Influence of Mechanical Ventilation

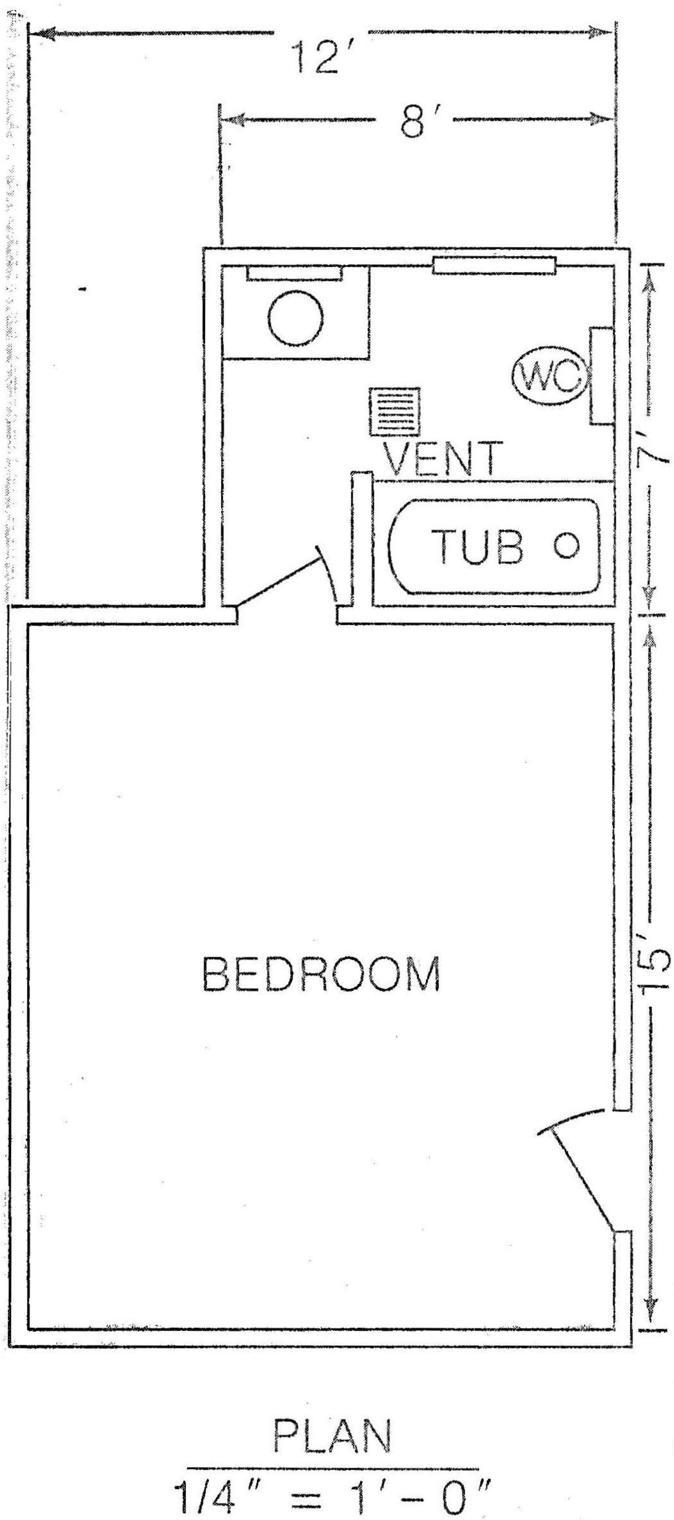

Fig. 1 Schematic drawing of test facility.

Due to the adverse effects of excess moisture within the indoor environment, experimental and theoretical studies were conducted to determine the build-up and removal rates of the moisture content of bathroom air during and after shower use. Data clearly show that without mechanical ventilation: (1) the minimum ventilation requirements for bathroom listed in ASHRAE Standard 62-73 cannot be met, and (2) excess moisture in the bathroom air cannot be controlled adequately.

F. H. SHAIR
Member ASHRAE
D.W. WOLBRINK
L.O. BOWEN
Associate Member ASHRAE
C.E. NEELLEY
K.E. SAMPSEL

BATHROOM ventilation helps remove moisture, excess heat, health-related gaseous and aerosol pollutants, and malodors.

Of particular interest is (1) the deleterious effects of condensation on materials and (2) on heat flow. With respect to the former, it should be noted that "*Water is either an essential or contributory factor in almost all cases of building material breakdown resulting from chemical changes such as the rusting of steel, physical changes such as the spalling of masonry by frost action, or biological processes such as the rotting of wood.*"[1] With respect to the effect of moisture on heat flow, "*moisture—when present in a material—is generally assumed to remain more or less stationary and to increase conductivity largely by adding to the path available for heat flow.*"[1] Sherwood and Peters[2] found that "even where there is no evidence of high moisture level within walls, moisture buildup in the wall and ceiling insulation can reduce its effectiveness."

Removing excess moisture and heat from the bathroom with an exhaust fan is also a way to save energy associated with air conditioning.[3] In most cases, exhaust fans require less energy than dehumidifiers.

Bridbord, *et al.*[4] reviewed those instances (including the use of sprays involving various propellants) where halogenated hydrocarbons in the indoor air environment may build up to concentrations of potential public health concern.

These concerns will grow in future years as increasing efforts are being made to insulate buildings more efficiently and to reduce infiltration and exfiltration as a means to conserve energy.

