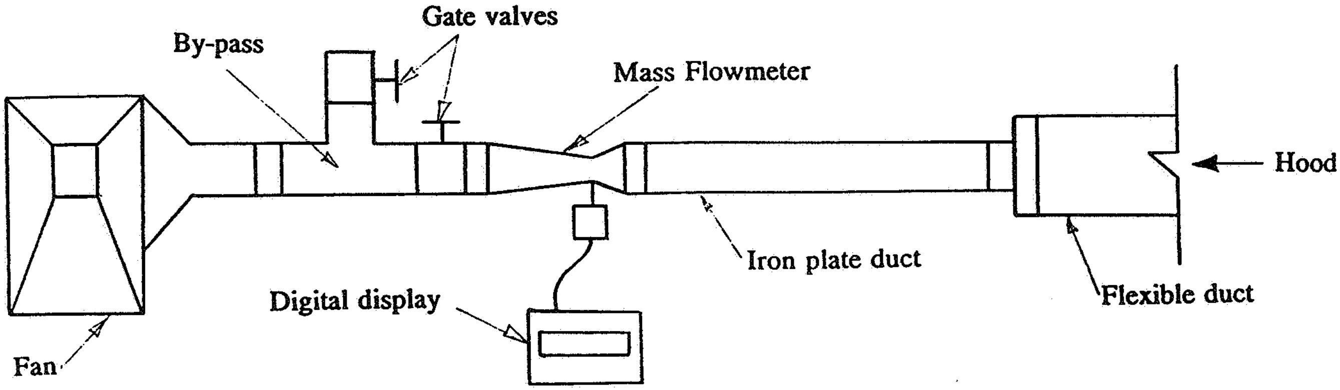


Fig. 2 - Layout of the air flow rate measurement and control.

Fig. 3 – Comparison of measured and calculated efficiencies

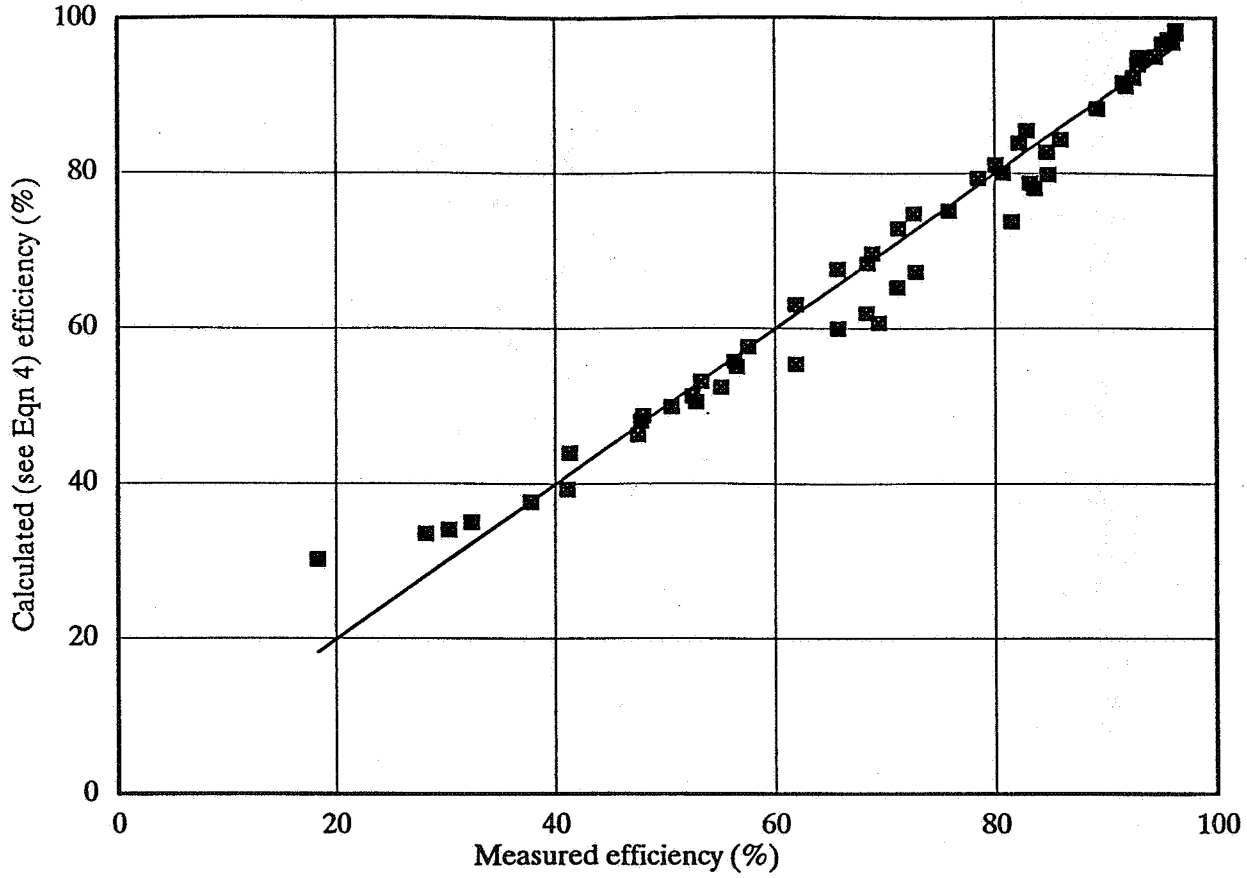