The need to remove malodors is of secondary concern since they are automatically dissipated by any practical ventilation process which adequately controls excess moisture and temperature. However, typical nonventilating processes which purportedly control malodors do not control temperature and moisture.

F. H. Shair is Professor of Chemical Engineering, California Institute of Technology at Pasadena; D.W. Wolbrink is with Broan Mfg. Co., Inc., Hartford, WI; L.O. Bowen is with Rangaire Corp., Cleburne, TX; C.E. Neelley is with Texas A & M University at College Station; and K.E. Sampsel is with the Home Metal Products Co., Plano, TX.

on Moisture Content of Bathroom Air

NOMENCLATURE

a $\equiv (q_0 + q_1 + q_2F + kA + S)/V$, reciprocal characteristic time, minutes^{-1}

A = total surface area within the bathroom upon which moisture may condense, ft^2 or m^2

C = concentration of moisture in bathroom air, $\frac{\text{grains of moisture}}{\text{pound of dry air}}$ or $\frac{\text{grams of moisture}}{\text{gram of dry air}}$

C_{ave} = average value of C defined by Eq. (5), $\frac{\text{grains of moisture}}{\text{pound of dry air}}$ or $\frac{\text{grams of moisture}}{\text{gram of dry air}}$

C_1 = concentration of moisture brought into the bathroom by way of mechanical ventilation, $\frac{\text{grains of moisture}}{\text{pound of dry air}}$ or $\frac{\text{grams of moisture}}{\text{gram of dry air}}$

C_0 = concentration of moisture brought into the bathroom by way of infiltration, $\frac{\text{grains of moisture}}{\text{pound of dry air}}$ or $\frac{\text{grams of moisture}}{\text{gram of dry air}}$

C_s = saturated concentration of air which is in equilibrium with the water droplets, $\frac{\text{pounds of moisture}}{\text{pound of dry air}}$ or $\frac{\text{grams of moisture}}{\text{gram of dry air}}$

C_w = effective concentration of moisture at the surfaces in contact with the bathroom air, $\frac{\text{pounds of moisture}}{\text{pound of dry air}}$ or $\frac{\text{grams of moisture}}{\text{gram of dry air}}$

F $\equiv (C_{inlet} - C_{outlet})/C_{inlet}$ is the factor which characterizes the filter associated with a recirculating system. C_{inlet} and C_{outlet} are the concentrations of moisture associated with the air entering and leaving the recirculating filter systems, respectively.

k = the average mass flux of moisture transported to the solid surfaces within the bathroom, $ft^3/(ft^2 - min)$ or $cm^3/(cm^2 - min)$

q_0 = the flow rate of air associated with infiltration, cfm or litres per minute

q_1 = the flow rate of makeup air associated with mechanical ventilation, cfm or litres per minute

q_2 = the flow rate of air associated with an internal recirculating system, cfm or litres per minute

S = the effective flow rate of air which is entrained within the shower droplets, cfm or litres per minute

$T_{wall\text{-}sat}$ = effective saturation temperature of the wall

t = time, minutes

Greek Symbols:

$\alpha \equiv q_0/(av)$, dimensionless parameter
$\beta \equiv q_1/(av)$, dimensionless parameter
$\delta \equiv kA/(av)$, dimensionless parameter
$\gamma \equiv S/(av)$, dimensionless parameter

$\rho \equiv$ density of air, $\frac{\text{pounds}}{ft^3}$ or $\frac{\text{grams}}{cm^3}$

Conversion Factors

1 in.	= 2.54 cm.
1 ft.	= 0.3048 m.
1 sq. ft.	= 0.0929 sq. m.
1 cubic foot	= 0.0283 cu. m. = 28.3 litres
1 cfm	= 28.3 litres per minute
1 gal. (U.S. liq.)	= 3.78 litres

$$1 \frac{\text{grain of moisture}}{\text{pound of dry air}} = \frac{1}{7000} \frac{\text{grams of moisture}}{\text{gram of dry air}}$$

$$°C = \frac{5}{9} (°F - 32)$$

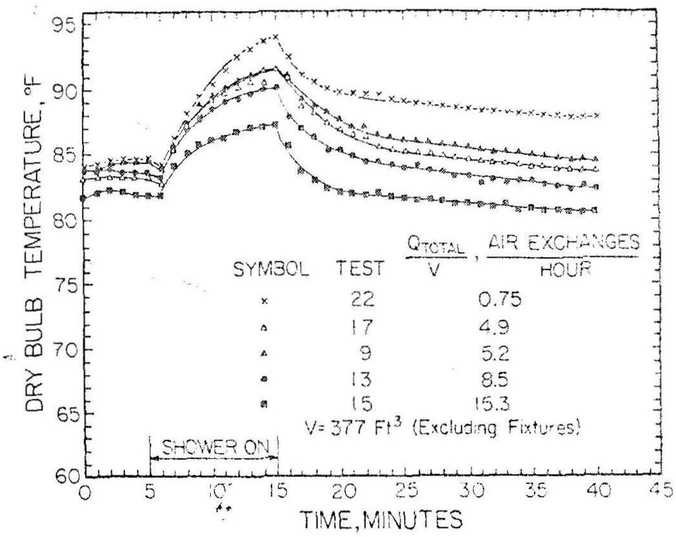

Fig. 2 Dry bulb temperature vs. time for various mechanical ventilation rates.

Chapter 20 of the 1977 ASHRAE HANDBOOK of FUNDAMENTALS presents the suggested method by which the flux of moisture transmission into and through a structural surface may be calculated; the driving force is the difference of humidity across the wall (as indicated by Fick's law of diffusion). One difficulty in using this general approach concerns the moisture build-up due to bathroom shower usage.

The purpose of this work was two-fold. First, the moisture vs. time relationships for bathroom air during and after conditions of shower were determined experimentally as a function of the rate of ventilation. Second, a theoretical model was developed to provide a perspective for interpretation of the experimental results and to permit the prediction of moisture content of bathroom air vs. time in general.

EXPERIMENTAL FACILITY AND PROCEDURE

The test facility, located at the Texas A & M University Engineering Experiment Station, is shown in Fig. 1. The Engineering Test Laboratory building consisted of a bedroom 12 ft. by 15 ft. (3.66 m by 4.57 m) and a bathroom 7 ft. by 8 ft. (2.13 m by 2.44 m.) connected by a door 30 in. × 80 in. (76 cm. × 203 cm.) with a ¾ in. (1.9 cm.) undercut [the area of the opening was 22.5 in.^{2}(145 cm.^{2})]. The ceiling height in both cases was 89 in. (226 cm.) above the floor. The empty bathroom had a volume of 415.5 cu. ft. (11.76 cu. m.); deducting the volume of the fixtures, the actual volume of air within the bathroom was estimated to be 376.6 cu. ft. (10.66 cu. m.).

The facility was constructed following standard wood frame methods. The exterior walls were sheathed with ½ in. (1.3 cm.) plywood and covered with ½ in. (1.3 cm.) siding. Neither the exterior floor nor the exterior roof was sheathed. The walls were insulated with fiberglass 3 in. (7.6 cm.) thick, the floor with fiberglass 5 ½ in. (14 cm.) thick, and the ceiling with fiberglass 6 in. (15.2 cm.) thick. The walls (a gypsum dry-wall system) were covered with one coat of an oil-based semi-gloss enamel.

The bathroom was equipped with a bathtub containing a shower, a lavatory with an overhead mirror, and a toilet. A shower curtain was installed across the bathtub. Standard construction practices were followed during installation of the fixtures.

The exhaust fan outlet was located in the ceiling and equipped with a grille. The opening was ducted directly to a standard air measuring chamber (tunnel) rather than to an exhaust fan; this arrangement enabled accurate control and monitoring of the mechanical ventilation rate.

Wet and dry bulb measurements were obtained using standard equipment. Tracer gas experiments were conducted (in a manner similar to those reported by Drivas, *et al.*[5]) in order to determine accurately the sum of the rates of infiltration plus mechanical ventilation. In these experiments, carbon monoxide in concentrations under 100 ppm was used as the tracer gas. The tracer gas concentration was continuously measured and recorded by means of an Ecolyzer system and a strip-chart recorder. The tracer technique was particularly useful in determining the overall exchange rate of air (*i.e.*, $q_0 + q_1$) when the rate of mechanical ventilation was not significantly greater than the rate of infiltration. The tracer technique also served as a useful check on the operation of the standard air measuring chamber.

THE THEORETICAL MODEL

A generalized theoretical model is available which permits accurate predictions of indoor gaseous pollutants.[6] However, for this study sink and source terms had to be specified in order to treat water vapor, taking into account condensation and re-evaporation from surfaces.

Consider a well-mixed bathroom air having a volume V, an infiltration rate q_0, and a make-up air flow rate q_1 (due to mechanical ventilation). Within the bathroom, there may exist an internal recirculating system, such as a ductless fan, involving a filter characterized by a factor $F = (C_{inlet} - C_{outlet})/C_{inlet}$. F is a function of the component to be filtered, the flow rate q_2, and of previous type and extent of exposure. When the shower is on, it represents the major source of moisture released into the air. The surfaces within the bathroom will react as either sources or sinks, depending upon the exact moisture concentration of the air and upon the moisture content associated with the surfaces.

The general transient expression of the mass balance of moisture associated with the bathroom air is given by

$$V\frac{dC}{dt} = (q_0C_0 + q_1C_1) - (q_0 + q_1)C - q_2FC + \text{Sources} - \text{Sinks} \qquad (1)$$

where C_0 represents the concentration of moisture brought

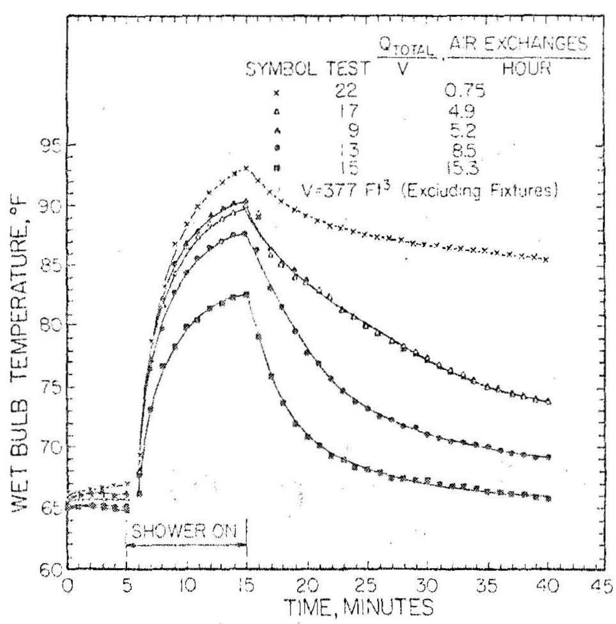

Fig. 3 Wet bulb temperature vs. time for various mechanical ventilation rates.