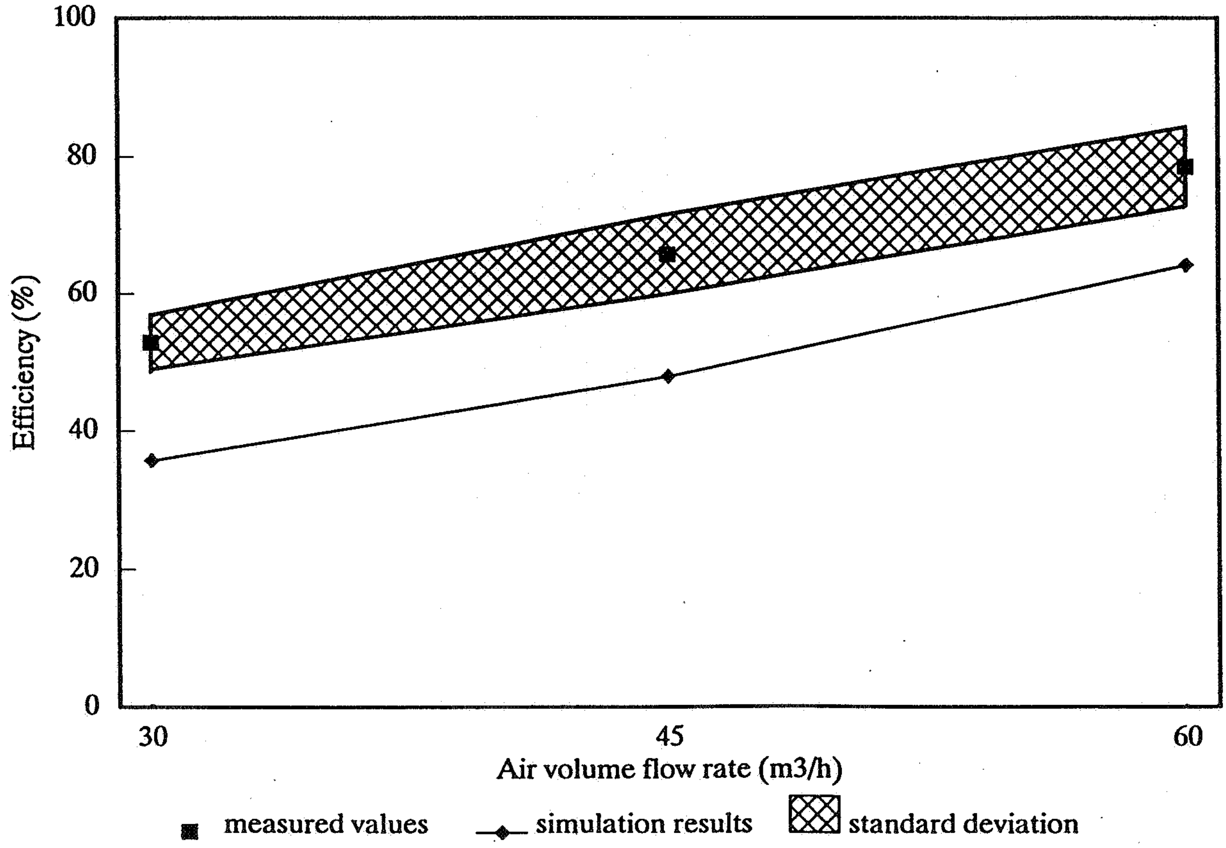
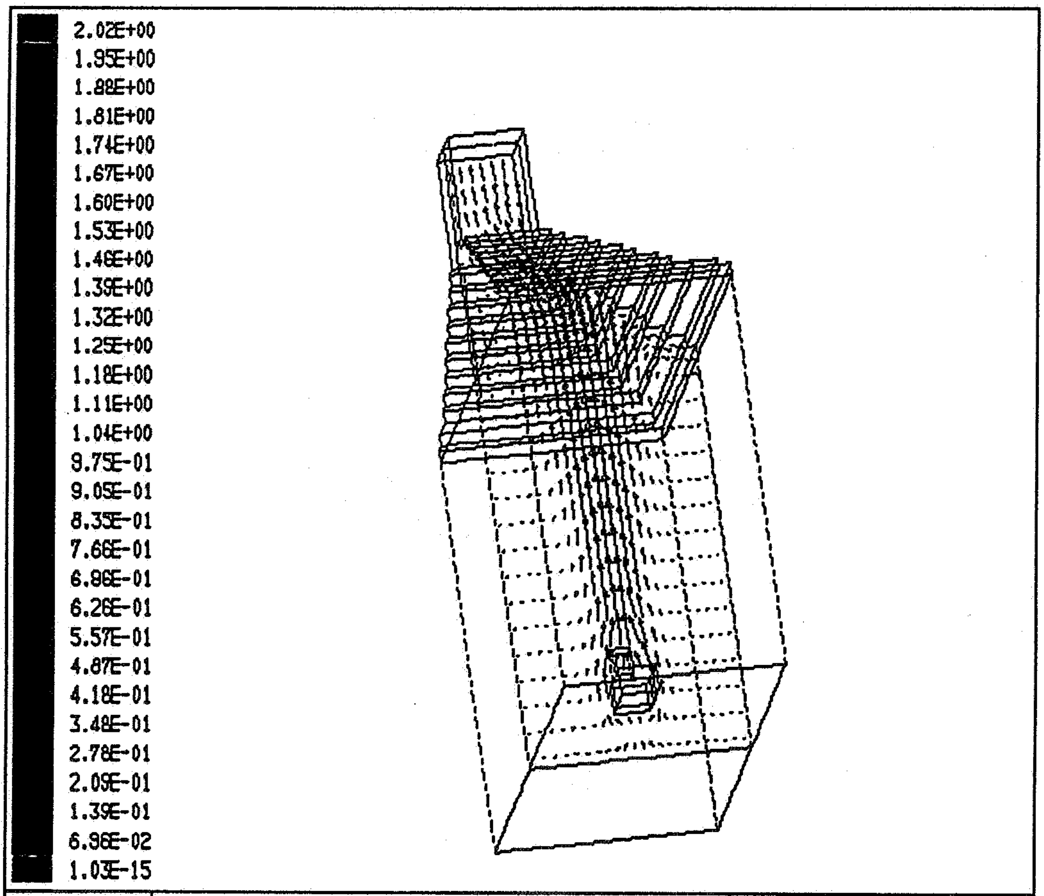