Table 1
Values of the Parameters Used in the Theoretical Model

Parameter	Test 22	Test 17	Test 9	Test 13	Test 15	Ave.
$(q_0 + q_1)/V$, $\frac{\text{Air Changes}}{\text{Hour}}$	0.75	4.9	5.2	8.5	15.3	—
$q_0 + q_1$, cfm	4.7	30.8	32.6	53.4	96.0	—
$C_0 = C_1$, $\frac{\text{Grains of Moisture}}{\text{Pound of Dry Air}}$	59.5	60.2	58.1	56.0	57.4	58.2
C_s, $\frac{\text{Grains of Moisture}}{\text{Pound of Dry Air}}$	390	393	379	381	375	384
C_w (Shower on) $\frac{\text{Grains of Moisture}}{\text{Pound of Dry Air}}$	140	160	180	160	140	156
C_w (Shower off) $\frac{\text{Grains of Moisture}}{\text{Pound of Dry Air}}$	190	180	180	140	140	166
$T_{wall\ sat.}$ (Shower on) °F	76.8	81.7	84.2	81.7	76.8	80.2
$T_{wall\ sat.}$ (Shower off) °F	85.8	84.2	84.2	76.8	76.8	81.6
kA cfm	30	30	30	30	30	30
S (Shower on) cfm	45	45	45	45	45	45
S (Shower off) cfm	0	0	0	0	0	0
F	0	0	0	0	0	0

into the room by way of infiltration, C_1 represents the concentration of moisture brought into the room by way of mechanical ventilation, and t represents time. The left-hand side of Eq. 1 represents the rate of accumulation of moisture within the bathroom air; the terms on the right represent either inputs (due to air-flows and internal sources) or outputs (due to air-flows and internal sinks).

$$C = C_1 \text{ at } t = 0 \tag{2}$$

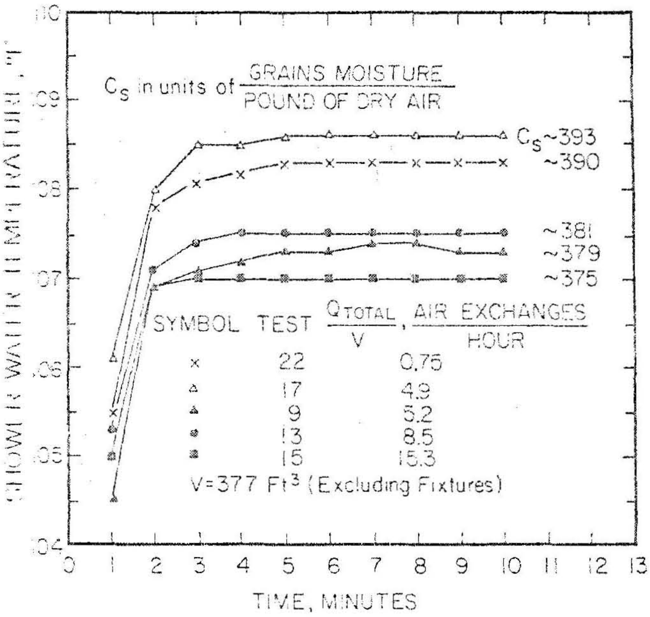

Fig. 4 Shower water temperature vs time for various mechanical ventilation rates.

The main thrust of the model development reduces to finding reasonably accurate representations of the source and sink terms in Eq. (1). The simplest reasonable representations of the source and sink terms lead to the expression:

$$V\frac{dc}{dt} = q_0C_0 + q_1C_1 - (q_0 + q_1)C - q_2FC + kA(C_w - C) + S(C_s - C) \tag{3}$$

where kA represents the average mass flux of moisture transport associated with the solid surfaces within the bathroom, A represents the surface area of the solid surfaces, C_w represents the effective concentration of moisture at the surface of the wall in contact with the bathroom air, S represents the effective flow rate of air which is entrained within the shower droplets, and C_s represents the saturated concentration of air which is in equilibrium with the water droplets.

Since most of the surface area is relatively porous (*i.e.*, enamel-covered gypsum dry-wall system), it is reasonable to expect that k (and thus the product of kA) will be relatively independent of air movement within the room; in other words, it is reasonable to expect that diffusion of moisture within the wall will be the rate-determining (slow) step involved with moisture transport to and from the walls. Based upon the work of Hales, *et al.*[7], it is reasonable to anticipate that k might have a value between 0.1 and 1.0 ft/min (3.05 and 30.5 cm/min). Since the wall temperature varied from one test to the next, it is reasonable to expect that C_w might also vary somewhat. Since the driving potential $(C_w - C)$ may be both positive and negative, the walls may act as a net source of moisture at some times and as a sink of moisture at other times.

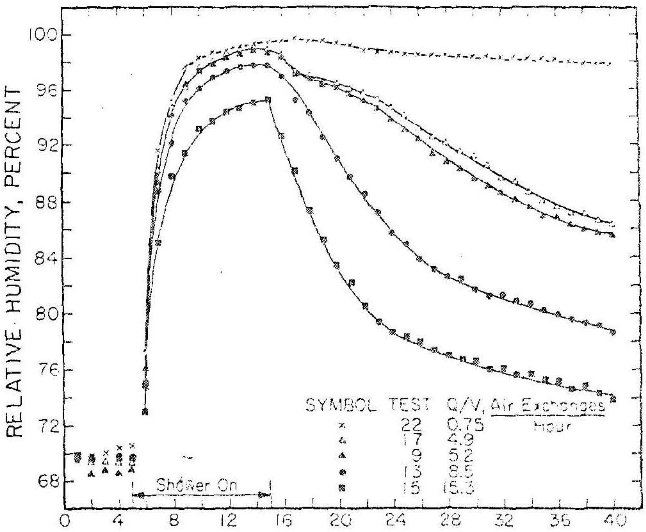

Fig. 5 Relative humidity vs. time for various mechanical ventilation rates.

The quantity S represents the flow rate of air which is intimately mixed with the vast number of shower droplets (*i.e.*, the flow rate of air which comes into intimate contact with the air-water interface). It is reasonable to expect that S will be a strong function (possibly directly proportional) to the water flow rate issuing from the shower and exit water temperature. It is also expected that S will be a function of shower-head design (*i.e.*, more or less spray). In the test results reported herein, the shower flow rate and shower-head design were dept constant. The water from the shower had a flow rate of about 3 gals. (11.3 litres) per minute and an initial inlet temperature of 107.8°C ± 0.8F° (42.1°C ± 0.44°C). Since the flow rate of air entrained within the shower is probably driven primarily by the flow of water issuing from the shower, it is reasonable to expect that S will be relatively independent of the ventilation rates. Since the shower conditions were more carefully controlled than were the exact conditions of the walls, it is expected that C_s will vary less between tests than C_w.

The solution to Eq. (3) is

$$C = \alpha C_0 + \beta C_1 + \delta C_w + \gamma C_S + (C_i - \alpha C_0 - \beta C_1 - \delta C_w - \gamma C_S)e^{-at} \qquad (4)$$

where

$a \equiv (q_0 + q_1 + q_2F + kA + S)/V$
$\alpha \equiv q_0/(aV)$
$\beta \equiv q_1/(aV)$
$\delta \equiv kA/(aV)$
$\gamma \equiv S/(aV)$

RESULTS

The value and purpose of the theoretical model is twofold. First, it provides a perspective from which experimental results can be interpreted. Second, it provides a means of predicting the moisture content of the air for bathrooms and conditions other than those experimentally determined within this study.

With respect to the first point, the theoretical model provides a means of determining how sensitive the humidity is to each of the influencing parameters. It should be noted that the mechanical ventilation rate q_0 is the parameter which most easily can be used to control the moisture level and temperature and, in so doing, also removes pollutants and malodors. Other parameters such as S (the value of which is reflected by the design of the shower stall and the shower nozzle) and C_w (the value of which is determined by the shower water temperature) can also be used to some extent to control the moisture content and temperature of the bathroom air. Other influencing parameters such as the infiltration rate, q_1, and the moisture content with which the walls are in equilibrium, C_w, are difficult to control.

The dbt and the wbt as functions of time are shown in Fig. 2 and 3, respectively. As indicated, the dbt and wbt rise sharply after about one minute following the start of the shower. The one-minute delay is due to the time required for the shower water to reach the desired temperature. Initially, the water which reaches the showerhead is cooled in transit by the relatively cool pipe; however, after the pipe becomes heated by hot water, the exit water temperature at the shower remains relatively constant (Fig. 4). Due to the strong dependence of the humidity ratio at saturation upon temperature, much effort was aimed at maintaining a constant shower water temperature throughout the series of experiments.

Fig. 5 shows the relative humidity as a function of time. It should be noted that in the case of no mechanical ventilation, the relative humidity remains at or about 98% for 20 minutes after the shower is turned off, which is in sharp contrast to the lowest mechanical ventilation rate (*i.e.*, 4.9 air exchanges per hour). The infiltration rate without mechanical ventilation was found to be 0.75 air changes per hour. (Infiltration rates for homes in general range between about 0.5 to about 1 air change per hour, according to current building practices.) Without mechanical ventilation so little moisture is removed through infiltration that, following shower, the moisture in the air remains in equilibrium with the moisture which has condensed on the surfaces of the building. Cooling of the warm, moist air is done primarily through heat transfer from the air to the bathroom surfaces; this cooling forces the air to release moisture (as indicated by the absolute humidity ratio shown in Fig. 7), primarily through condensation on surfaces throughout the