Fig. 4 – Comparison of measured and calculated efficiency as a function of volume flow rate

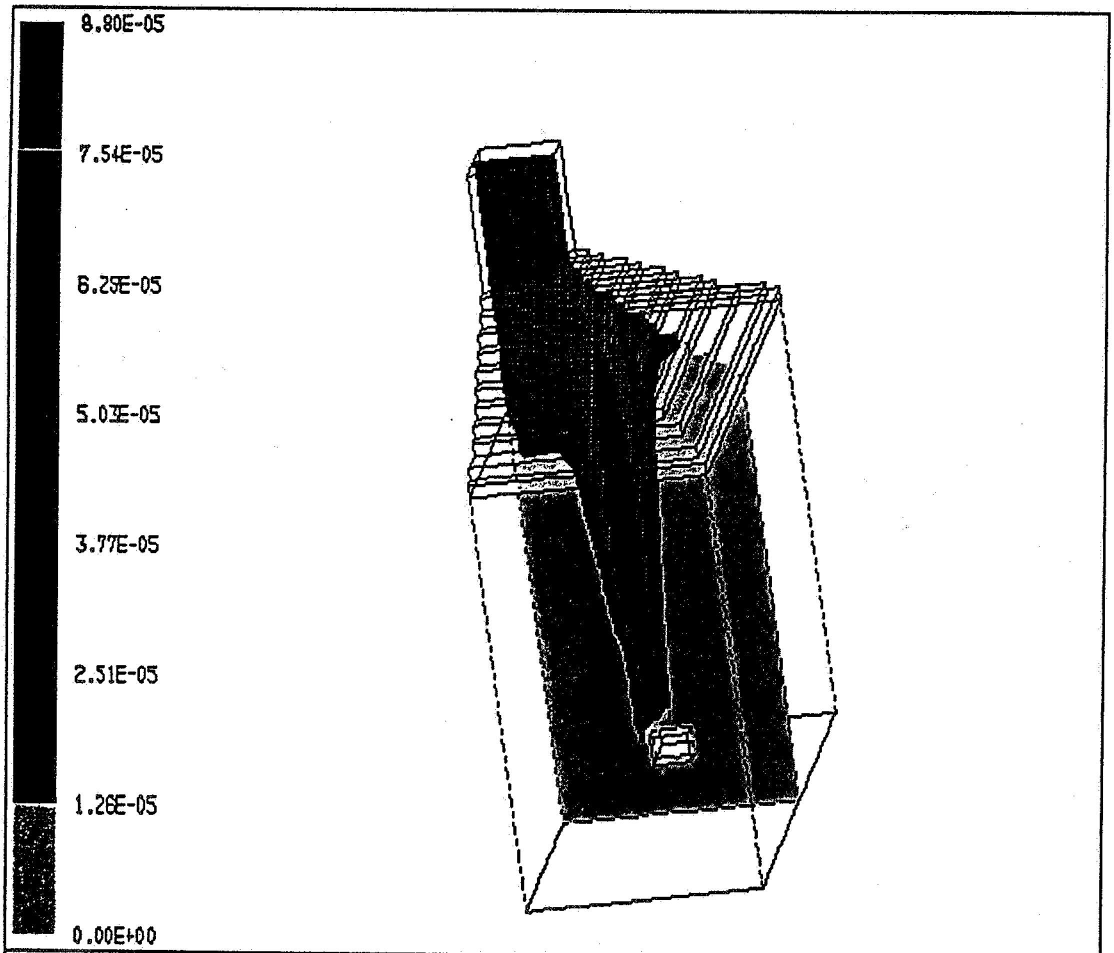




Z Y X

FIG. 5: FLOW RATE 750 L/MIN
Velocity Vectors (Meters/Sec)
Lmax = 1.694E+00 Lmin = 0.000E+00

Fluent 4.10
Fluent Inc.



Z Y X

FIG. 6: FLOW RATE 750 L/MIN
Trace Mole Fra (Dimensionless)
Lmax = 8.800E-05 Lmin = 0.000E+00

Fluent 4.10
Fluent Inc.