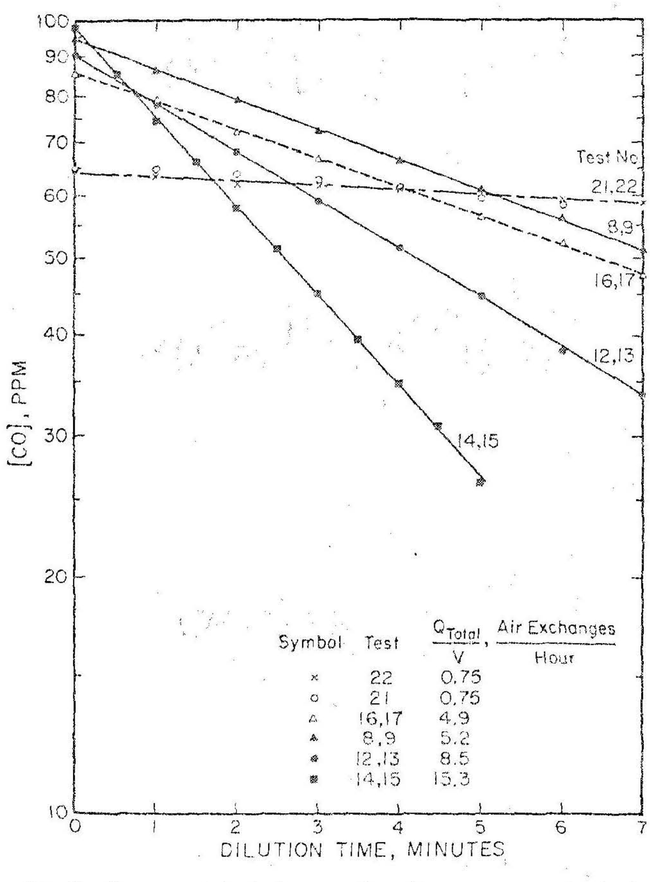

Fig. 6 Tracer concentration vs. time for various mechanical ventilation rates.

bathroom. At five air changes per hour, the primary mechanism by which the bathroom is cooled and by which excess moisture is removed is through mechanical ventilation; and the rate of moisture transfer to surfaces throughout the bathroom is *greatly* reduced. (At a rate of five air changes per hour, following the shower the relative humidity drops quickly enough to actually aid substantially in drying the surfaces throughout the bathroom.)

Fig. 6 shows the trace data from which the total exchange rate of air was determined accurately for each test; the total exchange rate was determined from the slope of the straight line on the semi-log plot.

As indicated in Fig. 7, it was possible to find reasonable values of each of the parameters which permitted excellent agreement between theory and experiment. The values of the parameters are listed in Table 1.

It should be noted that the values of kA, C_s and S are quite constant from one experiment to the next. Estimating the value of A to be about 43.4 ft.^{2} (4.03 m.^{2}) leads to a value of k = 0.7 ft/min (0.36 cm/second), which is in the range of values expected from previous work.[6] It is reasonable to use this value of k for any surface which has physical properties similar to those used in the test facility used in this study.

The values of C_w ranged from 140 to 180 and correspond to changes in the wall temperature from 76.8°F to 82.4°F (24.9°C to 28.0°C). The value of S was found to be quite constant at about 45 cfm or about ¾ ft^3/sec (21 litres/sec), which also seems quite reasonable.

For convenience, the average values of the parameters are also listed in Table 1. When no other data are available, it is recommended that the average values of the parameters (listed in Table 1) be used in the model (Eq. 4) to

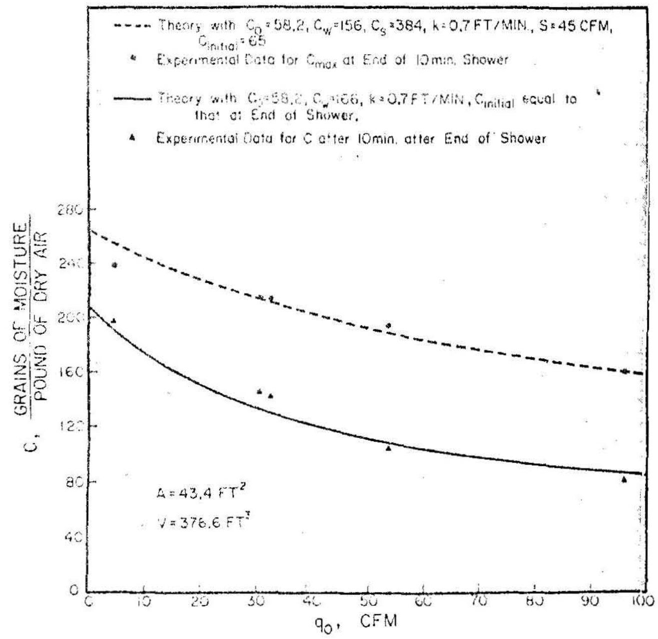

Fig. 8 Absolute humidity at the end of a ten-minute shower and at ten minutes after the shower vs. mechanical ventilation rate.

predict the absolute humidity ratio. An example of the use of the average values in Eq. 4 is shown in Fig. 8. The upper curve represents the absolute *humidity ratio predicted to* occur at the end of a 10-minute shower, as a function of the rate of mechanical ventilation. The lower curve represents the absolute humidity ratio predicted to occur at the end of 10 minutes *after* the end of a 10-minute shower, as a function of the rate of mechanical ventilation. As indicated, the agreement between theory and experiment is excellent.

Test 22 was conducted with no mechanical ventilation; the only exchange of air with the outside was due to infiltration. The infiltration rate, measured by means of the tracer technique, was found to be 0.75 air changes per hour. This case also corresponds to that for an internal recirculating ductless fan which does not remove moisture from the air.

Tests 9,13, and 15 were conducted with mechanical ventilation rates 5.2, 8.5, and 15.3 air changes per hour, respectively. Test 17 was conducted to determine the reproducibility of the experimental data; the mechanical ventilation rate for Test 17 was 4.9 air changes per hour. As shown in Fig. 2,3,5 and 7, the agreement between Test 9 and Test 17 is excellent.

It should be noted that these tests clearly show that without mechanical ventilation, the minimum ventilation requirements for bathrooms listed in ASHRAE Standard 62-73[7] cannot be met. At the 0.75 air change rate of the tests, air supply without exhaust fan was 4.7 cfm, far below the 20 cfm (per person) minimum required and 30-50 cfm recommended by the ASHRAE Standard. Even at a "loose" construction rate of 1.5 air changes per hour, the air supply without exhaust fan still would amount to less than half the standard's minimum. It is worthy of note that the ASHRAE Standard does not cover ventilation by openable window, but specifies *"installed capacity for intermittent use,"* which means the controlled air flow provided by an exhaust fan. It should also be noted that once the mechanical ventilating fan is installed, the cost of operation is negligible. For example, the power required to operate a 100 cfm fan (corresponding to 15.9 air changes per hour for the test facility bathroom) is less than 100 watts. Even if the fan required 100 watts, the cost of operating the fan for 40 minutes each day for one year with electrical power costing 4¢ per kilowatt hour would still be less than one dollar.

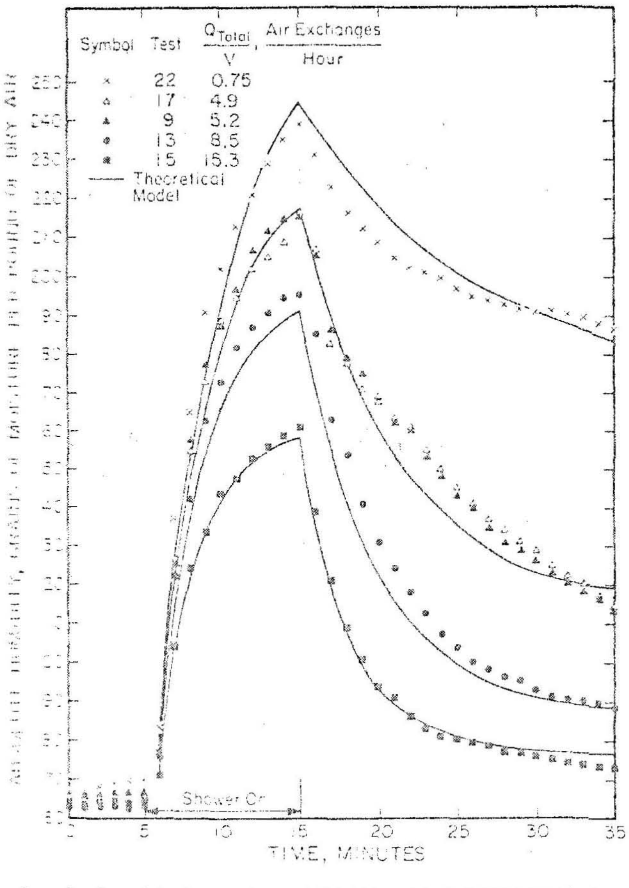

Fig. 7 Absolute humidity vs. time for various mechanical ventilation rates

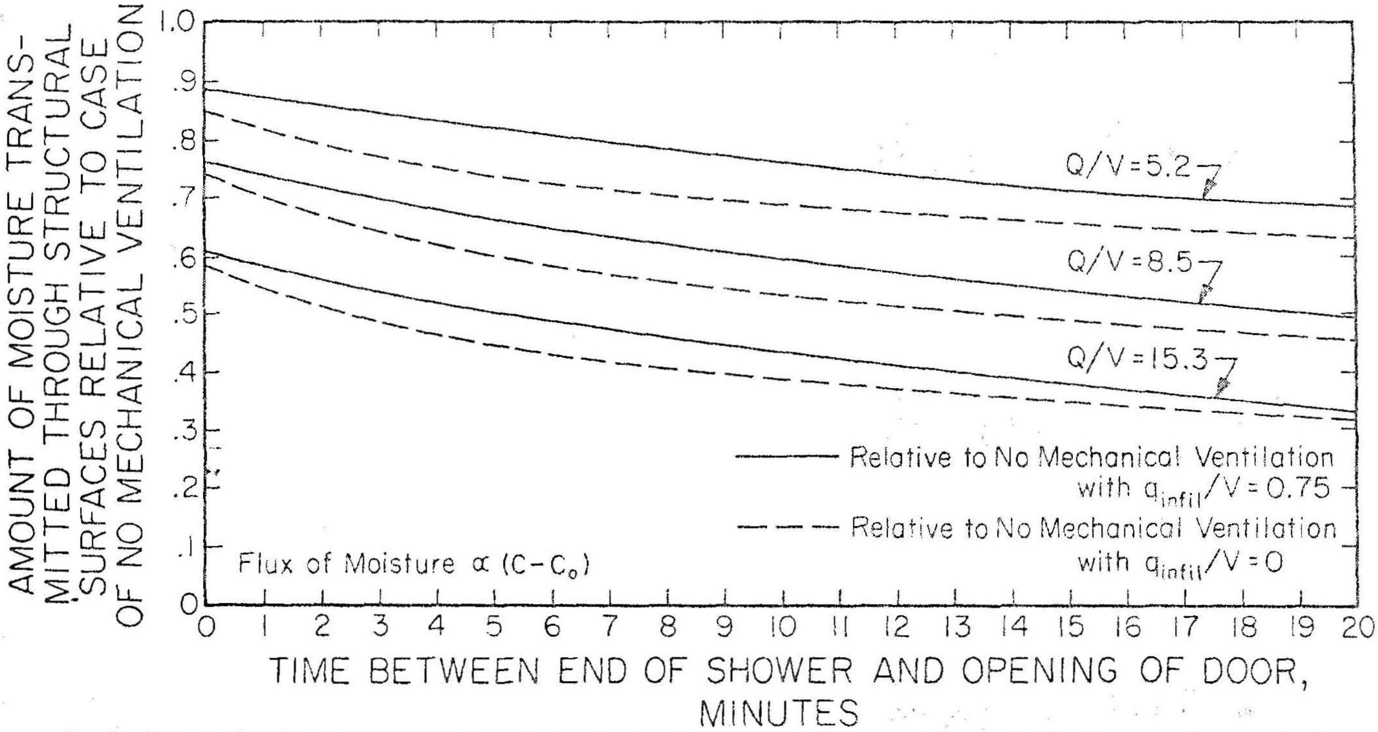

Fig. 9 Amount of moisture transmitted through structural surfaces in the bathroom relative to that for the case of no mechanical ventilation vs. time between end of shower and opening of door.

It is of interest to use the theoretical model to calculate the amount of moisture removed by a given airstream such as that associated with the mechanical ventilation system. In order to do this we must first determine the average concentration of the moisture during the time interval of interest. The average concentration of moisture between t_1 and t_2 is simply:

$$C_{ave} = \left(\frac{1}{t_2-t_1}\right) \int_{t_1}^{t_2} C dt = \alpha C_0 + \beta C_1 + \delta C_w + \gamma C_s + \frac{(C_i - \alpha C_0 - \beta C_1 - \delta C_w - \gamma C_s)}{a(t_2 - t_1)} \left(e^{-q_1 t_1} - e^{-q_2 t_2}\right) \ldots \quad (5)$$

The average rate of moisture removed by the airstream with flow rate q_1 and density p during time $t_2 - t_1$ is equal to pq_1C_{ave}, where C_{ave} is the average value of C during time $t_2 - t_1$.

The value of $(C_{ave} - C_0)$ may be used to calculate the rate of transmission of moisture through a structure when the moisture content of the air in contact with the outside surface of the wall is C_0. Fig. 9 indicates the amount of water transmitted through the wall for various ventilation rates and various times between the end of a ten-minute shower and the opening of the bathroom door relative to that for the case of no mechanical ventilation; the dotted line indicates what the ratio would be if infiltration were eliminated in Test 1 which served as the reference.

The data displayed in Fig. 2,3,5,7,8, and 9 indicate that a mechanical ventilation rate of eight air changes per hour provides reasonable control of the moisture content of bathroom air during and after a shower.

A more detailed discussion of the experimental program along with the tabulated data may be obtained from the Home Ventilating Institute.[9]

CONCLUSIONS

Experimental data show that without mechanical ventilation, the minimum ventilation requirements for bathrooms listed in ASHRAE Standard 62-73 cannot be met. In order to control excess moisture build-up in the air, even a mechanical ventilation rate of five air changes per hour is far superior to the case of no mechanical ventilation. Data suggest that a mechanical ventilation rate of eight air changes per hour provides reasonable control of the moisture content of bathroom air during and after a shower. A theoretical model, based upon a mass balance of the moisture associated with the bathroom air, was found to yield results in excellent agreement with the experimental data.

ACKNOWLEDGMENTS

The authors wish to thank Donald McClurg and Arnold Rodin for their interest and support. We also wish to thank Richard Kuehner for his help and encouragement. This work was supported by the Home Ventilating Institute.

REFERENCES

1. ASHRAE Handbook of Fundamentals, ASHRAE, 345 East 47th Street, New York, NY 10017, Chapter 20 (1977).
2. Sherwood, G.E. and C.C. Peters, "Moisture Conditions on Walls and Ceilings of a Simulated Older Home during Winter," USDA For. Serv. Res. Paper FPL 290 (1977).
3. "Home Energy Saving Tips," National Bureau of Standards GPO 871-542, U.S. Department of Commerce, NBS, Washington, D.C. 20234 (1976).
4. Bridbord, K., Brubaker, P.E., Gay, B., and J.C. French, "Exposure to Halogenated Hydrocarbons in the Indoor Environment," Env. Health Perspectives 11, 215 (1975).
5. Drivas, P.J., Simmonds, P.C., and F.H. Shair, "Experimental Characterization of Ventilation Systems in Buildings," Environ. Sci. and Tech. 6, 609 (1972).
6. Shair, F.H. and K.L. Heitner, "A Theoretical Model for Relating Indoor Pollutant Concentrations to Those Outside," Environ. Sci. and Tech. 8, 444 (1974).
7. Hales, C.H., Rollinson, A.M., and F.H. Shair, "Experimental Verification of the Linear Combination Model for Relating Indoor-Outdoor Pollutant Concentrations," Environ. Sci. and Tech. 8, 452 (1974).
8. ASHRAE Standard 62-73, ASHRAE, 345 East 47th Street, New York, NY 10017 (1973).
9. Home Ventilating Institute, 230 North Michigan Avenue, Chicago, Illinois 60601 (1978).